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Impact of Surface Characteristics on Radiant Panel Output

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ABSTRACT

This paper presents the experimental results of a research project conducted to determine the impact of surface characteristics on radiant panel output. The hemispherical and angular emittance and radiant panel output were measured for various modern building materials. The surfaces were found to have uniformly high hemispherical emittances and could be considered diffuse emitters. An office-sized environmental chamber was used to test radiant ceiling, floor, and wall panels under both natural and forced convection conditions. The results of these experiments showed that for the surfaces and conditions tested, surface texture did not have a significant impact on the rate of heat transfer from the radiant panels.

INTRODUCTION

Various materials are used as surfaces for radiant systems. Smaller, fast-acting systems usually have a thin painted layer over the heating element, which is often textured to improve appearance. Carpet, wood, and vinyl are commonly used as surfaces for low-temperature floor systems. Because of the diverse nature of these surfaces, it is important to know how surface characteristics will affect panel output.

This paper reports the results of experiments designed to measure the effects of surface characteristics on heat transfer from radiant panels. This information will assist system designers in predicting the actual panel output required for thermal comfort as well as energy performance.

The effort to measure radiative properties was primarily motivated by the fact that the emissivity of some modern building materials, such as carpet and vinyl, cannot be found in the literature. Since all previously measured nonmetal building materials have emissivities approximately equal to 0.9, it is commonly assumed that modern materials have similar properties. The research checked this assumption.

The operation of a radiant panel depends on a number of factors including the radiative surface properties of the panel and natural convection from the panel. The issue of radiative properties is simplified since most surfaces used as radiant panel surfaces are either nonmetal or painted metal. V. C. and A. Sharma (1989) noted that the hemispherical emittance of the building materials they measured (clay, concrete, cement, paper, wood, and paint) were high and varied from 0.8 to 0.92 ±0.02. Brewster (1992) made the same observation. Gubareff et al. (1960) compiled an extensive review of radiative properties that also show the same result for nonmetals. What this means is that surfaces that have the same emittance will transfer radiant heat equally as well, even though the surfaces may have significantly different appearance, feel, or texture.

A number of experiments dealing with various factors related to radiant heating and cooling were performed in the 1950s in the ASHVE Environmental chamber. Min et al. (1956) performed experiments to quantify the effects of natural convection and radiation in a panel-heated room. The natural convection correlations they developed are still used in the ASHRAE Handbook as an integral part of the currently recommended procedure for radiant panel design (ASHRAE 1996). Research by Schutrum and Humphreys (1954) reported additional panel performance data for rooms with nonuniform surface temperature environments. The results confirmed the previous conclusion that nonuniform surface temperatures had a small effect on both ceiling and floor systems. Panel performance was adequately predicted using the area weighted average, unheated surface temperature (AUST) of the room. Additional carpeted room tests were also performed. The results of these tests demonstrated that the heat transfer from a carpet surface is essentially the same as the heat transfer

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from an ordinary bare panel surface at the same surface temperature. They, of course, also noted the addition of carpet greatly increases the thermal resistance of the panel system.

BACKGROUND AND THEORY

Low-temperature radiant panels in buildings emit almost all of their energy in the infrared region. Thus, for purposes of calculating radiative heat transfer between building elements, the UV and the visible portions of the spectrum may be ignored. The radiative properties of building elements in the infrared region determine their performance as radiant heaters. Figure 1 shows the spectral emissive power for a blackbody. It is seen from this figure that most of the energy emitted by a blackbody that is at a temperature of 300°F is located in the infrared region.

Wien's displacement law comes from differentiating the blackbody intensity function and setting the result to zero (Incropera and DeWitt 1990).

$$\lambda_{max}T = 2898\ \mu m \cdot K \qquad (1)$$

This relationship gives the wavelength at which the emitted intensity is maximized for a given temperature and provides a quick estimate of the spectral energy distribution from a black surface. Approximately 25% of the energy is radiated at wavelengths shorter than the maximum and 75% at wavelengths longer than the maximum. For a surface at 300 K (80°F), more than 75% of the energy will be in the infrared region at wavelengths greater than 9.7 μ.

The emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a black surface at the same temperature. The *total, directional* emissivity, ε', is defined as the ratio of total directional emitted flux to total directional blackbody flux.

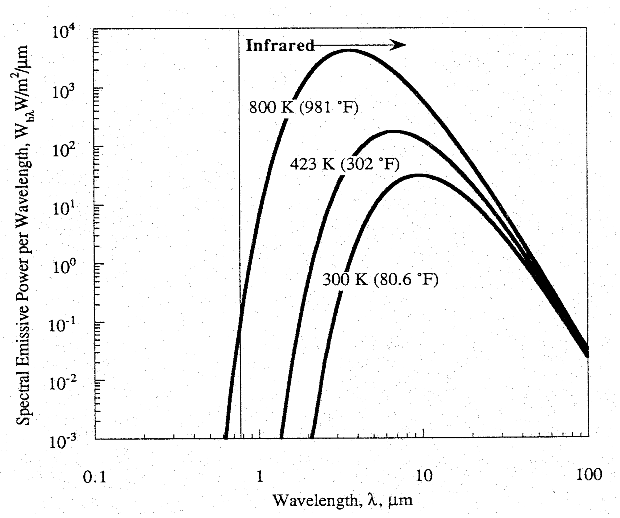

Figure 1 *Blackbody spectral emissive power.*

$$\varepsilon' = \frac{I_e(\theta, \phi, T)}{I_b(T)} \qquad (2)$$

The normal emissivity, which is the emissivity in the direction perpendicular to the surface, is the most commonly referred to directional emissivity. The *total hemispherical* emissivity, ε, is defined as the ratio of the total hemispherical flux to the total hemispherical blackbody flux (Incropera and DeWitt 1990).

$$\varepsilon = \frac{\int_0^{\infty}\int_{2\pi} I_{\lambda,e} \cos\theta \, d\varpi \, d\lambda}{\pi I_b} = \frac{W(T)}{W_b(T)} \qquad (3)$$

The literature usually gives values for either the hemispherical emissivity or normal emissivity of a surface. For nonmetals, the hemispherical emissivity is usually 95% of the normal emissivity. The experimental procedure used in this investigation measured the hemispherical emittance of various surfaces.

