ENGINEERING MANUAL - APPLICATION INFORMATION

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I. INTRODUCTION TO INFRARED

A. WHAT IS INFRARED?

Infrared is not new. Regardless of what we do, infrared is involved in almost every aspect of our daily lives. Infrared heating has been an important factor in the life of man since the very beginning.

The enormous amount of heat the earth receives from the sun is transmitted to the earth by infrared energy, along with visible light. The fact that we cannot see infrared complicates our understanding of it.

There are many complex formulas and constants involved in computing infrared or radiant heat transfer. However, if we concern ourselves only with the basics and the terminology involved, we find that the concept of infrared heating can be easily understood and applied to our needs.

The temperature and area of the infrared source (emitter) are by far the most important aspects of infrared heating. Next in line would be the radiant efficiency (emissivity). For the most part, these terms would only be used when conversing with someone knowledgeable in the infrared industry.

Infrared is a small portion of the electromagnetic spectrum located between visible light and the top end of the radar and microwave portion of the electromagnetic spectrum. The major difference between visible light energy and infrared energy is the source temperature and the fact that the human eye cannot see low temperature infrared energy. The infrared name is derived from the Latin word “infra,” meaning below, and the English word “red.” As the name implies, infrared simply means below red; therefore, we cannot see it. There is specially made photographic film available that is sensitive to infrared at selected wavelengths. The military also uses infrared to “see” enemy troop movements at night.

Since infrared is so similar to light energy, it follows the same basic law of optics. It radiates in all directions from a point source, travels in a straight line at the speed of light and can be polarized, focused or reflected the same as light.

The temperature of our sun, approximately 5800ºC, indicates that it is radiating at approximately 2,600,000 Btu per square foot per hour.

A great deal of this energy is given off as visible light due to the extremely high source temperature. Being so far away, the earth receives only a very small portion of the infrared energy given off by the sun. The radiant heat intensity throughout most of the United States during our summer months amounts to only 200 to 250 Btu per square foot per hour.

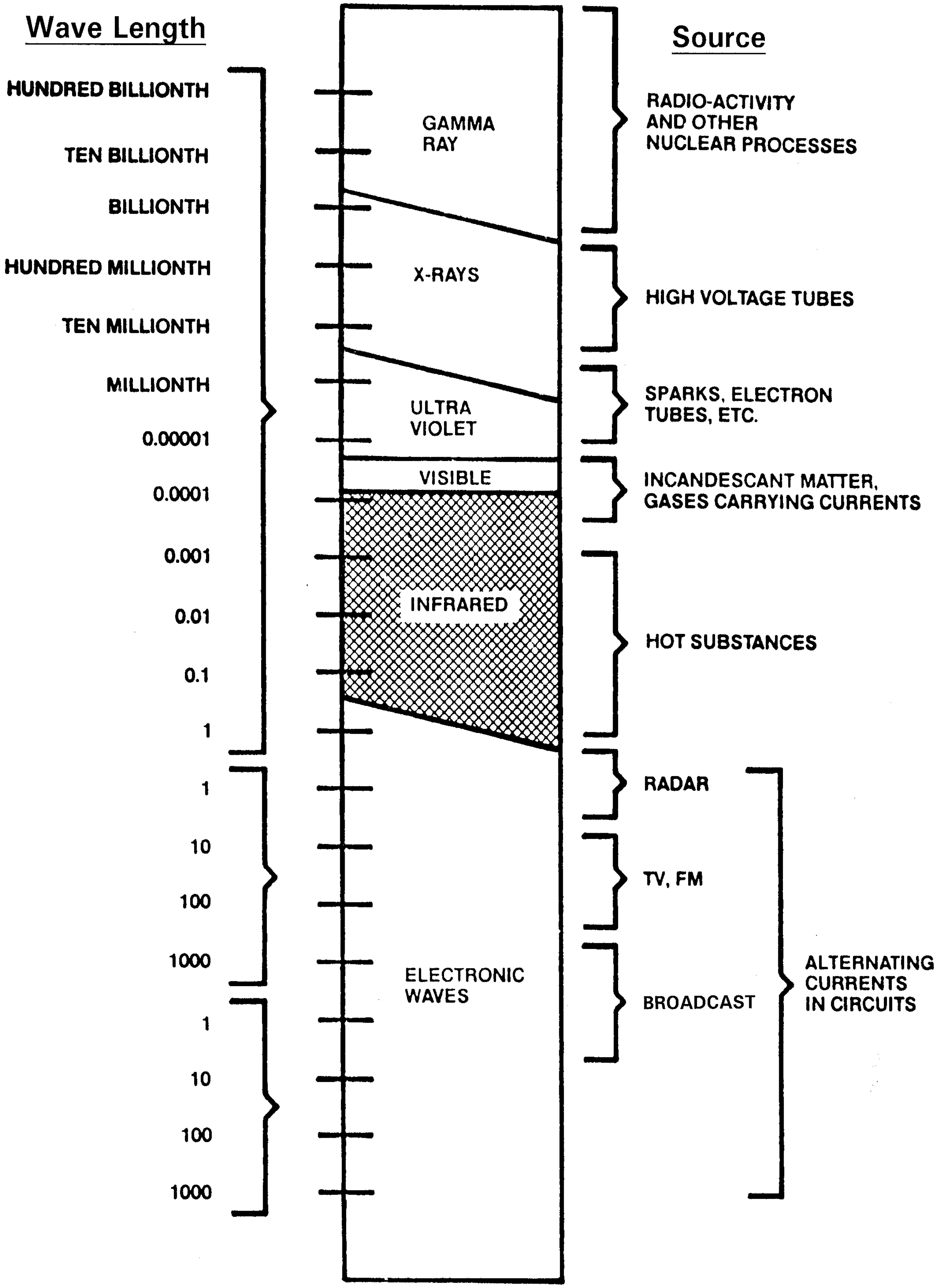
Every object at temperatures above absolute zero will give off infrared energy. The temperature, area and emissivity will determine the amount of energy given off. Any object heated to the point where it starts to glow gives off visible light.

If we have two objects, one larger than the other, made of identical material and heated to the same temperature, the larger one will radiate more heat than the smaller one. However, if we heat the smaller object to a higher temperature, we can make it radiate the same amount of infrared as the larger one. We have to be careful not to heat the smaller object too hot because then it becomes incandescent and starts to emit light energy. There is a point where the infrared energy starts to behave more like light energy. Instead of being absorbed as usable heat, it is reflected.

All non-transparent objects absorb infrared to varying degrees depending on the color, composition, surface texture and the wavelength of the infrared energy striking the surface. The temperature of the infrared source determines the wavelength. The higher the temperature, the shorter the wavelength. Short wavelength radiation generated by high temperature sources is more readily reflected by light colored objects. With long wavelength radiation generated by low temperature sources, the color has very little bearing on the amount of radiation reflected. Absorption is the opposite of reflection; therefore, the amount that is not reflected is absorbed.

If we have two identical objects sitting in the sun, one painted white and one painted black, the black object will absorb more of the high temperature radiation and become hotter. If we take the same two objects and put them under low temperature infrared heaters, each will absorb almost equal amounts of radiation and end up at very nearly the same temperature.

It has been determined that infrared energy with wavelengths from 2.4 microns to 20 microns is more readily absorbed by common construction materials. An object heated to 1700ºF generates infrared in the 0.75 to 20 micron range and peaks at 2.41 microns. Infrared energy with wavelengths less than 0.8 microns is actually visible light. Heating an object to a higher temperature will generate more infrared, but in the process, it also generates a greater percentage of visible light that contributes very little toward comfort heating.



ELECTROMAGNETIC SPECTRUM

B. HOW DO WE HEAT WITH INFRARED?

Infrared energy, upon striking an object, excites the molecular particles on the surface. The activity of these molecular particles generates heat, which in turn transfers by conduction through the object being heated. This object then becomes a radiant emitter but at a much lower temperature and intensity. All objects above the temperature of absolute zero emit infrared radiation, with temperature and emissivity (radiant efficiency) being the determining factors.

If two objects are placed in close proximity to one another, they will exchange radiant energy, with the higher temperature object radiating heat to the lower temperature object. Heat always flows from hot to cold, so if we place a high temperature infrared heater above a cold concrete floor, the infrared heater will transfer heat energy to the cold concrete until it becomes warm. The machinery and equipment at floor level also become warm from direct radiation and by conduction through direct contact with the floor slab.

The floor slab then becomes a giant low-temperature radiant emitter, radiating to the sidewalls and roof structure until they reach thermal equilibrium with the floor slab. The rate of heat loss through these structures will determine the temperature. Since the temperature differential is the driving force, this will determine the rate of radiant energy transfer from the floor slab.

An interesting aspect of this type of heating is the convected heat transfer from the warm floor slab and machinery to the cool air lying at floor level. We all know that warm air rises, so as the cool air sweeping across the floor picks up convected heat and rises, it is displaced by cooler air in a continuous cycle that gradually raises the air temperature in the building to a comfortable level. It is also interesting to note that gas-fired infrared heating systems are normally controlled by standard air temperature sensing thermostats, located at normal breathing level and shielded from the direct radiation of the heater.

Since a temperature differential is required to transfer heat from one medium to another, the floor slab must be at a higher temperature than the thermostat setting in order to elevate the air temperature sufficiently to satisfy the thermostat. Normally, the floor slab temperature will be 5º to 10º warmer than the thermostat setting. One reason for locating the heaters near the perimeter of a building is to heat that portion of the floor slab a little warmer. This warms the cool air infiltrating through the outside walls a little faster.

The major difference between a gas-fired infrared heating system and a forced hot air heating system is the method used to create a comfortable temperature. Gas-fired infrared heats the floor slab and machinery first and then the air. Personnel working in the comfort zone are blanketed by direct radiation from above, secondary radiation from below and warm air rising from the floor. In addition, personnel are in direct contact with the warm floor and all of the tools and machinery are warm to the touch.

In a forced hot air system, where the hot air rises to the ceiling and stratifies, gradually working its way down to thermostat level, the floor slab never becomes warm enough to be comfortable. It literally acts as a heat sink, draining heat from the air and personnel standing on the floor.

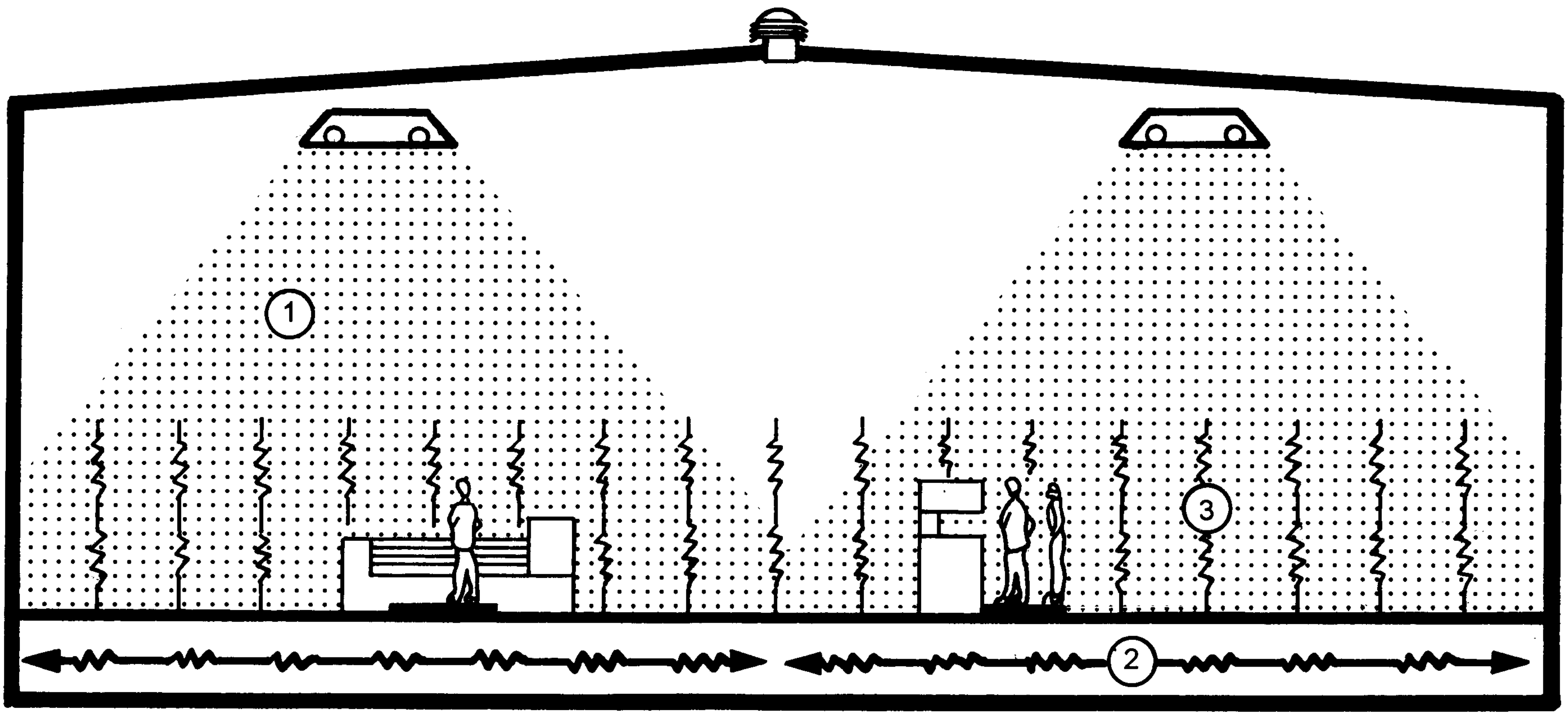
In many instances, a building heated with forced hot air will have a very high temperature differential between the floor and ceiling. The ceiling area of a high bay building can easily be 30º to 40º warmer than the floor area. Have you ever tried to heat a two-story house with a floor furnace?

In the same type building heated with an infrared system, the temperature is much more uniform. Direct radiation, indirect radiation, conduction and convection maintain a comfortable floor temperature. It is not uncommon to find a high bay building heated with infrared where the loft or roof area will be at a slightly lower temperature than the floor level … a good condition for minimizing heat loss.

Simply stated: A forced hot air system heats from the top down, making the floor the last (and most difficult) zone to heat. An infrared system heats from the floor up, satisfying the comfort zone (work area) first.

With these basic facts presented to us, is it any wonder that reported claims of 30% to 70% fuel savings over forced hot air heating systems are rather common?

MAIN HEAT TRANSFER METHODS



1) Direct radiant heat downward from heaters.
2) Horizontal conduction within floor slab.
3) Indirect radiant heat and convection upward from warm floor slab.

C. INFRARED HEAT FOR BODY COMFORT

Infrared warms the individual directly, but does not directly warm the air. To warm air that in turn warms the individual requires more energy. Remember, hot air rises…heat does not. Most people believe that the temperature of the air around them determines if they are warm or cold. This is only partly true. Try standing in the sun on a cold, calm day without your coat. You will find that you are still comfortable. However, if a cloud comes between you and the sun, you will feel a chill. The air temperature has not changed, but the amount of radiant energy that you were receiving from the sun has been reduced.

There are three modes of heat transfer: (1) **Conduction,** which is when one object transmits heat to another by contact, such as a pan on a stove; (2) **Convection,** which is when air is warmed by passing over a hot object, such as a forced air furnace; and (3) **Radiation,** when warm objects radiate heat to colder objects, such as the sun to the earth.

How does this relate to people and infrared? How can a body with a 92ºF surface temperature be kept comfortable in a 70ºF environment? The body is a machine that gives off heat. It just so happens that it is most comfortable in approximately a 20ºF temperature differential. The average person gives off approximately 400 Btu's per hour under these conditions. To achieve this, the heat is given up by a body in four ways.

Convection: The amount of convected heat lost depends upon the temperature difference between the body and the surrounding air. This temperature difference creates convection currents around the body. The greater the temperature difference, the greater the loss.

Radiation: Radiation losses are proportional to the temperature differential raised to the fourth power. Small differentials, when raised to the fourth power, have a big effect. Standing near a cold surface (such as a window) gives you a sensation of cold because your radiant heat loss goes up drastically.

Evaporation: The process of evaporation removes heat from the skin by vaporizing perspiration. Water is also vaporized from the lungs when breathing.

Conduction: Conduction losses are very small since people seldom stay in one spot long enough to lose very much heat by contact. The effects of conduction are most noticeable when you stand on a cold floor. You get "cold feet."

SUMMARY

Heat loss by conduction is negligible and evaporation stays relatively constant over a wide range of conditions. Therefore, body heat loss can be controlled by varying convection and radiation transfer. It is only the net heat loss that counts, and when this is under control, comfort exists.

Convection heat loss can be controlled by varying the velocity and/or temperature of the surrounding air. In well-insulated, tightly constructed, low ceiling buildings, this is not difficult. The velocity of air can be low and stratification is of very little consequence. High ceilings and/or uncontrolled air changes complicate the situation. Stratification becomes a factor, creating a need for more and more air to be heated hotter and hotter.

Infrared helps control body heat loss by surrounding it with warm surfaces. Air is a poor absorber; therefore, it absorbs very little energy from the emitter source. The energy (heat), therefore, is not dispersed into the upper regions of a building … stratification is less. The energy warms objects and floor below. The objects and floor in turn radiate heat. The air (at floor level) is warmed (by convection) from the objects and floor, providing a comfortable "occupied zone." Comfort can be maintained with lower air temperatures that will reduce infiltration and heat loss through the walls and roof.

D. ADVANTAGES OF HEATING WITH GAS-FIRED INFRARED

The advantages of heating with gas-fired infrared are as numerous as they are varied:

1. **A Large Variety of Models and Heat Exchanger Configurations Are Available …** With only a few exceptions, Space-Ray gas-fired infrared heaters can be applied effectively to most industrial and commercial heating needs. This allows you to design a heating system tailored to meet your customer's needs.

2. **Can Be Applied To All Types of Buildings … From Low Roof Warehouses To High Bay Industrial Buildings …** Despite the type of building to be heated, from low roof warehouses (12 ft. and up) covering acres of ground to high bay industrial buildings with 60 to 90 ft. ceilings, gas-fired infrared can usually be applied. This results in replacing old outmoded boiler and hot air systems with a far more efficient heating system, offering drastic reductions (up to 70% in very large buildings) in fuel consumption.

3. **Rust and Corrosion Can Be Eliminated on Expensive Sheet and Bar Stock …** Expensive raw material can be safely stored for long periods by setting up an infrared heating system to maintain the stored material temperature well above the dew point temperature. This system can eliminate the thousands of dollars lost annually due to rust and corrosion forming on precision machined parts, sheet steel and coil stock.

4. **Increased Worker Efficiency At Lower Fuel Consumption …** One or more Space-Ray infrared heaters can be located around a piece of operating machinery to maintain worker efficiency without regard to adjacent non-productive plant areas. This results in increased worker efficiency at the lowest possible fuel consumption.

5. **Sweating and Condensation Control …** Sweating and condensation forming on loading docks and on the inside of doorways can be eliminated, resulting in safer operation of forklifts and other hard rubber tired vehicles.

6. **<u>Spot and Zone Heating</u> …** You do not have to heat the entire building for the sake of a couple of production areas. Production and shipping areas can be zoned off to effectively heat only those areas requiring workable conditions. Warehouses can be zoned off to maintain above freezing temperatures for water-based paints, beverages and other commodities that can be damaged by freezing.

7. **<u>Quick Recovery of Heat Loss</u> …** Instead of losing all the building heat each time large overhead doors are opened, gas-fired infrared stores an enormous amount of heat energy in the floor slab. This helps the heating system to recover normal working temperatures faster and helps to conserve fuel.

8. **<u>Heats The "Unheatables"</u> …** Regardless of whether you are heating an automotive garage, a truck, bus, heavy equipment shop, or an aircraft hangar, gas-fired infrared does it better. Personnel working in, around and under the vehicles or airplanes are comfortable, and you save on fuel costs.

E. TYPES OF GAS-FIRED INFRARED HEATERS

There are several different types of gas-fired infrared heaters, each of them with a definite place in the heating industry. In many cases, their features and functions overlap, making it desirable to use a combination of the different types to achieve maximum efficiency.

The radiant tube heater is a very unique gas-fired infrared heater that may be used for a wide variety of applications. This unit uses an aluminized steel tube, calorized for high radiant efficiency. It is heated internally by a gas flame, with the hot gases pulled through the tube by a draft inducer. The pull-through system provides increased radiant efficiency and greater safety. The radiant tube heater may be used vented, unvented (indirect vented) or common vented and the design also makes it possible to duct fresh outside air into the unit in case of extremely dirty or high humidity atmospheres. These heaters may be mounted at different angles and different heights. Some large capacity units have been mounted successfully as high as 68' above finished floor level.

The high temperature, high intensity units have a porous ceramic emitter designed to operate in the 1700ºF to 1750ºF temperature range. These units may be suspended hanging horizontally to direct the radiation straight down or at an angle to direct the radiation into a designated area. With the reflector options available, they may be mounted at selected mounting heights and angles for a wide variety of heating applications. These units are primarily suited for spot and area heating.

The floor model infrared heaters have a perforated stainless steel emitter formed into a vertical cylinder. These units are designed primarily for spot heating and/or warming stations. An optional aluminum reflector is available to focus the radiation into a designated work area and to improve the performance where wind or strong drafts are a factor.

Space-Ray manufactures all of the different types of gas-fired infrared heaters listed above. All models are available for either natural or propane gas.

APPLIED MODEL HEATERS

RSTP SERIES

Low Intensity Radiant Tube Heater — 950ºF
150,000 Btu/hr and 175,000 Btu/hr

- Industry's workhorse since 1968
- The original unitized infrared tube heater design certified by CSA
- Completely factory assembled and tested. No field assembly required
- Proven in high bay installations . . . As high as 68' above the floor
- Ideal for large buildings, aircraft hangars, warehouses, steel mills and other hard-to-heat industrial heating applications
- Certified for indoor and outdoor installation
- High radiant efficiency
- Calorized-aluminized steel emitter and aluminized steel body construction
- Suitable for horizontal or angle mounting up to 90º

LTU/LTS SERIES

Low Intensity Radiant Tube Heater — 750ºF
38 Models, from 40,000 Btu/hr through 250,000 Btu/hr.

- Pull through system
- Completely factory pre-assembled for easy installation
- Calorized-aluminized steel emitter
- Highly efficient aluminum reflector design … ends are enclosed for maximum radiant heat output and minimum convection loss

ETU/ETS SERIES

Low Intensity Radiant Tube Heater — 700ºF
40,000 Btu/hr through 250,000 Btu/hr.

- Pull through system
- Contractor's choice series for easy field assembly and installation
- Ideal for light commercial and industrial applications
- Calorized-aluminized combustion chamber (10 feet) and heavy-duty hot-rolled steel emitter tube or the ALC Option.

PTU/PTS SERIES

Low Intensity Radiant Tube Heater — 700ºF
40,000 Btu/hr through 200,000 Btu/hr.

- Push through system
- Burner operates much quieter (less than 50db)
- External 24V connection
- Ideal for car washes and high humidity environments with the ALC option.

CB SERIES

Low Intensity Radiant Tube Heater
4 Models from 20,000 Btu/hr to 50,000 Btu/hr

- Pull through system
- Factory pre-assembled for easy installation
- Ideal for residential garages, workshops, small unheated areas as well as light commercial applications

RSCA SERIES

High Intensity Ceramic Heater — 1750ºF
26,000 Btu/hr • 52,000 Btu/hr • 104,000 Btu/hr

- Porous ceramic emitter surface with high emissivity coating
- Aluminized steel body construction
- Incoloy 800 reverberatory screen for increased radiant efficiency and safety
- Choice of controls: millivolt standing pilot or direct spark ignition
- One-piece cast iron Venturi
- Suitable for horizontal or angle mounting up to 45º

DK SERIES

High Intensity Ceramic Heater — 1800ºF
14 Models from 30,000 Btu/hr to 160,000 Btu/hr

- Special honeycomb tile design for increased radiant output
- Suitable for angle mount up to 35º
- Equipped with 100% gas shut-off safety control
- 24-volt direct spark ignition system
- Natural or propane gas
- Indirect vented operation

FLOOR MODEL HEATERS

RFPA SERIES

Medium Intensity Floor Model Heater - 1400ºF
100,000 Btu/hr • 212,000 Btu/hr • 250,000 Btu/hr

- Perforated stainless steel emitter
- Heavy gauge steel body construction
- Cast iron multi-port burner
- Optional reflector for directional heating

II. FEATURES, FUNCTIONS, AND BENEFITS OF GAS-FIRED INFRARED HEATING

Gas-fired infrared heating systems have many advantages over all other types of heating systems and are unsurpassed for their versatility, efficiency and economy.

They feature the unique ability to:
1. Efficiently heat large, open, high-bay buildings;
2. Maintain worker comfort at lower inside air temperatures;
3. Deliver the heat to floor level from 6' to 30' (and higher on specific applications) without the mechanical force needed to drive heated air to floor level;
4. Take advantage of nature by delivering the major portion of the available heat energy to the floor where it is released into the occupied areas:
   a) By low temperature radiation (from the warm floor slab),
   b) By conduction to all machinery and personnel in direct contact with the floor,
   c) By convected heat to warm the cool incoming air sweeping across the floor.

Gas-fired infrared is a relatively new concept of heating, using almost 100 percent of the available heat energy of clean burning natural or propane gas in a heating unit specifically designed to produce the most efficient and economical radiant heating source for industrial and commercial applications.

