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AN ENERGY EVALUATIVE COMPARISON OF A THERMAL COMFORT DESIGN MODEL

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ABSTRACT

This paper presents a new, improved method for designing radiant panel heating systems using accepted thermal comfort criteria, mean radiant temperature, and radiant asymmetry as bases for decision making. Peak design loads are calculated for radiant panel heating systems and convection heating systems in rooms with cold radiative interior spaces. An evaluative comparison of traditional methods and the new design method is also presented here. A parametric study of a number of enclosures establishes the differences among the ASHRAE Standard design method, the ASHRAE energy balance method, and the new thermal comfort design method, and the new thermal comfort/radiant asymmetry design method. The comparison variables include infiltration rates, metabolic rates, human location, posture, position, and radiant asymmetry within the enclosure.

INTRODUCTION

Traditional design methods for indoor environmental systems do not accurately respond to the individual's thermal condition within the space and do not adequately account for the complexities of human thermal comfort in radiative environments. Human thermal comfort is defined as that condition of mind that expresses satisfaction with the thermal environment. Thermal comfort is influenced primarily by air temperature and mean radiant temperature. Whenever artificial climates are created for human occupation, the aim is for the environment be designed so that individuals experience thermal comfort. Traditional design methods for radiative environments emphasize manipulation of air temperature alone, disregard radiant heat exchange within the enclosure, fail to account for thermal neutrality, and do not identify local discomfort. These methods often overestimate loads for radiant heating systems and underestimate loads for convective spaces resulting in overheated spaces. This simplistic approach to comfort sufficed as long as our space-tempering systems consisted primarily of convective (natural or forced) air in which the effects of cold radiative surfaces were small. However, in radiative environments the interaction of the human occupant in the space with the surfaces of the enclosure, as well as the surface-to-surface heat exchange, must be an integral part of the design considerations. The most frequently utilized design method is the ASHRAE Standard design method, in which a prescribed air temperature is chosen from the ASHRAE Comfort Envelope (ASHRAE 1989, p. 8.14) as the basis for room heat loss calculation. The ASHRAE Comfort Envelope, although it provides a simple and quick way of establishing load conditions, is applicable only for sedentary, normally clothed persons in low velocity, non-radiating convective environments in which the mean radiant temperature equals the room air temperature (ASHRAE 1989). The Fanger comfort charts (ASHRAE 1989, Fanger 1970) indicate that if a person's mean radiant temperature is raised the room air temperature can be lowered and vice versa. Radiant heating allows the room air temperature to be set a few degrees lower than that determined by the ASHRAE Standard method, while maintaining acceptable levels of thermal comfort. Based on the same principle, the room air temperature should be raised in environments with cold radiative interior surfaces. Determining the acceptable design room air temperature from the Fanger comfort charts requires knowledge of the person's mean radiant temperature, a time consuming task requiring an energy balance. Designers usually assume that the design air temperature equals the mean radiant temperature, thus simplifying the problem and reducing it to the ASHRAE standard design metthod. However, lower design room air temperature in radiant heating environments has an important effect on infiltration loads. Howell (1990) reports that in rooms with high infiltration losses, lowering the room air temperature can lead to a 16% reduction in room loads as compared to the ASHRAE standard design method. Zmeureanu (1988) reports a 72% overestimation in peak loads and daily loads, when the radiant system had been designed in a conventional manner as compared to one designed on a thermal comfort basis.

Another important problem associated with traditional design methods is non-uniformity of thermal comfort (hot and cold spots). Traditional methods have no provision for calculating thermal comfort throughout the environment, leaving this to the judgement of the designer. In addition, traditional methods have no provisions for calculating local discomfort due to radiant temperature asymmetry. Radiant temperature asymmetry (ΔT_{pr}) is the difference between the plane radiant temperature of the opposite sides of a small plane element (Olesen et al. 1989).

In the last 20 years there have been great advances in the prediction of human comfort (Fanger 1970, Olesen 1980). Design standards using thermal comfort were promulgated by ASHRAE 55-81, NKB guidelines (Nordic Committee on Building Regulations), and ISO 7730 (International Organization for the Standardization, 1983). These standards are the basis of the new design method described here: a method for the design of radiant heating ceiling systems and forced air convection heating systems.

In designing heating systems maximum sizing of equipment is important in order to maintain the proper air temperature for comfort. Additionally, the position of the radiant panels, cold walls, and glass surfaces have a large effect on the uniformity of thermal comfort, as well as localized discomfort due to radiant asymmetry effects. This study includes a comparison between the ASHRAE standard design method, the ASHRAE energy balance, the thermal comfort design method, and the thermal comfort/radiant asymmetry design method for a number of enclosures. Convective heating system loads have been compared to radiant heating system loads. Figure 1a shows a typical convective forced air heating system used in this study. This type of room can have cold walls, glass, and ceiling surfaces that may lead to lower mean radiant temperature. Figure 1b shows a typical radiant heating system within which the mean radiant temperature is higher than the optimum design air temperature of the space.