The *impact of surface characteristics* includes the effect of both micro- and macro-surface characteristics on radiative properties. This report will seek to distinguish between the length scales associated with surface characteristics by referring to micro-surface and macro-surface characteristics. *Micro-surface* characteristics refer to the surface characteristics that are on the order of the wavelength of thermal radiation. *Macro-surface* characteristics refer to the surface characteristics that are much greater in scale than the relevant wavelength of thermal radiation. Micro-surface characteristics include, but are not limited to, material properties. For example, the micro-surface characteristics of aluminum include both the chemical composition and crystalline structure of the metal as well as the polish of the surface. Macro-surface characteristics include the texture and roughness of the surface. The roughness of a surface for a metal can be a micro characteristic in terms of the above definitions. The *radiative properties* of a surface include its emissivity, absorptivity, reflectivity, and transmissivity. It should be noted that the radiative properties of metals and nonmetals (conductors and nonconductors) are often significantly different, and the impact of both micro- and macro-scale surface characteristics can also be different for a metal and nonmetal.

Figure 2 shows the parameters that affect radiative properties. Both micro- and macro-surface characteristics can be important. In addition, temperature can also have a strong influence on the radiative properties of a surface.

EXPERIMENTAL FACILITY

The experiments were performed in an office-sized environmental chamber where all the significant environmental parameters could be controlled. This controlled environment offered a number of substantial benefits in terms of experimental design. First, precise knowledge of the room surface

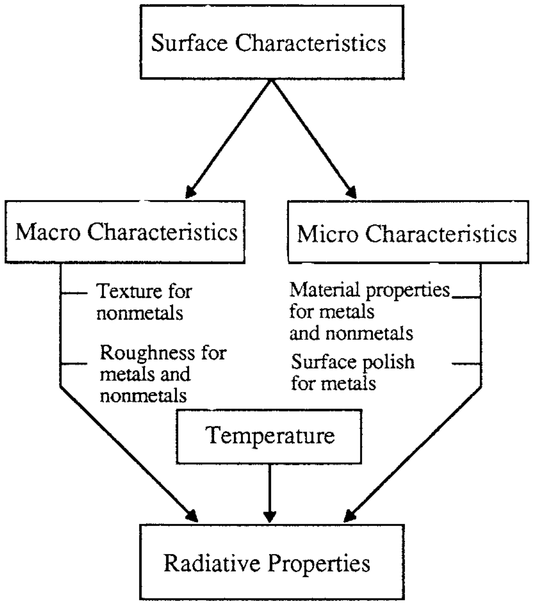

Figure 2 *Factors affecting radiative properties.*

temperatures permitted analytical calculation of the radiant output of the experimental panel. Second, knowledge of the temperatures and air speeds permitted the estimation of the rate of convective heat transfer. Finally, by controlling the environment, both realistic room conditions and extreme conditions could be simulated.

The experimental enclosure shown in Figure 3 consists of 53 individually controlled, heated wall sections as shown in Figure 4. (A complete description of the facility can be found in Fisher [1989, 1995] and the final reports for ASHRAE Research Projects 664 [Fisher et al.] and 876 [Pedersen et al. 1997]). The experimental radiant test panel was used on the floor, ceiling, and wall as shown in the figure. Validation tests performed by Fisher showed an absolute deviation of all surface temperatures of less than 0.2°C (Fisher 1989), thus demonstrating the ability of the control hardware and algorithms to maintain surface setpoint temperatures. The surface temperature is controlled in software, giving the investigator

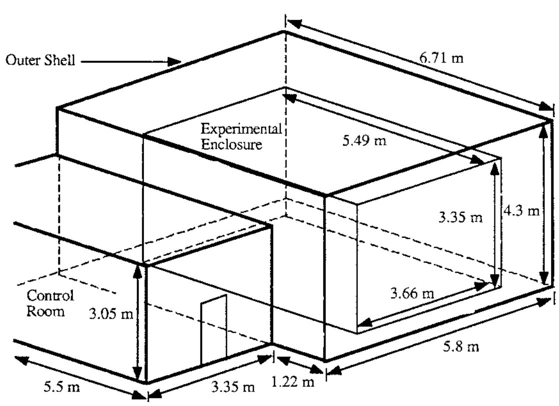

Figure 3 *Schematic of experimental facility (Fisher 1989).*

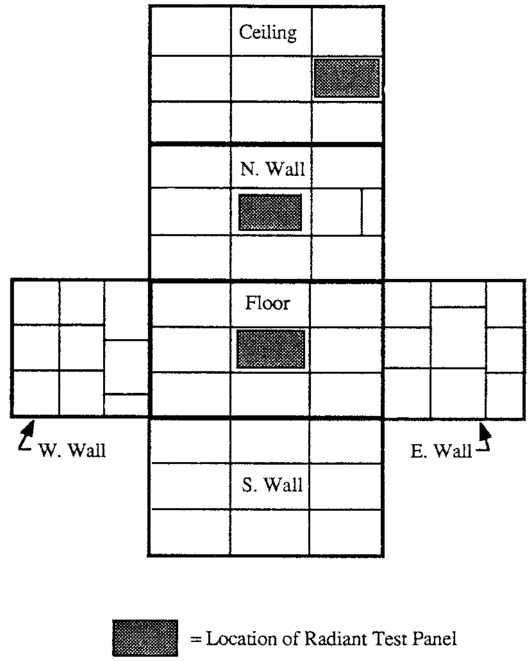

Figure 4 *Plan view of the experimental facility (Fisher 1989).*

flexibility in designing experiments with various inside surface temperature configurations.