Infrared heats the floor slab and machinery first. Instead of acting as a heat sink, the floor slab stores an enormous amount of thermal energy and becomes a giant low temperature radiant emitter, radiating to the sidewalls and roof structure until they reach thermal equilibrium with the floor slab. Cool air is warmed as it sweeps across the warm floor slab and the machinery, and then it rises, being displaced by cooler air in a continuous cycle that gradually raises the air temperature in the building to a comfortable level. Personnel working in the comfort zone are blanketed by direct radiation from above, secondary radiation from below, and warm air rising from the floor.

Heat always flows from hot to cold; therefore, to transfer heat to the building enclosure and incoming cold air, the temperature of the heat exchanger (floor slab) must always be higher than the inside design temperature. In an infrared system, the floor slab temperature will always be several degrees warmer than the air temperature with the following exceptions:
1. When starting with a cold building, and
2. Under certain spot heating conditions where the radiant heaters are used to spot heat personnel.

Infrared heat does not heat air directly, but indirectly <u>by convection from the warm floor slab and machinery</u>. The warm slab not only heats the air by convection but also serves to elevate the mean radiant temperature (MRT) of the building, and supplies heat to the personnel by radiation and conduction. The mean radiant temperature (MRT) is a term used to describe the radiant temperature for an enclosure where all objects within the enclosure, including the outside walls and roof, are emitting and receiving sufficient radiation to sustain their rate of heat loss.

The human body will ultimately reach an average condition of thermal equilibrium depending upon the heat balance established for the body and its environment. The requirements to maintain an adequate heat balance, MRT and air temperature depend largely upon the metabolic rate (body heat production - work rate) and clothing.

Since the purpose of infrared heating is to provide radiant energy in sufficient quantity to maintain an adequate heat balance between the workers and their environment, reasonable care must be exercised to maintain a reasonably uniform radiant flux density where radiant heat energy can offset the human body's radiation losses to colder surfaces (cold outside walls) and compensate for lower air temperatures. Infrared heat, when used to spot heat individual workstations, does little to raise the mean radiant temperature (MRT) but will compensate for the individual's convection losses due to cold air movement across the body, except in extreme cases.

The air temperature found to be satisfactory in the majority of industrial and commercial applications is 65ºF (and lower, depending on the workers' metabolic rates); whereas the mean radiant temperature (MRT) will be determined by the building heat loss (by ASHRAE methods) at the design temperature conditions. Naturally, if a particular building has an abnormally high heat loss (uninsulated metal walls, roof, etc.) and has a high rate of air change (process exhaust, welding hoods, etc.), then the heat loss will be high and the mean radiant temperature will be high relative to the air temperature.

The Raber-Hutchinson comfort equation, familiar to the radiant heating industry, theorizes the following:

$$\boldsymbol{T_a + \mathrm{Tmrt} = 140}$$

Where:	*Ta*	=	Temperature of ambient air
	Tmrt	=	Mean radiant temperature of surrounding surfaces
	140	=	Figure derived from considerable experimentation involving human subjects in a precisely controlled laboratory environment

The Raber-Hutchinson comfort equation suggests that a perfect balance is reached at 70ºF mean radiant temperature (MRT). For each 1ºF decrease in air temperature, the MRT must be raised 1ºF to maintain comfort conditions.

Ideal comfort conditions are extremely difficult to maintain in an industrial complex due to changing outdoor climatic conditions, workloads, door openings, differences in building size and construction, and other building operating factors. Gas-fired infrared heaters come closer to providing ideal comfort conditions at a lower operating cost than any other type of heating system.

The Raber-Hutchinson comfort equation gives us a clue to the reasons for the effectiveness of gas-fired infrared heating. The ideal inside design temperature is 65ºF. The floor slab temperature must be warmer than the design temperature in order for the air to absorb heat and maintain design conditions. Assuming the slab is 10ºF warmer than the air temperature: 65º air + 75º slab = 140º.

In addition to this, the individual workers are receiving direct radiation from the infrared heaters suspended overhead, and the individual workers are generating body heat at the rate of 200 to 800 Btu/hr per person, depending on their work rate. Another factor to consider is that the hot flue gases rising from indirectly vented heaters serve to offset the heat loss of the roof areas.

All of the above factors contribute to the overall efficiency of gas-fired infrared heaters and have the effect of over-compensating for the reduced air temperature by at least 15 percent in typical applications. This is referred to as the infrared compensation factor. A standard ASHRAE heat loss is computed and the resultant conduction and air change heat loss is multiplied by 0.85 (1.00 - 0.15). In some instances, the over-compensation is even higher than 15 percent. Lower infrared compensation factors may be used at lower mounting heights. (See *Infrared Heat Loss Compensation Factor* chart in Section IV.)

Remember that all of the radiant heat output from an infrared heater is absorbed by intercepting surfaces and ultimately dissipated to the building enclosure by re-radiation, conduction and convection.

The benefits derived from gas-fired infrared heating are numerous:

1. <u>Economical to Operate</u>

a) Delivers the heat to floor level where it can be utilized and stored.
b) Lower inside design temperature.
c) Infrared compensation factor of 0.85 (or less — see Section IV-C, *Infrared Heat Loss Compensation Factor*).
d) Minimum power requirements … electrical power is required only for burner ignition, the gas valve and the draft inducer (where applicable).

All of the above add up to a fuel savings of 30 to 70 percent over conventional warm air systems. Even greater fuel savings (50 to 70 percent) can be realized over some of the older, out-of-date boiler systems.

2. <u>Economical to Install and Maintain</u>

a) Factory assembled and pre-assembled units require minimum labor to install and maintain.
b) No filters, no fans and no lubrication. Infrared units require only an occasional cleaning.
c) Where applicable, indirectly vented units require no vent pipe or duct work, further reducing the installation costs.

3. <u>Efficient, Clean and Quiet</u>

a) Gas infrared heaters warm up quickly, responding immediately to thermostatic control.
b) No need for noisy blowers, stirring up dust and dirt.

Gas-fired infrared heaters are best suited to industrial applications, factories, machine shops, aircraft hangars and certain commercial applications such as warehouses, automotive or truck repair and practically any type of fabrication shops, car washes, fire stations, gymnasiums, indoor tennis courts, swimming pools and greenhouses. In high humidity and contaminated atmospheres, direct outside combustion air may be ducted to tube type heaters to insure complete combustion.

Areas in which these units are **NOT** suitable are low roof areas (less than 8' high), wind tunnels (open breezeways), combustible atmospheres, certain hazardous locations and inside paint booths. Infrared is not ideal for door heating, although it is frequently used to effectively spot-heat work areas inside of doorways.

III. FUNDAMENTAL DETAILS FOR GAS-FIRED INFRARED HEATING

As with any type of heating system, the basic design data must be gathered and good engineering practices followed. The following questions and comments are submitted for consideration on each gas-fired infrared heating application:

- **WHAT DOES THE CUSTOMER EXPECT FROM THE INFRARED SYSTEM?**

Fuel savings, ability to heat large high bay buildings and worker comfort are the obvious advantages of gas-fired infrared heating.

Spot and area heating are also extremely practical; however, the designer must realize that total building heat requires a conventional ASHRAE heat loss for the building; whereas, spot or area heating involves determining the Btu/hr per square foot heat requirement based on mean radiant temperature, air temperature, air velocity and the metabolic rate of the personnel working in the area.

In most instances, when determining the requirements for spot heating areas greater than 40-50% of the total building floor area, the most feasible method is to first compute a conventional ASHRAE heat loss for total building heat. This will help to determine the temperatures that can be expected in the areas surrounding those you want to heat and help you to adjust the Btu/hr per square foot spot heating requirements accordingly.

Since air temperature and mean radiant temperature are the basic criteria for spot heating, and there is no way to prevent the heat from migrating to the adjacent unheated areas, any heat added to the building structure will usually increase both the air temperature and the mean radiant temperature, thereby reducing the Btu/hr per square foot radiant intensity required for comfort in the working environment. The charts for determining spot heating requirements are based on metabolic rate, mean radiant temperature, air temperature, and air velocity.

- **IS THERE SUFFICIENT CEILING HEIGHT TO APPLY GAS-FIRED INFRARED HEATING EQUIPMENT?**

High-bay areas are generally well suited for infrared heating. Low-roof buildings (less than 8 feet high) and low ceiling, multi-story buildings are not well suited for infrared heating.

- **WHERE CAN YOU LOCATE THE INFRARED UNITS?**

Locate the heaters as high as possible to provide uniform radiant energy distribution on the floor level. Whenever possible, horizontal installation is preferred over angle mounted infrared heater installation. In large manufacturing facilities, you may have to locate heaters above the traveling crane or locate smaller units along the sidewalls or suspended below the crane rail.

- **DO YOU FORESEE A POSSIBILITY OF CONDENSATION ON THE UNDERSIDE OF THE ROOF?**

Uninsulated steel roofs (bare metal), single glaze skylights, lack of suitable vapor barrier on the underside of the insulation and unsealed edges, holes or cuts in the vapor barrier all contribute to the possibility of condensation. It may not be feasible to insulate the roof, so the vented radiant tube heater would then be the ideal choice.

- **IS THERE SUFFICIENT OPEN AREA TO EFFECTIVELY HEAT THE OCCUPIED PORTION OF THE BUILDING?**

The radiation from the heaters must be able to radiate directly to the floor or machinery and personnel at floor level.

High-density warehouses and those with remote controlled stackers require the placement of the heaters at the perimeter or through the major aisles for sufficient radiation to reach the floor.

- **INQUIRE AND MAKE NOTE OF ANY INFLAMMABLE OR CORROSIVE VAPORS PRESENT IN THE BUILDING.**

The heaters must be located in a manner that meets all safety code requirements. If in doubt, an inquiry should be made directly to the customer’s insurance carrier.

Take note, gas-fired infrared heaters are no more hazardous in the presence of inflammable vapors than unit heaters.

IV. HEAT LOSS CALCULATIONS

A. BASIC CRITERIA FOR INFRARED HEATER APPLICATIONS

Prior to the design of any heating system, an estimate must be made of the maximum hourly heat loss based on the greatest temperature differential that could be expected during the heating season.

The following steps must be followed to determine basic design criteria:

1. **<u>DETERMINE BUILDING USAGE</u>:**
   a. Factory: raw materials and chemicals used, power exhaust flow rates
   b. Warehouse: stacking heights, aisle width, sprinkler system location
   c. Machine Shop: raw material storage
   d. Automotive Repair: lift locations, auto exhaust removal system
   e. Truck Repair: truck lanes, cab height when raised, trailer height, canvas covered trailers
   f. Aircraft Hangars: NFPA 409 requires 10' heater clearance above engine and fuel tank of highest aircraft to be housed. Also, determine clearance required for aircraft tail section.
   g. Woodworking, millwork and pre-fabricated structures: Determine capacity and location of dust collector system. Check for airborne sawdust at heater level that could clog burners.

2. **<u>DETERMINE DESIGN TEMPERATURE CONDITIONS</u>:**
   a. Inside design temperature (normally 65º for industrial and commercial buildings)
   b. Lowest outside design temperature to be expected during the heating season

3. **<u>MATERIAL STORAGE AND OTHER BUILDING CONDITIONS THAT WOULD HAVE A DIRECT BEARING ON HEATER SELECTION AND LOCATION</u>:**
   a. Material storage and height
   b. Sprinkler system
   c. Crane location, height and clearance above

4. **<u>CONTAMINANTS IN BUILDING ATMOSPHERE</u>:**
   Refer to Section V, *Ventilation for Indirect Vented Infrared Heaters*, Item D-12, *General Precautions*.

5. **<u>GAS AND POWER AVAILABILITY</u>:**
   a. Determine type of gas to be used, system pressure and meter location
   b. Determine availability of power

6. **<u>SELECT THE MOST LOGICAL CHOICE OF HEATER MODELS TO BE USED AND METHOD OF MOUNTING</u>.**

7. **<u>OBTAIN COMPLETE SET OF BUILDING PLANS</u>.** If not available, make complete sketches showing the following:

a. Walls: Material, height, construction and thickness. Determine insulation type and thickness. Make certain the insulation has a suitable vapor barrier, intact and sealed on all edges.
b. Foundation (extending above floor level): Construction material and thickness
c. Windows: Quantity, size, and location
d. Doors: Construction, quantity, size and location
e. Skylights (in roof sections only): Material, quantity, size and location. If the roof has blanket-type insulation, make certain the raw edges of the insulation around the skylights are securely taped and sealed.
f. Roof: Material, roof pitch, construction and thickness. Determine type of roof insulation and thickness. The vapor barrier must be securely sealed on all edges, and must face the heated space.
g. Floor construction: Preferably concrete. Wood, dirt and asphalt floors are not as efficient in producing the flywheel effect so noticeable with a concrete floor.

8. **DETERMINE AIR CHANGES REQUIRED AND BUILDING EXHAUST RATE:** Pay particular attention to dust collectors, paint spray booths, welding hoods, process exhaust, heat treating furnaces, baking ovens and any other exhaust system that draws air out of the building. Any air exhausted must be replaced with outside air.

9. **DETERMINE INTERNAL HEAT GAINS OR LOSSES.**
   a. Consider heat gains from furnaces, motors or other heat producing machinery.
   b. Do not forget extra heat losses from incoming material, stock movement and vehicles.

10. **CHECK LOCAL CODE REQUIREMENTS AND RESTRICTIONS.** Also, check the customer's insurance carrier. They also may have certain restrictions on various heating appliances.

B. OVERVIEW OF NATURAL INFILTRATION LOAD

Natural infiltration load can be estimated by the following two methods:

1. **ASHRAE CRACK METHOD:** This method is described in the ASHRAE fundamentals guide for various types of structural material and for varying wind velocities. The infiltration load is calculated based on the leakage of the windows, doors and walls. This method provides more realistic infiltration loads for small residential applications, whereas the use of the crack method for large industrial applications is considered time consuming and cumbersome. After the determination of the volume of warm air replaced per hour, the infiltration load will be calculated with the following formula:

$$\mathbf{IL = L \times Q \times 0.018 \times TD}$$

IL = Infiltration Load in Btu/hr
L = Length of the crack in feet
Q = Volume of air (CFM per ft)
TD = Inside design temperature less outside design temperature in °F

The volume of air (Q) can vary from 5 to 100 CFM per ft. of crack depending on the severity of the crack width and building material construction.

2. **<u>AIR CHANGE METHOD</u>:** The air change method simplifies the infiltration load estimates for large industrial and commercial buildings. The following tables can be used as a guideline in determining the estimated air changes per hour. After determining the air change, the infiltration load (IL) can be estimated using the following formula:

$$\mathbf{IL = V \times 0.018 \times TD \times AC}$$

IL = Infiltration Load in Btu/hr
V = Volume of the building in cubic feet
TD = Inside design temperature less outside design temperature in ºF
AC = Air Changes per hour

Using the air change method requires good judgment and field experience. It is the responsibility of the designer to evaluate the special circumstances that must be considered for each individual installation.

a. GENERAL GUIDELINES FOR NATURAL INFILTRATION AIR CHANGE BASED ON BUILDING USAGE.

TYPE OF BUILDING	CONSTRUCTION	FLOOR AREA *(SQ. FT.)*	CEILING HEIGHT *(FT.)*	TYPICAL NUMBER OF AIR CHANGES *MIN*	*MAX*
Aircraft Hangar	Average	10,000	25	1.00	2.50
	Good	20,000	40	0.75	2.00
	Good	50,000	70	0.50	1.50
Auto Dealership					
Service Area	Good	5,000	16	1.50	2.00
Parts Storage	Good	5,000	18	0.75	1.00
Car Wash	Good	1,000	12	3.00	3.00
Fire Station	Good	4,000	15	0.75	1.25
Loading Dock Area	Average	5,000	12	1.00	3.00
Warehouse/Distribution	Average	10,000	18	0.75	1.25
Center		20,000	24	0.50	1.00
(with multiple large doors)	Good	10,000	18	0.50	1.00
		20,000	24	0.50	0.75

This chart is intended as a guide only. Special circumstances specific to the application and the building usage can alter the estimated air change (infiltration load) of the building (e.g., broken windows, unanticipated number of door openings, undisclosed roof ventilators or exhaust system, and/or strong prevailing wind conditions with large openings on the opposite side). Generally, the minimum estimated air change should be satisfactory in estimating the infiltration air change. The heating system designer must assume full responsibility for selecting and adjusting the values to be used.

For other types of buildings, please refer to the *Natural Infiltration Air Change based on Building Volume* chart.

b. NATURAL INFILTRATION AIR CHANGE BASED ON BUILDING VOLUME

The following chart may be used to estimate the infiltration air change based on building volume. This chart was devised on the theory that a small building will have a higher rate of infiltration than a large building. In a small building, the ratio of door and window area compared to total wall area is much greater than the large building. This chart does not take into account building usage.

BUILDING VOLUME (CU. FT.)	INFILTRATION AIR CHANGE/HR
5,000	1.00
10,000	0.90
20,000	0.80
40,000	0.70
80,000	0.60
100,000	0.55
200,000	0.50
500,000	0.35
1,000,000	0.25
2,000,000	0.15
5,000,000	0.10

This generalized information is provided based on field experience. The heating system designer must assume full responsibility for selecting and adjusting the values to be used.

C. INFRARED HEAT LOSS COMPENSATION FACTOR

Thirty-plus years of Space-Ray field experience, as well as published ASHRAE papers, indicates that it is necessary to adjust the heat loss when sizing an infrared heating system. It is a fact that the intensity of the radiation is inversely proportional to the square of the distance; therefore, it is recommended that the computed heat loss be multiplied by the following infrared heat loss compensation factors:

MOUNTING HEIGHT (FT)	COMPENSATION FACTOR
16	0.80
18	0.81
20	0.82
22	0.83
24	0.84
26	0.85
28	0.86
30	0.87
32	0.88
34	0.89
36	0.90
38	0.91
40	0.92

MOUNTING HEIGHT (FT)	COMPENSATION FACTOR
42	0.93
44	0.94
46	0.95
48	0.97
50	1.00
52	1.02
54	1.04
56	1.06
58	1.08
60	1.10
62	1.12
64	1.14
65	1.15

Please choose the appropriate factor based upon the mounting height or choose 1.0.

D. RADIANT VS. CONVECTION HEATING

An infrared heating system is always sized at a lower input capacity when compared to forced air (convection) heating. This is due to different modes of heat transfer (radiation vs. convection), thermal mass, and minimal stratification between ceiling and floor temperatures. For retrofit purposes, provided the unit heaters are maintaining the desired inside design temperature at ASHRAE design conditions, the following reduction can be utilized when recommending an infrared heating system:

Type	Thermal Efficiency	% Reduction in System Sizing
High Efficiency Unit Heater	80%	32%
Convectional Unit Heater	62%	48%

Assuming that the ASHRAE heat loss for a building is 100,000 Btu/hr, then the heater selection for this building would be as follows:

	Infrared Heater	Unit Heater
Building Heat Loss:	100,000 Btu/hr	100,000 Btu/hr
Infrared Compensation Factor[1]: *(for radiant heating)*	0.85	—
Thermal Efficiency[2]: *(for convection heating)*	—	80%
Heater Input Required:	85,000 Btu/hr (100,000 x 0.85)	125,000 Btu/hr (100,000 ÷ 0.80)

1 Infrared heat loss compensation factor based on 26’ AFF mounting height (see Section C).
2 High Efficiency Unit Heater for comparison purposes.

NOTE: Based on the different modes of heating (e.g., radiant vs. convection), heating systems for infrared can be sized 32% lower input capacity than convection heating systems.

E. HEAT LOSS DUE TO COLD MASS

The heating system designer should consider the effects of cold mass brought into a heated building, such as large trucks in a repair garage, cold steel in a manufacturing environment, and cold airplanes in an aircraft hangar. The heat absorbed in raising the cold mass to room temperature must be added to the heat loss.

The heat absorbed by cold mass can be estimated with the following formula:

$$\mathbf{A = M \times Cp \times (Ti - Tm)}$$

A= Heat absorbed in BTU
M = Average weight of material brought into space in lbs
Cp = Specific heat of material in Btu/lb per ºF
Ti = Inside design temperature (ºF)
Tm = Temperature at which material enters the building (ºF)

The specific heat of some common materials is:

MATERIAL	SPECIFIC HEAT
Aluminum	0.20
Cement	0.19
Concrete	0.16
Steel	0.12

MATERIAL	SPECIFIC HEAT
Cast Iron	0.11
Copper	0.09
Glass	0.16
Wood	0.51

If the desired recovery or warm-up time is less than one hour, then the heat absorbed (A) should by multiplied by 60 minutes divided by the desired recovery time in minutes desired (e.g., desired recovery time is 30 minutes, therefore 60 minutes ÷ 30 minutes = 2). If the desired recovery or warming time is more than one hour, then heat absorbed (A) should be divided by hours required.

For example, if you bring 5000 lbs. of cast iron that is at a temperature of 10ºF into a building with an inside design temperature of 65ºF, the additional BTU needed to bring the cold mass to design temperature is computed as follows:

A = M x Cp x (Ti - Tm)
A = 5000 x 0.11 x (65 - 10) = 30,250 BTU

If the desired recovery time is 2 hours: 30,250 ÷ 2 = 15,125 Btu/hr

If the desired recovery time is 15 minutes: 30,250 x (60 ÷ 15) = 121,000 Btu/hr

F. "R" AND "U" VALUES

R-value is the thermal resistance of materials under steady state conditions. The higher the R-value of the insulation, the better the resistance to heat transfer. For example, an insulation material with an R-value of 11 is better than an R-value of 6. R-values are used only to obtain the U-values.

U-value is the thermal transmittance of materials under steady state conditions. The "U" factor is usually the value of a combination of materials such as exterior wall surface (e.g., metal building), insulation in the wall (e.g., 2" vinyl covered blanket insulation or air space with insulation in a concrete wall), and interior wall covering (e.g., plywood sheets, etc.). A Table of "U" Factors for different building material combinations is provided on the following page.

