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# Residential Energy Recovery Radiant Heat System

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# RESIDENTAL ENERGY RECOVERY RADIANT HEAT SYSTEM

By

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RESIDENTAL FURNACE ENERGY RECOVERY RADIANT HEAT SYSTEM

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Energy recovery systems aim to recover waste energy that is normally lost to the environment. Waste energy comes in a variety of forms. Many processes involving building HVAC involves transferring energy from one working fluid to another. Governing codes put strict requirements on ventilation requirements for various occupancy types. Increased ventilation demands more from the HVAC equipment since outdoor air (ventilation air) is being brought into the building and requires tempering in order to retain occupant comfort. The energy required to temper this air increases as ventilation rates increase. Many commercial buildings have implemented exhaust air energy recovery, or runaround loops in order to re-capture the energy in the exhaust air streams.

Residential HVAC systems are no different than commercial systems, just less complex typically. There is little being done to recover energy lost in the residential sector. The commercial industry is much further along in energy recovery systems than the residential sector. This is partially due to the lack of interest by home owners. Residential buildings consumed 35% of the total U.S. natural gas consumption in 2008 accounting for more than $90 billion dollars annually. Typical residential furnaces are 80% thermally efficient. The wasted energy is largely found in the combustion products,

or flue gasses. By recovering heat from the flue gases, this energy can be reclaimed and put to use.

Having a source for energy recovery is only part of the predicament. A desirable application of the recovered energy is also necessary. Home owners need to see a quick payback as well as an incentive in order to be enticed to make capital investments. Without government aid or tax incentives many individuals lose interest.

This thesis researches using energy recovery from residential gas furnaces to apply reclaimed heat to a radiant driveway heating system. The incentive for this application is the not only a maintenance free driveway in the winter, but reduced costs for snow removal.

# Author's Acknowledgements

I want to thank Dr. Haorong Li, Dr. Yong Cho, and Daihong Yu for their advice and guidance through the thesis process. Special thanks go out to Alta Sharp for funding the experimental model that was built to test this theory.

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# Nomenclature

$\Delta T$ – Change in temperature

AFUE – Annual Fuel Utilization Efficiency

ASHRAE – American Society of Heating, Refrigeration, and Air-Conditioning Engineers

BMS – Building Management System

CFM – Cubic Feet per Minute

GPM – Gallon per minute

IECC – International Energy Conservation Code

IMC – International Mechanical Building Code

IPC – International Plumbing Code

MBH – Thousand Btu's per hour

NFPA – National Fire Protection Association

PEX – Cross linked Polyethylene

SEER – Seasonal Energy Efficiency Ratio

UL Listing – A product marking that signifies that a UL representative inspected the product and it met UL's safety standards.

UMC – Uniform Mechanical Building Code

VFD – Variable Frequency Drive

# CHAPTER 1: Introduction

## 1.1. Problem Statement

Residential energy conservation has been studied in the past in regards to reducing consumption. One study suggests replacing appliances with higher efficiency units (Chiras). This is the most common solution to decreasing energy consumption on the residential level. Another study suggests saving energy through the brushless direct current motors in residential furnaces (Kendall, 2004). Previous research offered modifications or replacement to the existing equipment with higher efficient parts or systems. Energy recovery from combustion flue gasses has been studied on large industrial plants, such as the research performed on Karlskoga in central Sweden (Teppler, Wood, & Buzzell, 2008). In the study the concentration was on treating recovered flue gas condensate to be reused as boiler makeup water. Recovered heat was utilized for space heating. Research has been performed to provide cleaner flue gas energy recovery on large goal fired boilers (Zhelev & Semkov, 2004). Latent energy recovery from the condensation of the flue gases has been analyzed as well (Osakabe). The vast majority of the research available to review was applied to large scale equipment in comparison to a residential furnace. Given this information, it was understood that research of the recovery process and application was necessary. Whether a mid-efficiency natural gas furnace had enough waste energy to supply a radiant system was unknown. Previous research tested recovery and various applications for the recovered energy but left an uncertainty as to what applications were viable options.

Specifically, an energy recovery system centered on a radiant heating application was uncertain.

## 1.2. Research Objectives

The main objective of this study was explore the feasibility of a flue gas energy recovery system on a residential furnace to supply waste energy to a radiant slab snow melt system. To achieve the desired information calculations to test the feasibility of the radiant heat application were necessary before a full-scale model was constructed to insure it was not a wasted effort. Data measurement and analysis of the full-scale system were then conducted to compare analytical results to insure accuracy. The sections to follow explain the experimental model that was built, data results, system analysis with final conclusions and recommendations.

# CHAPTER 2: Literature Review

## 2.1. Residential Energy Conservation

Residential energy conservation typically goes hand in hand with the building envelope (windows, doors etc.). It is true that when the envelope of the building is increased in thermal resistance (R-value) less heat transfer is allowed between the interior of the home to the exterior. This will directly reduce energy consumption for heating and cooling the residence. The envelope indirectly reduces energy consumption of household appliances, such as the furnace. Dishwashers, furnaces and water heaters directly impact the consumption of utilites in the residence. Many times when an appliance is considered to be inefficient the first recommendation is to replace that unit with a higher efficiency piece of equipment (Chiras). Replacement of appliances are costly and take years to recover the cost of the appliance (Fluger, Dumont, & MacDermott, 2005). This particular study suggests that the payback from replacing low efficiency equipment is better than the payback of a mutual fund. While replacement of equipment does increase energy conservation in a residence, it does not implement true energy recovery. Residential energy recovery options are limited. Commercial applications of energy recovery are mostly applied to outdoor air ventilation. The commercial building code requires ventilation air be brought into the building depending on the number of people and application of the building. The residential building code does not require dedicated outdoor air be brought into the building to the central HVAC equipment as the commercial building code does. Outdoor ventilation air is assumed to be brought in through operable windows and doors. Additionally, the commercial building codes

require energy recovery be applied during designs and renovations. This is not true of the residential building code.

Other applications of energy recovery in the commercial industry are typically applied to large equipment that are not necessary in a residence. Chillers, water towers, ice storage and the like have no logical application for a residence. It is possible for a residence to have this equipment, but is cost effective or logical.

Retrofitting equipment is a possibility for increased energy utilization. One such study modifies a mid-efficiency natural gas fired furnace with a second condensing heat exchanger (Che, Liu, & Gao, 2004). While this study achieved increased efficiencies, the improvement resulted into a reduced utility bill. The research contained within this thesis applies waste energy to a desired application of the home owner resulting in benefits and a desired application by home owners. While home owners can be excited about reduced utility bills, the application of the recovered waste energy does not indirectly save the home owner money. The increase in furnace efficiency reduces furnace firing time which results directly in a reduced gas bill. Payback on the modified furnace varied depending on the material of the second heat exchanger. Additionally, the secondary heat exchanger that yielded the shortest payback, was also corrosion sensitive and would require replacement after an extended period of time. The energy recovery system and application of the reclaimed energy shall have a long maintenance-free life. In addition, it is desired to have an application for the residential energy recovery system shall indirectly benefit the home owner with a favorable payback. Some benefits outside of decreased utility costs are desired.

## 2.2. Flue Gas Energy Recovery

The combustion process of fuels releases a substantial amount of energy. While a majority of this energy is utilized for the intended purpose, most of the time a large amount of energy is lost during transmission of the energy. Heat transfer occurs anytime there are two mediums at a different energy state thermally. To try to harness a combustion process to transfer 100% of the released energy is very hard to do, if not impossible. Research has been completed on recovering energy from large scale combustion processes. Mainly the application has been for large industrial plants. Applications of the recovered energy have ranged from space heating to treating condensate to use as boiler makeup water (Teppler, Wood, & Buzzell, 2008). One study utilized flue gasses from a solid waste incinerator to preheat air (Suvarnakuta, Patumsawad, & Kerdsuwan, 2010). As previously mentioned, pretreating ventilation air isn't necessary to residences. Re-treating heating supply air to the residence is possible, but would be comparable to increasing the efficiency of the furnace, as described above.

Recovering condensate and heat from the flue gasses has been studied on large industrial plants (Teppler, Wood, & Buzzell, 2008) as well. Of course, heating boiler make-up water is not applicable in a residence, but recovery of the condensate is possible. The main drawback to recovering this additional energy is the treatment of the flue gasses. Corrosive flue gasses require treatment equipment and chemicals to provide a fluid that is usable by a residence. Use for the treated condensate could be utilized for domestic hot water heating or water supply for an irrigation system. These types of applications require constant supply of chemicals that would not be desirable by a home owner or future home owners for that matter. Costs for reverse osmosis equipment would also

drive home owners away. Recovered storm water would be an easier application for residences to use as irrigation water. Recovery of this water is easy and doesn't require costly chemicals.

While flue gasses are present for any combustion process or piece of equipment, testing of recovery on a small scale residential furnace is uncertain. Secondly, applying this recovered energy to a radiant heating system outdoors for snow melting purposes has limited resources. Whether a typical residential furnace has enough waste energy in the flue gasses to successfully provide enough heat for a radiant heated driveway is also unknown. This application provides a benefit to the home owner financially that is indirect of utility bills. Other benefits are also realized from not having maintenance on snow removal equipment or investing the time required to do such maintenance.

## 2.3. Summary

Previous research has shown a potential for savings from flue gas energy recovery. Energy recovery has been applied to large scale equipment in the commercial or industrial building industry. While the researched concepts apply to smaller combustion equipment, the application of a radiant snow melt system in combination with a standard residential natural gas fired furnace remains unknown. Whether the system would function properly, payback, be provide savings to a home owner is unknown.

# CHAPTER 3: Development of Residential Energy Recovery System and Application

## 3.1. Introduction

The goal of this thesis is to explore energy recovery options on a residential level that will pay back within 12 years. An important factor to consider is to find an energy recovery option that is attractive to home owners. Capital investments for residences are harder to justify as home owners typically do not see a profit from the operation of the building, such is the case with a business or commercial building. In 2007 it was reported that $177.4 billion dollars were invested into residential improvements and repairs, an equivalent of $1,500 per residence[1]. This data is shown below in ***Table* 1**. Declining home values during the recession corresponded with a $63 billion decrease in home improvement spending between 2007 and 2009 (D&R International, LTD., 2011). Even elevated data from 2007, the average expenditure is not enough money to have a professional contractor replace a gas fired furnace.

[1] Based on 2011 General Housing Characteristics from the U.S. Census Bureau for total number of households.

*Buildings Energy Data Book: 2.6 Residential Home Improvement* *March 2011*

**2.6.1 Value of Residential Building Improvements and Repairs, by Sector ($2009 Billion) (1)**

| | | Improvements | Maintenance and Repairs | Total |
|---|---|---|---|---|
| 1980 | | 71.6 | 34.9 | 106.5 |
| 1985 | | 81.6 | 64.8 | 146.4 |
| 1990 | | 90.7 | 84.8 | 175.5 |
| 1995 | | 104.9 | 63.3 | 168.2 |
| 2000 | | 137.1 | 52.3 | 189.4 |
| 2003 | | 154.9 | 51.4 | 206.4 |
| 2004 | | 167.8 | 57.4 | 225.2 |
| 2005 | | 177.5 | 59.2 | 236.7 |
| 2006 | | 185.8 | 56.8 | 242.6 |
| 2007 | (2) | 177.4 | 56.6 | 233.9 |

Note(s): 1) Improvements includes additions, alterations, reconstruction, and major replacements. Repairs include maintenance. 2) The US Census Bureau discontinued the Survey of Residential Alterations and Repairs (SORAR) after 2007.

Source(s): DOC, Historic Expenditures for Residential Properties by Property Type: Quarterly 1962-2003 (Old structural purposes) for 1980-2000; DOC, 1997 Census of Construction Industries: Industry Summary, Jan. 2000, Table 7, p. 15; DOC, Historic Expenditures for residential Proerties by Property Type: Quarterly 2003-2007 (New structural purposes) for 2003-2007; and EIA, Annual Energy Review 2009, August 2010, Appendix D, p. 383 for GDP and price deflators.

**Table 1: Value of Residential Building Improvements and Repairs, By Sector (D&R International, LTD., 2011)**

Improvements and repairs consist of items such as replacing furnace filters, housing additions or renovations, and replacement of failed equipment. Monthly savings in a residence are easily noticeable on utility statements. Primary energy consumption in the residential sector totaled 21.54 quadrillion Btu (quads) in 2008, equal to nearly 54% of consumption in the buildings sector and 22% of the total primary energy consumption in the United States. Nearly half (47%) of this primary energy was lost during transmission and distribution (D&R International, LTD., 2011). These statistics warranted further investigation to see what can be done to recover some of the wasted energy.

Finding a reason, or incentive, for residences to make capital improvements is important. Most home owners don't live in the same house for 30+ years. Paybacks in this range are not very enticing and will not be given much thought. Without tax rebates or government funding to help pay for upgrades, there needs to be an incentive with the product in order to gain interest from home owners. This will be discussed further in the **Economic Analysis** section.

## 3.2. Commercial Energy Recovery Systems

Energy recovery has become a major part of the commercial building industry. As the world focuses on being more 'green' with our energy consumption there has been large pushes to implement saving techniques to conserve energy and lower our carbon footprint. A carbon footprint is a measure of the impact our activities have on the environment, and in particular climate change. It relates to the amount of greenhouse gases produced in our day-to-day lives through burning fossil fuels for electricity, heating and transportation etc. (Carbon Footprint - What Is A Carbon Footprint?). Having an increased awareness of our impact on the earth has led to building codes integrating energy recovery into building designs. The International Energy Code Council (IECC), among other organizations, has implemented an energy code that has been adopted around the United States for commercial buildings. For example, a commercial buildings being designed to the IECC 2009 standard is required to have ventilation energy recovery if the following is true of the designed system;

- Individual fan systems that have both a design supply air capacity of 5,000 cfm or greater and a minimum outside air supply of 70 percent or greater of the design supply air quantity shall have an energy recovery system that provides a change in the enthalpy of the outdoor air supply of 50 percent or more of the difference between the outdoor air and return air at design conditions. Provision shall be made to bypass or control the energy recovery system to permit cooling with outdoor air where cooling with outdoor air is required (International Code Council, Inc., 2009).

This section of the code requires that 50% of the difference in energy between the design return air condition and outdoor air condition be recovered and applied to the unconditioned outdoor air. Outdoor air is required by the governing building codes to provide ventilation to building occupants. Outdoor air is used to dilute and remove indoor air contaminants. The energy required to condition the outdoor air can be a significant portion of the total space-conditioning load (2009 ASHRAE Fundamentals, 2009). There are many options for energy recovery. They include the following;

- Air-to-air heat exchangers with metal walls.
- Air-to-air heat exchangers with permeable walls.
- Heat wheels.
- Enthalpy wheels.
- Heat pipes.
- Runaround coils. (Shah, 2011)

The recovery options above will help transfer energy to/from incoming outdoor air being supplied to building ventilation equipment. In the heating season when outdoor air is dry and cold, the exhaust air will transfer heat to the outdoor air thus reducing the heating load on the ventilation equipment. In the summer when the outdoor air is humid and hot, the outdoor air will transfer heat to the exhaust air before it is exhausted, thus cooling the outdoor air. Depending on the method of energy recovery utilized, latent energy can also be recovered or transferred. As energy recovery methods and technologies are perfected the cost of implementation will decrease. When it becomes cost effective for residences to add these types of systems it is suspected that the building code requirements will filter down to residential applications.

In the consulting industry, a good HVAC system will be designed to create a slightly positive pressure inside the building. In this situation, the HVAC equipment will bring in more air than is being exhausting, creating a net positive building pressure. Comfort is better controlled if tempered air is lost to the outdoors rather than have untreated air enter into the building through cracks in the building. Air entering through cracks and crevasses is known as infiltration. To maintain a positive building pressure the exhaust rate needs to be exceeded by the rate of incoming outdoor air. This outdoor air must be tempered to maintain comfort to occupants. Treating outdoor air is costly since it typically requires a substantial amount of energy. Using the energy recovery techniques mentioned above will help reduce the load on HVAC equipment.

Sensible energy transfer is easily transferred using a runaround coil. This configuration is shown below in ***Figure 1***. As the figure shows, a coil is inserted into each air stream and a transfer fluid, typically a water glycol mixture, is pumped through the coils to absorb energy from one airstream and transfers it to the other. This will recover sensible energy only since moisture is not transferred from one air stream to the other.

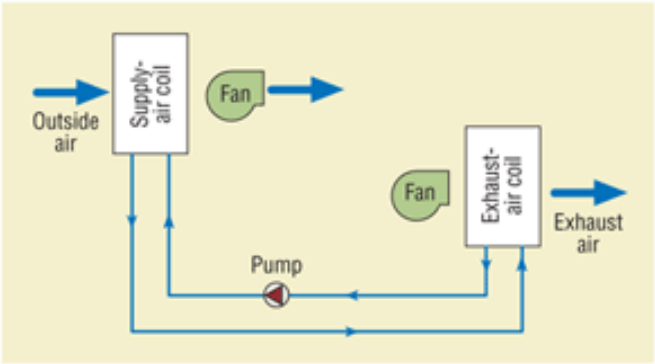
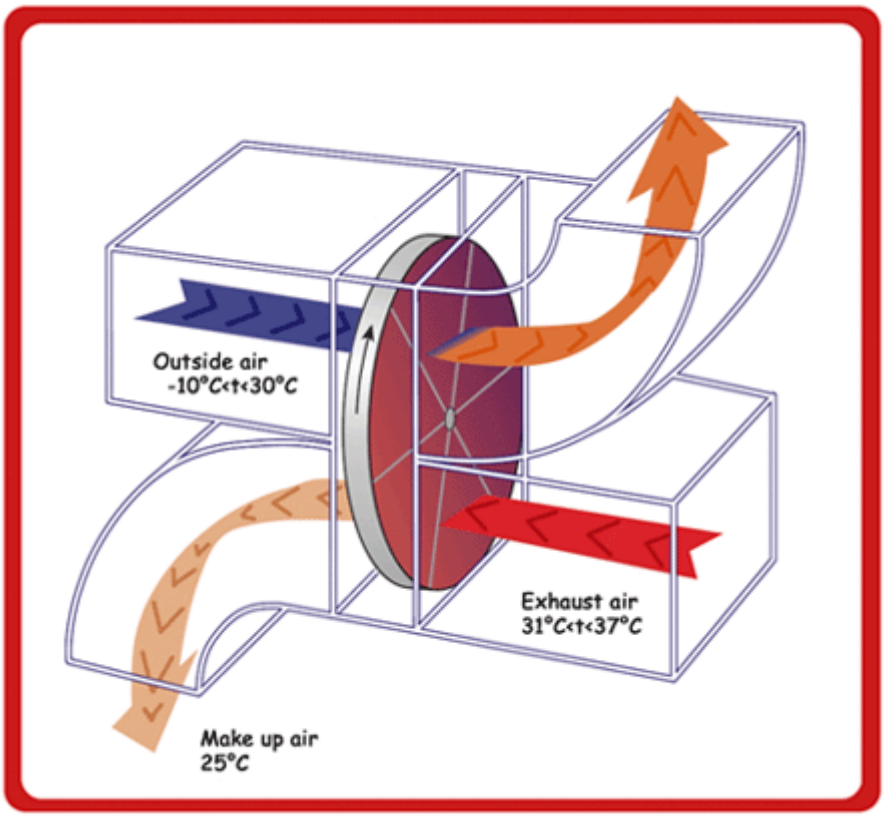



**Figure 1: Run Around Coil Schematic (Shah, 2011)**

Runaround coils have a unique feature of offering complete flexibility in terms of the location of supply- and exhaust-air streams; all of the other devices (listed above) require the air streams to be close together. Another advantage of runaround coils is the impossibility of cross-contamination between supply air and return air; this is a risk with all of the other devices listed above (Shah, 2011).

Latent energy recovery is possible, but requires a different recovery system than a runaround coil. Of course, latent energy can be recovered from condensing the flue gasses; however a runaround coil does not transfer moisture from one airstream to the other since it utilizes a heat transfer solution. Enthalpy wheels, or rotary heat exchangers, transfer sensible or sensible and latent energy between the exhaust air and the incoming outside air. Both sensible-only wheels and total energy wheels, sometimes referred to as desiccant wheels, are available (VanGeet, 2006). Desiccant wheels absorb energy, including moisture, and transfers it to the other air stream. ***Figure 2*** below shows how the total energy wheel operates. Careful consideration needs to be taken to be sure that the wheel transfers only moisture and not airborne contaminants (VanGeet, 2006).




A diagram of a rotary heat exchanger, or "heat wheel" (From Uptime Technology BV)

**Figure 2: Enthalpy Wheel Operation (Miller, 2008)**

Energy recovery is a growing concept and continues to expand in the commercial building marketplace.

## 3.3. Residential Energy Consumption

According to the *2010 Buildings Energy Data Book,* residences consumed 54.5% of the primary energy consumed by buildings in 2010 (D&R International, LTD., 2011). This percentage equates to 22 quadrillion Btu's of energy. This is equivalent to 792,000,000 tons of coal. This shows an increasing trend from the 2008 levels. With dwindling supplies of natural resources, a large potential for reducing energy consumption is found in the residential sector. ***Figure 3*** below shows the energy

consumption by the end use for a typical residence. As shown, space heating and water heating are the top two energy consumers in a residence. Space heating and cooling – which combined account for 54% of site energy consumption and 42% of primary energy consumption – drive residential energy demand (D&R International, LTD., 2011).

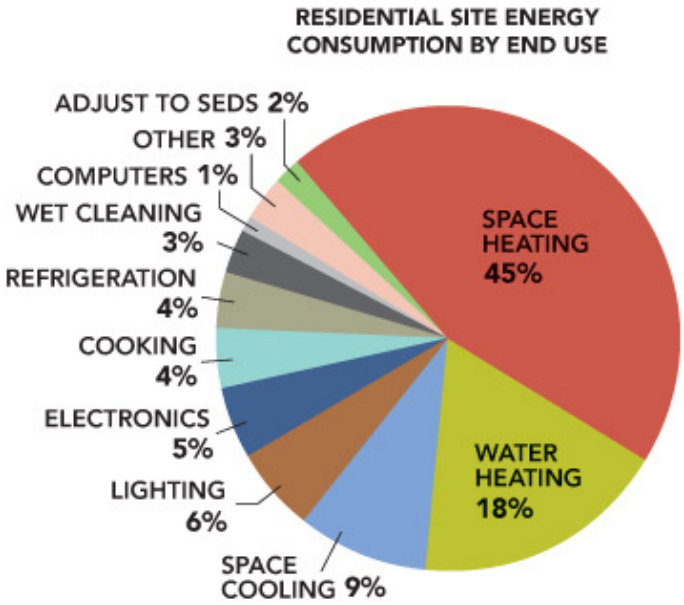



**Figure 3: Residential Site Energy Consumption by End Use (D&R International, LTD., 2011)**

Data on a typical single-family home is shown below in ***Table 2***. This data is an average of all residences in the United States. As shown the preferred method of space heating, on average, is through a central furnace system. The majority of new residences have central air handling system installed for heating as ***Table 3*** shows. The percentage of furnaces installed has dropped in recent years as the heat pump has been introduced. Heat pumps are versatile, but do not operate at extreme temperatures.

*Buildings Energy Data Book: 2.2 Residential Sector Characteristics* *March 2011*

**2.2.7 Characteristics of a Typical Single-Family Home (1)**

| | | | | | |
|---|---|---|---|---|---|
| Year Built | mid 1970s | Building Equipment | Type | Fuel | Age (5) |
| Occupants | 3 | Space Heating | Central Warm-Air Furnace | Natural Gas | 12 |
| Floorspace | | Water Heating | 49 Gallons | Natural Gas | 8 |
| Heated Floorspace (SF) | 1,934 | Space Cooling | Central Air Conditioner | | 8 |
| Cooled Floorspace (SF) | 1,495 | | | | |
| Garage | 2-Car | | | | |
| Stories | 1 | Appliances | Type / Fuel / Number | Size | Age (5) |
| Foundation | Concrete Slab | Refrigerator | 2-Door Top and Bottom | 19 Cubic Feet | 8 |
| Total Rooms (2) | 6 | Clothes Dryer | Electric | | |
| Bedrooms | 3 | Clothes Washer | Top-Loading | | |
| Other Rooms | 3 | Range/Oven | Electric | | |
| Full Bathroom | 2 | Microwave Oven | | | |
| Half Bathroom | 0 | Dishwasher | | | |
| Windows | | Color Televisions | 3 | | |
| Area (3) | 222 | Ceiling Fans | 3 | | |
| Number (4) | 15 | Computer | 2 | | |
| Type | Double-Pane | Printer | | | |
| Insulation: Well or Adequate | | | | | |

Note(s): 1) This is a weighted-average house that has combined characteristics of the Nation's stock homes. Although the population of homes with similar traits may be few, these are likely to be the most common. 2) Excludes bathrooms. 3) 11.5% of floorspace. 4) Based on a nominal 3' X 5' window. 5) Years.

Source(s) EIA, 2005 Residential Energy Consumption Survey: Characteristics, April 2008, Tables HC 1.1.1, HC1.1.3, HC 2.1, HC 2.2, HC 2.3,

**Table 2: Characteristics of a Typical-Single-Family Home (D&R International, LTD., 2011)**

*Buildings Energy Data Book: 2.2 Residential Sector Characteristics* *March 2011*

**2.2.8 Presence of Air-Conditioning and Type of Heating System in New Single-Family Homes**

| | | Type of Primary Heating System | | | | |
|---|---|---|---|---|---|---|
| Year | Total Homes (thousands) | Warm-Air furnace | Heat pump | Hot Water or steam (1) | Other or none (2) | Air-Conditioning |
| 1980 | 957 | 57% | 24% | 4% | 15% | 62% |
| 1985 | 1,072 | 54% | 30% | 5% | 11% | 70% |
| 1995 | 1,066 | 66% | 25% | 5% | 4% | 79% |
| 2000 | 1,242 | 71% | 23% | 4% | 2% | 85% |
| 2005 | 1,636 | 67% | 29% | 3% | 1% | 89% |
| 2006 | 1,654 | 63% | 33% | 3% | 2% | 89% |
| 2007 | 1,218 | 62% | 34% | 2% | 2% | 90% |
| 2008 | 819 | 60% | 34% | 3% | 3% | 89% |
| 2009 | 520 | 56% | 37% | 3% | 4% | 88% |

Note(s) 1) Includes both air source and geothermal (ground source) versions. 2) Includes electric baseboard, panel, radiant heat, space heater, floor or wall furnace, solar, and other types.