Fast-acting radiant heat panels having a surface area of 1.15 m^2 (1.37 m × 0.84 m) were selected as test panels for the investigation. The panel surfaces were covered with various building materials to determine the impact of surface characteristics on panel performance.

A number of commonly used building materials representing a broad range of typical surfaces encountered in buildings were tested as radiant panel surfaces:

1. Plastic: A thin, clear, high-temperature plastic surface that was attached to the heating element of the radiant panel (abbreviated as *plastic*).
2. Vinyl: A typical vinyl floor covering 0.18 cm (0.07 in.) thick (abbreviated as *vinyl*).
3. White: A thin layer of paint on top of the heating element of the radiant panel (abbreviated as *white*).
4. Textured: A textured layer of paint on top of the heating element of the radiant panel (abbreviated as *textured*).
5. Medium textured: A slightly smoother textured surface (abbreviated as *med.-tex.*).
6. Carpet: A nylon fiber carpet with a pile height of about 1.02 cm (0.4 in.). The carpet was plush with a medium weave density.

IMPACT OF MACRO-SURFACE CHARACTERISTICS ON RADIANT PANEL EMITTANCE

Procedure

Knowledge of surface temperature and the temperature of the surroundings provides enough information to calculate the

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total hemispherical emissivity, ε. The net radiometer used in the experimentation has a spectral sensitivity from 0.25 µm to 60 µm and measures both incoming and outcoming radiation in order to determine the net radiative flux from a surface. For an object at 300 K, 98% of the blackbody radiant energy is contained in this spectral region. It is, therefore, reasonable to say that the net radiometer can be used to measure total emissivity.

The procedure for measuring hemispherical emittance is as follows. The material of interest is heated to a temperature above that of the surroundings.The surface temperature is recorded, as well as the temperature of the surroundings and the radiative flux measured by the net radiometer. The emissivity is calculated as

$$\varepsilon_s = \frac{q''_{net}}{\sigma(T_s^4 - T_\infty^4)} \tag{4}$$

The above calculation is only valid when the surroundings are at a uniform temperature and when the surroundings are much larger than the surface of interest. The experimental chamber satisfied both requirements. Additionally, the view factor of the radiometer to the surface must be very close to 1 to make sure the radiometer is viewing only the radiant panel surface. An average of five to seven surface temperature measurements was used to determine the surface temperature as shown in Figure 5.

Results

Although the texture of a surface can change its emittance, there is a limit to which the radiative properties of the surface can be changed. The emissivity by definition cannot be greater than 1. In addition, roughening a surface will always

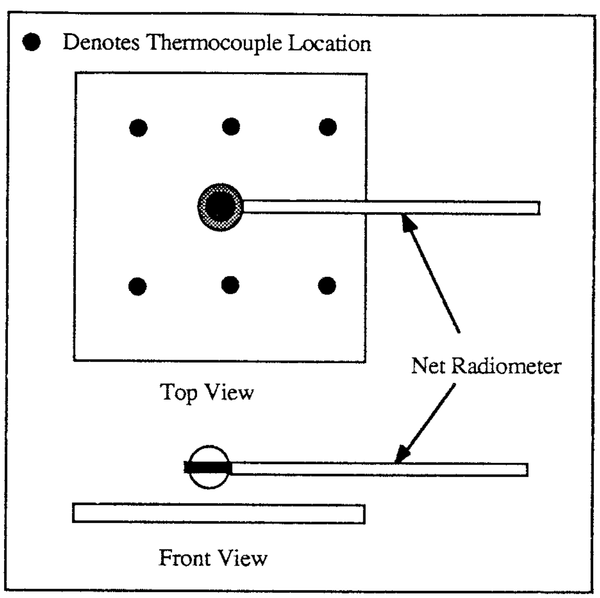

Figure 5 *Radiometer setup for emissivity measurements.*

tend to increase its effective emittance. As the surface roughness increases, the emissivity and absorptivity increase because of multiple reflection between surface peaks and valleys. This can be demonstrated theoretically by considering a textured surface as a collection of tiny cavities, with the "bumps" in the surface forming the cavity walls. To the extent that the cavity walls "see" each other, the cavity will appear to be thermally black (ε = 1) to a distant surface.

The fact that roughening a surface always increases its emissivity greatly simplifies the task of determining the emissivity of modern surfaces. For example, if a smooth painted surface has a measured emissivity of 0.9, a textured surface coated with the same paint must have an emissivity between 0.9 and 1.0. In this case, estimating the textured surface emissivity at 0.95 will result in a maximum error of ±5%.

The building materials for which emissivities are not reported in the literature can be summarized as petroleum-based polymers and fibers. These include plastics, vinyls, most carpet fibers, and many drapery fibers. The experimental measurements shown in Table 1 are reported for carpets, vinyl floor coverings, and a high-temperature plastic. In order to demonstrate the expected effect of texture on a high-emissivity surface, smooth and textured painted surfaces are also compared.

The uniformity of the results (even with errors and uncertainties considered) shows that the surfaces are very similar in terms of their radiative properties. Even though the materials

TABLE 1
Results of Emissivity Measurements for Various Surfaces

Surface	Surface Temperature (°C)	Emittance
Vinyl Floor Covering	35	0.93
Vinyl Floor Covering	45	0.98
Vinyl Floor Covering	55	1.04
Textured Paint	35	0.97
Textured Paint	45	0.97
Textured Paint	55	0.98
Plush Carpet	35	0.93
Plush Carpet	45	0.98
Plush Carpet	30	0.98
White Smooth Paint	35	0.96
White Smooth Paint	45	0.98
White Smooth Paint	55	0.99
White Smooth Paint	30	0.94
High-Temp. Plastic	35	0.90
High-Temp. Plastic	45	0.92
High-Temp. Plastic	55	0.95

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have different macro-scale properties, their radiative properties are similar.

