Sir W^{M} Herschel Infrared Handbook

DRAFT

P/N F2199 Rev C 06/11

Table of Contents

List of Figures and Tables

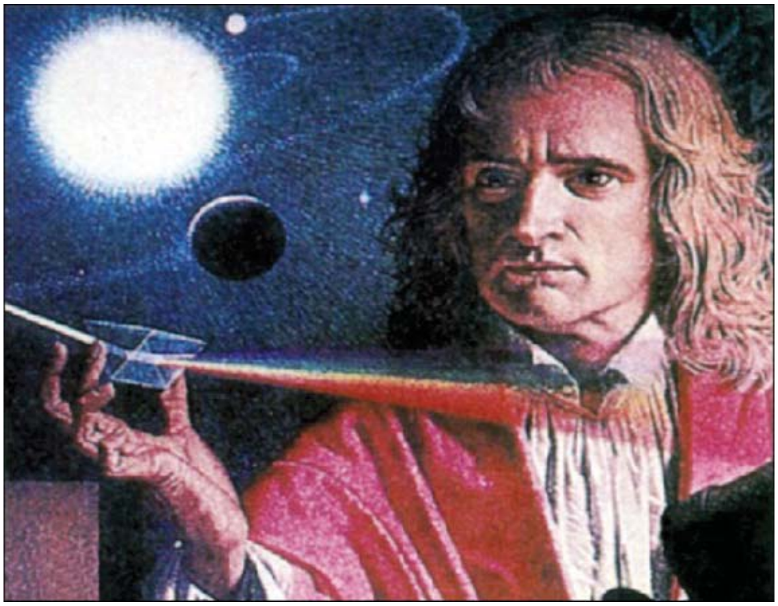
Figures

Tables

1.0 Introduction to Radiant Heating

1.1 Story of Sir William Herschel

An English astronomer discovered infrared in 1800. Using a thermometer to measure the heat of sunlight diffused through a prism, he discovered that, whereas blue light carried the least heat, the temperature on the thermometer rises as the color changes to the red spectrum.



It was actually beyond the red spectrum that the highest temperature was reached - *the infrared spectrum*.

Today's scientists know that it is through this range of infrared or "invisible" light waves, that we receive about 50% of the sun's energy.

In the late 1950's, Roberts-Gordon pioneered the concept of gas-fired infrared radiant tube heating which simulates the direct rays of the sun. This principle results in substantial fuel economy. Infrared radiant heat energy is directed downward to the occupants, the floor and building contents, warming them without warming the ceiling or the air. Air heating then occurs by convection from the floor and the objects warmed by the infrared. This gas-fired, infrared, radiant tube principle of heating is one of the most economical and the most comfortable heating methods available today.

1.2 What is Infrared?

Infrared is the transmission of energy by means of electromagnetic waves (rays). When rays strike an object they stimulate the molecules within the object, causing them to move rapidly and to generate heat. Infrared rays are invisible and travel at the speed of light in straight lines from the heat source to all surfaces and objects without heating the space (air), through which they pass. Energy in the rays is safely absorbed by cool surfaces (floors, equipment, people), and conduction carries some of this heat deeper into surfaces creating a reservoir of heat. The balance of the radiant energy is reflected from heated surfaces to be absorbed by other cooler surfaces. The temperature of air is raised by convection from heated surfaces.

Infrared heating equipment is most widely available today in two forms:

1. High-intensity equipment
2. Low-intensity equipment

High-intensity equipment, identified by an open flame and high temperature (1800^{o}F) ceramic surface, is what many engineers associate with the term “infrared heat”. This type of equipment is more suitable to a localized station heating or “spot” heating application and represents only a small portion of the infrared heating equipment available to the heating system designer. Low-intensity equipment, identified by a flame contained within a tube or network of tubes at a reduced temperature (maximum 900^{o} - 1000^{o} F), is recognized as being an efficient means of heating an entire space. Heating a continuous span rather than a series of intermittent spots provides a level of comfort surpassed only by the sun.

1.3 The Electromagnetic Spectrum

The electromagnetic spectrum differentiates all known types of electromagnetic wave energy via their wavelength as measured in microns. The shortest wavelength energy (10^{-8} microns) known is the cosmic ray, while the longest wavelength energy (5×10^{8} microns) known is the broadcast radio wave.

Visible light falls between these two extremes having wavelengths between 0.4 and 0.7 microns. Infrared energy waves are slightly longer than visible light, having wavelengths from 0.7 to 400 microns. However, the majority of heat producing radiant energy falls within a much narrower wavelength range of 2 to 12 microns. It is important to consider the heat energy wavelength for the following reasons.

a) The wavelength is directly related to the emitter source temperature, with higher temperature sources generally producing a majority of the energy at shorter wavelengths.
b) The energy transfer to the solid body receiver can be affected by wavelength. Many materials more completely utilize energy provided at longer wavelength (for example: concrete and water).

Page 4, Figure 1 illustrates the electromagnetic spectrum as it is known today.

Page 4, Figure 2 illustrates the infrared spectrum relative to common types of infrared heating equipment available today.

1.4 Utility of Infrared

Radiant heating equipment has demonstrated its utilitarian nature in many diverse applications. Thousands of cost effective installations exist in small automobile shops, as well as in large, high bay structures. To better understand these results, it is necessary to review the manner in which a radiant heating appliance warms a space.

Infrared energy from a radiant appliance heats objects, people and surfaces, not the air. Certain elemental constituents in the air (such as water vapor and carbon dioxide) do absorb radiant energy; however, the amount of these substances is typically so small, that the heating effect on the air is negligible. The warm objects and floor convert this energy to heat which:

- Is absorbed into the objects and floor creating a heat reservoir.
- Warms the air near the objects and floor via convection.
- Is reradiated to occupants and other surfaces of the space.

The radiant energy received by the occupants, directly from the heater or indirectly from the heater via reradiation by the floor and objects, serves to increase the mean radiant temperature (MRT) of the occupant. In a manner similar to direct sunlight, the increased MRT allows the occupant to perceive a comfort condition at a much reduced air temperature (sometimes as much as 7^{O} - 10^{O} F lower). The resulting reduced air temperature within the space provides the following advantages.

- Reduced stratification of air within the space.
- Reduced actual transmission heat loss due to lower temperature inside than assumed design condition as well as substantially lower ceiling and upper sidewall temperature due to reduced stratification (25° - 30° F lower is not usual).
- Reduced air change heat loss, to the extent that exfiltration through cracks or openings, near the roof, will be decreased due to decreased stack effect.

Each of the above advantages impacts favorably on fuel usage.

Figure 1

Electromagnetic Spectrum - Wavelength (microns)

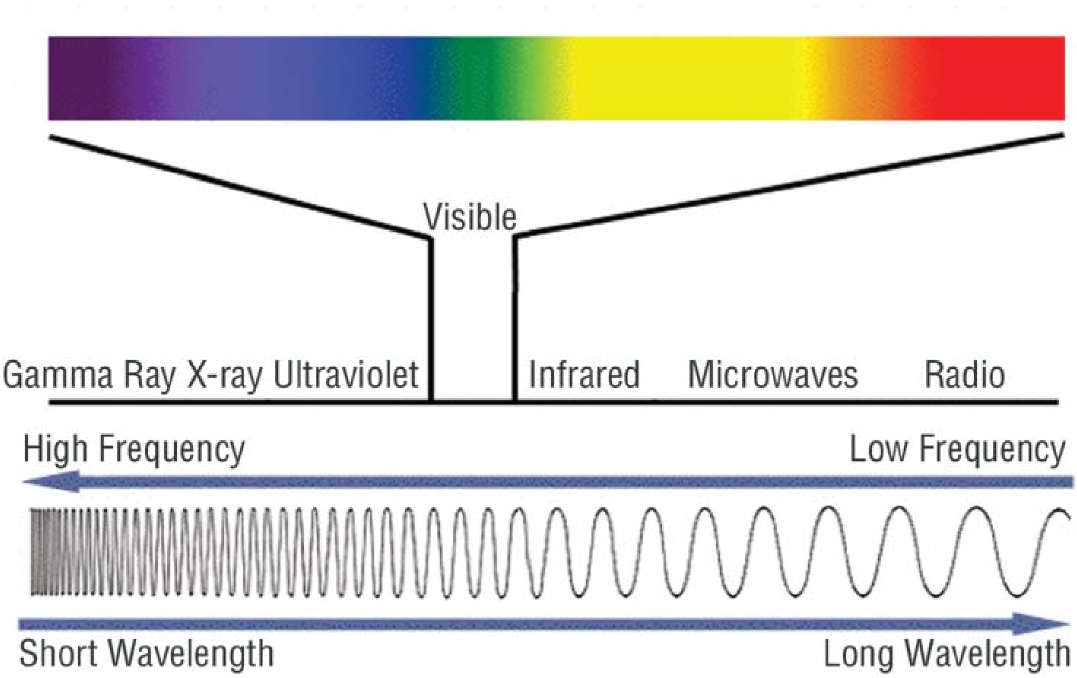
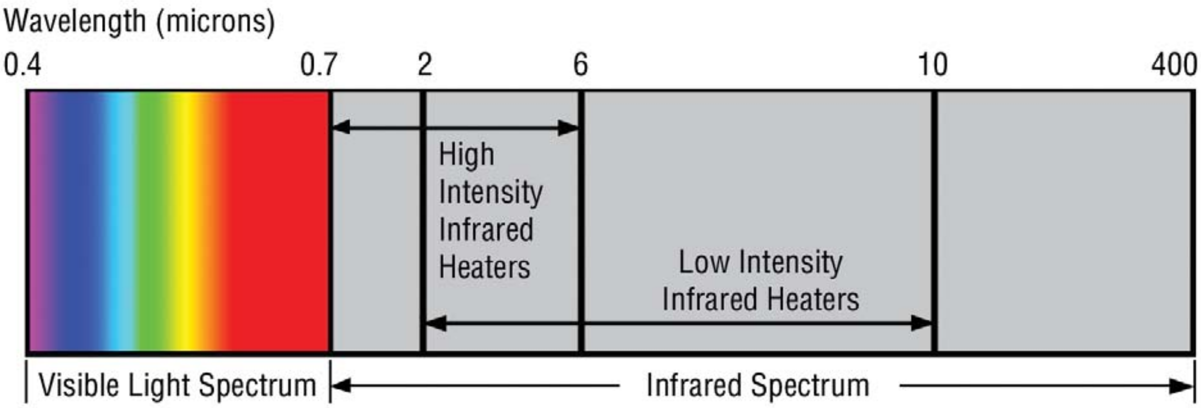


Figure 2

Infrared Spectrum - Wavelength (microns)



1.5 Method of Radiant Heating Appliances

All radiant heating appliances are not the same. Various material properties and performance criteria can be used to evaluate a radiant heating appliance relative to its major function, namely:

- Provide usable radiant energy to the space in sufficient quantity to provide comfort for the occupants.

Page 7, Figure 3 is a visual representation of the factors that effect the performance of a radiant appliance. These factors are reviewed briefly below.

Natural gas(NG) or Propane(LP) fuel contains an inherent chemical heating value (approximately 1000 BTU/cubic foot for natural gas(NG) and 2500 BTU/cubic foot for Propane(LP) gas). Of this total heating value available, only a percentage is available to the radiant heating appliance, the remainder being stack loss.

This percentage is known as the thermal efficiency and is described as follows:

$$\textbf{Thermal Efficiency} = \frac{\textbf{Total Input Energy - Stack Loss}}{\textbf{Total Input Energy}}$$

The tube is heated by the available energy from the fuel gas. A tube material property, called emissivity, helps determine the amount of energy that leaves the tube as radiant energy. The heat energy of the tube is dissipated by one of the following mechanisms:

1. A portion of the energy is released as radiant energy directly to the space.
2. A portion of the energy is released as radiant energy and is reflected by the fixture to the space.
3. A portion of the energy from the tube is convected to the space.
4. A portion of the energy is released to the fixture and “bounced back” into the tube.

The fixture efficiency is a measurement of the ability of the heating appliance to release radiant energy to the space. Note that the relationships of items 1 through 4 above can greatly influence the fixture efficiency. Equally influential to the fixture efficiency is the reflector material and the reflector shape. The property of the reflector material, known as reflectivity, and the overall configuration of the reflector determine the amount of usable radiant energy delivered to the space.

The pattern efficiency of a radiant heating appliance is a measurement of the ability of a radiant heating fixture to deliver energy into a usable, specific distribution pattern in the space. It is this distribution pattern, together with a material property of people or objects in the space known as absorptivity, that determines how much of the radiant energy released by the heating appliance is utilized by the space to provide comfort to the occupants.

For comparison purposes, Page 8, Figure 4 provides a visual representation of the methodology for a conventional, air heating appliance.

Figure 3

Energy Flow Chart - **Radiant Heating**

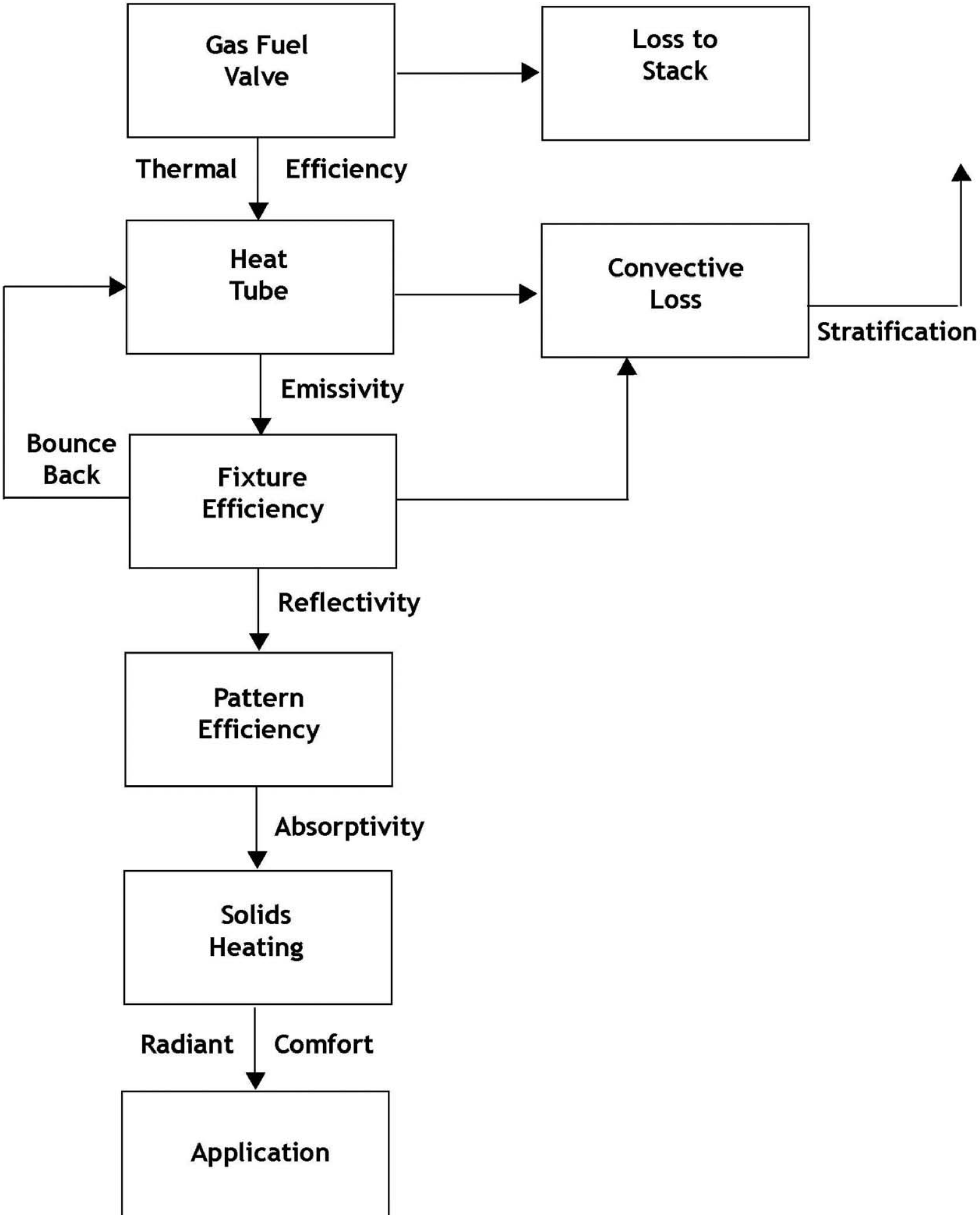
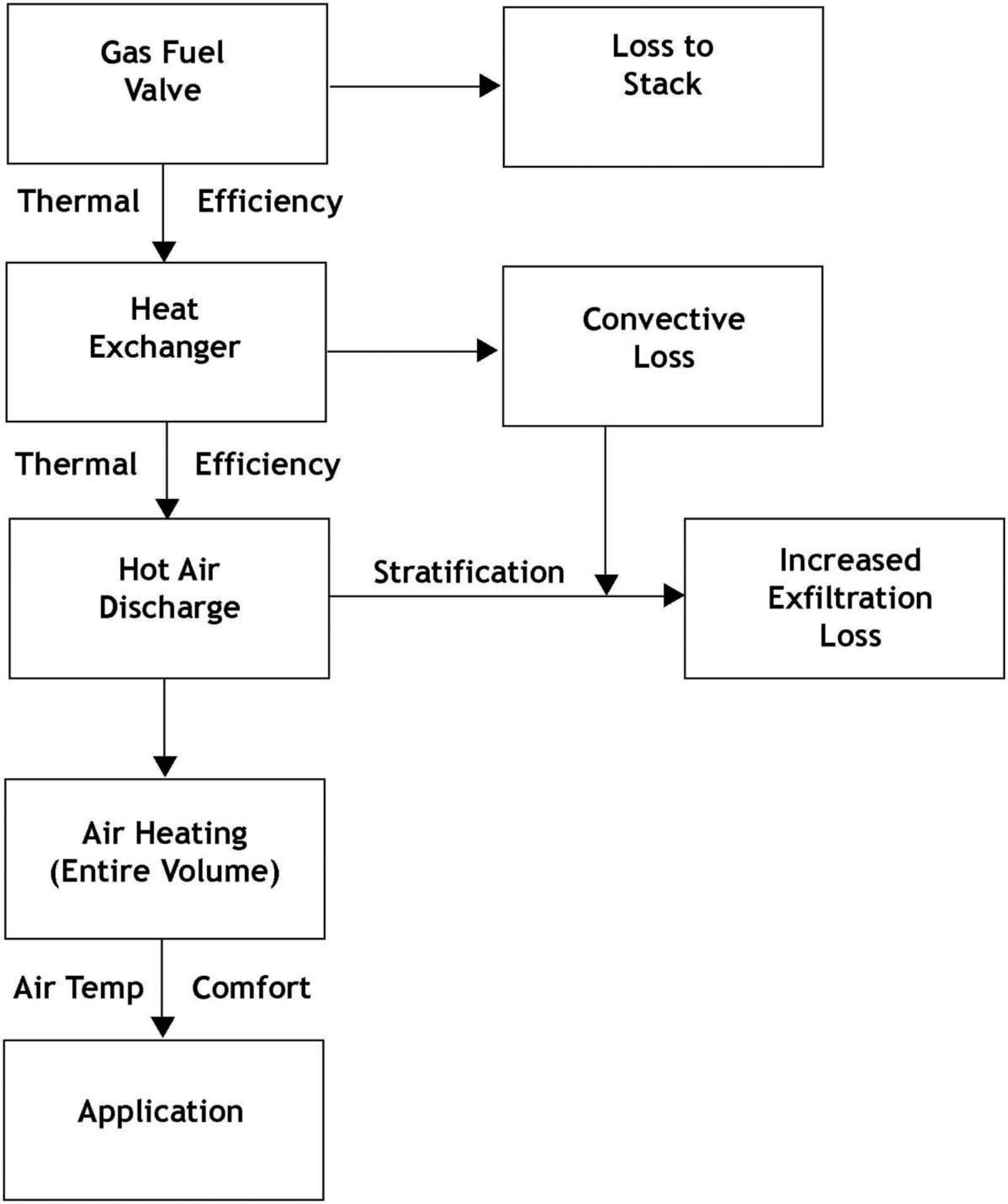


Figure 4

Energy Flow Chart - **Conventional Air Heating**



2.0 Types of Radiant Heating Equipment

2.1 Ashrae Defined Types

All radiant heating appliances are not the same. A recognized method of classifying radiant heating systems is according to the operating temperature of the emitting surface. For example:

1. High and Medium Intensity: above - 1500° F
2. Low Intensity: 500° F - 1500° F
3. Low Temperature: 120° F - 350° F

High and medium-intensity heaters usually take the form of open flame, non-vented appliances with incandescent ceramic faces.

Low intensity units are designed to operate below incandescent temperatures and frequently use steel tube or pipe as the emitter.

Low temperature radiant heating systems utilize large heated surfaces such as floors, walls, panels, or ceilings. The surface temperature is elevated by hot water piping or electrical resistance wire embedded in the surface.

ASHRAE recognizes three specific types of infrared heaters that are gas fired, See Page 10, Figure 5.

TYPE 1: Indirect Fired Units

Type 1 is characterized by burning a gas-air mixture inside a tube or enclosure, which radiates it’s energy to the space. The products of combustion are generally vented to the outside. Typical operating surface temperatures do not exceed 1200° F.

Type 1(a) units utilize an atmospheric burner venting products of combustion upward (for example: patio heater).

Type 1(b) units utilize multiple vacuum assisted burners operating in a horizontal tube.

Type 1(c) units utilize a power assisted (forced draft) burner operating in a horizontal tube.

Type 2: Direct Fired Units

Type 2 is characterized by burning the gas-air mixture in a porous matrix of refractory material, which radiates its energy into the space. The products of combustion are vented into the space. Temperatures of operating Type 2 units range from 1600° F to 1800° F.

Type 3: Catalytic

Type 3 is characterized by mixing gas and air in the presence of a catalyst. The mixture oxidizes without flame, and heat radiates into the space. The products of combustion are vented into the space. Temperature of these catalytic units range from 650° F to 700° F.

2.2 Market Defined Types

The ASHRAE defined radiant heating appliance types do not adequately reflect the recent evolution of new products as available in the market place.

The result is that the industry has moved beyond these definitions by introducing new appliances that fall into more then one of these categories. In addition, ASHRAE has not yet developed a way for the engineering community to distinguish between appliance performances within a category or between categories.

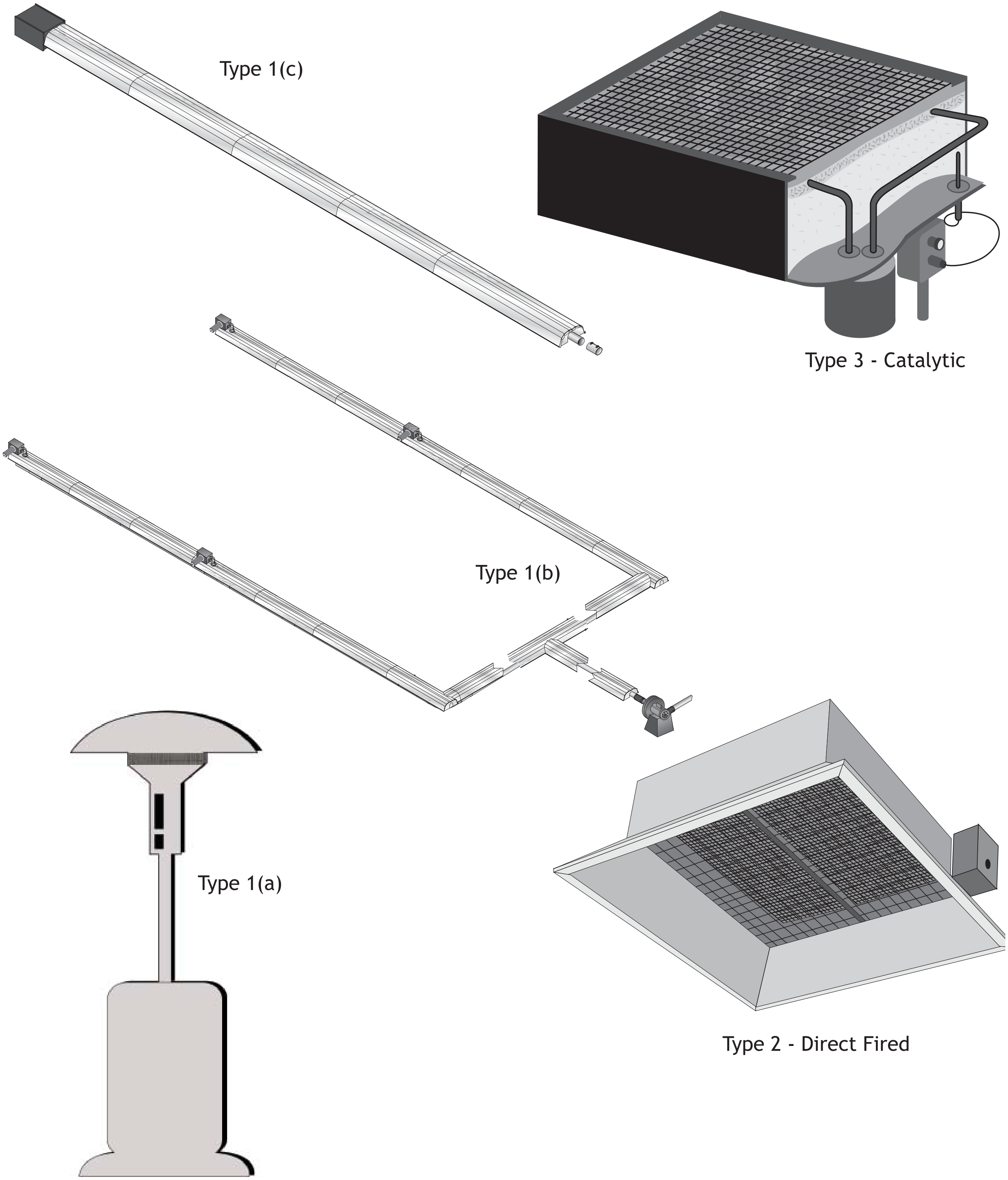
The most visible demonstration of this definition inadequacy exists within the increasingly popular ASHRAE type 1(b) and type 1(c) appliance market. Recently introduced radiant heating appliances have many of the characteristics of a type 1(b) system, but not all of them. Specifically, lower efficiency systems do not provide for condensation of the combustion gases before exhaust.

Additionally, many manufacturers combine multiple, individual, non-condensing burner appliances on a common exhauster and represent the resulting system as a type 1(b) condensing appliance. The lower efficiency of these systems more accurately reflects the performance characteristics inherent in an ASHRAE type 1(c) appliance.

In order to differentiate these appliances, the market has defined a radiant heating appliance category in between an ASHRAE type 1(b) and type 1(c). Burners in this category are referred to as Quasi type 1(b)/1(c) appliances or multi-burner.