Source(s): DOC, 2009 Characteristics of New Housing, June 2010, "Type of Heating System Used in New Single-Family Houses Completed" and "Presence of Air-Conditioning in New Single-Family Houses Completed"

**Table 3: Presence of Air-Conditioning and Type of Heating System in New Single-Family Homes (D&R International, LTD., 2011)**

Heat pumps use an energy input (almost always electricity) to transfer heat from a cold medium (the outside air or ground in the winter) to a warmer medium (the warm air or hot water used to distribute heating in a building). During hot weather, the heat pump

can operate in reverse, thereby cooling the indoor space. In winter, drawing heat from a relatively warm source (such as the ground rather than the outside air) and distributing the heat at the lowest possible temperature can dramatically improve the heat pump efficiency (Levine, et al., 2007). A typical air cooled operation in both cooling and heating are shown below in ***Figure 4*** and ***Figure 5***, respectfully. Air cooled heat pumps are most commonly installed over ground-source heat pumps since the installation cost of the well-field is expensive. A ground source heat pump exchanges energy with the earth rather than the outside air through the use of buried piping.

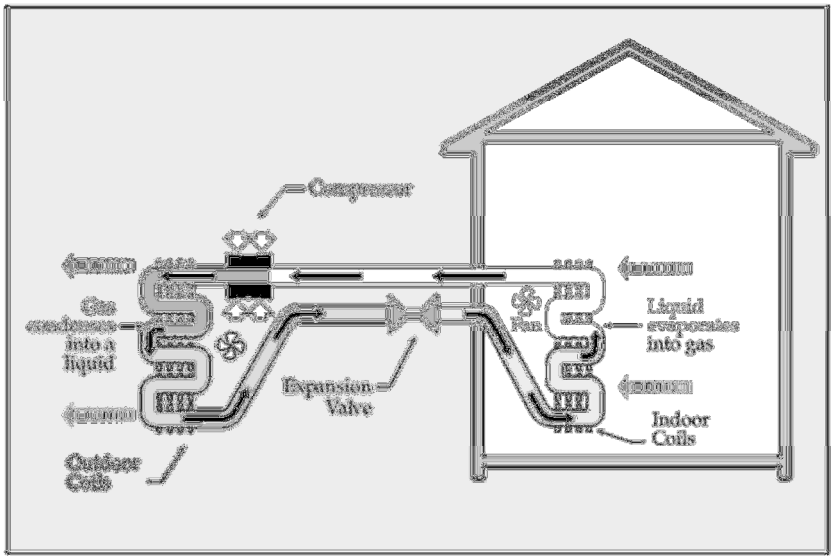



**Figure 4: Air Cooled Heat Pump in Cooling Mode**

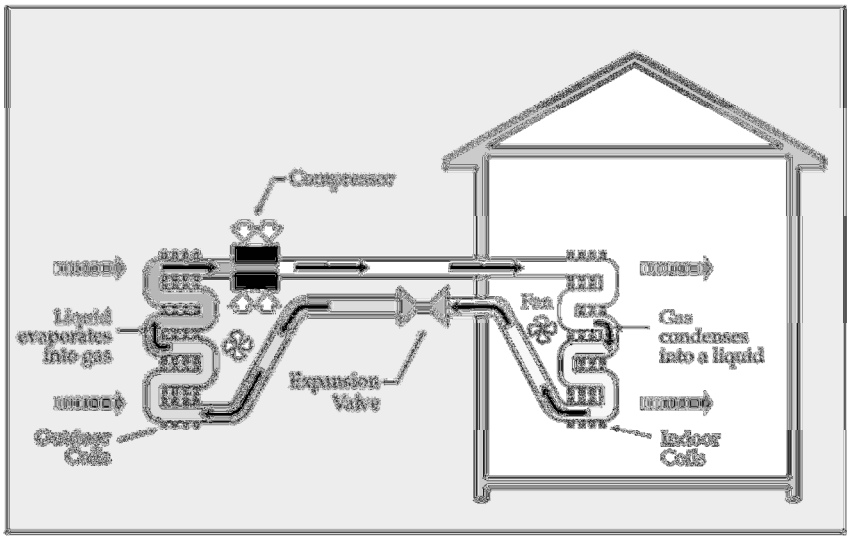



**Figure 5: Air Cooled Heat Pump in Heating Mode**

When outdoor temperatures fall below 40°F, a less-efficient panel of electric resistance coils, similar to those in your toaster, kicks in to provide indoor heating. This is why air-source heat pumps aren't always very efficient for heating in areas with cold winters. Fuel-burning furnaces generally can provide a more economical way to heat homes in cooler U.S. climates (National Renewable Energy Laboratory (NREL), 2001).

Gas fired domestic water heaters operate by passing flue gasses through a flue, which is surrounded by domestic water. ***Figure 6*** below shows a cut-away of a typical non-condensing gas-fired domestic water heater. Electric water heaters are available but are less commonly used when gas is available. As ***Table 2*** above listed, the average single-family residence has a natural-gas fired domestic water heater installed.

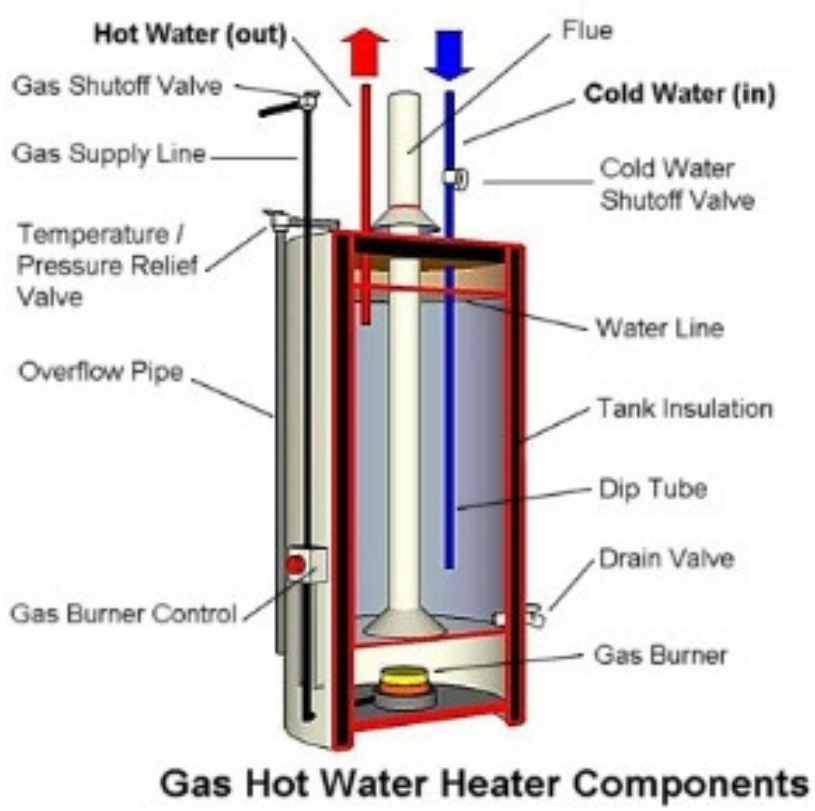



Figure 6: Gas Fired Domestic Water Heater Diagram (Guadalupe, 2010)

Since the furnace and water heater combine to consume over 60% of the energy consumed by a typical residence, energy recovery results are best applied to these appliances. This is due to the fact that the largest potential for savings can be realized where the most energy consumption lies.

## 3.4. Residential Indirect-Fired Gas Furnace Operation

Natural gas fired furnaces function to heat a supply airstream through the use of a heat exchanger. The heat exchanger keeps combustion gasses separate from the air supplied to the residence. This is why it is referred to as an 'indirect-fired' gas furnace. Combustion gas has harmful substances in it that are caustic to breath, so separation of the airstreams is crucial. Natural gas fired residential furnaces are typically an indirect

fired unit. There are applications that use direct-fired furnaces, but these types of furnaces are rarely (if ever) used in a residential application for central air. There are many variations of natural gas indirect-fired furnaces. ***Figure 7*** below is an example of a category I indirect-fired gas furnace. The flue from this furnace is under negative pressure as it uses the stack effect to exhaust combustion gasses to the outdoors. The negative pressure of the flue insures that combustion gasses don't escape into the residence. There are four vent categories which are classified depending on the furnace components and how the heat is transferred to the supply air stream. These vent categories are listed in the NFPA standards (National Fire Protection Association, 2002).

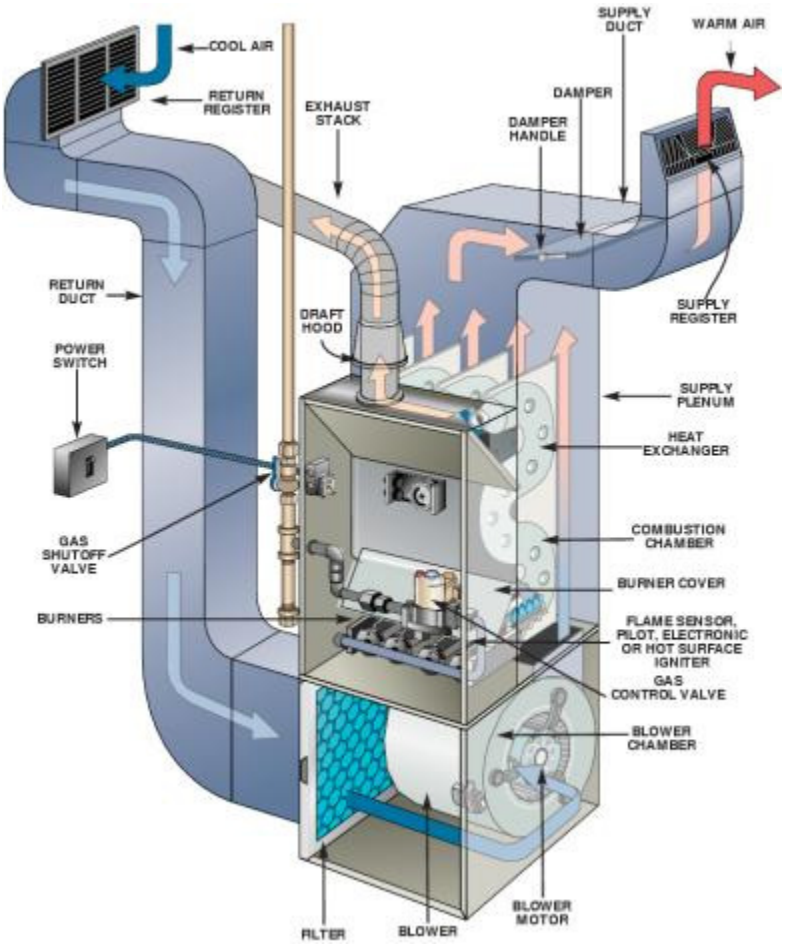



**Figure 7: Indirect Fired Gas Furnace Operation (1st Choice Home Inspection Service, 2012)**

- Category I – An appliance that operates with a non-positive vent static pressure and with a vent gas temperature that avoids excessive condensate production in the vent. This furnace is commonly referred to as a 'natural draft furnace.' These units were manufactured before 1992.
- Category II – An appliance that operates with a non-positive vent static pressure and with a vent gas temperature that may cause excessive condensate production in the vent. This is a variation of the natural draft furnace.
- Category III – An appliance that operates with a positive vent static pressure and with a vent gas temperature that avoids excessive condensate production in the vent. This furnace is commonly referred to as an 'induced-draft non-condensing furnace.'
- Category IV – An appliance that operates with a positive vent static pressure and with a vent gas temperature that may cause excessive condensate production in the vent. This furnace is commonly referred to as an 'induced-draft condensing furnace.'

Flue gas is very corrosive by nature. This is due to the fact that most conventional fuels contain small amounts of sulfur, which is oxidized to sulfur dioxide ($SO_2$) or sulfur trioxide ($SO_3$) during combustion, and the noncombustible substances such as mineral mater (ash) , water, and inert gas (2009 ASHRAE Fundamentals, 2009). When sulfur trioxide comes in the presence of water vapor, sulfuric acid forms. This is shown in ***Equation 1***.

$$SO_2(g) + O_2(g) \Leftrightarrow SO_3(g)$$

$$SO_3(g) + H_2O(l) \Leftrightarrow H_2SO_4(aq)$$

**Equation 1: Flue Gas Condensation Chemical Reaction**

Galvanized steel is commonly used for venting a non-condensing furnace (category I and III). When galvanized steel comes into contact with an acid, the zinc plating deteriorates from a chemical reaction with the sulfuric acid. After the zinc coating (sacrificial anode) is depleted, the exposed steel remains unprotected. The exposed steel will begin to corrode in the presence of an oxidizing agent, such as oxygen. For this reason, careful consideration needs to be taken in the selection of materials for natural gas fired furnaces. The material used for the heat exchanger is just as crucial.

Category III vents are used on non-condensing induced-draft furnaces. When this type of furnace operates it uses a fan to induce air into the heat exchanger where combustion occurs. Typical category III furnaces will use a horizontal venturi-type burner. Fuel exiting from the fuel gas orifice enters one end of the venturi as a jet. This induces the primary combustion air into the venturi where turbulent mixing occurs. About 60% of the air required for complete combustion (stoichiometric air) is drawn in by this process. An inner combustion cone sits at the outlet end of the venturi. The remaining or secondary combustion air is drawn into the flame zone at the outlet end of the venturi by the combined action of the induced draft from the fan-assist and the buoyancy of the flue gasses. Here the air mixes with the unburned fuel and products of combustion from the inner combustion zone. The combustion is controlled by diffusion and turbulent mixing, resulting in a secondary burn out region that stretches well into the

heat exchanger (Fleck, Arnold, Ackerman, Dale, Klaczek, & Wilson, 2009). ***Figure 8*** shows a single venturi-type burner. Furnaces have multiple burners installed for operation typically.

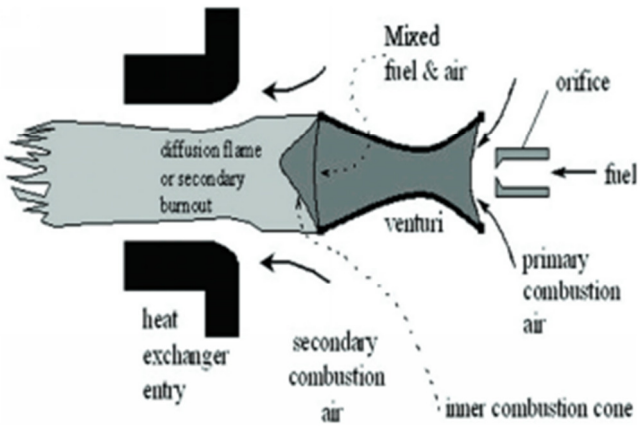



**Figure 8: Venturi-type Burner Schematic (Fleck, Arnold, Ackerman, Dale, Klaczek, & Wilson, 2009)**

A combustion process is considered complete if all the carbon in the fuel burns to $CO_2$, all the hydrogen burns to $H_2O$ and all the sulfur burns to $SO_2$ (Cengel & Boles, 2006). The three fundamental chemical reactions that occur during the combustion of a hydrocarbon fuel are shown below (2009 ASHRAE Fundamentals, 2009). Further discussion of the combustion process is found in the *Furnace Combustion Analysis* section.

$$C + O_2 \rightarrow CO_2$$

$$H_2 + 0.5O_2 \rightarrow H_2O$$

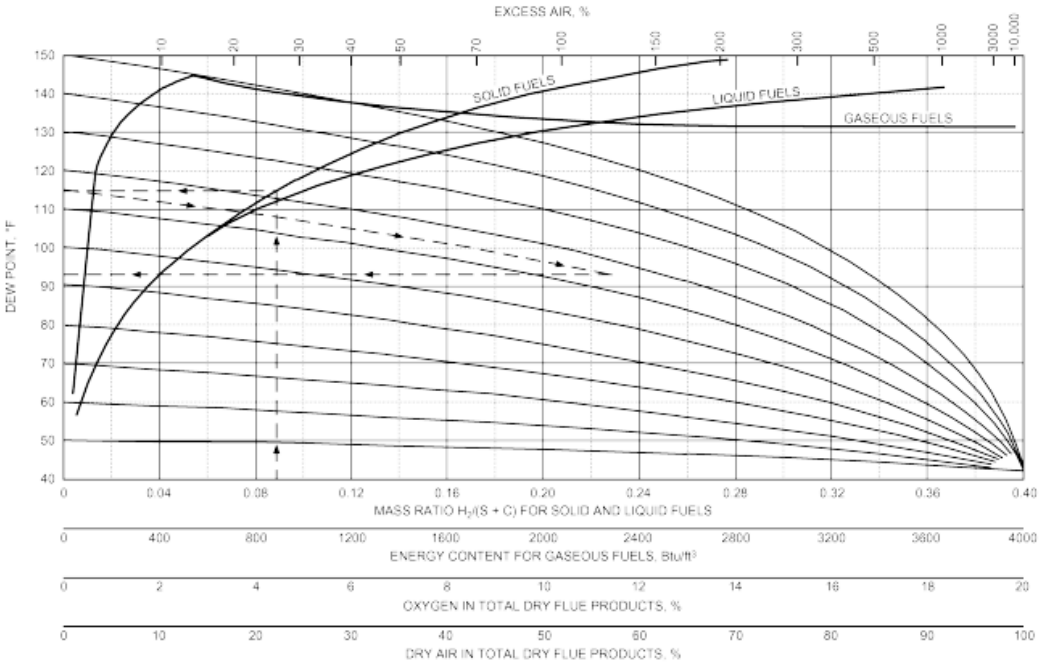
$$S + O_2 \rightarrow SO_2$$

**Equation 2: Hydrocarbon Combustion Process Fundamental Reactions**

Heat exchangers for induced-draft furnaces are made out of aluminized steel. Although aluminized steel has better corrosion properties than galvanized steel, it is still susceptible to oxidation (corrosion). The vent on this type of furnace (category III) can be galvanized steel since condensation is not likely in the flue piping.

A condensing gas furnace (high-efficiency) is much like an induced-draft furnace in that the flue is under positive pressure from the fan that aids in combustion. Burners are typically venturi-style as well. The main difference is that a condensing gas furnace has a series of heat exchangers. A second heat exchanger is located downstream of the standard aluminized steel exchanger. The second exchanger is typically made out of stainless steel or some material that is corrosion resistant. In the secondary heat exchanger, more energy is transferred from the combustion gasses to the point that condensation starts to precipitate out of the combustion gasses. In a condensing gas furnace, the most corrosive conditions exist at the leading edge of the condensing region, especially areas that experience evaporation during each cycle (Stickford, Talbert, Hindin, & Locklin, 1988). Draining condensate retards the concentration of acids on system surfaces; regions from which condensate partially or completely drains away before evaporation are less severely attacked than regions from which condensate does not drain before evaporation (2009 ASHRAE Fundamentals, 2009). As shown in ***Figure 7***, the dew point temperature of natural gas combustion fumes ranges from 130°F-

150°F depending on the amount of excess air applied to the combustion process. In general, any natural gas flue gas temperature below 130°F will start to condense.




**Theoretical Dew Points of Combustion Products of Industrial Fuels**
Adapted from *Gas Engineers Handbook* (1965). Printed with permission of Industrial Press and American Gas Association.

**Figure 9: Theoretical Dew Points of Combustion Products of Industrial Fuels (2009 ASHRAE Fundamentals, 2009)**

Efficiency of space heating systems for residential and commercial buildings are usually expressed in terms of the annual fuel utilization efficiency, or AFUE, which accounts for the combustion efficiency as well as other losses such as heat losses to unheated areas and start-up and cool-down losses (Cengel & Boles, 2006). Over the years, the AFUE ratings of gas-fired furnaces have increased. The minimum allowed AFUE rating for a non-condensing fossil-fueled, warm-air furnace is 78%; the minimum rating for a fossil-fueled boiler is 80%; and the minimum rating for a gas-fueled steam

boiler is 75% (U.S. Department of Energy, 2011). The following are typical AFUE ratings for different residential heating and ventilation equipment.

- Old, low-efficiency heating systems:
  - → Natural draft venting
  - → 68%-72% AFUE
- Mid-efficiency heating systems:
  - → Induced-draft combustion
  - → Positive pressure venting
  - → 80%-83% AFUE
- High-efficiency heating systems:
  - → Condensing flue gasses in a second heat exchanger for extra efficiency
  - → 90%-97% AFUE (U.S. Department of Energy, 2011)

The annual fuel utilization efficiency can be determined by following the testing procedure found in ASHRAE Standard 103 (Method of testing for annual fuel utilization).

## 3.5. Options for Residential Gas Fired Furnace Energy Recovery

Energy recovery from an indirect-fired gas furnace lies in the recovery of energy from flue gasses. High efficiency condensing gas furnaces take advantage of the extra energy in the flue gasses by having a secondary condensing heat exchanger, as previously discussed. While high efficiency furnaces take advantage of the excess energy, not many residences in the United States have high efficiency furnaces installed. As shown in ***Table 4***,

in 2006 over half of the gas-fired furnaces shipped in the United States were between 75%-88% efficient. The AFUE of most new heating systems is about 85%, although the AFUE of some old heating systems is under 60% (Cengel & Boles, 2006).

*Buildings Energy Data Book: 5.3 Heating, Cooling, and Ventilation Equipment* *March 2011*

**5.3.2 Residential Furnace Efficiencies (Percent of Units Shipped) (1)**

| Gas-Fired | | | | Oil-Fired | |
|---|---|---|---|---|---|
| AFUE Range | 1985 | AFUE Range | 2006 | AFUE Range | 1985 |
| Below 65% | 15% | 75% to 88% | 64% | Below 75% | 10% |
| 65% to 71% | 44% | 88% or More | 36% | 75% to 80% | 56% |
| 71% to 80% | 10% | Total | 100% | More Than 80% | 35% |
| 80% to 86% | 19% | | | Total | 100% |
| More than 86% | 12% | | | | |
| Total | 100% | | | | |

| | | | | | |
|---|---|---|---|---|---|
| Average shipped in 1985 (2): | 74% | AFUE | Average shipped in 1985 (2): | 79% | AFUE |
| Average shipped in 1995: | 84% | AFUE | Average shipped in 1995: | 81% | AFUE |
| Best Available in 1981: | 85% | AFUE | Best Available in 1981: | 85% | AFUE |
| Best Available in 2007: | 97% | AFUE | Best Available in 2007: | 95% | AFUE |

Note(s): 1) Federal appliance standards effective Jan. 1, 1992, require a minimum of 78% AFUE for furnaces. 3) Includes boilers.

Source(s): GAMA's Internet Home Page for 2006 AFUE ranges; GAMA News, Feb. 24, 1987, for 1985 AFUE ranges; LBNL for average shipped AFUE; GAMA, Consumer's Directory of Certified Efficiency Ratings, May 2004, p. 12 and 72-73 for 2004 best-available AFUEs; GAMA Consumer's Directory of Certified Efficiency Ratings for Heating and Water Heating Equipment, May 2007; GAMA Tax Credit Eligible Equipment: Gas- and Oil-Fired Furnaces 95% AFUE or Greater, May 2007; and GAMA AFUE press release 2006: U.S. shipments of gas warm-air central furnaces.

**Table 4: Residential Furnace Efficiencies (Percent of Units Shipped) (D&R International, LTD., 2011)**

A higher AFUE rating means that more energy is transferred to the supply air stream. For example, a 90% AFUE 100Mbh furnace transfers 90Mbh of useful energy to the supply air stream during the operation of the furnace. This leaves 10Mbh to be lost during the operation. A majority of these losses are in the flue gas as it exits the building. There is a potential for recovering all of this lost energy if the recovery method is efficient. Government regulations have placed a minimum of a 78% AFUE rating for gas-fired furnaces in residences, as previously mentioned. Depending on the efficiency of the furnace, a substantial amount of energy is being lost during the heating of a residence. Since a majority of waste energy is found in the

hot flue gasses, this is where the greatest possibility for energy recovery lies. The recovered energy from the flue gasses can be used for many purposes depending on the recovery method.

One option would be to heat the outdoor air (ventilation air) being brought into the home. However, not many homes have outdoor air introduced to the furnace directly since operable windows bring in ventilation air that meets the residential building code (IRC 2009). This will create a difficulty in treating the ventilation air since it is not coming into the residence from one location.

Another option would be to use the recovered energy to maintain water temperatures inside the hot water heater. This application would decrease gas or electrical costs to operate the hot water heater. Modification of the domestic water heater would be required to integrate a heat transfer coil inside the heater in order to transfer the energy. UL listed equipment would no longer be UL listed with these modifications, which creates a drawback to the proposed system.

The application chosen as the most attractive is to use the recovered waste energy to provide a radiant heated slab in the winter. The recovered energy will be used to heat the driveway and sidewalk concrete slabs to melt snow during winter storms in the Midwest. This offers an attractive benefit to homeowners as well as a payback that may fit inside the proposed timeline for a homeowner. Although a radiant heated driveway is more of a novelty

than a necessity, it provides many benefits and savings. This will be discussed further in the ***Economic Analysis*** section.

## 3.6. Proposed Energy Recovery System

As discussed, space heating accounts for nearly 45% of residential energy use (see ***Figure 3***). The opportunity for energy recovery in a residence is best applied to the largest source of energy consumption. For this reason an energy recovery system on a gas fired furnace offers great possibilities. Since most wasted energy from a residential furnace is in the flue gasses, energy recovery should be applied to the flue gasses. The installed energy recovery system shall not prohibit proper operation of the furnace. Flue gasses are poisonous to occupants of the home and need an unrestricted path to the outdoors. Careful consideration needs to be taken in the materials used to recover the energy from the flue gasses. Since condensing flue gasses is highly likely, special considerations need to be made as to what materials are selected for the recovery system. The recovered energy will be transferred to an intermediate heat transfer fluid, consisting of water and a glycol additive. A radiant heating system for a driveway will be the application for the recovered waste energy. A radiant heat system provides multiple benefits.

