HEAT PUMP AND AIR CONDITIONING SYSTEM ANALYSIS BASED ON VARIABLE SPEED COMPRESSOR

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by
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ABSTRACT

Experiments were carried out to investigate the effect of ambient air temperatures on the heat pump performance using a variable speed compressor. Ambient air temperatures were varied from 40 to 60 °F to simulate different seasons. The compressor frequencies of 45 Hz, 50 Hz, 55 Hz, and 60 Hz were studied to determine the optimal frequency under various heating loads. The investigation was carried out by showing the compressor power input, heating output, and coefficient of performance for each case. Thermal cycle analysis along with the heat exchanger theory was used to analyze the system energy balance, heat transfer rates, p-h diagrams, and coefficient of performance. The overall heat transfer coefficients were also determined for both the evaporator and the condenser. Only the capillary tube was used to regulate the refrigerant flow rate.

The variable speed compressor system used in this study will help save energy when compared with the traditional steady speed system. The variable speed compressor system will hopefully provide a more comfortable and steady indoor temperature than the traditional system, which is controlled by only an on-off switch.

The speed controlled compressor system proposed we believe will help saving more energy than traditional steady speed system. The variable speed compressor system will hopefully provide a more comfort and steady indoor temperature than the traditional system which is controlled by one switch. It is believed that the variable speed compressor system may allow the indoor temperature air to be steady-going and prevent the switch working frequently.

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NOMENCLATURE

Symbol	Description	Units
T_{air}	Room temperature	ºF
V_c	Condenser fan speed	ft/min
V_e	Evaporator fan speed	ft/min
P_1	Compressor discharge pressure	psia
T_1	Compressor discharge temperature	ºF
P_2	Condenser inlet pressure	psia
T_2	Condenser inlet temperature	ºF
P_2'	Compressor outlet pressure	psia
T_2'	Compressor outlet temperature	ºF
P_3	Condenser outlet pressure	psia
T_3	Condenser outlet temperature	ºF
h_1	Compressor discharge enthalpy	Btu/lb
h_2	Condenser inlet enthalpy	Btu/lb
h_3	Condenser outlet enthalpy	Btu/lb
COP	Coefficient performance	Dimensionless
ρ_{air}	Density of air	kg/m^3
L	Length	m
Q	Flow rate	m^3/s
R	Radius	m

Re	Reynolds number	Dimensionless
Re_p	Reynolds number of particle	Dimensionless
$T_{e,i}$	Evaporator air inlet temperature	ºF
$T_{e,o}$	Evaporator air outlet temperature	ºF
$T_{c,i}$	Condenser air inlet temperature	ºF
$T_{c,o}$	Condenser air outlet temperature	ºF
V	Velocity of fluid	m/s
μ	Dynamic viscosity	kg/m·s
λ	Molecular mean free path	m
ρ	Density of fluid	kg/m^3

CHAPTER 1

INTRODUCTION

According to the US Department of Energy approximately 56% of an average home's energy usage is for its heating and air conditioning (HVAC) system. The following chart displays household energy consumption percentages, provided by the US Department of Energy. Figure 1.1 shows the energy distribution of the home energy consumption (LEGRE INC 2006).

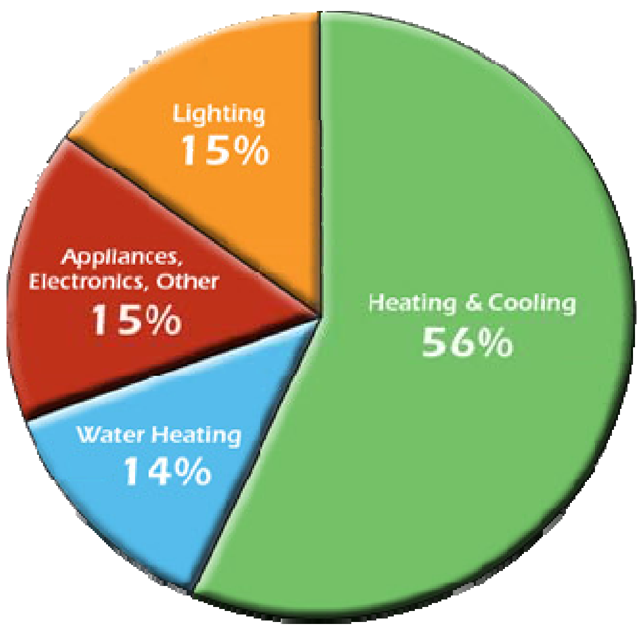


Fig 1.1 Household energy consumption percentages (LEGRE INC 2006)

This chapter presents the background information of heat pump system, current problems of heat pump, and objectives related to this thesis. Explanation is given of the advantages of high efficiency of the heat pump system through different ways which are air source heat pump, ground source heat pump and water source heat pump. The purpose of this study is proposed and the specific goals and scope of this research are defined.

1.1 Air Source Heat Pump

An air-source heat pump can provide efficient heating and cooling for your home, especially if you live in a warm climate. When properly installed, an air-source heat pump can deliver one-and-a-half to three times more heat energy to a home than the electrical energy it consumes. This is possible because a heat pump moves heat rather than converting it from a fuel, like in combustion heating systems (EREN 2001).

Although air-source heat pumps can be used in nearly all parts of the United States, they do not generally perform well over extended periods of sub-freezing temperatures. In regions with sub-freezing winter temperatures, it may not be cost effective to meet all your heating needs with a standard air-source heat pump (EREN 2001).

However, new systems with gas heating as a backup are able to overcome this problem. There is also a “Cold Climate Heat Pump” which shows promise, but is currently facing manufacturing problems. In addition, a version called the “Reverse Cycle Chiller” claims to be able to operate efficiently at below-freezing temperatures.

A heat pump is a machine or device that moves heat from one location to another location (the ‘sink’ or ‘heat sink’) using mechanical work. Most heat pump technology moves heat from a low temperature heat source to a higher temperature heat sink. Common examples are food refrigerators and freezers, air conditioners, and reversible-cycle heat pumps for providing thermal comfort (ASHRAE, Inc 2004).

Heat pumps can be thought of as a heat engine which is operating in reverse. One common type of heat pump works by exploiting the physical properties of an evaporating and condensing fluid known as a refrigerant. In heating, ventilation, and air conditioning (HVAC) applications, a heat pump normally refers to a vapor-compression refrigeration device that includes a reversing valve and optimized heat exchangers so that the direction of heat flow may be reversed. Most commonly, heat pumps draw heat from the air or from the ground. Some air-source heat pumps do not work as well when temperatures fall below around −5 °C (23 °F).

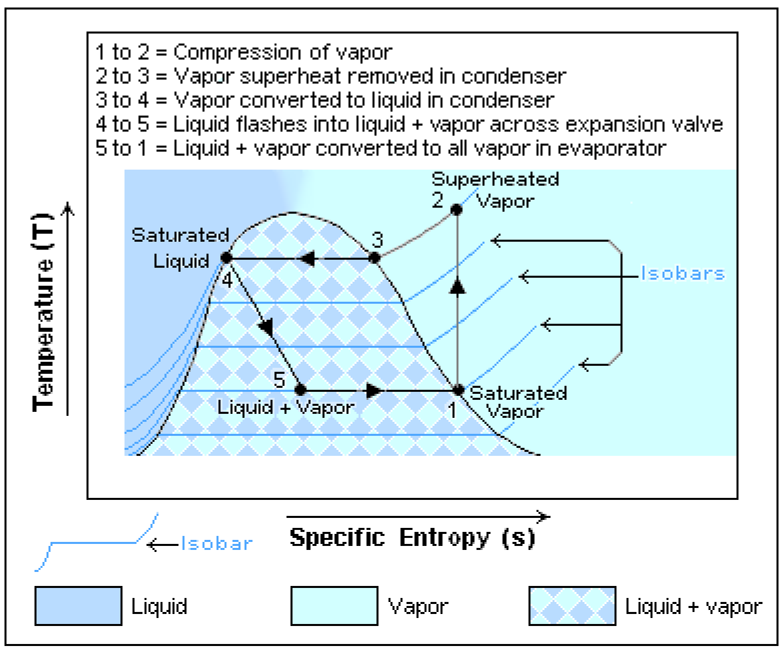


Fig 1.2 Temperature–entropy diagram

The thermodynamics of the vapor compression cycle can be analyzed on a temperature versus entropy diagram as depicted in Figure 1.2. At point 1 in the diagram, the circulating refrigerant enters the compressor as a saturated vapor. From point 1 to point 2, the vapor is isentropically compressed (i.e., compressed at constant entropy) and exits the compressor as a superheated vapor.

From point 2 to point 3, the superheated vapor travels through part of the condenser which removes the superheat by cooling the vapor. Between point 3 and point 4, the vapor travels through the remainder of the condenser and is condensed into a saturated liquid. The condensation process occurs at essentially constant pressure.

Between points 4 and 5, the saturated liquid refrigerant passes through the expansion valve and undergoes an abrupt decrease of pressure.

Between points 5 and 1, the cold and partially vaporized refrigerant travels through the coil or tubes in the evaporator where it is totally vaporized by the warm air (from the space being refrigerated) that a fan circulates across the coil or tubes in the evaporator. The evaporator operates at essentially constant pressure. The resulting saturated refrigerant vapor returns to the compressor inlet at point 1 to complete the thermodynamic cycle.

It should be noted that the above discussion is based on the ideal vapor-compression refrigeration cycle which does not take into account real world items like frictional pressure drop in the system, slight internal irreversibility during the compression of the refrigerant vapor, or non-ideal gas behavior of refrigerant.