DESIGN METHODS

A parametric study established the differences among the ASHRAE standard design method, the ASHRAE energy balance method, the thermal comfort design method, and the thermal comfort/radiant asymmetry design method for convection heating and radiant heating ceiling panels. The ASHRAE Standard design method is an approximate method and does not account for the radiant cooling effect on surfaces in convective environments and the radiant heating effect in radiative environments. The base design air temperature chosen for this study, 75° F, is the mean air temperature in the ASHRAE Comfort Envelope, and the temperature commonly used by designers. The ASHRAE energy balance design method incorporates an energy balance utilizing the base design temperature of 75° F to determine room loads and surface temperatures in a rigorous and exact solution. The thermal comfort design method establishes the optimum room air temperature necessary for thermal comfort based on the Fanger Comfort Equation (Fanger 1970). As a result, the mean radiant temperature for the radiant panel heating system is increased above the ASHRAE Standard design room air temperature, thus allowing the optimum air temperature to be a few degrees lower than the ASHRAE design temperature. In convective environments with cold surfaces the mean radiant temperature will be lower than the ASHRAE standard air temperature, resulting in an increase of the optimum air temperature. Placing a radiant heating panel over a person will greatly reduce the necessary room air temperature, but may cause local discomfort from the radiant asymmetry effect. Radiant asymmetry criteria, which were incorporated into the Thermal Comfort/Radiant Asymmetry design method, restrict the radiant surface size and position. Each method is compared to the others, varying the infiltration load, metabolic rate, human posture (seated or standing), and worst case position in the room. Three room configurations were compared.

An eight-step design procedure was utilized: 1) Radiant heating ceiling panels were designed at winter design outdoor temperatures; 2) all internal loads (people, solar loads, ventilation, equipment) were set to zero; 3) steady state heat transfer was considered for building envelope to maximize loads. 4) thermal neutrality (thermal comfort) was maintained within acceptable limits (using the Fanger Comfort Equation) at the coldest and warmest points in the room at winter design conditions; 5) uniformity of thermal neutrality (thermal comfort) was maintained within acceptable limits; 6) local thermal discomfort due to asymmetric thermal radiation was maintained within acceptable limits (radiant temperature asymmetry); 7) local discomfort due to droughts and vertical air temperatures difference were not evaluated by these design methods. The design procedures were based on mean air velocities *0.15 m/s* which are within local comfort levels (Olesen 1985). It has been found through tests of nine heating systems that vertical air temperature differences and in most cases mean air velocities were within acceptable comfort range when the room was well insulated and thermally neutral (Olesen 1986). In this study the design air room temperature was assumed to be uniform throughout the occupied zone (average space air temperature) without any temperature stratification; 8) infiltration was included.

Optimum Design Method

Figure 1 shows a logic diagram of the design methods (Steps 1 through 8 for radiant panel heating systems).

1. The occupant's clothing value *(ICL, CLO)*, activity level *(M, MET)*, and body posture (standing/sitting) are specified.

2. Room geometry (including windows, doors, and radiant panel locations) is entered into the computer. The designer specifies the floor grid so that the room's thermal profile is evaluated at predefined grid points. The designer also specifies heat transfer parameters (R-values, design outside air temperatures, etc.).

3. Space air temperature T_a in the room is specified by the model.

4. Surface temperatures T_i are calculated by the model using the matrix equation 11 in an iterative solution until equilibrium conditions are achieved.

5. Mean radiant temperature T_r is found at the coldest point in the room.

6. The design space air temperature T_{ad} is calculated from the determination of Fanger's Comfort Equation, equation 12.

7. The computer model compares the T_a derived in step 4 to the T_{ad} calculated in step 6. If $T_a \neq T_{ad}$ then T_{ad} is specified and steps 4 to 6 repeated, until there is convergence.

8. Radiant temperature asymmetry ΔT_{pr} is calculated vertically and horizontally at the critical areas. If ΔT_{pr} is not within an acceptable range, then the heating system should be modified. The NKB guidelines define the following limits: ΔT_{pr} <10° K (18° F) for cold windows; ΔT_{pr} > 5° K (9° F) for warm ceilings. Figure 1 also shows a logic diagram for convection heating systems (steps 1 to 8 for convective systems.) The procedures used are similar to the radiant heating panel systems. The space air temperature T_a is adjusted until optimum thermal comfort is achieved at the coldest point in the room (i.e., $T_a \equiv T_{ad}$), and the design air temperature T_{ad} is determined. Surface temperatures are calculated by the program using the matrix equation 10.