First is the reduction in physical work needed for snow removal. Snow removal creates an inconvenience for home owners. In the city of Omaha it is a requirement to clear the public sidewalks. In addition to obeying the law, snow can cause liability concerns should someone slip and be injured on your property. Most people will agree that the last thing they want to do on a blustery-cold day is crawl out of bed, typically early in the morning, to scoop snow before they head to work. The task of snow removal is such that

many people avoid it, even if it incurs a financial cost. Private contractors make a living on snow removal services during the winter season. This is no surprise. It proves that there is a substantial benefit to the home owner to pay for these services.

The second benefit of a radiant heat system is that the snow disappears on its own without costing extra money. There is no physical labor involved or capital expense to remove the snow. Payback is an important concept with investments. If the investment doesn't pay for itself or have a reasonable return on investment (ROI) it is not an attractive investment. In the Midwest it is not uncommon to pay $400-600 per winter for driveway cleaning by a private contractor (Sharp, 2012). If a furnace energy recovery system can be utilized as a radiant snowmelt system, the savings each season are substantial and can yield a short payback period depending on the cost of the energy recovery system.

The radiant heating system will consist or a runaround coil system. The pump will circulate a glycol solution through the radiant system outdoors and through an energy recovery coil indoors. The indoor recovery coil will extract excess heat from the furnace flue gasses as they travel through the flue. A revised flue system will be included in the considerations. The system shall extract as much heat from the gasses as necessary to provide a functioning snow melt system.

## 3.7. Economic Analysis

Unfortunately, many decisions in the world are driven by financial costs. Investments need to make a substantial impact so that the money is not wasted. The capital investment should produce outcomes that save money and produce a short

payback. The recovery system has little maintenance to be performed but should be considered in the payback analysis.

### 3.7.1. Analysis

Many people in the United States will own multiple houses in their lifetime so it is important to consider how long people typically own the same home. According to the 2010 American Community Survey completed by the U.S. Census Bureau, 61.6% of house owners in the state of Nebraska moved into their house in the year 2000 or later while only 17.6% moved into their house from 1990-1999 (U.S. Census Bureau, 2010). These statistics show that the vast majority of home owners live in one home for 12 years or less on average. For residential capital improvement to be beneficial to an owner the payback will need to be completed in less than 12 years.

A construction cost has been estimated per square foot of treated concrete slab. Each residence has a unique area that would be heated by such a system. Assuming an average snow removal cost of $100 per snowfall (industry average charge) will aid in the payback analysis. Costs for a hydronic radiant heat system range from $6-$12 per square foot of coverage (Anderson Radiant Heating, 2006). This price is the total installed cost, including a boiler to heat the glycol solution. Using the median price per square foot ($9/ft$^2$) a typical two-car driveway that is 35 feet long (~960 sq. ft.) would cost $8,640 to install a hydronic heating system. For a radiant system of this price to pay back in 12 years an average yearly cost for snow removal would be $720. This would yield a payback of 12 years. According to the National Oceanic and Atmospheric Administration, Omaha Nebraska averages 8.8 days per year where snowfall of at least 1” occurred (National Oceanic and Atmospheric Administration, 2010) (Current Results:

Research News & Science Facts, 2012). These average days of snow would yield a yearly cost of $880 per year at the stated rates.

Maintenance on the system would be negligible. The heat transfer solution wouldn't need to be drained or replaced very often on a closed system. To be conservative, a life maintenance cost for fluids has been estimated at $75. Of course the circulation pump could fail but is very unlikely given the yearly run duration. These pumps are only run when called upon and will not see many hours of operation each year. A conservative estimate of replacing the pump once every 12 years has been considered. The cost of the circulation pump used in this research was $80.

The cost of electricity in the city of Omaha Nebraska in the winter is listed below.

- $0.0964/kwh for the first 100 kwh
- $0.0834/kwh for the next 900 kwh
- $0.0579/kwh for the next 1000 kwh

The circulation pump utilized for this study was rated to 200 watts. The pump is considered to run for the entire day that the snow occurs. The power usage per snow day is below.

$$Power\ Usage = 200\ Watts * 24\ Hours * 8.8\ Days$$

$$Power\ Usage = 42{,}240\ watt - hours\ (42.24\ kwh)$$

**Table 5: Circulation Pump Electricity Cost (OPPD, 2012)**

| | **$0.0964/kwh** | **$0.0834/kwh** | **$0.0579/kwh** |
|---|---|---|---|
| **Yearly Cost** | $4.07 | $3.52 | $2.45 |

As ***Table 5*** above shows, the cost to run the circulation pump is negligible.

Considering the above analysis, a simple payback calculation has been shown below in ***Equation*** 3. As the calculation shows, there is a favorable payback considering the assumptions stated above.

$$Simple\ Payback = \frac{(Installed\ Cost + Operation * Years + Life\ Maintenance)}{Savings\ per\ Year}$$

$$Simple\ Payback = \frac{(\$8{,}640 + \$4.07X + \$155)}{\$880}$$

$$Simple\ Payback = \sim 10\ Years$$

**Equation 3: Simple Payback**

Finally, ***Figure 10*** shows the simple payback analysis. This analysis considers the maintenance cost as being divided up equally among the expected 12 year life of the pump and glycol solution. A typical contractor bill for snow removal in Omaha, Nebraska is provided in ***Appendix B***.

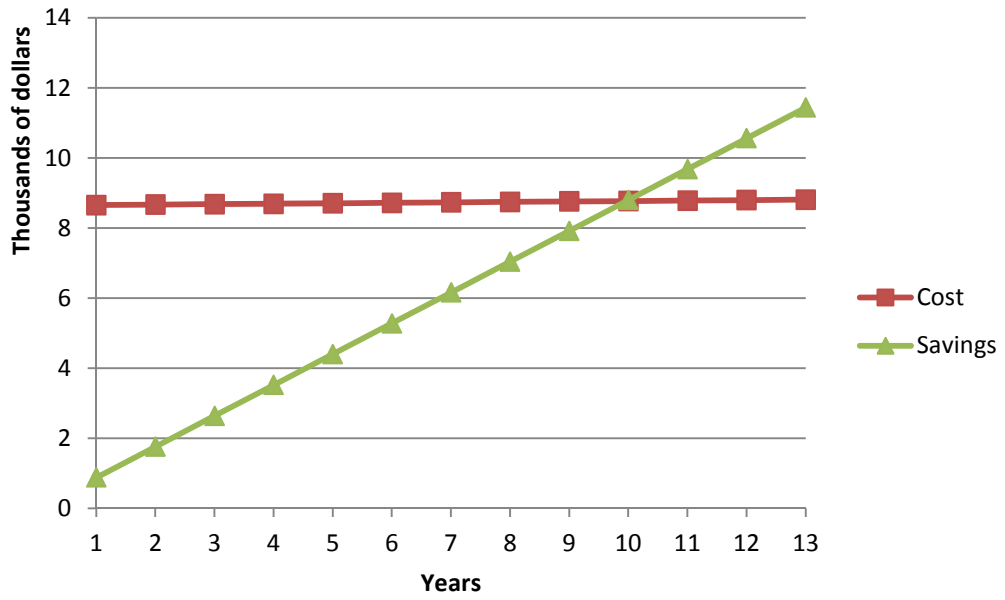



Figure 10: Simple Payback Graph

# CHAPTER 4: Initial Analytical Analysis

## 4.1. Radiant Heated Driveway Analysis

Analyzing a radiant snow-melting system has many variables to consider. Among these variables are (1) rate of snowfall, (2) snowfall-coincident air dry-bulb temperature, (3) humidity, (4) wind speed near the heated surface, and (5) apparent sky temperature (ASHRAE, 2011). By performing a heat balance on the radiant slab under a steady-state condition, a required heat flux for the concrete surface can be calculated. Equations listed below are sourced from the ASHRAE applications handbook.

$$q_o = q_s + q_m + A_r(q_h + q_e)$$

Equation 4: Radiant Slab Energy Balance

Where;

$$q_o = heat\ flux\ required\ at\ snow - melting\ surface, \frac{btu}{h \cdot ft^2}$$

$$q_s = sensible\ heat\ flux, \frac{btu}{h \cdot ft^2}$$

$$q_m = latent\ heat\ flux, \frac{btu}{h \cdot ft^2}$$

$$A_r = \frac{A_f}{A_t} = \frac{Snow\ Free\ Area}{Total\ Area} = snow - free\ area\ ratio, dimensionless$$

$$q_h = convective\ and\ radiative\ heat\ flux\ from\ snow - free\ surface, \frac{Btu}{h \cdot ft^2}$$

$$q_e = heat\ flux\ of\ evaporation, \frac{Btu}{h \cdot ft^2}$$

The sensible and latent heat flux variables are the energy required to raise the temperature of the falling snow, raise the temperature of the liquid film layer on the concrete, and to melt the snow. The snow-free area ratio is the ratio of concrete with no snow coverage to the total concrete surface area. When snow has covered the surface of the concrete it acts as an insulating layer preventing heat transfer from occurring. As the snow covered area increases, the amount of energy loss (transfer) from the concrete slab decreases.

Below is the equation for sensible heat flux $q_s$.

$$q_s = \rho_{water} s\left[c_{p,ice}(t_s - t_a) + c_{p,water}(t_f - t_s)\right]/c_1$$

**Equation 5: Sensible Heat Flux Equation**

Where;

$$c_{p,ice} = specific\ heat\ of\ ice, \frac{btu}{lb \cdot °F}$$

$c_{p,water} = specific\ heat\ of\ water, \frac{btu}{lb \cdot °F}$

$s = snowfall\ rate\ water\ equivalent, \frac{in}{hr}$

$t_a = ambient\ temperature\ coincident\ with\ snowfall,$ °F

$t_f = liquid\ film\ temperature,$ °F (usually taken as 33°F)

$t_s = melting\ temperature,$ °F

$\rho_{water} = density\ of\ water, \frac{lb}{ft^3}$

$c_1 = 12\ \frac{in}{ft}$

The heat flux of evaporation is the energy required to melt the snow. This value is calculated using ***Equation 6***.

$$q_m = \rho_{water} s h_{if} / c_1$$

**Equation 6: Heat Flux to Melt Snow**

Where;

$h_{if} = heat\ of\ fusion\ of\ snow, \frac{Btu}{Lb}$

As mentioned, the concrete slab will have heat losses associated with it if it is not being insulated by snow cover. The convective and radiative heat losses from the snow-free surfaces are described using ***Equation 7***.

$$q_h = h_c(t_s - t_a) + \sigma \varepsilon_s \left(T_f^4 - T_{MR}^4\right)$$

**Equation 7: Convective and Radiative Heat Loss**

Where;

$h_c = convection\ heat\ transfer\ coefficient\ for\ turbulent\ flow, \frac{Btu}{h \cdot ft^2 \cdot °\mathrm{F}}$

$T_f = liquid\ film\ temperature, °R$

$T_{MR} = mean\ radiant\ temperature\ of\ surroundings, °R$

$\sigma = Stefan - Boltzmann\ constant = 0.1712\ x\ 10^{-8} \frac{Btu}{h \cdot ft^2 \cdot °R^4}$

$\varepsilon_s = emittance\ of\ surface, dimensionless$

The convection heat transfer coefficient over a horizontal surface involvers turbulent airflow and uses a special heat transfer equation to calculate this coefficient. This equation is listed below in ***Equation 8***.

$$h_c = 0.037 \left(\frac{k_{air}}{L}\right) Re_L^{0.8} Pr^{1/3}$$

**Equation 8: Convective Heat Transfer Coefficient for Turbulent Flow over a Horizontal Plane**

Where;

$k_{air} = thermal\ conductivity\ of\ air\ at\ t_a, \frac{Btu \cdot ft}{h \cdot ft^2 \cdot °\mathrm{F}}$

$L = characteristic\ length\ of\ slab\ in\ direction\ of\ wind, ft$

$Pr = Prandtl\ number\ for\ air, taken\ as\ \mathrm{Pr} = 0.7$

$Re_L = Reynolds\ number\ based\ on\ characteristic\ length\ L$

Calculation of the Reynolds number is quantified using ***Equation 9***.

$$Re_L = \frac{VL}{\upsilon_{air}} c_2$$

**Equation 9: Reynolds Number Based on Characteristic Length**

Where;

$V = design\ wind\ speed\ near\ slab\ surface, mph$

$\upsilon_{air} = kinematic\ viscosity\ of\ air, \frac{ft^2}{h}$

$c_2 = 5280\ \frac{ft}{mile}$

The mean radiant temperature $T_{MR}$ is calculated using ***Equation 10***.

$$T_{MR} = \left[T_{cloud}^4 F_{sc} + T_{sky\ clear}^4 (1 - F_{sc})\right]^{1/4}$$

**Equation 10: Mean Radiant Temperature**

Where;

$F_{sc} = fraction\ of\ radiation\ exchange\ that\ occurs\ between\ slab\ and\ clouds$

$T_{cloud} = temperature\ of\ clouds, °R$

$T_{sky\ clear} = temperature\ of\ clear\ sky, °R$

The equivalent temperature of a clear sky is given by a curve fit of data in Ramsey et al. (1982).

$$T_{sky\ clear} = T_a - (1.99036\ x\ 10^3 - 7.562 T_a + 7.407\ x\ 10^{-3} T_a^2 - 56.325\phi + 26.25\phi^2$$

**Equation 11: Clear Sky Curve Fit Equation**

Where;

$T_a = ambient\ temperature, °R$

$\phi$ =*relative humidity of air at elevation which typical weather measurements are made*

Now that the convection heat transfer and radiative heat flux has equations to be calculated, the final term to evaluate is the evaporative heat flux from evaporating the film later of water on the surface of the concrete. The heat flux required to evaporate water from the wet concrete surface is equated using ***Equation 12***.

$$q_e = \rho_{dry\ air} h_m (W_f - W_a) h_{fg}$$

**Equation 12: Evaporative Heat Flux Required to Evaporate Water from Radiant Slab**

Where;

$h_m = mass\ transfer\ coefficient, \frac{ft}{h}$

$W_a = humidity\ ratio\ of\ ambient\ air, \frac{lb_{vapor}}{lb_{air}}$

$W_f = humidity\ ratio\ of\ saturated\ air\ at\ film\ surface\ temp, \frac{lb_{vapor}}{lb_{air}}$

$h_{fg} = heat\ of\ vaporization, \frac{Btu}{lb}$

$\rho_{dry\ air} = density\ of\ dry\ air, \frac{lb}{ft^3}$

In order to calculate the mass transfer coefficient, the convective heat transfer coefficient is needed. Using ***Equation 13*** the mass transfer coefficient can be calculated.

$$h_m = \left(\frac{Pr}{Sc}\right)^{2/3} \frac{h_c}{\rho_{dry\ air} c_{p,air}}$$

**Equation 13: Mass Transfer Coefficient**

Where;

$Pr = Prandtl\ number\ for\ air$

$Sc = Schmidt\ number$

Humidity ratios in the atmosphere and at the surface of the water film are calculated using the standard psychometric relation given in the following equation (from Chapter 1 of the 2009 ASHRAE Handbook – Fundamentals). This equation is listed below.

$$W = \left(\frac{p_v}{p - p_v}\right)$$

**Equation 14: Humidity Ratio**

Where;

$p_v = partial\ pressure\ of\ water\ vapor, psi$

$p = atmospheric\ pressure, psi$

Since atmospheric pressure varies depending on the location on earth and its relative altitude, ***Equation 15*** adjusts the atmospheric pressure based on location's altitude.

$$p = p_{std}\left(1 - \frac{A_z}{T_o}\right)^{5.265}$$

**Equation 15: Atmospheric Pressure Adjustment Based on Altitude**

Where;

$p_{std} = standard\ atmospheric\ pressure, psi$

$A = 0.00356 \frac{°R}{ft}$

$z = altitude\ of\ the\ location\ above\ sea\ level, ft$

$T_o = 518.7°R$

Using these equations, the heat flux required at the radiant slab surface can be calculated.

## 4.2. Heated Driveway Calculation

The list of variables and their specific values are found below in ***Table 6***. The variables describe a day where the ambient temperature is 30°F with a 17mph wind speed. The snowfall rate is described as 1" of snow per hour.

| | |
|---|---|
| $\boldsymbol{c_{p,ice} = 0.49\ \frac{btu}{lb \cdot °F}}$ | $\boldsymbol{c_{p,water} = 1.0\ \frac{btu}{lb \cdot °F}}$ |
| $\boldsymbol{\rho_{water} = 62.4\ \frac{lb}{ft^3}}$ | $s = 0.167\ \frac{in}{hr}\ (\sim 1"\ of\ snow\ a\ hr)$ |
| $\boldsymbol{t_a = 30}$ °F (dew point temp = 14°F) | $t_f = 33$ °F |
| $\boldsymbol{t_s = 32}$ °F | $h_{if} = 143.3\ \frac{Btu}{lb}$ |
| $\boldsymbol{V = 17\ mph}$ | $v_{air} = 0.49\ \frac{ft^2}{h}$ |

| | |
|---|---|
| $k_{air} = 0.0135\ \frac{Btu \cdot ft}{h \cdot ft^2 \cdot °F}$ | $L = 6\ ft$ |
| $Pr = 0.7$ | $Sc = 0.6$ |
| $\rho_{dry\ air} = 0.081\ \frac{lb}{ft^3}$ | $c_{p,air} = 0.24\ \frac{Btu}{lb \cdot °F}$ |
| $W_a = 0.00160\ \frac{lb_{vapor}}{lb_{air}}$ | $W_f = 0.00393\ \frac{lb_{vapor}}{lb_{air}}$ |
| $\varepsilon_s = 0.9$ | $h_{fg} = 1075\ \frac{Btu}{lb}$ |

**Table 6: Radiant Slab Calculation Variables**

With the equations listed in the previous section, a calculated energy flux can be calculated to determine the required heat for the radiant system to be feasible. In order to apply ***Equation 4***, various intermediate terms need to be calculated. The steps to calculate the energy required for the radiant slab are show below.

Applying ***Equation 5*** below, we find the sensible heat flux. This is the required heat to bring the snow temperature up to 32°F. Using ***Equation 6*** will calculate the heat flux required to melt the snow on the concrete slab. This is shown below.

$$q_s = 62.4\ x\ \frac{0.167}{12}[0.49(32 - 30) + 1.0(33 - 32)] = 1.72\ \frac{Btu}{h \cdot ft^2}$$

$$q_m = 62.4\ x\ \frac{0.167}{12}\ x\ 143.3 = 124.4\ \frac{Btu}{h \cdot ft^2}$$

The next term to be calculated is the heat loss from the slab surface because of convective and radiative heat transfer. This number is dependent on the snow-free area ratio. This calculation will be completed using various snow-free area ratios in order to simulate multiple conditions. If the radiant slab system was neglected to be turned on,

the snow would accumulate before the system was started. This would represent a snow-free area ratio of 0. If the radiant system was running from the start, then a snow-free area ratio of 1 would be used. Before the heat losses from the slab can be calculated using ***Equation 7***, the convective heat transfer coefficient must be calculated, which requires Reynolds number from ***Equation 9***.

$$Re_L = \frac{17\ x\ 6}{0.49}\ x\ 5280 = 1.09\ x\ 10^6$$

$$h_c = 0.037\left(\frac{0.0135}{6}\right)(1.09\ x\ 10^6)^{0.8}(0.7)^{1/3} = 4.99\frac{Btu}{h \cdot ft^2 \cdot °\mathrm{F}}$$

$$q_h = 4.99(32 - 30) + (0.1712\ x\ 10^{-8})(0.9)(492.67^4 - 489.67^4) = 12.17\ \frac{Btu}{h \cdot ft^2}$$

A couple final intermediate terms are required to apply the overall energy balance. One of those terms is the mass transfer coefficient which is calculated using ***Equation 13***.

$$h_m = \left(\frac{0.7}{0.6}\right)^{\frac{2}{3}} \frac{4.99}{(0.081\ x\ 0.24)} = 284.5\ \frac{ft}{h}$$

The final term to be calculated before the application of the energy balance is the evaporative heat flux from the surface of the concrete. This value represents the heat required to evaporate the water from the wet surface.

$$q_e = 0.081\ x\ 284.5(0.00393 - 0.00160)\ x\ 1075 = 57.7\frac{Btu}{h \cdot ft^2}$$

Finally, ***Equation 4*** can be applied. ***Table 7*** lists the heat required for a 10’ x 6’ slab.

$$q_o = 1.72 + 124.4 + A_r(12.17 + 57.7)$$

| | | |
|---|---|---|
| $\boldsymbol{A_r = 1.0}$ (no snow cover before startup) | $\boldsymbol{q_o = 196.0\ \frac{Btu}{h \cdot ft^2}}$ | $\boldsymbol{Q = 11.7\ \frac{kBtu}{h}}$ |
| $\boldsymbol{A_r = 0.5}$ (half snow cover before startup) | $q_o = 161.0\ \frac{Btu}{h \cdot ft^2}$ | $Q = 9.6\ \frac{kBtu}{h}$ |
| $\boldsymbol{A_r = 0}$ (full snow cover before startup) | $q_o = 126.1\ \frac{Btu}{h \cdot ft^2}$ | $Q = 7.6\ \frac{kBtu}{h}$ |

**Table 7: Heat Flux Required at Snow-Melting Surface**

As shown in the calculations, a bulk of the heat required is due to the evaporative energy required to evaporate water from the wet radiant surface. This is largely due to the wind speed of 17mph. Various conditions will adjust what is required of the radiant system to operate properly. Diversity needs to be considered in this calculation since there is a low possibility that the coldest day with the largest wind speeds will fall on the same day and time that the largest snowfall rate per hour occurs. Another important note is that the heat required for the radiant slab calculated above does not take into account the heat losses at edges. Edge loss is dependent on the slab construction and how it was insulated. Adlam (1950) demonstrated that these back and edge losses may vary from 4 to 50%, depending on factors such as slab construction, operating temperature, ground temperature, and back and edge insulation and exposure (ASHRAE, 2011).

## 4.3. Furnace Combustion Analysis

The combustion process at the furnace will determine how much heat is obtainable. The amount of heat released during the combustion process depends heavily upon the amount of natural gas that is combusted during the process. As discussed, the

amount of air available for the combustion process will determine if there is a complete burn of the fuel or not. In order to ensure complete combustion has taken place and every Btu of available heat from the gas has been released, the combustion system is analyzed. Using Avagadro's law, the theoretical air mass of dry air required for the complete combustion of a hydrocarbon can be calculated. This is shown below in ***Equation 16***.

$$m_a = 2.47CO + 34.28H_2 + 17.24CH_4 + 16.09C_2H_6 + 15.68C_3H_8 + 15.47C_4H_{10} + 13.27C_2H_2 + 14.78C_2H_4 + 6.08H_2S - 4.78O_2$$

**Equation 16: Theoretical Air (based on mass) (2009 ASHRAE Fundamentals, 2009)**

In actual combustion processes, it is common practice to use more air than the stoichiometric amount to increase the chances of complete combustion or to control the temperature of the combustion chamber (Cengel & Boles, 2006). The extra air introduced to the combustion process is called excess air. ***Equation 17*** below calculates the excess air percentage being introduced by the induced combustion fan.

$$Excess\ Air\ \% = \frac{Air\ Supplied - Theoretical\ Air}{Theoretical\ Air}$$

**Equation 17: Excess Air Percentage**

Combustion efficiency is usually maximized when just enough air is supplied and properly mixed with the combustible gasses to ensure a complete combustion (2009 ASHRAE Fundamentals, 2009). The amount of dry air supplied to the combustion can be calculated using a flue gas analyzer to determine the make-up of the flue gasses. A flue gas analyzer determines the compounds/elements present in the flue gas as well as

the volumetric percentage. The dry air supplied, calculated using ***Equation 18***; can be input into ***Equation 17*** to calculate excess air.

$$Dry\ Air\ Supplied = \frac{C(3.04N_2)}{CO_2 + CO}$$

**Equation 18: Dry Air Supplied**

Where;

$Dry\ Air\ Supplied = unit\ mass\ per\ unit\ mass\ of\ fuel$

$C = \frac{unit\ mass\ of\ carbon\ burned}{unit\ mass\ of\ fuel}$ $, corrected\ for\ carbon\ in\ ash$

$CO_2, CO, N_2 = percentages\ by\ volume\ from\ flue\ gas\ analysis$

Excess air can be calculated using a number of equations. Using volumetric percentages obtained from a flue gas analysis, ***Equation 19*** below can be used to calculate the excess air percentage.

$$Excess\ Air\ \% = 100\left(\frac{P}{A}\right)\left(\frac{U - CO_2}{CO_2}\right)$$

**Equation 19: Excess Air % Based On Volumetric Percentages (2009 ASHRAE Fundamentals, 2009)**

Where;

$U = CO_2\ percentage\ from\ stoichiometric\ combustion$

$CO_2 = carbon\ dioxide\ content\ of\ flue\ gases, \%$

$$P = dry\ products\ from\ stoichiometric\ combustion, \frac{unit\ volume}{unit\ volume\ of\ burned\ gas}$$

$$A = air\ required\ for\ stoichiometric\ combustion, \frac{unit\ volume}{unit\ volume\ of\ burned\ gas}$$

The ratio of P/A is approximately 0.9 for most natural gases. Since excess air calculations are almost invariable made from flue gas analysis results and theoretical air requirements are not always known, ***Equation 20*** can be used based on percentages by volume obtained from a flue gas analysis (2009 ASHRAE Fundamentals, 2009).

$$Excess\ Air\ \% = \frac{100[O_2 - (CO/2)]}{0.264N_2 - [O_2 - (CO/2)]}$$

**Equation 20: Excess Air % Based On Volumetric Percentages Alt. 2 (2009 ASHRAE Fundamentals, 2009)**

The analysis above will be used in further sections.