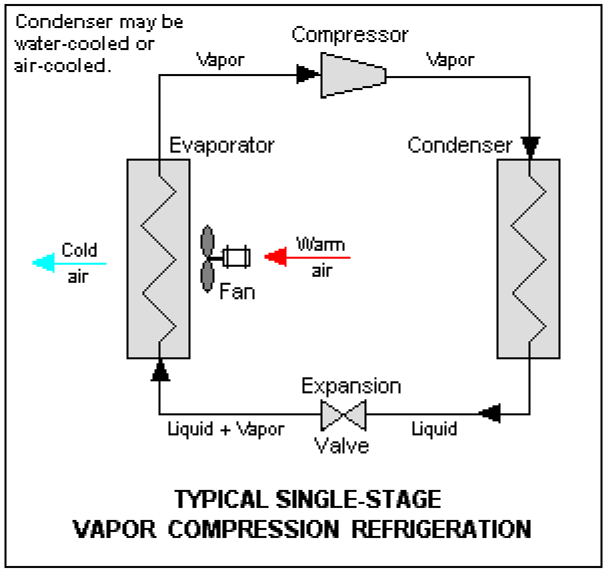


Fig 1.3 Vapor compression refrigeration

Heating and cooling is accomplished by moving a refrigerant through the heat pump's various indoor and outdoor coils and components. A compressor, condenser, expansion valve and evaporator are used to change states of the refrigerant from a liquid to hot gas and from a gas to a cold liquid. The refrigerant is used to heat or cool coils in a building or room and fans pull the room air over the coils. An external outdoor heat exchanger is used to heat or cool the refrigerant. This use of outside air has lead to the term "Air Source" Heat Pump. The overall operation uses the concepts described in classic vapor compression refrigeration.

The heat pump can also operate in a cooling mode where the cold refrigerant is moved through the indoor coils to cool the room air.

There are several advantages about the air source heat pump as below:

1. Typically draws approximately 1/3 to 1/4 of the electricity of a standard resistance heater for the same amount of heating, reducing utility bills.

2. This typical efficiency compares to 70-95% for a fossil fuel-powered boiler.

As an electric system, no flammable or potentially asphyxiating fuel is used at the point of heating, reducing the potential danger to users, and removing the need to obtain gas or fuel supplies (except for electricity).

The same system may be used for air conditioning in summer, as well as a heating system in winter.

Lower running costs – the compressor being the thing that uses most power.

1.2 Geothermal Heat Pump

A geothermal heat pump or ground source heat pump (GSHP) is a central heating and/or cooling system that pumps heat to or from the ground. It uses the earth as a heat source (in the winter) or a heat sink (in the summer). This design takes advantage of the moderate temperatures in the ground to boost efficiency and reduce the operational costs of heating and cooling systems, and may be combined with solar heating to form a geosolar system with even greater efficiency. Geothermal heat pumps are also known by a variety of other names, including geoexchange, earth-coupled; earth energy or water-source heat pumps. The engineering and scientific communities prefer the terms "geoexchange" or "ground source heat pumps" because geothermal power traditionally refers to heat originating from deep in the Earth's mantle (Rafferty et al 1997).Ground source heat pumps harvest a combination of geothermal power and heat from the sun when heating, but work against these heat sources when used for air conditioning (Hanova 2007).

Like a refrigerator or air conditioner, these systems use a heat pump to force the transfer of heat. Heat pumps are always more efficient at heating than pure electric heaters, even when extracting heat from cold winter air. But unlike an air-source heat pump, which transfers heat to or from the outside air, a ground source heat pump exchanges heat with the ground. This is much more energy-efficient because underground temperatures are more stable than air temperatures through the year. Seasonal variations drop off with depth and disappear below seven meters due to thermal inertia. Like a cave, the shallow ground temperature is warmer than the air above during the winter and cooler than the air in the summer. A ground source heat pump extracts ground heat in the winter (for heating) and transfers heat back into the ground in the summer (for cooling). Some systems are designed to operate in one mode only, heating or cooling, depending on climate (Eere.energy.gov 2009).

Most installed systems have two loops on the ground side: the primary refrigerant loop is contained in the appliance cabinet where it exchanges heat with a secondary water loop that is buried underground. The secondary loop is typically made of High-density polyethylene pipe and contains a mixture of water and anti-freeze (propylene glycol, denatured alcohol or methanol). After leaving the internal heat exchanger, the water flows through the secondary loop outside the building to exchange heat with the ground before returning. The secondary loop is placed below the frost line where the temperature is more stable, or preferably submerged in a body of water if available. Systems in wet ground or in water are generally more efficient than drier ground loops since it is less work to move heat in and out of water than solids in sand or soil. If the ground is naturally dry, soaker hoses may be buried with the ground loop to keep it wet.

Closed loop systems need a heat exchanger between the refrigerant loop and the water loop, and pumps in both loops. Some manufacturers have a separate ground loop fluid pump pack, while some integrate the pumping and valving within the heat pump. Expansion tanks and pressure relief valves may be installed on the heated fluid side. Closed loop systems have lower efficiency than direct exchange systems, so they require longer and larger pipe to be placed in the ground, increasing excavation costs.

1.3 Infrared Camera

Infrared imaging is used extensively for military and civilian purposes. Military applications include target acquisition, surveillance, night vision, homing and tracking. Non-military uses include thermal efficiency analysis, remote temperature sensing, short-ranged wireless communication, spectroscopy, and weather forecasting. Infrared astronomy uses sensor-equipped telescopes to penetrate dusty regions of space, such as molecular clouds; detect cool objects such as planets, and to view highly red-shifted objects from the early days of the universe.

Infrared radiation is popularly known as "heat" or sometimes "heat radiation", since many people attribute all radiant heating to infrared light and/or all infrared radiation to heating. This is a widespread misconception, since light and electromagnetic waves of any frequency will heat surfaces that absorb them. Infrared light from the Sun only accounts for 49% of the heating of the Earth, with the rest being caused by visible light that is absorbed then re-radiated at longer wavelengths. Visible light or ultravioletemitting lasers can char paper and incandescently hot objects emit visible radiation. It is true that objects at room temperature will emit radiation mostly concentrated in the 8 to 25 micrometer band, but this is not distinct from the emission of visible light by incandescent objects and ultraviolet by even hotter objects.

Heat is the energy in transient form that flows due to temperature difference. Unlike heat transmitted by thermal conduction or thermal convection, radiation can propagate through a vacuum.

The concept of emissivity is important in understanding the infrared emissions of objects. This is a property of a surface which describes how its thermal emissions deviate from the ideal of a black body. To further explain, two objects at the same physical temperature will not 'appear' the same temperature in an infrared image if they have differing emissivities.

At the atomic level, infrared energy elicits vibrational modes in a molecule through a change in the dipole moment, making it a useful frequency range for study of these energy states for molecules of the proper symmetry. Infrared spectroscopy examines absorption and transmission of photons in the infrared energy range, based on their frequency and intensity.

Infrared (IR) radiation is electromagnetic radiation whose wavelength is longer than that of visible light (400–700 nm), but shorter than that of terahertz radiation (100 μm – 1 mm) and microwaves. Infrared radiation spans more than three orders of magnitude (roughly 700 nm to 300 μm).

Bright sunlight provides an irradiance of approximately 1004 Watts per square meter at sea level at the Earth's surface, of which 32 W/m^2 is ultra-violet light; 445 W/m^2 is visible light; and 527 W/m^2 is infrared light.

1.4 Problem Statement

Ensuring the efficient operation of institutional and commercial facilities requires a coordinated system of automated diagnostics, interoperable controls, controls, and integrated HVAC components.

Important technology advances in recent years have created a more powerful array of diagnostic and monitoring tools — both handheld and system-based — that can help engineering and maintenance managers and front-line technicians detect and address small problems in HVAC system operation before they become larger and more expensive.

Many of these technology advances involve improved communication among manufacturers' components, subsystems, and building systems and information-technology (IT) services. They have affected the entire spectrum of HVAC system measurements and controls.

The resulting ability to better diagnose and monitor air handlers, chillers, pumps, and variable-air-volume (VAV) boxes is substantially reducing costs. Coupled with use of statistical analysis and system databases, these tools help technicians analyze deviations within operations to pinpoint the root causes and provide solutions so managers can implement the most appropriate solution quickly.

We also find more system interoperability with access control, security, closed-circuit and power monitoring. Now the same interoperable motion sensor can alert:

• The HVAC system to the need for heating or cooling

• The access-control system about an entry request

• The security system to the presence of an intruder

In short, one sensor can handle six system outputs. The advances offer more control for users, as well as more selection options and smarter operation for managers, more opportunities for suppliers.

Rather than one sensor for several spaces, wired sensors through the VFD are linked to one actuator, creating both better energy efficiency and greater comfort.

These devices provide a check against data from a building's monitoring system and add functionality. For instance, we can add enthalpy measurement if it is not available in the economizers. When connected to a computer, the devices also provide efficiency and analysis.

Testing revealed that rooftop units operated during unscheduled time. Some units had no working economizer. Heating and cooling were on at the same time. The heating mode compressor kicked in when the outside temperature was below 50 degrees. The COP will be the most important issue in our experimental consideration.

1.5 Research Goal

The purpose of this research is to understand the heat pump pattern with different power input, as well as to investigate the effect of cycle patterns on different temperature for the air input. We need to find out the best running situation of the system and save energy as much as possible. To achieve the goal of this current study, we will choose to do the experiments by changing the outdoor temperature and compressor frequency.

To simulate the changing of the temperature outside, we will do the experiments by putting the equipment near the window and wait until midnight to let the outdoor temperature decrease. By changing the power input, we will find the highest efficiency point to save the energy.

The investigation will be carried out by graph technique using the software Matlab and infrared imaging system. The data can be employed from the experiments and to simulate the different room temperature changing for different power input and air inlet patterns. The heat transfer equation will include the drag and the properties of material. The efficiency comparing with steady speed heat pump, geothermal heat pump will be studied as well.

CHAPTER 2

LITERATURE REVIEW

2.1 The Exergy and Coefficient analysis

First, below are mass, energy, entropy and exergy balance equations

For a general steady-state, steady-flow process, the four balance equations, namely mass, energy, entropy and exergy balance equations, are applied to find the heat input, the rate of exergy decrease, the rate of irreversibility, and the energy and exergy efficiencies (Kotas, 1985; Szargut et al., 1988; Cengel and Boles, 2001). The mass balance equation can be expressed in the rate form as

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{2.1}$$

Where $\dot{m}$ is the mass flow rate, and the subscript in stands for inlet and out for outlet. The general energy balance can be expressed as

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{2.2}$$

The energy (or first law) efficiency is simply a ratio of useful output energy to input energy and is referred to as a coefficient of performance (COP) for refrigeration systems. In this context, the energy efficiency of the GSHP unit itself (COPHP,act) and the whole GSHP system (COPsystem,act) can be defined as follows, respectively:

$$COP_{HP,act} = \frac{\dot{Q}_{sh}}{\dot{W}_{comp}} \tag{2.3}$$

Or

$$COP_{system,act} = \frac{\dot{Q}_{sh}}{\sum \dot{W}_{input}} \tag{2.4}$$

where $\dot{Q}_{sh}$ is the space heating load, $\dot{W}_{comp}$ is the work input to the compressor and $\sum \dot{W}_{input}$ is the total work input rate to the system.

