

sorption over the spectrum is proportional to that of a perfect black body. The combined radiation and convection coefficient in the normal temperature ranges has a value of slightly less than 2.0. In the usual installation, the actual performance is considerably greater due to transfer of high frequency radiation from the sun and lighting, which is practically independent of panel surface temperature. This mechanism was investigated in considerable detail by C. S. Leopold² who developed the concept of *Independent Radiant Transfer*, IRT, which he defined as follows:

$$\% \text{IRT} = \frac{\left(\text{Panel Transfer} - \text{Convection} + \text{Rad. due to temp. of other room surfaces} \right) 100}{\text{Energy of Source}} \quad (5)$$

When a method is available to evaluate the different mechanisms of radiation from luminaires, Equation 5 can be rewritten to eliminate the experimental error.

$$\% \text{IRT} = \frac{\left(\text{Short Wave Radiation} + \text{Long Wave Radiation} \right) \text{from lights absorbed by panel (Light)} - \text{absorbed by panel (Low temp from fixture surfaces)} 100}{\text{Energy Input to Lights}} \quad (6)$$

This method of analysis estimates the additional performance gained from lighting which is suspended in a room. It is a function of the type of ceiling and panel construction, and, above all, of the type of lighting fixture used. Table 1 shows some empirical results obtained both by model and full scale prototype tests. These percentages represent the additional pickup in Btu per sq ft of panel area as a percent of lighting input. For certain types of lighting, particularly incandescent, considerable improvement in performance may be obtained by using special copper phosphate heat absorbing paint on the panels.¹¹ It should also be noted that for incandescent lighting, a certain amount of the radiant energy from the lighting source is stored in the building structure, and will not materialize as a cooling load until a considerable time lag after the lights are turned on.¹²

ABSORPTION OF SOLAR RADIATION

Panels installed in the ceiling near large window areas will absorb quantities of energy from solar radiation, which are independent of panel surface temperature in the normal range, especially when a deflection device, such as a light colored venetian blind, is used to reflect the radiation upward to the panel. The quantity will depend on such factors as general window construction, window type and configuration, type of glazing, and type of shading device. Leopold reported an experiment with a model of exterior zone with 35 percent window strip of clear glass with white venetian blind inside.^{11,12}

With panels in the entire ceiling area, approximately 13 percent of the room load due to solar gain was absorbed by the panels as IRT, resulting in a *fictitious* h value of 2.77 for a panel temperature of 68.5 F with a room temperature of 78 F. A similar experiment using a panel area of 40 percent of the ceiling area in a suspended ceiling resulted in an h value of over 3, with the panels transferring approximately 6 percent of the room solar load independent of panel surface temperature. In this case, the total fictitious h value reflects not only the IRT but also the heat gain from surfaces in proximity of panels, as does Fig. 6.

USE OF TEST DATA

Example 1 illustrates the use of the performance data. It should be pointed out that this example is used only to

illustrate the significance of some of the existing test data. The data should not be extrapolated to cover any general condition which is different from that on which the tests in the reference material are based.

Example 1: Determine the heat transfer for the interior and exterior zones of a panel-air system installed in a multi-story building, 100 feet square, with 35 percent window strip of clear glass with white venetian blinds inside (data tilted 45 deg up). The ceiling panels are flush mounted, with a narrow (less than 10 ft) exterior zone fully paneled, and 12 ft by 4 ft panels forming 80 percent of the interior zone ceiling area. The panel surface temperature is assumed to be 65 F, with room and floor temperatures of 75 F. Lighting is from suspended fluorescent fixtures at 10 watts per sq ft, or 34.1 Btu/h per sq ft.

Solution a: (Interior Zone).

Radiation Transfer. The AUST will approach the floor temperature of 75 F. From Fig. 1, at AUST = 75 F and panel surface temperature = 65 F, the heat transferred by radiation is found to be 8.8 Btu per (hr) (sq ft).

Convection Transfer. The equivalent diameter D_e is $4 \sqrt{(12 \times 4) / (2(12 + 4))} = 6$. As the temperature of the room at approximately the mid-height is being used as a criterion, the convection transfer can be found from Fig. 3, using the curve for the ASHRAE Laboratory data, $D = 6$, and a temperature difference between room and panel = 10 F deg (75-65). This value is read as 7.0 Btu per (hr) (sq ft).

Forced Convection Increment. Assuming that the air outlets are installed to blow the full length of the panels, and assuming an average terminal velocity of 25 fpm, the additional heat transfer due to forced convection can be found from Fig. 2. The value of velocity times length is $25 \times 12 = 300$, and for laminar flow, the value of y from Fig. 3 is 4.9. Then, $h = y/1000 = 4.9 \times 25/1000 = 0.124$ Btu per (hr) (sq ft) (F deg). The heat transfer is therefore $0.124 \times 10 = 1.24$ Btu per (hr) (sq ft).

IRT. From Table 1 for suspended fluorescent direct lighting, IRT = 16 percent of the source energy. Since the panels form 80 percent of the ceiling area, the heat transfer per square foot of panel area is $0.80 \times 0.16 \times 34.1 = 4.2$ Btu (hr) (sq ft).

The total heat transfer per square foot of panel area is the sum of the radiation, convection, forced convection increment and IRT.

Radiation Transfer	= 8.8
Convection Transfer	= 7.0
Forced Convection Increment	= 1.24
IRT	= 4.2

Total Heat Transfer = 21.24 Btu per (hr) (sq ft).

Air must be introduced to absorb the remaining heat load and to provide ventilation and dehumidification. If the minimum air quantity required for ventilation and dehumidification is greater than that required to absorb the remaining heat load, the panel area can be decreased.