If an R-value of a building material is known, it can be easily converted to a U-value. For example:

R-value = 20
U-value = 1/R-value
U-value = 1/20 = 0.05

G. OUTSIDE DESIGN TEMPERATURES

Outside design temperatures and degree day charts for various cities throughout the United States may be found in Section VIII, *Estimating Fuel Consumption.*

H. TABLE OF "U" FACTORS — INDUSTRIAL HEATING

WALLS — LIGHT INDUSTRIAL

MATERIAL	TYPE OF INSULATION	THICKNESS OF INSULATION					
		0 IN.	*½ IN.*	*1 IN.*	*1½ IN.*	*2 IN.*	*3 IN.*
Flat Metal	Blanket or Batt	1.20	0.37	0.22	0.16	0.12	0.09
Flat Metal	Blown Polyurethane	1.20	0.26	0.15	0.10	0.08	0.06
Cement Board, 3/8"	Blanket or Batt	1.06	0.36	0.22	0.15	0.12	–
Plywood or Wood Panel, 3/4"	Blanket or Batt	0.56	0.28	0.18	0.14	0.11	–

WALLS — MASONRY

MATERIAL		THICKNESS OF MATERIAL		
		4 IN.	*8 IN.*	*12 IN.*
Brick (Face & Common)		–	0.48	0.35
Brick (Common only)		–	0.41	0.31
Stone (Lime & Sand)		–	0.67	0.55
Hollow Concrete Block:	Gravel Agg.	–	0.51	0.47
	Cinder Agg.	0.51	0.39	0.37
	Light Wt. Agg.	0.43	0.36	0.33
Solid Concrete Block:	Gravel Agg.	–	0.52	0.47
	Cinder Agg.	–	0.39	0.36
	Light Wt. Agg.	–	0.35	0.32
Veneer Wall (4" Face Brick, 4" Stone or 4"- 6" Precast Concrete)				
	Sand Agg.	0.49	0.41	0.38
	Cinder Agg.	0.41	0.33	0.31
	Light Wt. Agg.	0.35	0.30	0.28
Poured Concrete:	30#/Cu. Ft.	0.19	0.10	0.07
	80#/Cu. Ft.	0.42	0.25	0.18
	140#/Cu. Ft.	0.82	0.60	0.47

WALLS – INTERIOR

For walls against heated space, use 0.02

ROOFS

MATERIAL	TYPE OF INSULATION	THICKNESS OF INSULATION					
		0 IN.	*½ IN.*	*1 IN.*	*1½ IN.*	*2 IN.*	*3 IN.*
Built-up Metal Deck	Preformed	0.90	0.40	0.26	0.19	0.16	0.11
Built-up Wood Deck, 1"	Preformed	0.48	0.29	0.21	0.16	0.14	0.10
3"	Preformed	0.23	0.17	0.14	0.12	0.10	0.08
Concrete Slab, 2"	Preformed	0.30	–	0.16	0.13	–	–
4"	Preformed	0.18	–	0.12	0.10	–	–
Gypsum Slab, 2"	Preformed	0.36	–	0.20	0.15	–	–
4"	Preformed	0.25	–	0.16	0.13	–	–
Metal	Blanket or Batt	1.20	0.37	0.22	0.16	0.12	0.09
Wood, 1"	Blanket or Batt	0.48	0.26	0.17	0.13	0.11	0.08
2"	Blanket or Batt	0.32	0.20	0.15	0.11	0.10	0.07

WINDOWS/SKYLIGHTS

Glass — Vertical Outdoor Exposure	1.13
Glass — Horizontal Outdoor Exposure	1.22
Insulated Glass (Double) — 3/16" Air Space	0.75
1/2" Air Space	0.66
Glass Block	0.51
Plastic Bubbles (Skylights) — Single Wall	1.15
Double Wall	0.70
Fiberglass (Translucent Panels)	1.09

DOORS

Steel — Single Sheet	1.20
Steel — Insulated	0.75
Thermospan	0.15

FLOOR SLAB

Uninsulated Slab Edge	0.81
Insulated Slab Edge	0.55

I. HEAT LOSS ESTIMATING SHEET

1. Transmission Losses

Wall	Building Material	Size	Net Area or Length X	"U" X	TD* =	Btu/hr	Total Btu/hr
N O R T H							
E A S T							
S O U T H							
W E S T							
R O O F							
						TOTAL:	

2. Mechanical Exhaust Losses (Existing or specified ventilation that will be used in conjunction with heating system)

Exhaust	CFM X	60 X	0.018 X	TD*	Btu/hr	
		60	0.018			
		60	0.018			

3. Infiltration Losses

Length X	Width X	Avg. Height X	Air Change X	0.018 X	TD*	
				0.018		

4. Heat Loss Due to Cold Mass

Lbs. Material X	Specific Heat X	TD* ÷	Warm-Up Time =	

5. Heat Gain

5. Heat Gain	(Subtract)	–
* Difference between outside and inside design temperatures. Total Heat Loss:		
Infrared Compensation Factor:		x
Infrared Heat Required:		

J. INFORMATION FOR SAMPLE HEAT LOSS ESTIMATION

A. BUILDING DATA

Building Location:	Lincoln, Nebraska
Building Usage:	Warehouse
Building Age:	5 Years
Building Type:	Metal
Building Length:	240’
Building Width:	100’
Building Overview:	10 Overhead Doors
	10 Windows
	10 Skylights
	4 Entry Doors

B. BUILDING CONSTRUCTION DATA

Walls:	Metal with 3” Fiberglass Insulation
Wall Height:	Eave = 20’
	Peak = 30’
	Average Roof Height = 25’
Foundation:	4’ High 12” Concrete Block
Floor:	Concrete Slab, Insulated Slab Edge
Roof:	Flat Metal with 3” Fiberglass Insulation
Entry Doors:	7’ x 3’, Uninsulated Metal (Total of 4)
Overhead Doors:	12’ x 12’, Uninsulated Metal (Total of 10)
Skylights:	8’ x 3’, Double Wall Bubble (Total of 10)
Windows:	4’ x 4’, Insulating Glass, 1/2” Air Space (Total of 10)

C. BUILDING VENTILATION DATA

Existing Power Ventilation:	2500 CFM
Existing Gravity Ventilation:	None
Internal Heat Loss/Gain:	None
Estimated Air Change for Infiltration Heat Loss:	0.60/Hour
Additional Air Changes Required for Building Ventilation:	None

D. BUILDING OPERATING CONDITIONS

Inside Design Temperature:	65ºF
Outside Design Temperature:	–5ºF
Night Setback Inside Design Temperature:	50ºF
Hours Per Day at Setback:	12
Days Per Week at Normal Building Operating Conditions:	5

E. TYPE OF FUEL AVAILABLE

Type of Fuel:	Natural Gas
Fuel Cost:	$0.60/Therm
Type of Infrared Heater:	Vented Tube Heater
Heater Mounting Height:	

K. HEAT LOSS ESTIMATING SHEET — SAMPLE

1. Transmission Losses

Wall	Building Material	Size	Net Area or Length	X "U"	X TD *	= Btu/hr	Total Btu/hr
NORTH	Metal	240x16	3,840	0.09	70	24,192	
	3 Rollup Doors	12x12	432	1.20	70	36,288	
	2 Entry Doors	7x3	42	1.20	70	3,528	
	5 Windows	4x4	80	0.66	70	3,696	
	Foundation	240x4	960	0.33	70	22,176	
	Floor Slab Edge	2401	240	0.55	70	9,240	99,120
EAST	Metal	100x26	2,600	0.09	70	16,380	
	2 Rollup Doors	12x12	288	1.20	70	24,192	
	Foundation	100x4	400	0.33	70	9,240	
	Floor Slab Edge	100	100	0.55	70	3,850	53,662
SOUTH	Metal	240x16	3,840	0.09	70	24,192	
	5 Rollup Doors	12x12	720	1.20	70	60,480	
	1 Entry Door	7x3	21	1.20	70	1,764	
	5 Windows	4x4	80	0.66	70	3,696	
	Foundation	240x4	960	0.33	70	22,176	
	Floor Slab Edge	240	240	0.55	70	9,240	121,548
WEST	Metal	100x26	2,600	0.09	70	16,380	
	1 Entry Door	7x3	21	1.20	70	1,764	
	Foundation	100x4	400	0.33	70	9,240	
	Floor Slab Edge	100	100	0.55	70	3,850	31,234
ROOF	Metal	240x100	24,000	0.09	70	151,200	
	10 Skylights	8x3	240	0.70	70	11,760	
							162,960
						TOTAL:	468,524

2. Mechanical Exhaust Losses (Existing or specified ventilation that will be used in conjunction with heating system)

Exhaust	CFM	X 60	X 0.018	X TD *	= Btu/hr	
5 @ 500 cfm	2500	60	0.018	70	189,000	
		60	0.018			189,000

3. Infiltration Losses

Length	X Width	X Avg. Height	X Air Change	X 0.018	X TD *	
240	100	25	0.35	0.018	70	264,600

4. Heat Loss Due to Cold Mass

Lbs. Material	X Specific Heat	X TD *	÷ Warm-Up Time	=

5. Heat Gain

	(Subtract)	–
	Total Heat Loss:	922,124
	Infrared Compensation Factor:	x 0.81
	Infrared Heat Required:	746,920

* Difference between outside and inside design temperatures.

L. SAMPLE COMPUHEAT ANALYSIS

SPACE-RAY
INFRARED GAS HEATERS

COMPUHEAT BUILDING HEAT LOSS & DESIGN ANALYSIS

Date: 05-28-2013 Time: 13:58:06 Prepared by: RJ Herrington

Job Name: Warehouse - Lincoln, NE

2 Wall(s)	16 ' high	240 ' wide	16 ' peak	7680 sq ft	U = 0.09
2 Wall(s)	16 ' high	100 ' wide	26 ' peak	4200 sq ft	U = 0.09
2 Wall(s)	4 ' high	240 ' wide	4 ' peak	1920 sq ft	U = 0.33
2 Wall(s)	4 ' high	100 ' wide	4 ' peak	800 sq ft	U = 0.33
10 Window(s)	4 ' high	4 ' wide		160 sq ft	U = 0.66
10 Door(s)	12 ' high	12 ' wide		1440 sq ft	U = 1.20
4 Door(s)	7 ' high	3 ' wide		84 sq ft	U = 1.20
10 Skylight(s)	3 ' high	8 ' wide		240 sq ft	U = 0.70
1 Floor section	240 ' long	100 ' wide		24000 sq ft	
1 Roof section	240 ' long	100 ' wide	25.0' high	24000 sq ft	U = 0.09
2 Slab edge	240 ' long		total slab edge length =	480 ft	U = 0.55
2 Slab edge	100 ' long		total slab edge length =	200 ft	U = 0.55

Total Net Area Of Each Basic Surface

Walls	Skylights	Windows	Doors
12916 sq ft	240 sq ft	160 sq ft	1524 sq ft

Floor Area	Perimeter	Roof Area
24000 sq ft	680 ft	24235 sq ft

Basic Building Heat Loss

+ Conduction Loss	=	444861 BTU/hr	Power Vent Flow	=	2500 CFM
- Internal Heat Source	=	0 BTU/hr	Total Vent Flow	=	2500 CFM
+ Infil Air Heat Loss	=	269745 BTU/hr	Infiltration Air Flow	=	3500 CFM
Total Building Heat Loss	=	714606 BTU/hr			
Height Inlet to Outlet	=	24 Feet	Building Volume	=	600000 cu ft

Design Temperatures 65ºF Indoor -5ºF Outdoor 70ºF Temperature Difference

Recommended Heater Mounting Height 18 ft

COMPUHEAT ESTIMATED ANNUAL FUEL COST

Date: 05-28-2013 Time: 14:27:01 Prepared by: RJ Herrington

Job Name: Warehouse - Lincoln, NE

Condition 1 Heater requirements with existing building ventilation

Existing Building Vent Flow = 2500 CFM
Total Computed Heat Loss = 907281 BTU/hr
Infrared Heat Required = 734898 BTU/hr
Space - Ray Model LTU100 Input rating 100000 BTU/hr Natural Gas

Total No. of Space-Ray Heaters = 8

Input BTUH/cu. ft. Bldg. Volume = 1.3 Infrared Heater Input = 800000 BTU/hr
Input BTUH/sq. ft. Bldg. Area = 33.3 Total Computed Air Changes = 0.60 AC/hr
Maximum Temperature Rise With Recommended Heaters = 76.2 ºF

Building Operation Parameters

Normal Operating Conditions

Inside Design Temperature = 65 ºF
Outside Design Temperature = -5 ºF
Operating Hours per Day = 12
Operating Days per Week = 5

Setback Operating Conditions

Setback Temperature = 50ºF
Hours per Day = 12
Days per Week = 5
Weekend Hours per Day = 24
Weekend Days per Week = 2

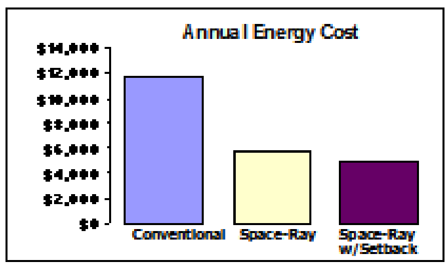
Degree Days @ Normal 65 ºF Inside Design Temperature = 6375
Degree Days @ Setback 50 ºF Inside Design Temperature = 3285

Fuel Specifications

Type of Fuel = Natural Gas
Fuel Cost = $ 0.60 per therm

Estimated Annual Fuel Cost

Conventional Unit Heaters = $ 11,649
Space-Ray InfraRed Heaters = $ 5,850
Space-Ray InfraRed Heaters w / Night Setback = $ 4,934



The foregoing COMPUHEAT heat loss analysis is based on certain data and assumptions provided to the Space-Ray division of Gas-Fired Products, Inc. However, deleted or inaccurate information and other factors not included within the data and assumptions could have a bearing on the results shown herein. The heat loss projection provided is intended only as an illustration and is provided only as a service to Gas-Fired Products' customers, and Gas-Fired Products, Inc. makes no warranties, express or implied, with respect thereto, and disclaims any liability for consequential or other damages.

COMPUHEAT ENERGY SAVINGS ANALYSIS

SPACE-RAY INFRARED GAS HEATERS

Date: 05-29-2013 **Time:** 15:12:05 **Prepared by:** RJ Herrington

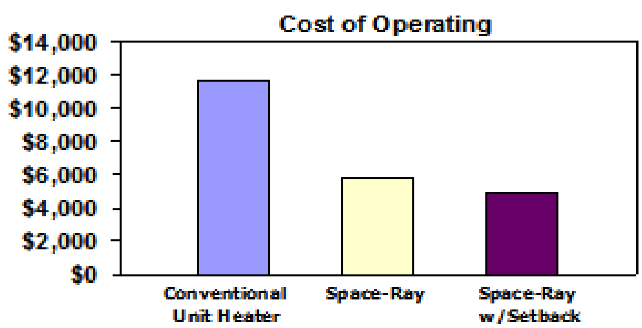
Job Name: Warehouse - Lincoln, NE

ENERGY SAVINGS ANALYSIS

Total Building Heat Loss	= 907281	BTU/hr			
Design Temperatures	65 °F Indoor	-5 °F Outdoor		70 °F Temperature Difference	
Space - Ray System Fuel	Natural Gas		Cost Of Fuel	$ 0.6	per Therm
Alternate System Fuel	Natural Gas		Cost Of Fuel	$ 0.6	per Therm
Annual Fuel Cost Increase	6.00 %				

System Savings

Year	Cost of Fuel	Cost of Operating Conventional Unit Heater	Cost of Operating Space-Ray	Savings Using Space-Ray	Savings Using Space-Ray w/Setback
1	0.60	11,649	5,850	5,799	6,715
2	0.64	12,348	6,201	6,147	7,118
3	0.67	13,089	6,573	6,516	7,546
4	0.71	13,874	6,968	6,906	7,998
5	0.76	14,706	7,386	7,320	8,477
6	0.80	15,588	7,829	7,759	8,986
7	0.85	16,523	8,298	8,225	9,525
8	0.90	17,514	8,796	8,718	10,096
9	0.96	18,565	9,324	9,241	10,702
10	1.01	19,679	9,884	9,795	11,344
		$153,535	$77,109		
	Total savings using Space-Ray heaters			$76,426	$88,507



The above energy savings analysis is based on certain data and assumptions provided to the Space-Ray division of Gas-Fired Products Inc. However, deleted or inaccurate information and other factors not included within the data and assumptions could have a bearing on the results shown herein. Actual fuel costs may vary with the usage of the building. The energy savings projection is intended only as an illustration and is provided only as a service to Gas-Fired Products' customers, and Gas-Fired products, Inc. makes no warranties, express or implied, with respect thereto, and disclaims any liability for consequential or other damages.

SPACE-RAY INFRARED GAS HEATERS

COMPUHEAT ENERGY SAVINGS ANALYSIS

Date: 05-29-2013 Time: 09:52:24 Prepared by: RJ Herrington

Job Name: Warehouse - Lincoln, NE

SIMPLE PAYBACK AND RETURN ANALYSIS

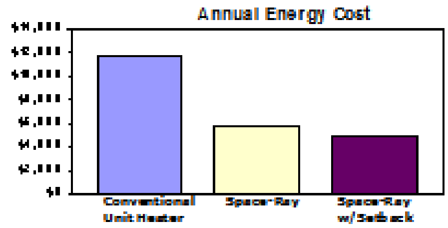
Total Building Heat Loss = 907281 BTU/hr

Design Temperatures 65ºF Indoor -5ºF Outdoor 70ºF Temperature Difference

Space - Ray System Fuel Natural Gas Cost Of Fuel $ 0.60 per Therm

Alternate System Fuel Natural Gas Cost Of Fuel $ 0.60 per Therm

Annual Fuel Cost Increase 6.00 %



Alternate System Conventional Unit Heater

Installed Cost of Space-Ray System $ 19,200

Installed Cost of Alternate System $ 11,000

Difference in Cost of Systems $ 8,200

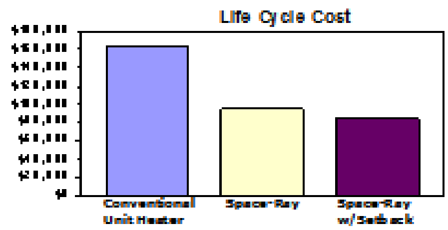
Estimated Annual Cost of Energy

With Alternate System $ 11,649

With Space-Ray System with Night Setback $4,934

Savings using Space-Ray $6,715

Simple Payback 1.22 Years



Percentage Return on Additional Investment 81.9 %

Life Cycle cost (Initial Cost Plus Energy Cost for Ten Years)

With Alternate System $ 164,535

With Space-Ray System with Night Setback $ 84,228

Savings Over a Ten Year Period with Space-Ray $ 80,307

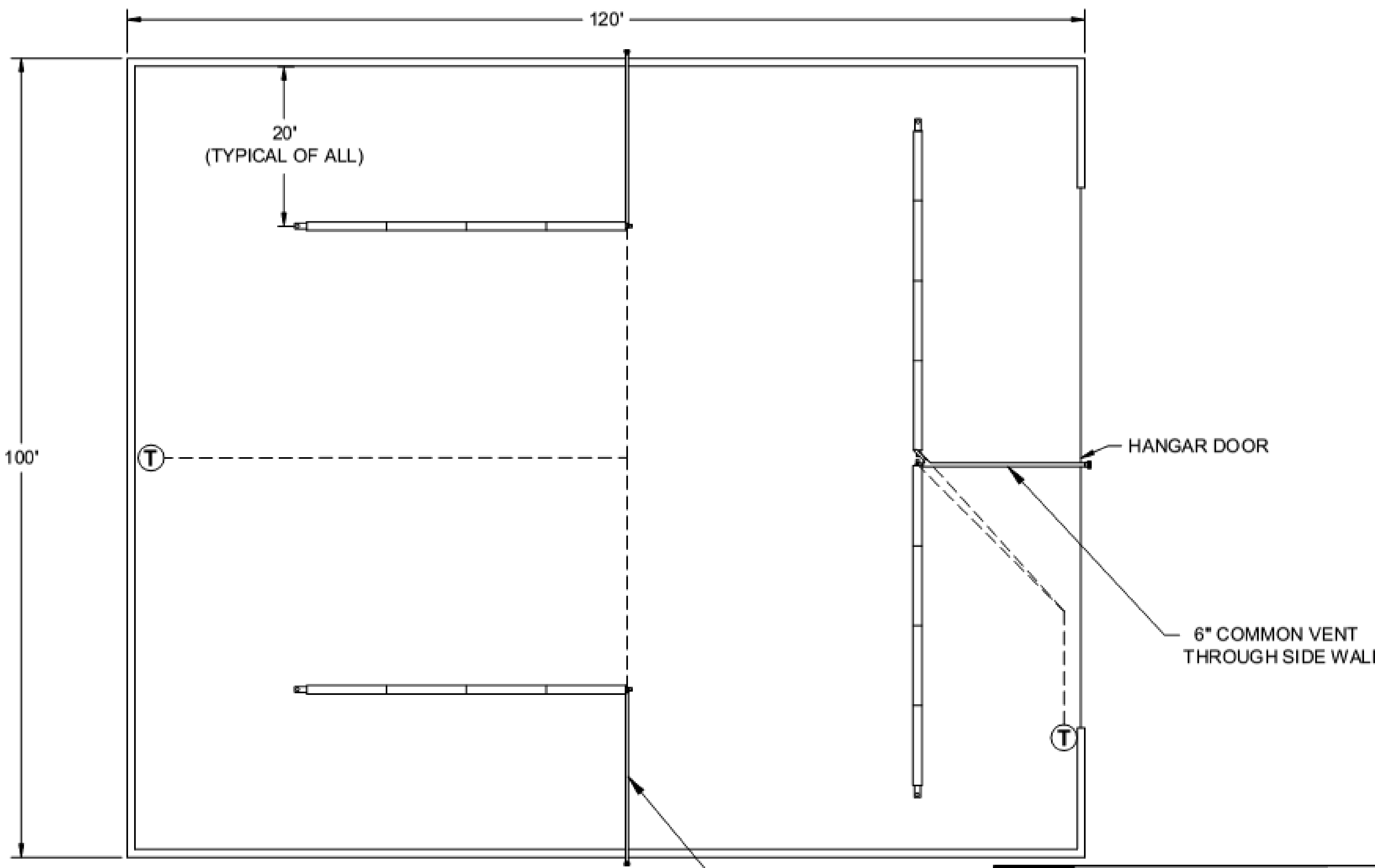
The above simple payback and return analysis is based on certain data and assumptions provided to the Space-Ray division of Gas-Fired Products Inc. However, deleted or inaccurate information and other factors not included within the data and assumptions could have a bearing on the results shown herein. Actual fuel costs may vary with the usage of the building. The energy savings projection is intended only as an illustration and is provided only as a service to Gas-Fired Products' customers, and Gas-Fired products, Inc. makes no warranties, express or implied, with respect thereto, and disclaims any liability for consequential or other damages.

M. SAMPLE LAYOUTS

The enclosed sample layouts illustrate the use and selection of infrared heaters in different applications. The unit and configuration selections are up to the heating system design engineer since every heating job is different. Please note: straight tube heaters are always cooler at the exhaust end than at the burner end and require special attention when used in lower mounting heights. Always observe the minimum recommended mounting heights as shown on the heater specification sheets. Study the building heat loss and infrared heater requirements carefully prior to the placement of the straight tube heaters. U-tube heaters provide more uniform energy distribution and are highly recommended at lower mounting heights (15' or less depending on the heater input) and for small area heating applications.

The sample layouts are as follows:

- Aircraft Hangar - Long Bay — No. 971
- Aircraft Hangar - Short Bay — No. 970
- Auto Dealership - Service Area with 14 Overhead Doors — No. 964
- Auto Dealership - Service Areas, Parts, Write-up Areas — No. 966
- Carwash - Gas Service Station Type — No. 985
- Carwash - Automated Long Bay Type — No. 984
- Fire Station - 4 Bay Type — No. 972
- Fire Station - 6 Bay Type — No. 969
- Greenhouse Heating — No. 1040
- Gymnasium Heating — No. 1036
- Ice Hockey Arena - Bleacher Area Heating — No. 961
- Indoor Tennis Court - with Partition Nets Between Courts and Individual Zone Control — No. 962
- Indoor Tennis Court - Complete Building Zone Heating — No. 963
- Swimming Pool - Residential — No. 1034
- Swimming Pool - University — No. 1038
- Golf Range Infrared Heating — No. 1105



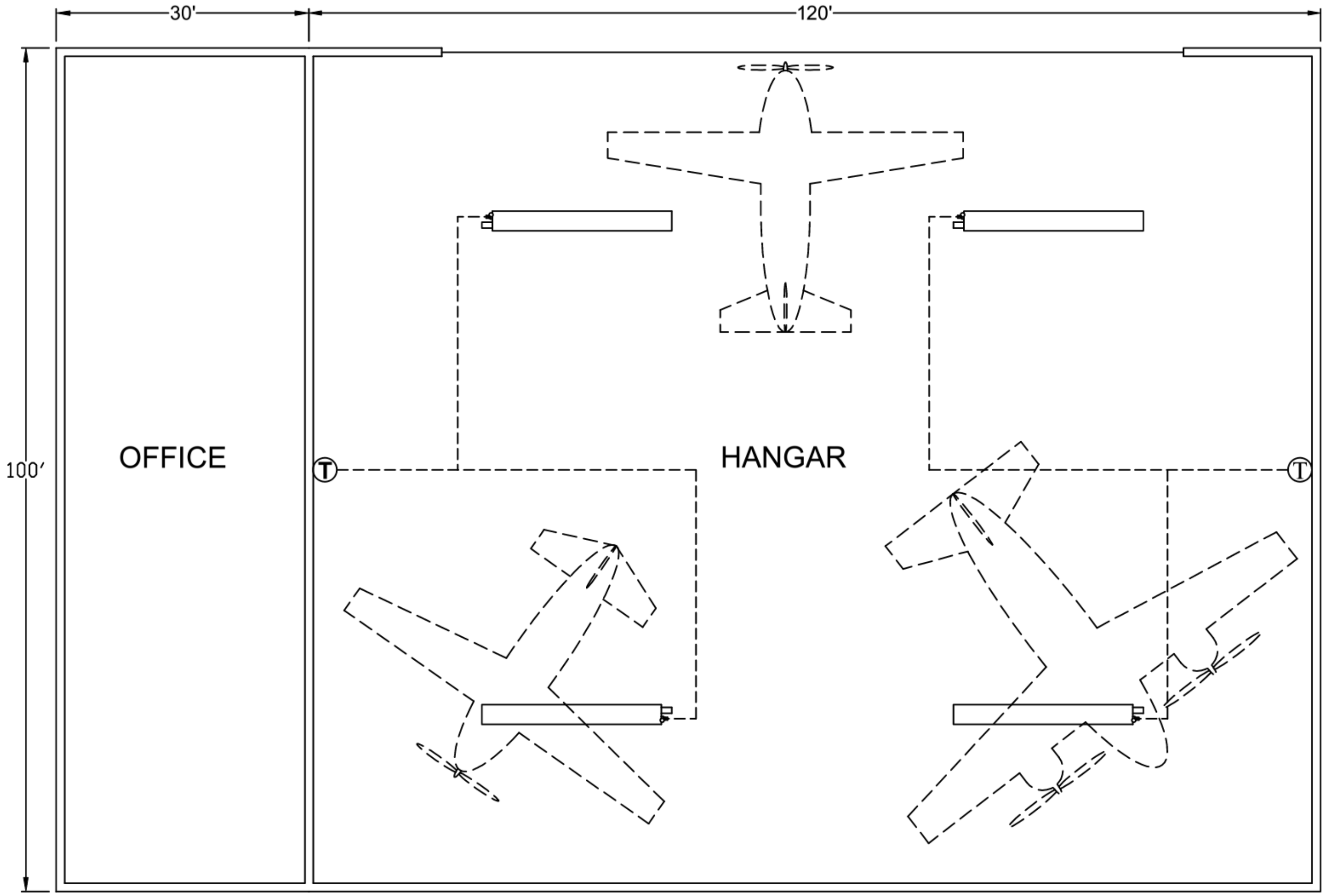
INFRARED HEATING REQUIREMENTS

1. (4) ETS150-40 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 25 FT. ABOVE FINISHED FLOOR.
2. (2) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND "I & O" INSTRUCTIONS FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: AIRCRAFT HANGAR INSTALLATION DETAIL-971