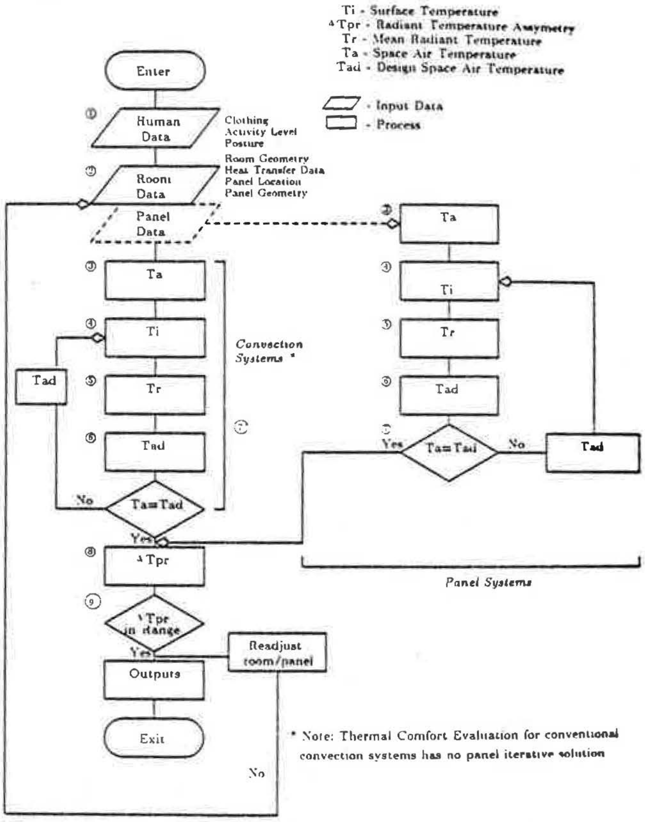


Figure 1 - Logic Diagram of the Thermal Comfort Design Method

ASHRAE Standard Design Method

The ASHRAE Standard heating load design method was used for both convection and radiant heating systems as set forth in Chapter 25 of the ASHRAE Fundamentals (ASHRAE 1989).

$$q_{cond_i} = U_i\,(T_{\infty} - 75) \tag{1}$$

where:

q_{cond_i} = total energy flux conducted by surface i - $\frac{btu}{h * ft^2}$

U_i = design U-value for each surface - $\frac{btu}{h * ft^2}$

T_{∞} = outdoor design temperature - °F

The infiltration load is:

$$q_{inf_i} = 0.0183 * V * (T_\infty - 75) \quad (2)$$

Ceiling heating panel surface temperatures are approximated using the following equation (ASHRAE 1987) based on the net heat q_r transformed by radiation from the panel:

$$q_r = \left\{\frac{(Tp + 460)}{100}\right\}^4 - \left\{\frac{(AUST + 460)}{100}\right\}^4 \quad (3)$$

T_p = panel surface temperature - °F

AUST = average unheated surface temperature. It is the surface temperature/area weighted average of the surfaces the panel faces (ASHRAE 1989).

Optimum Design Methods

The heating loads model can be incorporated into the design methodology by using a technique known as the "heat balance method." The general approach is to write an energy boundary equation for each enclosing surface in addition to an equation for room air mass. For this method, the following assumptions are made: a) steady-state heat transfer throughout all surfaces; b) isothermal surfaces; c) gray, diffuse surfaces; d) uniform space air temperature; e) negligible heat capacity of space air mass; f) surface emittance above 0.90.

The general surface boundary equation for each interior surface of N interior room surfaces is the sum of energy conducted q_{cond_i}, convected q_{conv_i}, and net radiation q_{rad_i} at surface i:

$$q_{cond_i} + q_{conv_i} + q_{rad_i} = 0 \quad i=1,N \quad (4)$$

$$q_{cond_i} = \frac{(T_\infty - 75)}{\Sigma R_j} \quad (5)$$

$$q_{conv_i} = h_i (T_i - T_a) \quad (6)$$

where: ΣR_j = building resistance from inside surface
T_i = inside surface temperature - °F
T_∞ = outside air temperature - °F
T_a = design space air temperature - °F

The convection film coefficients h_i were obtained from the ASHRAE 1989 Fundamentals and Howell (1990). The convection film coefficients were extracted from the combined film coefficients (convection and radiation) used by the ASHRAE Standard method, so that an appropriate comparison could be undertaken. The coefficients should be adjusted to accommodate differences in non-radiantly heated and radiantly heated environments. The following convection film coefficients were used: walls = . 49, floor/ceiling = .712, glass = .49, heated ceiling panel = .712. In the case in which surface emittances are higher than 0.90 (reflections are small) then the net radiation exchange can be expressed as:

$$q_{rad_i} = \varepsilon\, \sigma_i \,\Sigma_{j=1}^{N} A_{ij} (T_i - T_j) F_{ij} \quad (7)$$

where: $A_{ij} = \frac{T_4^i - T_4^i}{T_i - T_j}$ linearized substitution factor
ε = surface emittance
σ_i = Stefan-Boltzman constant
T_i = temperature of surface i - °F
F_{ij} = shape factor from surface i to j

Substituting equations 5, 6, and 7 into the energy balance equation for non-heated surfaces yields:

$$\frac{1}{\Sigma R_j}(T_\infty - T_i) - h_i (T_i - T_a) = \varepsilon_i\, \sigma \Sigma_{j=1}^{N} A_{ij} (T_i - T_j) F_{ij} \quad i=1,N \quad (8)$$