The ralationship of SEER to EER and COP is as below:

SEER is related to the Energy Efficiency Ratio (EER), which is the ratio of output cooling in Btu/Hr and the *input* power in watts W at a given operating point and also to the coefficient of performance (COP) commonly used in thermodynamics. Performance ratios can be a unitless output over input ratio, never to exceed one, and it is also proper to state what kind of energy is in the numerator and denominator. The COP of a heat pump is determined by dividing the power output of the heat pump by the electrical power needed to run the heat pump, with both powers measured using the same units, e.g. watts. The higher the COP, the more efficient the heat pumps. For example resistive heat has a COP = 1. The EER is the efficiency rating for the equipment at a particular pair of external and internal temperatures. EER (Btu/ (W*hr)) is converted to COP (Btu/Btu (Note: some may write W/W)) by dividing by 3.413 Btu/(Hr*W).

The SEER is calculated at a part loaded standardized ARI test. This more closely represents the performance from equipment cycling, instead of the steady state conditions under which the EER is measured.

Typical EER for residential central cooling units = 0.875 X SEER
SEER is always a higher value than EER for the same equipment.

A SEER of 13 is approximately equivalent to a COP of 3.43, which means that 3.43 units of heat energy are removed from indoors per unit of work energy used to run the heat pump.

The US government SEER standards provided that SEER rating more accurately reflects overall system efficiency on a seasonal basis and EER reflects the system's energy efficiency at peak day operations. Both ratings are important when choosing products. As of January 2006, all residential air conditioners sold in the United States must have a SEER of at least 13. ENERGY STAR qualified Central Air Conditioners must have a SEER of at least 14.

Today, it is rare to see systems rated below SEER 9 in the United States because aging, existing units are being replaced with new, higher efficiency units. The United States now requires that residential systems manufactured after 2005 have a minimum SEER rating of 13, although window units are exempt from this law so their SEERs are still around 10.

Substantial energy savings can be obtained from more efficient systems. For example by upgrading from SEER 9 to SEER 13, the power consumption is reduced by 30% (equal to 1 - 9/13). It is claimed that this can result in an energy savings valued at up to US$300 per year depending on the usage rate and the cost of electricity.

With existing units that are still functional and when the time value of money is considered, most often retaining existing units rather than proactively replacing them is the most cost effective. Maintenance should be performed regularly to keep their efficiencies as high as possible.

But when either replacing equipment, or specifying new installations, a variety of SEERs are available. For most applications, the minimum or near-minimum SEER units are most cost effective, but the longer the cooling seasons, the higher the electricity costs, and the longer the purchasers will own the systems, incrementally higher SEER units are justified.

With reference to Fig. 1, $\dot{W}$ is supplied to the cycle between states 1 and 2, heat of condensation, Q_{HC} is released to a reservoir at T_H, between states 2 and 3. Liquid refrigerant makes an adiabatic expansion through the expansion valve between states 3 and 4, then evaporates absorbing $\dot{Q}_{LC}$ from a reservoir at T_L between states 4 and 1. By considering heat transfer and irreversibilities, there will be finite temperature differences $(T_{HC} - T_H)$ at the condenser, $(T_L - T_{LC})$ at the evaporator and the internal heat leak from hot to cold end of the system,If a simple linear relation is assumed for the heat transfer (E. Bilgen 2002).

$$\begin{aligned} Q_{HC} &= C_H (T_{HC} - T_H) \\ Q_{LC} &= C_L (T_L - T_{LC}) \\ Q &= C_i (T_H - T_L) \end{aligned} \tag{2.5}$$

Heating and cooling capacity of the system can be expressed as

$$\begin{aligned} Q_H &= Q_{HC} - Q_i \\ Q_L &= Q_{LC} - Q_i \end{aligned} \tag{2.6}$$

The system power input is

$$W = Q_{HC} - Q_{LC} = Q_H - Q_L \tag{2.7}$$

From equations above and the carnot coefficient formula, we can express COP as a function of Q_H, Q_L, Q_i and the temperature ratios τ.

$$COP = \frac{1}{1-\tau_c^{-1}}(1-\frac{Q_i}{Q_{HC}})$$
$$COP_L = \frac{1}{\tau_C - 1}(1-\frac{Q_i}{Q_{LC}}) \tag{2.8}$$

We can obtain for the heating and cooling capacities or loads, Q_H and Q_L,

$$Q_H = \frac{C_L\, C_H}{C_H + C_L} T_L(\tau_C - \tau) - Q_i$$
$$Q_L = \frac{C_L C_H}{C_H + C_L} T_H(\frac{1}{\tau} - \frac{1}{\tau_C}) - Q_i \tag{2.9}$$

Where C_L and C_H are the temperature of low and high which correspond to the evaporator and condenser.

Earlier studies show that the major irreversibilities are at the condensation and evaporation stages (R.L. Akau et al 1980). Hence it makes sense to consider the related conductances as design constraints. If we define, for example, the following conductance ratio as a parameter,

$$x = \frac{C_L}{C_H + C_L} \tag{2.10}$$

And insert it in equation above it can be shown that the loads are maximized with respect to *x*, if *x* = 0.5. This result makes sense because in modern heat pumps, depending on the mode of operation, the two heat exchangers are served alternately for condensation or evaporation.

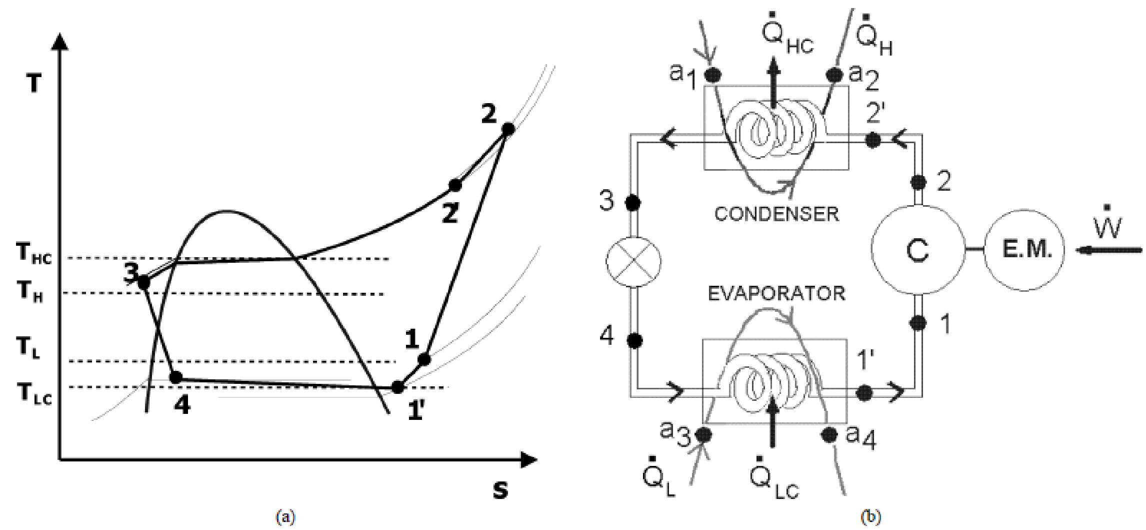


Fig 2.1 (a) T –s diagram of the heat pump system studies, (b) Simplified schematic of the heat pump system (E. Bilgen 2002)

Using x=0.5 as an optimum value, the maximum loads are obtained as

$$Q_{H,\max} = 0.25 C_e T_L (\tau_c - \tau) - Q_i$$
$$Q_{L,\max} = 0.25 C_e T_H (\frac{1}{\tau} - \frac{1}{\tau_c}) - Q_i \tag{2.11}$$

We can also obtain the COP maximized,

$$COP_{H,\max} = \frac{0.25 C_e T_L (\tau_c - \tau) - Q_i}{W}$$
$$COP_{L,\max} = \frac{0.25 C_e T_H (1/\tau - 1/\tau_c) - Q_i}{W} \tag{2.12}$$

The second law or exergy efficiency can be obtained if we apply the same definition of COP for a reverse Carnot cycle and use together. We obtain,

$$\eta_{H,E} = \frac{COP_H}{COP_{rev}} = \frac{1-\tau^{-1}}{1-\tau_c^{-1}} \left(1 + \frac{Q_i}{Q_H}\right)^{-1}$$
$$\eta_{L,E} = \frac{COP_L}{COP_{rev}} = \frac{\tau - 1}{\tau_c - 1} \left(1 + \frac{Q_i}{Q_L}\right)^{-1} \tag{2.13}$$

Exergetic COP for heating and cooling can be defined as

$$COP_{H,E} = \frac{E_H}{W}$$
$$COP_{L,E} = \frac{E_L}{W} \tag{2.14}$$

Experimental exergetic COP, on the other hand, may be determined from the measured loads, Q_H and Q_L or from above using experimental data as

$$COP_{H,E} = \frac{Q_H(1-T_0/T_b)}{W}$$
$$COP_{L,E} = \frac{Q_L(1-T_0/T_b)}{W} \tag{2.15}$$

Where T_b is the thermodynamic temperature at the boundary and T_0 is the temperature of the dead state.

Ground source or geothermal heat pumps (GSHPs or GHPs) are attractive alternatives to conventional heating and cooling systems owing to their higher energy utilizations efficiencies.GSHPs for direct utilization have had the largest growth of 9.7% annually since 1995. Most of this growth occurred in the U.S.A. and Europe, though interest is developing in other countries, such as Japan and Turkey. The installed capacity was 6850MW and annual energy use is 23 214 TJ in 26 countries in 2000. The actual number of installed units was around 500 000 in the same year, while the equivalent number of 12kW units installed was slightly over 570 000. The 12kW equivalent units installed are used as typical of homes in the U.S. and some western European countries (Lund and Freeston, 2000, 2001). It is also estimated that there are over a million today (Lund, 2004).