Drawn By:	Date:
RYAN RICHARDSON	21 JUN 16

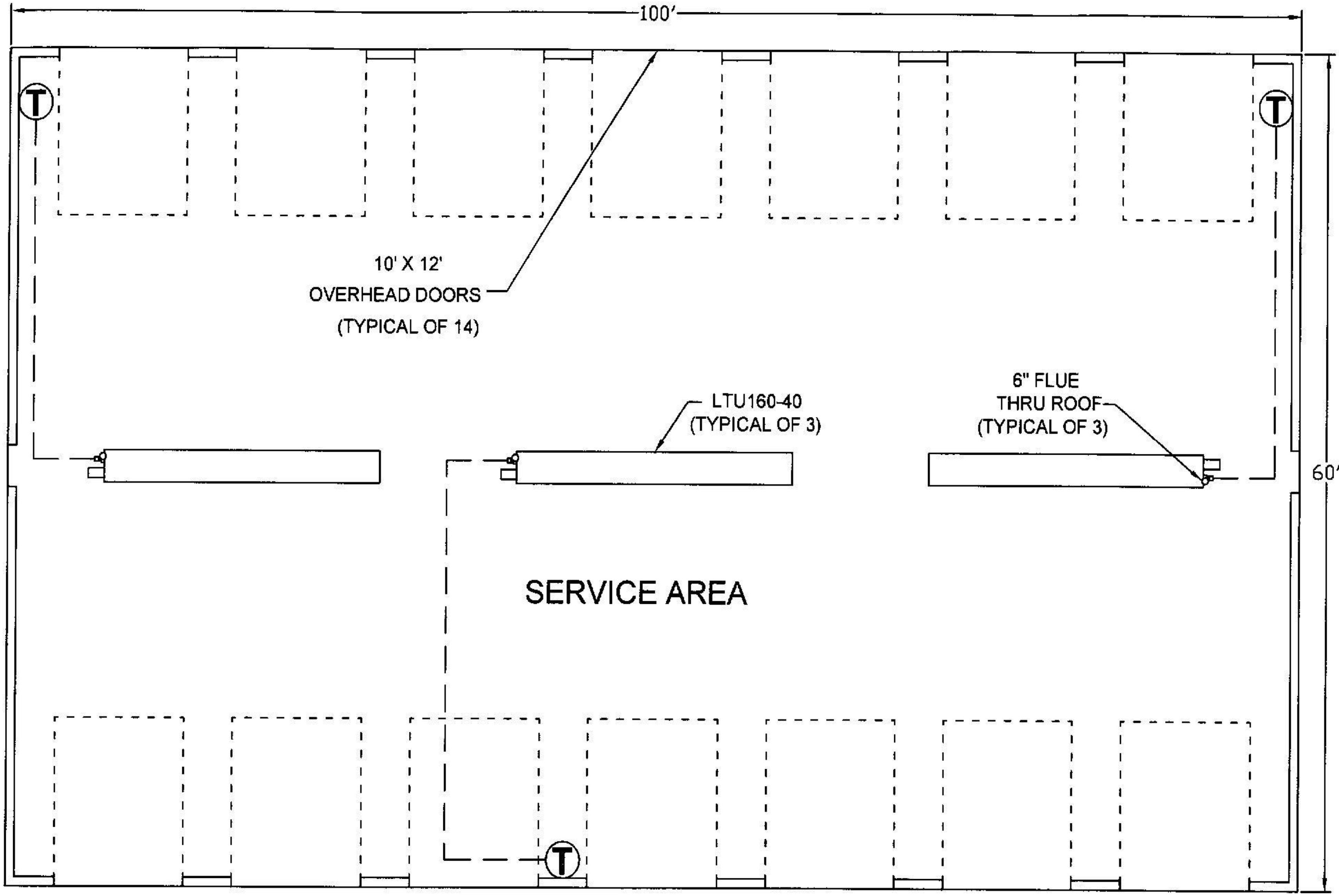


INFRARED HEATING REQUIREMENTS

1. (4) LTU130-40 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 22 FT. ABOVE FINISHED FLOOR.
2. (2) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND "I & O" INSTRUCTIONS FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY® INFRARED GAS HEATERS

Title: AIRCRAFT HANGAR INSTALLATION DETAIL-970	
Drawn By: RJ Herrington	Date: 07 MAR 06

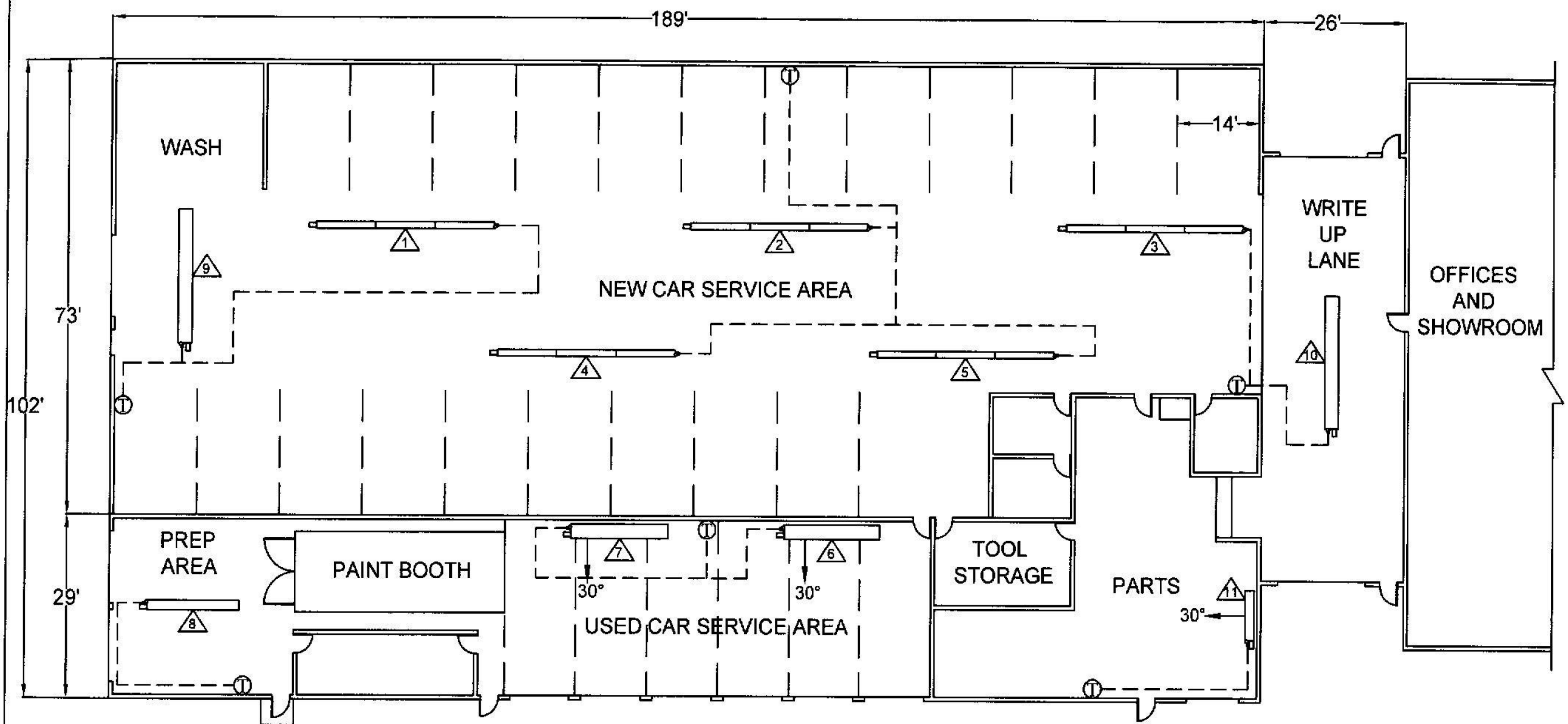


INFRARED HEATING REQUIREMENTS

1. (3) LTU160-40 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 18 FT. ABOVE FINISHED FLOOR.
2. (3) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND "I & O" INSTRUCTIONS FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY® INFRARED GAS HEATERS

Title: AUTO DEALERSHIP INSTALLATION DETAIL-964	
Drawn By: RJ Herrington	Date: 16 JUN 05



INFRARED HEATING REQUIREMENTS

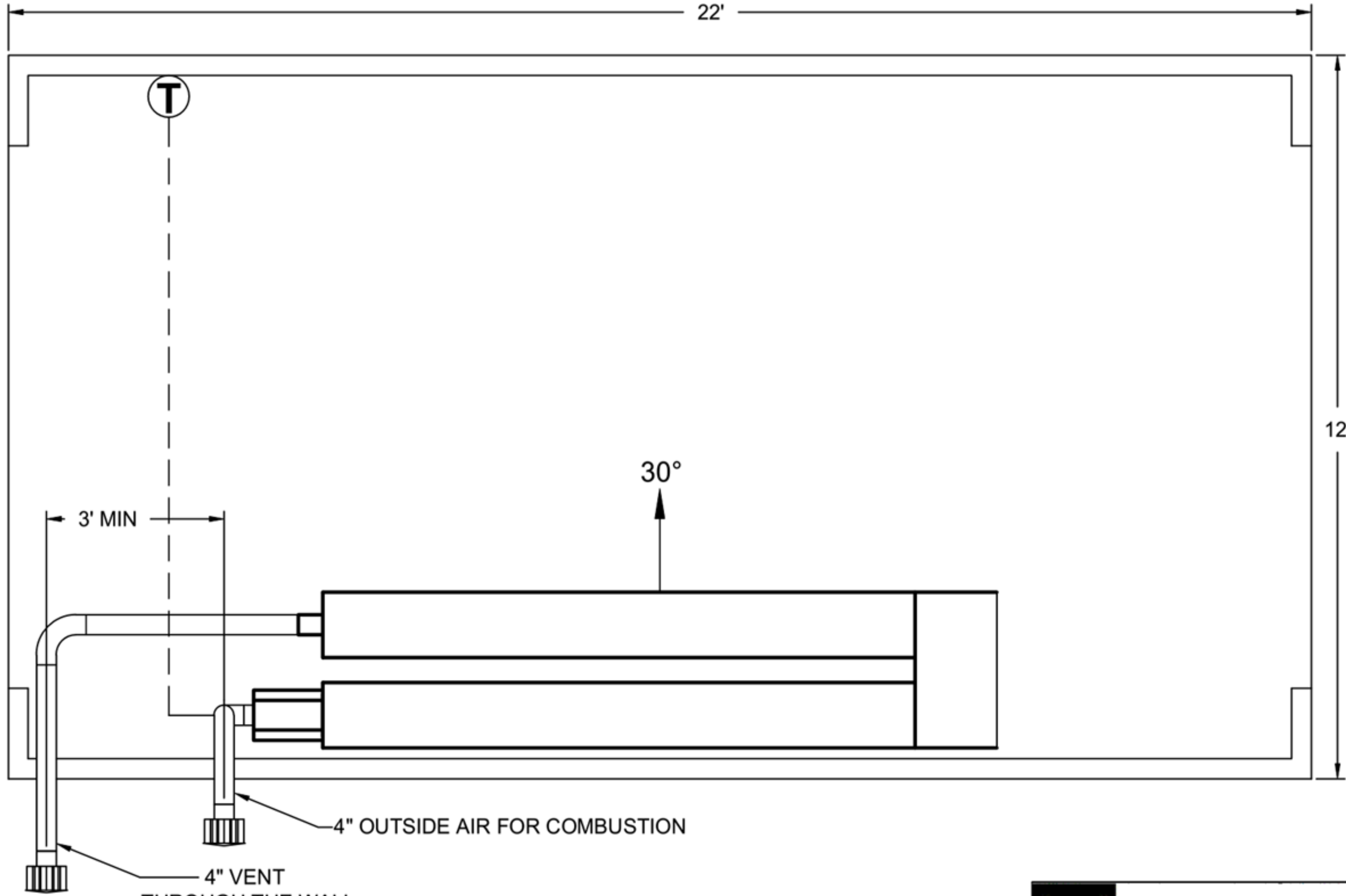
1. (5) LTS130-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 18 FT. ABOVE FINISHED FLOOR. (HEATERS 1 TO 5)
2. (2) LTU80-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 18 FT. ABOVE FINISHED FLOOR. (HEATERS 6 & 7)
3. (1) LTU75-30 VENTED SPACE-RAY INFRARED HEATER. INSTALL AT 18 FT. ABOVE FINISHED FLOOR. (HEATER 8)
4. (2) LTU130-40 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 18 FT. ABOVE FINISHED FLOOR. (HEATERS 9 & 10)
5. (1) CB50 VENTED SPACE-RAY INFRARED HEATER. INSTALL AT 18 FT. ABOVE FINISHED FLOOR. (HEATER 11)
6. (6) SPACE-RAY RT SERIES THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
7. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
8. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
9. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: AUTO DEALERSHIP INSTALLATION DETAIL-966

Drawn By: RJ Herrington

Date: 07 MAR 06

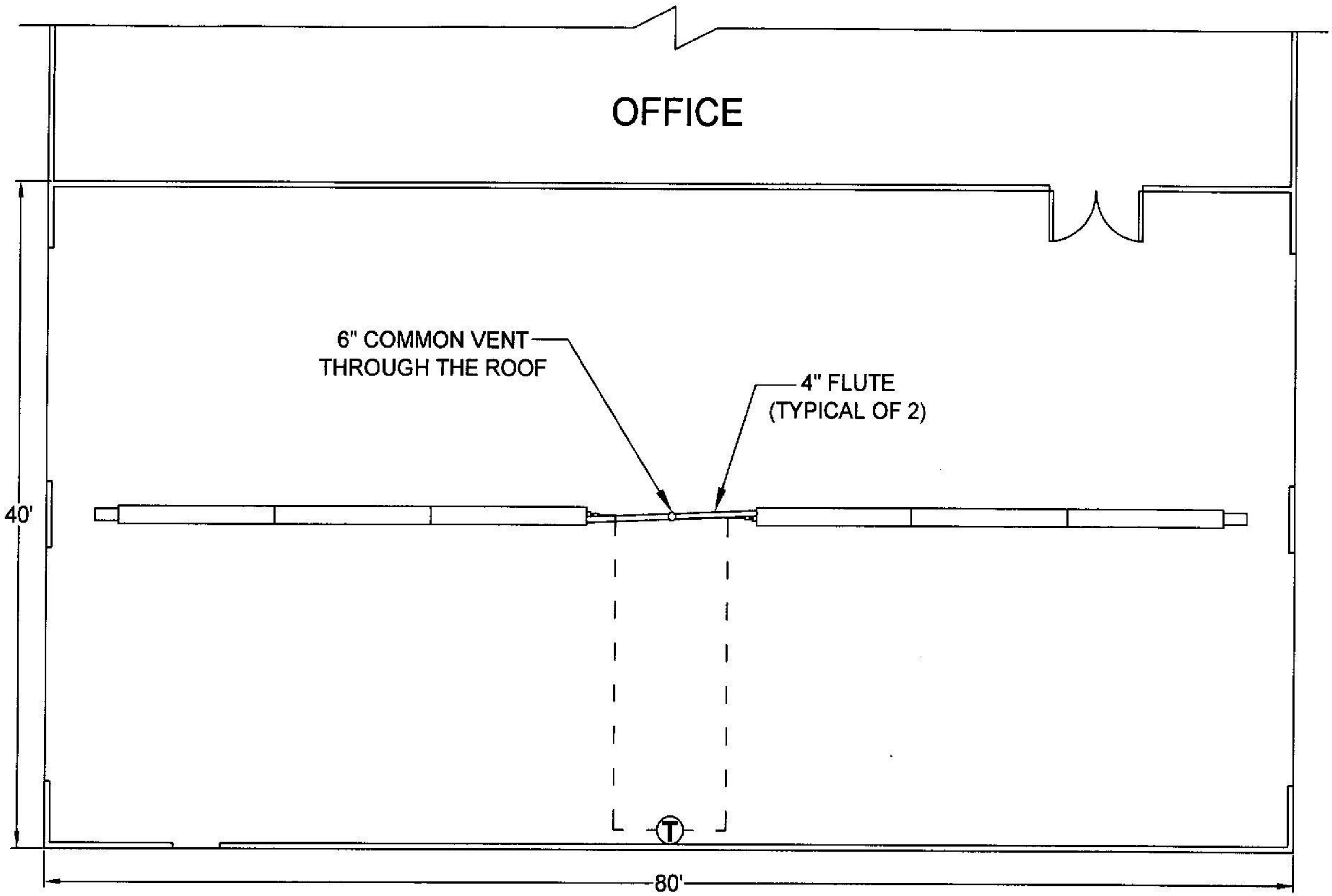


INFRARED HEATING REQUIREMENTS

1. (1) PTU75-30-ALC VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 12 FT. ABOVE FINISHED FLOOR.
2. (1) SPACE-RAY HIGH HUMIDITY ENVIRONMENT THERMOSTAT MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PUSH THROUGH" SYSTEMS WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY® INFRARED GAS HEATERS

Title: AMOCO CAR WASH INSTALLATION DETAIL-985	
Drawn By: RJ Herrington	Date: 08 MAR 06

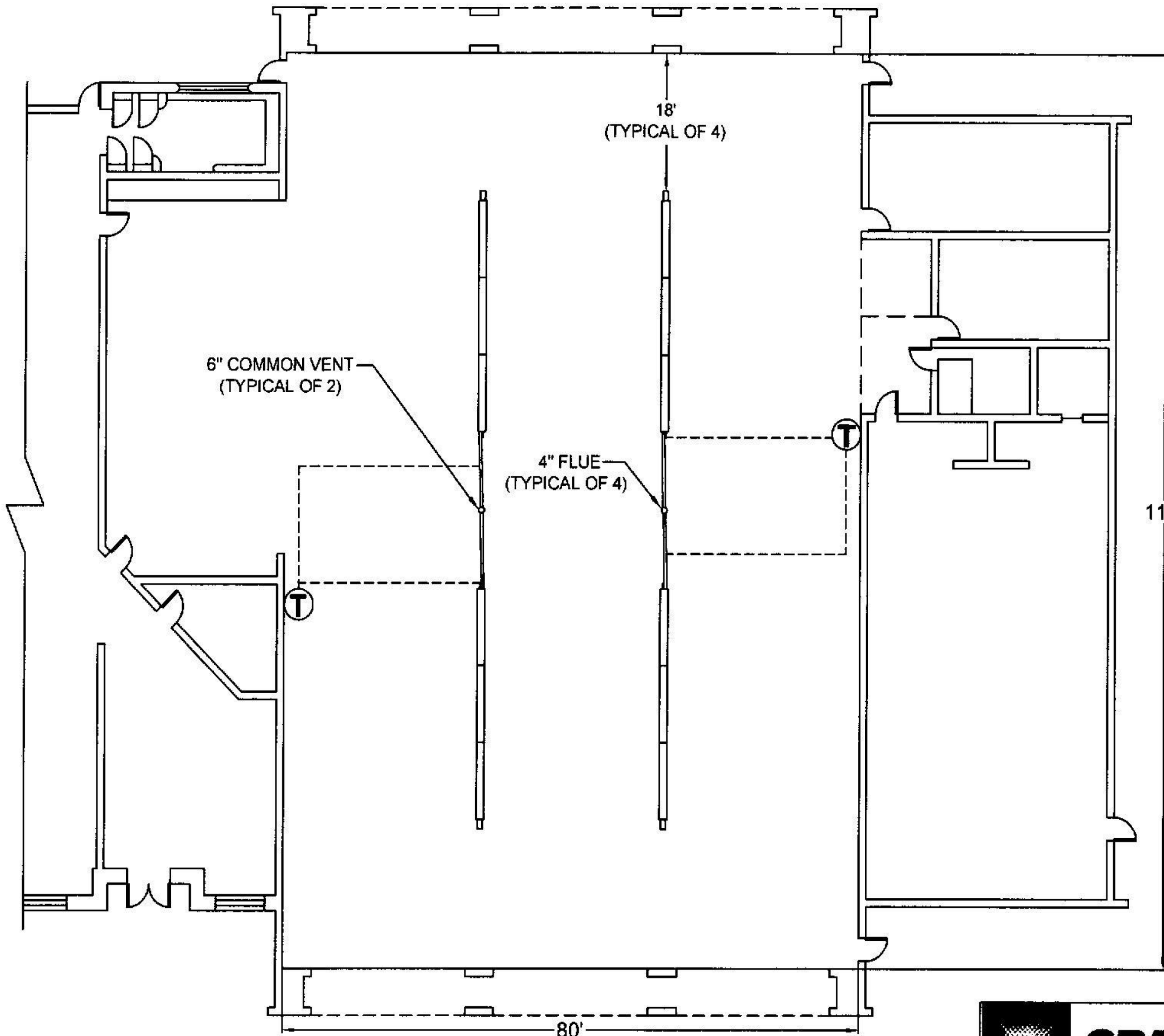


INFRARED HEATING REQUIREMENTS

1. (2) ET80-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 14 FEET ABOVE THE FINISHED FLOOR.
2. (1) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE THE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: 4 BAY FIRE STATION INSTALLATION DETAIL-972	
Drawn By: RJ Herrington	Date: 24 MAY 06

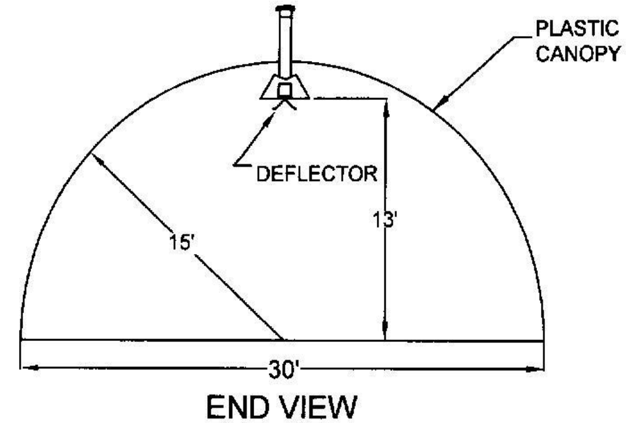


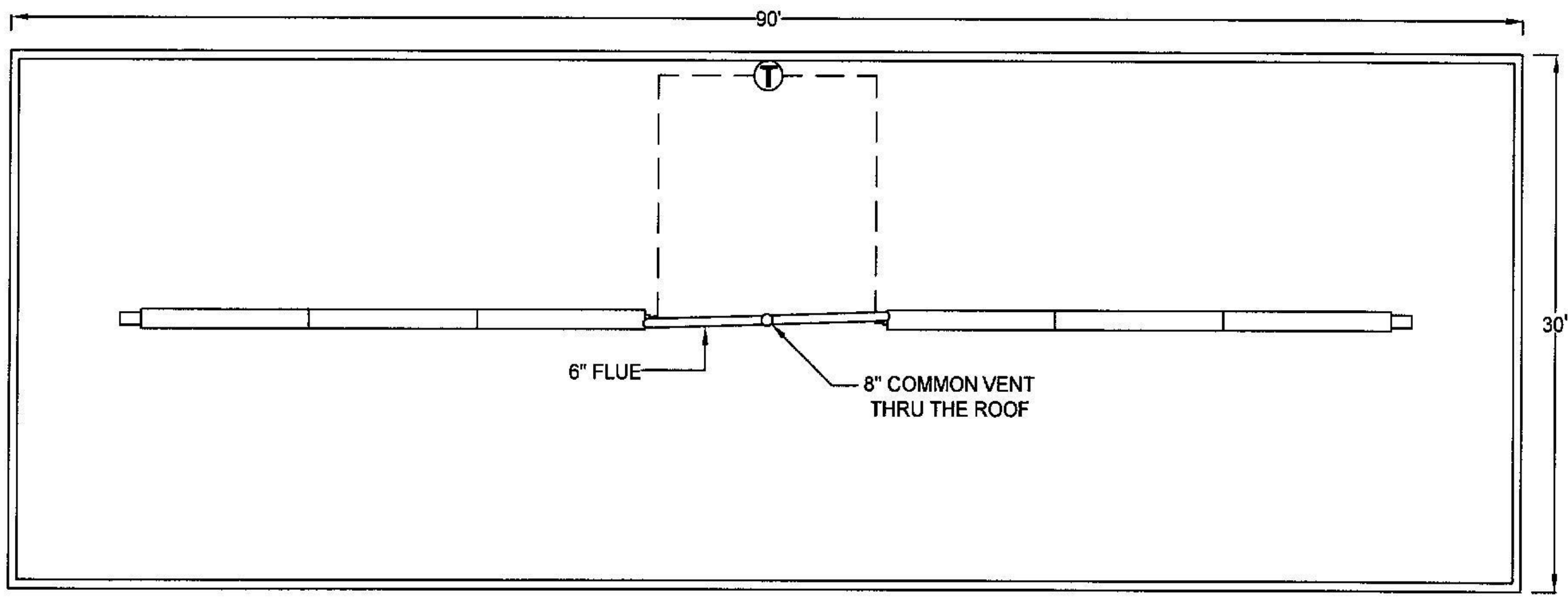
INFRARED HEATING REQUIREMENTS

1. (4) LTS60-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 14 FT. ABOVE FINISHED FLOOR.
2. (2) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY®
INFRARED GAS HEATERS

Title: 6 - BAY FIRE STATION INSTALLATION DETAIL-969	
Drawn By: RJ Herrington	Date: 16 JUN 05