A heat balance equation can also be written for the space air. Assuming negligible heat capacity of the air relative to the more massive building elements, the sum of all heat flow to the air must be zero, yielding:

$$\Sigma_{j=1}^{N} h_i A_i (T_i - T_a)\, Q_{inf} \quad (9)$$

where: A_i = area of surface i (ft2)
Q_{inf} = sensible infiltration load introduced into space (Btu/h)

Automated shape factors from occupant-to-surface and surface-to-surface are incorporated in the design method. The *shape factor* (angle factor, configuration factor) is used to describe the distribution of radiation about a room and is defined as the fraction of diffusely radiated energy leaving one surface that is incident on another surface. Occupant-to-surface shape factors in rectilinear enclosures have been determined by the authors (Summers et al. 1983). Occupant-to-surface shape factors for cone heaters (skewed surfaces) have also been developed and can be incorporated into the design methodology (Steinman et al. 1988).

People, equipment, ventilation and supplemental loads were set equal to zero for the purpose of this study . Simultaneous solution of equation is performed by matrix solution. For the convection environment equation 8 is expanded:

$$X_{ij}\, T = \Omega_j \quad (10)$$

where:

$$X_{ij} = -\varepsilon_i\, \sigma A_{ij} F_{ij} + \delta_{ij}\left[h_i + \varepsilon_i\, \sigma\, \Sigma_{k=1}^{N} A_{ik} F_{ik}\right] \quad i=1,N\ j=1,N$$

$$\Omega_j = \frac{1}{\Sigma R_i} T_\infty + h_i T_a$$

$T = T_i, T_2, \ldots\ldots T_N$ N=total number of surfaces

$\delta_{ij} = 1$ when i=j and $\delta_{ij} = 0$ when $i \neq j$

Simultaneous solution for panel heating environments equation (6) for unheated surfaces and equation (6) for air yields:

$$X_{ij}\, T = \Omega_j \quad (11)$$

where:

$$X_{ij} = -\varepsilon_i\, \sigma A_{ij} F_{ij} + \delta_{ij}\left[h_i + \varepsilon_i\, \sigma\, \Sigma_{k=1}^{N} A_{ik} F_{ik}\right] \quad i=1,N\ j=1,N$$

$X_{ij} = h_i A_j$ i=N j=N

$$\Omega_j = \frac{1}{\Sigma R_i} T_\infty + h_i T_a$$

$$\Omega_j = T_a\left[\Sigma_{j=1}^{N} A_j F_j + q_{inf}\right]$$

$T = T_i, T_2, \ldots\ldots T_N$ N=total number of surfaces

The air temperature used in the matrix solution equations 10 and 11 must satisfy the linearized Fanger Comfort equation (Fanger 1970) such that:

$$T_a = T_{comf}\left[1 + \frac{dT_a}{dT_r} - \frac{dT_a}{dT_r}\right] \quad (12)$$

In order for the comparison to the ASHRAE Standard method to be valid, T_{comf} must be set at 75 °F (Fanger 1970) for typical winter clothing of 1.0 *clo*, relative air velocity < *.15 m/s* and, sedentary activity *(1 met)*, and $dT_a / dT_r = 0.93$ from Figure 2.

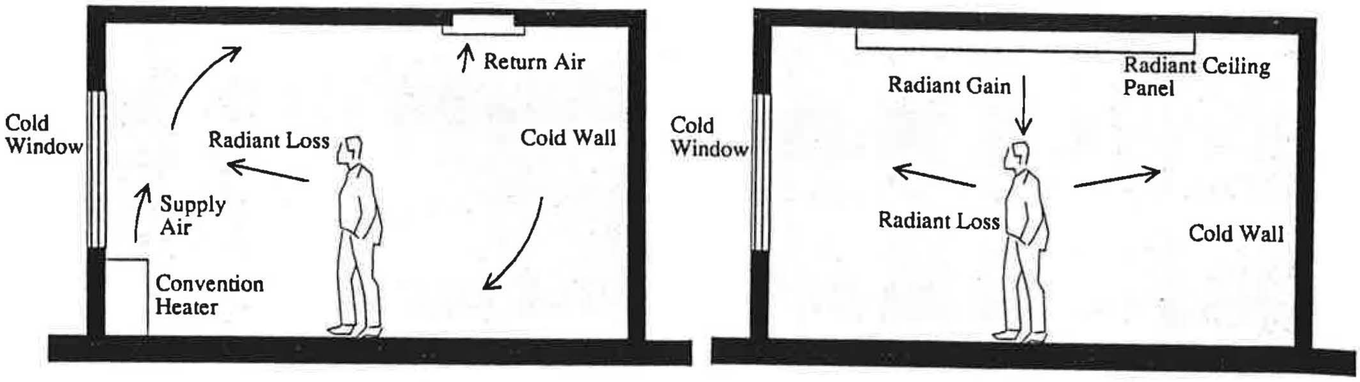


Figure 2 - Convection and Radiant Heating System Diagram

When mean radiant temperature T_r equals the room air temperature, then T_{comf} equals 75°F, the temperature which must be used for comparison to the ASHRAE Standard design method. When T_r increases, the room air temperature T_a decreases to accommodate comfort and vice versa. Other values used in this analysis are as suggested by Fanger (1970):