Different ways of formulating exergetic (or exergy) efficiency (second law efficiency, effectiveness, or rational efficiency) have been proposed in the literature. The first form of the exergy efficiency on the GSHP unit basis is as follows:

$$\eta_{HP} = \frac{COP_{HP}}{COP_{rev}} \tag{2.16}$$

The SEER and EER of an air conditioner are limited by the laws of thermodynamics (Cengel and Boles, 2001).. The refrigeration process with the maximum possible efficiency is the Carnot cycle. The COP of an air conditioner using the Carnot cycle is:

$$COP_{carnot} = \frac{T_H}{T_H - T_L} \tag{2.17}$$

The COP of the GSHP unit and the overall heating system (COPsystem) is calculated by the following equations, respectively:

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{2.18}$$

and

$$COP_{system} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp} + \dot{W}_{pump} + \dot{W}_{fc}} \tag{2.19}$$

$$COP_{system,act} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp,act} + \dot{W}_{fun.coil,act} + \dot{W}_{pumps,act}} \tag{2.20}$$

Where $\dot{Q}_{cond}$ is the heat transfer rate of the condenser (the space heating load), while $\dot{W}_{comp.act}, \dot{W}_{fan.coil,act}$ and $W_{pump,act}$ are the rates of actual work inputs to the compressor, fan-coil and circulating pumps, respectively.

2.2 Variable Speed compressor

The increased use of variable speed compressors (VSC) in refrigeration systems can potentially lead to the unstable operation when compressor speed is varied from time to time for capacity control. The causes of unstable operation may be classified into two groups, one relating to control algorithms and the other to the inherent characteristics of systems. This paper reports on a study on the operational stability of a VSC refrigeration system due to its inherent characteristics. Based on experimental results, a new modified minimal stable superheat (MSS) line having a maximum MSS value and a minimal MSS value has been proposed. Using the modified MSS line, and supported by a series of purposely designed experiments, a detailed analysis on the operational stability of a VSC refrigeration system due to its inherent characteristics when its compressor speed is changed for capacity control has been carried out and presented (Yiming Chen et, 2008).

Firstly, an experimental direct expansion (DX) air conditioning (A/C) plant, where all related experimental work was carried out, is briefly described. Secondly, the experimental work on qualitatively determining the relationship between MSS and the load imposed on a refrigeration system is presented, and a modified MSS line proposed. This is followed by reporting a detailed analysis on the operational stability of a VSC–EEV DX A/C system due to changes in compressor speed, using the modified MSS line and supported by a series of purposely designed experiments (Yiming Chen al et 2008).

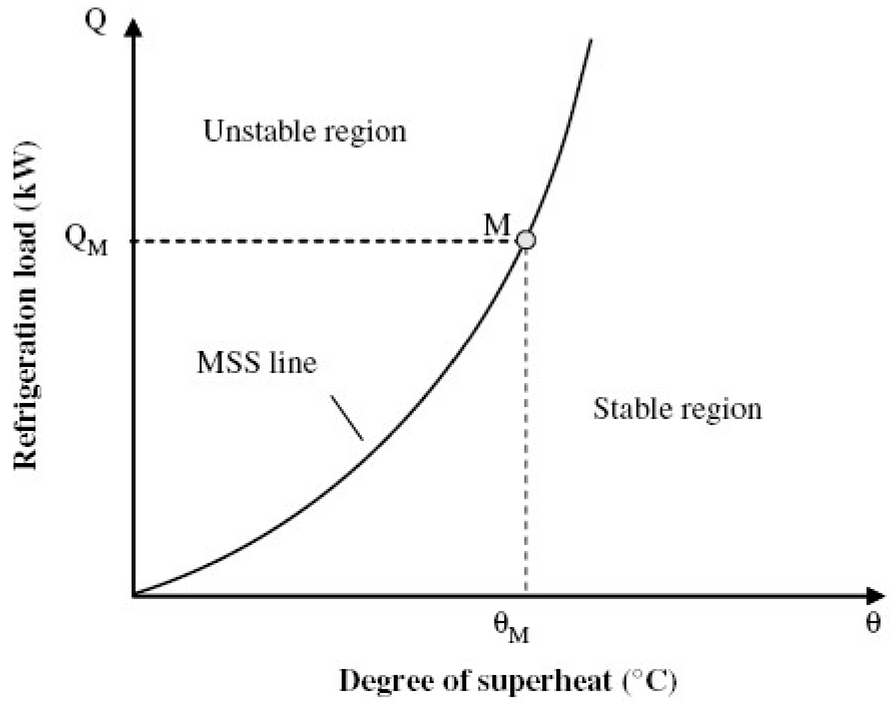


Fig 2.2 The MSS line as proposed by Huelle (1972)

The compressor performances are represented by means of diagrams in terms of cooling capacity, input power, refrigeration mass flow rate, when the condensation and evaporation temperatures vary. These diagrams are generally supplied only at the basic frequency; the manufacturers seldom provide the working curves for different frequency values of the compressor motor supply current. Hence, it is important to determine the compressor performances mapping when the frequency varies. In particular, the real working of a variable speed compressor can be considered as a set of infinite compressor performances at constant speed. In order to study the compressor performances when the frequency varies, the trend of refrigerant mass flow rate, input power and cooling capacity can be experimentally obtained and expressed by means of the second-order functions of the evaporation and condensation temperatures. For example, in fig 2.2, the cooling capacity trend of a reciprocating compressor is reported in terms of evaporation temperature, for a condensation temperature of 30℃ and for fixed frequencies (30, 40, 50 and 60 Hz). Additionally, referring to the reciprocating compressor, it is possible to note that, for different condensation and evaporation temperatures, the ratios of refrigerant mass flow rate, input power and cooling capacity values at any frequency with respect to the values at the basic frequency, remain constant (Huelle, 1972). The environment temperature, the superheating and sub cooling degrees of the refrigerant fluid are fixed too. In other words, the above-mentioned ratios are independent from condensation and evaporation temperatures and depend only on the compressor frequency, and hence can be expressed by means of the second-order functions of the frequency:

$$k = \frac{G}{G'} = a_1(f - f')^2 + a_2(f - f') + a_3 \tag{2.21}$$

Where G is the compressor input power (W), the refrigerant mass flow rates (M) or the cooling capacity (Q); a_i are constant coefficients; f the general frequency (Hz) and f' the basic frequency (Hz) (A.E. Dabiri et al 1981).

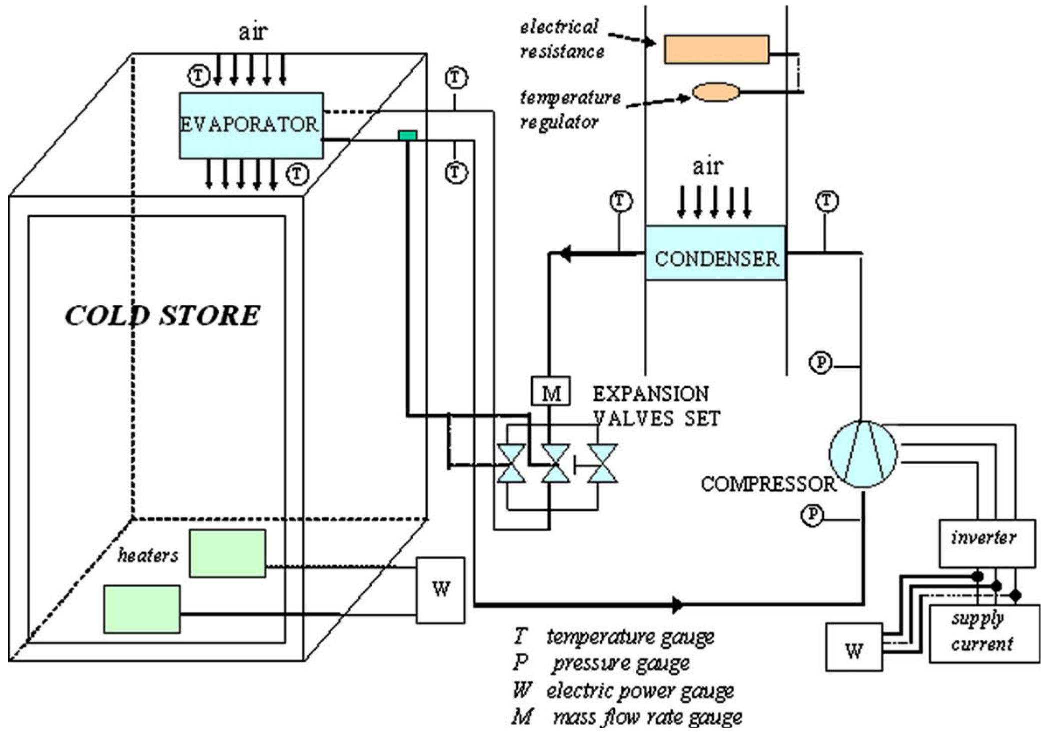


Fig 2.3 Experimental plant subjected to a cold store (C. Aprea et al 2009)

2.3 Geothermal heat pump

A GSHP unit is an assembly of an electrically driven compressor, two heat exchangers (refrigerant-air and refrigerant-water), and an expansion valve (throttle). Its primary aim is utilization of the ground as a heat source in winter and as a heat sink in a summer period. In winter/summer, natural heat of the ground is absorbed rejected by an antifreeze solution flowing in the GHE, which can be a series of plastic pipes installed below the ground surface (Fig.2.4) or submersed in a water reservoir (lake, river, sea, ocean, etc.). Majority of GSHP systems use closed-loop GHEs installed horizontally or vertically. The GSHP with a closed-loop GHE offers a coefficient of performance (COP) between 3 and 5. In the lower COP range, single-speed rotary or reciprocating compressors are used, while scroll compressors are common in mid-range units. The high

COP units use two-speed compressors and/or variable-speed indoor fan motors. For maximum cost-effectiveness, the GSHP is usually sized to meet 60–70% of the total maximum load demand (space heating and domestic hot water). Occasional peak heating demand during severe weather conditions can be met by using a supplementary heating system (wood stoves, electric resistance heaters, etc.).

The basic principle of a ground-source heat pump is shown in Figure 7. Heat can be extracted from the ground at a relatively low temperature the temperature is then increased through the heat pump and used in a heating system. For each kWh of heating output, only 0.22-0.30 kWh of electricity are required to operate the system (i.e., the seasonal performance factor is 3.3-4.5). For cooling in summertime, the system can be reversed, and heat from building cooling can be injected into the ground for highly effective space cooling.

The ground system links the heat pump to the underground and allows for extraction of heat from the ground or injection of heat into the ground. These systems can be classified generally as open or closed systems:

Open systems: Groundwater is used as a heat carrier, and is brought directly to the heat pump.

Closed systems: Heat exchangers are located in the underground (either in a horizontal, vertical or oblique fashion), and a heat carrier medium is circulated within the heat exchangers, transporting heat from the ground to the heat pump (or vice versa).