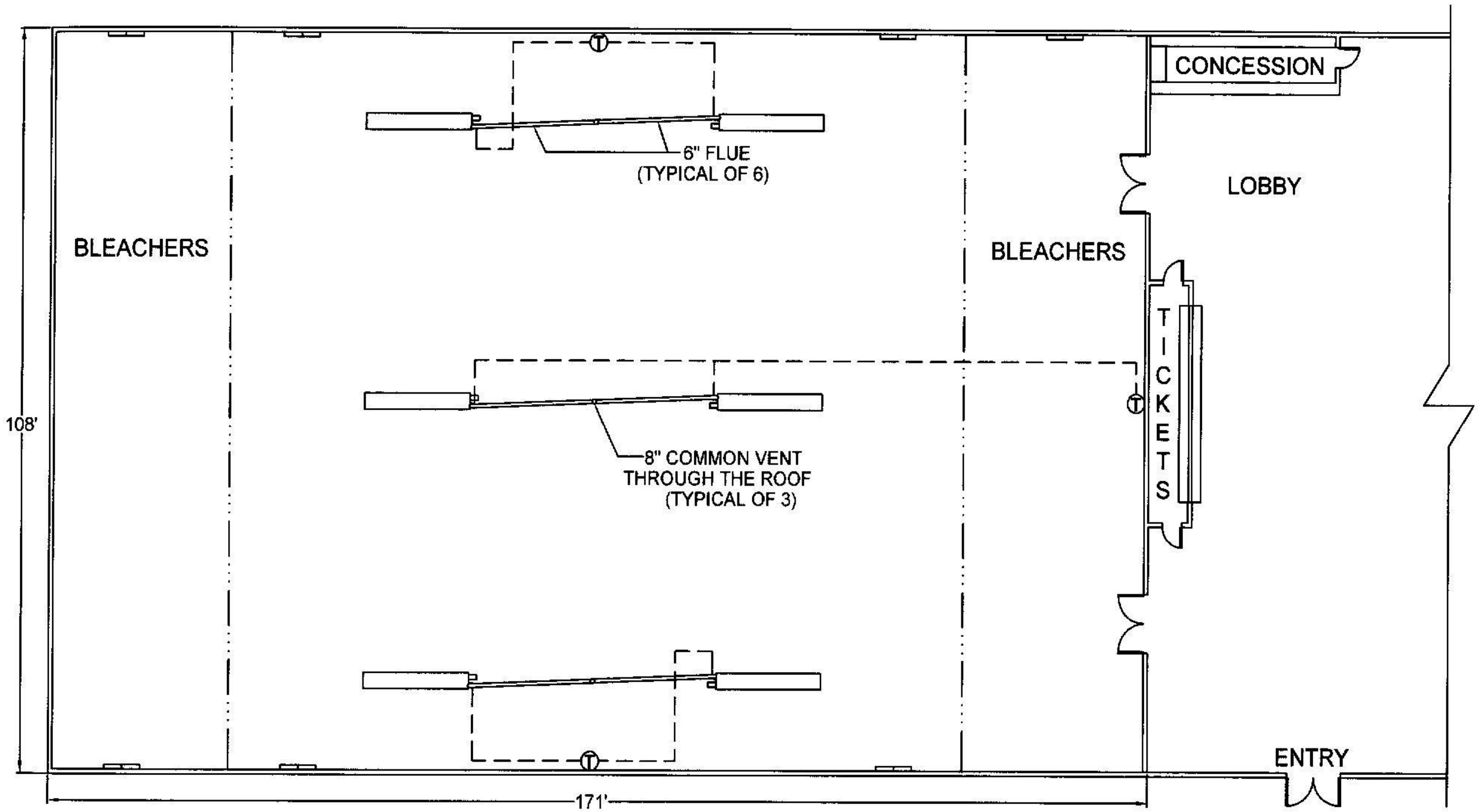
INFRARED HEATING REQUIREMENTS

1. (2) LTS100-30 VENTED SPACE-RAY INFRARED HEATERS WITH 10' DEFLECTOR ASSEMBLY. INSTALL AT 13 FT. ABOVE FINISHED FLOOR.
2. (1) SPACE-RAY RT SERIES THERMOSTAT MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY INFRARED GAS HEATERS

Title: GREENHOUSE HEATING INSTALLATION DETAIL-1040

Drawn By: RJ Herrington | Date: 16 JUN 05

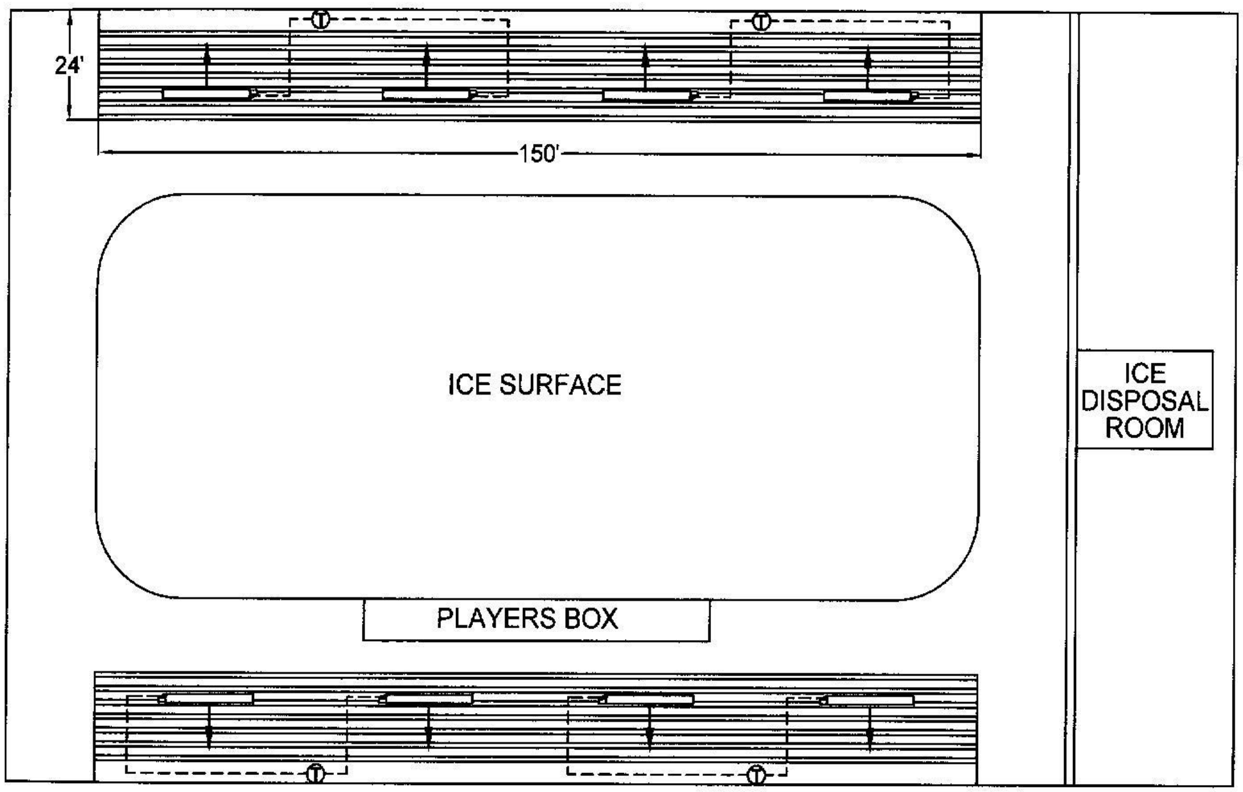


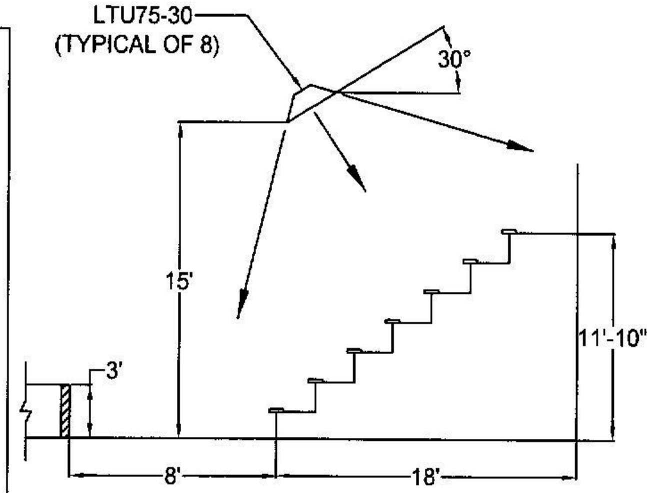
INFRARED HEATING REQUIREMENTS

1. (6) LTU130-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 20 FT. ABOVE FINISHED FLOOR.
2. (3) SPACE-RAY SETBACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: WESTSIDE GYMNASIUM INSTALLATION DETAIL-1036	
Drawn By: RJ Herrington	Date: 16 JUN 05



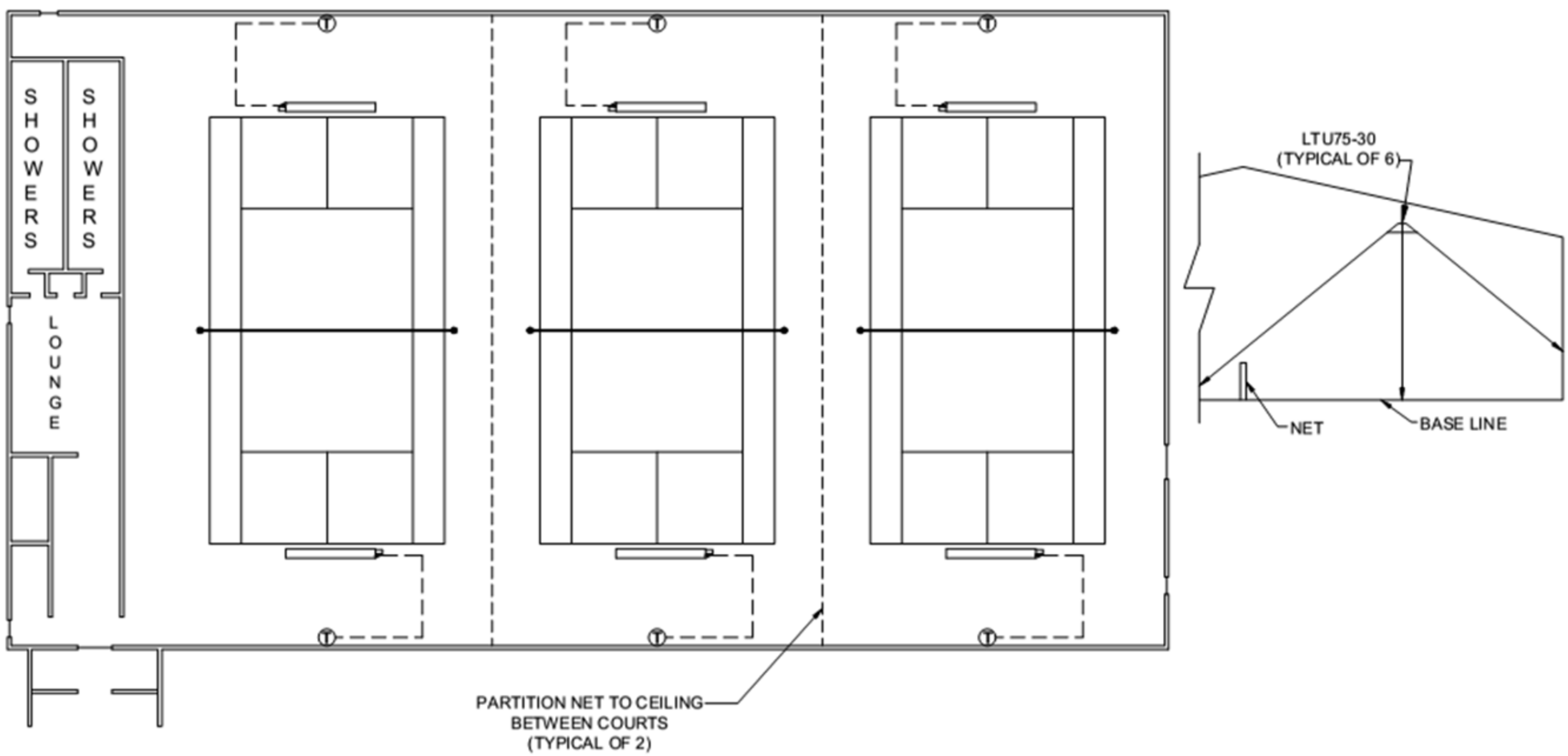


INFRARED HEATING REQUIREMENTS

1. (8) LTU75-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 15 FEET AND AT A 30° ANGLE ABOVE FINISHED FLOOR.
2. (4) SPACE-RAY RT SERIES THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY INFRARED GAS HEATERS

Title: ICE HOCKEY ARENA INSTALLATION DETAIL-961	
Drawn By: RJ Herrington	Date: 16 JUN 05



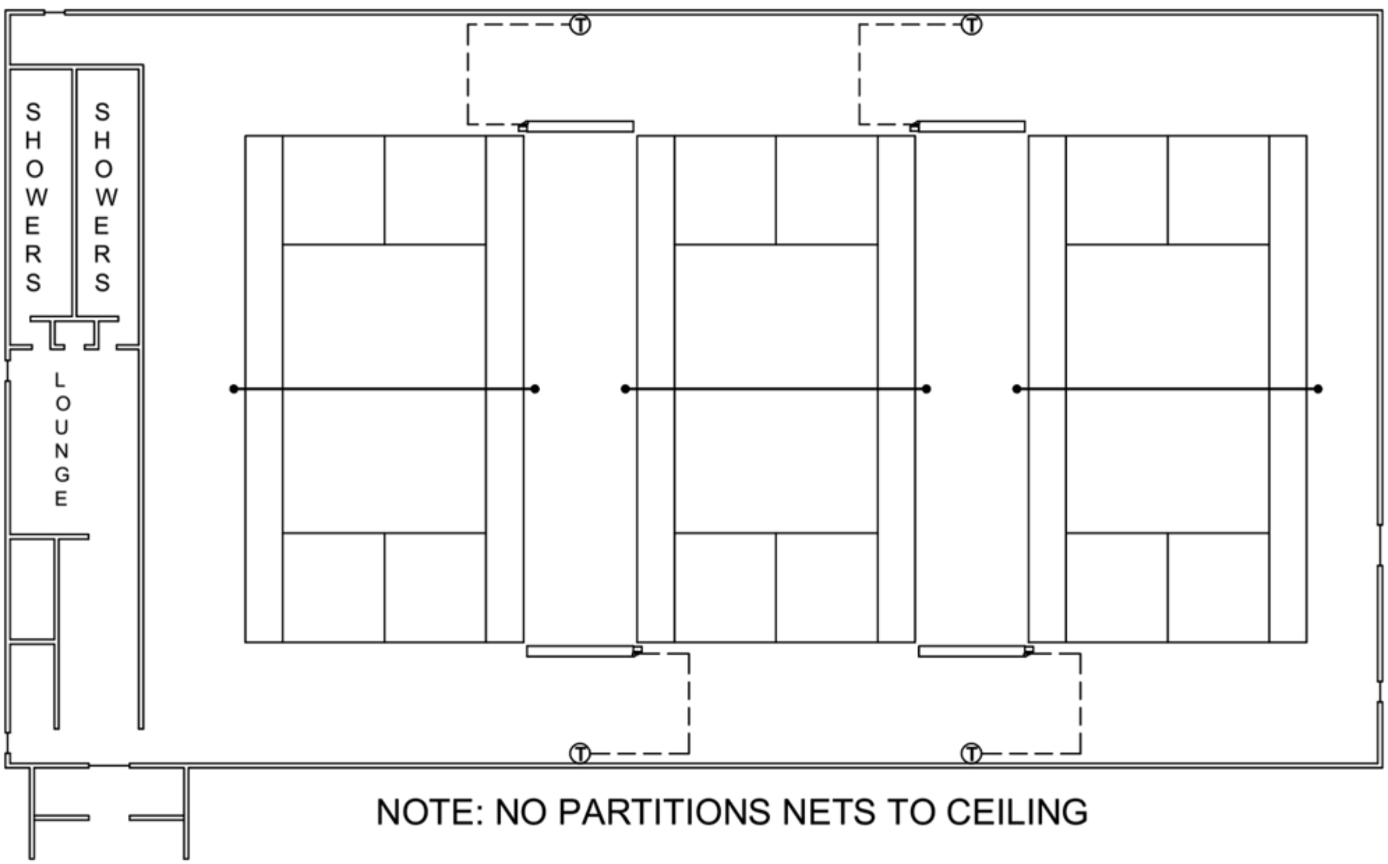
INFRARED HEATING REQUIREMENTS

1. (6) LTU75-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 30 FEET ABOVE FINISHED FLOOR.
2. (6) SPACE-RAY RT-SERIES THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

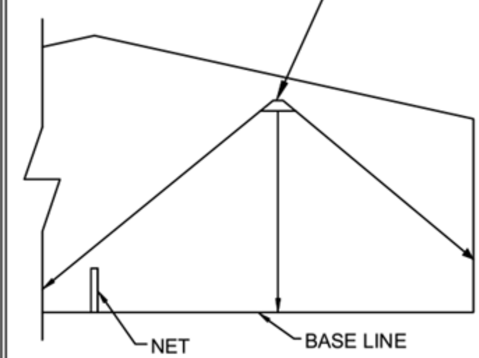
SPACE-RAY
INFRARED GAS HEATERS

Title: INDOOR TENNIS COURT INSTALLATION DETAIL-962

Drawn By:	Date:
RYAN RICHARDSON	3 JUN 16



NOTE: NO PARTITIONS NETS TO CEILING

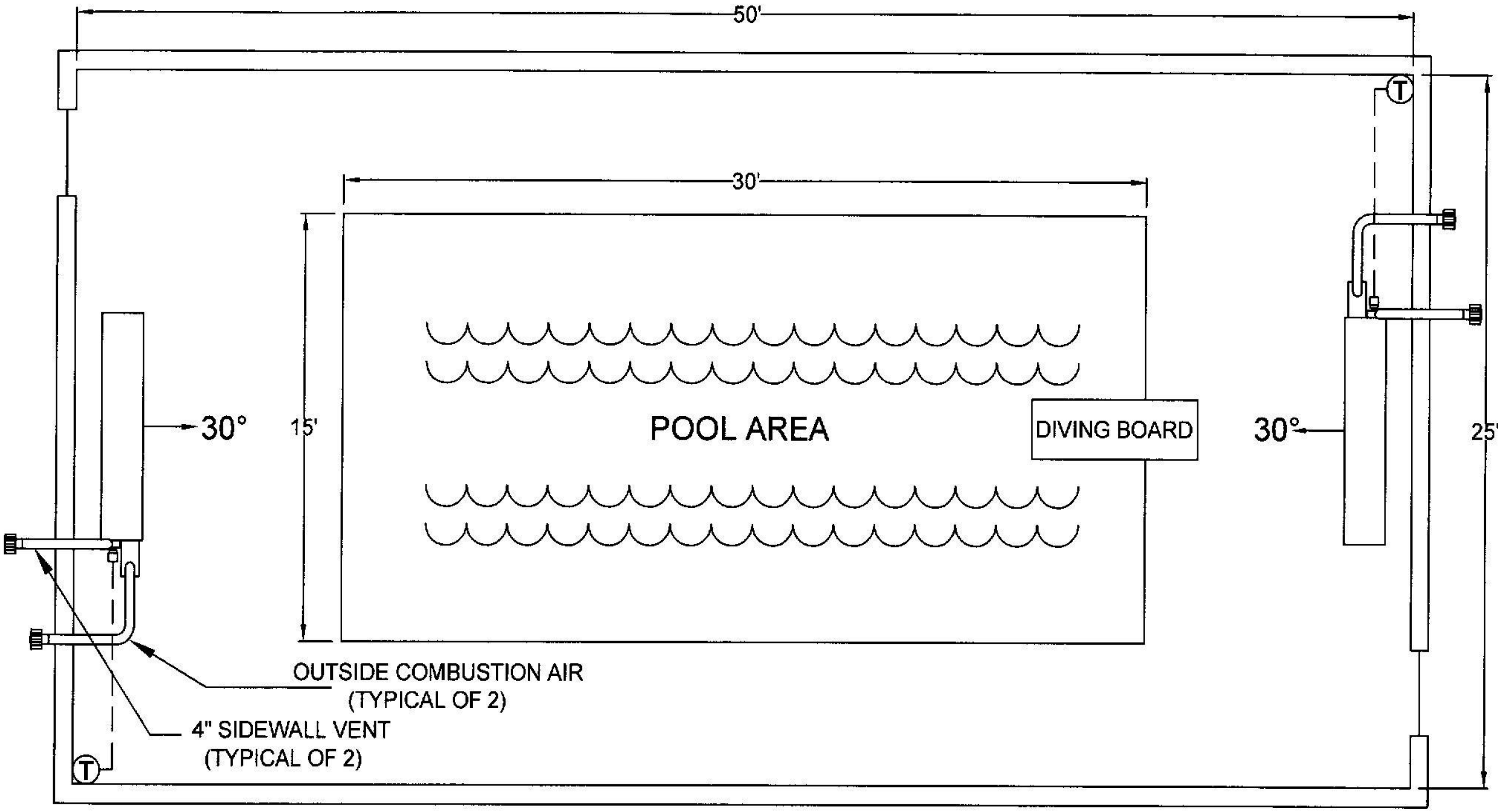


INFRARED HEATING REQUIREMENTS

1. (4) LTU130-30 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 30 FEET ABOVE FINISHED FLOOR.
2. (4) SPACE-RAY RT-SERIES THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY®
INFRARED GAS HEATERS

Title: INDOOR TENNIS COURT INSTALLATION DETAIL-963	
Drawn By: RYAN RICHARDSON	Date: 3 JUN 16

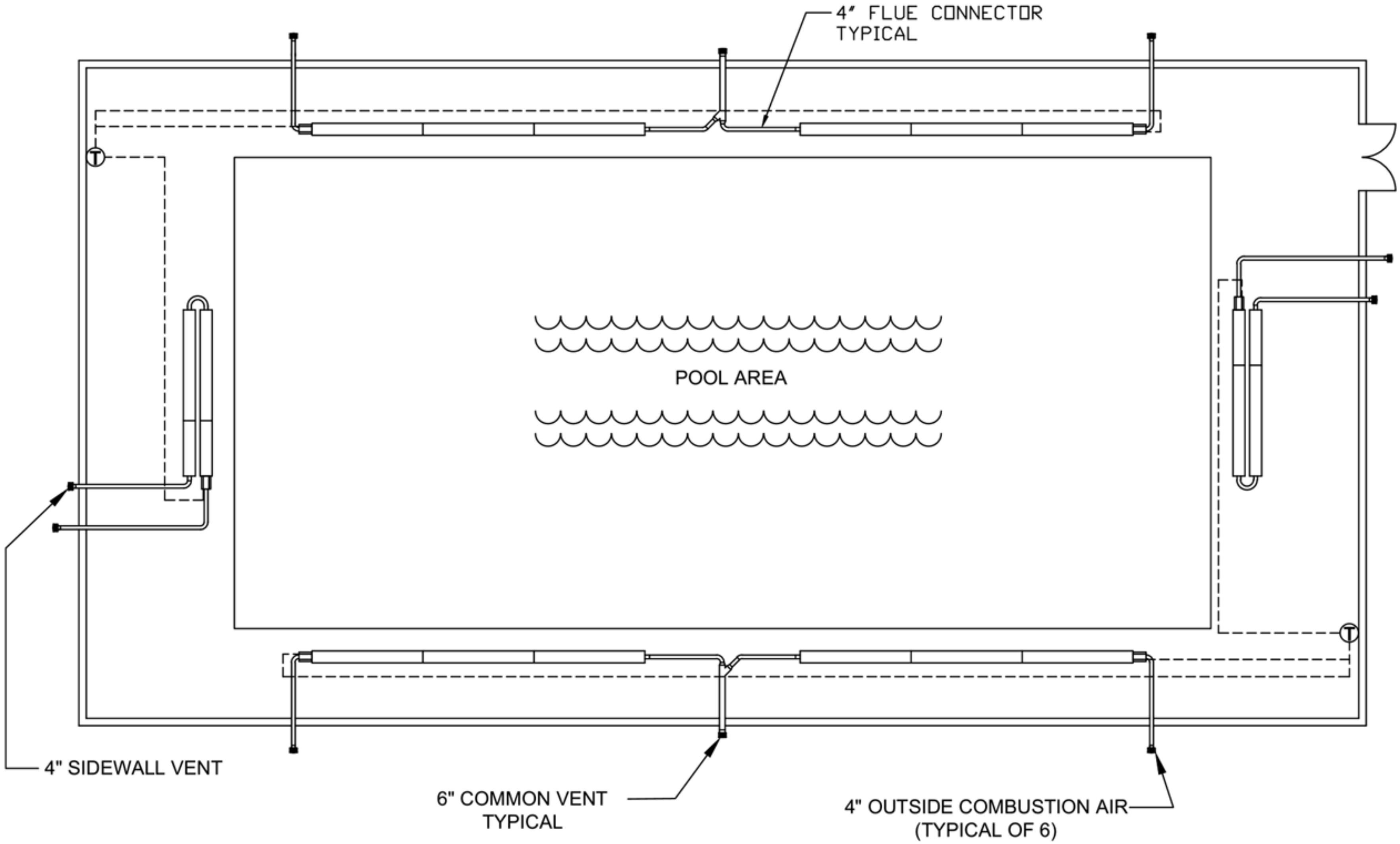


INFRARED HEATING REQUIREMENTS

1. (2) CB50-15 VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 12 FEET ABOVE FINISHED FLOOR.
2. (2) SPACE-RAY HIGH HUMIDITY ENVIRONMENT THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
3. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
4. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
5. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: INDOOR SWIMMING POOL INSTALLATION DETAIL-1034	
Drawn By: RJ Herrington	Date: 16 JUN 05