Medium Activity: T_{comf}= 59°F, and = $dT_a / dT_r = 0.85$

High Activity: T_{comf}= 45°F, and dT_a / dT_r= 0.73

Mean radiant temperature T_r is determined at any specific point in the room based on the following equation (Fanger 1970, ASHRAE 1987).

$$T_r^4 = \frac{1}{\sigma} \beta_1 F_{p-1} + \beta_2 F_{p-2} + \ldots\ldots + \beta_N F_{p-N}$$

where: β_i = surface radiosity (btu/h sf)

F_{p-1} = person to surface shape factors

σ = Stefan-Boltzman constant

RESULTS

The validity of the design method was established by means of a parametric investigation. A number of cases were analyzed for forced air heating and radiant panel ceiling systems in order to determine the effect of various parameters on an enclosure's heat loss. Three different room configurations were designed and compared as shown in Figure 3: Room I is a 30'-0" x 15'-0" x 8'-0" typical office, with one window and exposed ceiling; Room II is a 30'-0" x 30'-0" x 9'-0" space, which has a large room volume in order to determine the effects of higher infiltration rates; and Room III is a 30'-0" x 30'-0" x 9'-0" space with a window and exposed ceiling demonstrating radiantly cold room surfaces.

ASHRAE Standard Design method

A comparison of the three rooms indicates that the ASHRAE Standard design method predicts lower room loads that the ASHRAE Balanced method for panel heating systems. Table A documents a deviation of 5.6% and 18.2% for convective and radiant heating systems respectively in Room I at 2.0 ACH. This deviation is a result of the ASHRAE Standard method crudely approximating radiant balance: predicting lower surface temperatures for radiant heating systems and higher surface temperatures for cold radiating surfaces with convective heating systems.

Infiltration

An analysis of the effect of low infiltration rates indicates that load prediction differences between the ASHRAE Standard method and the thermal comfort method are negligible for both forced air and radiant panel heating systems. Table A shows a 1.3% deviation in load for convective and a 3.9% deviation for radiant panel heating at 0.5 ACH. At higher infiltration rates, the ASHRAE Standard method overestimates radiant panel loads while underestimating convective heating loads. Table C shows that for Room II, the ASHRAE Standard method overestimates the radiant heating system load by 5.0% to 14.7% for 1.0 to 4.0 ACH as compared to the Thermal Comfort design method. Similarly, Rooms I and III show deviation of 0.8% to 8.2% and 2.6% to 5.6%. Table C also shows that the ASHRAE Standard method underestimates convective heating loads for Room II by 3.6 to 5.3% for 1.0 to 4.0 ACH. Rooms I and III indicate deviation of 3.6 to 5.0% and 2.9 to 6.7% respectively. These results are in direct agreement with Howell (1990). The close agreement is explained by the fact the the ASHRAE Standard method underestimates surface temperatures for radiant heating and overestimates surface temperatures for convective cooling thus approaching optimum design conditions.

Radiant Temperature Asymmetry

The Thermal Comfort/Radiant Asymmetry method requires both optimum temperature and reduction of local discomfort from radiant temperature asymmetry effects. These requirements result in increased radiant heating panel area in order to reduce vertical radiant asymmetry and/or reduction of the cold glass/wall surfaces in order to achieve radiant symmetry. Rooms I and II indicate that the design of radiant heating panels with the additional requirement of radiant asymmetry effects increased the required panel area by 40-100%. Table A shows that for Room I at 1.0 ACH the vertical radiant asymmetry was reduced from 21.3°F to 11.0°F as compared to the ASHRAE Standard method. This reduction required an increase in panel surface area from 168 to 200 square feet. Consideration of radiant temperature asymmetry over and above thermal comfort considerations shows a modest reduction in room loads of 1.0-3.0% as compared to the ASHRAE Standard method.