An examination of the results leads to the following conclusions.

1. Emissivities of petroleum-based building materials fall between 0.9 and 1.0. Therefore, an estimated emissivity of 0.95 for these materials is reasonable and will not result in significant error in radiant panel calculations.
2. Texture has the expected effect of slightly increasing emissivity.

The apparent scatter in the data is actually a temperature bias most likely introduced by convection conditions that do not match the correction built into the net radiometer. The radiometer corrects for the natural convection from the surface to the radiometer in order to determine the radiative flux. It appears that this correction does not match actual flow conditions when the radiometer is located close to a surface. The data shown in Table 1 tend to support this hypothesis. The smoothest surfaces, which would tend to have the thickest boundary layers, show the greatest temperature dependence. The textured surfaces show less temperature dependence. For example, the first case reported in Table 1 (vinyl floor covering) shows a strong dependence on temperature. In addition, the measured emittance of 1.04 at 55°C is clearly in error. The result of this relatively insignificant, systematic measurement error is that the uncertainty associated with the direct radiation measurement is approximately the same as the uncertainty associated with the theoretical radiant exchange calculation based on measured surface temperatures and calculated view factors.

The uncertainty in the emittance measurements stems from uncertainty in measuring the actual surface temperature accurately. The average uncertainty for the emittance measurements is calculated to be ±0.06.

IMPACT OF MACRO-SURFACE CHARACTERISTICS ON ANGULAR EMITTANCE

Procedure

The primary piece of equipment used in the angular emittance experiments was a directional infrared thermometer. The thermometer measures the incoming thermal radiation incident on its detector. Knowing the approximate surface emissivity (in order to separate reflected from incident energy), the thermometer is able to determine a surface temperature by comparing the incident energy to that which would be emitted by a blackbody surface.

Through its optics, the thermometer views a part of the surface depending on its distance from the surface. At about 0.23 m (9 in.) from the surface, the “spot” size the thermometer sees is approximately 0.023 m in diameter. Since it has a narrow viewing angle, the thermometer is effectively making angular measurements. The thermometer is sensitive to radiation in the spectral range of 8 to 14 microns. Therefore, it compares the radiation it receives to what a blackbody would

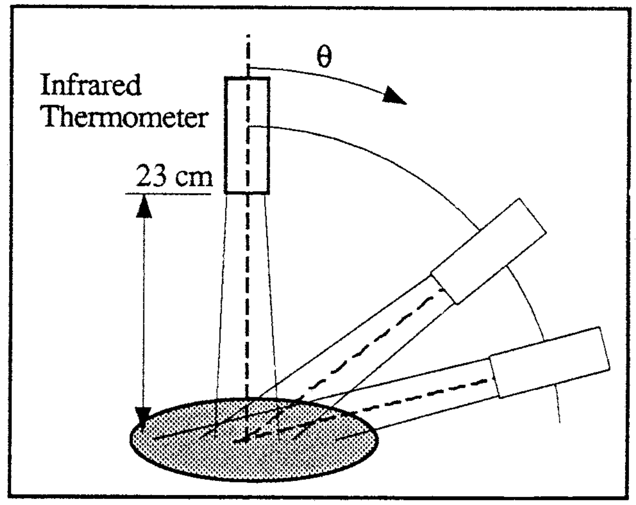

Figure 6 *Experimental setup for angular emissivity experiments.*

emit in that spectral region in order to determine the surface temperature. If the emissivity of the surface changes, then the measured surface temperature will also change.

The thermometer was positioned about 0.23 m from the panel surface. It was then rotated at a constant radius up to 80 degrees from the normal (see Figure 6). Temperatures were recorded at equal angle intervals.

The infrared thermometer receives a radiative flux proportional to

$$q'' \propto \varepsilon\sigma T_s^4 + \rho\sigma T_\infty^4 \qquad (5)$$

The thermometer views approximately the same spot at each angle, as shown in Figure 8. By selecting the approximate surface emissivity, the thermometer can distinguish between emitted and reflected energy coming from a surface. The temperature of the surroundings are measured by the thermometer. The emissivity was set to 0.95 for experiments using nonmetals. Therefore, the amount of reflected energy detected is much smaller than what is emitted from the heated surface. Since the actual surface temperature is constant and independent of the viewing angle, what the infrared thermometer displays as a change in temperature is actually a change in angular emissivity.

At zero degrees from normal, the surface temperature is determined. A radiative flux proportional to what the thermometer reads is then calculated using Equation 5. At other angles, the proportional radiative flux can also be determined using the measured surface temperature. The angular emissivity is then

$$\varepsilon' = \frac{\varepsilon_n\sigma(T_s^4 - T_\infty^4)}{\sigma(T_{s,n}^4 - T_\infty^4)} \qquad (6)$$

where

ε_n = normal emissivity for the surface,

$T_{s,n}$ = normal surface temperature measurement made by

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the infrared thermometer,

T_s = temperature measured by the infrared thermometer,

T_∞ = temperature of the surroundings (AUST).

Since the normal emissivity is not known exactly, it is more appropriate to calculate the ratio of the angular emissivity at a given angle to the normal emissivity. This ratio is expressed as

$$\varepsilon_r = \frac{(T_s^4 - T_\infty^4)}{(T_{s,n}^4 - T_\infty^4)} \tag{7}$$

where

ε_r = ratio of the emissivity to the normal emissivity.