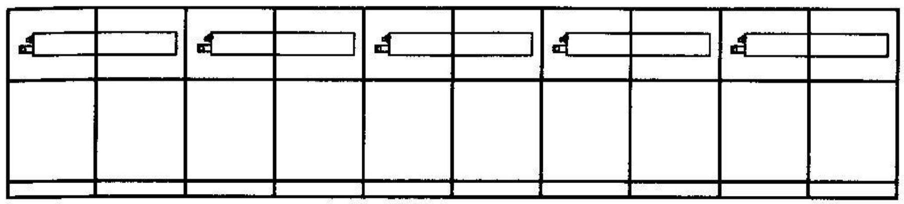
INFRARED HEATING REQUIREMENTS

1. (4) PTS100-30-ALC VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 15 FT.
2. (2) PTU100-30-ALC VENTED SPACE-RAY INFRARED HEATERS. INSTALL AT 15 FT.
3. (2) SPACE-RAY SET-BACK 5000 THERMOSTATS MOUNTED 5'6" ABOVE FINISHED FLOOR.
4. HEATERS SHALL BE "PUSH THROUGH" SYSTEMS (Positive Pressure) WITH CAST IRON BURNER AND CALORIZED EMITTER TUBES.
5. HEATER, THERMOSTAT LOCATIONS SUBJECT TO FIELD VERIFICATION & COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
6. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

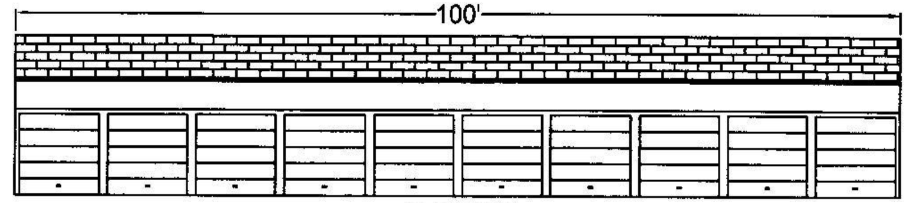
SPACE-RAY® INFRARED GAS HEATERS

Title: GARDNER-WEBB UNIVERSITY INDOOR SWIMMING POOL INSTALLATION DETAIL-1038

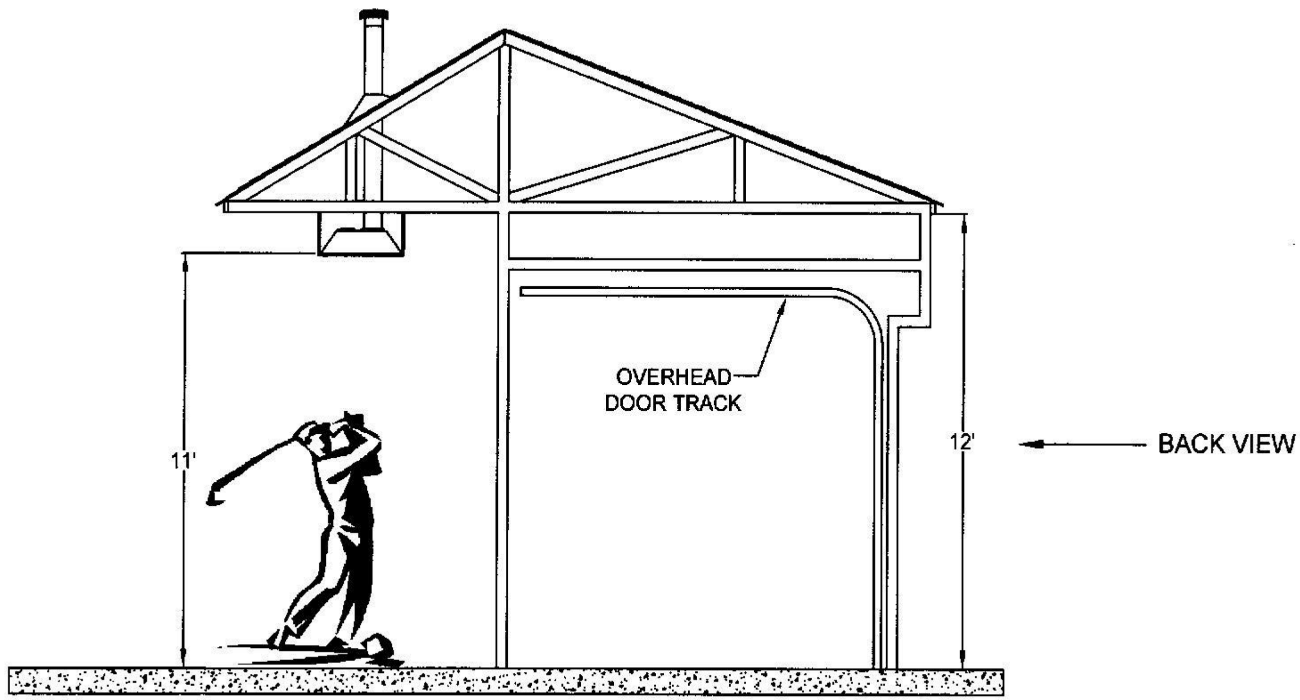
Drawn By: RYAN RICHARDSON | Date: 3 JUN 16



TOP VIEW



BACK VIEW



SIDE VIEW

INFRARED HEATING REQUIREMENTS

1. (5) LTU 130-30 VENTED SPARCE-RAY INFRARED HEATERS. INSTALL AT 11 FEET ABOVE FINISHED FLOOR.
2. HEATERS SHALL BE "PULL THROUGH" SYSTEMS (VACUUM) WITH CAST IRON BURNER.
3. EMITTER TUBE SHALL BE CALORIZED FOR HIGH EMISSIVITY AND CORROSION RESISTANCE.
4. HEATERS SHALL UTILIZE SINGLE REFLECTOR DESIGN THAT WILL COVER THE FIRING AND EXHAUST LEG AS WELL AS ENTIRE U-BEND.
5. HEATER AND SPECIFIED CONTROL SYSTEM LOCATION SUBJECT TO FIELD VERIFICATION AND TO BE COORDINATED WITH OTHER EQUIPMENT IN THE BUILDING.
6. CONSULT HEATER SPECIFICATION SHEET AND I & O INSTRUCTIONS. FOR FURTHER INFORMATION PLEASE VISIT WWW.SPACERAY.COM

SPACE-RAY
INFRARED GAS HEATERS

Title: GOLF RANGE
INSTALLATION DETAIL-1105

Drawn By: RJ Herrington
Date: 05 APR 06

V. VENTILATION FOR INDIRECT VENTED INFRARED HEATERS

The use of an infrared heating system indirect vented (unvented) requires additional ventilation of the building. The ventilation rate will be based on total heating capacity. The dilution air requirements are as follows:

Type Gas	CFM*	CFH
Natural	4.00	240
Propane	4.18	251

**Per 1,000 Btu/hr*

The additional ventilation can be provided by gravity or mechanical ventilation.

A. GRAVITY VENTILATION FOR INDIRECT VENTED HEATERS

Gravity ventilation, frequently called stack effect, can provide adequate ventilation if the designer exercises the proper precautions in its application.

Gravity ventilation is best applied in high-bay type buildings (steel industries, glass manufacturers) having high-bay areas with 30 to 60 foot ceilings. It is not always suitable for the modern, wide, low profile type of building even though the roof may peak at 20 to 28 feet.

$$Q = 9.4 \times A \times (H \times Td)$$

Where:

Q = Air Flow – CFM
A = Free Area – sq. ft. of inlet or outlet (use the lesser of the two)
Td = Temperature difference ^{o}F – inside to outside
H = Vertical Height – inlet to outlet

The air velocity through the ventilator openings will seldom exceed 450 fpm and is frequently 200 fpm or less in low profile buildings requiring excessively large outlet vent openings and an equal amount of inlet air opening. Since the vertical distance between inlet and outlet is rather small and air velocity leaving the roof ventilator is usually low on low profile buildings, mechanical ventilation is recommended.

Gravity ventilation may be aided by natural ventilation induced by wind forces. However, the designer should take the viewpoint that wind is not dependable as the motivating force where continuous ventilation is required. Such wind-induced ventilation is not adequate for buildings heated with infrared requiring positive vent flow.

B. MECHANICAL VENTILATION FOR INDIRECT VENTED HEATERS

When mechanical ventilation is used, the exhaust fan must be interlocked to the infrared-heating thermostat. Also, inlet air openings must be provided, and the greater the inlet area, the slower the air movement through the opening. To eliminate cold spots and drafty conditions, the inlets should be small and well distributed around the periphery and away from personnel work areas. In many cases, it is preferable to locate the inlets high (8' to 20' above the floor) with a deflector or scoop to direct the air flow up, but they must always be located below the heater level. An inlet velocity of 200 to 400 fmp is generally recommended. The following Exhaust Fan Capacity Chart is the result of averaging 12 to 14 AMCA certified ratings at 1/4" S.P. for each size and horsepower rating. This table in no way reflects the theoretical fan capacity ratings for each size and horsepower, and should only be used as a general guide to determine the approximate fan capacity when no other information is available.

The smaller sizes (12" to 16") and larger sizes (42" to 72") vary plus or minus 5% from the average for their particular size. The mid-range sizes, 1/2 HP to 3 HP and 18" to 36" diameter will vary from plus 15% to minus 35% depending on the quality and price range of the particular ventilator.

EXHAUST FAN CAPACITY (CFM) -- FAN HORSEPOWER --											
FAN SIZE	**1/4**	**1/3**	**1/2**	**3/4**	**1**	**1-1/2**	**2**	**3**	**5**	**7-1/2**	**10**
12	1,540	1,580	1,830	****	****	****	****	****	****	****	****
14	1,600	1,770	2,110	2,940	****	****	****	****	****	****	****
16	2,090	2,110	2,500	2,900	****	****	****	****	****	****	****
18	2,410	2,750	3,110	3,580	3,990	****	****	****	****	****	****
20	2,060	3,120	3,870	4,910	5,440	5,690	6,190	****	****	****	****
24	2,500	3,790	4,870	5,370	6,500	7,580	8,860	10,320	****	****	****
30	4,740	5,050	6,230	6,720	8,390	10,480	11,890	13,880	17,630	****	****
36	****	5,430	6,600	9,600	11,250	12,430	15,090	17,120	21,170	23,560	28,180
42	****	****	9,150	10,930	12,010	14,140	17,320	21,100	25,190	29,320	32,380
48	****	****	****	11,510	13,380	16,860	20,660	24,410	30,160	35,620	39,030
54	****	****	****	15,510	14,240	19,570	22,060	26,640	35,760	42,340	47,820
60	****	****	****	****	17,910	21,060	24,260	30,250	38,350	46,280	52,360
72	****	****	****	****	****	****	****	29,060	48,890	62,760	71,540

C. INCREASED BUILDING HEAT LOSS DUE TO UNVENTED USE OF INFRARED HEATERS

Most end users and contractors think that the ceramic type of infrared heaters do not require additional ventilation. In order to satisfy the National Fuel Gas Code NFPA54 and local codes, the use of unvented ceramic infrared heaters requires a minimum ventilation flow of 4 CFM per 1,000 Btu/hr of heater input by either mechanical or gravity ventilation.

This additional ventilation requirement increases the building heat loss and fuel cost.

Let us assume the following example:

Temperature Differential *(inside temperature less outside design temperature)*	=	65ºF
Building Heat Loss	=	125,000 Btu/hr
Infrared Compensation Factor *(based on 16' mounting height)*	=	0.80
Infrared Heat Required	=	100,000 Btu/hr

	TYPE OF INFRARED HEATER SELECTED	
	Tube	**Ceramic**
Input:	100,000 Btu/hr (direct vented)	100,000 Btu/hr *(requires gravity or mechanical ventilation of 4 CFM per 1,000 Btu/hr)*
Additional Ventilation Required:	0 CFM	4 CFM/1,000 Btu/hr = 400 CFM
Heat Loss Due to Additional Ventilation:	0 Btu/hr	Q = CFM x 60 min/hr x TD x 0.018 = 400 x 60 x 65 x 0.018 = 28,080 Btu/hr
Total Input Required:	100,000 Btu/hr	128,080 Btu/hr

CONCLUSION: It will require a 28% larger capacity ceramic infrared heating system than the tube infrared heating system to satisfy the building heat loss and comply with codes. Likewise, the fuel cost of a ceramic heating system can be as high as 28% more than a tube infrared heating system.

D. GENERAL PRECAUTIONS FOR VENTILATION OF INDIRECT VENTED HEATERS

1. NFPA 54, "Gas Appliances and Gas Piping" states, "where unvented infrared heaters are used, natural or mechanical means shall be provided to exhaust at least 4 CFM per 1,000 Btu/hr input of installed heaters" and "exhaust openings for removing flue products shall be above the level of the heaters." Many state and local codes have adopted similar requirements.

2. Additional airflow may be required to reduce the possibility of condensation, particularly in buildings where the roof has insufficient insulation qualities.

3. The inlet air openings must be located below the level of the heaters. Failure to do so often results in condensation forming within insulated buildings and insufficient airflow for the proper dilution of products of combustion.

4. Mechanical exhaust is required in some localities and advisable in all areas where other sources of products of combustion (e.g., carbon monoxide) exist, such as forklift trucks. The usual requirement is 5,000 CFM per 2-4 ton truck.

5. Mechanical exhaust should be accompanied by inlet air openings of 1.5 to 3.0 square inches (free area) per 1,000 Btu/hr installed input in order to maintain an inlet air velocity of 200 to 400 FPM.

6. The inlet air openings should be small (1 to 2 sq. ft.) and well distributed. They should be located on at least two outside walls and on all outside walls on larger installations. When sizing the inlet air openings with louvers or grillwork, be sure to use the effective open area (normally 60% open area).

7. Do not locate the inlet air opening directly under a heater mounted along an outside wall, and do not locate the heater directly under a ventilator. This will cause the incoming air to circulate up through the heater and straight out of the building, carrying with it 35 to 40 percent of the available heat.

8. Mechanical exhaust (power vents) should always be connected into the heater thermostat circuit to provide positive ventilation whenever the heaters are operating.

9. Consider all partitions or curtains above heater level as separate ventilating zones.

10. Locate the ventilators at the highest point in the building.

11. Be especially careful with sidewall ventilators and exhausters located in the gable ends of a building. Sidewall exhausters used for heater ventilation must be located above the heater level in order to comply with NFPA 54. Also, certain localities have code requirements specifying that when sidewall ventilators are used for heater ventilation, they must be sized to provide 10 CFM per 1,000 Btu/hr input of installed heaters. This is done to allow for the reduced efficiency of the ventilators when the wind is blowing against the side of the building where the exhausters are located. Exhaust fans in the gable ends of a building have a tendency to short circuit and exhaust air only from that end of the building where the exhausters are located, especially if large overhead doors are nearby.

12. Heaters should not be installed in areas where there are halogenated hydrocarbons present in the atmosphere. This includes, but is not limited to, perchloroethylene, trichloroethylene, methylene, chloride, methyl chloroform and freon. Some vapors frequently encountered in industry are decomposed to hazardous or corrosive gases in the presence of high temperature and air. Trichloroethylene, used for degreasing, may form phosgene and hydrogen chloride. Phosgene ($COCl_2$) is extremely toxic, the threshold limit being 1 PPM. Hydrogen chloride is not as toxic, 5 PPM, but is corrosive to practically all metals. Hydrogen chloride and phosgene are not formed in all cases, but this condition is possible and vapor producing equipment should be isolated by suitable exhaust or other means. Be careful not to exhaust the vapor in such a manner that it will re-enter the building through the outside combustion inlets.

E. CONDENSATION

Unvented gas-fired infrared heaters release all of the available heat as well as products of combustion into the building enclosure. They include carbon dioxide, water vapor and a minute amount of carbon monoxide.

The water vapor released by the burning gases rises with the products of combustion to the area above the heaters. The amount of water vapor released is not excessive (0.094 lb./1,000 Btu/hr Natural Gas) or (0.076 lb./1,000 Btu/hr Propane Gas); however, when added to the water vapor contained in the incoming dilution air, the humidity level is raised and condensation (sweating) may occur on cold building surfaces. This is a potential problem in a building with an uninsulated metal roof or one without a suitable vapor barrier covering the insulation on the heated side.

Relative Humidity (RH) is a function of the building air temperature (which will vary from floor to ceiling) and moisture content; therefore, it will be referred to as the dew point temperature. The dew point is the temperature at which condensation (sweating) will occur. The following general precautions should be observed for indirect vented infrared heaters:

1. Roof skylights unless insulated by double-glazing are a potential source of condensation (sweating).

2. When using Fiberglas® batt or blanket, or other hygroscopic type of insulation, it must have a suitable vapor barrier applied to the inside face (the heated side). All edges, cuts or holes must be tightly sealed with suitable tape, particularly on the underside of the roof. If moisture is allowed to penetrate the vapor barrier, it will condense on the underside roof surface and soak into the insulation to the point where it is of no value and may cause moisture damage. If allowed to continue, the weight of the added moisture will pull the insulation and vapor barrier completely away from the roof surface, requiring replacement.

3. Occasionally, condensation may form on the metal fastener used to hold the blanket type of insulation to a metal roof. The reason for this is that the metal fastener is attached directly to the roof surface and stays at relatively the same temperature as the roof skin temperature. If condensation does occur, it may be necessary to apply putty or a caulking compound to the fastener where it projects through the insulation on the underside of the ceiling. For the same reason, sweating may occur on the roof beams of a steel building. The insulation is compressed between the roof deck and the steel beams where it is held in place with a sheet metal screw. If sweating does occur, it will usually run down the inside of the beam and remain unnoticed.

4. Be especially careful of the spray-on foam type of insulation. When applied, it must completely cover the inside of the purlins and beams supporting the roof deck. Any exposed metal beams or portions thereof may sweat.

5. In order to eliminate potential condensation problems, provide the correct amount of dilution air for unvented infrared heaters as outlined in Section V, or whenever possible, direct vent tube-type infrared heaters outside the building.

VI. GAS PIPING

MAXIMUM CAPACITY OF PIPE IN CUBIC FEET OF GAS PER HOUR FOR GAS PRESSURES OF 0.5 PSI OR LESS AND A PRESSURE DROP OF 0.5 INCH WATER COLUMN, BASED ON 0.60 SPECIFIC GRAVITY GAS.

Nominal Iron Pipe Size *(Inches)*	Internal Diameter *(Inches)*	Length of Pipe *(feet)*													
		10	20	30	40	50	60	70	80	90	100	125	150	175	200
1/4	0.364	43	29	24	20	18	16	15	14	13	12	11	10	9	8
3/8	0.493	95	65	52	45	40	36	33	31	29	27	24	22	20	19
1/2	0.622	175	120	97	82	73	66	61	57	53	50	44	40	37	35
3/4	0.824	360	250	200	170	151	138	125	118	110	103	93	84	77	72
1	1.049	680	465	375	320	285	260	240	220	205	195	175	160	145	135
1-1/4	1.380	1,400	950	770	660	580	530	490	460	430	400	360	325	300	280
1-1/2	1.610	2,100	1,460	1,180	990	900	810	750	690	650	620	550	500	460	430
2	2.067	3,950	2,750	2,200	1,900	1,680	1,520	1,400	1,300	1,220	1,150	1,020	950	850	800
2-1/2	2.469	6,300	4,350	3,520	3,000	2,650	2,400	2,250	2,050	1,950	1,850	1,650	1,500	1,370	1,280
3	3.068	11,000	7,700	6,250	5,300	4,750	4,300	3,900	3,700	3,450	3,250	2,950	2,650	2,450	2,280
4	4.026	23,000	15,800	12,800	10,900	9,700	8,800	8,100	7,500	7,200	6,700	6,000	5,500	5,000	4,600

MAXIMUM CAPACITY OF PIPE IN CUBIC FEET OF GAS PER HOUR FOR GAS PRESSURE OF 1.0 PSI AND A PRESSURE DROP OF 10%, BASED ON 0.60 SPECIFIC GRAVITY GAS.

Pipe Size of Schedule 40 Standard Pipe *(Inches)*	Internal Diameter *(Inches)*	Total Equivalent Length of Pipe *(feet)*										
		50	100	150	200	250	300	400	500	1000	1500	2000
1.00	1.049	717	493	396	338	300	272	233	206	142	114	97
1.25	1.380	1,471	1,011	812	695	616	558	478	423	291	234	200
1.50	1.610	2,204	1,515	1,217	1,041	923	836	716	634	436	350	300
2.00	2.067	4,245	2,918	2,343	2,005	1,777	1,610	1,378	1,222	840	674	577
2.50	2.469	6,766	4,651	3,735	3,196	2,833	2,567	2,197	1,947	1,338	1,075	920
3.00	3.068	11,962	8,221	6,602	5,650	5,008	4,538	3,884	3,442	2,366	1,900	1,626
3.50	3.548	17,514	12,037	9,666	8,273	7,332	6,644	5,686	5,039	3,464	2,781	2,381
4.00	4.026	24,398	16,769	13,466	11,525	10,214	9,255	7,921	7,020	4,825	3,875	3,316
5.00	5.047	44,140	30,337	24,362	20,851	18,479	16,744	14,330	12,701	8,729	7,010	6,000
6.00	6.065	71,473	49,123	39,447	33,762	29,923	27,112	23,204	20,566	14,135	11,351	9,715
8.00	7.981	146,849	100,929	81,049	69,368	61,479	55,705	47,676	42,254	29,041	23,321	19,960
10.00	10.020	266,718	183,314	147,207	125,990	111,663	101,175	86,592	76,745	52,747	42,357	36,252
12.00	11.938	422,248	290,209	233,048	199,459	176,777	160,172	137,087	121,498	83,505	67,057	57,392

MAXIMUM CAPACITY OF PIPE IN CUBIC FEET OF GAS PER HOUR FOR GAS PRESSURE OF 5.0 PSI AND A PRESSURE DROP OF 10%, BASED ON 0.60 SPECIFIC GRAVITY GAS.

Pipe Size of Schedule 40 Standard Pipe *(Inches)*	Internal Diameter *(Inches)*	Total Equivalent Length of Pipe *(feet)*										
		50	100	150	200	250	300	400	500	1000	1500	2000
1.00	1.049	1,989	1,367	1,098	940	833	755	646	572	393	316	270
1.25	1.380	4,084	2,807	2,254	1,929	1,710	1,549	1,326	1,175	808	649	555
1.50	1.610	6,120	4,206	3,378	2,891	2,562	2,321	1,987	1,761	1,210	972	832
2.00	2.067	11,786	8,101	6,505	5,567	4,934	4,471	3,827	3,391	2,331	1,872	1,602
2.50	2.469	18,785	12,911	10,368	8,874	7,865	7,126	6,099	5,405	3,715	2,983	2,553
3.00	3.068	33,209	22,824	18,329	15,687	13,903	12,597	10,782	9,556	6,568	5,274	4,514
3.50	3.548	48,623	33,418	26,836	22,968	20,356	18,444	15,786	13,991	9,616	7,722	6,609
4.00	4.026	67,736	46,555	37,385	31,997	28,358	25,694	21,991	19,490	13,396	10,757	9,207
5.00	5.047	122,544	84,224	67,635	57,887	51,304	46,485	39,785	35,261	24,235	19,461	16,656
6.00	6.065	198,427	136,378	109,516	93,732	83,073	75,270	64,421	57,095	39,241	31,512	26,970
8.00	7.981	407,692	280,204	225,014	192,583	170,683	154,651	132,361	117,309	80,626	64,745	55,414
10.00	10.020	740,477	508,926	408,686	349,782	310,005	280,887	240,403	213,065	146,438	117,595	100,646
12.00	11.938	1,172,269	805,694	647,001	553,749	490,777	444,680	380,588	337,309	231,830	186,168	159,336

Source: ANSI Z223.1, 1996

VII. THERMAL VS. RADIANT EFFICIENCIES

A. *Gas-Fired Infrared Tube Heaters — Thermal and Radiant Efficiencies*

Some manufacturers are promoting the idea that thermal efficiency is the correct performance measurement for infrared tube heaters. We do not believe this to be the case. We believe the correct performance measurement should be radiant efficiency.

Currently, all infrared heaters are tested by the A.G.A. Labs to the American National Standard Z83.6b-1992. This standard requires all infrared heaters to pass a minimum radiant efficiency or radiant coefficient of 35%. Unfortunately, tube type heaters have physical dimensions that presently prohibit them from being tested for a minimum radiant efficiency. As an interim performance standard, the industry agreed to require tube type heaters to have a thermal efficiency of not less than 70%. Other infrared heaters, such as ceramic or high intensity infrared heaters, must be tested for radiant efficiency and are not tested for thermal efficiency.

Thermal efficiency simply measures the flue loss of a heater. In using this measurement, we know that some percentage of a heater's input goes up the flue, and the rest is left within the building in the form of either convection or radiation. The measurement of thermal efficiency makes no distinction between energy released in the form of convection or radiation. It is for this reason that convective air blown systems (e.g., unit heaters) use thermal efficiency as their performance measure.

As is clear, radiant infrared heating systems are very different from convective air-blown systems. This is because they seek to maximize the percentage of heat released in the form of radiation and minimize the amount of convection. This infrared radiation is generally recognized to be the principal source of fuel savings when compared to other types of heating systems. In fact, in some circumstances, radiant infrared systems have saved building owners up to 70% on their fuel bills when compared with convective forced air heating systems.