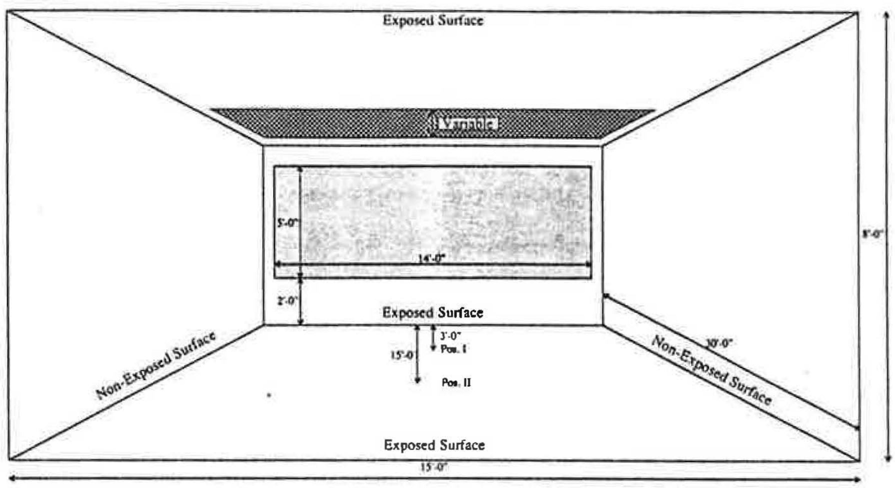
Metabolic Rate

The ASHRAE Standard design method only considers sedentary activity levels. However, designers commonly use the ASHRAE Comfort Envelope in determining heating loads in spaces in which occupants are under higher activity levels, due to the lack of appropriate information. Table D shows that for Room III the ASHRAE Standard method overestimated room loads by 35.4% at medium activity level, and 66.5% at high activity level. Rooms I and II showed similar results. This deviation is explained by the reduction in design room temperature in order to maintain thermal comfort at each activity level, as compared to the ASHRAE Standard method temperature of 75°F. The corresponding temperature reduction is documented in Table D.

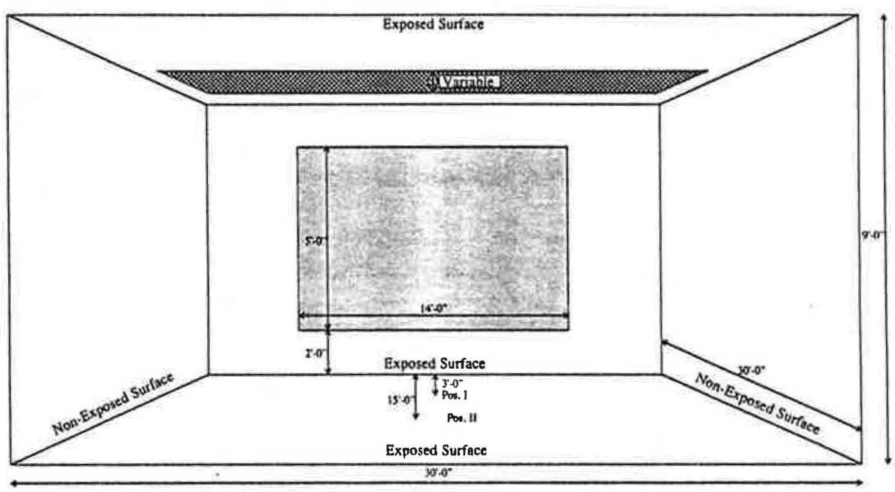
Posture Position

The analysis of the results indicates that there are small differences in heating loads as a result of the change in human posture (standing or seated). Table E shows an increase in in loads for convective heating systems and decrease in loads for radiant

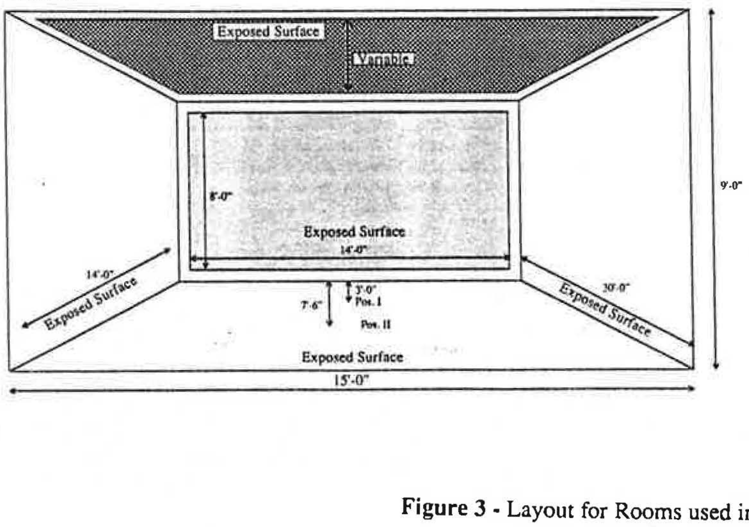
Room I



Room II



Room III



Base Temperature : Room Air - 75 °F (Values at Optimum)
Outside Temperature :10 °F
Emissitivities :.90
Metabolic Rate :variable
Clothing :1clo
Air Velocity :<.15 m/s
Relative Humadity :50%

Infiltration	1.0 ACH		
	2.0 ACH		
	3.0 ACH		
	4.0 ACH		
R-Values/	Walls	:10.0	.042
V-Values	Ceiling	:15.0	.063
	Floor	:5.0	.160
	Glass	:.87	.58
	Interior Walls	:5.0	.16

Figure 3 - Layout for Rooms used in the Parametric Study

heating systems of 0.9% to 2.3% for Room I. Rooms II and III also show negligible variation in room loads. Posture position has an important effect on radiant temperature asymmetry. A high radiant temperature asymmetry will cause an increase in panel area in order to decrease the vertical differences. Table D shows that the radiant temperature asymmetry increased from 11.0°F to 16.5°F when posture position changed from seated to standing. The radiant panel area needed to be increased to 224 square feet in order for the vertical radiant asymmetry to comply with accepted standards. Horizontal radiant asymmetry also increased from 15.0°F to 19.5°F. Rooms I and II exhibited similar increases, as it is documented in Table E.