Results

A diffuse surface radiates uniformly in all directions. Its emissivity is constant in all directions. All surfaces including nonconductors depart from the diffuse assumption. Figure 7 gives a representative description of the directional emissivity for a metal and nonmetal.

As shown in the figure, at viewing angles near the normal, both metals and nonmetals exhibit diffuse behavior. For metals, the emissivity remains nearly constant up to about 40 degrees from the normal, after which the emissivity increases until dropping off sharply to zero. For nonmetals, the typical trend is nearly constant emissivity to about 60 degrees from the normal. After that, emissivity sharply decreases to zero. For metals, the hemispherical emissivity is usually slightly larger than the normal emissivity, while for nonmetals, the opposite is true (Brewster 1992).

At first glance it appears that calculating radiant exchange using the assumption of diffuse behavior will result in large errors. There are two reasons why this is not true. First, for nonmetals the hemispherical emissivity is approximately 0.95 times the normal emissivity. That is, the error incurred by approximating the hemispherical emissivity as the normal emissivity is approximately 5%. It should be noted that for all simplified computational methods (two-surface method, mean radiant temperature method), it is the hemispherical emissivity that is used in the calculation.

The second reason why the diffuse assumption is reasonable despite the large differences at angles greater than 60 degrees is that the emissivity is multiplied by the view factor in the radiant exchange calculation. The view factor also exhibits a cosine dependence on θ; as θ approaches 90 degrees, the view factor approaches zero. Thus, in the region where departure from diffuse behavior is greatest, the view factor is the smallest.

The experimental procedure described earlier provides for a check on the diffuse assumption. The infrared thermometer is, however, limited in spectral range (8 µm to 14 µm), and the experimental apparatus and procedure is rather crude compared to what is required for precise measurements of diffuseness.

An initial check showed that the procedure was sufficient to differentiate between metals and nonmetals. Uncertainty calculations (Pedersen et al. 1997) confirmed the validity of the measurements. Figure 8 shows the angular emissivity normalized with respect to the normal emissivity for the high-temperature plastic, carpet, vinyl, and textured surface. The experimental uncertainty is shown for one of the carpet tests. A textured painted panel was compared to a previously measured smooth painted panel. Since the elemental surfaces of the textured panel are not co-planar, this panel is expected to be more diffuse than a similarly painted smooth surface. The experimental data demonstrate this effect. In fact, the emissivity ratios differ by more than 20% at 80 degrees from normal.

Caution must be exercised in interpreting the difference. A 20% error in the angular emissivity measured at 75 degrees represents less than a 3% error in the total power output of the panel. As stated previously, at large angles from the normal, large changes in angular emissivity have little impact on total panel output due to the magnitude of the view factors.

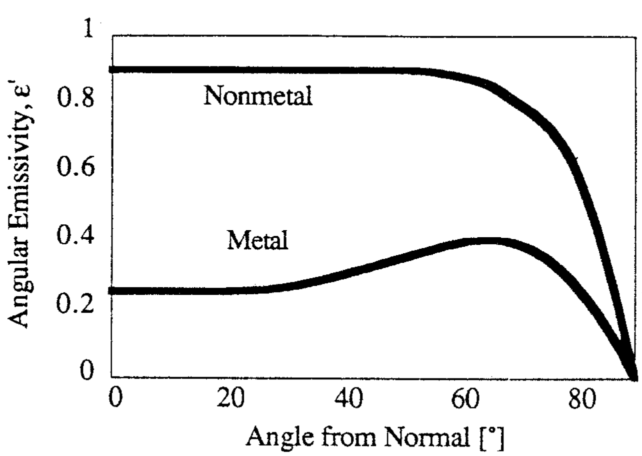

Figure 7 *Angular emissivity for nonmetal and metal (Incropera and DeWitt 1990).*

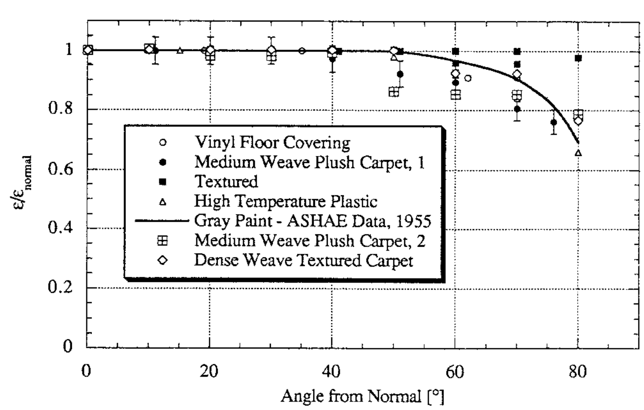

Figure 8 *Ratio of angular to normal emittance for various surfaces as a function of angle from normal.*

The data from these experiments also agree well with the ASHAE data (Umur et al. 1955). The results from the ASHAE experiments demonstrated that the hemispherical emissivity for surfaces such as asphalt, black paint, and gray paint was approximately equal to 0.95 times the normal emissivity. The results from the experiments were numerically integrated to obtain hemispherical emissivity ratios presented in Table 2.

As noted in the table, the data do indicate that the surfaces can be considered diffuse. Except for one of the carpet samples, all the other ratios are near 0.95, which is a typical value for a nonmetal. Although the most important conclusion that can be drawn from the data is that for all practical engineering calculations the surfaces are diffuse, there are several interesting features that should be mentioned. As has already been noted, texturing a surface does make it more diffuse.