The system cannot always be attributed exactly to one of the above categories. Examples of exceptions are use of standing column wells, mine water or tunnel water. To choose the right system for a specific installation, several factors have to be considered:

geology and hydro-geology of the underground (sufficient permeability is a must for open systems); area and utilization on the surface (horizontal closed systems require a certain area); existence of potential heat sources like mines; and, the heating and cooling characteristics of the building(s). In the design phase, more accurate data for the key parameters for the chosen technology are necessary; to size the ground system in such a way that optimum performance is achieved with minimum cost.

The main technical part of open systems are groundwater wells, to extract or inject water from/to water bearing layers in the underground (“aquifers“). In most cases, two wells are required (“doublette“), one to extract the groundwater, and one to re-inject it into the same aquifer from which it was produced.

With open systems, a powerful heat source can be exploited at comparably low cost. On the other hand, groundwater wells require some maintenance, and open systems in general are confined to sites with suitable aquifers. The main requirements are:

Sufficient permeability

Good groundwater chemistry (e.g., low iron content, to avoid problems with scaling, clogging and corrosion.

Open systems tend to be used for larger installations.

The closed system easiest to install is the horizontal ground heat exchanger (synonym: ground heat collector, horizontal loop). Due to restrictions in the area available, in Western and Central Europe the individual pipes are laid in a relatively dense pattern, connected either in series or in parallel (Figure 5).

To save surface area with ground heat collectors, some special ground heat exchangers have been developed. Exploiting a smaller area at the same volume, these collectors are best suited for heat pump systems for heating and cooling, where natural temperature recharge of the ground is not vital. Spiral forms

(Figure 6) are popular in USA, mainly in the form of the so-called “slinky” collectors (placed horizontally in a wide trench like in the figure, or vertically in a narrow trench).

The main thermal recharge for all horizontal systems is provided for mainly by the solar radiation to the Earth's surface. It is important not to cover the surface above the ground heat collector.

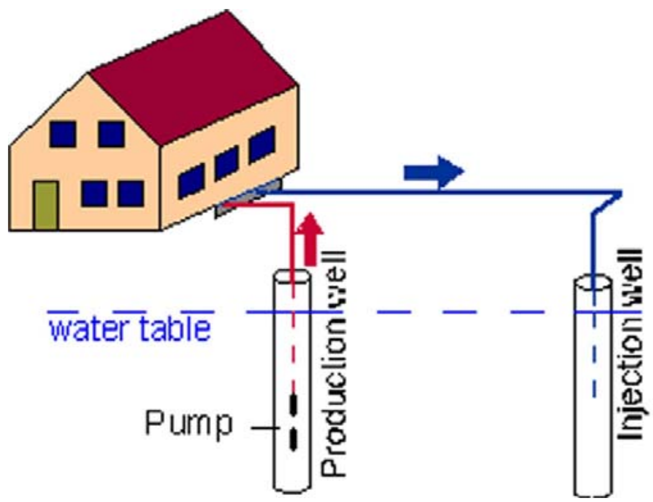


Fig 2.4 Groundwater heat pump (doublette)

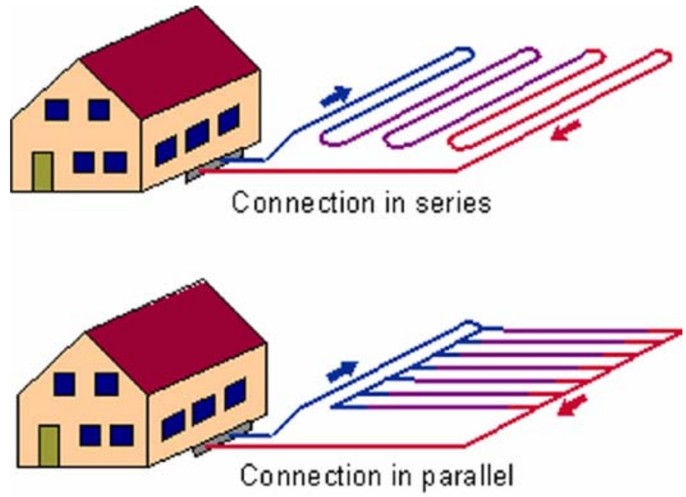


Fig 2.5 Horizontal ground heat exchanger (European style)

Evaluation of the GSHP thermodynamic performance is based on the following energy related characteristics. The total energy consumption (W_o) of the GSHP system includes all system components:

$$W_o = W_{co} + W_{fan} + W_{\sup} + W_{pump}, \tag{2.22}$$

The seasonal heating COP:

$$COP_H = \frac{Q_H}{W_{o-H}} \tag{2.23}$$

The seasonal cooling COP:

$$COP_C = \frac{Q_C}{W_{o-C}} \tag{2.24}$$

Finally, the annual overall COP is defined as follows.

$$COP_o = \frac{Q_H + Q_C}{W_{o-H} + W_{o-C}}, \tag{2.25}$$

Where W_{co} is electrical energy supplied to a compressor; W_{fan} is electrical energy supplied to an indoor circulation fan; W_{pump} is electrical energy supplied to a circulation pump of antifreeze solution; $W_{\sup}$ is electric energy required for supplementary heating, $(W_{\sup})_{heating} = Q_{heating-load} - Q_H$, or cooling, $(W_{\sup})_{cooling} = Q_{cooling-load} - Q_C$; Q_H and Q_C are the heating and cooling demand provided by the GSHP system, respectively. It is assumed that the efficiencies of the circulation pump and its electric motor are 70% and 90%, respectively

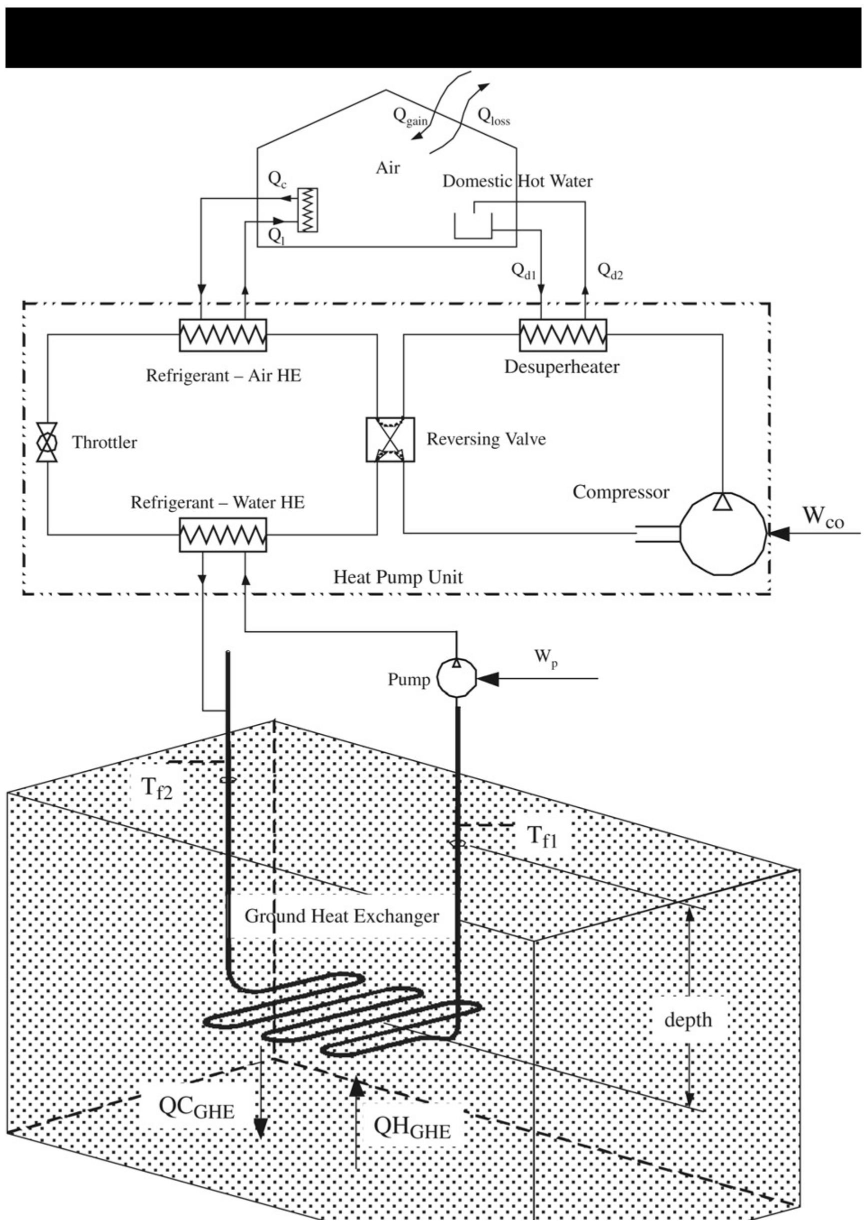


Fig 2.6 Schematic of a ground source heat pump system (V.R. Tarnawski et al 2009)

CHAPTER 3

THEORY

3.1 First Law of Thermodynamics

3.1.1 Energy Balance for Closed Systems

The energy of a closed system can be changed are through transfer of energy by work or by heat. A fundamental aspect of the energy concept is that energy is conserved which people call it the first law of thermodynamics. The energy balance can be expressed in symbols as:

$$E_2 - E_1 = Q - W \tag{3.1}$$

The symbol Q denotes an amount of energy transferred across the boundary of a system in a heat interaction with the system's surroundings. Heat transfer into a system is taken to be positive and heat transfer from a system is taken as negative.

The symbol W means the work for a change to the system.

The $E_2 - E_1$ means the change in the amount of energy contained within the system.

It requires that in any process of a closed system the energy of the system increases or decreases by an amount equal to the net amount of energy transferred across its boundary. It also shows that an energy transfer across the system boundary results in a change in one or more of the macroscopic energy forms: kinetic energy, gravitational potential energy, and internal energy.

The instantaneous time rate form of the energy balance is

$$\frac{dE}{dt} = \dot{Q} - \dot{W} \tag{3.2}$$

3.2 Refrigeration and Heat Pump Cycles

For the refrigeration and heat pump cycles, the equation can be shown as below:

$$W_{cycle} = Q_{out} - Q_{in}$$

(3.3)

For cycles of this type, Q_{in} is the energy transferred by heat into the system undergoing the cycle from the cold body, and Q_{out} is the energy discharged by heat transfer from the system to the hot body. To accomplish these energy transfers requires a net work input W_{cycle}. Since W_{cycle} is positive in this equation, it follows that Q_{out} is greater than Q_{in}.