Figure 5

ASHRAE Heater Types



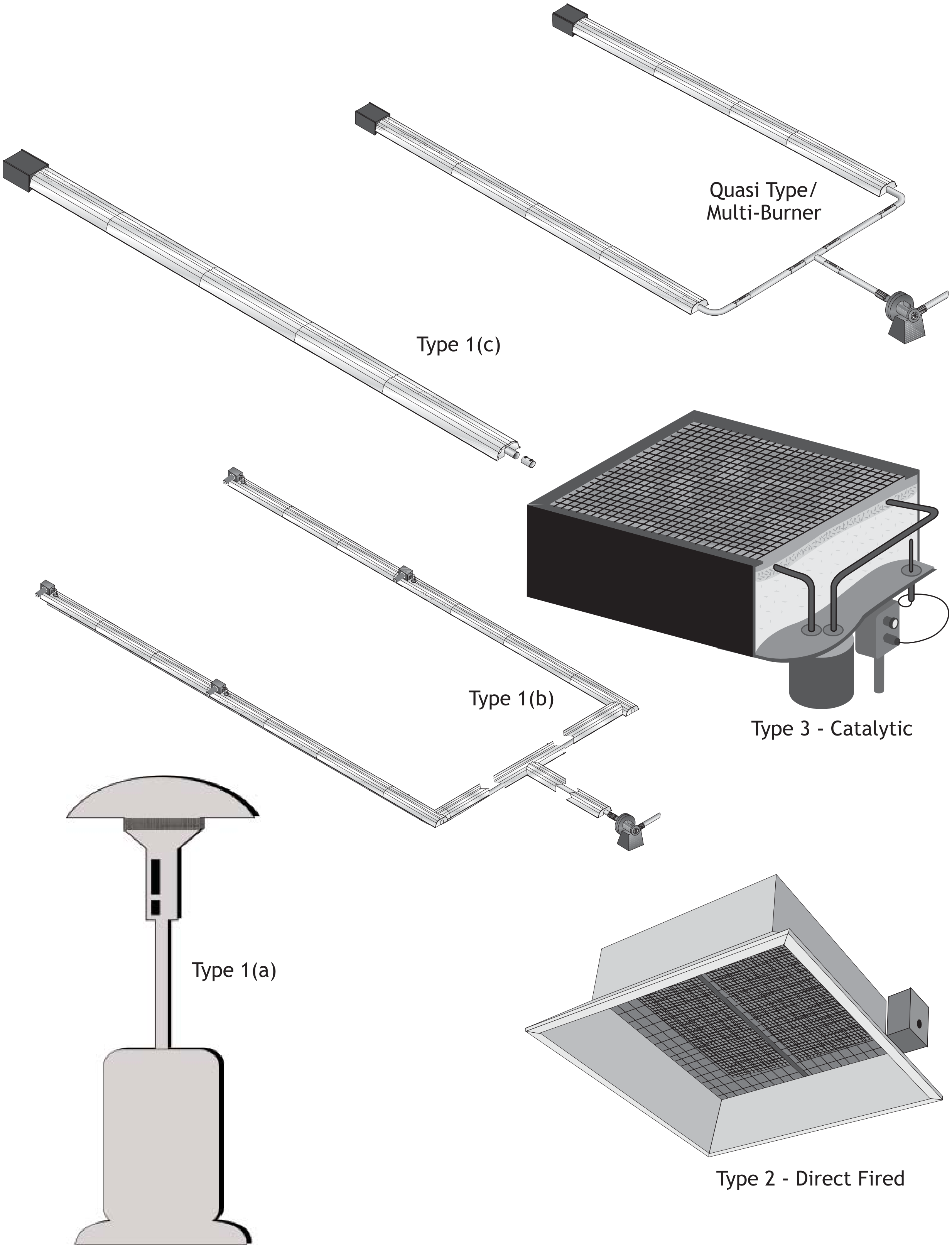
Market history, as well, dictates differentiating approaches to the implementation of type 1(b), type1(c), and Quasi type 1(b)/1(c) appliances.

Variation in market approach can be recognized as follows:

Description	Appliance Type
Engineered, custom designed, multiple burner, condensing appliances.	Type 1(b)
Engineered, custom designed, multiple burner, non-condensing appliances.	Quasi type 1(b)/1(c)
Factory assembled, single burner, non-condensing appliance.	Type 1(c)
Site assembled, single burner, non-condensing appliances.	Quasi type 1(b)/1(c) or Type 1(c)
Factory assembled, open flame.	Type 2
Factory assembled, catalytic combustion.	Type 3

Figure 6

Market Defined Heater Types



3.0 Concepts of Radiant Heating

All radiant heating appliances have several component design and material factors affecting their performance and utility. The factors include:

- Thermal Efficiency
- Emissivity
- Reflectivity
- Convection Losses
- Fixture Efficiency
- Pattern Efficiency
- Absorptivity
- Radiant Efficiency

Understanding the importance of these factors, relative to the radiant heater performance, can insure the suitability of a radiant heating appliance to a particular application.

While high thermal efficiency (for example) is desirable, it is not the only factor contributing to a successful radiant heating application. Following are additional areas that must be considered when determining the most suitable radiant heating appliance for a particular application.

- The burner efficiency and safety;
- The tube emissivity, temperature and total area;
- The reflector angularity, inherent ability to trap hot air near the tube and material reflectivity;
- The absorptivity of objects or occupants within the space.

Each of these items influence the ability of a radiant heating appliance to provide comfort with a space. Deficiencies in any of these areas can result in a poorly heated space or excessive fuel usage for a given comfort condition.

Following are in depth discussions of each of these important radiant heating concepts, See Page 15, Figure 7.

3.1 Thermal Efficiency

Thermal efficiency of a radiant heating appliance is defined as ratio of available energy output (at the point of use) to total energy input, See Page 16, Figure 8. In practice, the thermal efficiency can be determined by establishing stack loss and utilizing the following formula:

$$\textbf{Thermal Efficiency} = \frac{\textbf{Total Input Energy - Stack Loss}}{\textbf{Total Input Energy}}$$

Stack Loss can be determined by measuring the following:

a) Carbon Dioxide ($C0_2$) in flue gases (in % by volume)

b) Net flue gas temperature (defined as flue gas temperature - room temperature)

c) Latent heat recovered from water vapor in the flue gas when cooled to a temperature below dew point.

For most field test situations, it is impractical to determine c) above, and this contribution to stack loss is typically ignored.

Once the CO_2 % and net flue gas temperature are determined, a nomograph may be used to determine the stack loss, hence the thermal efficiency, See Page 17 Figure 9.

High thermal efficiency is desirable, but because of the formation of condensate within a high efficiency radiant heating appliance, certain equipment design features are required to insure satisfactory longevity. ANSI Z83.6a -1989 (test standard for gas-fired infrared heaters) requires that suitable means be provided to prevent corrosion and collect condensate formed in high efficiency radiant heating appliances.

The argument to differentiate “wet” (condensing) or “dry” (non-condensing) radiant systems has centered around the thermal efficiency that will define condensing or non-condensing operating conditions. Although ALL gas appliances produce some level of condensate at start-up, an appliance is not considered condensing unless it continually provides operational thermal efficiencies above 83%. Although this thermal efficiency does not provide a continuously condensing environment, it does produce sufficient condensate to cause corrosion in the appliance severe enough to require that special design considerations to be employed to provide adequate operational life. The 83% thermal efficiency threshold is well documented in corrosion testing completed by a number of gas appliance industry leading organizations including:

- Lennox
- Battelle Columbus Laboratories
- Gas Appliance Technology Center (GATC at Gas Research Institute (GRI)
- American Gas Association Laboratories (A.G.A.L.)

Figure 7

Concepts of Radiant Heating

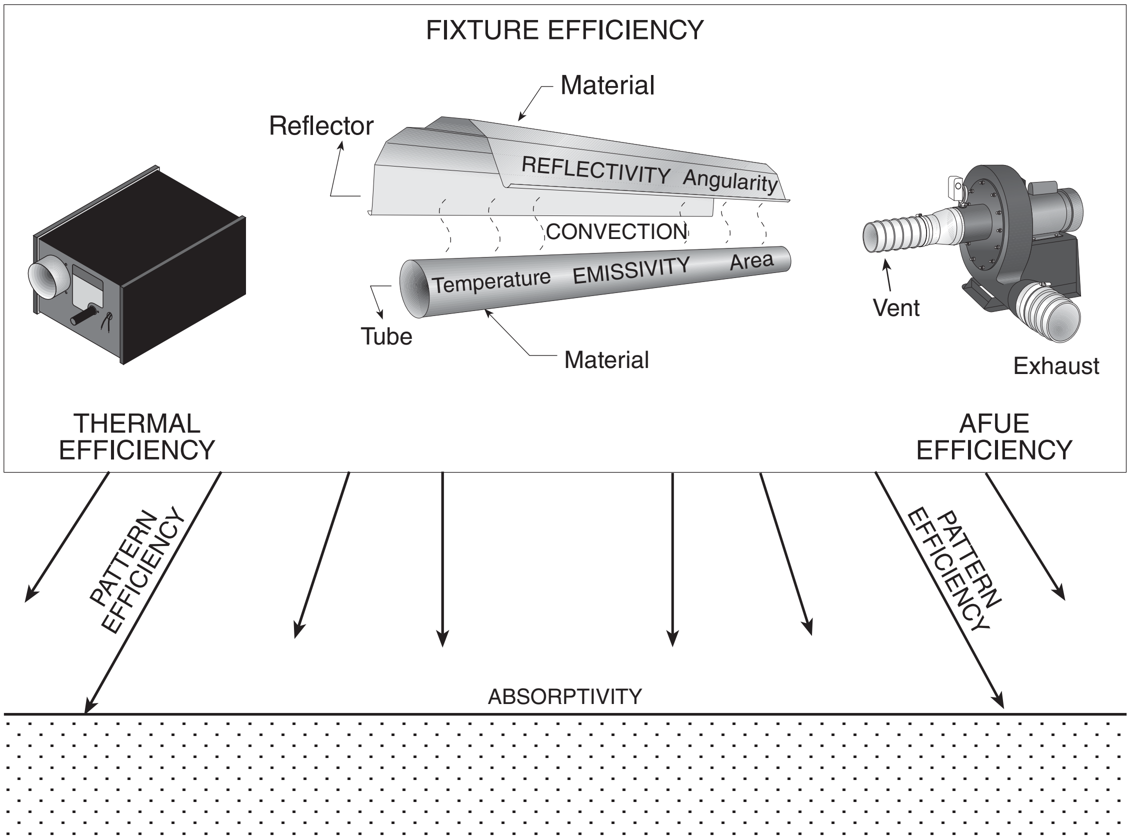
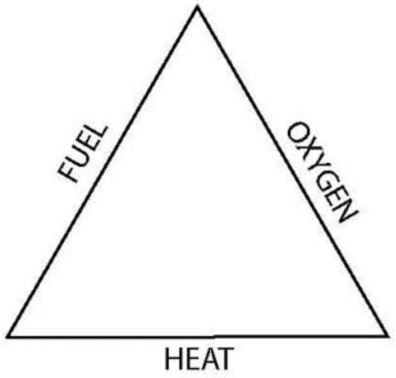
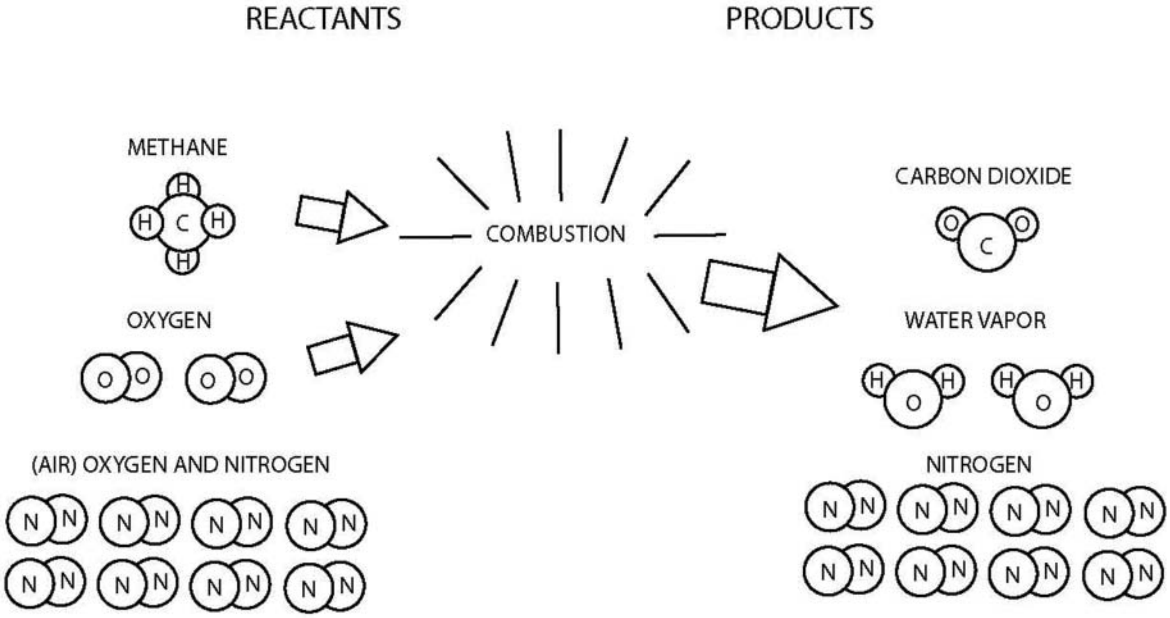


Figure 8

Combustion of Methane and Air



Requirements for Combustion



Excess air in terms of the ideal air needed for complete combustion, can be calculated for most common fuel gases as follows:

$$\text{Excess air \%} = \frac{\text{Ultimate } CO_2 - \text{Observed } CO_2}{\text{Observed } CO_2} \text{ X } 91$$

Where 91 is a constant for most fuel gases.

Ultimate CO_2 for natural gas = 11.95%

propane gas = 13.78%

(Reference: Fundamentals of Combustion A.G.A. Laboratories, 1973)

Figure 9

Flue Loss Calculations

On Page 24 and 25, Figures 10 and 11 are nomograph for use in calculating the flue loss (percent) of a hater operating on normally distributed gases whose characteristics falls within the following ranges:

	Natural Gas	Propane HD-5
Heater Value (gross), BTU per cu ft	970 - 1100	2466 - 2542
Specific gravity	0.57 - 0.70	1.522 - 1.574
Ultimate carbon dioxide (CO_2), %	11.7 -12.2	13.73 - 13.82

In lieu of these nomograph or for a heater operating on normally distributed gases any of whose characteristics fall outside of the range specified above, the flue loss (percent) may be calculated from the following formula:

$$L_f = \frac{1}{379}\left\{\begin{array}{l} \left(\frac{PU}{1000}\right)\left[16.2(T_f - T_r) + 6.53\text{x}10^3 \ \ln\frac{T_r}{T_r} + 1.41\text{x}10^6 \ (T_{r\text{-}1} - T_{f\text{-}1})\right] \\ +\left(\frac{P}{10}\right)\left(1-\frac{U}{100}\right)\left[9.47(T_f - T_r) + 3.47\text{x}10^3 \ \ln\frac{T_r}{T_r} + 1.16\text{x}10^6 \ (T_r - 1 - T_f - 1)\right] \\ +\left(\frac{P}{10}\right)\left(\frac{U-C}{C}\right)\left[9.46(T_f - T_r) + 3.29\text{x}10^3 \ \ln\frac{T_r}{T_r} + 1.07\text{x}10^6 \ (T_r - 1 - T_f - 1)\right] \\ +\left\{\left(\frac{T-P}{10}\right) = 0.00174hA\left[1-\frac{P}{A}\left(\frac{U-C}{C}\right)\right]\right\}\left[19.86(T_f - T_f) + 1194\left(\sqrt{T_r} - \sqrt{T_f}\right) + 7.50\text{x}10^3 \ \ln\frac{T_f}{T_r}\right] \end{array}\right\}$$

$$+ \quad 5.04\ (T - P)$$

Where:

A = Air required for complete combustion, SCF per 1000 BTU of gas burned

C = Carbon Dioxide (CO_2) in flue gases, percent of total dry constituents in the flue gases.

h = Relative humidity of air supplied for combustion, percent 100.

L_f = Flue loss, percent of heat input rate.

P = Dry constituents in flue gases from stoichiometric combustion SCF per 1000 BTU of gas burned.

T = Total constituents in flue gases from stoichiometric combustion SCF per 1000 BTU of gas burned.

Tf = Flue gas temperature, degree R.

Tr = Room temperature, degree R.

U = Ultimate carbon dioxide (CO_2) of fuel gas percent.

REFERENCE: Z83.20 - 2001

An understanding of the corrosive nature of condensate and its relationship to thermal efficiency is essential to insure that radiant heating applications perform in the best interest of the consumer.

3.1.3 Marketplace Report - Thermal Efficiency

Below is a summary of the range of thermal efficiencies generally experienced within the industry. These are categorized by appliance type.

Appliance Type	Thermal Efficiency Rate
Type 1 (a)	70 - 75 %
Type 1 (b)	83 - 90 + %
Type 1 (c)	70 - 82 %
Quasi Type 1(b)/1(c)	70 - 82 %
Type 2	75 - 90 + %
Type 3	75 - 80 %

3.2 Emissivity

Emissivity is a material property commonly discussed with radiant heating appliances. It is a ratio of the amount of infrared energy released by a material as compared to the amount that would be released by a black body surface at identical temperature and conditions. A black body surface is a theoretical concept for an emitter having the highest possible emissivity of 1.0.

The emissivity of a material is dependent on a number of factors.

- Temperature of the material: for many materials, variations in temperature produce significant changes in the emissivity.
- Surface condition of the material: the emissivity characteristics of a material are determined mainly by the surface layers of material. For this reason, coatings can be applied to base materials to improve emissivity.
- Wavelength of emitted energy: for most materials, variations in emissivity can be expected to be dependent on the wavelengths at which the measurement was made.

It is important to note that high emissivity does not automatically provide high radiant energy output for a radiant heating appliance. Emissivity is an indicator of the potential a radiant tube has to release energy. If, for example, the reflector on a radiant heating appliance can not direct energy released by the tube away from the appliance, high emissivity serves no purpose. Additionally, this “bounced back” energy into the tube can increase tube temperature, affecting emissivity, and creating the impression of increased heat release. Under these conditions this energy release is typically overstated (when calculated).

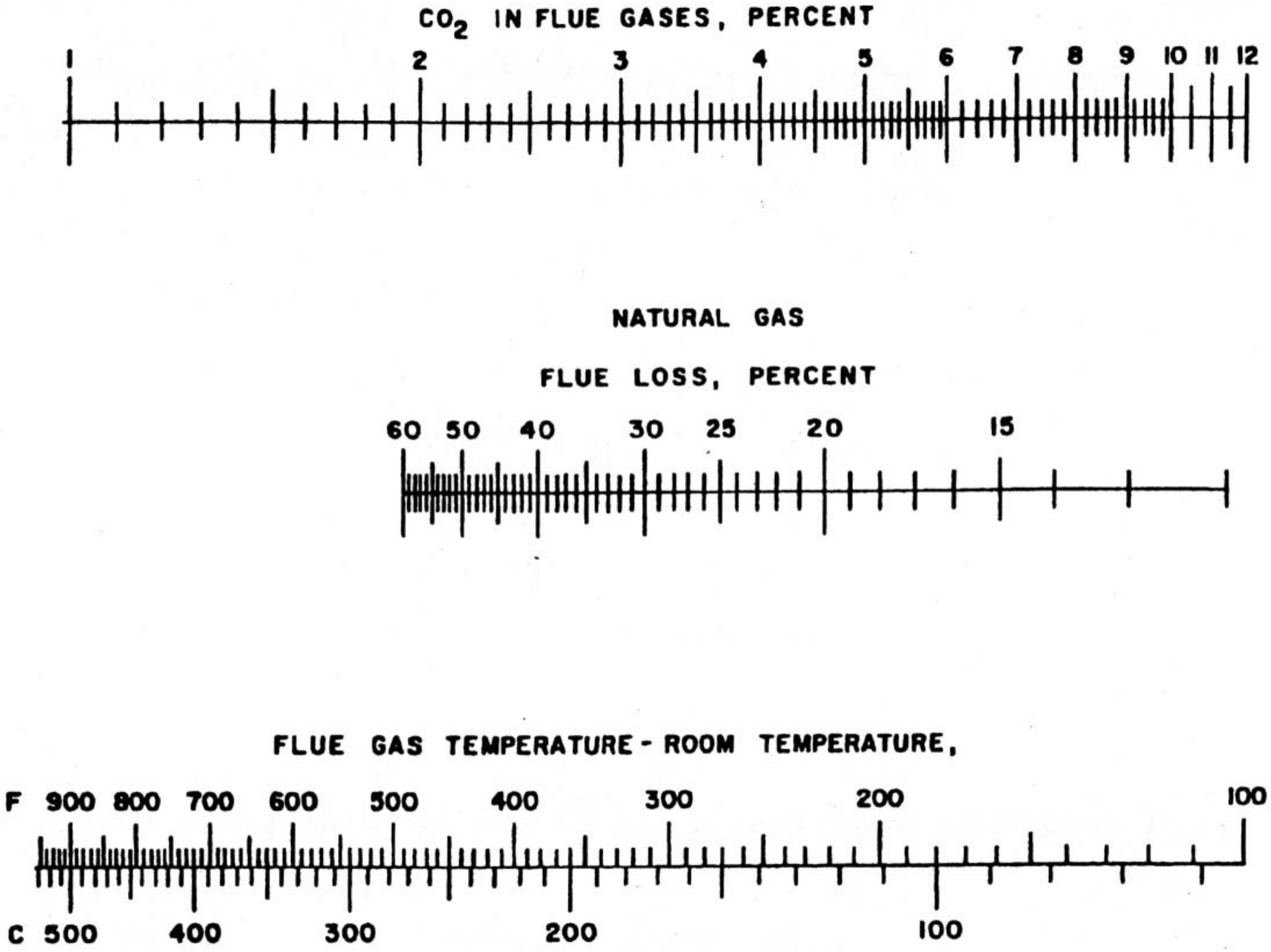
A more meaningful measurement of the quantity of radiant energy released by a radiant appliance is known as emittance.

Emittance is defined as the total energy released in radiant form over all wavelengths per unit area of surface,

Figure 10

Nomograph for Determining Flue Loss With Natural Gas

(See text of exhibits A for usage limitations)

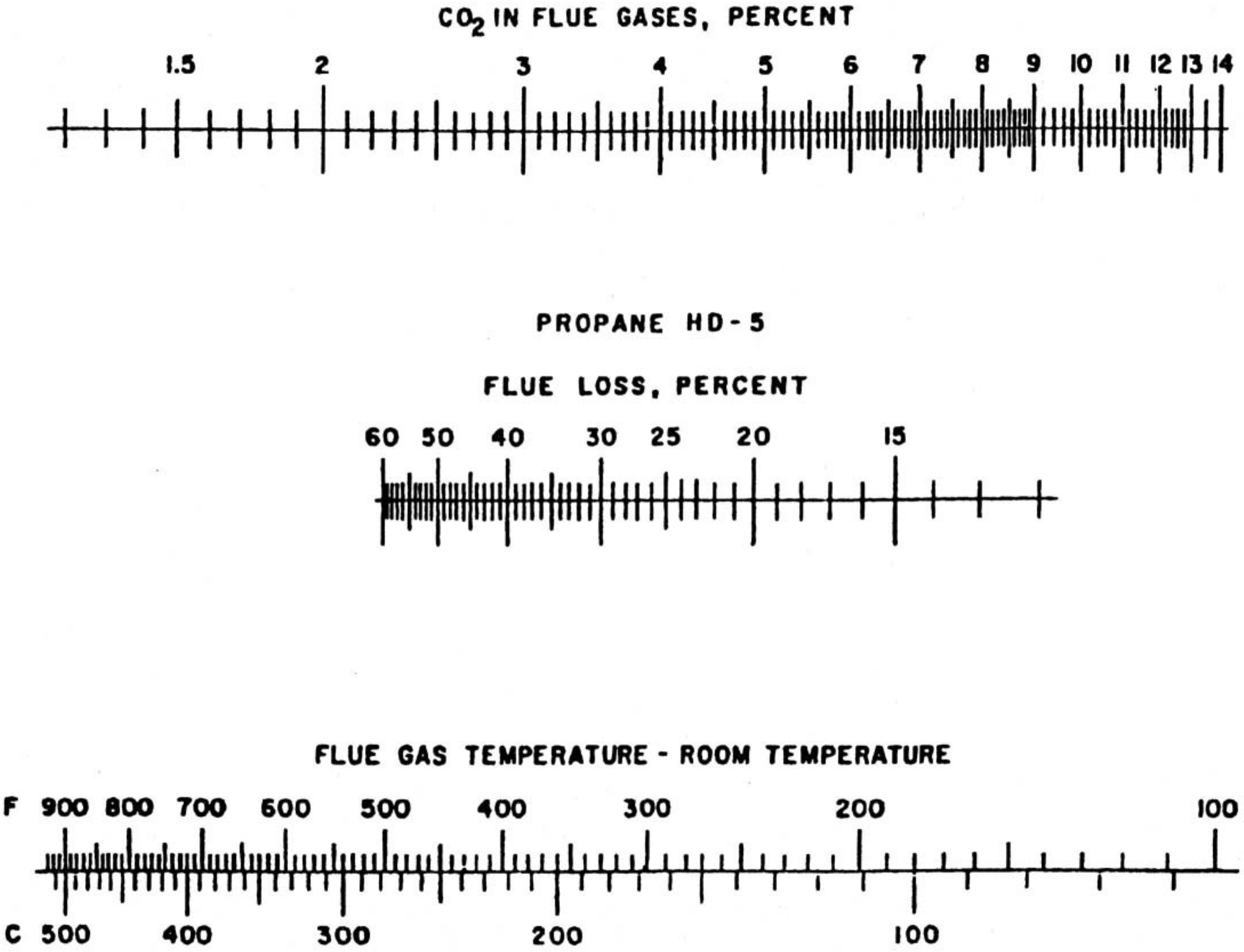


Reference ANSI Z83.20 - 2001

Figure 11

Nomograph for Determining Flue Loss With Propane HD-5

(See text of exhibits A for usage limitations)



Reference ANSI Z83.20 - 2001

and is determined by four factors.

- Temperature of the emitting surface.
- Temperature of the surrounding surfaces.
- Emissivity of the emitting surface.
- Emissivity of the surrounding surfaces.

By determining the emittance of each unit area of a radiant heating appliance an estimate could theoretically be made of the total radiant energy output from a radiant heating appliance using the Stefan - Boltzmann law. Note that this would require accurate surface temperature, emissivity and area calculations for each unit area on a radiant appliance, as well as similar information on each unit area of the surrounding surface. While this may be possible in theory, the complex, dynamic nature of heat transfer phenomenon renders this approach impractical.

It is essential to recognize that radiant heating involves an EXCHANGE of energy between surfaces (for example, an occupant and the floor). Understanding proper application of radiant heating equipment requires a realization of the importance of this radiant exchange.

Page 29 and 30, Figure 13 provides a listing of theoretical emissive power (radiant output) of a surface at temperature shown for various emissivities. This Table is based on the Stefan - Boltzmann law as follows:

$W = \varepsilon \sigma T^4$

Where: T = temperature

W = emissive power units

ε = emissivity of the material units

σ = Stefan - Boltzmann constant = 0.1714×10^{-8}

Page 28, Figure 12 indicates the distribution of the emissive power as a function of wavelength for various temperatures.

3.2.1 Market Report - Emissivity

Below is a summary of materials and emissivities generally available in the industry.

MATERIAL	EMISSIVITY	(wavelength) @ TEMPERATURE	REFERENCE
- Plain Steel	0.79 to 0.81	1000° F (3.6)	1,2,3
- Aluminized Steel (type 1)	0.20 to 0.50	1000° F (3.6)	3
- Aluminized Steel, heat treated (type 1)	0.80	1000° F (3.6)	3,4
- Fixed Ground coated steel	0.80	1000° F (3.6)	
- Porcelainized Steel	0.92 to 0.96	100° F (9.3)	3,5
- Cast Iron	0.95	1000° F (3.6)	2
- Stainless Steel (type 304)	0.44 to 0.62	1000° F (3.6)	2
- Galvanized Steel	0.28	100° F (9.3)	2,4
- Calorized Steel	0.57	1000° F (3.6)	1,2
- Aluminum	0.02 to 0.05	100° F (9.3)	2,4
- Stainless Steel (type 430 polished)	0.10 to 0.20	100° F (9.2)	3,4
- Pyromark® Paint (Silicon resin - properly cured and vitrified)	0.80	1000° F (3.6)	6

References

1. A.G.A. Research Bulletin #83
2. Radiative Transfer by Hottel and Sarofin
3. Technical Letters, Inland Steel Corp.
4. Critical Table, National Bureau of Standards
5. Mark's Std. Handbook for ME, 8th Edition
6. Literature from Manufacturer

A number of available radiant heating appliances utilize a silicone base high temperature paint (for example Pyromark® manufactured by Tempil Division of Big Three Industries). These coatings are said to provide emissivities of 0.95. This value can only be achieved through proper air drying, oven curing and vitrification of the coating to inorganic silica, unlikely within the radiant heating industry. Even under these careful curing conditions, this 0.95 emissivity value cannot be achieved for surface temperatures much below 1800° F. This same, carefully cured surface has an emissivity approximately equal to that of hot rolled steel at the same temperature (e = 0.8). Page 31 and 32, Figure 14 provides information on emissivity of high silicon resin paint from the manufacturer.