# CHAPTER 5: Experimental Testing Setup

## 5.1. Flue Gas Energy Recovery System Design

The purposed recovery system will consist of a runaround energy recovery loop with a large coil embedded in a concrete slab outdoors. This coil material will be PEX tubing routed throughout the slab. A copper coil in the indirect fired furnace flue will be used to recover heat from the combustion gasses. A glycol solution consisting of 30% propylene glycol and 70% water will act as the energy transfer medium and will be pumped runaround coil system. Fluid properties of the solution are listed below in ***Table 8***. The glycol additive is advantageous since it modifies the properties of water. It lowers the freezing temperature of water as well as raises the boiling temperature. The most important aspect of the glycol is the reduction of the freezing temperature. This is because we will be heating a concrete slab that will be below the freezing temperature of water. The potential of a freezing condition is nearly guaranteed.

| Percent Propylene Glycol | Freezing Point | Specific Gravity @ 60°F | Specific Heat @60°F |
|---|---|---|---|
| 30% | 7°F | 1.026 | 0.935 $\frac{Btu}{lb\,°F}$ |

**Table 8: Propylene Glycol Solution Properties**

Modifications to the flue system are next to be analyzed. Condensation is highly possible in the flue depending on the temperature of the glycol solution being circulated through the recovery coil. This causes issues with flue materials due to the corrosive nature of the flue gas condensate. Modifications to the flue need to maintain a free path to the outdoors. A higher glycol temperature will generate a greater ΔT with the radiant

system, thus more heat transfer to the cold concrete and snow. A higher glycol temperature will require a higher rate of energy recovery from the combustion gasses.

Modifications to the furnace installation may cause issues with the UL listing of the furnace. Manufacturers typically will not warranty equipment after modifications have been made to the packaged system. Additionally, any equipment with a UL listing will lose the listing when modifications have been made. So long as the performance and longevity of the equipment is not hindered, warranties would be put back into place pending manufacturer review of the modified product. Initial modifications to the flue will include placing a copper coil inside of the flue. When the furnace is fired the glycol solution will pick up excess heat from the flue gasses as they travel to the outdoors. A primary concern would be that condensate would come into contact with the furnace heat exchanger (See ***Figure 7***). In an 80% efficient furnace the heat exchanger is not built with corrosion resistant materials. If condensate comes into contact with the heat exchanger, deterioration of the material, over time, will allow flue gasses to contaminate supply air to the residence.

The recovery coil was designed to fit inside a 5" galvanized flue. Flue gas will pass all around the coil as the furnace runs. ***Figure 11*** below shows the initial design. Modifications to the recovery coil will be made and discussed in a later section. The initial design is ten (10) feet of coiled ½" copper tubing. After initial testing the recovery coil it was evident that an increase in heat transfer was necessary. Heat transfer was increased by increasing the area at which heat transfer was occurring. Discovering the increased requirement on heat transfer is what drove future modifications.

**Figure 11: Initial Copper Energy Recovery Coil**

The glycol solution returning from the outdoor radiant heat loop will run through the recovery coil to gain heat. A circulation pump will be used to circulate the glycol solution through the runaround coil system.

The information of the indirect-fired natural gas furnace to be experimented on is listed below. Nominally there are 21Mbh of waste energy available to be transferred to the energy recovery system assuming no heat is lost to the surroundings. This value is calculated using an AFUE of 80%.

- Model: Carrier 58PAV
- Gas Input Rating: 110,000 Btu/h
- Heat Output Rating: 89,000 Btu/h
- 80% AFUE

***Figure 12*** below shows the modified furnace flue with the energy recovery coil installed. The furnace configuration before modifications can be viewed in ***Appendix A***.

**Figure 12: Natural Gas Fired Furnace with Energy Recovery**

PEX tubing is connected between each coil, which will transport the glycol solution and thus transfer the recovered energy to the radiant system outdoors.

The pump used to circulate the glycol solution through the system is shown below in ***Figure 13***. The pump data for this pump is located in ***Appendix B***. Since the system will be tested as an open system, the pump will have to overcome the change in elevation as well as friction losses through the piping and coils. The furnace is located in a basement mechanical room, so the pump had to be capable of producing a minimum of 12' of head to overcome the change in height. Other losses will be present in the PEX tubing as the fluid flows through due to friction losses.


Figure 13: System Circulation Pump

## 5.2. Radiant Heat Snow Melt System Design

Radiant heating systems are becoming more common to residential applications as the years have passed. It is common for residences to have radiant heated floors in restrooms, laundry rooms and other rooms around the residence. Radiant heating systems not only create a warm surface to walk on but also help to control room temperatures. The heated floor helps heat the rest of the room through radiative heat transfer. Piping for the radiant snowmelt system will be installed in the concrete slab to provide a path for the glycol solution to travel through. The heated glycol solution will lose heat to the concrete slab which will raise the temperature of the slab. This will eventually transfer from the warm concrete slab to the snow to melt it. A piece of concrete at my house was identified as an issue after my first winter living there. Melting snow from the roof would drop onto the walkway when the sun was out, and then freeze once the sun set. This created black ice. Icy spots are a liability since neighbors; delivery persons, mail

persons and others could potentially slip and be injured. Additionally, the concrete pad was heaving as shown in ***Figure 14***. This slab ended up being perfect for the analysis. The concrete slab measures 10' x 6' in overall dimension.

**Figure 14: Radiant Heat Slab Proposed Location**

Two separate PEX heating loops were run in the concrete slab. As shown below in ***Figure 15***, the two loops run in nearly the same path but with flows opposite of each other. This is intentional to be sure that the concrete was getting an evenly heated surface. As the fluid travels through the piping, it will transfer heat to the lower temperature mass by laws of thermodynamics. This cross flow path will also help to even out the change in temperature from the inlet to the outlet across the heating loops. For additional control, the heating loops can be uncoupled to have different flow rates if desired. The two loops were made of continuous runs of ½" PEX tubing (polyethylene with cross-linking). Continuous loops were installed to insure that leaks would not occur in the concrete slab.

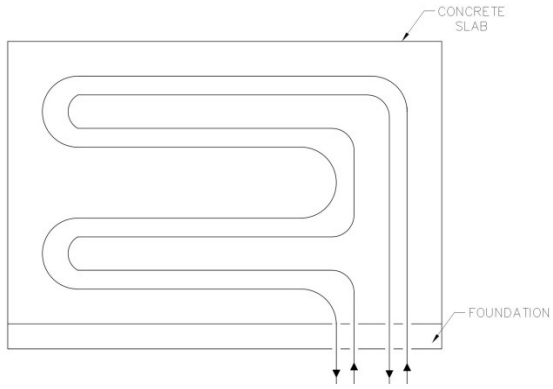



**Figure 15: Radiant Heat Piping Schematic**

Preparation below the PEX tubing is very important since the earth is a large heat sync and has the ability to absorb a lot of energy. This is due to the abundance of mass. A layer of 2" thick board insulation (polystyrene) was installed between the ground and the PEX tubing. This is also installed on the edges of the slab. ***Figure 16*** below shows a section of a typical slab preparation for the installation of a radiant heat system inside a home. Although the diagram is for the interior of a home, the preparation of the ground slab is identical.

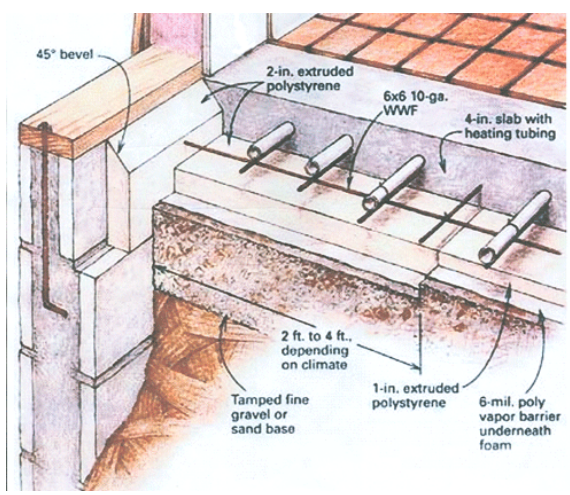



**Figure 16: Radiant Heat Slap Preparation (www.radiantcompany.com, 2009)**

Heat will transfer from a warm mass to any colder mass, as discussed. It is desired to have as much heat transfer to the concrete slab and remain there in order to melt the snow. Snow will first have to be brought up to 32°F, where it can begin melting thereafter. After it has melted, the concrete surface will have to evaporate the water film layer. This is evident from the calculations discussed in the *Radiant Heated Driveway Analysis* section. A 6 mil polyethylene vapor barrier was installed below with the insulation to prevent moisture migration from the earth. The vapor barrier will help keep the radiant system from absorbing moisture when it is at a higher temperature than the surrounding earth. ***Figure 17*** shows the radiant system before the concrete was poured. Rebar mesh was placed into the concrete for structural bracing. This will also help keep the concrete from cracking during expansion and contraction. The coverage of the PEX tubing is not uniform as shown. Initially it was unclear how well the PEX joints between tubes would seal. This was the first time I had installed PEX in concrete. The continuous loops solved leak worries but created difficulties for equal spacing and coverage of the concrete slab. Kinks were avoided to help conserve on pumping power. Kinks would reduce the cross-sectional area of the PEX tubing which would in-turn increase the frictional losses as the fluid is pumped through the system.

**Figure 17: Radiant Heating System before Pour**

PEX piping leaves the concrete slab and penetrates through the foundation and into the adjacent garage. From this location it was ran into the basement where the furnace is located. ***Figure 18*** below shows the penetration through the foundation. The loops are staggered so that flow passes in opposite directions for better coverage of the concrete slab.

**Figure 18: PEX Penetration through Foundation**

After the PEX tubing was zip tied into place on the rebar mesh and the piping was secured inside the garage, the slab was poured. The finished slab is shown below in ***Figure 19***. The concrete joints had pure silicon poured to fill gaps, then sealed with the proper sealing compound. Expansion joints were utilized between existing concrete slabs as well as between the adjacent foundation.

**Figure 19: Finished Radiant Heat Slab**

## 5.3. Data Collection

Initial measurements were taken separately at each coil to determine how effective the coils were operating. ***Figure 20*** shows how the collective system is tied together. Additionally, it shows the measurement points that will be collected. By measuring incoming and outgoing temperatures we can determine how much energy is being transferred by using an energy balance. A cumulative list of points measured is listed below in ***Table 9***. Surface temperatures were measured using an infrared temperature sensor while fluid temperatures were measured using a calibrated digital sensor.

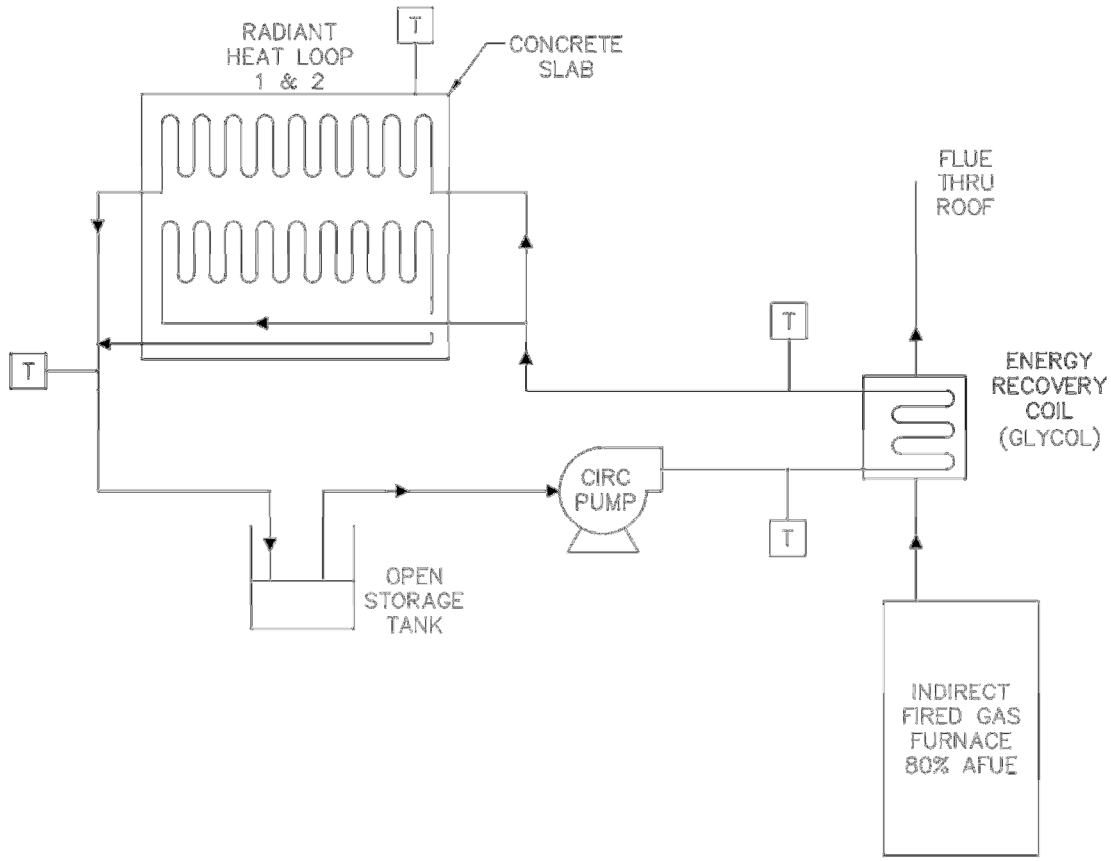



**Figure 20: Runaround Energy Recovery Schematic**

| | |
|---|---|
| AMBIENT TEMPERATURE | WIND SPEED |
| CONCRETE SURFACE TEMPERATURE | FLUID FLOW RATE |
| OUTDOOR COIL ENTERING FLUID TEMP | OUTDOOR COIL LEAVING FLUID TEMP |
| RECOVERY COIL ENTERING FLUID TEMP | RECOVERY COIL LEAVING FLUID TEMP |

**Table 9: Measured Points**

Wind speed and ambient temperature were recorded by a local weather station. The actual wind speed that the radiant slab will vary since the concrete slab is up against the house. The airflow patterns through this area will not be consistent.

Concrete slab temperatures are recorded using a Dewalt® model DCT414 cordless infrared thermometer. The thermometer is shown below in ***Figure 21***. Accuracy of this tool is the following;

- +/-1.5% or +/-1.5°C, whichever is greater, assumes ambient operating temperature of 23°C to 25°C (73°F to 77°F) on a black body source (EMS 0.95).


**Figure 21: Infrared Thermometer**

The digital thermocouple used to measure glycol solution temperatures is shown in ***Figure 22***. The accuracy for the different functions of the multi-meter is included in ***Appendix B***. The accuracy for the digital thermocouple is listed below;

- For 0-50°F » +/- 5.0% reading + 4 digits
- For 50-750°F » +/- 3.0% reading + 4 digits


**Figure 22: Craftsman Multi-Meter**

The combustion process will be analyzed using a flue gas analyzer. The analyzer has the ability to measure the volumetric percentages of the different gasses present in the flue gas airstream. Using the analysis discussed in the *Furnace Combustion Analysis* section, the combustion process can be analyzed. The flue gas analyzer used is shown below in ***Figure 23*** and ***Figure 24***. The accuracy of the combustion analyzer tool is listed in ***Appendix B***.


**Figure 23: Combustion Analyzer Computer**

**Figure 24: Combustion Analyzer Probe**

# CHAPTER 6: Experimental System Test Results

## 6.1. Initial Flue Gas Energy Recovery System Results

The flue gas temperature leaving the furnace heat exchanger was measured to be 370°F. This temperature is encouraging since there is a large temperature differential for our heat transfer process with the radiant system. The induced-draft furnace being tested has high flue gas temperatures coming out of the heat exchanger to avoid flue gas condensation from occurring. A large potential for heat recovery from the flue gas exists since the concrete slab will be at 32°F or below.

After installing the energy recovery flue system, the PEX distribution piping was connected to the circulation pump as well as the open storage container. For the initial set of measurements the outdoor coil was not connected. The furnace cycled to run from a cold start, so losses to the furnace cabinet are expected. This allows for accurate testing of a typical installation, with a properly selected/sized furnace. Cycle times will be consistent with standard residential installations. Initial fluid temperature and flow rate was set as listed below in ***Table 10***. For the indoor recovery coil test, pure water was used to eliminate variables from the experiment.

| Initial Fluid Temperature | Fluid Flow Rate |
|---|---|
| 89°F | 0.75 GPM |

Table 10: Indoor Initial Fluid Conditions

An average change in temperature across the recovery coil was measured to be around 6°F for this particular coil and flow rate. Shown below in ***Figure 25***, the temperature had a steady increase during the 7 minutes the furnace was firing. The average run-time of

the furnace is assumed to be consistent with standard residences. When programmable thermostats are used, a longer run-time will occur during changes in set points. This operation is commonly referred to night/day setbacks. This will allow for a greater temperature rise of the heating fluid due to the increased furnace operation time. This system test is worst case, assuming no changes in thermostat setpoints.

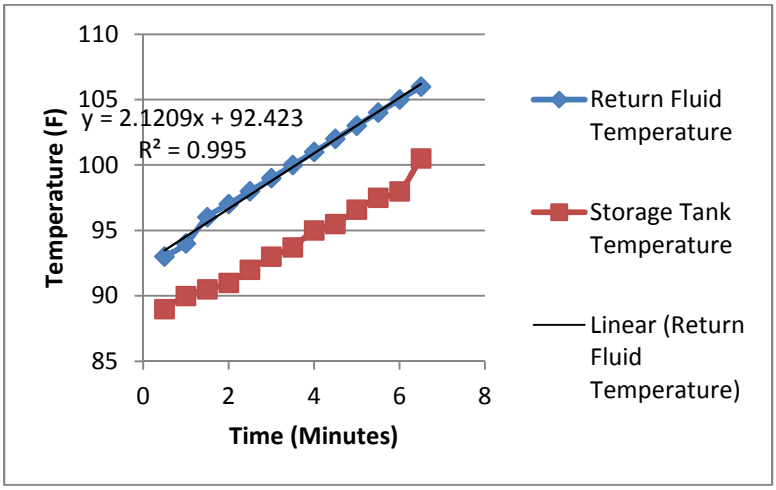



**Figure 25: Indoor Coil Fluid Return Temperature vs. Time**

As the correlation shows, the fluid is gaining about 2.3 degrees to the return temperature each minute that the system is in operation. The change in temperature from upstream/downstream glycol temperature measurements averaged out to be near 6°F, as shown below in ***Figure 26***. Using this temperature and the information listed above in ***Table 8*** the average energy transfer from the flue gasses to the glycol solution is around 2.1 MBH or 0.63 kilowatts. Given that the coil is very basic and had this amount of energy transfer is encouraging.

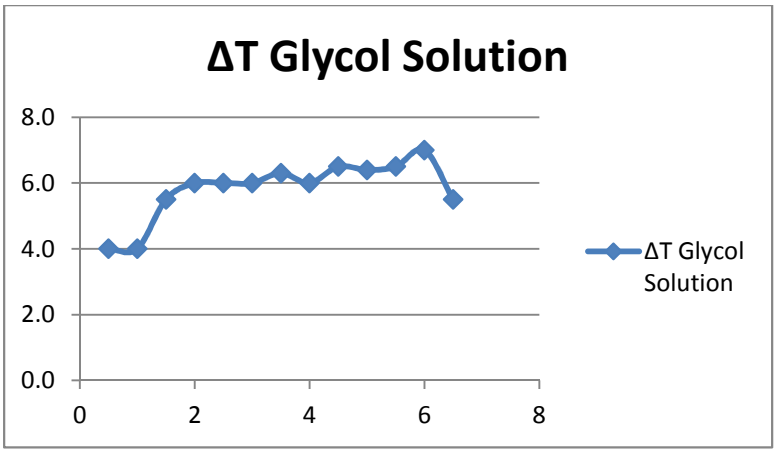



**Figure 26: Glycol Solution Recovery Change in Temperature**

During the testing of the indoor energy recovery coil, condensation formed inside of the flue. Condensation is occurring since the hot flue gasses are passing over the cold coil. The coil was around 90°F, which is below the flue gas dew point temperature of 130°F. Since the glycol solution is below the dew point temperature of the flue gasses, condensation is expected. The initial flue design was allowing condensation to make its way back into the main furnace heat exchanger so further modifications were completed. These modifications will be discussed in a later section.

## 6.2. Initial Radiant Heat Snow Melt System Results

Testing of outdoor radiant system involved patience since the winter didn't bring much snow to test with. Below, ***Figure 27*** shows the initial snow cover during the stand-alone testing. Initial conditions are listed below in ***Table 11***. The garage temperature is important to note since the open, un-insulated storage tank was located in the garage. This tank was not insulated so losses to the garage are significant. To simulate energy input from the recovery system at the furnace, a 1500W submersible heater was cycled on and off to input 750W over the duration of the experiment. The heater simulated the

heat recovered from the furnace accurately. This was a preliminary test run to view the performance of the outdoor coil system.

**Figure 27: Initial Snow Cover**

| AMBIENT TEMPERATURE (FEELS LIKE) | WIND SPEED |
|---|---|
| **29.1°F (19°F)** | **NW 21MPH, gusts up to 35MPH** |
| GARAGE TEMPERATURE | INITIAL GLYCOL SOLUTION TEMPERATURE |
| **46°F** | **48°F** |
| GLYCOL SOLUTION FLOW RATE | INITIAL SLAB TEMPERATURE |
| **0.7GPM** | **17°F** |

**Table 11: Ambient Conditions for Initial System Tests**

As the data shows, the supply temperature of the heating solution was not as warm as what is possible from the indoor recovery coil. Losses to the garage are noticeable since the fluid temperature decreased to the garage temperature over time. This signifies that heat transferred to the garage. ***Figure 28*** shows the return temperature measurement from the radiant heating coil. The temperature increase indicates that less than 750 watts

were being transferred from the glycol solution to the radiant slab, or lost to the environment, or both. Losses from the storage tank to the garage included.

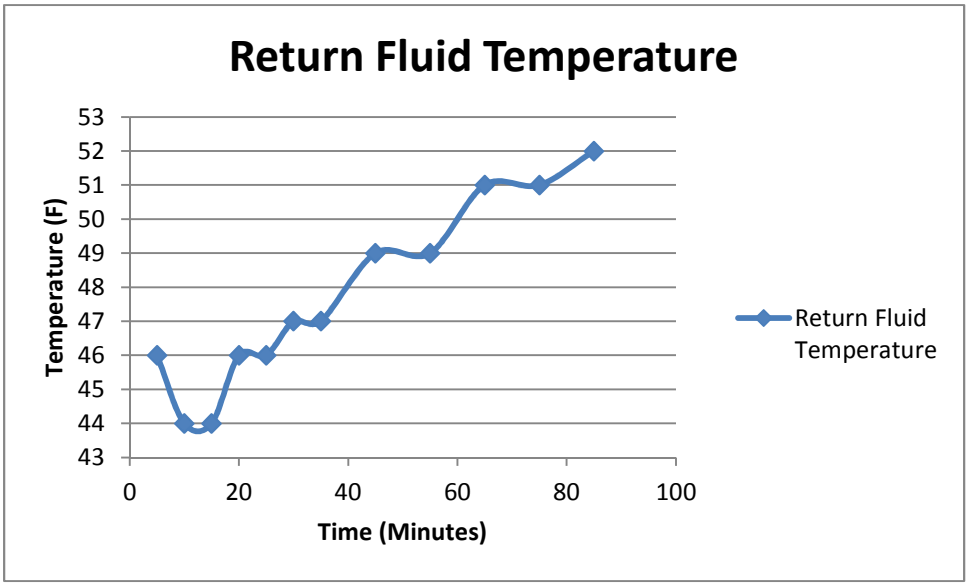



Figure 28: Radiant Heating Glycol Solution Return Temperature

As the system ran, pictures were taken at various times throughout the test. These pictures are included in ***Appendix A***.

## 6.3. Full System Test and Analysis

After the initial experimentation of the individual coils, it was time to test the system as a whole. Penetrations from the garage to the basement mechanical room were made in order to run insulated ½" PEX tubing to connect the coils. Modifications to the design of the energy recovery coil/flue were necessary to allow condensate to be drained off. This was done to prevent condensation from flowing back into the primary heat exchanger inside the furnace. As shown below in ***Figure 29***, a 'T' fitting was installed to prevent condensation from draining downwards into the furnace. This configuration allowed easy separation of the condensation. This fitting does create more static pressure

for the induced blower fan to overcome, but it proved to not be too much of an issue the furnace did not shut itself off. The furnace is equipped with a sensor that will not allow the furnace to operate if the flue static pressure is too high. This is a safety precaution to prevent the infiltration of flue gasses into the residence should the flue be blocked.

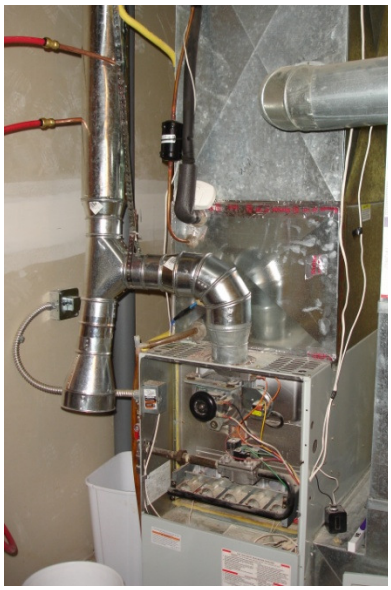


**Figure 29: Drainable Flue Configuration**

The circulation pump was inserted into the system to connect the whole operable system. An open storage container was utilized again to allow for easy temperature measurements of the glycol solution. Insulation was used on the PEX tubing to prevent losses while being transferred from the basement mechanical room to the outdoor radiant slab. PEX tubing was run exposed from the garage to the radiant slab. The open storage container was also wrapped in insulation to prevent losses to the environment. A commercial storage tank was not included as the budget didn't allow this.