TABLE 2
Ratio of Hemispherical to Normal Emittance for Various Surfaces Along with Average Uncertainty for the Measurements

Material	$\varepsilon/\varepsilon_n$ Hemispherical	Average Uncertainty
High-Temp. Plastic	0.92	±0.044
Vinyl Floor Covering	0.94	±0.054
Gray Paint (Umur et al. 1955)	0.95	-
Medium-Weave Plush Carpet	0.90	±0.078
Dense-Weave Textured Carpet	0.914	±0.090
Painted, Textured	0.97	±0.052

Although this may seem like a panel design aspect that warrants additional investigation, the potential return is small. It must be noted that these ratios do not necessarily indicate what surface has a higher hemispherical emissivity. Rather, they are a measure of "diffuseness." The diffuse assumption is an important approximation for radiant heat transfer. It allows for the definition of a view factor (shape factor or exchange factor), thus greatly simplifying radiant exchange calculations. The experiments show that although surface topography noticeably affects panel diffuseness, the diffuse assumption is valid for surfaces used in most buildings.

ESTIMATING IMPACT OF MACRO-SURFACE CHARACTERISTICS ON CONVECTIVE HEAT TRANSFER PANEL OUTPUT EXPERIMENTS

Panel Output Under Natural Convection Conditions

Six surfaces were tested in the natural convection experiments: plastic, vinyl, carpet, textured, medium textured, and white paint. The test panel was located on the ceiling, floor, and wall, as shown in Figure 4. Five to six thermocouples were used to measure surface temperatures. Two thermocouples were used to measure temperatures at the back of the panel,

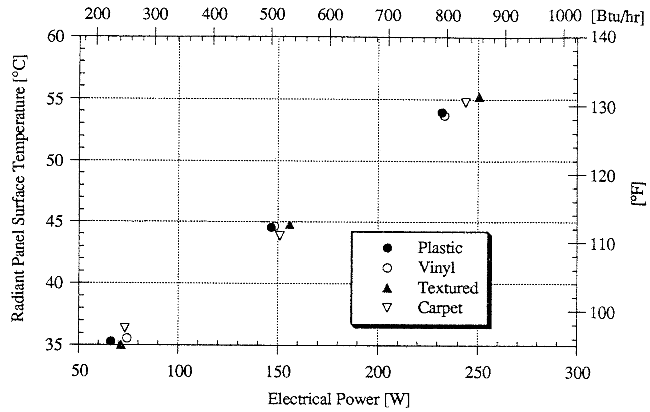

Figure 9 *Panel surface temperature vs. power input into the panels for ceiling panels.*

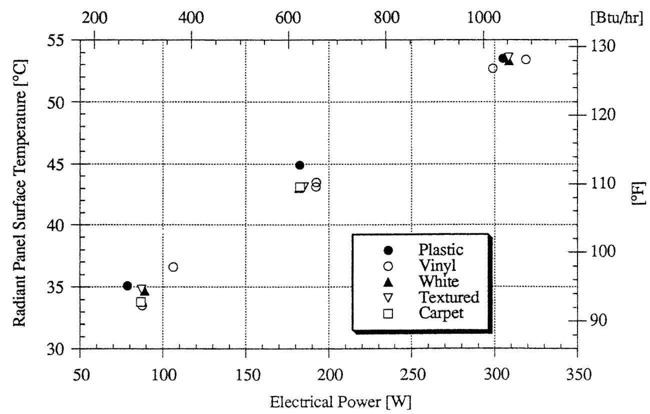

Figure 10 *Panel surface temperature as a function of power input to panel for floor tests.*

and for the carpet and vinyl surface tests, three thermocouples were used to determine the heating element temperature. The temperatures of all the other room surfaces were measured, allowing for a detailed calculation of the radiant exchange from the test panel to the rest of the room. For the floor tests, a direct measurement of the radiant flux was made using a net radiometer. Incidental measurements of air temperatures and flow rates were also made during the tests.

Three tests were made at each location and for each surface at approximate surface temperatures of 35°C, 45°C, and 55°C. The rest of the room surfaces were set to 25°C. Back losses were eliminated by heating the back of the test panel to the heating element temperature. The convective heat transfer from the panel was calculated from a heat balance on the panel.

Figures 9 through 11 display the surface temperature as a function of panel power for three different radiant panel locations—ceiling, floor, and wall. It is seen from the three figures that the surface temperature at a given power level is nearly the same regardless of the type of surface finish on the panel. For a constant power input and constant room surface temperature, a change in the experimental variable, surface tempera-

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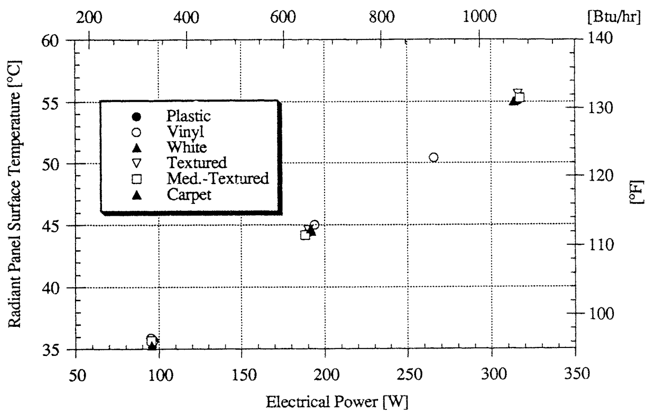

Figure 11 *Panel surface temperature as a function of power input to panel for wall tests.*

ture, could only be caused by either a change in the rate of convective heat transfer or a change in the rate of radiative heat transfer. Based on the hemispherical emittance and angular emittance experiments in the last section, the radiative flux was not expected to change for different surfaces. Since there was no change in surface temperature for different textures, the plots show that the different surface types are equivalent radiators. They also show that under the natural convection condition in the lab, surface texture had an insignificant effect on the rate of convective heat transfer. The floor tests (Figure 10) show a slight spread in the data, but the wall and floor locations show good agreement between the data.