Design Location

Two locations were compared in each room in order to optimize thermal comfort, position I at 3'-0" from the window surface and position II at the center of the room. Tables A and B show that evaluating loads at the center of the room results in 5% load reduction for either the convective or radiant panel systems. However, designing at the center of the room can lead to high vertical radiant temperature asymmetry. For example, Tables A and B show that for Room I at 1.0 ACH the radiant temperature asymmetry at position II is 7.1°F requiring 168.5 square feet of radiant panel for thermal comfort. However, position I indicates a radiant temperature asymmetry of 20.9°F, which is unacceptable. The design solution at position I requires a 280 square feet of panel area resulting in a radiant temperature asymmetry of 11.0°F, which is within acceptable levels. Under this design condition position II exhibits an unacceptable radiant temperature asymmetry of 17.2°F. Similarly, the design solution for Room II, position II results in a radiant temperature asymmetry of 9.1°F at 336 square feet and radiant temperature asymmetry of 20.9°F at position II. The design solution for position I requires an increase in panel area to 672 square feet leading to similar discrepancies in radiant temperature asymmetry.

CONCLUSIONS

The ASHRAE Standard design method does not accurately account for radiation balance in the room. When it is compared to an accurate energy balance method (ASHRAE Balanced) predicts lower room loads for panel heating and higher room loads for convective heating systems. This allows the ASHRAE Standard to more closely approach the Thermal Comfort design method. The ASHRAE Standard method overestimates room loads (15%) for radiant panel heating systems for high infiltration rates (3.0-4.0 ACH). Additionally, we have found that the ASHRAE Standard method can underestimate convective heating systems in cold radiative environments. It does not account for local discomfort due to radiant temperature asymmetry criteria, which can necessitate increased radiant panel area (40-100%), reduced glass area, and increased building insulation. Radiant temperature asymmetry considerations are important at low or high infiltrations rates. Activity level has a direct and substantial effect on prediction of room loads. The ASHRAE Standard method does not provide for the determination of different room loads at different human postures This was not important as different postures were found to have a negligible effect on room loads. However, human posture does have a dramatic effect on radiant temperature asymmetry and local discomfort, which the ASHRAE Standard method does not take into account.

We concluded that the choice of design location was more preferable near a window with a radiant heater overhead than than in the center of the room. However, designing in the center of the room, for convective heating systems, may be adequate. More study is required in order to determine an optimum design location.

In general radiant heating panel systems require less heating load output than convective heating systems when designed using the new Thermal Comfort/Radiant Asymmetry design method. This is due to reduced infiltration loads, lower room air temperature, and reduced ceiling loads where the panel is located.

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TABLE A - Comparison of Infiltration Rate Effect.

Room I (15'-0" 30'-0" 8'-0") - Position x=3'-0" y=7'-6"

System	Design Method	Infiltration - 0.5 ACH								Infiltration - 1.0 ACH							
		Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%	Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%
		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.	
Convection	ASHRAE STD	9722.0	[illegible]	[illegible]	49.3	NA	NA	NA	- - -	10863.0	70.0	75.0	49.3	NA	NA	NA	- - -
	ASHRAE BAL	7818.0	[illegible]	[illegible]	46.2	[illegible]	9.4	4.4	-11.	9958.0	70.0	75.0	46.2	66.8	9.1	9.1	-8.3
	TC	9839.0	[illegible]	[illegible]	48.5	[illegible]	9.5	4.5	1.3	11252.0	70.0	79.6	48.5	70.1	9.5	9.5	3.6
	TC/RA	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR
Panel	ASHRAE STD	9034.0	[illegible]	[illegible]	[illegible]	NA	NA	NA	- - -	10175.0	168.0	75.0	127.0	NA	NA	NA	NA
	ASHRAE BAL	9696.0	[illegible]	[illegible]	[illegible]	[illegible]	10.7	26.9	20.6	12082.0	168.0	75.0	139.6	87.0	15.3	24.3	18.7
	TC	9470.0	[illegible]	[illegible]	[illegible]	[illegible]	10.3	23.8	-5.4	10366.0	168.0	69.8	127.9	80.4	14.2	21.3	1.9
	TC/RA	9014.0	[illegible]	[illegible]	[illegible]	[illegible]	13.9	7.9	0.3	10092.0	280.0	71.0	109.0	79.3	15.8	14.0	0.8

Note: Person is Sedentary and seated

Room I (15'-0" 30'-0" 8'-0") - Position I x=3'-0" y=7'-6"