On February 4th, 2012 there was a large snow storm that came through the Omaha area that allowed for testing of the complete system. Conditions that day are listed below

in ***Table 12***. The pump was started at 7:45AM after 3.5" of snow had already fallen. Initial snow cover is shown below in ***Figure 30***. Blowing snow also added to the test. There was an official total of 9.1" of snowfall that day.

| AMBIENT TEMPERATURE (FEELS LIKE) | WIND SPEED |
|---|---|
| **30.6°F (22°F)** | **N 17MPH, gusts up to 23MPH** |
| GARAGE TEMPERATURE | INITIAL GLYCOL SOLUTION TEMPERATURE |
| **46°F** | **55°F** |
| GLYCOL SOLUTION FLOW RATE | INITIAL SLAB TEMPERATURE |
| **2.1 GPM** | **~27°F** |

**Table 12: Ambient Conditions for Full System Test**

**Figure 30: Full System Test Initial Snow Cover**

During the test the thermostat for the house was set at 70°F and remained there for the duration of the test. As shown in ***Figure 31***, for a human with a 1.0 clo clothing level, this would be a comfortable setpoint. A temperature between 68°F and 77°F would be comfortable to a human depending on the humidity ratio. For reference, a winter business suit has about 1 clo of insulation, and a short-sleeved shirt and trousers has about 0.5 clo (2009 ASHRAE Fundamentals, 2009).

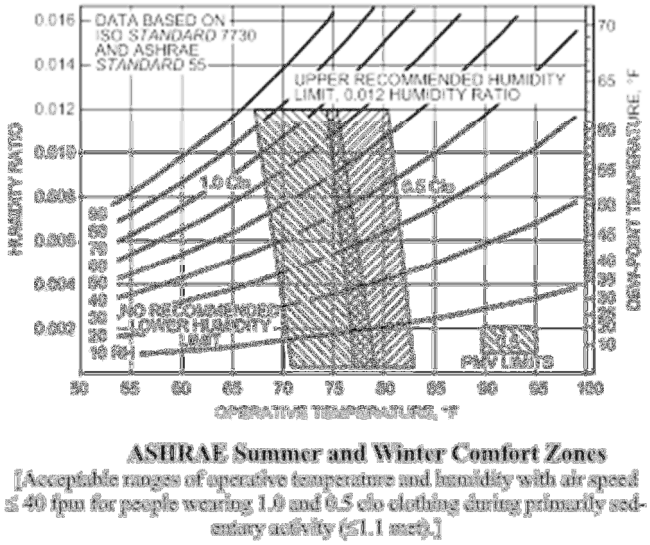



**Figure 31: Human Comfort Zones (2009 ASHRAE Fundamentals, 2009)**

A full progression of the radiant snow melt system can be seen in ***Appendix A***. As the system systematically melted snow, the storm kept dropping more snow onto the radiant slab. In addition to the 9.1" of snow that fell that day, the snow was a very dense wet snow which added a degree of difficulty to the test. This dense snow added to the energy required to evaporate the water film layer from the concrete surface. Midway through the test it was deemed necessary to increase the energy recovery coil capacity by increasing the heat transfer area of the coil. A second coil was built to fit inside the original circular coil. This is shown below in ***Figure 32***. The coils were run in series so that the heat transfer fluid would flow through the circular coil first, then through the vertical coil. This larger coil was put into the 5" galvanized flue and into operation.

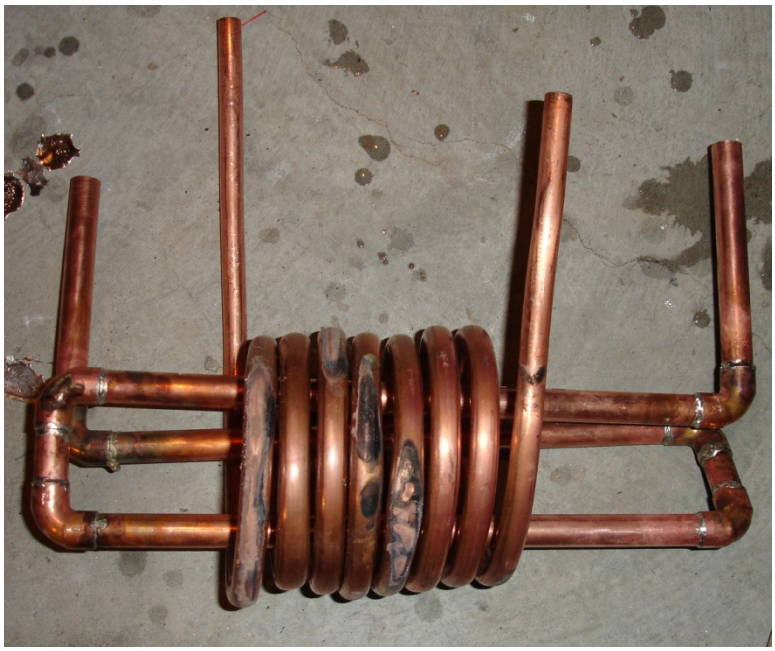


**Figure 32: Modified Energy Recovery Coil**

With the increase in heat transfer area, the ΔT of the heat transfer fluid increased an additional 3-4°F. This is a substantial increase in heat transfer and aided in the timeliness of the snow melting process. The second coil was intentionally placed in the center of the flue gas air stream to access the hottest part of the flue gas flow. Boundary layers of a laminar flow occur at the edges which act as insulating layers, keeping hot flue gas from the edges. This is shown below in ***Figure 33***. This concept would cause the original circular coil to not recover as much heat as if the tubing was installed in the center of the air stream.

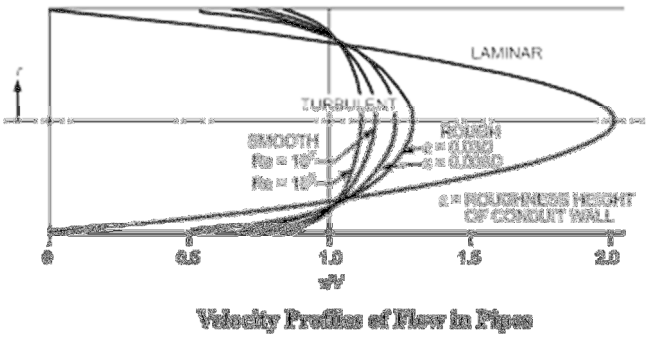



**Figure 33: Velocity Profiles of Flow in Pipes (2009 ASHRAE Fundamentals, 2009)**

Cold spots on the flue were present where the circular outer coil was touching the flue. The cold areas represented places where flue gasses were not flowing. This could also be due to the formation of boundary layers from the flow of flue gasses. ***Figure 34*** shows the installation of the modified recovery coil onto the flue. As shown, the heat transfer fluid was circulated through the coils in series. A substantial increase in the production of condensate was observed after the addition of the second coil. The increase reinforces the fact of more heat transfer occurred and thus more energy recovery was taking place.

**Figure 34: Modified Coil Installed in Flue**

The final picture of the system's progress before being shut down is shown below in ***Figure 35***. This is a picture 24 hours after the system was initially started. There are portions of the slab that were not covered by PEX piping as shown.

**Figure 35: Final Radiant System Result**

## 6.4. Furnace Combustion Analysis Results

The tested Carrier furnace has a rated fuel input of 110Mbh with an 80% AFUE. Using the flue gas analyzer, the volumetric percentages of the various gasses can be measured. From the measured data, the excess air percentage can be calculated to ensure complete combustion of the natural gas. During the use of the combustion analyzer, it was apparent that the sensitivity of the sensor was not up to the manufacturer's specifications. The readings were out of spec for atmospheric air. Several attempts were made to calibrate the sensor but none corresponded to accurate measurements. Given the expense of the tool, it was apparent that this analysis would not take place. One publication listed the average combustion analyzer results (Bergmann, 2006). These are shown below;

- Efficiency: 80-82%
- Oxygen: 7.0%-9.0%
- Carbon Dioxide: 6.5%-8.0%
- Stack Temp: 325°F to 400°F
- Draft: -0.02”WC to -0.04”WC
- Carbon Monoxide: < 100 ppm (undiluted)

The flue gas temperature was measured to be ~370°F. This measurement is in line with the normal measurements taken for an induced-draft furnace.

Due to the combustion analyzer not operating properly, the furnace combustion operation can be spot checked using ***Equation 21***. The measured incoming/outgoing supply air temperatures are listed below in ***Table 13***. The tested furnace is known to provide 1,410 CFM while in heating mode. Using this equation it provides a total sensible heat transfer equivalent. Based on the measured supply air temps, the sensible heat transfer can be checked against the manufacturer’s claims of 88 Mbh sensible heat transfer.

$$Total\ Sensible\ Heat = 1.08 * CFM * \Delta T$$

**Equation 21: Total Sensible Heat Transfer**

| | **Upstream of Furnace Heat Exchanger (return air temp)** | **Downstream of Furnace Heat Exchanger** |
|---|---|---|
| **Temperature (°F)** | 66°F | 124°F |

**Table 13: Average Incoming/Outgoing Supply Air Temp**

When measuring the supply air temperature downstream of the heat exchanger, it is important to be far enough away from the heat exchanger as to not see elevated readings from the radiation of the heat exchanger. Measuring the supply air temperature close to the furnace is important so that heat losses to the environment from non-insulated supply ductwork won't affect the calculated heat transfer. Overall, we can see that the furnace is operating within the specifications of the manufacturer. In conclusion, we can assume that the amount of heat available for recovery is near the nominal value. The manufacturer data for the furnace tested is located in ***Appendix C***.

# CHAPTER 7: Analysis and Discussion

## 7.1. Flue Gas Energy Recovery Analysis and Discussion

Energy recovered from the flue gas has a high potential since the temperature of the gasses are around 370°F from the furnace heat exchanger. This gives a large ΔT with the glycol solution temperature. One of the limiting factors for energy recovery from the furnace is the duration at which the furnace is being fired. As more energy was recovered from the flue gasses, the heating solution temperature continued to rise. As mentioned, the increased glycol solution temperature will decrease the time required to melt snow that has accumulated. This is again due to the larger ΔT of the heat transfer process. The maximum temperature achievable would be that of the flue gasses. This temperature would be nearly impossible to achieve. The hand built coil that was tested is not the most efficient coil available. Increases to coil heat transfer surfaces, such as fins, were not implemented into the design. Coil manufacturers incorporate turbulators as a tool to increase heat transfer. Turbulators increase turbulence of the fluid flowing through the coil so that boundary layers associated with laminar flow do not occur. Laminar flow through a pipe creates a boundary layer at the pipe surfaces. This was shown in ***Figure 33***. Boundary layers prevent liquid inside the pipe from transferring energy. The boundary layers act as an insulator preventing heat transfer. The same concept could be applied to the flue gases before crossing the recovery coil. Turbulators could be placed in the flue to ensure turbulence of the flue gasses.

## 7.2. Radiant Heat Sub-System Analysis and Discussion

The radiant heat system had losses associated with it that made testing tricky. The storage tank used was not insulated, which allowed the heat to be released into the surroundings rather than being transferred to the radiant system. Losses from the storage tank can be minimized by installing a domestic hot water storage tank. Storage tanks are required, per IPC 2009, to have an insulating value of R-12.5. This would dramatically reduce losses to the environment from the storage tank.

The data was collected when the wind speed was substantial. Gusting winds create turbulent airflows which will cool the concrete slab dramatically. This makes the radiant heat system less effective since heat is removed from the water film layer quickly. Even though the conditions were not favorable, the system still performed as desired. The system was tested in a conservative fashion and the desired outcome was achieved. As components to the system are upgraded and improved, the performance of the system will continue to increase. This is encouraging since the tested system was done on a budget with minimum equipment costs.

## 7.3. Furnace Combustion Analysis and Discussion

Operation of the furnace is required for energy transfer to the concrete slab. The furnace generates the waste heat required to actively melt the snow that has fallen onto the radiant slab. It was assumed that the published furnace data of an 80% AFUE was accurate. This would leave 21Mbh of wasted heat from the furnace. A quick calculation of the percentage of excess air allows assumptions to be made as to whether the combustion is complete or not. Having found that the combustion analyzer was giving

incorrect readings, a quick check was conducted in order to ensure that the waste heat was available to be recovered. The heat transferred to the residence supply air, in combination with system and flue gas losses, accounts for the 110Mbh of gas input to the furnace. The AFUE rating is determined by vigorous lab facility testing. Since the AFUE rating can't be measured in the field (Bergmann, 2006) we must indirectly analyze the system. Through the data collected the heat transferred to the residence supply air was calculated to be 88Mbh. This is consistent with manufacturer data as well. Given this information, it is a safe assumption that 22Mbh is available to be recovered from the operation of the furnace. One study found that the furnaces tested were generally within the manufacturer's specifications for all parameters except for external static pressure, which exceeded specifications (Pigg & Parkhurst, 2007). Given this research, the combustion and heat transfer process of the indirect-fired gas furnace is accurate. This would reinforce the assumption that the AFUE rating of the furnace is accurate.

## 7.4. Uncertainty Analysis and Discussion

Uncertainties are present in energy recovery analysis. Each measurement device had an accuracy associated with its measurement. Since the data from the combustion analyzer was not utilized, it will be neglected from this section. Shown below in ***Table 14***, the uncertainty of the calculated heat transfer across a coil does not have a large uncertainty.

| Independent Variables | Inputs | Uncertainty |
|---|---|---|
| Entering Glycol Solution °F | 55°F | +/- 1.7°F |
| Leaving Glycol Solution °F | 62°F | +/- 1.9°F |
| Fluid Flow Rate oz/sec | 1.5oz/sec | +/- 0.25 oz |
| **Dependent Variables** | **Output** | **Uncertainty** |
| Sensible Heat Transfer | 2.5 Mbh | +/- .2 Mbh |

**Table 14: Uncertainty of Heat Transfer across a Coil**

The calculations don't have many dependent variables. Uncertainty of the measurements has a smaller effect than if there were many dependent variables in numerous calculations. This minimizes uncertainties since the individual measurement uncertainties don't compound on each other. Quantities multiplied or divided by each other require that the *Propagation of Uncertainty* principal be applied. In addition, the findings of the experiment are not dependent on the calculations directly since a full scale model was constructed and tested successfully.

# CHAPTER 8: Closing Statements

## 8.1. Conclusion and Recommendations

Residential energy recovery has a large potential for reducing the amount of waste energy from residences in the United States. An energy recovery radiant heating system successfully applies waste energy from a typical residential furnace to a radiant heat system. The recovered energy was computed to be 9.8Mbh for a basic coil constructed out of copper from an 80% AFUE 110MBH furnace. This coil was placed inside the flue of the furnace to recover waste heat. A 30% glycol heat transfer solution was utilized to transfer waste energy to a radiant heated slab outdoors. The energy recovery system was very similar to a runaround coil. A small circulation pump was utilized to pump the solution through pex tubing to transfer energy. The basic runaround system was proven to be enough to melt snowfall during a large Midwest snowstorm. Calculated energy requirements for a radiant heated slab are an accurate assessment while neglecting edge losses. Additionally, the application of a radiant heating system is proven to be advantageous to home owners and provides an incentive for the capital improvement. Payback on the recovery system is within a reasonable time period for which a home owner resides in the same home for the city of Omaha, Nebraska. Savings each year are observed by reduced spending to contractors for snow removal during winter months.

If this type of system is incorporated into the design of the residence, the installation cost would be much lower than that of a renovation. It is expected that decreased installation costs would decrease payback periods and increase attractiveness to new home owners. Increased efficiencies of the system are possible through the

incorporation of various components. In conclusion, a residential energy recovery radiant heat system is a viable option for residential energy recovery in its basic state.

# CHAPTER 9: Future Considerations

## 9.1. Glycol Heating Solution Storage Tank (closed system)

The system, as tested, ran a continuous loop when radiant heat for the driveway was desired. By incorporating a storage tank into the system as well as some two-way control valves, the indoor recovery system can be operated separate from the outdoor radiant system. With this system modification, the indoor circuit could continue to recover heat from the flue gasses when the radiant system is not being utilized. An external hot water storage tank could be used to store the heated glycol solution. As the furnace cycles over a time period, the heating fluid temperature would get hotter and eventually plateau an equilibrium temperature. By monitoring the liquid temperature inside the storage tank, a control valve would control whether the flow continues through the storage tank or bypasses it. The maintained temperature would be dependent on how long the furnace cycles and how often it cycles. When it snows and the radiant driveway system is desired, fluid from the storage system can be circulated to the driveway. This will increase the capacity of the radiant system, allowing for a greater treated area. Two-way control valves and a basic controls system would be accompanied with this upgrade. A BMS would easily handle this operation. Another important factor in providing a sealed storage tank would be that the glycol heating loop would be a closed system. After the system is purged of air, the pump head required to circulate the loop will be substantially less than pumping with an open system. In a closed system, the pump does not have to overcome changes in elevation. This will decrease the size of the pump required for the system. A decrease in the pump size will result in a smaller motor

and thus a smaller operating cost. A schematic of this setup is shown below in ***Figure 36***. In lieu of two-way control valves, modulating control valves could be utilized for finite temperature control of each loop. Please note unions and ball valves are not shown in the diagram for simplicity. A VFD could also be provided to allow for a desired ΔT of the radiant system to be achieved. This is to say that the VFD speed could be controlled by a calculation of the ΔT of the radiant coil. The operation of the pump by a VFD would also allow for savings in pumping energy.

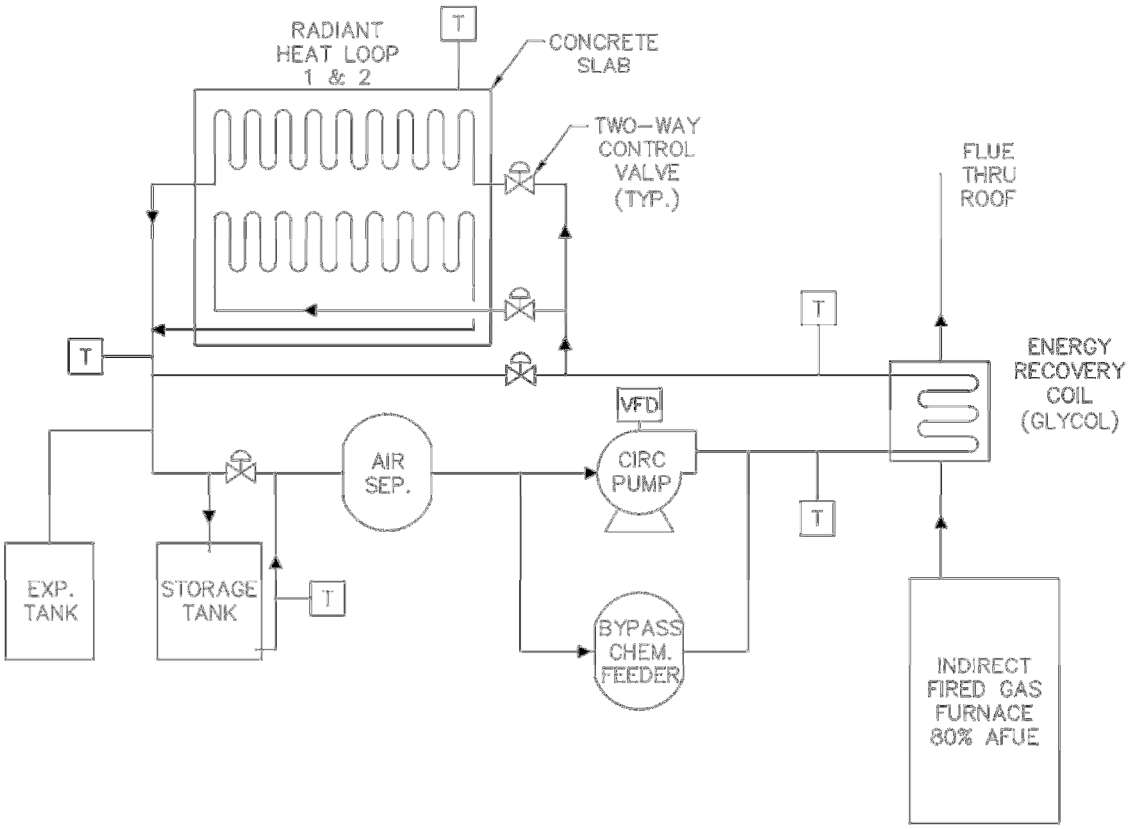



**Figure 36: Storage Tank Schematic Diagram**

## 9.2. Three-Way Control Valve

The system tested provided full-flow through the system when in operation. By installing a modulating three-way control valve, the ΔT at one of the coils could be monitored and controlled. The major advantage of this would be to allow the temperature of the heating fluid at the flue to increase. By increasing the temperature of the heating fluid the snow melt system will work substantially faster. ***Figure 37*** below shows a schematic diagram of the installation of the three-way control valve. The control valve can be modulated based on the temperature of a desired point or difference in temperature. The most advantageous approach would be to maintain a ΔT between the exiting temperature from the radiant loop and the exiting temperature from the recovery coil. This system utilizes a full-speed pump, so no pumping energy is conserved.

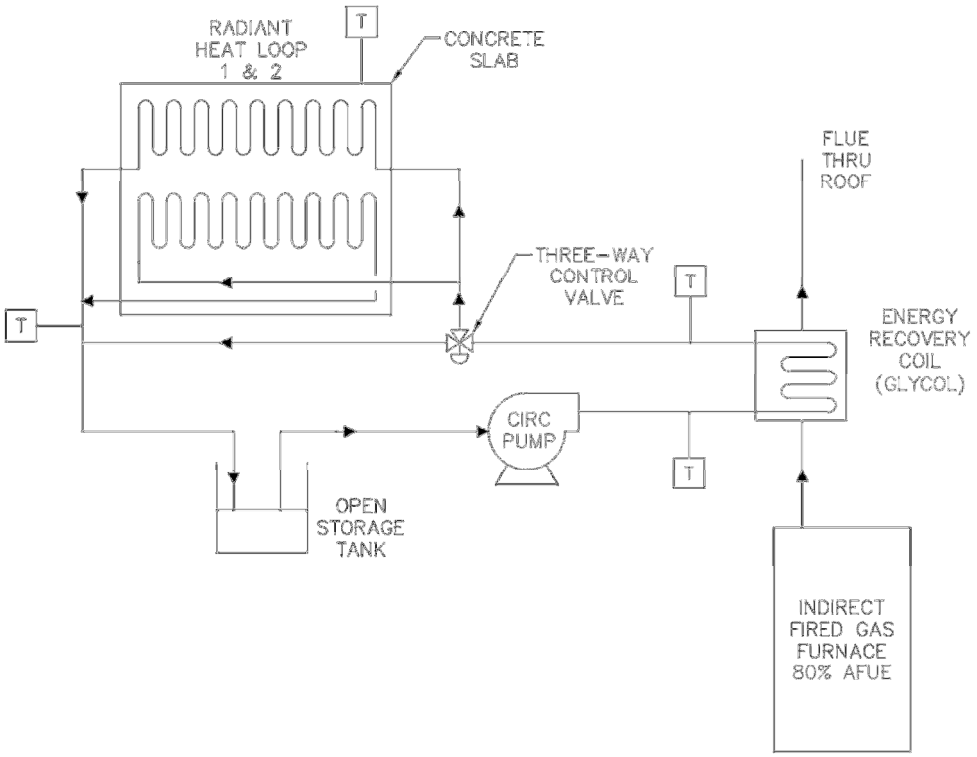



Figure 37: Three-way Valve Schematic Diagram

It is important to note that when the three-way control valve is either fully-open or closed, one of the coils would be stagnant (no flow). Since the fluid has a glycol additive, freezing of the fluid is not a concern.

## 9.3. Condensing Flue Gasses for Increased Recovery Capacity

During the experiments, flue gasses started condensing on the recovery coil since it was below the dew point of the flue gasses. This condensation is very corrosive, as noted. In order to recover large amounts of heat off the flue gasses, condensation is inevitable. Flue materials need to be chosen wisely since our furnace produces gas in excess of 370°F. A PVC venting system will melt at those temperatures since the melting point of PVC is around 156°F. The max recommended service temperature of PVC is 140°F (Harvel Plastics, Inc, 2012). Additionally we do not want the condensation coming in contact with the furnace burners or heat exchanger since it will corrode the components. If flue gasses are routed away from the furnace, such as in the tested system, condensation can be contained and prevented from coming into contact with the furnace. If multiple coil sections were installed, eventually there would need to be a transition from galvanized type B venting material to sealed PVC pipe. PVC could be used after the temperature of the flue gas has dropped below the melting temperature. A more in depth analysis of the energy recovery heat transfer process would be required to find when the transition could be made. Changes in the coil design and flue routing would be important to consider in the research. Another option would be to install a stainless steel vent pipe. The downfall of this would be the cost associated for the materials. To further increase the energy recovered, the condensation could be treated to

allow for a coil to be submerged in the collected condensate. The condensation could surround the coil to transfer energy from the condensation to the glycol solution. This would require a chemical treatment plan, which would incur a chemical cost.

## 9.4. Energy Recovery Coil Design and Considerations

The coil used for testing was basic. This was a conservative measure taken to prove the functionality of the system. Coils of various shapes and sizes can be ordered through commercial manufacturers. The number of rows in the coil depends on the application of the coil. Primary coils for an air handler can have anywhere from 1 row to 4 or 6 rows depending on the airflow across the coil and desired capacity. The number of fins per inch is also a variable in coil design. Typical coils utilize around 10-12 fins per inch of coil. A special flue system would need to be designed around the coil selected. The increased performance of the recovery coil would produce more condensate however. In addition to a properly placed and sized condensate drain, the venting material would need to be corrosion resistant. This type of system modification would need to be installed on an induced blower furnace rather than a traditional furnace that utilizes the stack effect for venting (category I). As coil performance increases, the static pressure drop across the coil also increases. A larger induced blower fan could be sized to account for the increased flue static pressures.