Several points can be made about the figures. First, texture has no effect on the rate of heat transfer from the surface. The smooth painted white surface and the textured surface (Figures 10 and 11) have the same surface temperature for a given power input. Since the emissivity of the white surface is already high, the effect of giving the surface a rough texture is not significant. Second, there is good agreement between all the surfaces, supporting the fact that they all have similar radiative properties. The fact that some surfaces were slightly more diffuse does not have any impact either. Third, it must be noted that the carpet and vinyl surfaces agree with the other surfaces (which have no surface resistance) because back losses for all surfaces were made negligible. If this were not the case, surface temperature would be a strong function of the surface resistances. Finally, it is clear that the location of the panel does affect the panel output. Nearly 325 W were needed to maintain a surface temperature of 55°C for the wall location compared with roughly 250 W for the ceiling location.

The location of the panel, whether on the floor, ceiling, or wall, will affect the split between the fraction of total power output that is radiated to the room and the fraction of total power that is convected to the room. Figure 12 shows the radiative percentage of the total output for the floor and wall panels. Shown on this figure are both the experimental results and the theoretical results from the ASHRAE Handbook (ASHRAE 1996). The figure illustrates that the difference

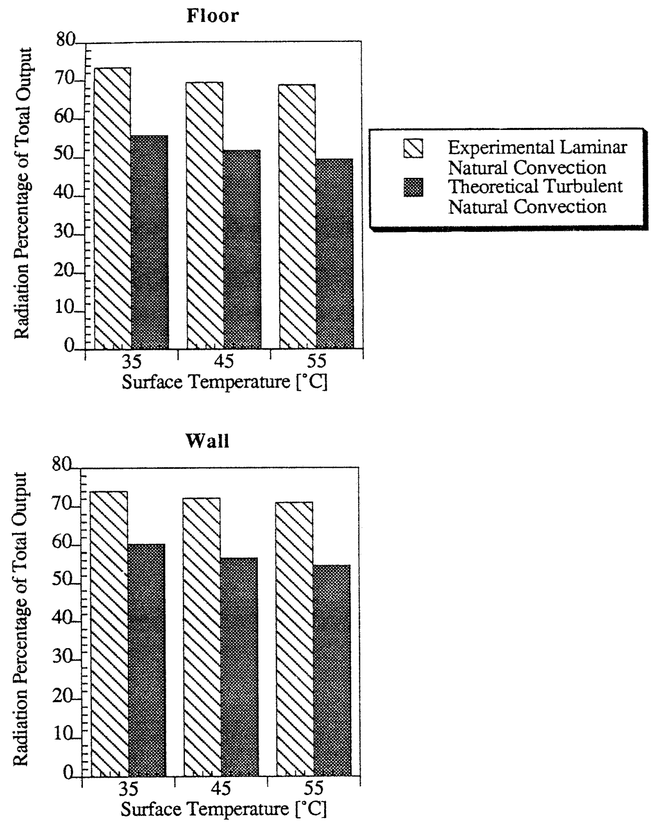

Figure 12 *Average radiation percentage of total output for floor and wall panels.*

between laminar and turbulent natural convection, in terms of the radiation percentage of total output, is approximately 15% to 18% for the floor and wall locations. Due to the size of the panels and the operating conditions, there was laminar natural convection from the panels, and, therefore, the radiative percentage of the total output is relatively high. The Handbook equations, based on the research done by Min et al. (1956), predicts turbulent natural convection from the panels, which lowers the radiative percentage of the total output. For the ceiling, both the experimental data and the theoretical equations show the radiative percentage of panel output to be approximately 90%. The difference in the experimental and theoretical is a result of differences in operating conditions and panel size and indicates the significance of natural convection and the importance of using the proper correlation. The experimental uncertainty in the radiation percentage of total output for the floor and wall panel is calculated to be approximately 8% and 9%, respectively.

Panel Output Under Forced Convection Conditions

Three surfaces were tested in the forced convection experiments: textured, vinyl, and carpet. The test panel was located on the floor in the center of the room in the same location used for the natural convection floor tests. A horizontal floor jet was discharged into the room from a 27.9 cm × 121.9

cm (11 in. × 48 in.) diffuser. The panel, which was located about .254 m (10 in.) from the air exit, was washed by the jet.

Two velocity probes measured air speeds 5.1 cm (2 in.) above the panel surface. An additional two probes measured air speeds 10.2 cm (4 in.) above the panel surface. Seven air temperature measurements were also made along with six panel surface temperature measurements. As in other tests, the panel power was kept constant during each test.

Six tests were made for each surface. Approximate air velocities of 1.0 m/s, 0.76 m/s, 0.51 m/s, and 0.30 m/s (200 fpm, 150 fpm, 100 fpm, and 60 fpm) were used along with two different panel surface temperatures: 25°C and 30°C. The rest of the room surface temperatures were controlled to 25°C. The inlet air temperature was approximately 21°C, although it was higher for the lower air speed tests.

The forced convection experiments performed were designed to answer the question: Does surface texture significantly affect the rate of forced convective heat transfer at low air speeds up to 1.0 m/s (200 fpm)? The experiments were designed to give average heat transfer coefficients. Therefore, the seven air temperature measurements were averaged in order to establish a reference temperature. Since the convective fluxes were fairly small in all of the experiments (always less than 65 W/m^2), the panel surface temperature was nearly isothermal. The six surface temperature measurements were also averaged. Additionally, the four air velocity measurements were also averaged in order to calculate a reference velocity. All the Reynolds numbers were calculated for the length of the panel (1.37 m, 54 in.). As a result, the heat transfer coefficients are not local values but average values for the entire surface.