% of Total Radiant Output in Various Wavelength Bands for Black Body Radiation*

Figure 12

WAVELENGTH BAND MICRONS	TEMP° F																			
	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	3000	4000	5000
.45 - .7	-	-	-	-	-	-	-	-	-	*	*	*	*	*	*	*	*	*	3.58	9.02
.7 - 1.0	-	-	-	-	*	*	*	*	*	*	*	*	*	*	*	*	*	5.62	13.3	20.1
1.0 - 1.5	*	*	*	*	*	*	*	*	*	1.37	1.88	2.5	3.33	4.23	5.3	6.43	7.62	19.9	27.5	29.2
1.5 - 2.0	*	*	*	*	*	1.54	2.38	3.17	4.25	5.4	6.85	8.3	9.5	10.8	12.0	13.3	14.6	20.8	20.0	16.6
2 - 3	1.075	2.22	3.78	5.95	8.3	10.8	13.4	16.1	18.5	20.7	22.6	24.1	25.8	26.9	27.8	28.5	28.8	26.2	19.5	14.3
3 - 4	4.9	7.7	10.5	12.9	15.1	17.0	18.7	19.9	20.6	20.8	21.1	21.0	20.5	20.2	19.8	19.1	18.6	12.0	7.8	5.2
4 - 5	9.0	11.5	13.0	14.8	15.7	16.2	16.3	15.8	15.5	15.2	14.5	14.0	13.5	12.6	11.9	11.3	10.6	5.9	3.5	2.1
5 - 6	11.0	12.3	13.5	13.4	13.3	13.0	12.3	11.8	11.3	10.5	9.7	9.0	8.3	7.9	7.3	6.7	6.2	3.2	1.8	1.1
6 - 7	11.0	11.3	11.0	10.5	10.2	9.5	9.0	8.2	7.3	6.9	6.3	5.9	5.3	4.9	4.5	4.2	3.8	1.9	1.0	*
7 - 8	10.5	9.7	9.0	8.8	8.0	7.2	6.4	6.0	5.5	4.8	4.5	3.9	3.6	3.3	2.9	2.7	2.6	1.1	*	
8 - 9	8.5	8.0	7.2	6.7	5.8	5.4	4.7	4.1	3.7	3.3	2.9	2.7	2.4	2.2	2.1	1.9	1.6	*	*	
9 - 10	7.0	6.6	5.9	4.8	4.6	3.9	3.5	3.1	2.7	2.4	2.2	1.9	1.8	1.6	1.5	1.2	1.1	*		
10 - 11	6.0	5.2	4.2	4.0	3.3	3.0	2.6	2.3	2.0	1.8	1.6	1.5	1.3	1.1	*	*	*	*		
11 - 12	5.0	4.0	3.4	3.0	2.6	2.2	1.9	1.7	1.5	1.4	1.1	1.0	*	*	*	*	*			
12 - 13	3.7	3.5	2.9	2.5	2.1	1.7	1.6	1.3	1.2	1.0	*	*	*	*	*	*	*			
13 - 14	3.4	2.8	2.3	2.0	1.5	1.4	1.2	1.1	*	*	*	*	*	*	*	*	*			
14 - 15	2.7	2.2	1.9	1.5	1.4	1.2	1.0	*	*	*	*	*	*	*	*					
15 - 20	8.2	6.8	5.5	4.6	3.8	3.1	2.6	2.3	1.9	1.6	1.4	1.2	1.1	1.0	*					
20 +	8.0	6.1	5.6	4.1	3.4	2.7	2.2	1.8	1.7	1.5	1.2	1.1	*	*	*	*	*			

*** Any Figure Less Than 1% Has Been Omitted**

Emissive Power (W) at Varying Temperatures and Emissivities
BTUH/FT^2

Figure 13

	EMISSIVITIES									
°F	**1.0**	**0.9**	**0.8**	**0.7**	**0.6**	**0.5**	**0.4**	**0.3**	**0.2**	**0.1**
100	169	152	135	118	101	84	68	51	34	17
150	238	214	190	167	143	119	95	71	48	24
200	326	293	261	228	196	163	130	98	65	33
250	437	393	350	306	262	218	175	131	87	44
300	574	517	459	402	344	287	230	172	115	57
350	740	666	592	518	444	370	296	222	148	74
400	941	847	753	659	565	470	376	282	188	94
450	1179	1061	943	825	707	590	472	354	236	118
500	1461	1315	1169	1023	877	730	584	438	292	146
550	1790	1611	1432	1253	1074	895	176	537	358	179
600	2172	1955	1738	1520	1303	1086	869	652	434	217
650	2611	2350	2089	1828	1567	1306	1044	783	522	261
700	3114	2803	2491	2180	1868	1557	1246	934	623	311
750	3687	3318	2950	2581	2212	1844	1475	1106	737	369
800	4335	3902	3468	3034	2601	2168	1734	1300	867	434
850	5065	4558	4052	3546	3039	2532	2026	1520	1013	506
900	5884	5296	4707	4119	3530	2942	2354	1765	1177	588
950	6798	6118	5438	4759	4079	3399	2179	2039	1360	680
1000	7815	7034	6252	5470	4689	3908	3126	2344	1563	782
1050	8942	8048	7154	6259	5365	4471	3577	2683	1788	894
1100	10186	9167	8149	7130	6112	5093	4074	3056	2037	1019
1150	11557	10401	9246	8090	6934	5778	4623	3467	2311	1156
1200	13060	11754	10448	9142	7836	6530	5224	3918	2612	1306

Emissive Power (W) at Varying Temperatures and Emissivities BTUH/FT^2

Figure 13

				EMISSIVITIES						
°F	**1.0**	**0.9**	**0.8**	**0.7**	**0.6**	**0.5**	**0.4**	**0.3**	**0.2**	**0.1**
1250	14707	13236	11766	10295	8824	7354	5883	4412	2941	1470
1300	16504	14854	13203	11553	9902	8252	6602	4951	3301	1650
1350	18460	16614	14768	12922	11076	9230	7384	5538	3692	1846
1400	20586	18527	16469	14410	12352	10293	8234	6176	1447	2059
1450	22891	20602	18313	16024	13735	11446	9156	6867	4578	2289
1500	25384	22846	20307	17769	15230	12692	10154	7615	5077	2538
1550	28074	25267	22459	19652	16844	14037	11230	8422	5615	2807
1600	30974	27877	24779	21682	18584	15487	12390	9292	6195	3097
1650	34092	30683	27274	23864	20455	17046	13637	10228	6818	3409
1700	37441	33697	29953	26209	22465	18720	14967	11232	7488	3744
1750	41030	36927	32824	28721	24618	20515	16412	12309	8206	4103
1800	44871	40384	35897	31410	26923	22436	17948	13461	8974	4487
1850	48975	44078	39180	34282	29385	24488	19590	14692	9795	4898
1900	53355	48020	42684	37348	32013	26678	21342	16006	10671	5336
1950	58022	52220	46418	40615	34813	29011	23209	17407	11604	5802
2000	62990	56691	50392	44093	37794	31495	25196	18897	12598	6299
2100	73873	66486	59098	51711	44324	36936	29549	22162	14775	7387
2200	86110	77499	68888	60277	51666	43055	34444	25833	17222	8611
2300	99808	89827	79846	69866	59885	49904	39923	29942	19961	9980
2400	115076	103568	92061	80553	69046	57538	46030	34523	23015	11508
2500	132037	118833	105630	92426	79222	66018	52815	39611	26407	13204
2600	150804	135724	120643	105563	90482	75402	60322	45241	30161	15080
2700	171505	154354	137204	120053	102903	85752	68602	51452	34301	17150

Figure 14

EXCERPT

(A portion of Pages 7, 8, 12, and 13 - plus all of Pages 19 and 20)

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION TECHNICAL NOTE D - 998

MEASUREMENTS OF TOTAL EMITTANCE OF SEVERAL REFRACTORY
OXIDES, CERMETS, AND CERAMICS FOR TEMPERATURES
FROM 600° F TO 2,000° F.

By William R. Wade and Wayne S. Slump

Reproduced by permission of NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Refractory Paints

Pyromark® paint - Included in the materials investigated was a refractory paint, Pyromark®, manufactured by the Tempil Corporation (132 West 22nd St., New York, N.Y.). Although the exact composition of this paint is not available from the manufacturer, it probably contains finely ground chromium oxide powders which produce a flat black. The paint may also contain a small amount of graphite, plus a silicate binder and an organic vehicle. An X-ray diffraction analysis indicated only the presence of CrO. This paint is of interest because it will allow normal metal forming and fabrication procedures that may not be possible with some other types of coatings.

For the Pyromark® coating, specimen preparation consisted of applying the paint in thickness of approximately 0.001 inch to solvent - cleaned metal strips of "as-rolled" stainless steel 321 and of polished Inconel. The coated specimens were then air-dried at room temperature for at least 24 hours before testing. A preliminary investigation to determine the adherence of this paint to a metallic substrate and the emittance stability at elevated temperatures was conducted on both the stainless steel and Inconel specimens. This investigation indicated that the Pyromark® coating has a high total normal emittance that remains essentially stable at temperatures up to 2,000° F as shown on Page 33, Figure 15. Visual examination of the test samples indicated no appreciable effort on the coating after exposure to temperatures as high as 2,000° F for a period of 15 minutes.

Subsequent measurements of the total normal emittance for both the Inconel and stainless-steel specimens over a temperature range from 600° F to 2,000° F are shown on Page 33, Figure 16. These curves indicate that slightly different values of emittance may be obtained by applying the Pyromark® to different substrate materials, the values increasing from 0.81 at 600° F to 0.94 at 2,000° F for the polished Inconel and from 0.78 at 600° F to 0.90 at 2,000° F for the as-rolled stainless-steel substrate.

To determine whether the Pyromark®-coated specimens emit diffusely is accordance with Lambert's cosine law, measurements were made of radiant intensities from test specimens at angles of observation from 0^{o} to 60^{o} for a temperature range from 600^{o} F and $2{,}000^{o}$ F are shown on Page 34, Figure 17, where the circle corresponds to Lambert's cosine law for perfectly diffuse emission. The close agreement between the measured values and Lambert's cosine law indicates that this coating emits diffusely for the temperatures and angles considered. Therefore, the measured values of total normal emittance are close approximations of total head spherical emittance.

SUMMARY OF RESULTS

The investigation of chemically oxidized Inconel indicates that, although a high stable value of emittance may be attained by this process, exposure to temperatures in excess of $1{,}600^{o}$ F causes a change in the surface coating due to the formation of another oxide. This change in the oxide results in a change of total emittance, limiting this coating to fairly low temperature applications where stable emittance is desired. This measured values for this coating were found to vary from 0.91 to 0.94 over a temperature range from 600^{o} F to $1{,}600^{o}$ F.

Results obtained for the paint coating tested, specifically chromium oxide paint (Pyromark®) and aluminum oxide paint, indicate the coatings have high stable values of total emittance. The emittance values ranged from 0.78 at 600^{o} F to 0.90 at $2{,}000^{o}$ F for Pyromark® coating on "as-rolled" stainless steel 321 and from 0.81 at 600^{o} F to 0.94 at $2{,}000^{o}$ F for this same coating on polished Inconel.

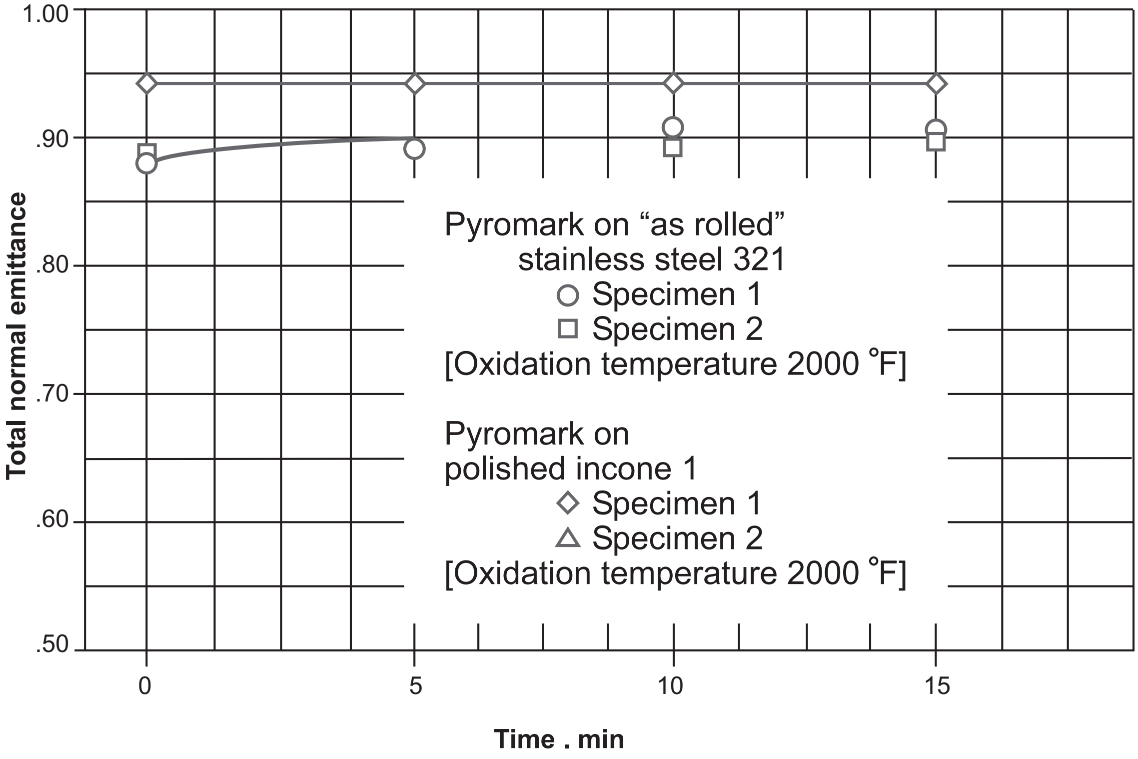
Emittance measurements of a variety of silicon carhide and silicon nitride materials show fairly high values of total emittance, which proved to be quite stable, after oxidation in air at $1{,}800^{o}$ F. The measured values for these materials show a decrease of emittance with increasing temperatures over the temperature range considered, a characteristic of most ceramic materials. The total emittance for these stably oxidized materials varied from 0.94 at 600^{o} F to 0.62 at $1{,}600^{o}$ F.

The ceramics investigated were primarily chromium/aluminum oxide composites. Certain modifications of these basic components result in three distinct materials which were designated by Haynes Stellite Co. as LT-1, LT-2, LT-LB. These materials were all reported to have good thermal shock properties, resistance to oxidation and to mild abrasion, and high stable values of total emittance. The results of measurements conducted on these cerments show a nearly linear increase of emittance over a temperature range from 600^{o} F to $1{,}600^{o}$ F, ranging from 0.76 to 0.94 for the three materials tested.

Investigation of the total emittance conducted on flame-sprayed coatings of the cermets indicates that they can have slightly higher values of emittance. The measured values for these coatings varied from 0.84 at 800^{o} F to 0.94 at $1{,}800^{o}$ F.

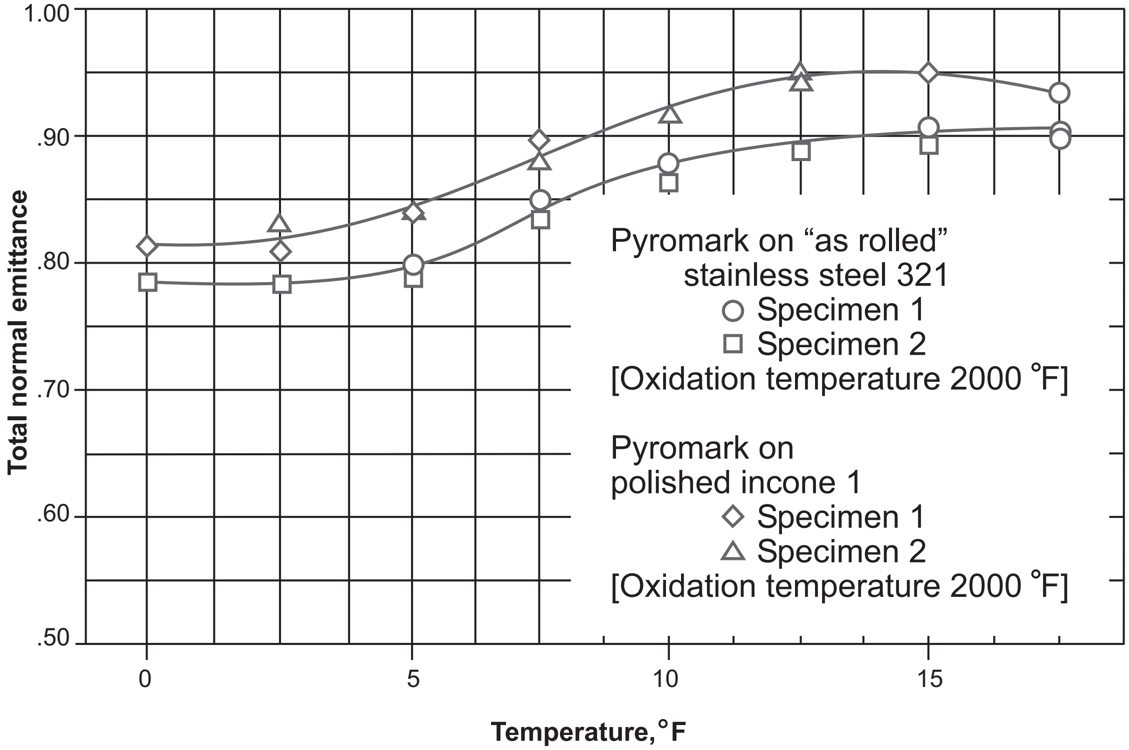
Langley Research Center, National Aeronautics and Space Administration, Langley Air Force Base, VA. Oct. 13, 1961

Figure 15



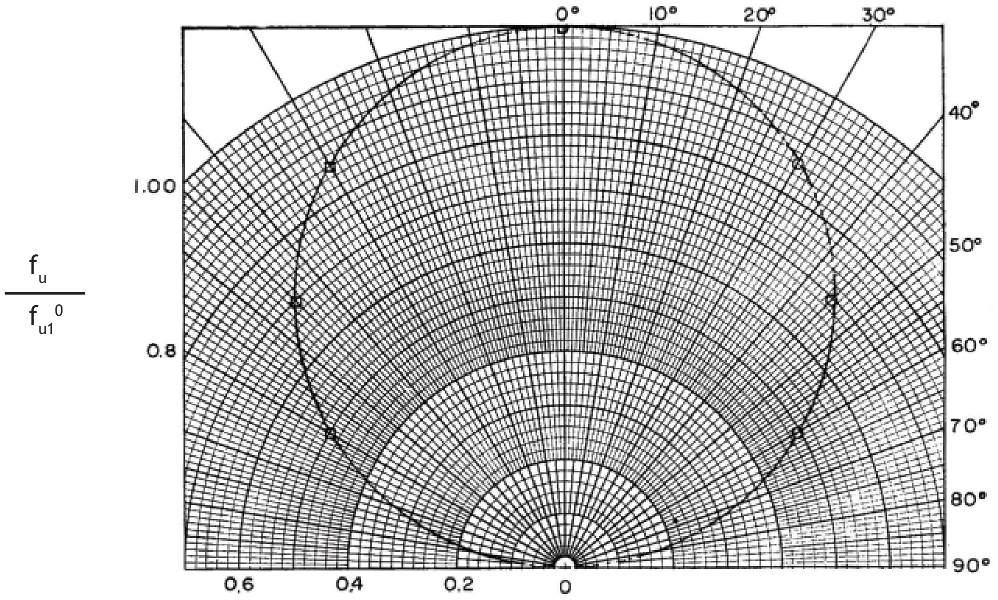
Variations of the total normal emittance of Pyromark® coating with time of heating in quiescent air.

Figure 16



Variations of the total normal emittance of Pyromark® coating as a function of temperature.

Figure 17



On “as-rolled” stainless
stainless steel 321
Temperature

○ 600 °F
△ 2000 °F

On polished Inconel
Temperature

◇ 600 °F
△ 2000 °F

Comparison of the radiant flux density of Pyromark® coating on as-rolled stainless steel 321 and on polished Inconel heated for 15 minutes at 2,000° F with Lambert’s cosine law for diffuse emission.

PYROMARK® is a product of Tempil° Division So. Plainfield, N.J.

For literature and a 2oz. sample of black PYROMARK® write to:

REFRACTORY COATINGS DIVISION

Tempil°

Division of Big Three Industries, Inc.

HAMILTON BOULEVARD

SO. PLAINFIELD, N.J. 07080

Phone: 201.757.8300 Teies 138662

Manufacturers of Tempestias° Tempilac° and Tempil° Pellets

3.3 Reflectivity

Reflectivity is a material property related to emissivity as follows

Emissivity + Reflectivity = 1.0

It is a measure of the amount of energy redirected by a surface as compared to the amount that would be reflected by a prefect reflector (a material having emissivity equal to zero).

Reflectivity of a surface is dependent on a number of factors, the most important of which are as follows:

- Surface condition of the material: in a manner similar to emissivity, reflectivity is mainly determined by the surface layers of the material. A smooth, dense shiny surface generally produces highly reflective qualities.
- Wavelength of emitted energy: for most materials, some variations in reflectivity can be expected as wavelength changes. For infrared energy, this variation is small and is usually neglected. It is for this reason that infrared radiant energy (particularly, long wave infrared energy) is considered essentially "color-blind."

The reflector (or shade, as it is sometimes called) is a key component of a radiant heating appliance. In addition to a highly reflective material, the configuration of the reflector can effect the radiant output of the appliance as follows.

- Overall depth of the reflector can control the extent of convective loss from the appliance by trapping hot air.
- Angularity of the reflector determines the amount of energy directed away from the appliance, evenly downward toward the occupied space.

3.3.1 Market Report - Reflectivity

Below is a summary of materials and reflectivities generally available within the radiant heating appliance industry.

MATERIAL	REFLECTIVITY**
- Tin	0.94
- Chrome	0.92
- Aluminum (mill finish)	0.91 to 0.95
- Aluminum (polished)	0.91 to 0.95
- Nickel	0.90
- Aluminized Steel (type 1)	0.50 to 0.80
- Galvanized Steel	0.72
- Stainless Steel (type 304)	0.48 to 0.66
- Stainless Steel (type 430 polished)	0.80 to 0.90

**References: A.G.A. Research Bulletin #83 and Critical Table, National Bureau of Standards

Page 38, Figure 17 indicates a number of reflector configurations available throughout the industry. Consideration is given to each configuration regarding reflectivity, angularity and ability to prevent convection losses.

3.4 Convection Losses

Convection is the transfer of heat from one point to another by moving and mixing masses of fluid (either air or liquid). Natural convection occurs when the motion is due to differences in density as caused by temperature differentials. Forced convection occurs when the motion is imparted via mechanical means, independent of temperature.

Convection losses occur when a hot object is cooled by natural and/or forced convection. Convection loss of heat from a radiant heating appliance causes a condition within the heated space referred to as stratification. Building heat stratification, when experienced, often results in increased fuel usage as follows:

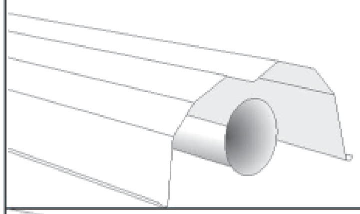
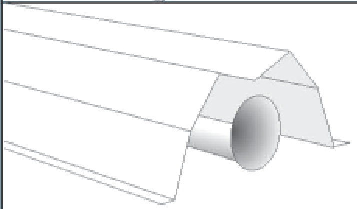
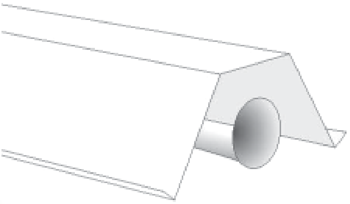
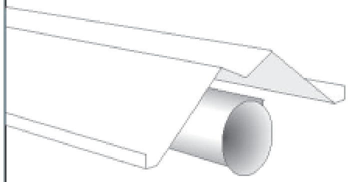
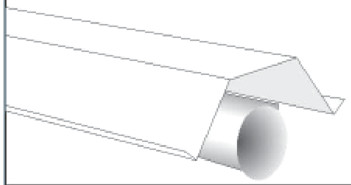
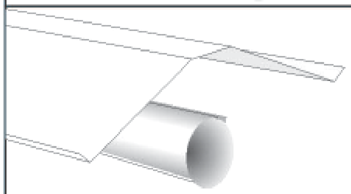
- Transmission heat loss is increased through the roof and upper wall areas by virtue of the increased temperature difference from inside to outside.
- Air change heat loss in increased to the extent that there is ex-filtration through cracks or openings in or near the roof as a result of the stack effect.

For convective heated spaces it is not unusual for ceiling and upper wall areas to be as much as 25^{o} F to 30^{o} F warmer than areas near the floor. This can be controlled somewhat through the use of large circulating fans, however it should be noted that the electric power requirement for operation of these fans is substantial.

For buildings heated with radiant heating appliances, the subject of stratification is essentially a non-issue, pro-

Figure 18

CONFIGURATION COMPARISON

FIXTURE CONFIGURATION	REFLECTIVITY	ANGULARITY	PREVENT CONVECTION LOSS	FIXTURE EFFECTIVENESS	PATTERN EFFECTIBENESS	RADIANT EFFICIENCY
	E	E	E	E	E	A
	E	E	A	E	A	A
	E	B	B	B	A	A
	E	A	B	A	B	B
	E	B	B	B	B	A
	E	B	B	B	B	B

**Comparison Key: E = Above Average, A = Average, B = Below Average vided the reflector on the appliance traps hot air surrounding the radiant tube. Additionally, if the hot air is not contained, a substantial reduction in radiant output may result due to cooling of the radiant tube. For example, a reduction in tube temperature from 1100° F to 1000° F (a 9% decrease) can result in a radiant output reduction as high as 23%.

3.4.1 Market Report - Convection Losses

Page 37, Figure 18 summarizes available reflector configurations. It is important that the reflector system adequately prevent convection losses from the radiant heating appliance. While the practice of tilting reflectors is common in the radiant heating industry, the convection losses caused by this provide the same negative effects on performance as described above. To properly redirect radiant energy to the sides of a space, side extension shields are available from some manufacturers.

3.5 Fixture Efficiency

Fixture efficiency is a measurement index of the ability of a radiant heating appliance to release available radiant energy to the heated space. All radiant heating appliance fixtures absorb some infrared energy and convert it to heat, which is convected away. An inability to control this convection loss, as well as the direction and distribution of radiant energy results in a low fixture efficiency.

3.5.1 Market Report - Fixture Efficiency

Fixture efficiency is affected by many key components and properties of the components of a radiant heating appliance. Clearly, high emitter temperature, high tube emissivity and high reflector material reflectivity have a great influence on the fixture efficiency, however, a discussion of these areas without consideration for tube length, reflector shape, or the ability of the reflector to control convective loss is misleading. A determination of the effectiveness of a radiant heating appliance must include consideration of all of these factors acting as a system in order to be accurate.

3.6 Pattern Efficiency

Pattern efficiency is a measurement index of the ability of a radiant heating appliance to distribute radiant energy to the space in a manner consistent with the needs of the space. For example, consider a shipping dock. A relatively high, localized, direct radiant appliance output level may prove effective in meeting the needs of the space. In contrast, general comfort heating of a large space can be better served by a much lower evenly distributed symmetric radiant appliance output. As illustrated, the pattern efficiency of a radiant heating appliance is highly subjective, and must be considered in concert with the application in question in order to be meaningful.

3.6.1 Market Report - Pattern Efficiency

With the advent of numerous new products into the radiant heating market and the unavailability of adequate industry standards to differentiate the market approaches, it is easy to understand the confusion regarding suitability of one type of appliance versus another type in a particular application. Resolution of the problem is possible if, as a first step, the requirements of the space can be identified. Is it localized heating or general space heating? Do we want to warm people to comfort level or just keep the floors dry? What are the ambient conditions? Is it drafty? Is the activity level high or low? Answers to these (and many others) questions provide a strong design basis for a successful radiant heating application.

Secondly, an understanding of the performance capabilities of the radiant heating appliance (its fixture efficiency) is necessary to adequately match equipment to the requirement of the space. A single, larger burner on a very long tube may provide adequate BTU, however, it may not provide sufficiently even coverage to be appropriate for the application. Only when the requirements of the space are matched to the capabilities of the equipment can the heating equipment designer be assured of successful application of radiant heating equipment.

Page 37, Figure 18 indicates relative pattern efficiencies for various equipment configurations.