Solution b: (Exterior Zone). The transmission and solar loads should be calculated separately and expressed as Btu per (hr) (sq ft of panel area), using the full area of the exterior zone. Assume that the transmission load is 10 Btu per (hr) (sq ft) and the solar load is 30 Btu per (hr) (sq ft). With the assumed glass configuration (similar to that used in References 11 and 13) the value of IRT would be similar to that found in test conditions, or 13 percent of the source energy. Thus, IRT = $0.13 \times 30 = 3.9$ Btu per (hr) (sq ft).

The performance for the panel must be $(30 + 10) - 3.9 = 36.1$ Btu per (hr) (sq ft). Since the exterior zone will operate with a higher AUST due to the effects of warm wall and windows, Fig. 6 can be used to determine panel and air heat transfer. Follow the curve for a 10 F deg temperature difference ($t_a - t_p$) until the sum of the heat transferred to the panel (read on the ordinate) and the heat transferred to the air (read on the abscissa) is 36.1 Btu per (hr) (sq ft). Thus, at 0.63 cfm per sq ft of floor area, the heat transfer is found to be:

Panel	= 25.8 Btu per (hr) (sq ft)
Air	= 10.3 Btu per (hr) (sq ft)
	36.1

The panel may also remove an additional amount of IRT from the lighting system. If the lighting intensity is the same as for the interior zone, the total heat transfer to the panel would be:

Solar IRT	= 3.9
Panel Performance (Fig. 6)	= 25.8
Lighting IRT = 0.16×34.1	= 5.4

35.1 Btu per (hr) (sq ft)

The air supply (assumed here at a temperature of 60 F) would

Panel-Air Systems for Air Conditioning

Table 1 . . . Values of IRT for Lighting from Specific Tests

Full Ceiling—Flush Panel				
Lighting	Air Supply F	Type of Paint	IRT	Reference
1000 w Silver Bowl Suspended Incandescent	60	Heat Absorbing	33	3
	75	Heat Absorbing	35	3
	60	Regular	20.4	3
300 w Bare Bulb Suspended Incandescent	60	Heat Absorbing	16.7	3
	60	Regular	10.2	3
6 Single-Tube Fluorescent	60	Heat Absorbing	12.3	3
	60	Regular	3.1	3
4 Fluorescent Tubes Semi-Indirect	60	Heat Absorbing	24.5	3
	60	Regular	12.5	3
13 RLM Type 6 Suspended-Direct	—	Regular	16	10
13 RLM Type 6 Suspended-Indirect	—	Regular	18	10

then have to be increased to meet the remaining load due to lighting, people, and other sources.

Regardless of the quantity of heat removed by the panels, air must be introduced to provide ventilation and to remove the moisture introduced to the space by infiltration, people, and other sources. In addition, experience has shown that a definite minimum amount of air motion is desirable. Fig. 7 shows the air supply rate in cfm per square foot, and dew point, required for two room conditions and at two rates of infiltration. These curves were derived by assuming 95 F dry-bulb and 78 F wet-bulb for the infiltration air, and assuming one person per 100 sq ft of floor area. The infiltration rate was calculated for a space 11 ft from floor surface to underside of overhead slab.

The use of the curve for 4 hr per air change is not recommended unless the building under consideration is of very tight construction.

If the supply air is dehumidified by refrigeration, the dry-bulb temperature will be near the dew point, and some cooling will be achieved. Thus, the required panel area may be reduced and only part of the ceiling area used for the panel.

PANEL SURFACE TEMPERATURE

Surface temperatures discussed in this chapter are assumed to be average panel surface temperatures. Pipes or tubing are bonded or clipped to metal panels, or for plaster panels, the pipe is embedded in the plaster. The difference in temperature between the water and the panel surface will depend on the effectiveness of the tube bond to the panel, the conductivity and thickness of the panel, and the tube spacing.

Since panel cooling systems operate with much smaller temperature differences than panel heating systems, the difference between the panel temperature at the tube and between the tubes has much greater significance, and should be considered in the design of cooling panels.

Fig. 8 shows an approximate surface temperature distribution assuming a given heat transfer rate with several combinations of panel thickness and tube spacing.

The effect of the tube bond and heat transfer section from the tubing to the panel must be evaluated separately. The

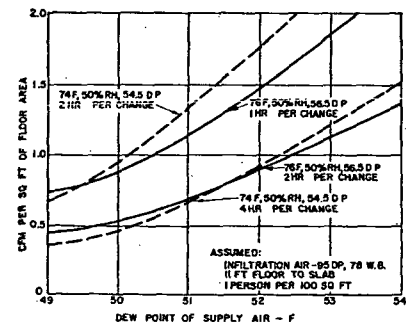


Fig. 7 . . . Minimum Dew Point of Supply Air Required to Maintain Room Conditions

efficiency of various types of systems may be represented by an overall resistance factor r which is defined as

$$r = \frac{t_a - t_p}{q} \quad (8)$$

where

r = overall resistance factor, (hour) (square foot) (Fahrenheit degree) per Btu.

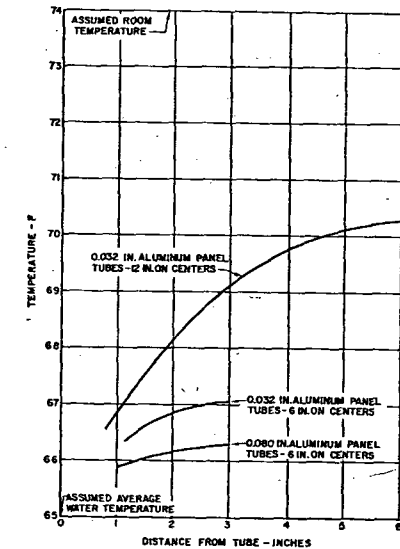


Fig. 8 . . . Estimated Panel Surface Temperatures for Flush Panels