In order to improve the performance measurement for all infrared heaters, the Infrared Division of the Gas Appliance Manufacturers Association (GAMA) has been working closely with the Gas Research Institute to develop a method for testing the radiant efficiency of tube type heaters. To date, excellent progress has been made in this area. We are hopeful that some additional work will soon be completed and adopted which will allow all types of gas infrared heaters to be tested to this performance standard.

B. *Space-Ray Tube Heaters*

At Space-Ray, we have designed and manufactured gas infrared tube type heaters since 1968. Our design objective has always been to maximize the radiant output and minimize the convective output of our tube heaters. To accomplish this, we concentrate on three things:

1) Temperature of the emitter (tube) surface,
2) Emissivity of the emitter surface, and
3) Size of the emitter surface.

These have always been the principal factors necessary to achieve high radiant efficiency.

The formula for determining the radiant output (radiant efficiency) of a gas infrared heater is:

$$\mathbf{R = SEA\,(T^4 - Ta^4)}$$

Where:	R	=	radiant output
	S	=	Stefan-Boltzmann constant
	E	=	emissivity of radiating surface
	A	=	surface area
	T	=	emitter surface temperature in ºR
	Ta	=	ambient temperature ºR

In reviewing this formula, you will note that the most significant variables in the equation are actually the net temperature above ambient of the emitter surface as well as the emissivity of the radiating surface. Any incremental increase in the emissivity or the emitter temperature (increased exponentially to the fourth power) greatly increases the radiant output of the tube heaters.

1) Temperature of the Emitter

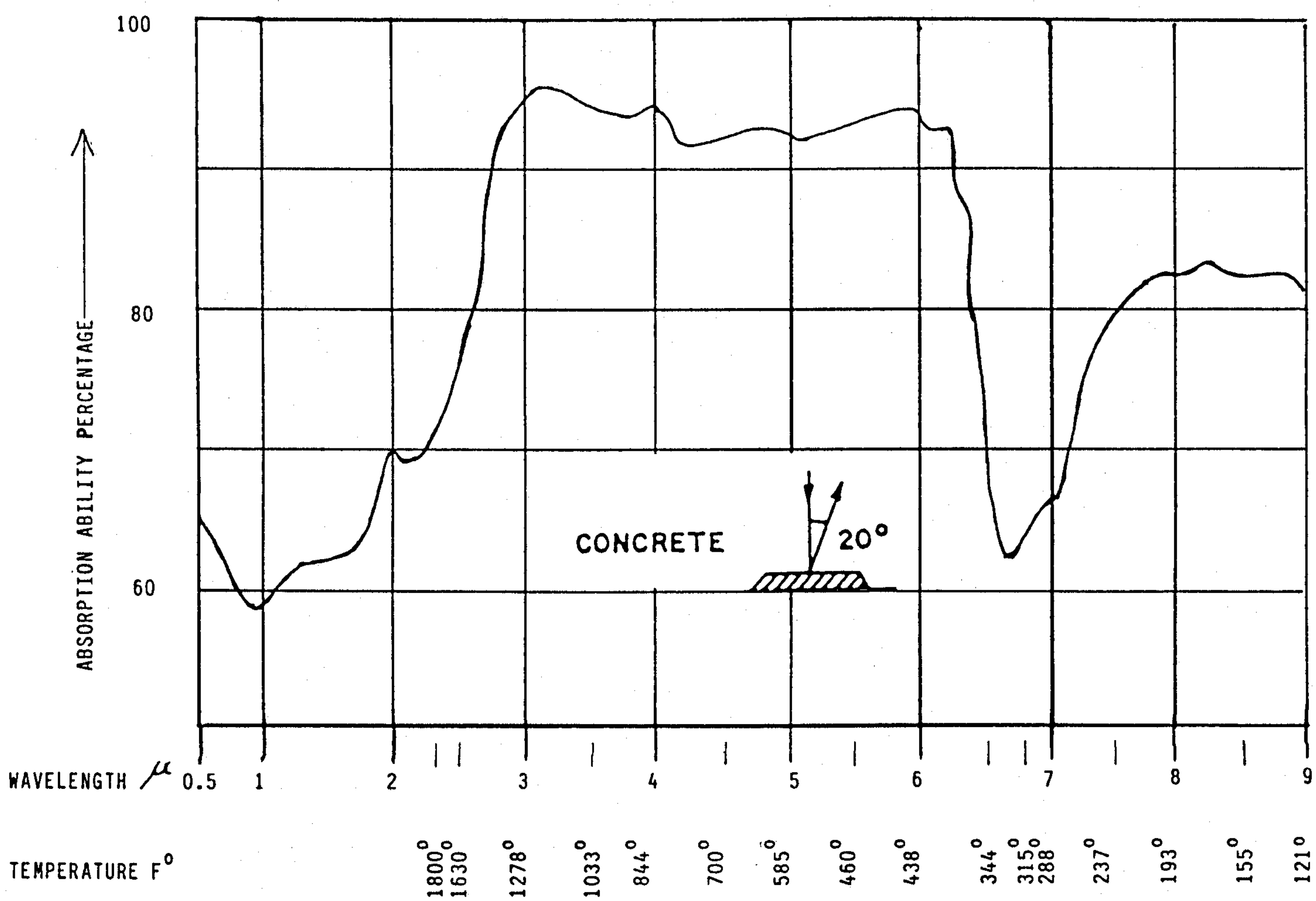
Because the temperature of the emitter (tube) is such a significant variable when determining a heater's radiant efficiency, we have always sought to obtain the highest average surface temperature within the limits of a given material. In the case of our LTU series heaters, they average a reasonably uniform temperature of 750ºF. The uniformity of our tube temperatures is enhanced by:

- Pulling the products of combustion through the tube via a draft inducer,
- Use of a tube restrictor plate, and
- Unique design of the burner creating a highly luminous flame.

The emitter temperatures and the corresponding wavelength of infrared radiation that is emitted play an important role in the design of infrared tube heaters. If we evaluate the basic heat transfer modes in the building (e.g., direct radiation from the heating system, conduction in the concrete slab and the convected heat transfer from the warm floor and machinery), we would like to maximize the radiant energy absorbed by the concrete floor. The infrared heating system works from the bottom up, heating the concrete slab and machinery first, then the surrounding ambient air.

The chart on the next page shows the absorption of infrared energy by concrete at various wavelengths and temperatures.

ABSORPTION OF INFRARED ENERGY BY CONCRETE AT VARIOUS WAVELENGTHS AND TEMPERATURES



ABSORPTION OF INFRARED ENERGY BY CONCRETE IS GREATER THAN 90% IN THE RANGE OF 2.8 TO 6.2 MICRONS.

The absorption of infrared energy by the concrete will depend on the emitter tube temperature as well as the wavelength of the infrared radiation. For example, if the emitter temperature is 1800ºF, then 70% of the infrared energy generated will be absorbed by the concrete (30% reflected). Conversely, if the emitter temperature is 1250ºF, then 95% of the infrared energy generated will be absorbed by the concrete (only 5% reflected). As more radiant energy is absorbed by the concrete floor, more thermal energy is stored in the floor slab. The floor slab, functioning as a giant, low-temperature radiant emitter, will radiate to the surrounding ambient air and satisfy the thermostat setting more quickly. Evaluating the absorption of infrared energy by concrete, you will note that greater than 93% of infrared energy emitted between 3.0 and 6.2 microns is absorbed by the concrete. Therefore, it is desirable to design a tube heater with emitter tube temperatures between 1278ºF and 380ºF. As the emitter temperatures drop below 380ºF, the absorptivity of infrared radiation by concrete drops as low as 62%. Some products promote the idea of longer tubes operating at near condensing modes. After a certain distance from the burner, the tube temperatures drop below 380ºF, where the amount of radiant energy emitted (or radiant output) is decreased to a negligible amount and the absorption of infrared energy by the concrete drops tremendously. Therefore, the long continuous tube systems with low emitter temperatures do very little to maintain the thermal mass of the concrete floor and do not have the quick recovery capability of the tube heaters with higher operating temperatures.

2) Emissivity of the Emitter Surface

Space-Ray provides a variety of emitter materials (calorized-aluminized steel, calorized Alumi-Therm steel, aluminized steel, hot-rolled steel and stainless steel) which can affect the radiant output of the tube heaters.

The emissivities of various materials vary widely with wavelength, temperature and surface conditions. The emissivities of emitter materials offered by Space-Ray are as follows:

Calorized Aluminized Steel:	**0.80 - 0.86**
Hot Rolled Steel:	**0.66**
Aluminized Steel:	**0.25 - 0.30**
Stainless Steel:	**0.20 - 0.25**

A comparison of various emitter materials is presented in Table 1.

The use of calorized-aluminized steel emitters, as well as Space-Ray's tube heater design, translates to the highest radiant efficient tube heaters. The calorization process creates a highly absorptive interior tube surface as well as a highly emissive exterior surface. This in turn maximizes the radiant energy transfer from the luminous flame. Our measured tube emissivities range from 0.83 to 0.86, with the maximum possible being 1.0. These measured results, in fact, exceed the estimated emissivity of 0.80 provided to us by ARMCO Steel Company. Space-Ray's 27+ years of field experience shows that the calorization process increases:

- The corrosion resistance of the emitter surface,
- The longevity of the emitter surface.

Some Space-Ray users report an average emitter life of 22 years even in an 11,000 degree day operating environment. Please consult your Space-Ray Representative for further information regarding the calorization process and the benefits of calorized-aluminized steel emitter tubes.

From time to time, infrared tube heaters are utilized in highly corrosive and humid environments. Due to the material's high corrosion resistance, some heating engineers are specifying aluminized or stainless steel emitter tubes. Space-Ray can provide aluminized-steel or stainless steel emitter tubes in these applications, but we do not recommend these emitter materials due to their poor surface emissivities. Radiant emitter tubes made of these materials are poor radiant efficient heaters. Space-Ray's 27+ year's field experience with tube heaters show that calorized emitter tubes provide an excellent life expectancy of 12+ years in high humidity and corrosive environments.

3. Size of the Emitter Surface

We also seek to maximize the size of the emitter surface subject to maximizing temperature and emissivity, as well as providing our customers with a product that is easily and affordably installed and serviced.

4. Other Considerations

Each of our tube heaters is completely encased above the tube in a highly efficient aluminum reflector. This permits over 97.5% of the radiation emitted from the tube's surface to be reflected directly into the space to be heated. Also, we utilize cast iron burner assemblies for long life, Honeywell gas control systems and permanently lubricated maintenance free draft inducer motors for their reliability.

Utilizing the Stefan-Boltzmann formula, the calculated radiant efficiencies of Space-Ray tube heaters are as follows:

RADIANT EFFICIENCY OF TUBE HEATERS

HEATER	TUBE TYPE	AVERAGE TUBE TEMPERATURE *(°F)*	RADIANT EFFICIENCY
LTU[1]	U-Tube	730	65.6%
LTS[1]	Straight Tube	679	58.3%
ETS[2]	Straight Tube	711	52.8%

1) with calorized-aluminized steel emitter
2) with calorized-aluminized steel combustion chamber (10 ft.), the remaining hot-rolled steel emitter.

The development of radiant efficiency methodology is still continuing. As an industry group, we are observing the difficulty in measuring the radiant output of tube heaters. The current proposed methodology shows significant deviation between the participating laboratories. Space-Ray will support the development of the radiant efficiency standards with the hope that the above-calculated values can be validated by independent testing laboratories.

In summary, in designing our Space-Ray tube heaters, we seek to maximize the radiant output and minimize the convective output. We firmly believe that this combination of features translate into additional fuel savings for Space-Ray customers.

-- TABLE I --

THE COMPARISON OF VARIOUS EMITTER MATERIALS USED BY INFRARED HEATER MANUFACTURERS

EMITTER TUBE MATERIAL	EMISSIVITY *(External)*	ABSORPTIVITY *(Internal)*	CORROSION RESISTANCE *External*	CORROSION RESISTANCE *Internal*	MAXIMUM OPERATING TEMPERATURE	PEELING, FLAKING, SCRATCHING RESISTANCE	RADIANT OUTPUT
ALUMINIZED STEEL	0.25 - 0.30 Very poor	Very poor	Very good	Very good	1100°F	Excellent	Poor
CALORIZED ALUMINIZED STEEL	0.80 - 0.86 Very good	Very good	Very good	Very good	1250°F	Excellent	Excellent
CALORIZED ALUMI-THERM STEEL	0.80 - 0.86 Very good	Very good	Very good	Very good	1400°F	Excellent	Excellent
HOT-ROLLED STEEL	0.66 Good	Good	Average	Average	900°F	Average	Good
HOT-ROLLED STEEL with high emissive paint on the exterior	0.66 - 0.75[1] Good	Good	Very good	Average	900°F	Fair[2]	Good
ALUMINIZED STEEL with high emissive paint on the exterior	0.80[1] Very good	Very poor	Very good	Very good	1100°F	Fair[2]	Fairly good
STAINLESS STEEL – SS409	0.20 - 0.25 Very poor	Very poor	Very good	Very good	1250°F	Excellent	Poor

[1] Emissivity of high temperature paint @ 1100ºF operating temperature
[2] Dependent on surface preparation

Sources:

ASHRAE Handbook Fundamentals, 1989
Combustion Technical Manual, Industrial Heating Equipment Association, 1988
Gas Engineers Handbook, Industrial Press, 1965
Heat Transfer, J. P. Holman, McGraw-Hill, 1963
American National Standard Z83.6, American National Standards Institute, Inc. 1985

VIII. ESTIMATING FUEL CONSUMPTION

The formula for estimating probable fuel consumption is based on the following variables, many of which must be determined for each individual job:

$$\text{Estimated cost per year} = \frac{\mathbf{HL \times 24 \times DD \times cost\ per\ unit\ of\ fuel}}{\mathbf{FCF \times Td \times V}} \times \mathbf{C_D}$$

Where:

HL = Building Heat Loss - Total calculated heat loss for the building. For infrared systems, you would use the adjusted heat loss (see Section IV, Item C, *Infrared Heat Loss compensation Factor*).

FCF = Fuel Conversion Factor based on annual operating efficiency -- primarily intended for comparing infrared heating systems to unit heaters, furnaces, and boilers.

Td = Design Temperature Difference (inside design temperature minus outside design temperature).

24 = Hours per day.

V = Heating Value of fuel (Btu/hr per unit of fuel).

DD = Degree days - total accumulated degrees per year (accumulated each 24 hour day) that the average mean temperature for the day has fallen below a pre-determined base temperature.

C_D = Empirical Correction Factor for heating effect vs. 65ºF degree days, as indicated in chart below:

DEGREE DAYS	CORRECTION FACTOR
2000	0.750
3000	0.700
4000	0.650
5000	0.600
6000	0.607
7000	0.621
8000	0.635
9000	0.649

Be careful to use the correct degree day figures. If your customer requests that you prepare a fuel cost estimate for heating his facility with infrared, he will probably compare it with last year's total fuel cost (if replacing existing heating equipment), so it is recommended that you prepare a cost estimate for the ten-year average degree days as well as the previous year's degree days if they differ by a substantial amount. Natural gas usually is billed in either CCF (100 cubic feet) or MCF (1,000 cubic feet). Natural gas contains approximately 1,000 Btu/hr per cubic feet of gas, so 1 CCF = 100,000 Btu/hr = (1 therm) and 1 MCF = 1,000,000 Btu/hr.

Example: Natural gas at \$0.60 per CCF or \$0.60 per therm =
\$0.60 per 100,000 Btu/hr or \$6.00 per 1,000,000 Btu/hr.

(Natural gas usually is sold on a sliding scale where the unit cost depends on the quantity used per month. The average cost will have to be determined for estimating fuel cost.)

Propane gas = \$.80 per gallon = 91,326 Btu/hr per gallon
Electricity = \$.08 per kilowatt = 3,410 Btu/hr per kilowatt

ESTIMATED FUEL COST

$$\text{Estimated cost per year} = \frac{\mathbf{HL \times 24 \times DD \times cost\ per\ unit\ of\ fuel}}{\mathbf{FCF \times Td \times V}} \times \mathbf{C_D}$$

The major variables for any given locality are the design temperature difference, building heat loss and cost per unit of fuel. The variables may then be reduced to multipliers that will greatly simplify the computations.

Estimated fuel cost of a building in Pittsburgh, PA is as follows:

Building Heat Loss:	2,665,000
Infrared Compensation Factor:	0.80
Adjusted Building Heat Loss:	2,132,000
Degree Days at 65ºF Base:	5950
Temperature Differential:	65ºF
Fuel Cost:	\$6.00/MCF

$$\text{Estimated Fuel Cost per Year} = \frac{2{,}132{,}000 \times 24 \times 5950 \times \$6.00}{1 \times 65 \times 1{,}000{,}000} \times 0.607$$

Estimated Fuel Cost per Year = \$17,047.89

NIGHT SETBACK

Night Setback will normally conserve enough fuel to make a considerable reduction in annual fuel costs depending on the ratio of degree days at night setback temperature versus the degree days at inside design temperature.

Assume the inside temperature is 65ºF for ten hours per day, five days per week. The night setback is 55ºF for 14 hours per day, for five days per week and 24 hours per day for two days per week.

24 hours x 7 days	=	168 hours	
10 hours x 5 days	=	50 hours	
14 hours x 5 days + 24 hours x 2 days	=	118 hours	
Average hours per day = (50/168) x 24	=	7.143	@ 65ºF
Average hours per day = (118/168) x 24	=	16.857	@ 55ºF
Degree days at 65ºF base	=	5950	
Degree days at 55ºF base	=	3631	

$3631/5950 = 0.61$ (61%) ratio DD @ Design vs. DD @ Night Setback

$\frac{7.143}{65} + \frac{16.857 \times .61}{55} = 0.297$ (Multiplier for Night Setback)

Estimated cost per year = 0.297 (multiplier) x 2.132 (million Btu/hr) x 5950 (DD) x $6.00 x 0.607 ($C_D$)

Estimated cost per year = $13,721.47 with Night Setback

> CAUTION: Be very careful to word your proposal to the customer that your fuel cost estimate is based on the best information you have available. You have no control over the unit fuel cost, weather conditions or building usage; therefore, you cannot guarantee fuel cost figures.

A. DEGREE DAYS

The degree day figure for the day will be the difference between the inside design temperature (degree day base) and the average mean temperature for the day.

Example:	Maximum temperature for the day	=	33º
	Minimum temperature for the day	=	20º
	Average Mean temperature for the day	=	27º

Most industrial heating jobs will be designed for a 65 degree inside design temperature; therefore, they will use the 65 degree base.

@ 65º base:	65-27 = 38 D.D.
@ 55º base:	55-27 = 28 D.D.
@ 45º base:	45-27 = 8 D.D.

The total accumulated degree days (for the year) multiplied by 24 (hours) gives an estimate of the total number of hours and degrees the thermometer has registered a temperature below a predetermined base throughout the entire heating season.

Example:	Pittsburgh, PA		
	Degree days at 65º base	=	5950
	Degree days at 60º base	=	4711
	Degree days at 55º base	=	3631
	Degree days at 50º base	=	2709
	Degree days at 45º base	=	1934
	Lowest mean temperature (winter)	=	0.0º
	Highest mean temperature (winter)	=	73.0º
	Average mean temperature (winter)	=	43.8º

The above figures were extracted from the National Climatic Data Center Reports.

Annual Degree Days for major US cities are provided on the following pages. In addition, the US Weather Bureau and/or local utility companies in your area can usually provide the degree day figures for the previous two years and a ten year average.

A. Degree Days

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
ALABAMA	Birmingham	17	2943	2066	1368	851	500	256
	Huntsville	11	3279	2379	1636	1083	676	393
	Mobile	25	1695	1073	633	347	160	73
	Montgomery	22	2277	1519	960	568	306	136
ALASKA	Anchorage	-23	10816	9038	5308	3604	2127	1030
	Elmendorf AFB	-51	10940	9173	7559	6140	4962	3962
	Fairbanks	-51	14274	12616	11086	9707	8477	7358
	Juneau	-4	9105	7308	5626	4190	3037	2143
	Nome	-31	14371	12562	10799	9140	7624	6349
ARIZONA	Flagstaff	-2	7254	5729	4395	3253	2277	1486
	Phoenix	31	1442	830	403	150	38	0
	Tucson	28	1734	1026	520	215	61	7
	Winslow	5	4839	3720	2748	1934	1287	801
ARKANSAS	Fort Smith	12	3477	2561	1794	1188	718	403
	Little Rock	15	3152	2269	1558	988	578	302
	Texarkana AP	18	2501	1714	1104	643	338	141
	Fayetteville AP	7	4174	3143	2270	1560	996	591
CALIFORNIA	Bakersfield	30	2128	1318	722	353	140	43
	Eureka	31	4725	2950	1451	530	138	22
	Fresno	28	2647	1725	993	482	193	51
	Long Beach	41	1485	689	246	48	0	0
	Los Angeles	37	1595	702	235	42	0	0
	Oakland	34	2877	1555	689	244	54	6
	Sacramento	30	2772	1785	1002	469	175	43
	San Diego	42	1284	491	124	12	0	0
	San Francisco	35	3161	1744	787	278	60	5
COLORADO	Alamosa	-21	8717	7133	5752	4554	3532	2664
	Colorado Springs	-3	6346	5007	3830	2840	1997	1330
	Denver	-5	6014	4734	3599	2634	1818	1186
	Grand Junction	2	5683	4518	3505	2632	1902	1315
	Pueblo	-7	5465	4279	3241	2362	1633	1068
CONNECTICUT	Bridgeport	6	5501	4300	3243	2327	1577	992
	Hartford	3	6174	4939	3842	2886	2075	1414
	Norwalk	6	5847	4601	3499	2541	1748	1123
DELAWARE	Dover	11	4356	3279	2371	1623	1042	625
	Wilmington	10	4986	3854	2864	2021	1358	849
D.C.	Washington Nat. AP	14	4122	3090	2211	1510	960	575
FLORIDA	Daytona Beach	32	900	483	220	77	22	0
	Jacksonville	29	1402	846	476	233	95	38
	Miami	44	274	89	26	0	0	0
	Orlando McCoy AFB	35	656	336	132	50	15	0
	Pensacola	25	1571	980	574	304	136	61
	Tallahassee	27	1652	1030	595	314	134	57
	Tampa	36	739	390	166	60	19	0

OUTSIDE DESIGN TEMPERATURES AND DEGREE DAYS TO SELECTED BASES
(Continued)

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
GEORGIA	Athens	18	2965	2080	1364	832	467	223
	Atlanta	17	3021	2128	1417	885	518	270
	Augusta	20	2568	1730	1102	649	351	158
	Columbus	21	2356	1575	997	582	310	133
	Macon	21	2279	1512	948	548	286	121
	Rome	17	3122	2204	1470	923	544	291
	Savannah	24	1921	1242	753	424	202	84
IDAHO	Boise	3	5802	4501	3353	2391	1599	1023
	Coeur d’Alene	-8	6461	5049	3796	2718	1832	1133
	Lewiston	-1	5429	4138	3008	2081	1335	810
	Pocatello	-8	7123	5740	4488	3396	2483	1729
ILLINOIS	Carbondale	2	4563	3516	2639	1900	1316	868
	Chicago	-5	6177	5007	3964	3040	2266	1623
	Moline	-5	6498	5312	4272	3355	2551	1897
	Peoria	-8	6226	5059	4028	3128	2351	1721
	Rockford	-9	6952	5715	4615	3654	2004	2091
	Springfield	-3	5654	4539	3561	2709	2002	1425
INDIANA	Evansville	4	4729	3658	2749	1995	1395	922
	Fort Wayne	-4	6320	5113	4038	3091	2288	1627
	Indianapolis	-2	5650	4498	3490	2631	1908	1331
	South Bend	-3	6377	5141	4060	3111	2302	1644
	Terre Haute	-2	5521	4390	3400	2560	1860	1298
IOWA	Cedar Rapids	-10	6671	5472	4431	3511	2696	2023
	Des Moines	-10	6554	5394	4363	3463	2672	2015
	Dubuque	-12	7375	6092	4971	3996	3117	2374
	Ottumwa	-8	6339	5186	4170	3275	2498	1860
	Sioux City	-11	6947	5754	4699	3775	2951	2255
	Waterloo	-15	7537	6270	5155	4174	3293	2537
KANSAS	Dodge City	0	5059	3973	3034	2223	1567	1034
	Goodland	-5	6099	4866	3763	2813	2009	1368
	Salina	0	5187	4106	3162	2339	1685	1141
	Topeka	0	5319	4219	3272	2443	1774	1218
	Wichita	3	4787	3740	2843	2083	1469	972
KENTUCKY	Ashland	5	4900	3775	2839	2022	1395	917
	Bowling Green	4	4309	3274	2408	1688	1124	714
	Covington	1	5247	4117	3131	2298	1619	1081
	Lexington	3	4814	3720	2795	2000	1386	905
	Louisville	5	4525	3465	2577	1824	1243	809
	Paducah	7	4130	3141	2303	1614	1062	666
LOUISIANA	Alexandria	23	1961	1283	779	445	220	93
	Baton Rouge	25	1673	1062	626	339	152	67
	Lake Charles	27	1579	982	559	284	120	46
	New Orleans	29	1490	920	532	273	118	47
	Shreveport	20	2269	1521	961	558	292	127

OUTSIDE DESIGN TEMPERATURES AND DEGREE DAYS TO SELECTED BASES

(Continued)