3.7 Absorptivity

Absorptivity is a material property commonly discussed with the application of radiant heating appliances to a space. It is a measure (ratio) of the amount of radiant energy that can be absorbed by a material as compared to the total radiant energy received by the material. For most materials, the absorptivity and the emissivity have identical values.

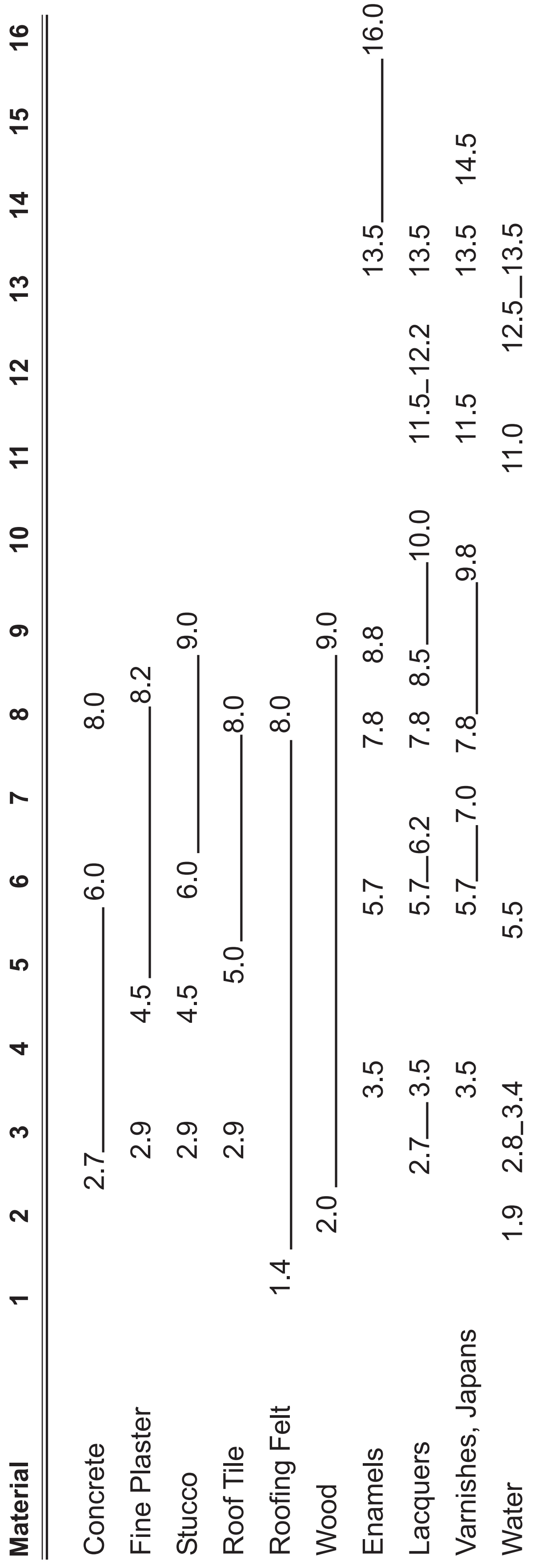
Absorptivity is most important when considering the effect a radiant heating appliance will have on a space. Because some objects (for example: water, concrete and people) absorb varying wavelengths of radiant energy more completely, it is important that the emitting radiant source produce the infrared energy in a wavelength that can be best utilized by the space.

3.7.1 Market Report - Absorptivity

Below is a summary report of major absorption wavelengths for miscellaneous materials.

MAJOR ABSORPTION WAVELENGTHS OF SOME MATERIALS

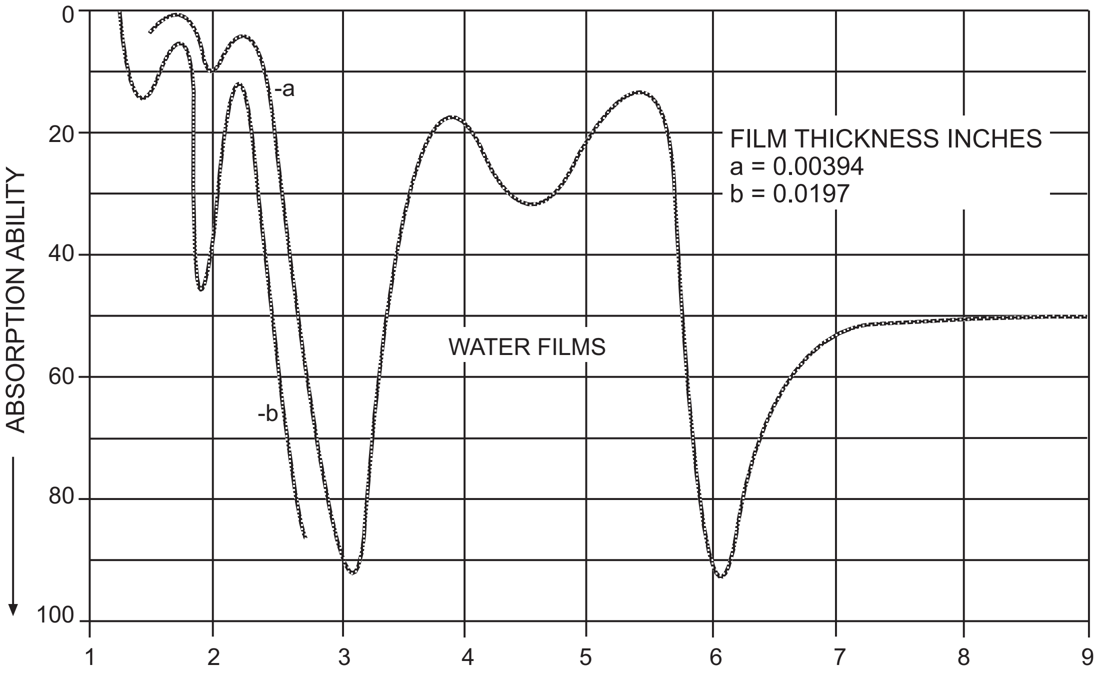
Major Absorption Wavelengths, Microns



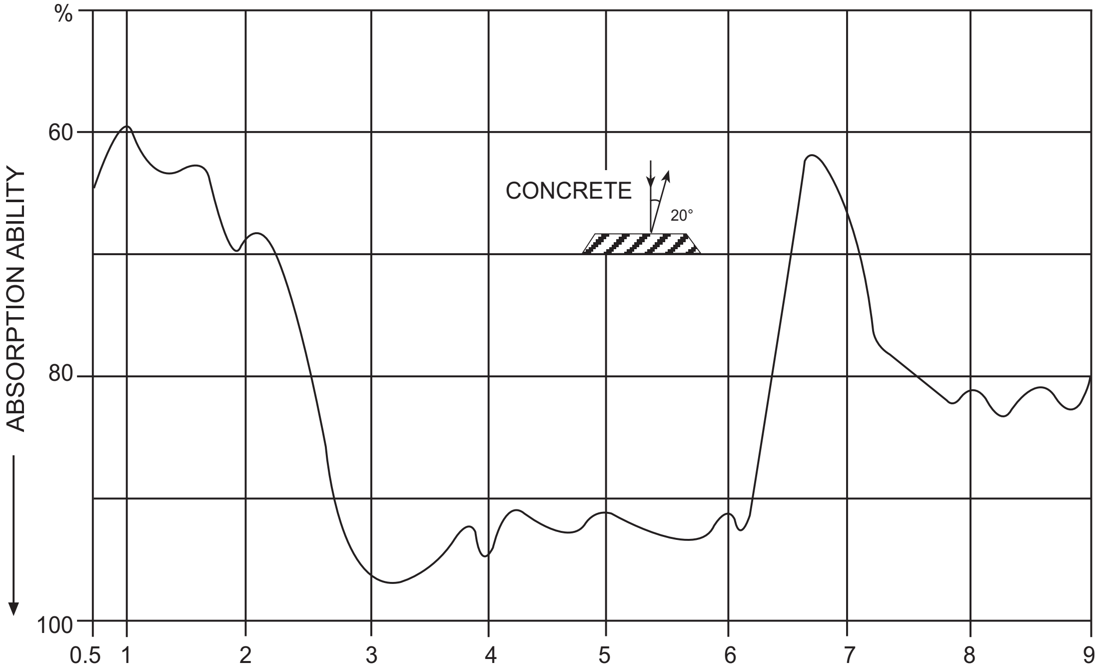
(*Reference A.G.A. Research Bulletin #92)

Absorption Ability for Water and Concrete

Figure 19



WAVE - LENGTH



WAVE - LENGTH

Absorbed Energy Comparison for Water and Concrete at
nitter Temperature of 9000° F (low intensity) and 1800° F (high intensity)

Figure 20

CHART 1 WAVELENGTH - MICRONS

BTU/SQ. FT. -HR	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12+	**TOTAL THEORETICAL BTU AVAILABLE (1)**
TEMP.														
1800° F	--	7792	12474	8884	5339	3275	2019	1301	942	673				**44870**
900° F	--	90	635	1000	952	764	558	423	317	229	176	129	607	**5880**

CHART 2 WAVELENGTH - MICRONS

	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12+
ABSORPTIVITY													
Concrete	--	0.65	0.80	0.95	0.93	0.95	0.75	0.8	0.70	0.70	0.70	0.70	0.70
Water	--	0.01	0.18	0.50	0.40	0.30	0.65	0.55	0.05	0.45	0.45	0.45	0.45

CHART 3 (1) **WAVELENGTH - MICRONS**

BTU/SQ. FT. -HR	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12+	**TOTAL BTU ABSORBED (2)**	**% OF THEORETICAL BTU AVAILABLE (3)**
HIGH INTENSITY 1800° F															
Concrete (BTU)	--	5045	9979	8440	4965	3111	1514	1041	735	741				**35301**	**78.7**
Water (BTU)	--	77	2245	4442	2135	982	1312	715	471	302				**12681**	**28.3**
LOW INTENSITY 900° F															
Concrete (BTU)	--	58	508	950	885	726	418	338	247	160	123	90	425	**4928**	**83.8**
Water (BTU)	--	0.90	114	500	381	229	363	233	159	103	79	58	273	**2492**	**42.4**

NOTE:

(1) CHART 3 VALUES CALCULATED FOR AN INDIVIDUAL WAVELENGTH AS FOLLOWS:
VALUE FROM CHART 1 X VALUE FROM CHART 2 = CHART 3

(2) TOTAL VALUES ARE SUMMATION OVER ALL WAVELENGTHS

(3) % OF THEORETICAL AVAILABLE BTU CALCULATED AS FOLLOWS:
TOTAL BTU ABSORBED / TOTAL THEORETICAL AVAILABLE BTU X 100% = % OF THEORETICAL AVAILABLE

Page 41, Figure 19 illustrates average absorption ability for water films and concrete in graphical form. By comparing the wavelength of maximum absorption for these materials with the energy distribution for various wavelengths on Page 29 and 30, Figure 13 of an estimate of the utilization of radiant energy from different source temperatures can be made. Page 41, Figure 19 indicates that for an 1800° F emitter source (high intensity), approximately 78.7 % of the emitted energy is absorbed by concrete and 28.3 % of the emitted energy is absorbed by water.

For a 900° F emitter source, (low intensity) approximately 83.8 % of the emitted energy is absorbed by concrete and 42.4 % of the emitted energy is absorbed by water.

3.8 Radiant Efficiency

Radiant efficiency is a measurement index of a radiant heating appliance comparing the actual radiant energy output to the fuel energy input. Radiant output is a function of many system characteristics. Among these are:

- Emitter temperature
- Heat exchanger emissivity
- Total emitter surface area
- Fixture efficiency

Large surface areas, together with high emissivity and uniformly high emitter temperatures provide the highest percentage of input as radiant output.

The ANSI standard for testing gas fired infrared heaters (ASNI Z83.6a -1989) requires a minimum radiant efficiency of 35 %, however, for radiant heating appliances physically too large to have the radiant output measured according to the method of test in the standard, this requirement is waved. Instead, a thermal efficiency of not less than 70 % must be attained (refer to sections 2.9.1 and 2.9.2 in ANSI Z83.6a-1989 for more information).

3.8.1 Market Report - Radiant Efficiency

As discussed previously on Page 23, Section 3.2, the determination of radiant output from a radiant heating appliance cannot be calculated in a practical (simple) manner. Manufacturer claims of radiant efficiencies of 55-60 % or more cannot be substantiated through measurement. Measurement is the only way to insure accurate radiant efficiency values. In order to make radiant efficiency information comparable for manufacturer to manufacturer, The Gas Research Institute and the Institute of Gas Technology are developing a standard efficiency measurement method for radiant appliances. Until this work is complete, it is prudent to review so-called radiant efficiency information thoroughly to determine it validity.

3.9 Glossary of Terms

Absorptivity - An inherent property of a material evaluated by the ratio of the radiant energy absorbed to that falling upon it. It is equal to the emissivity for radiation of the same wavelength.

Air Change - (1) Introduction of new, cleansed, or recirculated air to a space. (2) A method of expressing the amount of air movement into or out of a building or room in terms of the number of building volumes or room volumes exchanged in unit time.

Air Shutter - An adjustable device for varying the size of the primary air inlet(s).

Aluminized Steel - Steel having resistance to oxidation due to formation of an aluminum/aluminum allow coating by hot dipping, hot spraying or diffusion processes. Emissivity typically 0.2-0.5 (unheat treated), 0.7-0.8 (heat treated).

Ambient Air - The surrounding air (usually outdoor air or the air in an enclosure under study).

Annual Fuel Utilization Efficiency (AFUE) - The ratio of annual output energy to annual input energy which includes any non-heating season pilot input loss.

Atmospheric Burner - A device for the final conveyance of the gas, or a mixture of gas and air at atmospheric pressure, to the combustion zone.

Black Body - (1) A body that absorbs all the radiant energy falling upon it. (2) A body that has the maximum theoretical radiant energy emittance at a given absolute temperature.

British Thermal Unit (BTU) (AN I-P UNIT) - The heat energy in a BTU was defined by the Fifth International Conference on the Properties of Steam (1956) as exactly 1055.055 852 62 J. It was related through specific heat to the IT calorie so that 1 cal/kg K = 1 Btu/lb F for 1 lb = 453.592 37 g. The mechanical equivalent energy of a Btu is approximately 778.169 262 ft lb. The heat energy of a Btu is approximately that required to raise the temperature of a pound of water from 59^{o} F to 60^{o} F.

Burner Control Assembly - An assembly of various valves, burner head, ignition system, filter, etc. necessary to operate and control the burner.

Burner, End - The burner control assembly installed at the end of a series of burners farthest from the vacuum pump.

Calorie - Heat required to raise the temperature of 1 gram of water 1 degree Celsius, specifically from 4^{o} C to 5^{o} C. Mean calorie = 1/100 part of the heat required to raise 1 gram of water from 0^{o} to 100^{o} C. Great calorie or kilocalorie - 1000 calories.

Calorized Steel - Steel having resistance to oxidation due to heating in an aluminum powder at 1472° F to 1832° F. Emissivity typically 0.6.

Chimney - One or more passageways, vertical or nearly so, for conveying flue gases to the outside atmosphere.

Chimney Effect - The rising of air or gas in a duct or other vertical passage, as in buildings, inducted when the density of air in the chimney is lower than that of surrounding air or gas.

CLO - A non-SI unit of clothing insulation defined as the thermal insulation necessary to keep a sitting person comfortable in normally ventilated room at 70° F (21° C) and 50 % relative humidity. In physical terms, the thermal resistance of one CLO - 0.88 F-ft^{2}h/Btu (0.155 K m2/W).

Combustion Air - The air required for complete combustion of fuel, and usually consisting or primary air and, excess air.

Combustion Chamber - A 20 inch long section of 4” tubing with a top fitting to accept a burner. A window is provided to observe the flame.

Combustion Chamber, End - A combustion chamber that received the first burner in a series of burners. (Farthest from the vacuum pump).

Comfort Chart - A chart showing dry-bulb temperatures , relative humidities and air motion so the effects of the various conditions on human comfort may be compared.

Comfort Zone - A condition in an environment or enclosure whereby a suitable operative temperature is maintained. The required range of operative temperature for human comfort is defined by the comfort chart (refer to ANSI 55-1981).

Condensate - Liquid formed by condensation of a vapor. In combustion of hydrocarbon fuels, water condensed from flue products (this is typically slightly acidic). NOTE: Combustion of natural gas produces 11.2 gallons of condensate for each 1 x 10^6 BTU burned. Combustion of propane gas produces 8.9 gallons of condensate for each 1 x 10^6 BTU burned. Condensation begins at/below the dew point.

Condensation - The change of state of vapor into a liquid by extracting heat from the vapor.

Conduction (Heat Conduction) - Process of heat transfer through a solid.

Control, Single Stage - A control that cycles a burner from the maximum heat input rate and off.

Convection - (1) Transfer of heat by a fluid moving by natural variations in density. (2) Transfer of heat by the movement of a fluid.

Forced Thermal Convection - Heat transmission by mechanically induced movement of fluid.

Free Thermal Convection (Natural Convection) - Heat transmission by movement of a fluid caused by density difference.

Coupling - A device used to connect sections of tubing. A standard unlines coupling is used to connect radiant tubes.

Coupling, Damper - A coupling with a damper. This is installed where needed to adjust the vacuum in system.

Coupling, Lined - A coupling lined with Inconel. It is used to connect tail pipe tubes.

Decorative Grille - A 1/2" square honeycomb grille installed below the radiant tube. This is for decorative purposes only. A 1-foot wide model installs directly on the reflector. A 2-foot wide model installs in a suspended ceiling.

Degree Day - A unit of accumulated temperature departure, based on temperature difference and time. Used in estimating fuel consumption and specifically nominal heating load of a building in winter. For any one day, the number of degree days of temperature difference between a given base temperature usually 65^{o} F (18.3^{o} C and 18^{o} C in Canada) and the mean outside temperature over 24 hours.

Direct Exhaust System - A mechanical venting system supplied or recommended by the manufacturer through which the products of combustion pass directly from the furnace, heater or boiler to the outside which does not employ a means of draft relief.

Direct Vent System - A system consisting of (1) a central furnace, heater or boiler for indoor installation, (2) combustion air connections between the furnace, heater or boiler and the outdoor atmosphere. (3) flue gas connections between the furnace, heater or boiler and the vent cap, and (4) vent cap for installation outdoors, supplied by the manufacturer and constructed so all air for combustion is obtained from the outdoor atmosphere and all flue gases are discharged to the outdoor atmosphere.

Draft Hood - A device installed on gas-fired appliances designed to protect the appliance from chimney draft disturbances.

Dry-Bulb Temperature - The temperature of air indicated by an ordinary thermometer.

Dual Fuel Burner - A burner designed with two separate orifices and gas trains for both pilot gas flow and main gas flow. This permits a fuel conversion to be made by selective energizing of the gas trains (i.e. and without physical change of orifices).

Efficiency - The ratio of the energy output to the energy input of a process of a machine.

Efficiency, Thermal - The ratio of the useful/available energy at the point of use the thermal energy input over a designated time.

Efficiency, Radiant - The measure of the percentage of gross BTU input that is realized/available as direct radiant BTU output.

Emissivity (e) - The ratio of the total amount of radiant energy emitted by a body to that omitted by a perfect black body at the same temperature. (Perfect black body emissivity (e) = 1, perfect reflector (e) = 0.).

End Vent Plate - A plate (approx. 4" dia.) that is attached to the end combustion chamber. This plate is pre-drilled and the correct size must be installed to match the burner. This is used with or without outside air system.

Excess Air - In combustion, the percent or air greater than that required to completely oxidize the fuel.

Flow Unit - The amount of fuel-air mixture required for firing the rate of 10,000 BTU per hour. The would equal 1.83 SCFM. Flow units are used as a measure of flow rate for both combustion air and the air entering through the end vent.

Flue - The general term for the passages and conduits through which flue gases pass from the combustion air and the air entering through the end vent.

Flue Gases - Productions of combustion and excess air.

Flue Losses - The sensible heat and latent heat above room temperature of flue gases leaving the appliance.

Forced Draft - Combustion air supplied under pressure to the fuel burning equipment.

Gas Connector Assembly - A semi-rigid or flexible connection between the gas line and the burner control assembly. This includes a shut-off valve with a 1/2" female pipe connection.

Halogenated Hydrocarbon Compounds - Hydrocarbon, compounds which contain halogen elements such as hydrogen, chlorine, fluorine, bromine, and iodine. These are generally non-corrosive until after being heated at several hundred degrees (as during a combustion process). At this point a decomposition takes place, freeing halogen compounds. When these compounds are combined with moisture for combustion products, extremely corrosive acids are formed.

Heat - A form of energy that is exchanged between a system and its environment or between parts of the system induced by temperature difference existing between them.

Heat Gain - The quantity of heat absorbed by an enclosed space or system.

Heat, Latent - Change of enthalpy during a change of state.

Heat Reservoir - An ideal system that can absorb or reject an indefinitely large amount of heat.

Heat, Sensible - Heat that causes a change in temperature.

Heat, Specific - The ratio of the quantity of heat required to raise the temperature of a given mass of any substances one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 59° F (15° C)) one degree.

Heating Value, Higher (HHV) - The heat produced per unit of fuel when complete combustion takes place at constant pressure and the products of combustion are cooled to the initial temperature of the fuel and air when the vapor formed during combustion is condensed.

Heating Value, Lower (LHV) - The gross heating value minus the latent heat of vaporization of the water vapor formed by the combustion of the hydrogen in the fuel.

Induced Draft - Drawing air from the combustion chamber by mechanical means.

Inductive Load - An alternating current load in which current lacks voltage.

Infiltration - The uncontrolled inward air leakage through cracks and interstices in any building element around windows and doors of a building, caused by the pressure effects of wind or the effect of differences in the indoor and outdoor air density.

Lanced Stop - A tab in the lined coupling used to prevent racking of the coupling when driving lock member. (Provided on all couplings produced after July 1979).

Liquefied Petroleum Gases - The terms “Liquefied Petroleum Gases”, “LPG” and “LP-Gas” include any material which is composed predominantly of any of the following hydrocarbons, or mixtures of them; propane, propylene, butanes (normal butane or isobutane), and butylenes. This high heating value gas is stored under high pressure in liquid form.

Make-up Air - Air brought into a building from the outside to replace that exhausted.

Mean Radiant Temperature (MRT) - The single temperature of all enclosing surfaces which would result in the same heat emission as the same surface with various different temperatures.

Orifice - The opening in an orifice cap, orifice spud, or other device whereby the flow of gas is limited and through which the gas is discharged.

Orsat Apparatus - A gas analyzer based on absorption of CO_2, O_2, etc. by separate chemicals that have a selective affinity for each of those gases.

Power Burner - A burner in which either gas or air, or both, are supplied at pressures exceeding, for gas, the line pressure, and for air atmospheric pressure, this added pressure being applied at the burner.

Primary Air - The air introduced into a burner which mixes with gas before it reaches the port(s).

Pyranometer - An instrument that measures the combined direct and indirect radiation by means of a calibrated sensing element.

Radiant Branch - A section of radiant pipe with one or more burners firing in series which is connected to the tail pipe.

Radiation - The transfer of energy in wave form for a hot substance to another independent substance cooler in temperature with no material means of heat transfer.

Radiant Tube - That section of tubing run between burners and up to 50 feet (depending upon burner size) downstream from the last burner in a series. This tubing is of hot rolled steel.

Reflector - A device conFigured to direct radiant energy to the point of use in the space while absorbing little energy.

Reflector Support - A device that orients and maintains reflector position.

Residential Application - Providing comfort heating for single family living quarters.

Resistive Load - (1) An electric load without capacitance or induction, or one in which inductive portions cancel capacitive portions at the operating frequency. (2) An electric load with all energy input converted to heat.

Single Fuel Burner - This is the standard burner in which the pilot and main orifices can be changed to fire with either natural gas or propane. No change is required in the regulator settings.

Stack - (1) A structure that contains a flue, or flues, for the discharge of gases. (2) The vertical train of a system of soil, waste, or vent piping extending through one or more stories.

Stack Effect - The impulse of a heated gas to rise in a vertical passage such as in a chimney, small enclosure or building due to density differences.

Stack Gases - The mixture of flue gases and air that enters the draft diverter, draft hood, integral draft diverter or stack.

Stainless Steel - Any of several steels containing 12 to 30 % chromium as a principle alloying element; they usually exhibit passivity in aqueous environments; providing corrosion resistance. Typical emissivity (e) = 0.45.

Stoichiometric Combustion (Perfect Combustion) - Fuel burning completely; all combustibles are consumed with no excess air. Only the theoretical amount of oxygen is used (chemically correct ratio of furl to air).

Stratification - Division into a series of graded layers, as with thermal gradients across a stream.

Tail Pipe - That section of tubing connecting the last section of radiant tubing in a series of burners to the vacuum pump. This tubing is porcelain lined.

Shared - A section carrying the flow of combustion gases of more than one radiant branch.

Unshared - A section carrying the flow of combustion gases on only one radiant branch.

Therm - A quantity of heat equal to 100,000 BTU.

Thermal Expansion - Increase in or more of the dimensions of a body, caused by a temperature rise.

Thermostat - (1) Automatic control device, responsive to temperature, used to maintain a constant (static) temperature. (2) A temperature-activated switch. (3) An instrument that responds to changes in temperature, and directly or indirectly controls temperature. (4) A temperature sensitive device that automatically opens and closes an electric circuit to regulate the temperature of the space with which it is associated.

Tube and Reflector Hanger - A device for supporting heat exchanger and reflector.

U - Factor - The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids. It is evaluated by first evaluating the R - value and then computing its reciprocal.

U Factor - Fuel use factor per 1000 BTU/hr calculated heat loss.

Vacuum Pump System - A complete combustion system consisting of a vacuum pump, burners, 1 control panel, thermostats and 4 inch O.D. steel tubing for heat exchanger surface in the form of radiant and tail pipe plus assorted reflectors and other hardware. The number of such systems required is based primarily on the heat loss of the building.

Vacuum, End Vent - Vacuum measurement taken at the beginning of a radiant branch (the end vent) on a CORAY-VAC® System.

Vacuum, Orifice - Vacuum measurement taken at the orifice access plug on a CO-RAY-VAC burner.

Vent / Air Intake Terminal - A device which is located on the outside of a building and is connected to a furnace, boiler or heater by a system of conduits. It is composed of an air intake terminal through which the air for combustion is taken from the outside atmosphere, and an exhaust terminal from which flue gases are discharged.

Vent Pipe - Passages and conduits in a direct vent system through which gases pass from the combustion chamber to the outdoor air.

Zone (Control Zone) - A space or group of spaces within a building with heating or cooling requirements sufficiently similar that comfort conditions can be maintained by a single controlling device.

Glossary References

American National Standard for Gas Fired Infrared Heaters, ANSI Z83.6-1982

American Gas Association, Cleveland, Ohio, 1982.

North American Combustion Handbook, 2nd Edition

North American Manufacturing Company, Cleveland, Ohio 1978.

Terminology of Heating, Ventilation, Air Conditioning and Refrigeration.

American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1986.

4.0 Radiant Heating Design Issues

In order to insure successful application of radiant heating appliances, a number of design issues must be considered. Proper utilization of the inherent performance advantages of radiant heating depends heavily on a thorough analysis and understanding of the application space, its utilization, and the expected benefits from the radiant heating system. In essence, what is to be accomplished? What are the requirements of the application? How will the radiant heating system interface with the occupants, scheduling parameters and activity level? These questions and others must be answered to properly execute the design thought process. Additional issues, among them

- Utilization factors,
- Radiant adjustment to heat loss, and
- Height adjustment for radiant energy can effect the design approach. These issues are discussed briefly below.

4.1 Utilization Factors

Consideration must be given to the utilization parameters for a particular application. Of major importance in such a consideration is the determination of an accurate heating load (heat loss) calculation. *The American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE)* has well documented methods for determining the heating load (See ASHRAE Handbook of Fundamentals - 1985, ch. 25). The importance of an accurate heat loss cannot be over emphasized, as the best equipment designs can be rendered ineffective if poorly applied. Areas of special concern in determining heat loss relate to:

- Accurate constructional descriptions
- Air change estimation to determine infiltration loss
- External factors (such as proper design temperature, wind velocity and direction)
- Internal heat gains
- Additional heating load to compensate for cold masses (for example, machinery or trucks)

In addition to an accurate heating load, special consideration must be given to allow for radiant effects (as compared to conventional air heating). The realization that maximum comfort is only achieved by providing warm surfaces which exchange gentle, well distributed radiant energy with the occupants, will insure a successful radiant heating application. Additionally, possible radiant shadows cast by beams or pillars can effect proper placement of the radiant appliance.