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# Appendix A

**Figure 38: Initial Snow Cover (Test 1)**

**Figure 39: Radiant Slab (Test 1, T=15min)**

**Figure 40: Radiant Slab (Test 1, T=49 Min)**

**Figure 41: Radiant Slab (Test 1, T=76 Min)**

**Figure 42: Radiant Slab (Test 1, T=87 Min)**

**Figure 43: Radiant Slab (Test 1, T= 94Min)**

**Figure 44: Radiant Slab (Full-System Test T= 0Min)**

**Figure 45: Radiant Slab (Full-System Test T= 2.5 Hours)**

**Figure 46: Radiant Slab (Full-System Test T= 4.1 Hours)**

**Figure 47: Radiant Slab (Full-System Test T= 5.8 Hours)**

**Figure 48: Radiant Slab (Full-System Test T= 7.8 Hours)**

**Figure 49: Radiant Slab (Full-System Test T= 9 Hours)**

**Figure 50: Radiant Slab (Full-System Test T= 10 Hours)**

**Figure 51: Radiant Slab (Full-System Test T= 11.1 Hours)**

**Figure 52: Radiant Slab (Full-System Test T= 12 Hours)**

**Figure 53: Radiant Slab (Full-System Test T= 15.5 Hours)**

**Figure 54: Radiant Slab (Full-System Test T= 21 Hours)**

**Figure 55: Radiant Slab (Full-System Test T= 23.5 Hours)**

**Figure 56: Radiant Slab (Full-System Test T= 25.5 Hours)**

**Figure 57: Radiant Slab (Full-System Test T= 26.5 Hours)**

**Figure 58: Radiant Slab (Full-System Test T= 27 Hours)**

**Figure 59: Radiant Slab (Full-System Test T= 28 Hours)**


**Figure 60: Unmodified Furnace Installation**

**Figure 61: Unmodified Furnace Installation #2**

# Appendix B

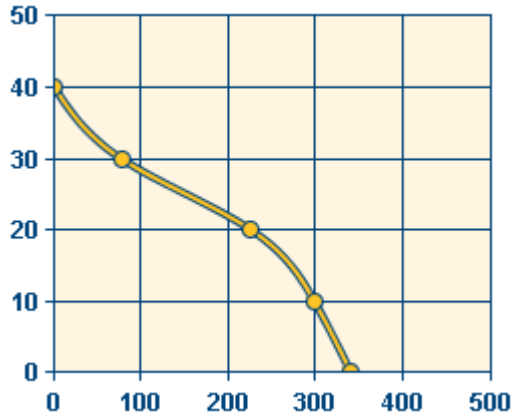


Figure 62: Wayne 'PC2' Utility Pump Curve

# Series CA-6200 CA-CALC™ Multi-Gas Combustion Analyzers

*TSI's Series CA-6200 CA-CALC™ Multi-gas Combustion Analyzers are rugged, fast-response, easy-to-use instruments ideal for tuning burners for maximum efficiency and safety–from large industrial systems to small residential appliances. The basic unit measures $O_2$, CO, draft, ambient temperature and stack temperature. Optional sensors can be added for measuring NO, $NO_2$, $SO_2$ and high CO concentrations.*


## Features

- Industry-leading service
- Intuitive operation with large backlit display
- Real-time data displayed on a single screen
- Operates on AC power or C-Cell batteries (24 hrs)
- $O_2$ Reference adjustable from 0% to 18%
- Rugged case, built-in magnets, heavy-duty pump
- Pump status monitored continuously
- Calculated emission rates include lb/MBtu and ng/J

### Advanced Sensor Technology

- Quick field installation of factory-calibrated sensors
- Four sensor slots designed for easy field upgrades
- $H_2$-compensated CO sensor with manual isolation valve and high-limit protection
- Automatic baseline calibration of sensors
- Recalibrates easily for critical safety checks
- Sensors protected by a condensation moisture barrier

### Superior Data Logging and Printing

- Store up to 680 sets of combustion test data
- Define up to 35 customers and 75 boilers/burners
- Test data time/date stamped for easy on-site access
- Download data to a computer or print hard copy report

## Applications

- Tune burners in industrial boilers, heat exchangers, residential furnaces, process heaters
- Supplement preventative maintenance
- Monitor and service diesel systems
- Measure $NO_X$ and $SO_2$ emissions
- Monitor emissions controls and burner performance
- Check carbon monoxide safety

## Models available for:

- Boiler/Burner Service and Repair Contractors
- Boiler/Furnace Maintenance Companies
- Plant Engineers
- Process Boiler Technicians
- Utility Companies
- Boiler Owners and Manufacturers

TSI®

# Specifications

## Series CA-6200 CA-Calc Multi-Gas Combustion Analyzers

### Sensors

| | |
|---|---|
| **Oxygen ($O_2$)*** | |
| Range | 0 to 25% |
| Resolution | 0.1% $O_2$ |
| **Carbon Monoxide (CO)–$H_2$ Compensated*†** | |
| Range | 0 to 5,000 ppm |
| Resolution | 1 ppm |
| **Carbon Monoxide (CO)–High Concentration*†** | |
| Range | 0 to 2% |
| Resolution | 5 ppm |
| **Nitric Oxide (NO)*†** | |
| Range | 0 to 4,000 ppm |
| Resolution | 1 ppm |
| **Nitrogen Dioxide ($NO_2$)*†** | |
| Range | 0 to 500 ppm |
| Resolution | 1 ppm |
| **Sulfur Dioxide ($SO_2$)*†** | |
| Range | 0 to 4,000 ppm |
| Resolution | 1 ppm |
| **Flue Gas Temperature Probe** | |
| Range | 32 to 1,800°F (0 to 1,000°C) |
| Resolution | 1°F (1°C) |
| **Draft Pressure** | |
| Range | ±30 in. $H_2O$ (±80 mBar) |
| Resolution | 0.01 in. $H_2O$ (0.01 mBar) |
| **Supply Air Temperature Probe (Optional)**** | |
| Range | -40 to 302°F (-40 to 150°C) |
| Resolution | 1°F (1°C) |

### Calculated Data

| | |
|---|---|
| Carbon Dioxide ($CO_2$)†† | 0 to $CO_2$ Max |
| Excess Air (EA) | 0 to 1,000% |
| Loss ASME | -25 to 100% |
| Efficiency ASME (net) | 0 to 125% |
| Loss qA (Siegert) | -25 to 100% |
| Efficiency (η) Based on qA | 0 to 125% |
| Lambda (λ) | 0 to 10 |
| $CO/CO_2$ Index ($CO_r$) | 0 to 1.0000 |
| NOx | 0 to 4,500 ppm |

### Operating Conditions

| | |
|---|---|
| **Instrument Temperature Range** | |
| Operating Range | 32 to 113°F (0 to 45°C) |
| Storage Range | -22 to 140°F (-30 to 60°C) |
| **Instrument Humidity Range** | |
| Continuous | 15 to 90% non-condensing |
| Intermittent | 0 to 99% |
| **Maximum Flue Gas Probe Temperature** | |
| Continuous | 1,800°F (1,000°C) (with handle shielded) |

### General Data

| | |
|---|---|
| **Instrument** | |
| External Dimensions | 6 × 10 × 2.5 in. (15 × 25.4 × 6.4 cm) |
| Weight | 2.5 lbs/3.9 lbs with probe (1.12/1.75 kg) |
| Display | 1/4 VGA B/W |
| **Pump** | |
| Flow Rate | Nominal 700 cc/min |
| Maximum Flue Pressure | ±30 in. $H_2O$ (±80 mBar) |
| **Standard Flue Gas Sampling Probe** | |
| Probe/Hose Material | Stainless steel/rubber |
| Probe Length | 12 in. std (30 cm) |
| Hose Length | 9 ft (2.74 m) |
| Probe Diameter | 5/16 in. (0.80 cm) |
| **Emission Flue Gas Sampling Probe†††** | |
| Probe/Hose/Material | Stainless steel/rubber/Teflon® |
| Probe Length | 12 in. std (30 cm) |
| Hose Length | 9 ft (2.74 m) |
| Probe Diameter | 1/2 in. (1.27 cm) |
| **Communication Interface** | |
| Type | Serial |
| Baud Rate | 1,200 to 19,200, selectable |
| **Power Requirements** | |
| Batteries | 4 size C alkaline batteries |
| Battery Life | >24 hours (pump on); >48 hr (pump off) |
| AC Adapter | P/N 2613033 (NA), 2613078 (EU) |
| Backup Battery | Lithium |
| Backup Battery Life | 3 yrs |

* Electrochemical sensor
** P/N 3013003
† Selectable units include mg/m³, g/bhphr, ppm, lb/MBtu, ng/J, g/MWh, mg/MJ, mg/kJ, mg/kWh and g/kWh.
††Calculated from $O_2$ and fuel type
††† Required for $NO_2$ and $SO_2$ measurement

Teflon is a registered trademark of E.I. du Pont de Nemours and Company.
Specifications are subject to change without notice.

## Ordering Information

| Model | Carbon Monoxide (CO) | Carbon Monoxide 0-2% ($CO_{hi}$) | Oxygen ($O_2$) | Nitric Oxide (NO) | Nitrogen Dioxide ($NO_2$) | Nitrogen Oxides ($NO_x$)* | Sulfur Dioxide ($SO_2$) | Draft Pressure | Ambient, Stack Temp. | Carbon Dioxide ($CO_2$)** | Efficiency (Loss) (qA) | Excess Air (λ) | $CO/CO_2$ Index ($CO_r$) | Adjustable $O_2$ Reference*** | Water Trap and Filter | Data Storage/Review/Print | 8 Factory and 5 User-Defined Fuels | Emissions Probe for $NO_2$ and $SO_2$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| CA-6210 | • | | • | | | | | • | • | • | • | • | • | • | • | • | • | |
| CA-6211 | • | | • | • | | • | | • | • | • | • | • | • | • | • | • | • | |
| CA-6212 | • | • | • | | | | | • | • | • | • | • | • | • | • | • | • | |
| CA-6213 | • | | • | • | | • | • | • | • | • | • | • | • | • | • | • | • | • |
| CA-6214 | • | • | • | • | | • | | • | • | • | • | • | • | • | • | • | • | |
| CA-6215 | • | | • | • | • | • | | • | • | • | • | • | • | • | • | • | • | • |
| CA-6216 | • | | • | | • | | • | • | • | • | • | • | • | • | • | • | • | • |

* $NO_X$=NO+$NO_2$ in Model CA-6215
** Calculated from fuel type and $O_2$
***CO undiluted (air free) when $O_2$ reference=0


**TSI Incorporated**
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CERTIFIED ISO 9001


*P/N 2980225 2/03*



*Printed in U.S.A.*

## Series CA-6210 thru CA-6216 CA-CALC™ Combustion Analyzer Specifications*

***Carbon Monoxide (CO)– H2 compensated All Models***
Range:.....0-5000 ppm
Accuracy:.....0–100 ppm ±5 ppm or 10% of reading
100–2000: ±5% of reading
2000–5000: ±10%
Resolution: .....1 ppm
Response Time***: .....<30 seconds to 90% of step change

***Oxygen (O2*) All Models***
Range:.....0–25%
Accuracy:.....±0.3% O2
Resolution: .....0.1% O2
Response Time**: <30 seconds to 90% of step change

***Nitric Oxide (NO) Models CA-6211, CA-6213, CA-6214, CA-6215***
Range:.....0–4000 ppm
Accuracy:.....0–100 ±5 ppm
> 100–1000 ±5% of reading
>1000–4000 ±10% of reading
Resolution: .....1 ppm
Response Time**: .....<60 seconds to 90% of step change

***Nitrogen Dioxide (NO2) Models CA-6215, CA-6216***
Range:.....0–500 ppm
Accuracy:.....0–100 ±5 ppm
> 100–200 ±5% of reading
>200–500 ±10% of reading
Resolution: .....1 ppm
Response Time**: .....<40 seconds to 90% of step change.

***Sulfur Dioxide (SO2) Models CA-6213, CA-6216***
Range:.....0–4000 ppm
Accuracy:.....0–200 ±5 ppm
> 200–1000 ±5% of reading
>1000–4000 ±10% of reading
Resolution: .....1 ppm
Response Time**: .....<40 seconds to 90% of step change.

***Carbon Monoxide (CO)- High Range Models CA-6212, CA-6214***
Range:.....0–2% ppm
Accuracy:.....0–500 ±25 ppm
>500–10000 ±5% of reading
>1%–2% ±10% of reading
Resolution: .....5 ppm
Response Time**: .....<30 seconds to 90% of step change

***Flue/stack Temperature***
Range:.....32–1800°F (0-1000°C)
Accuracy:.....±2°F (1°C) to 390°F (199°C)
±0.5% of reading >390°F (199°C)
Resolution .....1°F (1°C)

***Draft Pressure***
Range .....±30"$H_2O$ (±80mbar)
Accuracy:.....±1% of rdg. or 0.01" $H_2O$ (zeroed)
Resolution .....±0.01"$H_2O$

***Supply Temperature (optional)***
Range:.....-40 to 302°F (-40 to 150°C)
Accuracy:.....±2°F (1°C)
±0.5% of reading
Resolution .....1°F (1°C)

*Note: These specifications assume the instrument is allowed to stabilize at the operating temperature before being turned on.*
***With a 9' sampling tube length and water trap*

---

*Specifications are subject to change without notice. Specification giving lowest accuracy is applied.

Search Here

# DCT414S1

12V MAX* Infrared Thermometer Kit

5 of 5 (2 reviews)

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## Product Features

- Non-contact temperature measurements (-20°F to 932° F) for ease of use over distance and accuracy
- LED hot (red) & cold (blue) indicator for visual and/or audio alarm for quickly locating trouble spots
- Data storage for recording max, min, and average temperatures
- Adjustable emissivity improves accuracy across a wide range of materials

Powered by an ENERGY STAR® qualified battery charger for a better environment

12V MAX* LITHIUM ION

*Maximum initial battery voltage (measured without a workload) is 12 volts. Nominal voltage is 10.8.

**Product Tour**

SPECS & WARRANTY | QUESTIONS & ANSWERS | USER REVIEWS (2)

## Specifications

| | |
|---|---|
| Display Type | 1.4" Numeric Display with Backlit LCD |
| Temperature Measurement Range | -20º F to 932º F (-29º C to 500º C) |
| Accuracy | +/-1.5% or +/- 1.5º C, whichever is greater, assumes ambient operating temperature |
| Field of View; Distance: Spot | (D:S) 12:1 |
| Adjustable Emissivity | 0.1 to 1.0, Adjustable in Increments of 0.01 |
| Data Storage | Min, Max and Avg Temperature of Last Reading |
| Voltage | 12V MAX* |
| Operating Temperature Range | 14º F to 122º F (-10º C to 50º C) |
| Temperature Storage Range | -25º F to122º F (-32º C to 50º C) |
| Shipping Weight | 4.8lbs |

## Warranty Information

This DEWALT® High Performance Industrial Tool comes with a warranty package that includes:

- 3 Year Limited Warranty
- 1 Year Free Service Contract
- 90 Day Money Back Guarantee

**More information on the general DEWALT warranty**

## DCT414S1 Includes:

- DCT414 IR thermometer
- (1) 12V MAX* Li-Ion battery
- fast charger
- kit box

## RELATED TOOLS

**DCT410S1**

12V MAX* 17mm Inspection Camera with Wireless Screen Kit

4 of 5 (12 reviews)

**DCL510**

12V MAX* LED Worklight

5 of 5 (9 reviews)

**DCF610S2**

12V MAX* 1/4" Screwdriver Kit

5 of 5 (14 reviews)

**DCF815S2**

12V MAX* 1/4" Impact Driver Kit

5 of 5 (18 reviews)

**DCD710S2**

12V MAX* 3/8" Drill Driver Kit

4.5 of 5 (13 reviews)

PETERSEN"S DEPENDABLE QUALITY LANDSCAPING
17326 PIERCE CIRCLE
OMAHA, NEBR. 68130

# Invoice

Date 2/9/2012
Invoice # 07-1328

PAID

Bill To

Ship To

P.O. #
Terms Due on receipt

Ship Date 2/9/2012
Due Date 2/9/2012
Other

| Item | Description | Qty | Price | Amount |
|---|---|---|---|---|
| SNOW REM... | SNOW REMOVAL SERVICES : AS PER CONTRACT. { 2 / 4 / 2012 } 10.5 in. Snow. | 10.5 | 10.00 | 105.00 |
| SNOW REM... | SNOW REMOVAL SERVICES : AS PER CONTRACT. { 1 / 23 / 12 } 1 + in. Snow | 1 | 20.00 | 20.00 |
| | Subtotal | | | 125.00 |
| TAX | | | 5.50% | 6.88 |

| | |
|---|---|
| Subtotal | $131.88 |
| Sales Tax (0.0%) | $0.00 |
| Total | $131.88 |
| Payments/Credits | $-131.88 |
| Balance Due | $0.00 |

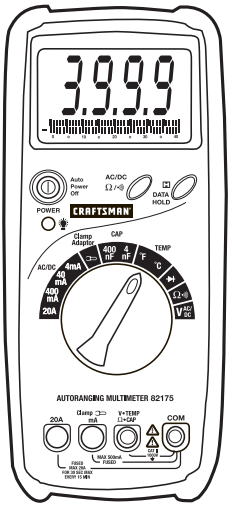
***PETERSEN"S DEPENDABLE QUALITY LANDSCAPING***
pdqlandscaping@yahoo.com 402-691-8346
pdqlandscaping.net 402-691-4558

**Owner's Manual**

CRAFTSMAN

# AutoRanging Digital MultiMeter

**Model No.**
**82175**




**CAUTION**: Read, understand and follow Safety Rules and Operating Instructions in this manual before using this product.

- Safety
- Operation
- Maintenance
- Español


**www.craftsman.com** **061906**

# TABLE OF CONTENTS

## ONE YEAR FULL WARRANTY

ONE YEAR FULL WARRANTY ON CRAFTSMAN AUTORANGING MULTIMETER

If this CRAFTSMAN AutoRanging MultiMeter fails to give complete satisfaction within one year from the date of purchase, RETURN IT TO THE NEAREST SEARS STORE OR OTHER CRAFTSMAN OUTLET IN THE UNITED STATES, and Sears will replace it, free of charge.

If this CRAFTSMAN AutoRanging MultiMeter is used for commercial or rental purposes, this warranty applies for 90 days from the date of purchase.

This warranty gives you specific legal rights, and you may also have other rights which vary from state to state.

Sears, Roebuck and Co., Dept. 817WA, Hoffman Estates, IL 60179

**For Customer Assistance Call 9am-5pm (EST)**
**Monday through Friday 1-888-326-1006**

**WARNING: USE EXTREME CAUTION IN THE USE OF THIS DEVICE.** Improper use of this device can result in injury or death. Follow all safeguards suggested in this manual in addition to the normal safety precautions used in working with electrical circuits. DO NOT service this device if you are not qualified to do so.

## SAFETY INSTRUCTIONS

This meter has been designed for safe use but must be operated with caution. The rules listed below must be carefully followed for safe operation.

1. **NEVER** apply voltage or current to the meter that exceeds the specified maximum:

| Input Limits | |
|---|---|
| Function | Maximum Input |
| mV DC/AC | 250V DC or AC |
| V DC/AC | 1000V DC or 750V AC |
| Clamp Adapter Input | 250V DC or AC |
| mA DC/AC | 400mA DC/AC |
| 20A DC/AC | 20A DC/AC<br>(30 seconds max every 15 minutes) |
| Resistance, Capacitance, Diode test, Continuity, Temperature | 250V DC/AC |

2. **USE EXTREME CAUTION** when working with high voltages.

3. **DO NOT** measure voltage if the voltage on the "COM" input jack exceeds 1000V above earth ground.

4. **NEVER** connect the meter leads across a voltage source while the function switch is in the current, resistance, or diode mode. Doing so can damage the meter.

5. **ALWAYS** discharge filter capacitors in power supplies and disconnect the power when making resistance or diode tests.

6. **ALWAYS** turn off the power and disconnect the test leads before opening the doors to replace the fuse or battery.

7. **NEVER** operate the meter unless the back cover and the battery and fuse doors are in place and fastened securely.

## SAFETY SYMBOLS

This symbol adjacent to another symbol, terminal or operating device indicates that the operator must refer to an explanation in the Operating Instructions to avoid personal injury or damage to the meter.

**WARNING**

This **WARNING** symbol indicates a potentially hazardous situation, which if not avoided, could result in death or serious injury.

**CAUTION**

This **CAUTION** symbol indicates a potentially hazardous situation, which if not avoided, may result damage to the product.

**MAX 1000V**

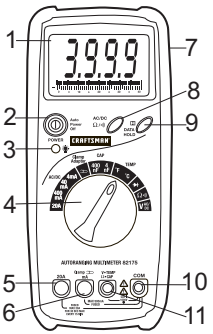
This symbol advises the user that the terminal(s) so marked must not be connected to a circuit point at which the voltage with respect to earth ground exceeds (in this case) 1000 VAC or VDC.

This symbol adjacent to one or more terminals identifies them as being associated with ranges that may, in normal use, be subjected to particularly hazardous voltages. For maximum safety, the meter and its test leads should not be handled when these terminals are energized.

This symbol indicates that a device is protected throughout by double insulation or reinforced insulation.

## CONTROLS AND JACKS

1. 4000 count Liquid Crystal Display
2. Power switch
3. Display backlight switch
4. Rotary function switch
5. 20A current input jack
6. mA and Clamp Adapter input jack
7. Meter holster
8. AC/DC and Resistance/Continuity push button
9. Data Hold push-button
10. COM negative input jack
11. Positive input jack




## SYMBOLS AND ANNUNCIATORS

| | |
|---|---|
| | Audible Continuity |
| | Low Battery |
| | Diode |
| | Data Hold |
| Ω | Resistance (ohms) |
| | Clamp Adaptor |
| ~ | AC (Alternating voltage or current) |

# SPECIFICATIONS

| Function | Range | Resolution | Accuracy |
|---|---|---|---|
| DC Voltage (V DC) *Clamp adapter input | 400mV* | 0.1mV | ±(2.0% reading + 3 digits) |
| | 4V | 1mV | ±(0.7% reading + 3 digits) |
| | 40V | 10mV | |
| | 400V | 100mV | |
| | 1000V | 1V | ±(1.0% reading + 3 digits) |
| AC Voltage (V AC) (40 - 400Hz) *Clamp adapter input | 400mV* | 0.1mV | ±(2.0% reading + 8 digits) |
| | 4V | 1mV | ±(1.2% reading + 5 digits) |
| | 40V | 10mV | |
| | 400V | 100mV | |
| | 750V | 1V | ±(1.5% reading + 5 digits |
| DC Current (A DC) | 4mA | 1μA | ±(1.2% reading + 5 digits) |
| | 40mA | 10μA | |
| | 400mA | 100μA | |
| | 20A | 10mA | ±(2.0% reading + 8 digits) |
| AC Current (A AC) (40 - 400Hz) | 4mA | 1μA | ±(1.5% reading + 8 digits) |
| | 40mA | 10μA | |
| | 400mA | 100μA | |
| | 20A | 10mA | ±(3.0% reading + 8 digits) |
| Resistance | 400Ω | 0.1Ω | ±(1.2% reading + 3 digits) |
| | 4kΩ | 1Ω | |
| | 40kΩ | 10Ω | |
| | 400kΩ | 100Ω | |
| | 4MΩ | 1kΩ | |
| | 40MΩ | 10kΩ | ±(3.0% reading + 5 digits) |
| Capacitance | 4nF | 1pF | ±(4.0% reading + 3 digits) |
| | 400nF | 100pF | |
| Temperature | 0 to 50°F | 1°F or C | ±(5.0% reading + 4 digits) |
| | 50 to 750°F | | ±(3.0% reading + 3 digits) |
| | 750 to 1800°F | | ±(3.0% reading + 5 digits) |
| | -20 to 0°C | | ±(5.0% reading + 4 digits) |
| | 0 to 400°C | | ±(3.0% reading + 3 digits) |
| | 400 to 1000°C | | ±(3.0% reading + 5 digits) |

**NOTE:** Accuracy specifications consist of two elements:

- (% reading) – This is the accuracy of the measurement circuit.
- (+ digits) – This is the accuracy of the analog to digital converter.

**NOTE:** Accuracy is stated at 64°F to 82°F (18°C to 28°C) and less than 75% RH ambient MultiMeter conditions.

## SPECIFICATIONS

| | |
|---|---|
| **Diode Test** | Test current of approximately 1mA, open circuit voltage 2.8V DC typical |
| **Continuity Check** | Audible signal will sound if the resistance is less than 40Ω approx;<br>Open circuit: voltage 2.8V DC typical |
| **Temperature sensor** | Type K temperature probe (sold separately) |
| **Autorange** | For voltage and resistance |
| **Input Impedance** | 10MΩ (VDC and VAC)<br>100kΩ for millivolt range |
| **Display** | 4000 count LCD with bargraph indication |
| **Overrange indication** | “O.L” is displayed |
| **Auto Power Off** | Meter automatically shuts down after 15 minutes of inactivity |
| **Polarity** | Automatic polarity;<br>No indication for positive polarity;<br>Minus (-) sign for negative polarity. |
| **Measurement Rate** | 0.4 seconds, nominal |
| **Low Battery Indication** | **“ ⊟+ ”** is displayed if battery voltage drops below operating voltage |
| **Battery** | Requires one 9V NEDA 1604 or 6F22 battery (sold separately) |
| **Fuses** | mA range, 500mA/250V fast blow ceramic<br>20A range, 20A/250V fast blow |
| **Operating Temperature** | 32°F to 104°F (0°C to 40°C) |
| **Storage Temperature** | 10°F to 122°F (-10°C to 50°C) |
| **Relative Humidity** | <70% operating |
| **Operating Altitude** | 2000 meters (7000ft.) maximum. |
| **Weight** | 12.34oz. (350g). |
| **Size** | 7.52” x 3.23” x 1.42”<br>(191mm x 82mm x 36mm) |
| **Safety** | For indoor use and in accordance with Overvoltage Category II, Pollution Degree 2. Category II includes local level, appliance, portable equipment, etc., with transient overvoltages less than Overvoltage Category III. |

# OPERATING INSTRUCTIONS

**WARNING**: Risk of electrocution. High-voltage circuits, both AC and DC, are very dangerous and should be measured with great care.

1. ALWAYS press the POWER switch to the OFF position when the meter is not in use.
2. If “O.L” appears in the display during a measurement, the value exceeds the range you have selected. Change to a higher range.

**NOTE**: On some low AC and DC voltage ranges, with the test leads not connected to a device, the display may show a random, changing reading. This is normal and is caused by the high-input sensitivity. The reading will stabilize and give a proper measurement when connected to a circuit.

## DATA HOLD

The Data Hold function allows the meter to "freeze" measurements.

1. Press the DATA HOLD (H) button to “freeze” the reading on the indicator. The indicator “ H “ will be appear in the display.
2. Press the DATA HOLD button to return to normal operation.