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
MAINE	Augusta	-7	7598	6176	4937	3839	2879	2082
	Bangor	-11	7947	6491	5220	4094	3103	2279
	Caribou	-18	9616	8043	6650	5424	4330	3367
	Portland	-6	7501	6042	4766	3651	2688	1900
MARYLAND	Baltimore	10	4706	3600	2646	1848	1218	743
	Cumberland	6	5106	3952	2967	2119	1457	926
	Hagerstown	8	5086	3941	2960	2125	1466	940
	Salisbury	12	4016	2970	2089	1395	867	511
MASSACHUSETTS	Boston	6	5593	4368	3303	2383	1630	1030
	New Bedford	5	5305	4069	3004	2107	1389	849
	Springfield	-5	5953	4718	3642	2706	1922	1288
	Worcester	0	6950	5588	4404	3364	2461	1727
MICHIGAN	Alpena	-11	8410	6893	5572	4425	3411	2530
	Detroit	3	6563	5308	4195	3212	2346	1639
	Escanaba	-11	8547	7030	5700	4550	3529	2639
	Flint	-4	7068	5737	4585	3569	2675	1927
	Grand Rapids	1	6927	5622	4481	3477	2591	1859
	Lansing	-3	6987	5676	4537	3535	2648	1914
	Marquette	-12	9520	7955	6570	5346	4291	3331
	Sault Ste. Marie	-12	9305	7719	6321	5105	4040	3098
MINNESOTA	Duluth	-21	9901	8338	6959	5759	4698	3766
	International Falls	-29	10604	9066	7699	6496	5416	4472
	Minneapolis/St. Paul	-16	8007	6728	5586	4583	3690	2891
	Rochester	-12	8277	6929	5758	4717	3802	2981
MISSISSIPPI	Biloxi	28	1498	908	502	246	99	35
	Jackson	21	2839	1628	1056	647	363	177
	Natchez	23	1941	1278	789	460	231	100
	Tupelo	14	3088	2207	1492	946	555	292
	Vicksburg	22	2201	1465	919	535	279	123
MISSOURI	Columbia	-1	5206	4114	3167	2356	1696	1148
	Hannibal	-2	5613	4501	3534	2682	1985	1407
	Joplin	6	4321	3306	2450	1750	1190	762
	Kansas City	2	5283	4195	3261	2438	1791	1244
	Popular Bluff	11	4101	3107	2263	1580	1033	640
	Springfield	3	4660	3589	2691	2698	2000	1425
	St. Louis	2	4938	3873	2954	2178	1555	1052
MONTANA	Billings	-15	7212	5837	4616	3591	2719	1997
	Bozeman	-20	7997	6495	5161	3979	2986	2147
	Great Falls	-21	7766	6339	5126	4042	3141	2391
	Havre	-18	8660	7255	6012	4933	4025	3221
	Lewiston	-22	8613	7060	5672	4492	3467	2622
	Missoula	-13	7839	6332	4989	3809	2811	1986
NEBRASKA	Chadron	-8	7031	5713	4537	3524	2648	1910
	Grand Island	-8	6482	5271	4213	3295	2490	1837
	Lincoln	-5	6375	5208	4181	3285	2500	1850
	Omaha	-8	6194	5048	4047	3174	2421	1798
	Scottsbluff	-8	6702	5409	4244	3235	2375	1661

OUTSIDE DESIGN TEMPERATURES AND DEGREE DAYS TO SELECTED BASES
(Continued)

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
NEVADA	Carson City	4	5766	4393	3187	2184	1368	778
	Las Vegas	25	2532	1700	1038	540	238	71
	Reno	5	6030	4620	3392	2361	1513	883
	Winnemucca	-1	6409	5033	3797	2739	1846	1161
NEW HAMPSHIRE	Concord	-8	7482	6089	4867	3781	2840	2054
	Keene	-12	7035	5698	4526	3489	2594	1852
NEW JERSEY	Atlantic City	10	5086	3908	2882	2023	1338	815
	Newark	10	4972	3847	2851	2013	1352	841
	Trenton	11	4950	3809	2813	1978	1314	812
NEW MEXICO	Alamogordo	14	3059	2142	1389	811	408	163
	Albuquerque	12	4414	3327	2402	1616	1000	542
	Hobbs	13	2881	2004	1311	769	398	164
	Roswell	13	3126	2221	1475	891	478	218
	Tucumcari	8	3930	2922	2058	1368	827	449
NEW YORK	Albany	-6	6927	5626	4475	3462	2595	1871
	Buffalo	2	6798	5477	4330	3304	2425	1704
	Elmira	-4	6927	5570	4391	3351	2467	1730
	New York City	11	4868	4237	2752	1925	1277	785
	Oswego	1	6841	5499	4329	3292	2415	1682
	Rochester	1	6713	5413	4276	3266	2402	1688
	Syracuse	-3	6787	5472	4325	3311	2453	1742
	Utica	-12	7368	6006	4807	3763	2860	2093
NORTH CAROLINA	Asheville	10	4294	3146	2224	1472	923	547
	Charlotte	18	3342	2383	1606	1015	592	313
	Elizabeth City	12	3235	2307	1547	987	596	326
	Goldsboro	18	3102	2204	1471	917	539	280
	Greensboro	14	3874	2851	2000	1321	802	458
	Raleigh	16	3342	2396	1626	1043	620	333
NORTH DAKOTA	Bismarck	-23	9075	7664	6430	5313	4353	3516
	Fargo	-22	9343	7970	6748	5653	4688	3811
	Grand Forks	-26	9553	8168	6939	5842	4871	3993
	Minot	-24	9415	7988	6719	5588	4596	3723
OHIO	Akron	1	6241	4995	3899	2949	2148	1503
	Bowling Green	-2	6023	4834	3782	2860	2087	1465
	Cincinnati	1	4950	3856	2910	2111	1473	970
	Cleveland	1	6178	4925	3828	2871	2074	1430
	Columbus	0	5686	4500	3466	2582	1836	1256
	Dayton	-1	5255	4142	3166	2335	1652	1115
	Lima	-1	5910	4726	3676	2763	2002	1393
	Toledo	-3	6570	5315	4197	3217	2376	1678
	Youngstown	-1	6560	5266	4126	3141	2307	1622
OKLAHOMA	Ardmore	13	2609	1830	1219	739	416	201
	Bartlesville	6	3842	2908	2109	1471	945	569
	Muskogee	10	3409	2504	1764	1172	718	412
	Oklahoma City	9	3735	2802	2012	383	872	517
	Tulsa	8	3731	2813	2028	1408	901	548

OUTSIDE DESIGN TEMPERATURES AND DEGREE DAYS TO SELECTED BASES
(Continued)

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
OREGON	Baker	-1	7286	5777	4431	3268	2316	1561
	Eugene	17	4799	3358	2149	1224	612	255
	Grants Pass	20	4325	3048	1994	1156	570	223
	Portland	17	4691	3289	2134	1248	646	286
	Salem	18	4974	3521	2294	1330	668	281
PENNSYLVANIA	Allentown	4	5815	4603	3526	2597	1823	1206
	Erie	4	6768	5416	4261	3243	2364	1646
	Harrisburg	7	5335	4181	3168	2296	1584	1026
	Johnstown	-3	5768	4557	3499	2582	1823	1229
	New Castle	2	5885	4656	3577	2649	1885	1283
	Philadelphia	10	4947	3826	2847	2012	1352	847
	Pittsburgh	1	5950	4711	3631	2709	1934	1336
	Scranton	1	6330	5070	3955	2988	2145	1478
	West Chester	9	5370	4198	3165	2287	1562	999
	Williamsport	2	6047	4810	3714	2772	1968	1328
	York	8	5203	4052	3049	2194	1505	970
RHODE ISLAND	Providence	5	5908	4638	3525	2560	1776	1152
SOUTH CAROLINA	Charleston	25	1868	1189	703	382	171	64
	Columbia	20	2629	1803	1168	708	402	190
	Florence	22	2561	1750	1143	704	404	198
	Greenville	18	3239	2298	1531	953	551	280
	Sumter	22	2480	1680	1084	662	378	185
SOUTH DAKOTA	Aberdeen	-19	8570	7233	6052	4996	4069	3255
	Brookings	-17	8646	7277	6067	4997	4055	3210
	Huron	-18	8103	6795	5639	4611	3717	2927
	Pierre	-15	7571	6280	5145	4157	3290	2557
	Rapid City	-11	7301	5955	4751	3720	2842	2090
	Sioux Falls	-15	7885	6593	5447	4439	3546	2768
TENNESSEE	Bristol	9	4356	3273	2377	1637	1086	682
	Chattanooga	13	3583	2620	1812	1190	731	425
	Columbia	10	3761	2788	1962	1327	840	510
	Dyersburg	10	3559	2648	1883	1280	809	484
	Knoxville	13	3658	2689	1870	1245	779	467
	Memphis	13	3207	2324	1613	1051	636	350
	Nashville	9	3756	2799	1992	1365	876	539
	Tullahoma	8	3618	2657	1857	1245	786	472
TEXAS	Abilene	15	2621	1814	1187	695	371	168
	Amarillo	6	4231	3202	2321	1589	1015	592
	Austin	24	1760	1131	658	352	160	60
	Bay City	29	1339	792	428	237	98	38
	Corpus Christie	31	945	528	261	105	38	10
	Dallas	18	2407	1672	1086	648	351	161
	Houston	27	1549	957	538	268	111	39
	Laredo	32	926	521	260	102	39	10
	Lubbock	10	3516	2576	1788	1163	679	351
	Pecos	16	2512	1710	1074	593	280	98
	Plainview	8	3809	2825	1992	1318	789	428
	San Antonio	25	1606	999	555	275	103	32
	Vernon	13	2791	1974	1329	811	457	224
	Waco	21	2126	1431	889	508	258	105

OUTSIDE DESIGN TEMPERATURES AND DEGREE DAYS TO SELECTED BASES
(Continued)

STATE	CITY	OUTSIDE DESIGN TEMP.	DEGREE DAYS 65ºF	60ºF	55ºF	50ºF	45ºF	40ºF
UTAH	Logan	-3	6751	5419	4239	3223	2357	1647
	Ogden	1	5973	4711	3591	2649	1857	1238
	St. George	14	3253	2319	1532	930	483	215
	Salt Lake City	3	5802	4566	3465	2522	1731	1119
	Vernal	-5	7667	6288	5061	3988	3095	2350
VERMONT	Burlington	-12	7953	6559	5344	4256	3308	2496
	Rutland	-13	7155	5809	4625	3585	2687	1945
VIRGINIA	Charlottesville	14	4189	3145	2255	1545	984	586
	Danville	14	3856	2844	1991	1313	794	452
	Fredericksburg	10	4360	3301	2395	1655	1073	653
	Norfolk	20	3446	2477	1688	1087	655	361
	Richmond	14	3960	2940	2080	1395	867	509
	Roanoke	12	4315	3221	2321	1578	1019	623
	Staunton	12	5069	3897	2887	2044	1372	866
	Winchester	6	4823	3696	2732	1932	1305	819
WASHINGTON	Aberdeen	25	5320	3682	2311	1298	614	232
	Bellingham	10	5724	4110	2724	1657	907	448
	Kennewick	21	4845	3634	2606	1769	1146	718
	Olympia	16	5709	4125	2757	1661	855	373
	Seattle	21	5121	3615	2341	1354	691	297
	Walla Walla	0	5049	3797	2723	1847	1181	719
	Wenatchee	7	5696	4418	3300	2376	1634	1063
	Yakima	-2	6031	4662	3459	2470	1656	1067
WEST VIRGINIA	Beckley	-2	5577	4310	3229	2354	1635	1092
	Bluefield	-2	5217	3999	2969	2125	1457	948
	Charleston	7	4697	3583	2647	1852	1248	788
	Huntington	5	4676	3581	2664	1877	1277	827
	Morgantown	4	5348	4169	3152	2304	1614	1079
	Parkersburg	7	5107	3964	2987	2163	1506	996
	Wheeling	1	5450	4275	3244	2358	1645	1086
WISCONSIN	Appleton	-14	7728	6405	5250	4216	3295	2479
	Ashland	-21	9060	7571	6259	5119	4093	3194
	Eau Claire	-15	8463	7103	5916	4854	3930	3078
	Fond du Lac	-12	7568	6261	5124	4111	3211	2435
	Green Bay	-13	8143	6754	5543	4461	3522	2693
	La Crosse	-13	7540	6281	5161	4187	3305	2540
	Madison	-11	7642	6311	5150	4130	3223	2439
	Milwaukee	-8	7326	5983	4828	3789	2877	2117
	Racine	-6	6919	5616	4487	3488	2618	1900
	Sheboygan	-10	7232	5879	4699	3667	2757	1997
	Wausau	-16	8565	7162	5940	4848	3890	3029
WYOMING	Casper	-11	7642	6242	4970	3844	2889	2063
	Cheyenne	-9	7310	5875	4594	3466	2526	1727
	Cody	-19	7332	5908	4641	3566	2644	1880
	Lander	-16	7905	6506	5232	4108	3162	2334
	Rock Springs	-9	8356	6879	5541	4337	3286	2409

Design Temperatures are from ASHRAE “1985 FUNDAMENTALS HAND BOOK.” Degree Days Data is reprinted from the National Climatic Data Center Publication, “ANNUAL DEGREE DAYS TO SELECTED BASES.” (Data derived from the 1951-80 Normals.) Degree Days Data for other selected cities is available from Space-Ray.

B. *DETERMINING BUILDING HEAT LOSS WITH INFRARED AND OTHER HEATING SYSTEMS*

- Gas-Fired Infrared Heaters:
  Total Building Heat Loss x Infrared Compensation Factor based on mounting height (See Section IV, Item C) or 0.85

- Furnaces (Gas and Oil):
  Total Building Heat Loss x 1.25

- Electric Infrared Heaters:
  Total Building Heat Loss x 0.85

- Furnaces (Electric):
  Total Building Heat Loss x 1.00

- Unit Heaters (Gas and Oil):
  Total Building Heat Loss x 1.20

C. *FUEL CONVERSION FACTORS*

HEATER TYPE	CONVERSION FACTOR
Conventional Unit Heater	0.62
High Efficiency Unit Heater	0.80
Oil Fired Unit Heater	0.75
Electric Unit Heater	0.98
Central Furnace — Gas	0.70
Central Furnace — Oil	0.60
Boiler w/Steam Unit Heater	0.66
Boiler w/Hydronic Unit Heater	0.70
Air Rotation	0.85

D. ESTIMATION OF ELECTRICAL POWER CONSUMPTION

For a true operating cost analysis, do not fail to add the cost of electrical power consumption for the fan motors driving unit heaters or furnaces. The electrical cost can be determined by the following formula:

Electrical Cost = (EL x TOH x KWC) + KWD

Where:	EC	=	Total Electrical Cost ($)
	EL	=	Electrical Load (KW)
	TOH	=	Total Operating Hours (Hrs.)
	KWC	=	Electrical Cost per Unit (e.g., $.08/KWH)
	KWD	=	Demand Charge per Unit (consistent with total electrical load)

Total Operating Hours can be estimated by:

$$\mathbf{TOH = \frac{DD \times 24}{TD}}$$

Where:	DD	=	Degree Days based on 65ºF
	TD	=	Design Temperature Difference

Note: 1 HP = 1.341 KW*

*Based on average power and load factors, motor efficiency and initial starting amp draw.

E. COMPARATIVE FUEL COSTS

1 Therm	=	100,000 Btu/hr
	=	1 CCF (100 cubic feet) natural gas
	=	39.683 cubic feet propane gas
	=	1.095 gallons propane (liquid)
	=	4.638 lbs. propane (liquid)
	=	0.714 gallons fuel oil (No. 2)
	=	29.300 kilowatt hours electric

IX. SPOT AND AREA HEATING APPLICATION

One of the most interesting applications of Space-Ray infrared is providing heat to a specific area with little regard to size, type or construction of the building. This application even extends to outdoor use. Outdoor applications will require wind protection with some models, depending on the application.

Certain factors must be taken into account when considering the use of infrared for spot heating. Among them are certain empirical equations such as the Raber and Hutchinson comfort equation (John Wiley & Sons, 1947).

$$\mathbf{Ta + Tmrt = 140}$$

Where: Ta = Temperature of Ambient Air
Tmrt = Mean Radiant Temperature of Surrounding Surfaces

The above formula approximates the amount of radiant energy necessary for comfort when the subject is exposed to various temperatures. The formula is based on body heat production and radiation and convective losses in still air, and assumes the following:

a. Evaporative loss remains constant,
b. Normally clothed sedentary individual, and clothed temperature of 82.4ºF maintained.

The formula also takes into account, on a limited basis, the ratio of radiation loss to convective loss with respect to air speed.

The temperature of the ambient air, either inside or outside, can be readily determined, but the temperature of the surrounding surfaces is not so easily determined. Normally, if there is no heat input into an area, the surface temperature will be the same as the outside air and should be considered as such to provide a safety factor.

Usually when determining the radiant heat requirement for an area, the temperature of the ambient air is used as a guide; and, if the surrounding surface temperature is higher, this is considered as a gain. Radiant energy will be exchanged between two surfaces with unequal temperatures, as the fourth power of the absolute temperature.

Every person will generate heat in varying degrees according to his activity. His comfort will depend largely on the rate at which he loses this heat. If he loses it too quickly, he will shiver, and if too slowly, he will perspire.

The rate at which a person will generate heat is known as his rate of metabolism or metabolic rate, which will increase with his activity. It will be about 200 Btu/hr for a person sleeping, about 400 Btu/hr for a person seated at rest, and up to 1,500 Btu/hr or more for a person doing strenuous work.

The average person has a surface area of approximately 19.5 square feet; consequently, if he is working and producing 800 Btu/hr, he will be losing heat at the rate of 800 ÷ 19.5 = 41 Btu/hr per square foot. This amount of heat is considered as a heat gain and is subtracted from the total amount that must be radiated to the person to maintain comfort.

A person will lose heat by convection, radiation and evaporation. The amount that is lost by evaporation is, for all practical purposes, negligible.

The amount lost by radiation follows the basic radiation laws in that the amount lost is proportional to the fourth power of the absolute temperature difference. This absolute temperature is the temperature above absolute zero, which is 460 degrees below zero on the Fahrenheit scale. This amount will depend also on the relation between the surfaces, areas of each surface, configuration of each surface, attitude in respect to parallel surfaces, emissivity of each surface and other factors.

The figures used in the tables take into account various aspects of radiation losses and are the amounts that will be lost from a body temperature of 85ºF to the environment which is considered to have an average temperature and does not take into account temperature gradients or surrounding surfaces that vary in their surface temperature. The convection losses are computed as the losses from an exposed skin surface corresponding to the wind velocity and air temperature.

All of these factors are taken into consideration with an added safety factor in the charts provided, and the value given is the Btu/hr per square foot that must be provided. This is not the total input to the heaters, but is the net radiant output. This is computed as Total Radiant Input x Radiant Efficiency (approximately 62%) = Total Radiant Output.

Example:

Indoor application — standing, light assembly, 30ºF inside air temperature, air velocity 75 ft/min., 20 x 50 area to be heated.

By looking at the chart for Indoor Applications (B) for Standing, Light Assembly (Chart B-2) the figure given is 134 Btu/hr per square foot.

134 x (20 x 50) = 134,000 Btu/hr

NOTE: For spot heating applications, U-tube heaters provide more uniform coverage than straight tube heaters. If you choose to use straight tube heaters for spot heating applications, please note that straight tube heaters are always cooler at the exhaust end than at the burner end. There will be a noticeable temperature differential, particularly at low mounting heights.

A. OUTDOOR APPLICATIONS

1. Sitting (Patio, Sundeck, etc.)
   20ºF and up -- 5 to 15 MPH wind velocity

2. Entrances and Walkways
   0ºF and up -- 5 to 15 MPH wind velocity

3. Heavy Work (Loading Docks, Etc.)
   0ºF and up -- 5 to 15 MPH wind velocity

B. INDOOR APPLICATIONS

1. Sitting (Lounges, Desk Work)
   20ºF and up -- 50 feet per minute to 4 MPH wind velocity

2. Standing, Light Assembly (Bench Work)
   20ºF and up -- 50 feet per minute to 4 MPH wind velocity

3. Standing, Heavy Assembly (Machine Shop, Welding Shop, Auto Repair)
   20ºF and up -- 50 feet per minute to 4 MPH wind velocity

4. Walking, Heavy Work (Active Factory Work, Loading Docks)
   20ºF and up -- 50 feet per minute to 4 MPH wind velocity

A. OUTDOOR APPLICATIONS

CHART A-1

T* ºF	AIR VELOCITY 5 MPH	10 MPH	15 MPH
65	69	114	157
60	96	152	208
55	122	168	253
50	149	224	301
45	174	262	352
40	200	298	397
35	226	334	443
30	251	373	491
25	275	406	538
20	299	443	586

CHART A-2

T* ºF	AIR VELOCITY 5 MPH	10 MPH	15 MPH
45	115	203	293
40	141	238	338
35	166	275	384
30	192	314	432
25	216	347	478
20	240	384	526
15	264	416	570
10	288	453	616
5	314	490	664
0	336	522	709

CHART A-3

T* ºF	AIR VELOCITY 5 MPH	10 MPH	15 MPH
45	66	154	243
40	91	189	288
35	117	226	334
30	142	264	382
25	166	298	429
20	190	334	477
15	214	366	520
10	238	403	566
5	264	440	614
0	286	472	659

* Present temperature of area to be heated

B. INDOOR APPLICATIONS

CHART B-1

T* °F	AIR VELOCITY 50 Ft/Min	75 Ft/Min	1 MPH	2 MPH	3 MPH	4 MPH
65	26	27	29	38	46	56
60	42	46	46	58	69	80
55	58	61	64	77	88	102
50	72	78	80	96	112	126
45	83	88	91	107	126	144
40	106	112	115	134	154	174
35	120	126	131	152	174	195
30	138	144	149	173	195	221
25	152	158	163	190	218	245
20	168	174	181	206	237	264

CHART B-2

T* °F	AIR VELOCITY 50 Ft/Min	75 Ft/Min	1 MPH	2 MPH	3 MPH	4 MPH
65	16	18	19	29	37	46
60	32	37	37	48	59	70
55	48	51	54	67	78	94
50	62	69	70	86	102	117
45	74	78	82	98	117	138
40	96	102	106	125	144	165
35	110	117	122	142	165	186
30	128	134	139	163	186	211
25	142	149	154	181	208	235
20	157	165	171	198	227	254

CHART B-3

T* °F	AIR VELOCITY 50 Ft/Min	75 Ft/Min	1 MPH	2 MPH	3 MPH	4 MPH
65	0	0	0	10	18	27
60	13	18	18	29	40	51
55	29	32	35	48	59	74
50	43	50	51	67	83	98
45	54	59	62	78	98	115
40	77	83	86	106	125	146
35	91	98	102	123	146	166
30	109	115	120	144	166	192
25	123	130	134	162	189	216
20	138	146	152	178	208	235

CHART B-4

T* °F	AIR VELOCITY 50 Ft/Min	75 Ft/Min	1 MPH	2 MPH	3 MPH	4 MPH
65	0	0	0	0	10	19
60	2	10	10	21	32	43
55	21	24	27	40	51	66
50	35	42	43	59	75	90
45	46	51	54	70	90	107
40	69	75	78	98	117	138
35	83	90	94	115	138	158
30	101	107	112	136	158	184
25	115	122	126	154	181	208
20	130	138	144	170	200	227

***** Present temperature of area to be heated.

NOTE: For a typical indoor spot-heating application, air velocity of 50-75 ft/min. generally is sufficient. Use a larger air velocity if greater air movement in the facility is prevalent.

X. UNIT SELECTION

The first steps in designing any gas-fired infrared heating application are to compute the building heat loss and then select the heater quantity, type and reflector option to do the best heating job.

Building conditions usually dictate the mounting height and spacing between rows of heaters. Large, high bay buildings with overhead cranes almost always require the larger units at higher mounting heights, primarily because of the spacing between rows.

Normally, 70 to 80 percent of the building heat loss is around the outside perimeter walls. The transmission loss will amount to 50 percent of the total and the infiltration air will add another 20 to 30 percent. When designing an infrared heating system, the heaters should be located to cover the areas where the greatest heat loss occurs. This simply means that the majority of the heaters should be placed either on or near the outside perimeter walls. They can either be angle mounted on the outside walls facing the center of the building or hanging horizontally (facing down) at a distance from the outside wall approximately equal to the mounting height. You also need to be careful not to locate the heaters so they radiate too much heat to the outside walls. This will only increase the transmission heat loss.

The spacing between rows of heaters should not exceed three times the heater mounting height for uninsulated or drafty buildings. This is also true for all buildings where low inside design temperatures (50º-55º) are specified. This spacing can be exceeded under certain conditions but it would generally be advisable to install another row of heaters down the center of the building even if this means installing large units above an overhead crane.

Warehouses can be handled somewhat differently since the prime concern usually is to heat just the aisles. With material stacking on both sides, the radiation from overhead heaters is contained in the aisles and the remainder of the warehouse is heated by conduction and convection. With the high mounting heights and relatively low heat loss required for most warehouses, it is often necessary to space infrared heaters as much as 70 feet apart.

The recommended spacing between rows of heaters in a tight, well-insulated building with a 60º to 70º inside design temperature should not exceed four times the heater mounting height for reasonably uniform comfort heating.