Heaters should be located sufficiently inboard of exterior walls to prevent direct radiation from the heater from contacting the wall and being lost to the outside. Page 54, Figure 21 represents graphically the relationship of radiant output intensity to horizontal distance D from center line of burner emitter for a burner mounted at height H. Distance D from center line is given in multiples of mounting height H. As shown, for distance D greater than 1.5H the intensity drops below 50 % of the intensity at center line (D=O). As a practical limit, radiant appliances with high fixture efficiency should be located no closer to exterior walls than 1.5 times the mounting height.

Page 54, Figure 21 can also be used to estimate the coverage capability of a radiant heating appliance. (See Section 4.1.1 below for more information on coverage). The specific application will determine the degree of coverage is a major issue when applying radiant heating appliances to station heating (spot heating).

Note, relationships illustrated on Page 33, Figure 15 relates to a deep dish, high fixture efficiency radiant system.

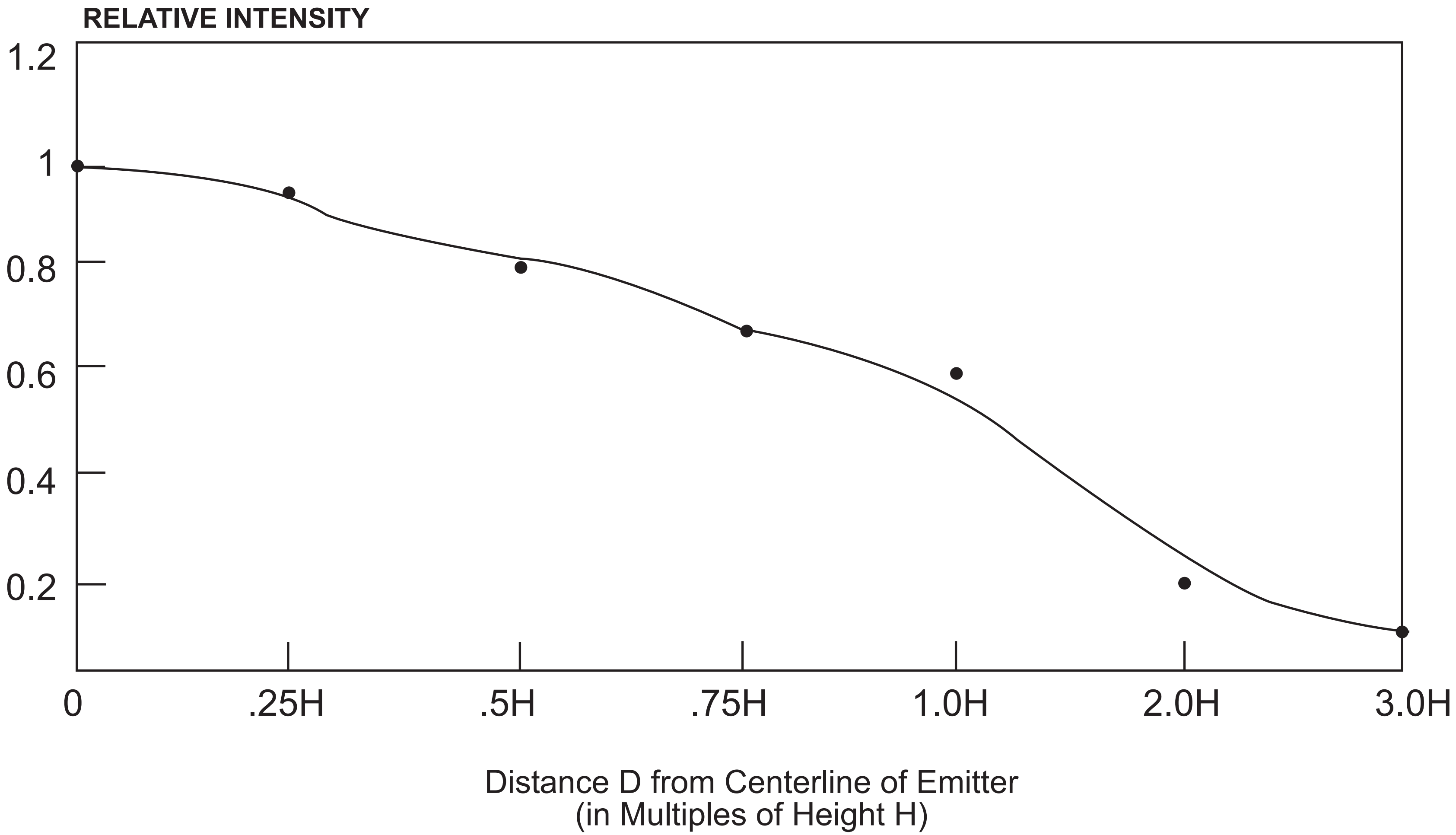
4.1.1 Coverage

Page 55, Figure 22 demonstrates the physical coverage differences between a well distributed, multiple burner system and a single large burner. Because floor temperature rise is a major contributor to comfort, a well distributed pattern (high pattern efficiency) is desired to expose as high as a percentage of floor area as possible to direct radiant energy. (See Page 90, Figure 38, Page 81, Section 7.1 for additional information).

It is evident from Page 55, Figure 22 that a well distributed energy pattern can satisfy the requirement for maximum comfort more effectively than a large output (from a single burner) energy pattern.

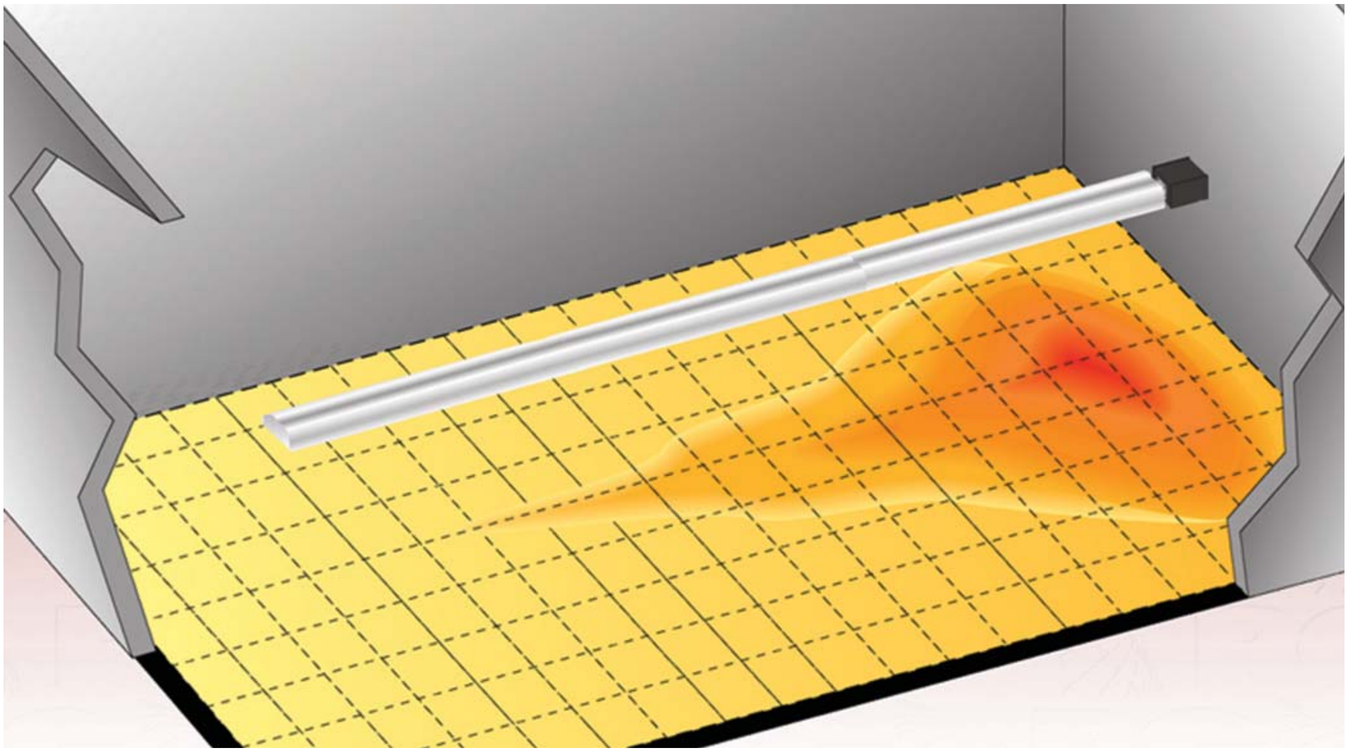
Intensity Variation with Width

Figure 21

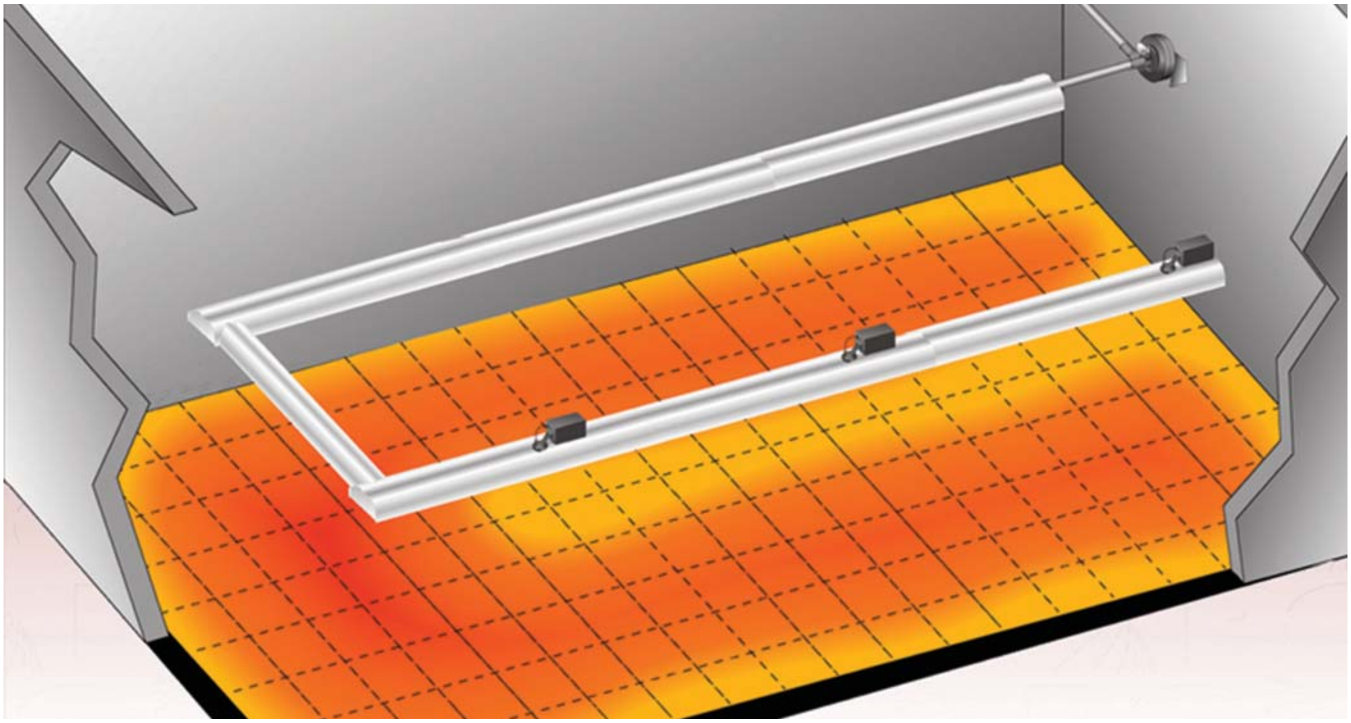


Coverage Comparison
(Equal BTU Input and Constant Mounting Height)

Figure 22



Single Burner, Concentrated Pattern, Low Pattern Efficiency



Multiple Burner, Even Pattern, High Pattern Efficiency

4.2 Radiant Adjustment to Heat Loss

The practice of applying an adjustment factor to heat loss calculations for radiant heating systems is well known within the radiant heating industry, having been used by manufacturers for over 25 years. Recently, a number of studies have been conducted to identify the values of the adjustment factor in the range of 0.8 to 0.85 (ASHRAE Transactions Volume 93, part 1). This adjustment can be more thoroughly understood when considering the following radiant effect issues.

- Infrared energy heats objects, not the air;
- Lower ambient air temperatures reduce the amount of air infiltration;
- Less air stratification with radiant heat;
- Lower ambient air temperatures reduce the transmission heat loss across walls and roof;
- Elevated floor temperatures provide a thermal reserve capacity;
- Increased mean radiant temperature allows the occupants to perceive thermal comfort at the reduced air temperature.

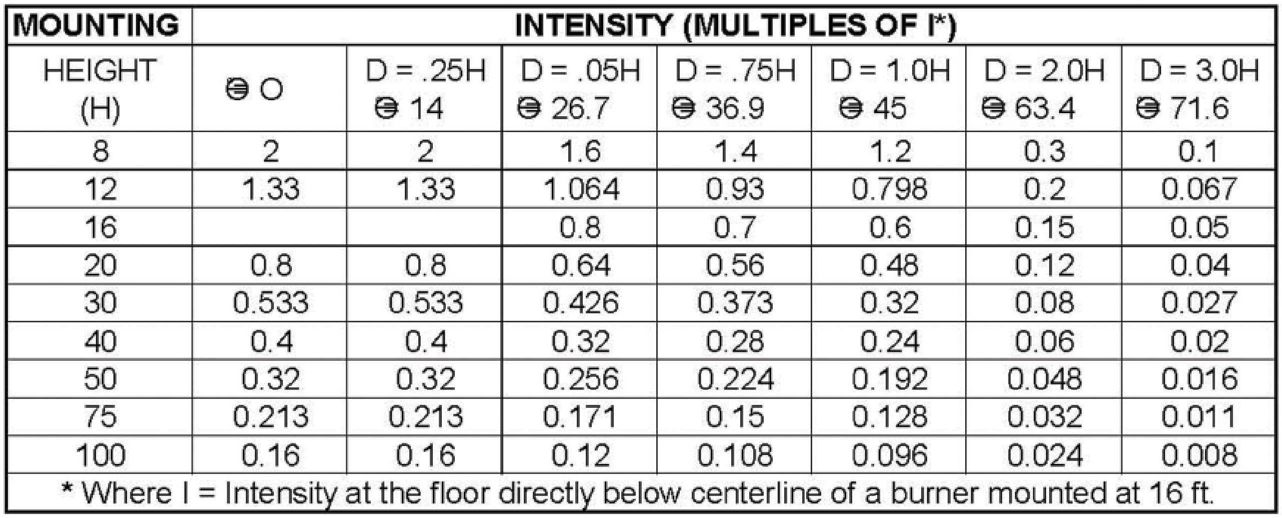
Each of these issues impact favorably on the utilization of the installed capacity of the radiant heating system. This fact, together with realization that the ASHRAE heat loss calculation methods (particularly the transmission heat loss coefficients) have been developed specifically for conventional hot air systems, demonstrates the need for the heat loss adjustment factor.

4.3 Radiant Height Adjustment Factor

As discussed above in Section 4.2, the installed capacity of radiant heating systems is typically reduced compared to the calculated heat loss due to the radiant effects associated with a properly designed radiant heating system. The ability of a radiant system to provide the advantages of these radiant effects rests largely with the ability of this system to establish a reserve heat capacity in the floor. Without this reserve capacity, radiant comfort cannot be achieved. (The exception is in station heating / spot heating applications where sufficiently high levels of direct radiation are received from the heater.) The height adjustment factor is a means to insure adequate floor level radiant intensity to “charge” the floor heat reservoir. Page 57, 59, 58, and 60 illustrate the relationship of floor level intensity to height for single and multiple (overlapping) burner runs incorporating a high fixture efficiency system.

Calculated Intensity Chart
(Single Burner Runs, High Fixture Efficiency Equipment)

Table 1

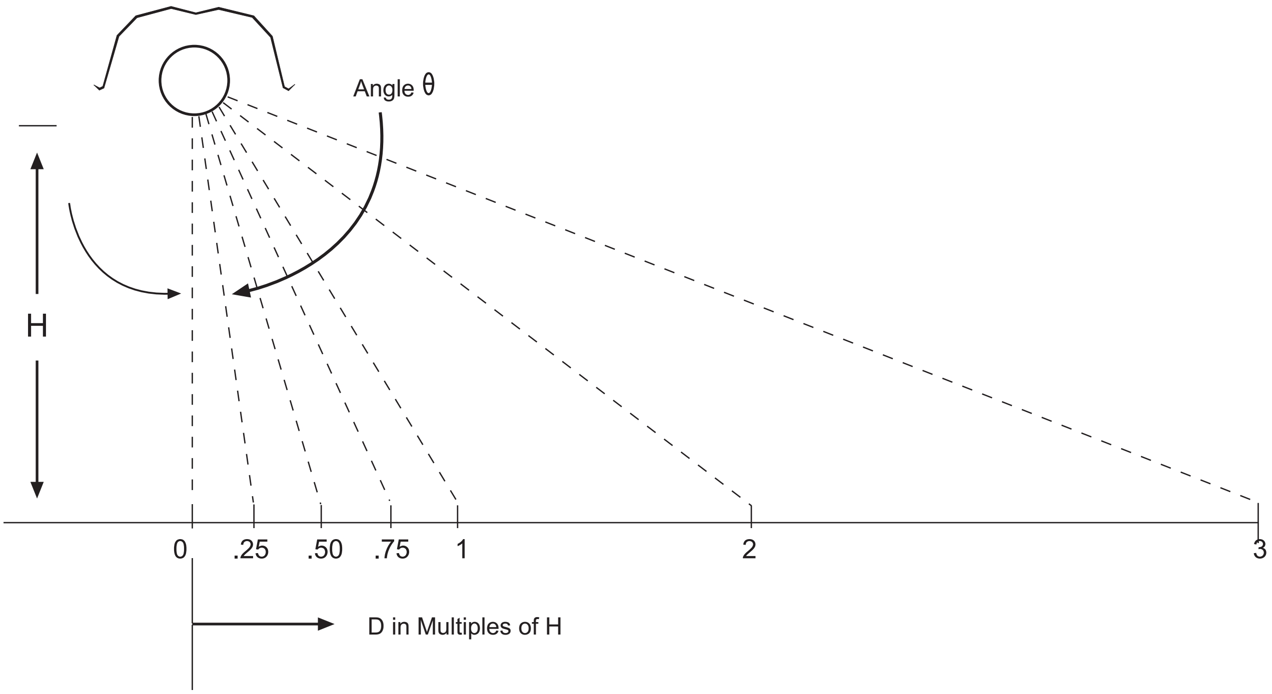


MOUNTING	INTENSITY (MULTIPLES OF I*)						
HEIGHT (H)	θ O	D = .25H θ 14	D = .05H θ 26.7	D = .75H θ 36.9	D = 1.0H θ 45	D = 2.0H θ 63.4	D = 3.0H θ 71.6
8	2	2	1.6	1.4	1.2	0.3	0.1
12	1.33	1.33	1.064	0.93	0.798	0.2	0.067
16			0.8	0.7	0.6	0.15	0.05
20	0.8	0.8	0.64	0.56	0.48	0.12	0.04
30	0.533	0.533	0.426	0.373	0.32	0.08	0.027
40	0.4	0.4	0.32	0.28	0.24	0.06	0.02
50	0.32	0.32	0.256	0.224	0.192	0.048	0.016
75	0.213	0.213	0.171	0.15	0.128	0.032	0.011
100	0.16	0.16	0.12	0.108	0.096	0.024	0.008

* Where I = Intensity at the floor directly below centerline of a burner mounted at 16 ft.

Burner Location for Intensity Charts (Single Burner Run)

Figure 23

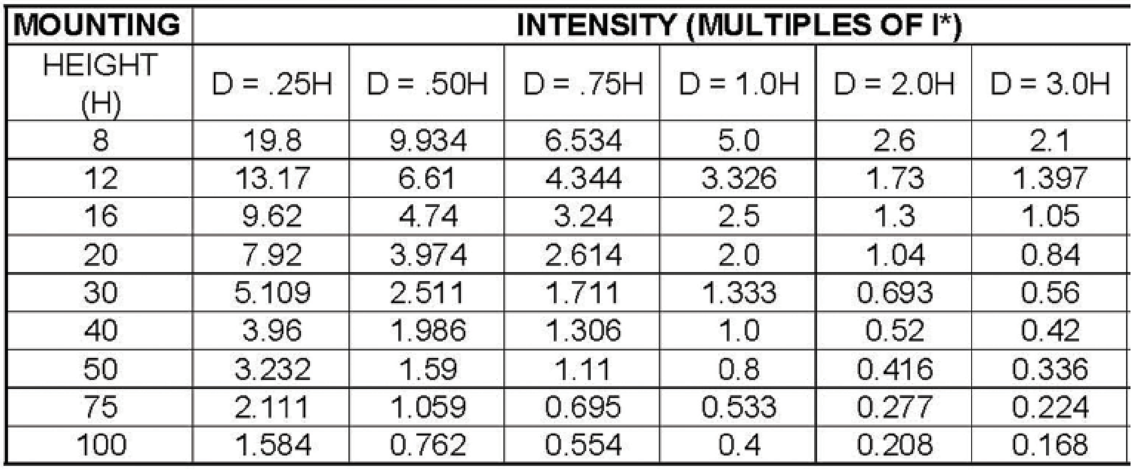


Where H = Mounting Height

D = Horizontal distance from center line of burner emitter (at floor level)

Calculated Intensity Chart
(Multiple Burner Runs, High Fixture Efficiency Equipment)

Table 2



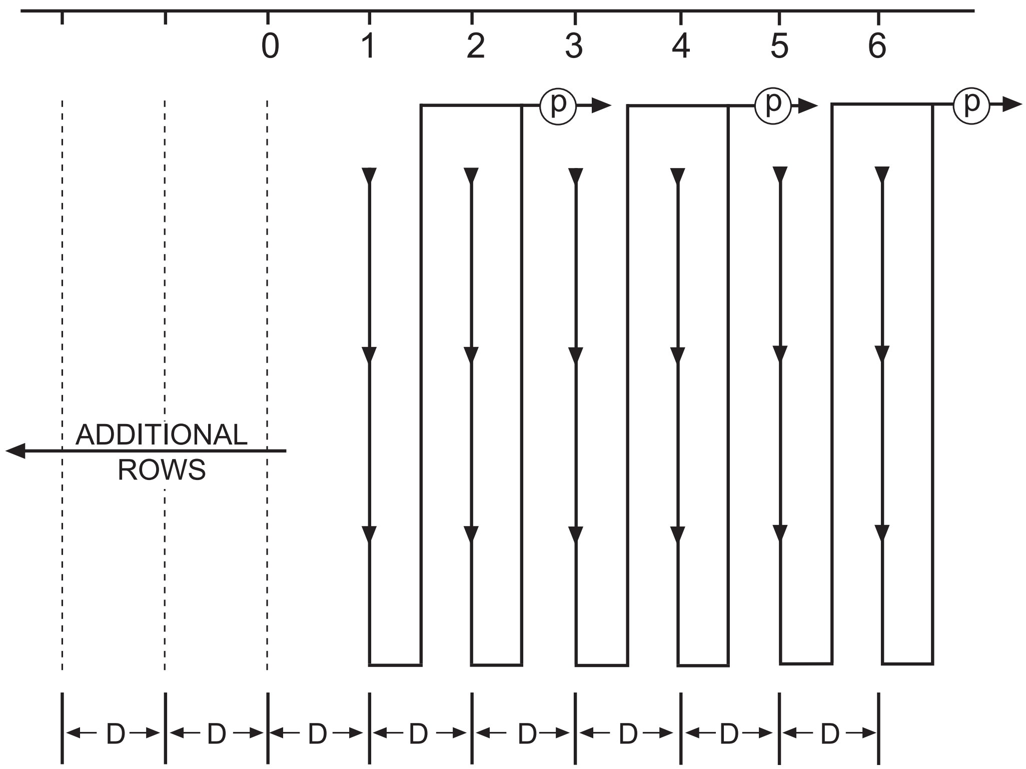
MOUNTING	INTENSITY (MULTIPLES OF I*)					
HEIGHT (H)	D = .25H	D = .50H	D = .75H	D = 1.0H	D = 2.0H	D = 3.0H
8	19.8	9.934	6.534	5.0	2.6	2.1
12	13.17	6.61	4.344	3.326	1.73	1.397
16	9.62	4.74	3.24	2.5	1.3	1.05
20	7.92	3.974	2.614	2.0	1.04	0.84
30	5.109	2.511	1.711	1.333	0.693	0.56
40	3.96	1.986	1.306	1.0	0.52	0.42
50	3.232	1.59	1.11	0.8	0.416	0.336
75	2.111	1.059	0.695	0.533	0.277	0.224
100	1.584	0.762	0.554	0.4	0.208	0.168

Where I = - Intensity at the floor directly below center line of a burner mounted at 16 ft.

- This calculation assumes 100% of the output will be allocated to a floor area as defined by a 120^{o} pattern width.
- The values shown include the overlap and are minimum values with at least two rows of burners on either side. The values of intensities are less between all other rows.
- With spacing of 3.0 H between rows, the added intensity from adjacent rows of burners is less than 5%.

Burner Layout for Intensity Chart (Multiple Burner Runs)

Figure 24



Where D = Horizontal distance between burner runs.

Additionally, higher mounting heights for radiant heating appliances increase the probability for direct radiant energy loss due to exposure of longer wall surfaces. Proportionately larger wall surfaces also remove energy from the floor to a larger degree, decreasing floor heat reservoir.

The increased input capacity recommended by a height adjustment factor is not extraneous as compared to the heat loss calculation. Rather, it is a realization that in order to maintain radiant comfort conditions (and the economic benefits) a minimum radiant level must be maintained at the floor.

It is recommended that an adjustment to the heat loss of 1% per foot for mounting heights above 20 feet be added up to 50-60 feet. Above this height, additional correction overstates the BTU requirement as determined by the heat loss.

Below are two examples of how to implement the height adjustment factor.

Example 1

- Mounting Height 30 feet
- Thermal efficiency 80%, therefore 0.85 radiant heat loss adjustment used.
- Calculated heat loss 350,000 BTU/HR.

Heat Loss	= 350,000	BTU/HR
Radiant Adjustment	x .85	BTU/HR
	297,500	BTU/HR
1%/FT Height Adjustment		
(30ft - 20ft = 10%)	x 1.1	
	327,250	BTU/HR Installed Capacity Required

Example 2

- Mounting Height 60 feet
- Thermal efficiency 90%, therefore 0.80 radiant heat loss adjustment used.
- Calculated heat loss 500,000 BTU/HR.

Heat Loss	= 500,000	BTU/HR
Radiant Adjustment	x .8	BTU/HR
	400,00	BTU/HR
1%/FT Height Adjustment		
(60ft - 20ft = 40%)	x 1.4	
	560,000	BTU/HR Installed Capacity Required

(Note in example 2, if equipment had been conventionally sized based on thermal output only, a nearly identical input requirement would result.) For mounting heights above 50-60 feet, no further correction is generally necessary provided:

- The floor level radiant intensity is sufficient to establish a reserve heating capacity (hence radiant comfort);
- The heat loss requirement is satisfied based on thermal output.

For additional information on specific application, the manufacturer should be consulted.

5.0 Radiant Heating Application Issues

Previously, on Page 52, Section 4 a number of areas relating to successful radiant heating design issues were discussed. Equally influential to the success of a radiant heating system are a number of application issues that relate the function of the radiant heating system to the application. Following are areas that must be considered when determining how effective a radiant system will be, once installed:

- Parameters for comfort.
- Design input levels, and how they relate to comfort.
- Control philosophy with radiant heating.
- Installation considerations.
- Occupancy

Each of these issues is described below in more detail.

5.1 Parameters for Comfort

Comfort is an environmental quality easily recognized when it is experienced. However, it is more difficult to define comfort in such a way that it can be used to control the thermal environment in an industry or commercial application. Comfort is associated with a neutral thermal sensation during which the human body regulates its internal temperature with a minimum of physiological effort. Relative to the radiant heating industry, perceived comfort, temperature and thermal acceptability are relative to an individual's metabolic heat production, its transfer to the environment and resulting physiological adjustments. The heat transfer is influenced by the environmental factors of:

- Air Temperature
- Thermal Radiation
- Air Movement (velocity)
- Humidity

and by the personal factors of:

- Activity
- Clothing

ASHRAE Standard ANSI / ASHRAE 55-1981 provides a comprehensive technical explanation of the interaction of these factors relative to comfort.