## BACKLIGHT

The backlight function illuminates the display to improve readability in dimly lighted areas.

1. Press the “ 💡 “ button (located directly beneath the power button) to turn the backlight on.
2. In order to conserve power, the backlight will automatically turn off after 5 seconds.

## BARGRAPH DISPLAY

The LCD display indicates measurement data numerically and in a bargraph. The bargraph is shown on the bottom of the LCD window.

## AUTO RANGE

The meter automatically finds the proper range for voltage and resistance measurements. All other functions are manually ranged.

## INPUT SHUTTERS

The yellow input shutters either block or open the input jacks to ensure proper test lead connections for the measurement function selected.

## DC VOLTAGE MEASUREMENTS

**CAUTION:** Do not measure DC voltages if a motor on the circuit is being switched ON or OFF. Large voltage surges may occur that can damage the meter.


1. Set the rotary function switch to **V** position.
2. Press the AC/DC button to select DC.
3. Insert the black test lead banana plug into the negative (COM) jack.
   Insert the red test lead banana plug into the positive (V) jack.
4. Touch the black test probe tip to the negative side of the circuit. Touch the red test probe tip to the positive side of the circuit.
5. Read the voltage in the display. The display will indicate the proper decimal point, range, and value. If the polarity is reversed, the display will show (-) minus before the value.

## AC VOLTAGE MEASUREMENTS

**WARNING:** Risk of Electrocution. The probe tips may not be long enough to contact the live parts inside some 240V outlets for appliances because the contacts are recessed deep in the outlets. As a result, the reading may show 0 volts when the outlet actually has voltage on it. Make sure the probe tips are touching the metal contacts inside the outlet before assuming that no voltage is present.

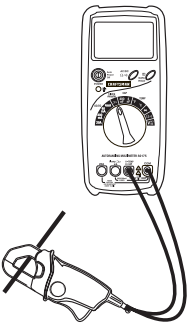
**CAUTION:** Do not measure AC voltages if a motor on the circuit is being switched ON or OFF. Large voltage surges may occur that can damage the meter.

1. Set the rotary function switch to the **V** position.
2. Press the AC/DC button to select AC.
3. Insert the black test lead banana plug into the negative jack.
   Insert red test lead banana plug into the positive (V) jack.
4. Touch the black test probe tip to the negative side of the circuit.
   Touch the red test probe tip to the positive side of the circuit.
5. Read the voltage in the display. The display will indicate the proper decimal point and value.

## CLAMP ADAPTOR MEASUREMENTS

The Clamp Adaptor input on the meter accepts the output from a clamp adapter such as the Craftsman 82796. The meter displays the current value measured by the adaptor. The clamp adaptor must output 1mV (AC or DC) per amp.

1. Set the rotary function switch to the **Clamp Adapter** position.
2. Select AC or DC using the AC/DC button.
3. Insert the black banana plug from the adaptor into the meter's negative (COM) jack. Insert the red banana plug from the clamp adapter into the meter's clamp input jack .
4. Clamp the adaptor jaw around the conductor under test.
5. Read the current in the display.



## DC CURRENT MEASUREMENTS

**CAUTION:** Do not make current measurements on the 20A scale for longer than 30 seconds. Exceeding 30 seconds may cause damage to the meter and/or the test leads.


1. Insert the black test lead banana plug into the negative (COM) jack.
2. For current measurements up to 400mA, set the rotary function switch to the highest mA position and insert the red test lead banana plug into the (mA) jack.
3. For current measurements up to 20A, set the rotary function switch to the 20A range and insert the red test lead banana plug into the (20A) jack.
4. Use the AC/DC pushbutton to select DC.
5. Remove power from the circuit under test, then open up the circuit at the point where you wish to measure current.
6. Touch the black test probe tip to the negative side of the circuit.
   Touch the red test probe tip to the positive side of the circuit.
7. Apply power to the circuit.
8. Read the current in the display. For mA measurements, reset the function switch to successively lower mA positions to obtain a higher resolution reading. The display will indicate the proper decimal point and value.

## AC CURRENT MEASUREMENTS

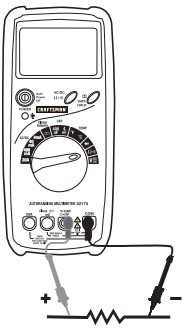
**WARNING:** To avoid electric shock, do not measure AC current on any circuit whose voltage exceeds 250V AC.

**CAUTION:** Do not make current measurements on the 20A scale for longer than 30 seconds. Exceeding 30 seconds may cause damage to the meter and/or the test leads.

1. Insert the black test lead banana plug into the negative (COM) jack.
2. For current measurements up to 400mA set the rotary function switch to the highest mA position and insert the red test lead banana plug into the (mA) jack.
3. For current measurements up to 20A, set the function switch to the 20A range and insert the red test lead banana plug into the (20A) jack.
4. Use the AC/DC pushbutton to select AC.
5. Remove power from the circuit under test, then open up the circuit at the point where you wish to measure current.
6. Touch the black test probe tip to the negative side of the circuit. Touch the red test probe tip to the positive side of the circuit.
7. Apply power to the circuit.
8. Read the voltage in the display. For mA measurements, reset the function switch to successively lower mA positions to obtain a higher resolution reading. The display will indicate the proper decimal point and value.

## RESISTANCE MEASUREMENTS

**WARNING:** To avoid electric shock, disconnect power to the unit under test and discharge all capacitors before taking any resistance measurements. Remove the batteries and unplug the line cords.




1. Set the rotary function switch to the **Ω** position.
2. Use the Ω •))) pushbutton to select resistance (Ω).
3. Insert the black test lead banana plug into the negative (COM) jack
   Insert the red test lead banana plug into the positive Ω jack.
4. Touch the test probe tips across the circuit or part under test. It is best to disconnect one side of the part under test so the rest of the circuit will not interfere with the resistance reading.
5. Read the resistance in the display. The display will indicate the proper decimal point and value.

## CONTINUITY CHECK

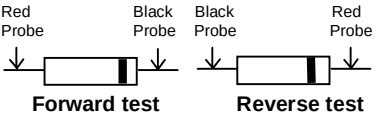
**WARNING:** To avoid electric shock, never measure continuity on circuits or wires that have voltage on them.

1. Set the rotary function switch to the •))) position.
2. Use the Ω •))) pushbutton to select continuity.
3. Insert the black lead banana plug into the negative (COM) jack
   Insert the red test lead banana plug into the positive (Ω) jack.
4. Touch the test probe tips to the circuit or wire you wish to check.
5. If the resistance is less than approximately 40Ω, the audible signal will sound. If the circuit is open, the display will indicate “1”.

## DIODE TEST

**WARNING:** To avoid electric shock, do not test any diode that has voltage on it.

1. Set the rotary function switch to the ⊳| position.
2. Insert black test lead banana plug into the negative (COM) jack. Insert the red test lead banana plug into the positive (V) jack.
3. Touch the test probe tips to the diode or semiconductor junction you wish to test. Note the meter reading.
4. Swap the probe polarity as shown by switching probe position. Note this reading.



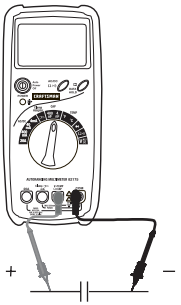

5. The diode or junction can be evaluated as follows:
   A. If the Forward test displays a value and the Reverse test displays “O.L”, the diode is good.
   B. If both tests display “O.L”, the device is open.
   C. If both tests are very small or 0, the device is shorted.

**NOTE:** The value indicated in the display during the diode check is the forward voltage.

## CAPACITANCE MEASUREMENTS

**WARNING:** To avoid electric shock, disconnect power to the unit under test and discharge all capacitors before taking any capacitance measurements. Remove the batteries and unplug the line cords.

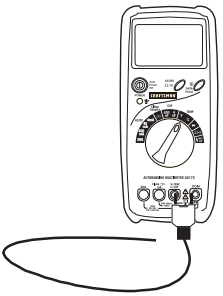
1. Set the rotary function switch to the **4nf** or **400nF** position.
2. Insert the black test lead banana plug into the negative (COM) jack.
   Insert the red test lead banana plug into the positive (CAP) jack.
3. Touch the test leads to the capacitor to be tested.
4. Read the capacitance in the display. The display will indicate the proper decimal point and value.



## TEMPERATURE MEASUREMENTS

**WARNING:** To avoid electric shock, disconnect both test probes from any source of voltage before making a temperature measurement.

1. Set the function switch to a **TEMP °F** or **TEMP °C** position.
2. Insert the Temperature Probe into the negative (COM) and the positive (TEMP) jacks, making sure to observe the correct polarity.
3. Touch the Temperature Probe head to the part whose temperature you wish to measure. Keep the probe touching the part under test until the reading stabilizes (about 30 seconds).
4. Read the temperature in the display. The digital reading will indicate the proper decimal point and value.




**WARNING:** To avoid electric shock, be sure the thermocouple has been removed before changing to another measurement function.

## AUTO POWER OFF

The meter will automatically beep and shut off after 15 minutes of inactivity. This feature prevents unnecessary battery drain if the meter is accidentally left on. Press the POWER switch to OFF and then press to ON turn power on.

# MAINTENANCE

**WARNING**: To avoid electric shock, disconnect the test leads from any source of voltage before removing the back cover or the battery or fuse doors.

**WARNING:** To avoid electric shock, do not operate your meter until the battery and fuse doors are in place and fastened securely.

This MultiMeter is designed to provide years of dependable service, if the following care instructions are performed:

1. **KEEP THE METER DRY**. If it gets wet, wipe it off.
2. **USE AND STORE THE METER IN NORMAL TEMPERATURES.** Temperature extremes can shorten the life of the electronic parts and distort or melt plastic parts.
3. **HANDLE THE METER GENTLY AND CAREFULLY.** Dropping it can damage the electronic parts or the case.
4. **KEEP THE METER CLEAN.** Wipe the case occasionally with a damp cloth. DO NOT use chemicals, cleaning solvents, or detergents.
5. **ALWAYS USE A FRESH BATTERY OF THE RECOMMENDED SIZE AND TYPE.** Remove old or weak battery so it does not leak and damage the unit.
6. **IF THE METER IS TO BE STORED FOR A LONG PERIOD OF TIME**, the battery should be removed to prevent meter damage.

## REPLACING THE BATTERY

**WARNING**: To avoid electric shock, disconnect the test leads from any source of voltage before removing the battery door.

1. When the battery drops below the operating voltage, "[-+]" will appear in the LCD display. The battery should be replaced.
2. Disconnect the test leads from the meter.
3. Open the battery door by lifting the rear stand and removing the two screws using a Phillips head screwdriver.
4. Insert the battery into battery holder, observing the correct polarity.
5. Put the battery door back in place. Secure with the two screws.
6. Dispose of the old battery properly.

**WARNING**: To avoid electric shock, disconnect the test leads from any source of voltage before removing the battery door.

**WARNING:** To avoid electric shock, do not operate the meter until the battery door is in place and fastened securely.

**NOTE**: If your meter does not work properly, check the fuses and battery to make sure that it they are still good and that they are properly inserted.

**REPLACING THE FUSES**

**WARNING**: To avoid electric shock, disconnect the test leads from any source of voltage before removing the fuse door.

1. Disconnect the test leads from the meter and any item under test.
2. Remove the protective holster and then remove the three screws and lift off the back cover.
3. Remove the old fuse from its holder by gently pulling it out.
4. Install the new fuse into the holder.
5. Always use a fuse of the proper size and value.
6. Replace the rear cover and secure it with the three screws.

**WARNING**: To avoid electric shock, do not operate your meter until the fuse door is in place and fastened securely.

**UL LISTED**

The UL mark does not indicate that this product has been evaluated for the accuracy of its readings.

## TROUBLESHOOTING

There may be times when your meter does not operate properly. Here are some common problems that you may have and some easy solutions to them.

**Meter Does Not Operate:**

1. Always read all the instructions in this manual before use.
2. Check to be sure the battery is properly installed.
3. Check to be sure the battery is good.
4. If the battery is good and the meter still does not operate, check to be sure that both ends of the fuse are properly installed.

**If You Do Not Understand How the Meter Works:**

1. Purchase the instructional book *Multitesters and Their Use for Electrical Testing* (Item No. 82303) at your local Sears store.
2. Call our Customer Service Line **1-888-326-1006**.

## SERVICE AND PARTS

| Item Number | Description |
|---|---|
| 82375 | Fuse kit |
| 93894 | 9V Battery |
| 82398 | Set of black and red Test Leads |
| 82175-DB | Replacement battery door |
| 82175-CS | Rear cover screws |
| TP872 | Type k temperature probe |

For replacement parts shipped directly to your home
Call Monday through Friday, 9 AM – 5 PM Eastern Time
**1-888-326-1006**

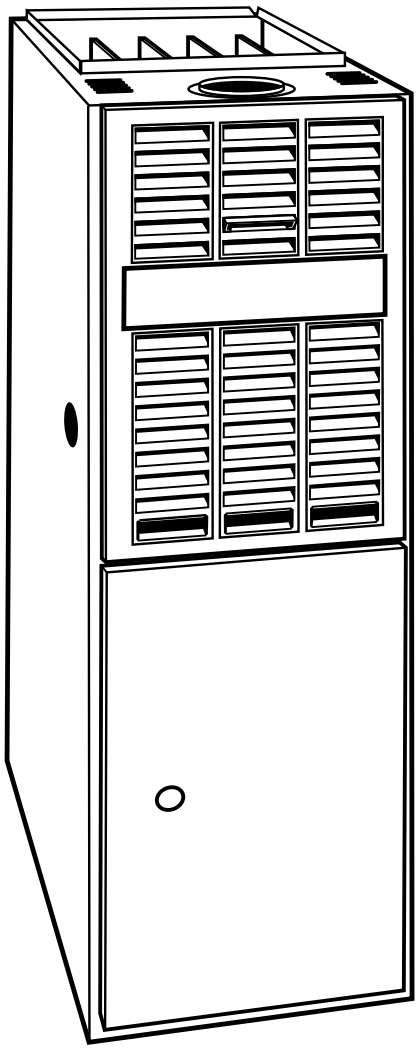
# Appendix C

Carrier®
HEATING & COOLING

# Product Data

# 58PAV
## Energy-Efficient Induced-Combustion Upflow Furnace

Input Capacities:
45,000 thru 155,000 Btuh



### 80% AFUE At Budget Price

Carrier provides an 80 percent Annual Fuel Utilization Efficiency (AFUE) gas furnace for the budget conscious consumer and builder. The 58PAV offers the same high quality you demand and receive from Carrier.

The cabinet is constructed from a specially selected galvanized steel. There is also double protection for the cabinet. First, a galvanized steel substrate provides resistance to rusting. Then the cabinet is constructed of prepainted steel — the same high-quality finish found on refrigerators and dishwashers.

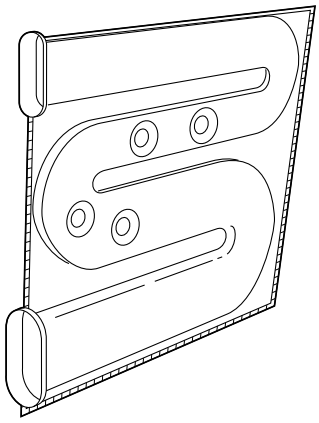
The 58PAV offers a hot surface ignition system which provides a superior and more reliable ignition system than older spark relight systems.

The heat exchangers are constructed of aluminized steel and covered by a 20-year Limited Warranty. They are Carrier's patented Super-S heat exchangers that improve heat transfer and enable downsizing of this furnace to only 40-in. tall.

A patented draft safeguard will stop furnace operation if the vent system is not operating properly.

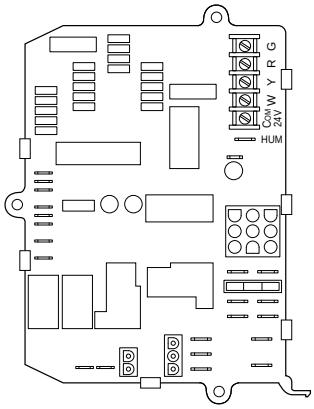
To improve the sound level, we have incorporated a soft mount inducer assembly and a slow opening gas valve.

The control board is the brain of this induced-combustion gas furnace. It offers a unique self-test feature that checks all the major functions of the furnace within 1 minute. The control board also features a 3-amp fuse that protects the transformer and control board. Another feature on the control board is an LED status indicator light to ensure top furnace performance.



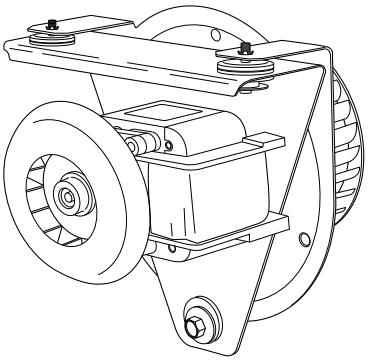
A89217

**HEAT EXCHANGER**




A95246

**CONTROL BOARD**



A92074

**INDUCER BLOWER**

# Model number nomenclature

**58PAV** — **045** — **160** — **08**

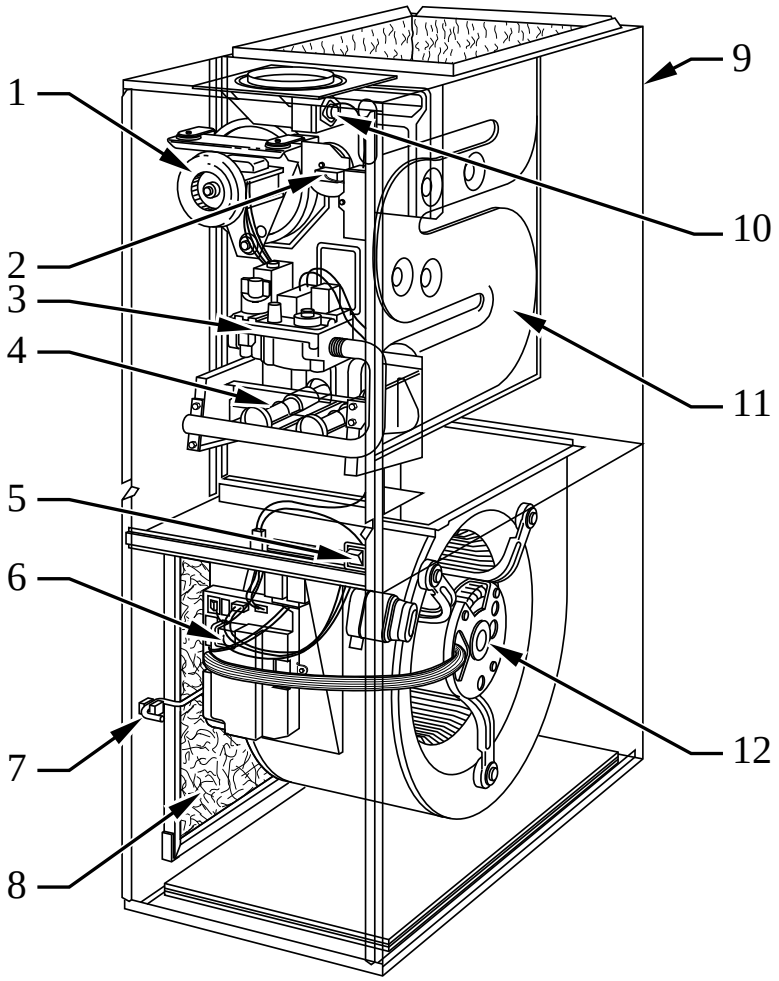
**Induced-Combustion Upflow Furnace**

**Input Capacity**

| | |
|---|---|
| 045 — 44,000 Btuh | 111 — 110,000 Btuh |
| 070 — 66,000 Btuh | 135 — 132,000 Btuh |
| 090 — 88,000 Btuh | 155 — 154,000 Btuh |

**Series Number**

**Cooling Size (Airflow)**
(400 CFM per 12,000 Btuh)
08 — 800 CFM
12 — 1200 CFM
14 — 1400 CFM
16 — 1600 CFM
20 — 2000 CFM

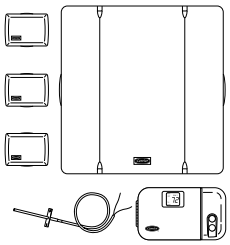



A98062

**NOTE:** The 58PAV Furnaces are for use with natural gas. These furnaces can be field-converted for propane gas with a factory-authorized and listed accessory conversion kit.
**NOTE:** Control location and actual control may be different than shown above.

1. Inducer Assembly
2. Pressure Switch
3. Gas Control Valve
4. Burner Assembly
5. Blower Door Safety Switch
6. Control Box
7. Air Filter Retainer
8. Air Filter
9. Wrap-Around Casing
10. Draft Safeguard Switch
11. Heat Exchanger
12. Blower and Blower Motor

# Carrier accessories*

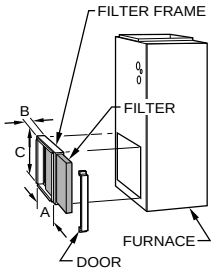


A97432

### CONTROLS: THERMOSTATS AND ZONING

Available in programmable and non-programmable models, Carrier thermostats maintain a constant, comfortable temperature level in the home.

For the ultimate in home comfort, Carrier's 2, 4, and 8-zone systems allow temperature control of individual "zones" of the home. This is accomplished through a series of electronic dampers and remote room sensors. The 4-zone system is shown.

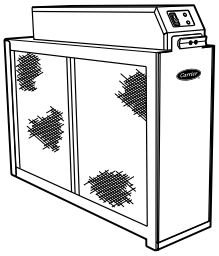



A93068

### SIDE FILTER RACK

Custom made filter rack for easy connection when a return plenum already exists. Provides easy access for cleaning filter.

| | |
|---|---|
| **A** | 23-1/8 in. |
| **B** | 2-3/8 in. |
| **C** | 14-1/2 in. |

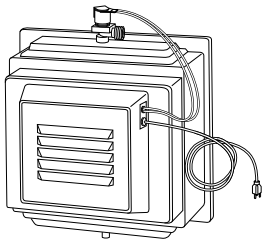


A97380

### MECHANICAL OR ELECTRONIC AIR CLEANER

Cleans the air of smoke, dirt, and many pollens commonly found. Saves decorating and cleaning expenses by keeping carpets, furniture, and drapes cleaner.

Electronic air cleaner is shown.



A91365

### MODEL 49FH HUMIDIFIER

By adding moisture to winter-dry air, a Carrier humidifier can often improve the comfort and keep furniture, rugs, and draperies in better condition. Moisturizing household air also helps to retain normal body heat and provides comfort at lower temperatures.

| UNIT SIZE | 045-08 & 12 | 070-08 & 12 | 090-14 & 16 | 111-12, 16 & 20 | 135-16 & 20 | 155-20 |
|---|---|---|---|---|---|---|
| **ELECTRONIC AIR CLEANER (EAC)** | Model AIRA | | | | | |
| **MECHANICAL AIR CLEANER** | Model 31MF | | | | | |
| **HUMIDIFIER** | Models 49BF, 49BG, 49FH, 49FP, and 49WS | | | | | |
| **HEAT RECOVERY VENTILATOR** | Models VA3B, VB5B, or VC5B | | | | | |
| **ENERGY RECOVERY VENTILATOR** | Model VL3A | | | | | |
| **THERMOSTAT — NON-PROGRAMMABLE** | For Use with 1-Speed Air Conditioner — TSTATCCNAC01-A<br>For Use with 2-Speed Air Conditioner — TSTATCCN2S01-A<br>For Use with 2-Speed Heat Pump — TSTATCCN2S01-A | | | | | |
| **THERMOSTAT — PROGRAMMABLE** | For Use with 1-Speed Air Conditioner — TSTATCCPAC01-A<br>For Use with 2-Speed Air Conditioner — TSTATCCP2S01-A<br>For Use with 1-Speed Heat Pump — TSTATCCPDF01-A<br>For Use with 2-Speed Heat Pump — TSTATCCP2S01-A or TSTATCCPDF01-A | | | | | |
| **THERMIDISTAT — PROGRAMMABLE THERMOSTAT with Humidity Control** | TSTATCCPRH01-A | | | | | |
| **ZONING — 2 ZONE** | ZONECC2KIT01, ZONEKIT2ZCAR | | | | | |
| **ZONING — 4 ZONE** | ZONECC4KIT01 | | | | | |
| **ZONING — 8 ZONE** | ZONECC8KIT01 | | | | | |
| **SIDE FILTER RACK (Accepts one 16 x 25 x 1 Filter)** | KGAFR0201ALL | | | | | |
| **TWINNING KIT†** | KGATW0401HSI | | | | | |
| **GAS CONVERSION KIT* — NATURAL-TO-PROPANE** | KGANP2001ALL | | | | | |
| **PROPANE-TO-NATURAL** | KGAPN1601ALL | | | | | |

* Factory-authorized and field installed. Gas conversion kits are A.G.A./C.G.A. recognized.
† 16 and 20 sizes only.