The results of the forced convection experiments are shown in Figure 13. The figure displays the average heat transfer coefficient as a function of length Reynolds number. The figure indicates no obvious definitive trends in the data. At some velocities, the carpet surface has a higher heat transfer coefficient, while at other velocities, the vinyl or textured surface has a higher heat transfer coefficient than the other surfaces. A power curve fit of the data did show small differences between the surfaces. Yet all differences in the data based on the curve fit of the data were smaller than the experimental uncertainty. Therefore, for low air velocities (1 m/s and less), the results indicate no significant impact in convective output based on surface texture. At higher air velocities, one would expect to see differences due to surface roughness. The average uncertainty in the measurements is calculated to be nearly 25% due to small temperature differences and low convective fluxes.

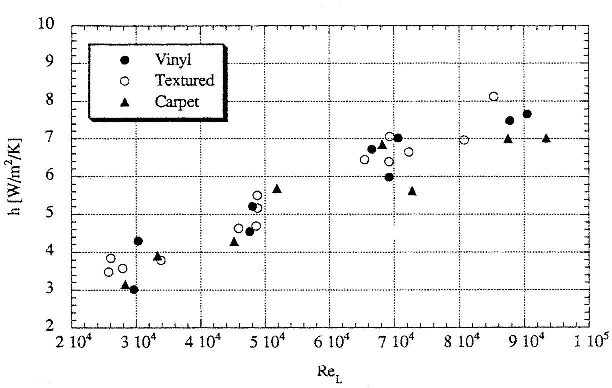

Figure 13 *Average convective heat transfer coefficient vs. length Reynolds number.*

Even though the air velocities were relatively low, it was found that the heat transfer from the surface was turbulent and not laminar.The most likely cause would be free stream turbulence in the airstream. This seems to indicate that the convection resulting from air drafts might be better predicted using a turbulent correlation rather than a laminar correlation, even when the Reynolds number would not suggest using a turbulent correlation.

CONCLUSIONS AND RECOMMENDATIONS

Investigation of modern building materials and radiant panel coverings showed emissivities on the order of 0.95 and diffuse behavior comparable or better than materials for which the diffuse assumption is typically used. Specific conclusions drawn from the experimental measurements are:

1. Surface emissivities of 0.90 - 0.95 can be used for engineering calculations of radiative heat transfer from most modern building materials without significant error. This includes all previously measured and cataloged paints (non-aluminum-based), woods, plasters, concretes, and fabrics, as well as the plastics, vinyls, carpets, and floor tiles measured as part of this study. Painted aluminum and metals behave like any other painted surface.
2. The diffuse assumption can also be used for engineering calculations of radiative heat transfer without significant error.
3. Changes in surface emittance due to texturing of the surface were not measurable for most of the surfaces and did not significantly affect panel output calculations for any of the surfaces tested.

The immediate objective of the research project with regard to convective heat transfer was to determine the impact of surface characteristics on the rate of convection for typical radiant panel coverings. The investigation showed that for the experimental configuration operating under both natural and forced convection, panel surface texture and material did not measurably affect the rate of convective heat transfer from the panel. Under the same operating and environmental conditions, carpeted and smooth surfaces convected heat from the panel at the same rate. Other results of the convection experiments are summarized as follows:

1. A single set of natural convection correlations is not applicable over the entire range of configurations and operating conditions typical of radiant systems. Both

turbulent and laminar flow regimes can exist for buoyantly driven flows in radiantly heated rooms.

2. Turbulent forced convection correlations may be applicable for flow velocities as low as 60 fpm.
3. The radiative percentage of total panel output for a ceiling panel is approximately 90%. This experimental value agrees well with what the ASHRAE Handbook predicts.
4. The fraction of total panel output convected from the radiant floor and wall test panel surface was as high as 30%. Based on the convection equations in the Handbook, this percentage can be as high as 50%. This fact highlights the importance of accurately predicting the rate of convective heat transfer for radiant panel applications.

A significant amount of research remains to be done in the area of convective heat transfer for radiant panel applications, including work in the following areas:

1. Development of natural convection correlations for surface temperature configurations typical of radiant systems—this research, which should include the effect of panel placement and cold surfaces, should produce correlations that are based on known environmental conditions and design parameters.
2. Development of correlations for both mixed and forced convection in radiantly heated rooms with various surface temperature configurations—the experiments should account for drafts, as well as buoyantly driven flows.
3. Determination of the applicability and range of existing correlations for radiant heating and cooling applications—although many natural and forced convection correlations exist, the literature does not provide designers of radiant systems with guidelines for selecting appropriate correlations for a given set of parameters.
4. Development of mixed convection correlations for hybrid systems that use radiant cooling panels in conjunction with a forced air ventilation system.

Since natural convection can be a significant portion of the total panel output, it deserves further research. While a significant amount of research on natural convection has been done, there is not a great deal of research addressing natural convection in the context of radiant heating. Future research needs to address the transition from laminar to turbulent flow in an enclosure in order to properly size and design radiant systems.

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NOMENCLATURE

Variables

AUST = average area weighted surface temperature
h = heat transfer coefficient, W/m^2°C
I = intensity, W/m^2 sr/μm
p''_{pc} = convective flux from panel, W/m^2
p''_{pr} = radiative flux from panel, W/m^2
Re_L = length Reynolds number = UL/v
T = temperature, °C
W = total hemispherical radiative power, W

Greek symbols

ε,ρ = surface emissivity and reflectivity (dimensionless)
σ = Stefan-Boltzmann constant
θ = zenith angle (angle from normal)
ϕ = azimuthal angle

Subscripts

b = blackbody
c = convection
e = element (water) or emitted flux
λ = spectral dependence
net = net radiant flux from radiant panel, W/m^2
r = radiation
s = surface
∞ = surroundings

Superscripts

$'$ = directional (angular) dependence

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