Even people treat them as the same to this point, refrigeration and heat pump cycles actually have different objectives. The objective of a refrigeration cycle is to cool a space or to maintain the room temperature lower than the outside temperature. The objective of a heat pump is to maintain the temperature above the surroundings where the air needs to be heated.

Since refrigeration and heat pump cycles have different objectives, their performance parameters, called coefficients of performance, are defined differently.

For the refrigeration cycles, the performance of refrigeration cycles can be described as the ratio of the amount of energy received by the system undergoing the cycle from the cold body, Q_{in} to the net work into the system to accomplish this effect, W_{cycle}. Thus the coefficient of performance, β is

$$\beta = \frac{Q_{in}}{W_{cycle}} \tag{3.4}$$

For a household refrigerator, Q_{out} is discharged to the space in which the refrigerator is located. W_{cycle} is usually provided in the form of electricity to run the motor that drives the refrigerator.

For the heat pump cycles, the performance of heat pumps can be described as the ratio of the amount of energy discharged from the system undergoing the cycle to the hot body, Q_{out}, to the net work into the system to accomplish this effect, W_{cycle}. The coefficient of performance, γ is

$$\gamma = \frac{Q_{out}}{W_{cycle}} \tag{3.5}$$

For residential heat pumps, the energy quantity Q_{in} is normally drawn from the surrounding atmosphere, the ground, or a nearby body of water. W_{cycle} is usually provided by electricity.

3.3 Second Law of Thermodynamics

3.3.1 Refrigeration and Heat Pump Cycles Interacting

The second law of thermodynamics places limits on the performance of refrigeration and heat pump cycles as it does for power cycles. In accord with the conservation of energy principle, the cycle discharges energy Q_H by heat transfer to the hot reservoir equal to the sum of the energy Q_C received by heat transfer from the cold reservoir and the net work input. This cycle might be a refrigeration cycle or a heat pump cycle, depending on whether its function is to remove energy Q_C from the cold reservoir or deliver energy Q_H to the hot reservoir.

If W_{cycle} were identically zero, the system would withdraw energy Q_C from the cold reservoir and deliver energy Q_C to the hot reservoir, while undergoing a cycle. However, this method of operation would violate the Clausius statement of the second law and thus is not allowed. It follows that these coefficients of performance must invariably be finite in value. This may be regarded as another corollary of the second law.

3.3.2 Maximum Performance Measures

When the cooling and heating system is operating between thermal reservoirs at temperatures T_H and T_C. That is

$$\eta_{\max} = 1 - \frac{T_C}{T_H}$$

(3.6)

Which is known as the Carnot efficiency.

Also the coefficient of performance of any system undergoing a reversible refrigeration cycle while operating between the two reservoirs:

$$\beta = \frac{T_C}{T_H - T_C} \tag{3.6}$$

Similarly, for the heat pump cycle:

$$\gamma_{\max} = \frac{T_H}{T_H - T_C} \tag{3.7}$$

Note that the temperatures used must be absolute temperatures on the Kelvin or Rankine scale.

3.4 Constant Surface Temperature

Results for the total heat transfer rate and the axial distribution of the mean temperature are entirely different for the constant surface temperature condition.

From the definition of the average convection heat transfer coefficient, we can get that:

$$\frac{\Delta T_o}{\Delta T_i} = \frac{T_s - T_{m,o}}{T_s - T_{m,i}} = \exp\left(- \frac{PL}{\dot{m} c_p} \bar{h} \right) \tag{3.8}$$

Where $\bar{h}$ is the average value of convection heat transfer coefficient for the entire tube. T_s is constant and the temperature for outside flows, $T_{m,o}$ is the temperature for outlet flows. $T_{m,i}$ is the temperature of inlet flows. P is the surface perimeter. $\dot{m}$ is the internal flow rate. c_p is the heat capacity.

It is usually reasonable to neglect heat transfer by conduction in the axial direction. So we can get the heat transfer term:

$$q_{conv} = \dot{m} c_p \left(T_{m,o} - T_{m,i} \right) \tag{3.9}$$

To determine an expression for the total heat transfer rate q_{conv} is complicated by the exponential nature of the temperature decay. We can obtain:

$$q_{conv} = \bar{h} A_s \Delta T_{lm} \tag{3.10}$$

Where A_s is the tube surface area and T_{lm} is the log mean temperature difference.

$$\Delta T_{lm} = \frac{\Delta T_o - \Delta T_i}{\ln\left(\Delta T_o / \Delta T_i \right)} \tag{3.11}$$

T_{lm} is the appropriate average of the temperature difference over the tube length.

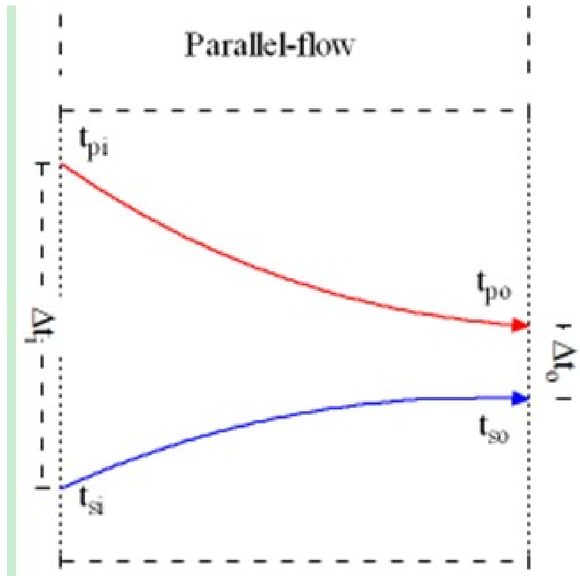


Figure 3.1 Flow in the same direction

The log mean temperature difference (LMTD) is used to determine the temperature driving force for heat transfer in flow systems, most notably in heat exchangers. The LMTD is a logarithmic average of the temperature difference between the hot and cold streams at each end of the exchanger. The larger the LMTD, the more heat is transferred. The use of the LMTD arises straightforwardly from the analysis of a heat exchanger with constant flow rate and fluid thermal properties.

It is important to note that, in many applications, it is the temperature of an external fluid, rather than the tube surface temperature. In this case, we can get:

$$\frac{\Delta T_o}{\Delta T_i} = \frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = \exp\left(-\frac{\overline{U}A}{\dot{m}c_p}\right) \tag{3.12}$$

And

$$q = \overline{U}A_s \Delta T_{lm} \tag{3.13}$$

Where $\overline{U}$ is the average overall heat transfer coefficient. T_∞ is the free stream temperature of the external fluid.

CHAPTER 4

EXPERIMENTAL METHOD

In this section the results of heating cycle field fractions for the Part 1: Experiment, Part 2: Experiment results, Part 3: Data analyze and Part 4: Infrared imaging are presented.

4.1 Equipment

4.1.1 Heat pump

The Hampden H-ACT-1-CDL-L air conditioning system is a completely functional mechanical air conditioning system. Its compact layout includes: two horsepower condensing unit, scroll type hermetic compressor with a liquid receiver, forced air evaporator, electronic high and low pressure control, filter/drier and sight glass with moisture indicator, temperature control.

One of the three liquid control devices (capillary tube, thermostatic expansion valve, hand expansion valve) may be used, illustrating different operating characteristics. Further, the system is capable of reverse cycle operation to demonstrate heat pump principles.

The 0-220 ºF thermometer should be used in the well following the compressor. Gauges indicate high side (condensing) pressure and low side (evaporating) pressure. Sight glasses in the liquid and gas lines helps to understand the refrigerant change of state in the evaporator and condenser.

This system uses the R-410A refrigeration which is a blend of R-32 and R-125 refrigerants. They have almost the same condensing and evaporating temperatures, so and it can be operated down. Because R-410A air conditioning systems operate at higher pressure, modifications need to be made for assembling refrigerant pipe and tubing.

A compressor connected to an AC source of the required voltage will run at 60 Hz and operate at one speed. With use of a variable frequency drive (VFD), the compressor will change speeds as the output frequency of the VFD change. The VFD must work within the parameters of the compressor it is controlling. Normal operation is at 60 Hz it can be operated down to a frequency of 45 Hz. Differences in high and low pressure, voltage, and wattage can be observed as the compressor is run up and down the frequency range scale.

The compressor should not be operated below a speed of 45 Hz. Inadequate circulation of fluids can take place below this speed resulting in damage to compressor. The VFD is programmed to prevent this.

Metering the refrigerant flow to the evaporator is the sole function of a thermal expansion valve (TEV). What's critical is that it must meter that flow at the same rate it is being vaporized by the heat load.

To do this, it keeps the coil supplied with the proper amount of refrigerant to maintain the right superheat of the suction gas leaving the evaporator.

Fig 4.1 Variable speed compressor system

4.1.2 Data acquisition system

The VFD can be operated in both local and remote mode. In local it is operated by the keypad attached to the VFD in figure 4.2.

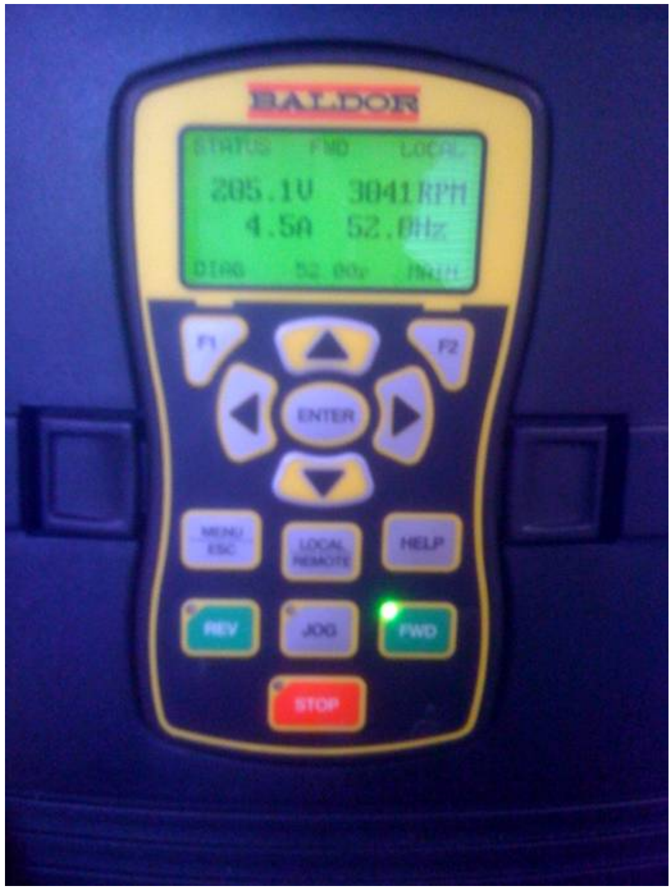


Figure 4.2 LOCAL control panel

The remote is operated at the computer screen in Figure 4.3. It is important to become familiar with running the compressor through the operation of the VFD as it will be needed to operate the unit.