A less technical description is provided herein. When designing radiant heating appliance systems for comfort, consideration must be given to the following.

- Energy received by the occupant as radiation from the heating appliance.
- Energy received by the occupant from the floor (and other surfaces, if applicable due to re-radiation.
- Ambient air temperature.

Note that surfaces within a room (walls and roof) can substantially reduce comfort by “removing” radiant energy from occupants.

Mean radiant temperatures (MRT) is a concept used to describe the net amount of radiant energy available to provide comfort. MRT is defined as the uniform surface temperature of a black body enclosure in which an occupant would exchange the same amount of radiant heat as in the actual nonuniform space. Very roughly, MRT can be indicated by an average of the floor, walls and ceiling temperatures. It is for this reason that sufficiently warmed floors with reserve heating capacity are required if a comfort condition is to be realized in application of radiant heating.

The ANSI/ASHRAE Standard 55-1981 defines another quantity, operative temperature, to define comfort. For low air velocities (below 80 FPM), operative temperature is described as the average of air temperature (+a) and MRT as follows:

$$to = \frac{ta + MRT}{2}$$

Where to = Operative temperature
ta = Air Temperature
MRT = Mean Radiant Temperature

The comfort chart (Ref. ANSI Standard 55-1981, Figure 2) indicates comfort ranges for operative temperatures:

- In Summer from 73^{o} F to 81^{o} F
- In Winter from 68^{o} F to 71^{o} F

Note these ranges are for varying conditions of humidity.

Generally speaking, an operative temperature of approximately 70^{o} F is considered comfortable during heating season. Variation to this can occur depending on clothing and activity level.

With radiant heating properly applied, floor temperatures can be elevated as much as 10^{o} F to 15^{o} F as compared to floor temperatures with conventional air heating. By re-radiation of energy from the floor, the corresponding walls and inside roof surface temperatures are also increased.

From this consideration of higher temperatures of inside building surfaces as obtained with radiant heating it is apparent why the comfort level can be maintained with a lower air temperature with radiant heating.

Table 3, below, has been prepared to show the Air Temperature (+a) required under various conditions to achieve comfort at (+o) of 70° F.

TABLE 3

Heating Method	Inside Surface Temp. MRT	Air Temp. +a	Operative Temp +o
Convective Air Heating	65° F	75° F	70° F
Radiant Heating	75° F	65° F	70° F

Section 5.2, below, discusses typical installed capacities to produce comfort in various applications.

5.2 Design Input Levels

Typical design input levels vary according to the type of heating that is required. Basic types of utilization are:

a) General space heat, including perimeter heating;
b) Area heating;
c) Spot or station heating;
d) Process heating.

As the installed capacity increases to match the desired utilization level ("a" through "d" above) comfort becomes increasingly dependent on direct radiant output from the heating appliance.

Page 66, Figure 25 lists typical installed capacities for various utilization categories. These are tabulated for typical values in varying climatic zones. For example, in an area with 6000 degree-days and typical design temperature rise (80° F for this), an installed capacity from 25 to 50 BTU per Ft^2 per Hr would provide adequate comfort for general space heating.

For spot (station) or area heating, Page 67, Figure 26 indicates BTU output levels required for a given coverage as a function or required comfort temperature rise. The required input level may be calculated as follows:

$$\frac{\text{Output Level}}{\text{Radiant Efficiency}} = \text{Input Level}$$

Page 68, Table 4 lists input levels for spot or area heating as a function of air temperature, air velocity and activity level.

Note that for heating areas or spot stations, it is essential that the radiant heating appliances provide energy from at least two (preferably three) directions, none of which are directly overhead.

5.3 Control Philosophy

Night setback is a well known, accepted method of conserving fuel in the heating industry. Because radiant heating does not utilize the same mechanisms as convective air heat to provide comfort in a space, the concept of night setback must be treated differently.

Because of the importance in maintaining a floor heat reservoir to provide comfort with radiant heating, night setback must be accomplished is such a manner so as not to deplete the floor heat "charge."

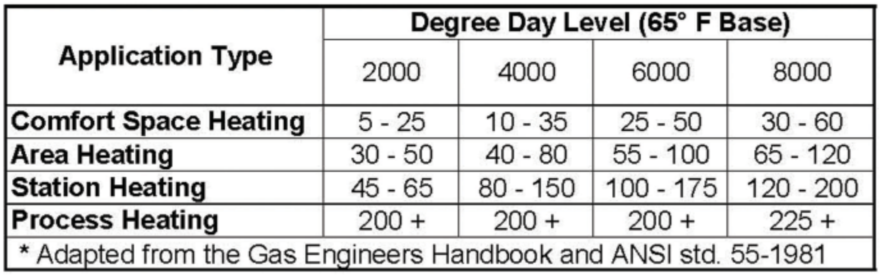
When determining the setback parameters, consideration must be given as to the degree direct radiant output from a radiant heating appliance provides space comfort. As a space increases its dependence on direct radiant output for comfort (as in spot heating), setback can be increased. In buildings that do not have large levels of floor intensity (as in high bay buildings), setback should be minimized.

Below is a summary guide for setback parameters for various applications.

Application	Recommended Setback °F
• High Bay Areas	Very little to none
• Average Construction Commercial Buildings	Up to 5° -7° F
• New Construction Commercial Building	Up to 8° - 10° F

Typical Range of Installed Capacity Requirements (BTU/Ft^2 - Hr) for Various Application Types*

Figure 25



Application Type	Degree Day Level (65° F Base)			
	2000	4000	6000	8000
Comfort Space Heating	5 - 25	10 - 35	25 - 50	30 - 60
Area Heating	30 - 50	40 - 80	55 - 100	65 - 120
Station Heating	45 - 65	80 - 150	100 - 175	120 - 200
Process Heating	200 +	200 +	200 +	225 +

* Adapted from the Gas Engineers Handbook and ANSI std. 55-1981

Assumes	-	Typical Design Temperatures rise for each Degree Day Level.
	-	Radiant Efficiency range 35 - 60%

Output Levels for Comfort in Spot or Area Heating

Figure 26

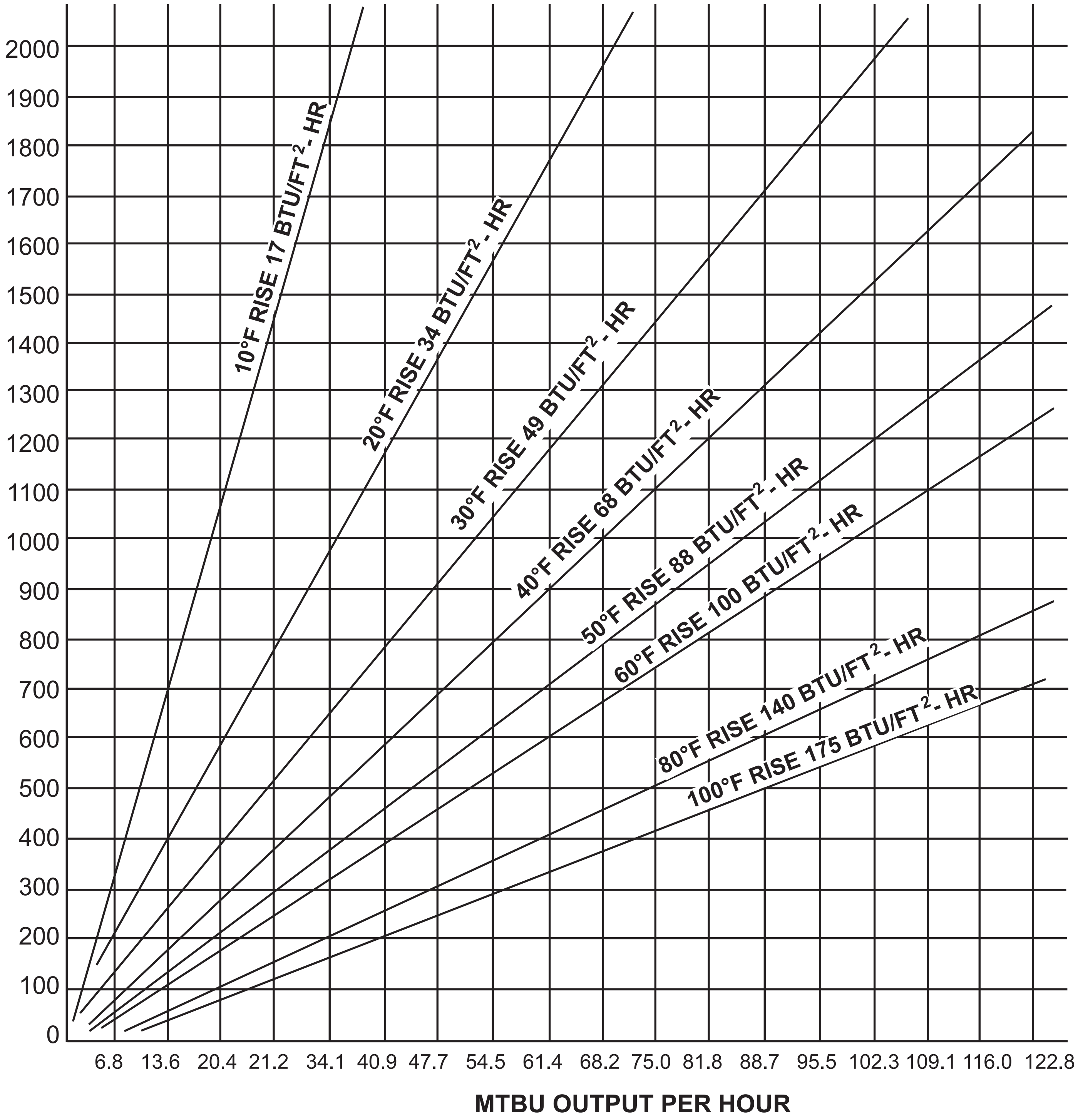
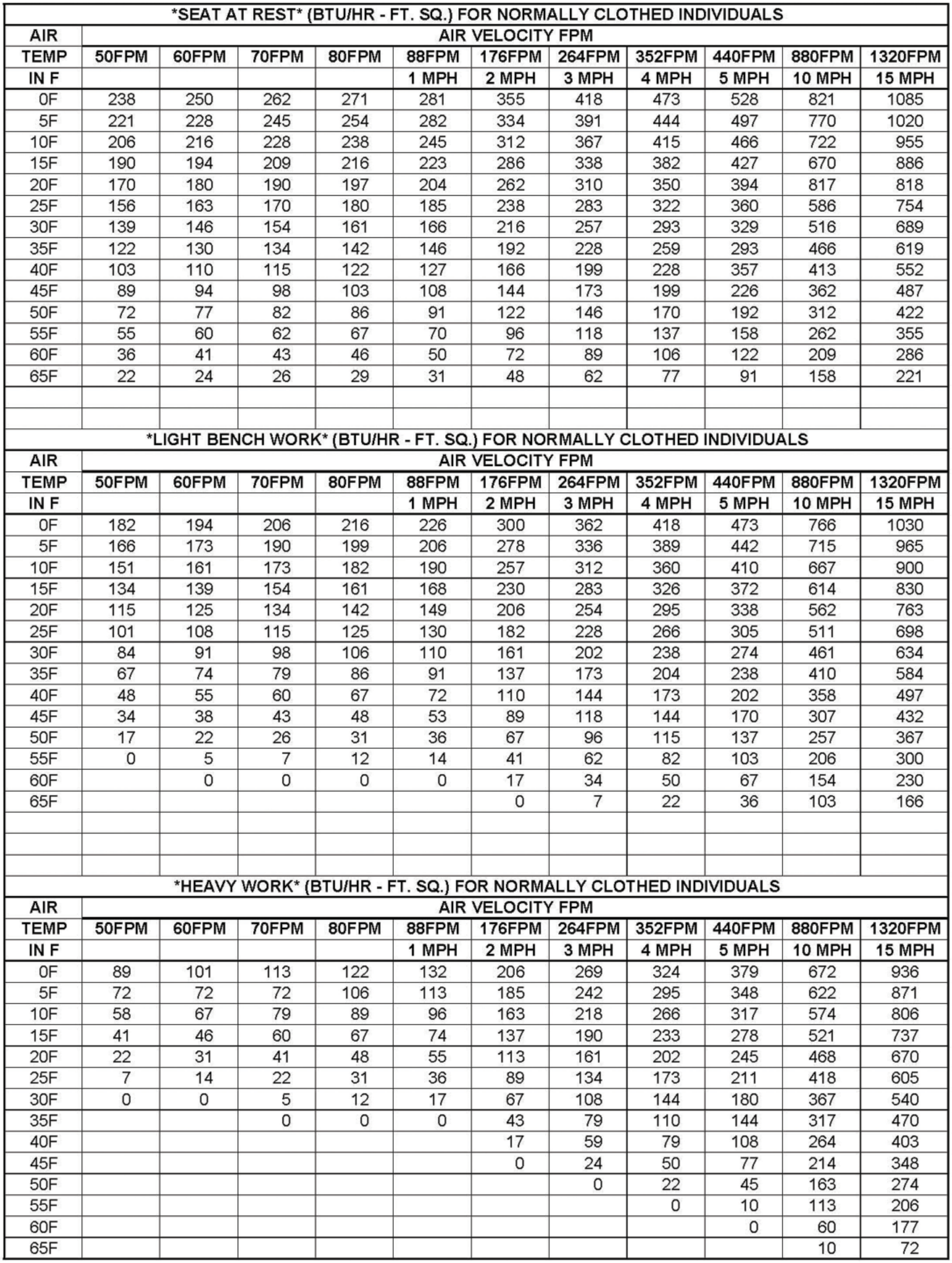


Table 4



SEAT AT REST (BTU/HR - FT. SQ.) FOR NORMALLY CLOTHED INDIVIDUALS											
AIR	AIR VELOCITY FPM										
TEMP	50FPM	60FPM	70FPM	80FPM	88FPM	176FPM	264FPM	352FPM	440FPM	880FPM	1320FPM
IN F					1 MPH	2 MPH	3 MPH	4 MPH	5 MPH	10 MPH	15 MPH
0F	238	250	262	271	281	355	418	473	528	821	1085
5F	221	228	245	254	282	334	391	444	497	770	1020
10F	206	216	228	238	245	312	367	415	466	722	955
15F	190	194	209	216	223	286	338	382	427	670	886
20F	170	180	190	197	204	262	310	350	394	817	818
25F	156	163	170	180	185	238	283	322	360	586	754
30F	139	146	154	161	166	216	257	293	329	516	689
35F	122	130	134	142	146	192	228	259	293	466	619
40F	103	110	115	122	127	166	199	228	357	413	552
45F	89	94	98	103	108	144	173	199	226	362	487
50F	72	77	82	86	91	122	146	170	192	312	422
55F	55	60	62	67	70	96	118	137	158	262	355
60F	36	41	43	46	50	72	89	106	122	209	286
65F	22	24	26	29	31	48	62	77	91	158	221

LIGHT BENCH WORK (BTU/HR - FT. SQ.) FOR NORMALLY CLOTHED INDIVIDUALS											
AIR	AIR VELOCITY FPM										
TEMP	50FPM	60FPM	70FPM	80FPM	88FPM	176FPM	264FPM	352FPM	440FPM	880FPM	1320FPM
IN F					1 MPH	2 MPH	3 MPH	4 MPH	5 MPH	10 MPH	15 MPH
0F	182	194	206	216	226	300	362	418	473	766	1030
5F	166	173	190	199	206	278	336	389	442	715	965
10F	151	161	173	182	190	257	312	360	410	667	900
15F	134	139	154	161	168	230	283	326	372	614	830
20F	115	125	134	142	149	206	254	295	338	562	763
25F	101	108	115	125	130	182	228	266	305	511	698
30F	84	91	98	106	110	161	202	238	274	461	634
35F	67	74	79	86	91	137	173	204	238	410	584
40F	48	55	60	67	72	110	144	173	202	358	497
45F	34	38	43	48	53	89	118	144	170	307	432
50F	17	22	26	31	36	67	96	115	137	257	367
55F	0	5	7	12	14	41	62	82	103	206	300
60F		0	0	0	0	17	34	50	67	154	230
65F						0	7	22	36	103	166

HEAVY WORK (BTU/HR - FT. SQ.) FOR NORMALLY CLOTHED INDIVIDUALS											
AIR	AIR VELOCITY FPM										
TEMP	50FPM	60FPM	70FPM	80FPM	88FPM	176FPM	264FPM	352FPM	440FPM	880FPM	1320FPM
IN F					1 MPH	2 MPH	3 MPH	4 MPH	5 MPH	10 MPH	15 MPH
0F	89	101	113	122	132	206	269	324	379	672	936
5F	72	72	72	106	113	185	242	295	348	622	871
10F	58	67	79	89	96	163	218	266	317	574	806
15F	41	46	60	67	74	137	190	233	278	521	737
20F	22	31	41	48	55	113	161	202	245	468	670
25F	7	14	22	31	36	89	134	173	211	418	605
30F	0	0	5	12	17	67	108	144	180	367	540
35F			0	0	0	43	79	110	144	317	470
40F						17	59	79	108	264	403
45F						0	24	50	77	214	348
50F							0	22	45	163	274
55F								0	10	113	206
60F									0	60	177
65F										10	72

5.4 Installation Considerations

Prior to the design, the thought process should focus on extraordinary factors possibly existing in the building's usage. Is there potential hazardous or corrosive vapors present? Is there combustible merchandise being stored up or near the roof line? Is it feasible to recess the radiant heating system in a drop ceiling for more suitable aesthetics? Is it practical to install radiant heating in a public forum where codes require high levels of ventilation?

Hazardous Environment:

Referencing national and local fire codes can quickly determine the suitability of individual radiant heating systems. In buildings where occasional vapors may be present, it is prudent to consider application of fresh air supply to create an independently enclosed system, which does not ingest building air. Use of cooler outside air can decrease thermal efficiency slightly and should be considered only when necessary.

Corrosive Environment:

Occasionally such products as cleaners, preservatives, refrigerators, etc. can contain corrosive contaminants which are harmful to any heating system. Just traces of fluore/chloro hydrocarbons in the building atmosphere can greatly accelerate the rate of corrosion on heat exchanger surfaces. To protect against such deterioration, a fresh air supply is recommended.

Safety:

With any heating source, consideration must be given to performance related items, such as the clearances to combustible materials defined by all laboratories during the appliance's certification process. Attention should be focused on such situations as stacking of combustible products, penetration of combustible walls, and proximity of materials to the radiant fixture. In most cases, a solution is available by application of uniquely designed side reflector, universal shield, triple wall thimble or other specialty items. The thought process should include sensitive consideration of configuration and location of the radiant tube heaters during the design phase.

Aesthetics:

Often radiant heating is preferred in large expanses of low ceiling areas in decorative form such as showrooms, stores, theatres, bowling alleys and classrooms. With the advent of low rated output radiant burners, 2' x 4' metal ceiling mesh, and side reflector, these applications have received wide acclaim as safe, efficient, and cost effective.

Ventilation:

Such public facilities as civic centers, shopping mall commons, theatres and gymnasiums are fundamentally sound applications for radiant heating systems because of the high efficiency yield and even distribution of comfort. Even as codes require a high volume of air change, the application of radiant heating with roof-top makeup air units is considered more functional when comparing cost of operation. Often public occupancy time represents less than 20% of the day, leaving 80% duty to the more cost efficiency radiant system. Further, radiant systems can typically be sized to handle up to three air change volumes each hour, requiring less heat makeup with the coded air volume.

5.5 Occupancy

Of major importance during the design stages of a radiant heating system is a consideration for the occupancy of the building in question. As discussed, personal factors such as activity and clothing level can greatly influence the perception of comfort for the occupants. Occasionally, a particular sensitivity to comfort can be identified for intended occupants of a structure, and these specialized needs must be addressed to insure a satisfactory application. Examples of these types of sensitive installations are:

- Zoos
- Kennels
- Residences for the elderly

Fitting the design concept to the requirement of the building and its components can insure a successful application.

6.0 Fuel Utilization

Radiant heating has long been recognized as a cost effective method of providing heat. Determining fuel utilization characteristics for radiant heating is possible utilizing accepted ASHRAE methods. Because of this, fuel usage comparisons to conventional heating equipment can also be made.

Section 6.1 derives the necessary equations to calculate fuel usage. Because this calculation is applied to a design (maximum) heating load, and is subject to errors from a number of other sources, namely:

1. Difference between historical and actual degree day numbers,
2. No consideration for the effects of internal heat gains (from lights, etc.),
3. Differences between estimated and actual air change or ventilation heat loss,

The estimated annual fuel usage calculated is usually conservative. ASHRAE has recently implemented a correction coefficient C_D, based on degree-days, to provide for more realistic fuel use estimates.

It is necessary to recognize the range of values that can exist for the fuel utilization factors, depending on application circumstances. As an application increases it’s dependency on radiant energy to provide comfort, a correspondingly larger fuel savings can be expected over conventional heating equipment.

Page 77, Section 6.2 gives a comparative analysis for various types of heating equipment and includes consideration for operating costs as well as fuel costs.

6.1 Formula Derivation

1. Formula Definition Roberts-Gordon (RG) vs. ASHRAE

(Reference ASHRAE 1985 Fundamentals pp. 28.2 - 28.3)

A) (RG) Formula

$$F = \frac{U^1 \times N_b \times D\ 70^O}{1000 \times DTR} \quad (C_D) \qquad \text{(EQ 1)}$$

Where F = Estimated Annual Fuel Consumption (units same as U^1)

U^1 = Factor for Fuel unit usage per Unit Heat Loss N_u

N_b = Calculated Heat Loss at design temperatures

D = Number of Degree Days Annual

DTR = Design Temperature Rise

C_D = Empirical corrections factor for heating effect vs 65^O F degree days, usually less than 1.0 and based on actual degree day (from Page 75, Figure 27)

U1 Factor Definition Formula:

$$U^1 = \frac{N_u \times C\ 24}{E \times 70^O \times V} \qquad \text{(EQ 2)}$$

Where N_u = Unit heat loss of 1000 BTU/HR with temperature rise of 70^O

C = Experienced base correction factor based on reduction of ASHRAE calculated heat loss due to radiant effects.

E = Thermal Efficiency

V = Heating value of fuel used to define F in terms of fuel units rather than BTU

Substituting N_u into EQ 2

$$U^1 = \frac{1000\ BTU/HR \times C \times 24}{E \times 70^O \times V} \qquad \text{(EQ 2A)}$$

When EQ 1 and EQ 2A are combined:

$$F = \frac{\frac{1000 \times C \times 24}{E \times 70^O \times V} \times N_b \times D\ 70^O}{1000 \times DTR} \quad (C_D)$$

Simplifying:

$$F = \frac{C \times 24 \times N_b \times D}{E \times DTR \times V} \quad (C_D) \qquad \text{(EQ 3)}$$

B) Compare to ASHRAE Formula (pp, 28.2 - 28.3 Fundamental Handbook 1985):

$$F = \frac{HL \times D \times 24}{t \times K \times \Delta V} \quad (C_D) \qquad \text{(EQ 4)}$$

Where E = Fuel or energy consumption (= F in RG terminology)

H_L = Calculated Heat Loss (= N_b in RG terminology)

D = Degree Days (= D in RG terminology)

t = ΔDesign temperature rise (= DTR in RG terminology)

K = A correction factor that includes effect of efficiency and other load performance factors (= E/C in R-G terminology)

C_D = Empirical correction factor for heating effect vs 65° F degree days usually less than 1.0 and based on actual degree day

V = Heating value of fuel used to define E in terms of fuel units rather than BTU

Substituting RG terminology into EQ 4:

$$F = \frac{N_b \times D \times 24}{DTR \times E/C \times V} \quad (C_D) \qquad \text{(EQ 5)}$$

Rearranging EQ 5:

$$F = \frac{C \times 24 \times N_b \times D}{E \times DTR \times V} \quad (C_D) \qquad \text{(EQ 6)}$$

Note that EQ 3 and EQ 6 are identical. Values of (C_D) can be read from Page 75, Figure 27 at the appropriate degree day value.

2. U^1 Factors Calculation (Using EQ 2)

A) CRV - B Series

Where C = 0.7, E = 0.90, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times .7 \times 24}{0.90 \times 70\ 1000} = 0.267 \qquad \frac{CF}{1000\ BTU/HR\ Heat\ Loss}$$

B) CRV - E Series

Where C = 0.7, E = 0.82, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times .7 \times 24}{0.82 \times 70\ 1000} = 0.293 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

C) VANTAGE

Where C = 0.7, E = 0.78, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times .7 \times 24}{0.78 \times 70\ 1000} = 0.307 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

D) UNIT HEATERS

Where C = 1.0, E = 0.75, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times 1.0 \times 24}{0.75 \times 70\ 1000} = 0.457 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

E) DIRECT FIRED MAKE-UP AIR

Where C = 1.06, E = 0.92, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times 1.06 \times 24}{0.92 \times 70\ 1000} = 0.395 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

(NOTE: C & E Factor for direct fired make-up air are defined for average winter temperature of 55° F and 8% nonrecoverable latent heat due to inability to condense all combustion flue products. This is for purposes of the example to follow).

F) AIR ROTATION

Where C = 0.9, E = 0.80, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times .9 \times 24}{0.80 \times 70\ 1000} = 0.386 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

G) HIGH INTENSITY

Where C = 0.85, E = 0.92, V = 1000 for Natural Gas.

$$U^1 = \frac{1000 \times .85 \times 24}{0.92 \times 70\ 1000} = 0.317 \qquad \frac{\text{CF}}{\text{1000 BTU/HR Heat Loss}}$$

NOTE: Additional U^1 factors for other equipment and fuel are tabulated on Page 76, Figure 28.