# Physical data

| UNIT SIZE | | 045 | | 070 | | 090 | | 111 | | | 135 | | 155 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 08 | 12 | 08 | 12 | 14 | 16 | 12 | 16 | 20 | 16 | 20 | 20 |
| OUTPUT CAPACITY (BTUH)† | Nonweatherized ICS | 35,000 | 35,000 | 53,000 | 53,000 | 71,000 | 71,000 | 89,000 | 89,000 | 89,000 | 107,000 | 107,000 | 124,000 |
| INPUT BTUH* | | 44,000 | 44,000 | 66,000 | 66,000 | 88,000 | 88,000 | 110,000 | 110,000 | 110,000 | 132,000 | 132,000 | 154,000 |
| SHIPPING WEIGHT (Lb) | | 114 | 116 | 124 | 126 | 140 | 144 | 150 | 156 | 172 | 168 | 182 | 192 |
| CERTIFIED TEMP RISE RANGE (8F) | | 25-55 | 15-45 | 40-70 | 30-60 | 40-70 | 30-60 | 55-85 | 45-75 | 25-55 | 50-80 | 40-70 | 50-80 |
| CERTIFIED EXT STATIC PRESSURE | Heating | 0.10 | 0.10 | 0.12 | 0.12 | 0.15 | 0.15 | 0.20 | 0.20 | 0.20 | 0.20 | 0.20 | 0.20 |
| | Cooling | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 |
| AIRFLOW CFM‡ | Heating | 855 | 1140 | 830 | 1140 | 1170 | 1445 | 1175 | 1410 | 1750 | 1645 | 1775 | 1770 |
| | Cooling | 930 | 1250 | 880 | 1195 | 1360 | 1740 | 1205 | 1575 | 2225 | 1620 | 2025 | 2055 |
| LIMIT CONTROL | | SPST | | | | | | | | | | | |
| HEATING BLOWER CONTROL | | Solid-State Time Operation | | | | | | | | | | | |
| BURNERS (Monoport) | | 2 | 2 | 3 | 3 | 4 | 4 | 5 | 5 | 5 | 6 | 6 | 7 |
| GAS CONNECTION SIZE | | 1/2-in. NPT | | | | | | | | | | | |
| GAS VALVE (Redundant) Manufacturer | | White-Rodgers | | | | | | | | | | | |
| Minimum Inlet Pressure (In. wc) | | 4.5 (Natural Gas) | | | | | | | | | | | |
| Maximum Inlet Pressure (In. wc) | | 13.6 (Natural Gas) | | | | | | | | | | | |
| IGNITION DEVICE | | Hot Surface | | | | | | | | | | | |

* Gas input ratings are certified for elevations to 2000 ft. For elevations above 2000 ft, reduce ratings 4 percent for each 1000 ft above sea level. Refer to National Fuel Gas Code Table F4 or furnace Installation Instructions. In Canada, derate the unit 10 percent for elevations 2000 ft to 4500 ft above sea level.
† Capacity in accordance with U.S. Government DOE test procedures.
‡ Air delivery above 1800 CFM requires that both sides, or a combination of 1 side and bottom, or bottom only, of the furnace be used for return air. A filter is required for each return-air supply.
ICS—Isolated Combustion System

# Performance data

| UNIT SIZE | | 045 | | 070 | | 090 | | 111 | | | 135 | | 155 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 08 | 12 | 08 | 12 | 14 | 16 | 12 | 16 | 20 | 16 | 20 | 20 |
| DIRECT-DRIVE MOTOR Hp (PSC) | | 1/5 | 1/3 | 1/5 | 1/3 | 1/3 | 1/2 | 1/3 | 1/2 | 3/4 | 1/2 | 3/4 | 3/4 |
| MOTOR FULL LOAD AMPS | | 2.9 | 5.8 | 2.9 | 5.8 | 5.8 | 7.9 | 5.8 | 7.9 | 11.1 | 7.9 | 11.1 | 11.1 |
| RPM (Nominal) — SPEEDS | | 1075-3 | 1075-4 | 1075-3 | 1075-4 | 1075-4 | 1075-4 | 1075-4 | 1075-4 | 1075-4 | 1075-4 | 1075-4 | 1075-4 |
| BLOWER WHEEL DIAMETER x WIDTHS (In.) | | 10 x 6 | 10 x 6 | 10 x 6 | 10 x 6 | 10 x 7 | 10 x 8 | 10 x 7 | 10 x 8 | 11 x 10 | 10 x 8 | 11 x 10 | 11 x 10 |
| WASHABLE 16 x 25 x 1-In. FILTER | Qty | 1 | 1 | 1 | 1 | 1 | — | 1 | — | — | — | — | — |
| WASHABLE 20 x 25 x 1-In. FILTER | Qty | — | — | — | — | — | 1 | — | 1 | — | 1 | — | — |
| WASHABLE 24 x 29 x 1-In. FILTER | Qty | — | — | — | — | — | — | — | — | 1 | — | 1 | 1 |

PSC—Permanent Split Capacitor

## ENERGY EFFICIENCY

| UNIT SIZE | | 045 | | 070 | | 090 | | 111 | | | 135 | | 155 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 08 | 12 | 08 | 12 | 14 | 16 | 12 | 16 | 20 | 16 | 20 | 20 |
| CAPACITY BTUH* | Nonweatherized ICS | 35,000 | 35,000 | 53,000 | 53,000 | 71,000 | 71,000 | 89,000 | 89,000 | 89,000 | 107,000 | 107,000 | 124,000 |
| AFUE%* | Nonweatherized ICS | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 | 80.0 |

* Capacity and AFUE in accordance with U.S. Government DOE test procedures.
ICS—Isolated Combustion System

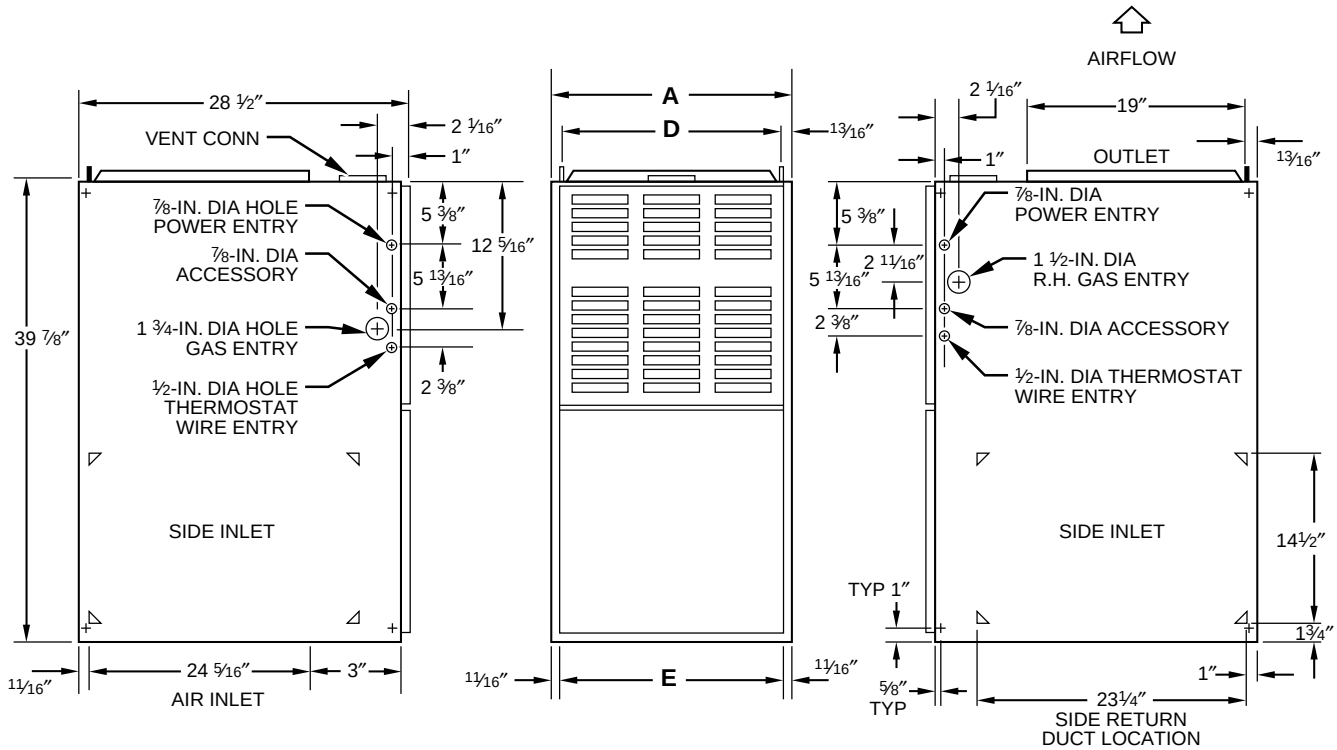
## AIR DELIVERY—CFM (With Filter)*

| UNIT SIZE | SPEED | EXTERNAL STATIC PRESSURE (In. wc) | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 |
| 045-08 | High | 1030 | 1005 | 970 | 925 | 880 | 815 | 745 | 615 |
| | Med-High | 855 | 830 | 800 | 765 | 720 | 670 | 595 | 485 |
| | Med-Low | 755 | 725 | 695 | 650 | 605 | 555 | 475 | 400 |
| 045-12 | High | 1490 | 1430 | 1385 | 1325 | 1250 | 1175 | 1085 | 975 |
| | Med-High | 1335 | 1305 | 1270 | 1230 | 1160 | 1090 | 1005 | 915 |
| | Med-Low | 1140 | 1130 | 1105 | 1075 | 1030 | 975 | 900 | 830 |
| | Low | 980 | 975 | 965 | 915 | 875 | 840 | 785 | 715 |
| 070-08 | High | 1040 | 1010 | 975 | 935 | 880 | 810 | 735 | 640 |
| | Med-High | 855 | 830 | 800 | 765 | 715 | 660 | 600 | 490 |
| | Med-Low | 745 | 715 | 690 | 650 | 605 | 550 | 475 | 385 |
| 070-12 | High | 1430 | 1380 | 1325 | 1265 | 1195 | 1125 | 1045 | 945 |
| | Med-High | 1310 | 1275 | 1235 | 1190 | 1135 | 1065 | 990 | 900 |
| | Med-Low | 1140 | 1130 | 1100 | 1065 | 1000 | 965 | 910 | 815 |
| | Low | 990 | 965 | 960 | 935 | 905 | 855 | 800 | 710 |
| 090-14 | High | 1570 | 1535 | 1480 | 1415 | 1360 | 1280 | 1185 | 1070 |
| | Med-High | 1370 | 1355 | 1330 | 1290 | 1240 | 1170 | 1080 | 955 |
| | Med-Low | 1170 | 1165 | 1150 | 1115 | 1085 | 1035 | 970 | 880 |
| | Low | 1010 | 1005 | 990 | 965 | 950 | 905 | 845 | 745 |
| 090-16 | High | 2010 | 1950 | 1875 | 1810 | 1740 | 1660 | 1550 | 1455 |
| | Med-High | 1675 | 1660 | 1625 | 1600 | 1545 | 1490 | 1395 | 1295 |
| | Med-Low | 1445 | 1430 | 1415 | 1400 | 1370 | 1325 | 1265 | 1170 |
| | Low | 1260 | 1260 | 1260 | 1250 | 1210 | 1180 | 1115 | 1030 |
| 111-12 | High | 1470 | 1415 | 1340 | 1270 | 1205 | 1115 | 985 | 845 |
| | Med-High | 1340 | 1305 | 1245 | 1185 | 1130 | 1045 | 915 | 780 |
| | Med-Low | 1185 | 1175 | 1140 | 1075 | 1030 | 950 | 835 | 705 |
| | Low | 1015 | 1010 | 980 | 955 | 910 | 840 | 725 | 600 |
| 111-16 | High | 1880 | 1815 | 1745 | 1690 | 1575 | 1500 | 1400 | 1265 |
| | Med-High | 1660 | 1615 | 1570 | 1505 | 1435 | 1355 | 1260 | 1170 |
| | Med-Low | 1455 | 1410 | 1375 | 1350 | 1290 | 1235 | 1145 | 985 |
| | Low | 1265 | 1265 | 1240 | 1210 | 1180 | 1110 | 995 | 855 |
| 111-20 | High | — | 2465 | 2385 | 2305 | 2225 | 2125 | 2020 | 1910 |
| | Med-High | 2125 | 2120 | 2075 | 2040 | 1985 | 1900 | 1825 | 1715 |
| | Med-Low | 1745 | 1750 | 1740 | 1720 | 1685 | 1635 | 1565 | 1500 |
| | Low | 1545 | 1545 | 1545 | 1520 | 1495 | 1450 | 1390 | 1330 |
| Both Sides or 1 Side & Bottom | High | 2475 | 2465 | 2425 | 2365 | 2305 | 2230 | 2145 | 2015 |
| | NO Filter | — | 2510 | 2465 | 2415 | 2355 | 2280 | 2190 | 2070 |
| 135-16 | High | 1900 | 1845 | 1780 | 1705 | 1620 | 1530 | 1445 | 1320 |
| | Med-High | 1695 | 1645 | 1580 | 1520 | 1460 | 1385 | 1280 | 1155 |
| | Med-Low | 1460 | 1415 | 1375 | 1340 | 1290 | 1205 | 1110 | — |
| | Low | 1275 | 1260 | 1245 | 1230 | 1180 | 1135 | — | — |
| 135-20 | High | — | 2260 | 2185 | 2085 | 2025 | 1935 | 1835 | 1730 |
| | Med-High | 2110 | 2070 | 2015 | 1960 | 1885 | 1810 | 1715 | 1620 |
| | Med-Low | 1790 | 1775 | 1755 | 1720 | 1670 | 1610 | 1535 | 1455 |
| | Low | 1495 | 1500 | 1515 | 1485 | 1450 | 1400 | 1345 | 1270 |
| 155-20 | High | — | 2300 | 2235 | 2155 | 2055 | 1980 | 1870 | 1775 |
| | Med-High | — | 2050 | 2005 | 1945 | 1880 | 1805 | 1720 | 1640 |
| | Med-Low | 1790 | 1770 | 1740 | 1710 | 1635 | 1580 | 1520 | 1440 |
| | Low | 1520 | 1525 | 1505 | 1485 | 1450 | 1410 | 1350 | 1290 |

* Air delivery above 1800 CFM requires that both sides, or a combination of 1 side and bottom, or bottom only of the furnace be used for return air. A filter is required for each return-air supply.

—Indicates unstable operating conditions.

# Dimensions




**NOTES:** 1. Two additional ⅞-in. dia holes are located in the top plate.
2. Minimum return-air opening at furnace:
   a. For 800 CFM–16-in. round or 14½ x 12-in. rectangle.
   b. For 1200 CFM–20-in. round or 14½ x 19½-in. rectangle.
   c. For 1600 CFM–22-in. round or 14½ x 23¼-in. rectangle.
   d. For airflow requirements above 1800 CFM, use both side inlets, a combination of 1 side inlet and the bottom, or the bottom only.

A98025

**DIMENSIONS (In.)**

| UNIT SIZE | A | D | E | VENT CONN* (Dia) |
|---|---|---|---|---|
| **045-08** | 14-3/16 | 12-9/16 | 11-11/16 | 4 |
| **045-12** | 14-3/16 | 12-9/16 | 11-11/16 | 4 |
| **070-08** | 14-3/16 | 12-9/16 | 11-11/16 | 4 |
| **070-12** | 14-3/16 | 12-9/16 | 11-11/16 | 4 |
| **090-14** | 17-1/2 | 15-7/8 | 16 | 4 |
| **090-16** | 21 | 19-3/8 | 18-1/2 | 4 |
| **111-12** | 17-1/2 | 15-7/8 | 16 | 4 |
| **111-16** | 21 | 19-3/8 | 18-1/2 | 4 |
| **111-20** | 24-1/2 | 22-7/8 | 22 | 4 |
| **135-16** | 21 | 19-3/8 | 18-1/2 | 5† |
| **135-20** | 24-1/2 | 22-7/8 | 22 | 5† |
| **155-20** | 24-1/2 | 22-7/8 | 22 | 5† |

* Refer to the furnace Installation Instructions for proper venting procedures.
† Oval collar

## MINIMUM INCHES CLEARANCE TO COMBUSTIBLE CONSTRUCTION

This forced air furnace is equipped for use with natural gas at altitudes 0 - 10,000 ft (0-3,050m).

An accessory kit, supplied by the manufacturer, shall be used to convert to propane gas use or may be required for some natural gas applications.

This furnace is for indoor installation in a building constructed on site.

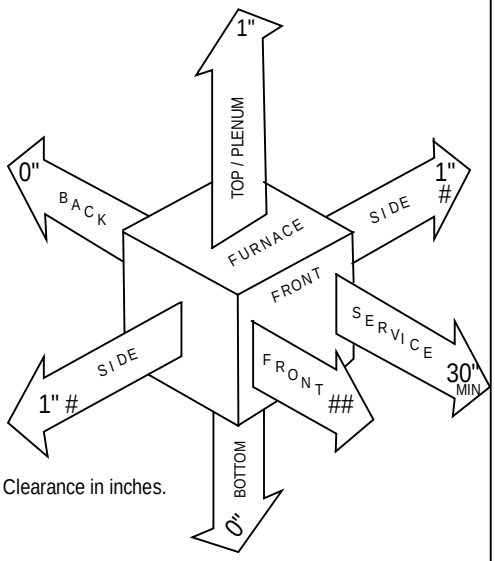
This furnace may be installed on combustible flooring in alcove or closet at minimum clearance from combustible material.

This furnace may be used with a Type B-1 Vent and may be vented in common with other gas-fired appliances.

# For furnaces wider than 14.25 inches (362mm) may be 0 inches.

## For single wall vent type 6 inches. For Type B-1 vent type 3 inches.

This furnace is approved for UPFLOW installations only.




Clearance in inches.

Vent Clearance to combustibles:

For Single Wall vents 6 inches (6 po).

For Type B-1 vent type 1 inch (1 po).

320325-101 REV. G (LIT)

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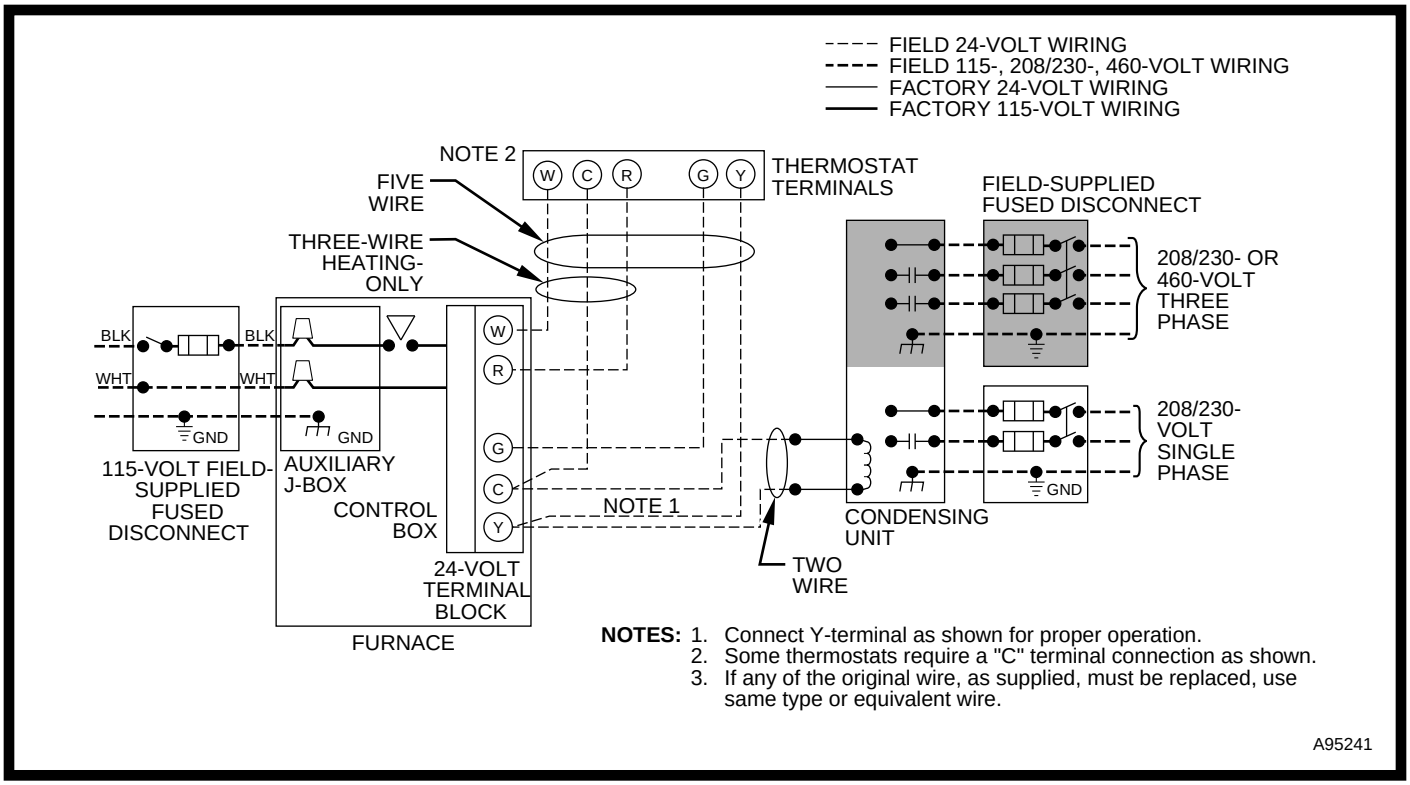
**MEETS DOE RESIDENTIAL CONSERVATION SERVICES PROGRAM STANDARDS.**

Before purchasing this appliance, read important energy cost and efficiency information available from your retailer.

# Typical wiring schematic

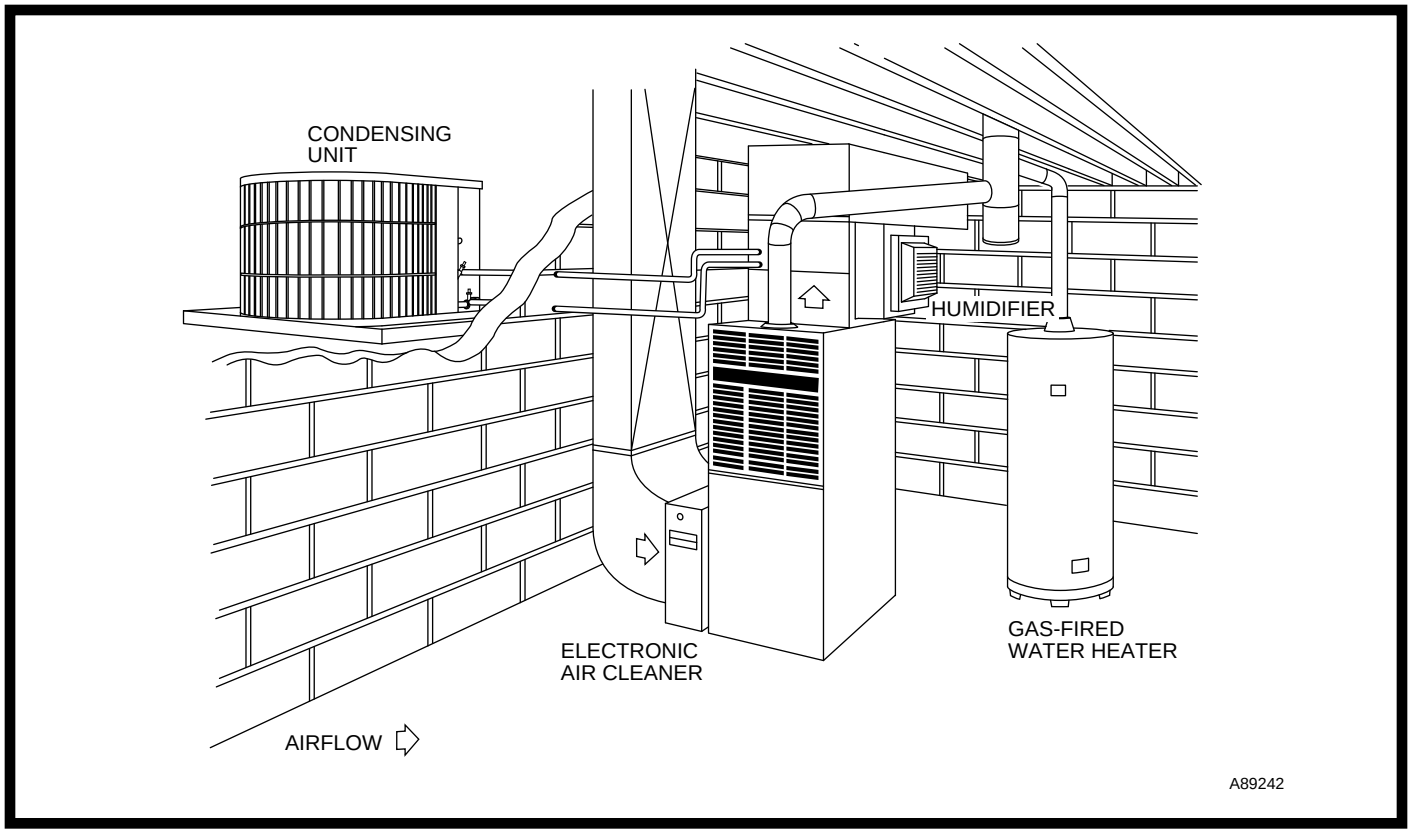



# Electrical data

| UNIT SIZE | | 045-08 | 045-12 | 070-08 | 070-12 | 090-14 | 090-16 | 111-12 | 111-16 | 111-20 | 135-16 | 135-20 | 155-20 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| UNIT VOLTS — HERTZ — PHASE | | 115 — 60 — 1 | | | | | | | | | | | |
| MINIMUM WIRE SIZE | | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 12 | 14 | 12 | 12 |
| MAXIMUM WIRE LENGTH (Ft)* | | 47 | 34 | 47 | 32 | 31 | 27 | 35 | 28 | 31 | 28 | 33 | 31 |
| MAXIMUM UNIT AMPS | | 6.0 | 8.3 | 5.9 | 8.7 | 9.0 | 10.4 | 8.0 | 10.1 | 14.4 | 10.1 | 13.3 | 14.0 |
| OPERATING VOLTAGE RANGE (Min—Max)† | | 104 — 127 | | | | | | | | | | | |
| MAX FUSE SIZE OR HACR-TYPE CKT BKR (Amps)‡ | | 15 | 15 | 15 | 15 | 15 | 15 | 15 | 15 | 20 | 15 | 20 | 20 |
| TRANSFORMER (24v) | | 40va | | | | | | | | | | | |
| EXTERNAL CONTROL POWER AVAILABLE | Heating | 12va | | | | | | | | | | | |
| | Cooling | 35va | | | | | | | | | | | |
| AIR CONDITIONING BLOWER RELAY | | Standard | | | | | | | | | | | |

* Length shown is as measured 1 way along wire path between unit and service panel for maximum 2 percent voltage drop.
† Permissible limits of the voltage range at which the unit will operate satisfactorily.
‡ Time-delay fuse is recommended.

# Typical installation



# SERVICE TRAINING

**Packaged Service Training** programs are an excellent way to increase your knowledge of the equipment discussed in this manual, including:

- Unit Familiarization
- Maintenance
- Installation Overview
- Operating Sequence

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