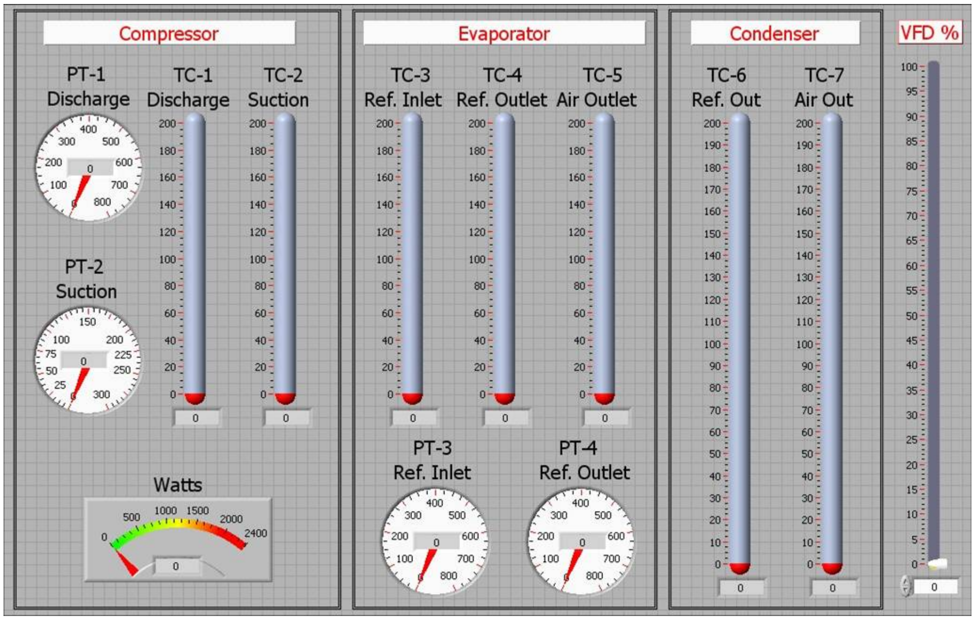


Figure 4.3 Variable frequency driver panel

4.1.3 Infrared

Infrared (IR) radiation is electromagnetic radiation whose wavelength is longer than that of visible light (400–700 nm), but shorter than that of terahertz radiation (100 µm – 1 mm) and microwaves. Infrared radiation spans more than three orders of magnitude (roughly 700 nm to 300 µm).

Bright sunlight provides an irradiance of approximately 1004 Watts per square meter at sea level at the Earth's surface, of which 32 W/m^2 is ultra-violet light; 445 W/m^2 is visible light; and 527 W/m^2 is infrared light.

4.2 Operation procedure

The temperature control thermostat turns the compressor on and off, based on the temperature of its sensing element relative to its temperature setting. Sensing bulbs are normally mounted in the air stream in back of the evaporator. On the trainer, the sensor will be sensing room temperature. Therefore, in order for the compressor to operate, the thermostat must be set below room temperature.

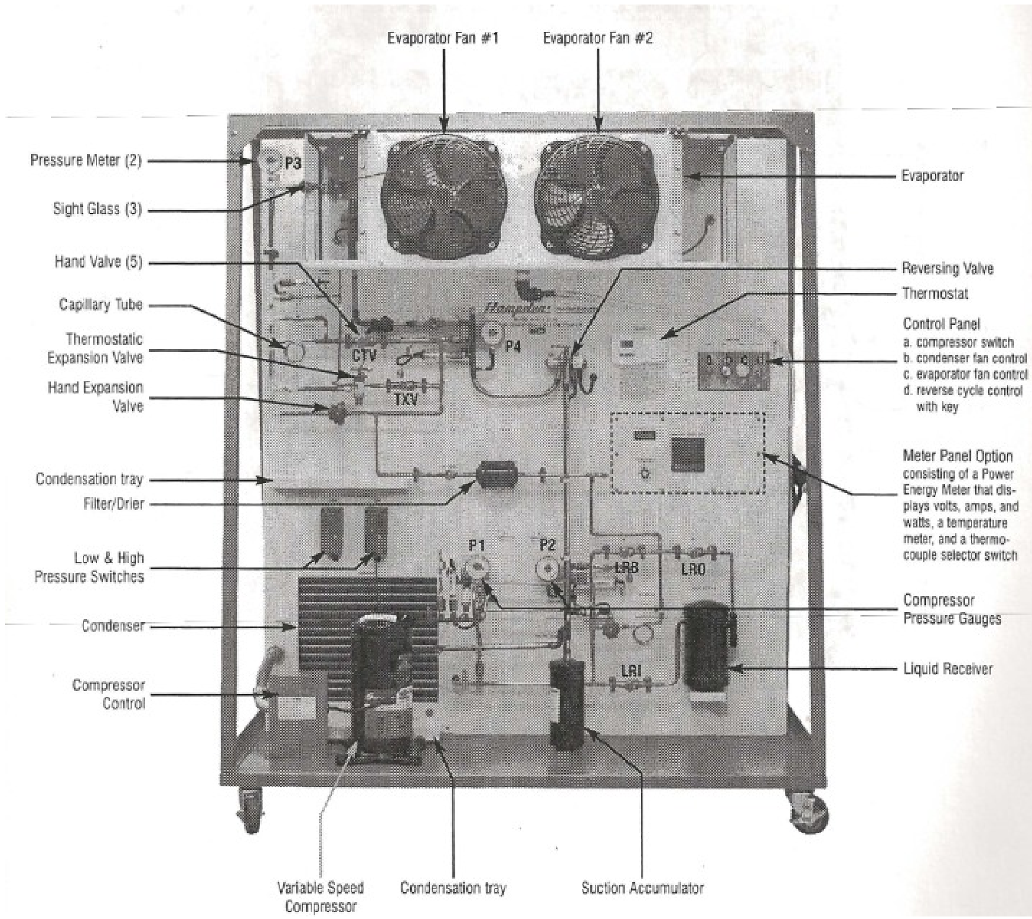


Figure 4.4 Trainer layout

Below is the operation procedure for the heating mode:

1. Close the thermostatic Expansion Valve (TXV). Open the Capillary Tube Valve (CTV).

2. Close the Liquid Receiver (LRI) and LRO, which isolates the liquid receiver. Please note that the capillary tube system relies on an exact amount of refrigerant.
3. Open the LRB to bypass the liquid receiver.
4. Start the training unit and let it operate for about 15 minutes. If the system appears to have less refrigerant than it needs to operate with 5°F to 10°F evaporator superheat, consider it normal, as the evaporator is under a full load (ambient temperature), which affects the operation of the capillary tube. If the evaporation superheat is excessive, open the LRO valve to add refrigerant to the system. If the high side pressure is excessive, open the LRI valve to allow excess refrigerant to enter the liquid receiver.
5. With pressure $\geq$ 15 PSI change reversing valve to reverse.
6. Turn thermostat off then to heat system should start. If it does not press the up arrow on the thermostat.
7. If more refrigerant is required, crack opens the blue handle valve from the receiver and then closes it back.

Figure 4.5 shows the process from 1 to 4 and figure 4.6 shows the rest procedures.