Correction Factor C_D vs. Degree Days

Figure 27

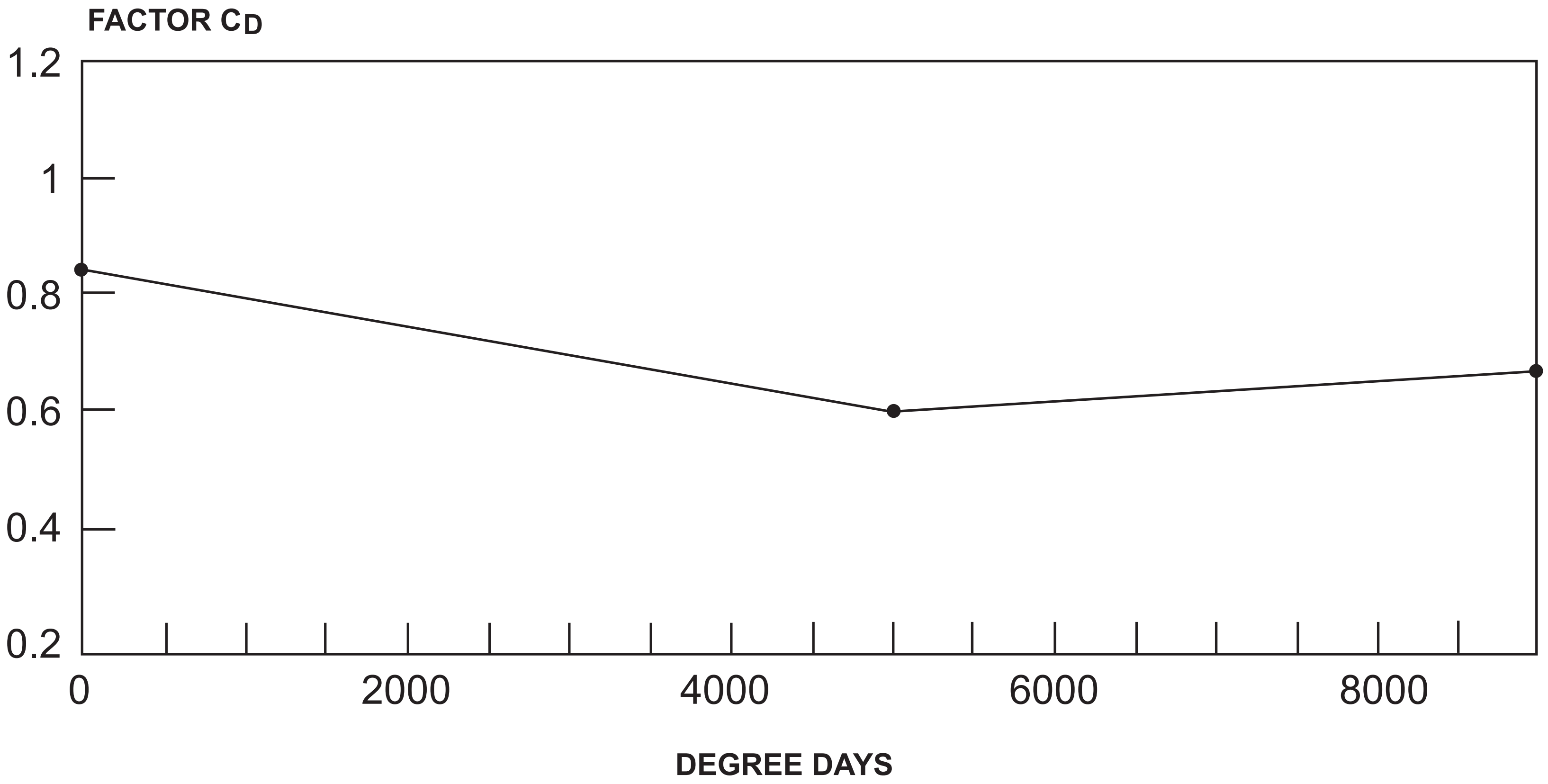


Figure 28

Fuel Utilization (U^1) Factors

Equipment	Fuel/Units	Correction Factor 1/K (= C/E)	U^1 Factor Per Degree Day
CORAYVAC® B/A	Natural Gas/CF	.778	.267 CF
CORAYVAC® E	Natural Gas/CF	.854	.293 CF
Vantage	Natural Gas/CF	.897	.307 CF
Gordon-Ray	Natural Gas/CF	.897	.307 CF
High Intensity	Natural Gas/CF	.924	.317 CF
Unit Heater	Natural Gas/CF	1.334	.457 CF
Direct Fired Make-Up Air: (Note 1)			
Outdoor Temp. 0° F	Natural Gas/CF	1.609	.555 CF
Outdoor Temp. 20° F	Natural Gas/CF	1.424	.487 CF
Outdoor Temp. 55° F	Natural Gas/CF	1.152	.395 CF
Air Rotation	Natural Gas/CF	1.125	.386 CF
Central Furnace	Natural Gas/CF	1.429	.490 CF
Boiler	Natural Gas/CF	1.538	.527 CF
CORAYVAC® B/A	LP/Gallons	.778	.00300 GAL.
CORAYVAC® E	LP/Gallons	.854	.00320 GAL.
Vantage	LP/Gallons	.897	.00336 GAL.
Gordon-Ray	LP/Gallons	.897	.00336 GAL.
High Intensity	LP/Gallons	.924	.00345 GAL.
Unit Heater	LP/Gallons	1.334	.00499 GAL.
Air Rotation	LP/Gallons	1.125	.00433 GAL.
Boiler	LP/Gallons	1.538	.00575 GAL.
Unit Heater	Oil/Gallons	1.429	.00350 GAL.
Central Furnace	Oil/Gallons	1.538	.00376 GAL.
Boiler	Oil/Gallons	1.667	.00407 GAL.

Assumptions:	Natural Gas	1,000	BTU/CF
	L.P.	91,500	BTU/GAL
	#2 Oil	140,000	BTU/GAL
	Electricity	3,413	BTU/KWH

NOTE 1: The U^1 factor is based on useful heat available after heating the required ventilation rate of 4 CFM per 1000 BTU/HR of input. If this level of ventilation U^1 Factor for Natural Gas .373 and the outside air temperature is not a consideration.

6.2 Illustrative Example

1. Operating Expenses Include Both
    - a) Fuel Usage Costs
    - b) Electric Usage and Demand Costs

A) Fuel Usage Costs

Fuel usage can be determined via (EQ 1) by utilizing U1 factors as derived above.(Page 76, Figure 28), As previously stated, (EQ 1) is interchangeable with accepted ASHRAE procedures.

Fuel usage costs are determined by the following:

$$CF = F \times A \qquad \text{(EQ 7)}$$

Where CF is total annual fuel cost

F is estimated annual fuel consumption

A is cost per fuel unit (units consistent with F)

B) Electric Usage Costs (usage and demand)

To determine electric usage costs, an estimate of equipment running time must be made. This can be accomplished as follows:

$$TR = \frac{DD \times 24}{DHR} \qquad \text{(EQ 8)}$$

Where TR = Estimated annual equipment running time

DD = Degree Days

DHR = Estimated average degree rise hours per hour (determined by operating conduction)

Example:

For a typical warehouse:

8 hour (day shift)

1st hour: 10^{o} rise (from night setback to day set point) = 10^{o} - HRS.

2nd - 8th hours: 5^{o} rise x 4 cycles per hour = 5 x 4 x 7 = 140^{o} - HRS.

16 hour (night shift)

1st - 2nd hours: no equipment run time (building cools)

3rd - 16th hours: 5^{o} rise x 4 cycles per hour = 5 x 4 x 14 = 280^{o} - HRS.

$$\text{Average degree hours} = \frac{10 + 140 + 280}{24}$$

$$= 18 \text{ degree - hours/hour}$$

Using (EQ 8), determine TR for Dalias, TX: (to conform to example below)

$$TR = \frac{2407 \times 24}{18} = 3209 \text{ HRS.}$$ Estimated annual equipment running time.

Electric usage and demand costs are determined by:

$$CE = (L \times TR \times KWC) + KWD \qquad \text{(EQ 9)}$$

Where CE = Total annual electric cost

L = Equipment connected electric load

TR = Estimated annual equipment running time

KWC = Cost per electric unit (units consistent with L)

KWD = Demand charge per electric unit (units consistent with L)

2. Comparative Analysis for Illustrative Purposes

Assumption for analysis:

Location: Dallas, Texas

Degree Days: 2407

Design Temperature Rise: 48^{o} F (68O F inside, 20^{o} F outside)

Transmission Loss Factors:

Walls U = .65

Roof U = .35

Doors U = .65

Room Size: 400ft x 250ft x 22ft ceiling

Infiltration Rate: 1/3 ACPH (or approximately 12,100 CFM)

C_D = 0.76 (For 2407 Degree Days - From ASHRAE Chart, Page 75, Figure 27

HEAT LOSS CALCULATED PER ASHRAE PROCEDURE (See Page 79, Figure 29)

Cost of Natural Gas = \$0.40 per therm

Cost of Electric Usage = \$0.05 per KWH

Using equations and factors previously derived as noted, comparative analysis is tabulated on Page 80, Figure 30.

Job: “Sample Warehouse Cost Study”

Figure 29

LENGTH 400 WIDTH 250 HEIGHT 22 INSIDE TEMP 68 OUTSIDE TEMP 20 DESIGN TEMPRISE 48

			Btu/HR
INFILTRATION:	NUMBER OF AIR CHANGES .33 * VOLUME * DTR	=	627,264
MECH. EXHAUST:	CFM * 1.08 = 0 * DTR	=	0
FLOOR LOSS:	PERIMETER * FACTOR (INSULATED = 0.55, UNINSULATED = 0.81) * DTR	=	50,544
	INFILTRATION PLUS FLOOR LOSS	=	677,808

SURFACE	QUANTITY *	WIDTH *	LENGTH =	AREA *	“U” =		* DTR =	
ROOF	1	250	400	100,000	0.350	35000.0	48	1,680,000
SIDE 1	1	22	400	8,800				
LESS DOORS				1,000	0.650	650.0	48	31,200
LESS WINDOWS				0	1.000	0.0	48	0
NET SIDE 1				7,800	0.650	5070.0	48	243,360
SIDE 2	1	22	400	8,800				
LESS DOORS				0	0.650	0.0	48	0
LESS WINDOWS				0	1.000	0.0	48	0
NET SIDE 2				8,800	0.650	5750.0	48	274,560
SIDE 3	1	22	250	5,500				
LESS DOORS				0	0.650	0.0	48	0
LESS WINDOWS				0	1.000	0.0	48	0
NET SIDE 3				5,500	0.650	3575.0	48	171,600
SIDE 4	1	22	250	5,500				
LESS DOORS				0	0.650	0.0	48	0
LESS WINDOWS				0	1.000	0.0	48	0
NET SIDE 4				5,500	0.650	3575.0	48	171,600

Transmission Heat Loss	2,572,320 Btu/Hr
Total Calculated Heat Loss	3,250,128 Btu/Hr

Comparative Equipment Analysis
For 100,000 Sq. Ft. Warehouse in Dallas, TX.

Figure 30

EQUIPMENT	FUEL USE FACTOR U' NOTE: 1	ESTIMATED ANNUAL FUEL USAGE OF NOTE: 2	ANNUAL FUEL USAGE COST $ NOTE: 3	ANNUAL ELECTRIC USAGE KWH NOTE: 4	CONNECTED LOAD KW	ANNUAL ELECTRIC COST - $ NOTE: 5	TOTAL TOTAL OPERATING COST NOTE: 6	% MORE THAN CRV B	OPERATING COST PER SQ. FT. NOTE: 7
CORAYVAC® B	0.267	2,315,038	9,260.15	13,157.00	4.1	657.85	9,918.000	---	0.099
CORAYVAC® E	0.293	2,540,473	10,161.89	18,599.00	5.8	929.95	11,091.840	11.8	0.11
VANTAGE®	0.307	2,661,860	10,647.44	22,142.00	6.9	1,107.11	11,754.550	18.5	0.117
UNIT HEATERS	0.457	3,962,444	15,849.78	25,030.00	7.8	1,251.51	17,101.280	72.5	0.171
DIRECT FIRED MAKEUP AIR	0.395	3,424,870	13,699.48	47,878.00	14.9	2,393.91	16,093.390	62.3	0.16
AIR ROTATION	0.386	3,346,835	13,387.34	43,090.00	13.4	2,154.52	15,541.860	56.7	0.155
HIGH INTENSITY	0.317	2,748,566	10,994.26	5,776.00	4.0 NOTE: 8	641.80	11,636.060	17.3	0.116

NOTES:

1. Factors as calculated in Section II units are CF/1000 BTU/HR HEAT LOSS
2. Calculated per equation 1
3. Calculated per equation 7, using stated assumptions
4. Calculated per equation 10, using stated assumptions for heating season only
5. Calculated per equation 7, using stated assumptions; Note: For illustration purpose demand charge, KWD, was assumed zero.
6. Total operating cost = Annual fuel usage cost plus annual electric cost. (Does not include electric demand charges.)
7. Based on typical warehouse example @ 100.000 Ft^2
8. Includes electric load for required ventilation of CFM per 1000 BTU/HR input.

7.0 Appliance Approach to Market Place

7.1 Characteristics for Performance

Ever since the late 1950's, radiant heating has been synonymous with fuel efficiency and uniform comfort in commercial, industrial, and agricultural applications.

The radiant heater which pioneered the transition from warm air heat to "heat on the feet" comfort, provided uniform distribution of radiant energy onto the total floor area by means of multiple burners in series. Its high efficiency was the result of its better emitter surface averaging 1.0 square feet per thousand BTU input, operated under vacuum in a condensing mode. CORAYVAC® by Roberts Gordon, LLC
remains the most recognized radiant heater in the ever expanding market place, and has set the standards for uniform comfort, operational efficiency, longevity, and safety.

Through the years, and mainly for budgetary considerations, many different types of radiant tube heaters have emerged to provide broad choices in cost, efficiency and radiant distribution, See Page 83, Figure 31.

In tier one, there remains only one high efficiency radiant heater with multiple burners in series that operates in the condensing mode, and is rated at over 90% efficiency (Type 1(b)), See Page 84, Figure 32.

In the second tier, several radiant heaters are available in the 80% efficiency range. Operated in the dry, non-condensing mode, they usually are conFigured with multiple end burners connected to a power exhauster, and provide an emitter surface from .42 to .61 square feet per thousand BTU input. (Quasi 1(b)/1(c) type).

A third tier of radiant tube heater falls within the 75% efficiency range. They are conFigured as single burners under power of draft venting, and provide emitter surface from .28 to .50 square feet per thousand BTU input. They are either factory or field assembled. (Quasi type 1(b)/1(c) or type 1(c)).

Differentiating the various types of radiant tube heaters requires understanding of products and knowledge of materials used to best accomplish its design performance, See Page 84, Figure 32. Stainless steel is an impressive material, but what is it emissive and reflective value? Understanding the grade and character of aluminized tube, has it been heat treated to effectively nearly double its emissive power? Is black paint applied to emitter tube functional if its longevity is only 12 to 18 months. Since convective heat is substantially undesirable, is the reflector contour deep and broad enough to prevent its escape to the ceiling (stratification)?

Is there sufficient angularity to the reflector top to prevent radiant energy from reradiating back onto the emitter? All these differences, and more, eventually determine whether a good or deficient fixture is present to produce and control the required radiant effect over the long haul.

Since the inception of radiant tube heat, design for most applications has concentrated radiant distribution on total floor areas without significant heat stratification between floor and ceiling, See Page 90, Figure 38. With radiant heat covering becomes a secondary source of heat by absorption at no extra cost and quickly some energy into convective heat. Additionally energy is reradiated to the objects in the space, contributing to comfort. Hence, air temperature is substantially lower for comfort that is normally associated with warm air-convection heating. This is the proven standard established by CORAYVAC®.

With the advent of other types of radiant tube heaters with much larger burner output and shorter emitter lengths, the comfort principle requiring total span distribution of radiant energy cannot be satisfied. Consequently, perimeter and area distribution designs requires attention.

With sufficient emitter length to cover only a perimeter distribution (outside walls), center areas of the building remain that must await convectional heat coverage could be achieved by the costlier means of adding more output (heaters) than the heat loss requires. Some radiant heater manufacturers provide side reflector to direct some of the perimeter radiant energy toward the center. Others suggest tilting the reflector up to 45^0 to reach the center area without considering the significant loss of convection heat to the ceiling. The resulting substantial stratification works counter to the radiant heat loss adjustment principle through increased transmission loss.

With even larger burner outputs and shorter emitter lengths, a third level of radiant distribution results in area distribution design, See Page 90, Figure 38. This distribution pattern is the least effective for people comfort since spotted areas of radiant and convective heat are intermixed creating identifiable discomfort zones, at least until convective heat transcends the entire span of the building. Greater stratification and fuel use occur in this design than any other. Larger output burners also require greater attention in design to clearance to combustibles.

In evaluating the good-better-best approach to radiant product and its resulting comfort distribution patters, certain cost factors are of vital consideration. Judgements such as longevity (life of fixture) and its resulting energy savings pay back, value of employee comfort and its measure to productivity, value engineering and its cost / benefit ratio to type and performance of equipment selected are all significant and justifiable considerations.

Figure 30

APPLIANCE SUMMARY TABLE

MFG	RATE BTU/HR				HTG SURFACE FT^2/MTBU/HR
A	120	vb	105’ x 4” OD	20’ vp OPT	.92/OPT 1.09
A	120	vb	60’ x 4” OD	vp	.52
A	125	pb	40’ x 4” OD	10’ vt OPT	.34
A	75	pb	20’ x 4” OD vt		.28
B	130	vb	60’ x 3½” OD OPT	30’ vp	.42/OPT .63
C	75	vb	58’ x (3” + 6” OD)	vp	.61
C	100	vb	35’ x 3” OD vp		.27
D	120	vb	80’ x 3½” OD	vp	.61
D	125	vb	45’ x 3½” OD	20’ vp OPT	.33/OPT .48
E	125	pb	50’ x 4” OD	vt	.42
F	125	pb	40’ x 4” OD	20’ vt OPT	.34/OPT .50
G	125	vb	50’ x 4” OD	vt	.42
H	75	pb	32’ x 3” OD vt		.34
H	100	pb	12’ x 3” OD	13’ vp OPT	.09/OPT .20
I	125	pb	140’ x 4” OD	vt	.34
J	125	pb	40’ x 4” OD	10’ vt OPT	.34/OPT .42
K	125	pb	40’ x 4” OD	20’ vt OPT	.34/OPT .50

7.2 Appliance Comparison

Figure 32

APPLIANCE COMPARISON (A)

MANUFACTURER	A	A	A
THERMAL EFFICIENCY	90%	80%	75%
EMITTER (FIRST SECTION)			
Size	20' - 50' x 4" OD	10' x 4" OD	4' x 4" OD
Material	Hot Rolled Steel	Heat Treated Aluminized Steel	Heat Treated Aluminized Steel
Gauge	16	16	16
Emissivity	.8	.8	.8
(SECOND SECTION)			
Size	65' - 110' x 4" OD	50' - 70' x 4" OD	16' x 4" OD
Material	Hot Rolled With Porcelain	Hot Rolled Steel	Hot Rolled Steel
Gauge	16	16	16
Emissivity	.9	.8	.8
REFLECTOR			
Size	5-27/32" x 14½"	5-27/32" x 14½"	5-27/32" x 14½"
Material	Finished Aluminum	Finished Aluminum	Finished Aluminum
CONFIGURATION	A	A	A
TYPE	1b	Quasi 1b/1c	1c

Figure 33

APPLIANCE COMPARISON (B)

MANUFACTURER	A	B	B
THERMAL EFFICIENCY	75%	80%	75%
EMITTER (FIRST SECTION)			
Size	10’ x 4” OD	10’ x 3½” OD	10’ x 3½” OD
Material	Aluminized	Aluminized Spiral Steel	Painted Aluminized Spiral Steel
Gauge	16	22	22
Emissivity	.8	.8	.8
(SECOND SECTION)			
Size	10’ - 50’ x 4” OD	50’ - 80’ x 3½” OD	30’ x 3½” OD
Material	Hot Rolled Steel	Aluminized Spiral Steel	Painted Aluminized Spiral Steel
Gauge	16	22	16
Emissivity	.8	.5	.5 - .8
REFLECTOR			
Size	5-27/32” x 14½”	3¾” x 12¼”	3¾” x 12¼”
Material	Finished Aluminum	Finished Aluminum	Finished Aluminum
CONFIGURATION	A	D	D
TYPE	1c	Quasi 1b/1c	1c

Figure 34

APPLIANCE COMPARISON (C)

MANUFACTURER	C		C	D
THERMAL EFFICIENCY	80%		75%	80%
EMITTER (FIRST SECTION)				
Size	7¾” x 3” OD		7¾” x 4” OD	10’ x 3½” OD
Material	Painted Stainless Steel 304		Painted Stainless Steel 304	Aluminized Steel
Gauge	16		16	16
Emissivity	.6 - .8		.5 - .8	.5
(SECOND SECTION)				
Size	17¼” x 3” OD	23’ x 3½” OD	27¼ x 4” OD	70’ x 3½” OD
Material	Mild Steel	Aluminized Spiral	Mild Steel	Hot Rolled Steel
Gauge	14	22	16	16
Emissivity	.8	.8	.8	.8
REFLECTOR				
Size	5¼” x 11½”		5¼” x 11½”	4” x 14”
Material	Stainless Steel 304 or Aluminum		Stainless Steel 304 or Aluminum	Polished Aluminum
CONFIGURATION	C		C	D
TYPE	Quasi 1b/1c		Quasi 1b/1c	Quasi 1b/1c

Figure 35

APPLIANCE COMPARISON (D)

MANUFACTURER	D	E	F
THERMAL EFFICIENCY	75%	75%	75%
EMITTER (FIRST SECTION)			
Size	10’ x 3½” OD	10’ x 4” OD	10’ x 4” OD
Material	Aluminized Steel	Aluminized Steel	Painted Aluminized Steel
Gauge	16	12	16
Emissivity	.5	.5	.5 - .8
(SECOND SECTION)			
Size	35’ - 55’ x 3½” OD	20’ - 50’ x 4” OD	20’ - 50’ x 4” OD
Material	Hot Rolled Steel	Aluminized Steel	Painted Aluminized Steel
Gauge	16	16	16
Emissivity	.8	.5	.5 - .8
REFLECTOR			
Size	4” x 14”	3½” x 19”	5½” x 11”
Material	Polished Aluminum	Polished Aluminum	Polished Aluminum
CONFIGURATION	D	E	F
TYPE	Quasi 1b/1c	1c	1c

Figure 36

APPLIANCE COMPARISON (E)

MANUFACTURER	G	H	H
THERMAL EFFICIENCY	75%	75%	75%
EMITTER (FIRST SECTION)			
Size	10’ x 4” OD	10’ x 3” OD	10’ x 3” OD
Material	Aluminized Steel	Heat Treated Aluminized Steel	Heat Treated Aluminized Steel
Gauge	16	16	16
Emissivity	.5	.8	.8
(SECOND SECTION)			
Size	20’ - 50’ x 4” OD	22’ x 3” OD	2’ - 15’ x 3” OD
Material	Painted Steel	Heat Treated Aluminized Steel	Heat Treated Aluminized Steel
Gauge	16	16	16
Emissivity	.8	.5	.8
REFLECTOR			
Size	5” x 11”	5¼” x 18 1/8”	5¼” x 18½”
Material	Polished Aluminum	Polished Aluminum	Polished Aluminum
CONFIGURATION	D	E	F
TYPE	1c	1c	1c

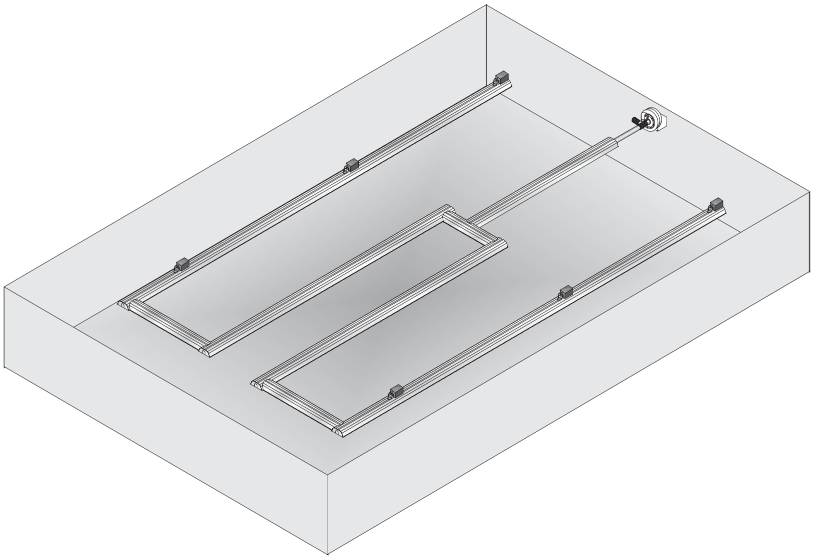
Figure 37

APPLIANCE COMPARISON (F)

MANUFACTURER	I	J	K
THERMAL EFFICIENCY	75%	75%	80%
EMITTER (FIRST SECTION)			
Size	10’ x 4” OD	10’ x 4” OD	10’ x 4” OD
Material	Aluminized Steel	Painted Aluminized Steel	Aluminized Steel
Gauge	16	14	16
Emissivity	.5	.5 - .8	.5 - .8
(SECOND SECTION)			
Size	30’ - 50’ x 4” OD	10’ - 50’ x 4” OD	30’ - 50’ x 4” OD
Material	Hot Rolled Steel	Hot Rolled Steel	Aluminized Steel
Gauge	16	12	16
Emissivity	.8	.8	.5
REFLECTOR			
Size	5” x 11”	6½” x 16”	6’ x 13½”
Material	Polished Aluminum	Polished Aluminum	Polished Aluminum
CONFIGURATION	D	B	D
TYPE	1c	1c	Quasi 1b/1c

Distribution Layouts

Figure 38

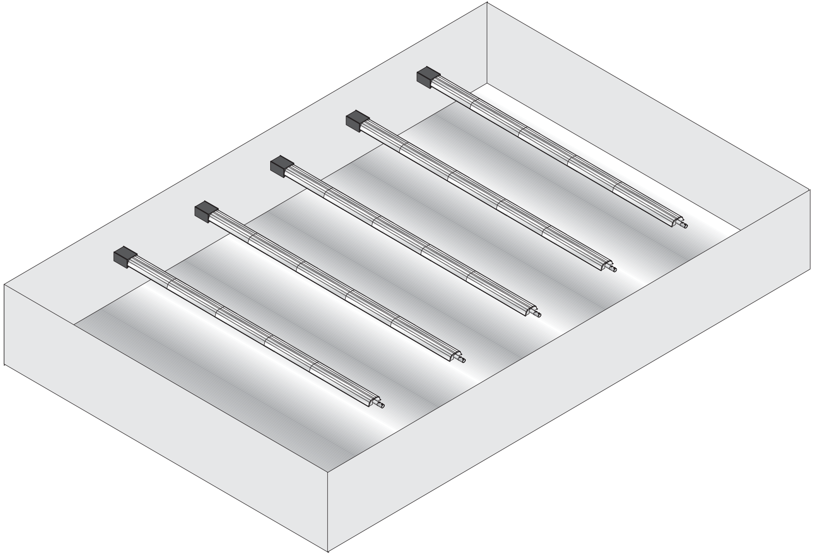


Total Span Distribution

- Provided by multiple burners in series

Perimeter Distribution

- Provided by multiple end burners



Area Distribution

- Provided by individual burner

APPENDIX A:

ANNUAL DEGREE DAYS TO SELECTED BASES

This information is extracted from a publication entitled Degree Days to Selected Bases that is available from the National Climatic Center, U.S. Department of Commerce, National Oceanic and Atmospheric Administration.