System	Design Method	Infiltration - 2.0 ACH								Infiltration - 4.0 ACH							
		Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%	Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%
		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.	
Convection	ASHRAE STD	15145.0	[illegible]	[illegible]	49.3	NA	NA	NA	- - -	23709.0	70.0	75.0	49.3	NA	NA	NA	- - -
	ASHRAE BAL	14240.0	[illegible]	[illegible]	[illegible]	[illegible]	9.1	4.4	-5.6	22804.0	70.0	75.0	46.3	66.8	9.1	4.4	-3.8
	TC	15731.0	[illegible]	[illegible]	[illegible]	[illegible]	9.5	4.5	3.9	24900.0	70.0	79.6	48.5	70.1	9.5	4.5	5.0
	TC/RA	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR
Panel	ASHRAE STD	14228.0	224.0	[illegible]	[illegible]	NA	NA	NA	- - -	22333.0	336.0	75.0	131.0	NA	NA	NA	- - -
	ASHRAE BAL	16814.0	224.0	[illegible]	[illegible]	[illegible]	12.9	22.4	18.2	27520.0	336.0	75.0	159.6	106.1	23.8	24.6	23.2
	TC	13999.0	224.0	[illegible]	[illegible]	81.9	16.2	19.6	-1.6	20520.0	336.0	63.0	132.6	87.9	20.3	20.1	-8.1
	TC/RA	13798.0	392.0	[illegible]	[illegible]	80.9	16.1	8.6	-3.1	20491.0	392.0	63.4	126.7	87.5	20.1	16.4	-8.2

Note: Person is Sedentary and seated

TABLE B - Comparison of Infiltration Rate and Location.

Room I (15'-0" 30'-0" 8'-0") - Position II x=15'-0" y=7'-6"

System	Design Method	Infiltration - 0.5 ACH								Infiltration - 1.0 ACH							
		Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%	Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%
		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.	
Convection	ASHRAE STD	[illegible]	[illegible]	[illegible]	46.3	NA	NA	NA	- - -	10863.0	70.0	75.0	49.3	NA	NA	NA	- - -
	ASHRAE BAL	[illegible]	[illegible]	[illegible]	46.2	69.6	2.0	0.1	10.4	9958.0	70.0	75.0	46.7	69.6	2.0	0.1	-8.3
	TC	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2.0	0.1	-2.7	10723.0	70.0	78.0	47.7	71.8	2.0	0.1	-1.3
	TC/RA	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR
Panel	ASHRAE STD	[illegible]	[illegible]	[illegible]	135.0	NA	NA	NA	- - -	10175.0	168.0	75.0	127.0	NA	NA	NA	- - -
	ASHRAE BAL	[illegible]	[illegible]	[illegible]	147.8	80.3	6.4	1.6	20.6	12082.0	168.0	75.0	139.6	84.9	11.4	8.6	18.7
	TC	[illegible]	[illegible]	[illegible]	141.2	77.8	5.9	1.5	-11.	10595.0	168.0	70.6	129.5	79.8	10.1	7.1	4.1
	TC/RA	[illegible]	[illegible]	[illegible]	[illegible]	77.5	4.5	0.4	17.2	9618.0	280.0	69.7	106.2	80.7	2.4	17.2	-5.5

Note: Person is Sedentary and seated

Room I (15'-0" 30'-0" 8'-0") - Position II x=15'-0" y=7'-6"

System	Design Method	Infiltration - 2.0 ACH								Infiltration - 4.0 ACH							
		Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%	Load	Area	T_a	Ti	MRT	ΔT_{cr} °F		Δ%
		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.		BtuH	Sq. Ft.	°F	°F	°F	Hor.	Ver.	
Convection	ASHRAE STD	[illegible]	[illegible]	[illegible]	49.3	NA	NA	NA	- - -	23709.0	70.0	75.0	49.3	NA	NA	NA	- - -
	ASHRAE BAL	[illegible]	[illegible]	[illegible]	46.3	66.8	9.1	4.4	5.8	22805.0	70.0	75.0	46.2	66.8	2.0	0.1	-3.8
	TC	[illegible]	[illegible]	[illegible]	47.7	62.0	0.1	2.4	3.8	24163.0	70.0	75.0	47.7	71.8	2.0	0.1	1.9
	TC/RA	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR	NR
Panel	ASHRAE STD	[illegible]	224.0	[illegible]	130.0	NA	NA	NA	- - -	22333.0	336.0	75.0	131.0	NA	NA	NA	- - -
	ASHRAE BAL	[illegible]	224.0	[illegible]	147.4	94.9	13.6	24.7	21.2	27521.0	336.0	75.0	159.6	111.4	2.0	37.7	23.2
	TC	[illegible]	224.0	[illegible]	124.1	83.9	11.3	21.0	-2.9	19492.0	336.0	61.4	128.4	89.7	1.3	30.2	-12.7
	TC/RA	[illegible]	392.0	[illegible]	107.0	83.9	11.6	18.5	-8.9	19350.0	392.0	61.6	122.2	89.5	1.5	26.6	-13.4

Note: Person is Sedentary and seated