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Alabama												
Birmingham	2844	1995	1233	838	488	17	21	1928	2916	4073	5403	6877
Huntsville	3302	2414	1670	1103	686	11	16	1808	2747	3828	5080	6492
Mobile	1684	1062	619	330	148	25	29	2577	3780	5162	6698	8342
Montgomery	2269	1508	945	547	282	22	25	2238	3302	4568	5991	7551
Alaska												
Anchorage	10911	9122	7492	6081	4896	-23	-18	0	36	224	647	1279
Annette	7053	5315	3773	2513	1543			14	99	386	940	1808
Barrow	20265	18440	16615	14803	13009	-45	-41	0	0	0	13	44
Barter Island	19994	18169	16344	14528	12738			0	0	0	9	44
Bethel	13203	11404	9685	8140	6835			0	25	142	411	938
Bettles	15925	14180	12548	11060	9718			17	97	289	626	1110
Big Delta	13698	11985	10410	8977	7735			34	145	395	787	1370
Cold Bay	9865	8040	6230	4532	3096			0	0	16	138	533
Fairbanks	14345	12561	11115	9714	8451			52	196	467	898	1468
Gulkana	13938	12162	10507	8985	7548			9	63	228	537	1027
Homer	10364	8539	6745	5133	3840			0	0	24	240	777
Juneau	9007	7222	5557	4107	2925	-4	-1	0	39	197	573	1219
King Salmon	11562	9773	8047	65936	5304			0	12	112	465	1023
Kodiak	8850	7049	5327	3819	2593	10	13	0	7	117	436	1032
Kotzebue	15039	14237	12491	10852	9337			0	23	102	288	588
McGrath	14487	12736	11107	9634	8348			14	88	284	642	1184
Nome	14325	12503	10721	9047	7528	-31	-27	0	0	46	197	503
St. Paul Island	11119	9294	7469	5667	4021			0	0	0	24	199
Shemya	9735	7910	6085	4298	2693			0	0	0	30	254
Summa FAA	14368	12556	10790	9146	7640			0	10	71	253	578
Talkeetna	11708	9934	8306	6848	5609			6	57	254	620	1207
Unaiakleet	14027	12238	10515	8943	7565			0	31	138	391	842
Yakutat	9533	7711	5942	4420	3181			0	0	56	362	947
Arizona												
Flagstaff	7322	5776	4421	3257	2299	-2	4	140	416	894	1562	2418
Phoenix	1552	899	431	165	45	31	34	3508	4680	6039	7596	9297
Prescott FAA	4456	3303	2321	1507	863	4	0	882	1560	2400	3414	4612
Tucson	1752	1050	541	229	65	28	32	2814	3937	5253	6765	8431
Winslow	4733	3623	2683	1862	1249	5	10	1203	1921	2802	3828	5018
Yuma	1005	507	211	59	8	36	39	4195	5518	7045	8719	10498
Arkansas												
Fort Smith	3336	2442	1667	1075	613	12	17	2022	2949	4015	5239	6595
Little Rock	3354	2442	1687	1075	624	15	20	1925	2843	3908	5128	6436
California												
Bakersfield	2185	1367	760	371	147	30	32	2179	3185	4400	5835	7437
Bishop	4313	3179	2230	1437	848			1037	1728	2603	3641	4876
Blue Canyon	5704	4271	3037	2015	1206			302	698	1283	2079	3106
Daggert FAA	2203	1420	824	410	166			2729	3765	5004	6415	7996
Eureca	4679	2925	1494	507	194	31	33	0	55	480	1414	2816
Fresno	2550	1724	995	493	205	28	30	1671	2563	3667	4986	6525
Long Beach	1506	772	292	70	8	41	43	985	1982	3325	4928	6696
Los Angeles Int'l.	1819	833	295	66	7	41	43	615	1464	2755	4348	6115
Los Angeles C.C.	1245	522	158	25	0	37	40	1185	2289	3747	5442	7244
Mount Shasta	5890	4458	3215	2177	1338			286	680	1263	2045	3035
Oakland	2909	1570	714	263	61	34	36	128	622	1598	2963	4587
Red Bluff	2688	1762	1018	505	208			1904	2803	3895	5196	6727
Sacramento	2843	1837	1043	493	185	30	32	1159	1971	3011	4286	5812
Sacramento City	2587	1627	893	406	148			1291	2158	3249	4584	6151
Sandberg	4427	3177	2107	1250	622			800	1374	2123	3100	4293
San Francisco AP	1507	648	213	42	9	35	38	722	1694	3084	4746	6532
San Francisco FB	3080	1576	608	169	25			39	368	1230	2519	4258
Santa Maria	3053	1624	690	229	42	31	33	84	484	1377	2738	4380
Stockton	2806	1835	1072	537	219	28	30	1259	2100	3167	4455	5968

Appendix A: Annual Degree Days to Selected Bases

ROBERTS-GORDON, LLC

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Colorado												
Alamosa	8509	7029	5654	4473	3457	-21	-16	86	329	780	1428	2227
Colorado Springs	6473	5131	3954	2949	2089	-3	2	461	945	1582	2417	3383
Denver-Stapleton	6016	4723	3601	2653	1852	-5	1	625	1159	1857	2739	3759
Denver city	5505	4246	3175	2271	1533			742	1312	2071	2993	4074
Eagle	8425	6864	5505	4319	3317			117	385	845	1487	2313
Grand Junction	5605	4441	3425	2551	1814	2	7	1140	1810	2619	3565	4653
Purblo	5394	4221	3220	2351	1628	-7	0	981	1632	2456	3412	4514
Connecticut												
Bridgeport	5461	4264	3216	2321	1583	6	9	735	1362	2140	3064	4152
Harford	6350	5085	3971	3005	2173	3	7	584	1143	1855	2715	3706
District Columbia												
Wash DC (Dules	5005	3881	2896	2054	1380			940	1636	2474	3456	4616
Wash DC (Nat AP)	4211	3182	2293	1563	984	14	17	1415	2210	3152	4237	5489
Delaware												
Wilmington	4980	3824	2839	2003	1330	10	14	952	1697	2537	3525	4573
Florida												
Apalachicola	1361	792	426	189	67			2663	3930	5377	6967	8669
Dayton Beach	897	480	215	71	25	31	32	2919	4321	5881	7563	9341
Fort Myers	457	189	56	9	0	41	44	3711	5265	6958	8741	10553
Jacksonville	1327	788	429	194	70	29	32	2596	3890	4349	6938	8641
Key West	59	7	0	0	0	55	57	4838	6660	8474	10299	12124
Lakeland	678	330	128	40	7	39	41	3929	4774	6398	8135	9927
Miami	206	54	8	0	0	44	47	4038	5715	7494	9306	11131
Orlando	733	370	151	48	9	35	38	3226	4686	6291	8016	9806
Pensacola	1578	991	575	302	135	29	33	2595	3932	5341	6893	8551
Tallahassee	1563	996	550	279	116	27	30	2563	3792	5200	6755	8415
Tampa	718	364	151	50	14	36	40	3366	4836	6447	8172	9963
West Palm Beach	299	88	27	0	0	41	45	3786	5408	7159	8960	10785
Georgia												
Athens	2975	2064	1370	830	462	18	22	1722	2661	3767	5052	6508
Atlanta	3095	2189	1451	911	524	17	22	1589	2511	3604	4880	6316
Augusta	2547	1729	1106	652	348	20	23	1995	2999	4204	5573	7094
Columbus	2378	1600	1010	593	313	21	24	2143	3188	4429	5831	7375
Macon	2240	1492	934	536	271	21	25	2294	3373	4643	6066	7626
Rome	3342	2422	1653	1068	537			1615	2514	3576	4816	6210
Savannah	1952	1258	751	413	185	24	27	2317	3444	4766	6248	7851
Hawaii												
Hio	0	0	0	0	0	61	62	3066	4887	6712	8537	10362
Honolulu	0	0	0	0	0	62	63	4221	6046	7871	9696	11521
Kahului	0	0	0	0	0			3732	5555	7380	9205	11030
Lihue	0	0	0	0	0			3719	5535	7360	9185	11010
Idaho												
Boise	5833	4533	3399	2434	1626	3	10	714	1233	1929	2793	3811
Idaho Falls	8619	7129	5800	4609	3590	-11	-6	237	573	1064	1703	2511
Lewiston	5464	4168	3050	2112	1366	-1	6	657	1186	1886	2780	3856
Pocatello	7063	5687	4454	3401	2504	-8	-1	437	883	1477	2252	3177

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Illinois												
Carro	3633	2895	2090	1447	925			1806	2687	3710	4893	6197
Chicago Midway	6127	4952	3912	2998	2219	-5	0	925	1575	2361	3272	4317
Chicago O'Hare	6497	5245	4163	3220	2404	-8	-4	664	1243	1986	2863	3872
Decatur	5344	4247	3293	2461	1778	-3	2	1197	1925	2797	3791	4932
Moline	6395	5202	4170	3263	2462	-9	-4	893	1530	2324	3236	4262
Peoria	6098	4930	3910	3013	2239	-8	-4	968	1631	2431	3360	4412
Rockford	6845	5600	4507	3555	2713	-9	-4	714	1298	2032	2899	3883
Springfield	5558	4437	3468	2615	1913	-3	2	1116	1821	2670	3654	4772
Indiana												
Evansville	4624	3578	2685	1929	1327	4	9	1364	2139	3064	4140	5367
Fort Wayne	6209	4992	3930	2996	2193	-4	1	748	1358	2117	3008	4030
Indianapolis	5577	4430	3431	2568	1856	-2	2	974	1653	2478	3441	4554
South Bend	6462	5213	4118	3156	2333	-3	1	695	1271	2002	2865	3867
Iowa												
Burlington Radio	6149	4988	3970	3077	2306	-7	-3	994	1657	2466	3386	4447
Des Mones	6710	5521	4470	3546	2728	-10	-5	928	1561	2335	3225	4243
Debuque	7277	5992	4871	3893	3028	-12	-7	605	1146	1850	2897	3657
Mason Cisy	7901	6586	5430	4425	3529	-15	-11	580	1088	1763	2581	3508
Sioux City	6953	5745	4574	3738	2898	-11	-7	932	1545	2298	3182	4177
Spencer	7770	6474	5329	4334	3448			641	1171	1857	2685	3620
Waterloo	7415	6153	5040	4070	3199	-15	-10	675	1236	950	2806	3760
Kansas												
Condordia	5623	4498	3509	2646	1912			1302	1998	2832	3795	4886
Dodge City	5046	3963	3011	2184	1512	0	5	1411	2153	3022	4025	5176
Goodland	6119	4891	3804	2857	2041	-5	0	925	1515	2253	3131	4139
Russell FAA	5312	4220	3259	2423	1735	0	4	1485	2219	3081	4076	5210
Topeka	5243	4152	3203	2378	1700	0	4	1361	2093	2974	3974	5112
Wichita	4687	3654	2750	1977	1346	3	7	1673	2464	3386	4438	5638
Kentucky												
Covington	5070	3965	3001	2189	1527	1	6	1080	1798	2654	3672	4834
Lexington	4729	3652	2743	1963	1350	3	8	1197	1941	2858	3904	5120
Louisville	4640	3584	2676	1906	1303	5	10	1268	2032	2942	4005	5227
Louisiana												
Alexandria	2200	1443	880	490	234	23	27	2193	3260	4525	5958	7531
Baton Rouge	1670	1036	582	295	120	25	29	2585	3775	5150	6685	8340
Lake Charles	1496	908	500	240	91	27	31	2739	3978	5391	6956	8638
New Orleans Aud	1343	805	439	202	82			2876	4166	5622	7215	8916
New Orleans N O	1465	893	492	239	96	29	33	2706	3960	5383	6956	8638
Shreveport	2167	1438	883	490	233	20	25	2538	3634	4906	6335	7903
Maine												
Bangor	7950	6496	5222	4103	3122	-11	-6	268	640	1194	1896	2470
Caribou	9632	8044	6634	5409	4319	-18	-13	128	365	784	1379	2118
Old Town FAA	8648	7133	5600	4628	3589			209	519	1016	1660	2454
Portland	7498	6035	4764	3658	2705	-6	-1	252	616	1169	1890	2758
Maryland												
Baltimore	4729	3631	2662	1873	1235	10	13	1108	1840	2708	3728	4918
Massachusetts												
Blue Hill	6335	5020	3885	2895	2071			457	968	1659	2493	3498
Boston	5621	4383	3313	2405	1659	6	9	661	1250	2000	2920	4000
Nantucket	5929	4520	3323	2311	1513			284	708	1332	2143	3170
Worchester	6848	5498	4326	3296	2421	0	4	387	863	1514	2303	3259

Appendix A: Annual Degree Days to Selected Bases

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Michigan												
Alpena	8518	6982	5635	4473	3464	-11	-6	208	497	981	1642	2459
Denver	6419	5167	4072	3113	2280	3	6	654	1227	1961	2832	3815
Flint	7041	5705	4540	3529	2540	-4	1	438	923	1586	2399	3335
Grand Rapids	6801	5514	4383	3396	2524	1	5	575	1108	1807	2646	3598
Houghton Lake	8347	6861	5579	4455	3486			250	590	1132	1832	2689
Lansing	6604	5608	4464	3470	2595	-3	1	535	1059	1747	2578	3528
Marquette	8351	6635	5517	4379	3378	-12	-8	216	531	1031	1725	2549
Muskegon	6890	5850	4390	3373	2482	2	6	469	953	1620	2421	3360
Sault Ste Marie	9193	7614	6215	5017	3971	-12	-8	139	386	816	1443	2217
Traverse City	7698	6272	5035	3953	3013	-3	1	376	773	1362	2101	2589
Minnesota												
Duluth	9756	8185	6793	5581	4540	-21	-16	176	425	864	1482	2259
Intl. Falls	10547	8995	7623	6413	5348	-29	-25	176	454	908	1523	2283
Minn. - St. Paul	8159	6842	5677	4558	3765	-15	-12	585	1097	1758	2575	3491
Rochester	8227	6868	5682	4643	3733	-17	-12	474	943	1579	2370	3280
St. Cloud	8868	7481	6255	5187	4241	-15	-11	426	862	1468	2220	3098
Mississippi												
Jackson	2300	1548	988	590	319	21	25	2316	3394	4664	6086	7639
Meridian	2388	1621	1042	623	339	19	23	2231	3289	4538	5940	7483
Missouri												
Columbia	5078	3997	3064	2259	1605	-1	4	1269	2009	2901	3919	5089
Kansas City	5161	4089	3157	2351	1694	2	6	1421	2169	3061	4085	5249
St. Joseph	5435	4341	3378	2544	1847	-3	2	1334	2064	2925	3911	5046
St. Louis	4750	3701	2798	2031	1419	3	8	1475	2252	3174	4232	5445
Springfield	4570	3517	2611	1844	1235	3	9	1382	2149	3068	4126	5342
Montana												
Billings	7265	5898	4697	3641	2766	-15	-10	496	951	1581	2354	3298
Butte	9719	8059	6557	5225	4078	-24	-17	58	222	545	1038	1718
Cut Bank	9033	7474	6096	4907	3886	-25	-20	140	406	856	1489	2299
Dilon	8354	6821	5457	4255	3237			199	492	953	1570	2382
Glasgow	8969	7572	6329	5238	4302	-22	-18	438	867	1449	2185	3074
Great Falls	7652	6248	5022	3965	3074	-21	-15	339	760	1365	2132	3066
Havre	8687	7282	5073	5005	4104	-18	-11	395	818	1432	2191	3113
Helena	8190	6710	5309	4247	3258	-21	-16	256	606	1106	1786	2629
Kaispell	8554	6959	5542	4304	3233	-14	-7	117	348	755	1342	2096
Lewiston FAA	8586	7038	5676	4487	3467	-22	-16	192	468	933	1567	2379
Miles City	7889	6562	5392	4369	3479	-20	-15	752	1252	1905	2706	3641
Missoula	7931	6410	5066	3884	2876	-13	-6	188	497	970	1616	2428
Nebraska												
Grand Island	6420	5224	4166	3239	2434	-8	-3	1036	1662	2428	3326	4245
Lincoln Ap	6218	5062	4040	3139	2362			1148	1809	2611	3536	4585
Lincoln Ap	6012	4875	3870	2993	2234	-5	-2	1187	1865	2685	3634	4701
Norfolk	6981	5745	4663	3710	2863	-8	-4	925	1520	2263	3131	4118
North Platte	6743	5470	4345	3354	2509	-6	-4	802	1359	2060	2898	3874
Omaha-Eppley	6049	4907	3911	3237	2290			1173	1862	2691	3637	4715
Omaha-North	6601	5400	4349	3427	2624	-8	-3	949	1573	2346	3249	4270
Scottsbluff	6774	5473	4304	3289	2415	-8	-3	666	1188	1845	2653	3605
Valentine	7300	6006	4859	3847	2956			736	1267	1945	2758	3682
Nevada												
Elko	7483	6027	4714	3586	2625	-8	-2	342	706	1228	1910	2785
Ely	7814	6327	5004	3826	2829	-10	-4	207	550	1052	1694	2526
Las Vegas	2601	1770	1120	625	306	25	28	2946	3938	5114	6443	7950
Lovelock FAA	5890	4695	3550	2579	1747	8	12	684	1217	1894	2743	3740
Reno	6022	4612	3387	2360	1534	5	10	329	739	1344	2150	3140
Tonopah	5800	4610	3492	2532	1723	5	10	631	1167	1869	2739	3753
Winnemucca	6629	5241	3994	2931	2015	-1	3	407	845	1423	2185	3096

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	65	60	55	50	45	99%	97%	65	60	55	50	45
New Hampshire												
Concord	7360	5867	4757	3682	2762	-8	-3	349	781	1394	2150	3051
Mt. Washington	13678	12053	10253	8534	6960			0	0	25	132	379
New Jersey												
Atlantic City	4946	3783	2784	1941	1257	10	13	864	1533	2349	3339	4485
Atlantic City Marina	4693	3534	2530	1713	1076			835	1503	2317	3333	4517
Newark	5034	3911	2920	2074	1391	10	14	1024	1721	2543	3533	4677
Trenton	4947	3818	2832	1996	1323	11	14	968	1661	2483	3482	4634
New Mexico												
Albuquerque	4292	3234	2330	1557	963	12	16	1316	2080	2996	4053	5288
Clayton	5207	3999	2966	2089	1374			767	1380	2176	3120	4231
Roswel	3697	2729	1898	1226	706	13	18	1580	2417	3412	4566	5872
Truth or Consque.	3392	2447	1636	1007	542			1558	2429	3447	4647	6008
Tucmcari FAA	4047	3015	2135	1415	858	8	13	1357	2148	3096	4200	5467
Zura FAA	5815	4507	3381	2437	1648			473	983	1685	2567	3605
New York												
Albany	6888	5596	4451	3457	2595	-6	1	574	1111	1787	2619	3583
Binghamton	7285	5908	4714	3677	2767	-2	1	369	820	1452	2231	3151
Buffalo	6927	5591	4429	3403	2508	2	6	437	928	1580	2388	3319
Massena FAA	8237	6827	5596	4510	3552	-13	-8	343	759	1352	2088	2958
NY Centr. Park	4848	3739	2771	1958	1299	11	15	1068	1784	2636	3653	4814
NY JFK	5184	4023	2994	2130	1422	12	15	861	1520	2321	3278	4395
NY LaGuardia	4909	3787	2806	1980	1311	11	15	1048	1752	2587	3589	4740
Oswego East	6792	5444	4274	3243	2376	1	7	435	915	1570	2360	3319
Rochester	6719	5417	4285	3291	2434	1	5	531	1062	1750	2580	3549
Syracuse	6678	5379	4250	3267	2429	-3	2	551	1081	1778	2621	3607
North Carolina												
Asheville	4237	3129	2224	1488	937	10	14	872	1587	2508	3595	4868
Cape Hateras	2731	1846	1166	702	380			1550	2485	3635	4991	6500
Charlotte	3218	2300	1552	984	585	18	22	1596	2503	3579	4842	6263
Greensboro	3825	2811	1984	1324	825	14	18	1341	2158	3149	4318	4640
Raleigh-Durham	3514	2542	1744	1123	670	16	20	1394	2242	3273	4482	5850
Wilmington	2433	1632	1028	610	321	23	26	1964	2995	4225	5622	7162
North Dakota												
Bismark	9044	7656	6425	5326	4374	-23	-19	487	928	1518	2248	3116
Fargo	9271	7891	6663	5573	4615	-22	-18	473	919	1515	2251	3123
Minot FAA	9407	7964	6685	5564	4573	-24	-20	370	758	1299	2002	2837
Wiliston	9161	7753	6504	5387	4450	-25	-21	422	841	1415	2128	3011
Ohio												
Akron-Canton	6224	4971	3883	2936	2129	1	6	634	1205	1943	2820	3839
Cincinnati Abbe Obs	4844	3763	2830	2040	1412			1188	1931	2819	3864	5060
Cincinnati AP	5070	3965	3001	2189	1527	1	6	1080	1798	2654	3572	4834
Cleveland	6154	4901	3819	2875	2079	1	5	613	1183	1826	2807	3836
columbus	5702	4513	3480	2597	1846	0	5	809	1449	2244	3183	4257
Dayton	5641	4483	3468	2600	1866	-1	4	936	1603	2414	3370	4460
Mansfield	5818	4518	3573	2579	1917	0	5	818	1445	2225	3152	4219
Toledo Express	6381	5136	4049	3091	2274	-3	1	685	1268	2001	2870	3877
Youngstown	6426	5145	4032	3054	2232	-1	4	518	1065	1774	2621	3623
Oklahoma												
Oklahoma City	3695	2760	1962	1326	809	9	13	1876	2768	3788	4980	6289
Tulsa	3680	2750	1950	1306	778	8	13	1949	2850	3665	5052	6347
Oregon												
Astoria	5295	3620	2233	1215	570	25	29	13	159	596	1415	2598
Burns	7212	5740	4436	3299	2343			289	649	1168	1851	2724
Eugene	4739	3313	2141	1226	607	22	17	239	638	1286	2201	3417
Meacham	7863	6249	4817	3556	2495			103	317	712	1275	2034
Medford	4930	3614	2496	1577	882			562	1077	1779	2685	3813
North Bend	4688	2985	1642	756	292			0	131	597	1553	2913
Pendelton	5240	3968	2868	1970	1254	-2	5	656	1211	1935	2858	3982
Portland	4792	3385	2234	1333	708	17	23	300	711	1378	2309	3520
Redmond	6643	5106	3767	2621	1680			170	459	943	1620	2512
Salem	4852	3424	2246	1317	667	18	23	232	620	1272	2169	3355
Sexton Summit	6430	4859	3477	2311	1374			137	381	837	1499	2386

Appendix A: Annual Degree Days to Selected Bases

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Pennsylvania												
Allentown	5827	4618	3550	2633	1843	4	9	772	1392	2150	3053	4088
Bradford	7804	6294	5006	3894	2931			170	482	1022	1735	2596
Erie	6851	5485	4304	3283	2411	4	9	373	832	1482	2282	3235
Harrisburg	5224	4087	3097	2238	1541	7	11	1025	1711	2545	3511	4644
Philadelphia	4865	3753	2788	1965	1312	10	14	1104	1817	2671	3679	4849
Pittsurg City	5278	4135	3138	2294	1603			948	1630	2456	3440	4573
Pittsburg AP	5930	4694	3637	2720	1938	1	5	647	1240	2004	2914	3961
W-Barre-Scranton	6277	5018	3928	2972	2149	1	5	608	1181	1909	2778	3783
Williamsport	5981	4757	3695	2764	1971	2	7	698	1299	2059	2952	3986
Rhode Island												
Block Island	5771	4432	3289	2306	1517			359	844	1523	2368	3409
Providence	5972	4682	3565	2599	1803	5	9	532	1067	1774	2625	3662
South Carolina												
Charleston	2146	1406	864	496	240	24	27	2078	3163	4454	5903	7478
Charleston City	1904	1230	741	412	188	25	28	2354	3502	4839	6334	7937
Columbia	2598	1738	1154	686	374	20	24	2087	3094	4292	5647	7159
Florence	2566	1748	1127	676	374	22	25	1952	2960	4171	5538	7060
Grnvle-Spartenburg	3163	2246	1493	921	519	18	22	1573	2477	3552	4809	6229
South Dakota												
Aberdeen	8617	7267	6078	5014	4067	-19	-15	566	1046	1678	2440	3337
Huron	8055	6751	5600	4052	3678	-18	-14	711	1239	1912	2714	3641
Pierre	7677	6401	5271	4273	3409	-15	-10	858	1406	2102	2928	3889
Rapid City	7324	5982	4799	3762	2868	-11	-7	661	1148	1786	2575	3511
Sioux Falls	7838	6543	5401	4394	3498	-15	-11	719	1253	1933	2746	3681
Tennessee												
Bristol	4306	3255	2373	1646	1093	9	14	117	1880	2823	3922	5197
Chattanooga	3505	2574	1785	1180	737	13	18	1636	2526	3566	4791	6169
Knoxville	3478	2557	1775	1187	744	13	19	1569	2475	3518	4753	6135
Memphis	3227	2352	1624	1058	640	13	18	2029	2984	4077	5339	6744
Nashville	3696	2758	1964	1338	852	9	14	1694	2576	3613	4812	6151
Oak Ridge	3944	2955	2119	1445	933			1367	2202	3187	4338	5656
Texas												
Abilene	2610	1801	1162	664	342	15	20	2466	3481	4670	5995	7498
Amarillo	4183	3156	2278	1548	976	6	11	1433	2230	3177	4274	5527
Austin	1737	1097	620	316	127	24	28	2903	4095	5443	6962	8600
Brownsville	650	336	146	54	19	35	39	3874	5385	7020	8753	10543
Corpus Christi	930	514	243	98	28	31	35	3474	4880	6438	8111	9872
Dallas	2290	1544	949	526	250	18	22	2755	3835	5073	6467	8016
Del Rio	1523	923	494	230	80	26	31	3363	4596	5986	7548	9222
El Paso	2678	1833	1149	653	326	20	24	2098	3077	4229	5548	7048
Fort Worth	2382	1616	1007	562	274	17	22	2587	3642	4862	6239	7775
Galveston	1224	704	369	157	54	31	36	3004	4312	5800	7413	9139
Houston	1434	864	471	215	81	27	32	2889	4147	5576	7150	8835
Lando No 2	876	481	230	87	32	32	36	4137	5568	7143	8824	10593
Lubbock	3545	2603	1807	1163	666	10	15	1647	2535	3559	4745	6068
Lufkin	1940	1253	731	385	163	25	29	2592	3730	5033	6512	8114
Midland	2621	1808	1159	656	333	16	21	2245	3258	4434	5757	7258
Port Arthur	1518	924	504	238	86	16	27	2798	4028	5431	6990	8669
San Angelo	2240	1498	918	493	227	18	22	2702	3798	5031	6432	7993
San Antonio	1570	956	518	242	92	25	30	2994	4206	5594	7146	8818
Victoria	1227	702	364	150	51	29	32	3140	4440	5925	7537	9262
Waco	2058	1357	807	437	195	21	26	2863	3988	5271	6717	8303
Wichita Falls	2904	2061	11384	832	451	14	18	2611	3594	4741	6015	7458
Utah												
Blanding	6163	4869	3732	2757	1912			600	1129	1827	2670	3646
Byce Canyon	9133	7459	5949	4616	3480			41	193	505	1005	1686
Cedar city	6137	4833	3690	2717	1897	-2	5	615	1130	1813	2671	3678
Milford	6412	5109	3957	2969	2121			688	1212	1885	2721	3704
Salt Lake city	5983	4733	3633	2676	1864	3	8	927	1502	2221	3094	4108
Wndover	5760	4558	3511	2621	1870			1137	1760	2538	3475	4547
Vermont												
Burlington	7876	6488	5270	4190	3246	-12	-7	396	833	1440	2180	3066

Degree Day Information

	Heating Degree Day Base					Design Temp Dry Bulb	Design Temp Dry Bulb	Cooling Degree Day Base				
State	**65**	**60**	**55**	**50**	**45**	**99%**	**97%**	**65**	**60**	**55**	**50**	**45**
Virginia												
Lynchburg	4233	3172	2269	1536	966	12	16	1100	1861	2783	3873	5128
Norfolk	3488	2516	1710	1100	663	20	22	1441	2284	3315	4530	5918
Richmond	3939	2916	2061	1388	866	14	17	1353	2157	3127	4276	5580
Roanoke	4307	3234	2326	1580	1011	12	16	1030	1778	2690	3771	5029
Wallops Island	4240	3170	2268	1531	978			1107	1865	2788	3881	5149
Washington												
Olympia	5530	3970	2653	1617	854	16	22	101	365	880	1657	2731
Omak	6858	5476	4253	3230	2355			522	965	1573	2367	3324
Quillayute	5951	4232	2750	1603	813			8	116	458	1137	2172
Seattle-Tacoma	5185	3657	2386	1416	731	21	26	129	423	984	1832	2981
Seattle-(Urban)	4727	3269	2091	1194	602	22	27	183	549	1197	2127	3358
Spokane	6835	5420	4173	3088	2188	-6	2	388	797	1377	2120	3040
Stampede Pass	9400	7643	6006	4532	3256			16	83	274	623	1176
Walla Walla	4835	3616	2600	1760	1126	0	7	862	1471	2279	3260	4457
Yakima	6009	4655	3483	2502	1688	-2	5	479	945	1604	2452	3455
West Virginia												
Beckley	5615	4356	3279	2390	1652	-2	4	490	1061	1809	2745	3833
Charleston	4590	3500	2590	1809	1216	7	11	1055	1790	2699	3750	4981
Elkins	5975	4659	3533	2616	1834	1	6	389	905	1601	2508	3555
Huntington	4624	3533	2624	1843	1249	5	10	1098	1829	2746	3790	5020
Parkersburg	4817	3720	2786	1987	1363	7	11	1045	1770	2657	2686	4888
Wisconsin												
Eau Claire	8388	7033	5832	4786	3860	-15	-11	459	928	1554	2331	3231
Green Bay	8098	6689	5473	4405	3478	-13	-9	386	805	1411	2168	3066
La Crosse	7417	6158	5050	4088	3219	-13	-9	695	1264	1978	2841	3798
Madison	7730	6373	5188	4156	3250	-11	-7	460	923	1572	2361	3279
Milwaukee	7444	6080	4898	3860	2946	-8	-4	450	911	1554	2342	3252
Wyoning												
Casper	7555	6167	4914	2813	2857	-11	-5	458	895	1468	2193	3061
Cheyenne	7255	5825	4562	3452	2512	-9	-1	327	734	1288	2003	2886
Lander	7869	6471	5207	4080	3140	-16	-11	383	814	1376	2078	2965
Rock Springs	8410	5922	5592	4412	3393	-9	-3	227	563	1059	1703	2515
Sheridan	7708	6298	5024	3935	3000	-14	-8	446	860	1411	2147	3037

Degree Day information is extracted from a publication entitled "Degree Days Selecyed Bases" that is available from the National Climatic Center, U.S. Department of Commerce, National Oceanic & Atmospheric Administration

Design Temperature Dry Buld is extracted from the 1989 ASHRAE Handbook of Fundamentals Section 24.4 Table 1 of Climatic Conditions for the United States.