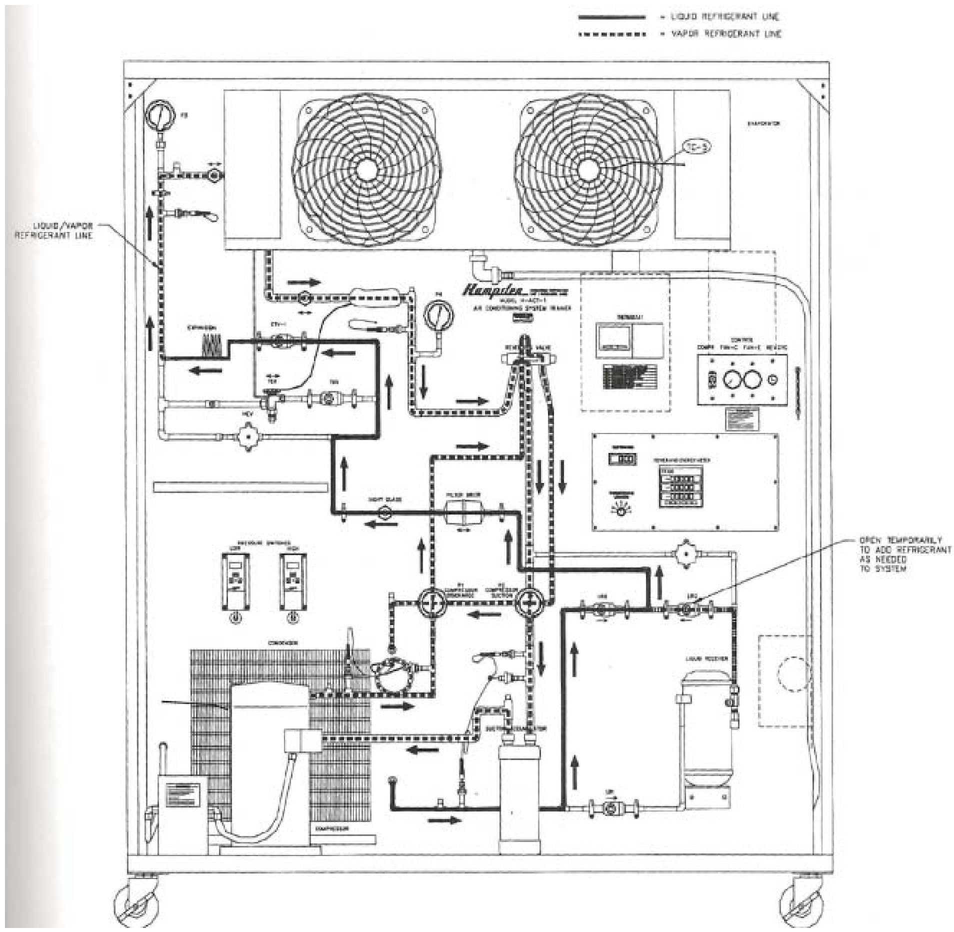
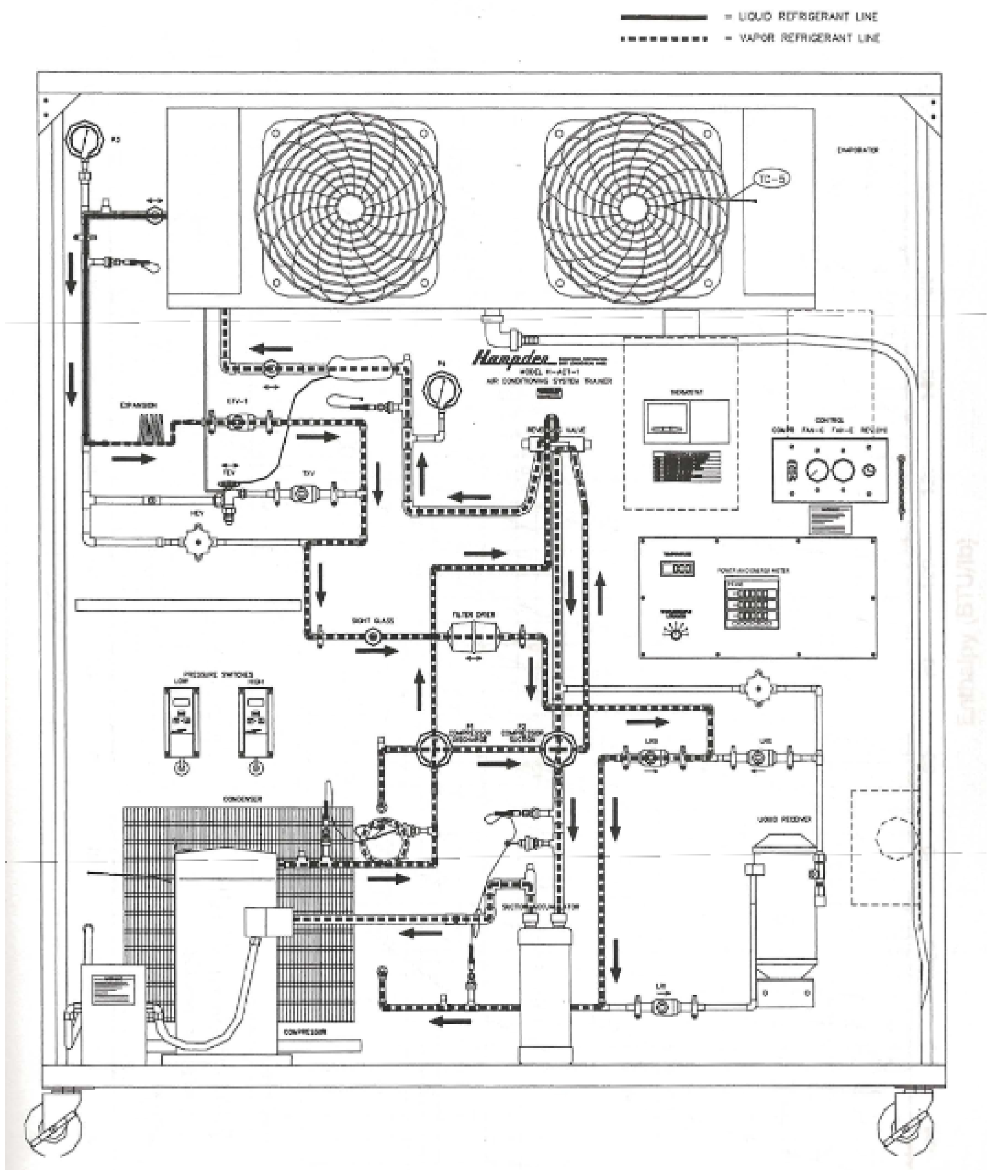


Figure 4.5 Flow diagram for cooling mode



4.6 Flow diagram for heating mode

The procedure to operate the compressor through the VFD is as below:

1. Turn on the main AC circuit breaker mounted on the back of the unit. The VFD is also mounted on the back. On power up the VFD display

will illuminate and go through its diagnostics. On completion it will be ready to display voltage, RPM, frequency, and motor amperage. The PR300 power energy meter will be ready to display voltage, amperage, and wattage.

2. To operate it from the keypad press LOCAL then FWD. The VFD will start the compressor and bring it up to the minimum speed of 45 Hz. Press the up arrow and bring it to 60 Hz.

3. To operate from Remote after the main circuit breaker has been turned on, bring up the Labview screen. This will display temperatures, pressures, and the wattage of the compressor. The slide is provided to control the compressor speed.

4. Before operating the compressor from the computer screen. Go to the VFD and in the following order press LOCAL, FWD, STOP, and REMOTE. The VFD will start the compressor and bring it to whatever speed the slide is set at. If it is all the way down the compressor will run at 45 Hz. If it is all the way up, it will run at 60 Hz. Moving the slide on the screen up and down will change the speed of the compressor.

4.3 Experimental parameter

The movement of the molecules in a solid can be thought of as a vibration—in a liquid, swimming, and in a gas, flying. The vibrating molecules of a solid are no particular interest to us, since refrigerant does not become solid during the refrigeration cycle. It is interesting to note, however, that the meaning of a ton of refrigeration is supposed to be the refrigerating effect of a ton of ice as it is changed from 32°F liquid in a 24 hour period.

Any pressure created, as with a pump, is in addition to the 14.7 psi. A pressure gauge reads zero at atmospheric pressure. Its reading, then, is in terms of pounds per square inch gauge (psig). This simply means we are ignoring the atmospheric pressure.

If your gauge reads in pounds per square inch absolute (psia), it would read 14.7 when no additional pressure is present.

New tables of the thermodynamic properties of 410A refrigerant have been developed and are presented here. Equations have been developed, based on the Martin-Hou equation of state, which represent the data with accuracy and consistency throughout the entire range of temperature, pressure, and density. Vapor enthalpy and entropy are calculated from the standard Martin-Hou equations. Additional equations have been developed for the calculation of saturated liquid enthalpy, latent enthalpy, and saturated liquid entropy, and are presented here.

Table 4.1 Physical properties of R-401A

Chemical Formula	CH_2F_2 / CHF_2CF_3
Molecular Weight	72.58
Boiling point at one atmosphere	-60.84°F (-51.58°C)
Critical temperature	161.83°F (72.13°C)
Critical pressure	714.50 psia (4926.1 kPa)
Critical Density	30.51 lb/ft^3 (488.90 kg/m^3)
Critical volume	0.0328 ft^3/lb (0.00205 m^3/kg)

4.3.1 Test conditions

Test conditions are summarized in Table 4.2. The Hampden Model H-ACT-1-CDL-L has a total of thirteen measurement inputs: four inputs, one watts input, seven temperature input measurements, and one VFD control output. This is the RS-232/RS-485 communications module at address zero.

The cycle-matching tests were performed at the ASHRAE standard test conditions for both cooling and heating modes (ASHRAE, 1983) using different levels of charge and sizes of capillary tube. For these individual combinations, performance measurements were made for different compressor speeds.

\

Table 4.2 Test conditions

H-ACT-1 Fieldpoint CDL					
Transmitter	Description	Measurement	Output	Address	Fieldpoint
Watts	Compressor Watts	0 - 4000 W	0.0 to 0.024 mA	1/0	Module #1 FP-AI- 100
PT-1	Discharge	0 -750 psi	0.0 to 0.024 mA	1/1	
PT-2	Suction	0 – 300 psi	0.0 to 0.024 mA	1/2	
PT-3	Evaporator Inlet	0 – 750 psi	0.035 to 0.024 mA	1/3	
PT-4	Evaporator Outlet	0 – 750 psi	0.0 to 0.024 mA	1/4	
TC-1	Discharge	454 to 3218°F	Type T	2/0	Module #2 FP-TC- 120
TC-2	Suction	454 to 3218°F	Type T	2/1	
TC-3	Evaporator REF Inlet	454 to 3218°F	Type T	2/2	
TC-4	Evaporator REF Outlet	454 to 3218°F	Type T	2/3	
TC-5	Evaporator Air Outlet	454 to 3218°F	Type T	2/4	
TC-6	Condenser REF Outlet	454 to 3218°F	Type T	2/5	
TC-7	Condenser Air Outlet	454 to 3218°F	Type T	2/6	
VFD-1	VFD Hz Control	45 – 60 Hz 0-100%	1- 10 VDC	3/0	Module #3 FP-AO-210

CHAPTER 5

RESULTS AND DISCUSSION

In this section the results of heating cycle field fractions for Part 1: Experiment results, Part 2: Data analysis and Part 3: Infrared imaging are presented.

5.1 Experiment Result

5.1.1 Data from AC system

One needs to understand the cycle structures in the different temperature to be able to analyze refrigerant cycle because the mass flow is one of the important factors, which affect the system's coefficient in the cycle. Such an understanding will benefit the efficiency of energy consuming.

Record the temperature and pressures for the following Table 5.1. In Table, we recorded the data which comes from the meters directly without calculations. The sixteen records located in the cycle for different parts.

Table 5.1 Experimental data

Ambient Temperature	55.4 °F
Power	1498 Watt
Compressor Frequency	55.2 Hz
Compressor Discharge temperature	165 °F
Compressor Suction temperature	56 °F
Temperature of Inlet Refrigerant from Condenser	160 °F
Temperature of Outlet Refrigerant from Condenser	86 °F
Outlet Air Temperature from Condenser	79 °F
Temperature of Outlet Refrigerant from Evaporator	27 °F
Outlet Air Temperature from Evaporator	46 °F
Pressure of Refrigerant Discharge from Compressor	338 psig
Pressure of Refrigerant Suction from Compressor	77.9 psig
Pressure of Refrigerant Inlet from Condenser	344 psig
Compressor Current	5.4 A
Compressor Voltage	219.7 V
Compressor Speed	3203 RPM

The data in Table 5.1 is one set out of the sixteen. The pressure of refrigerant suction from compressor is taken at (A) in figure 5.3. This is the service valve at the compressor. This chart corresponds to the VFD channel as Figure 5.1

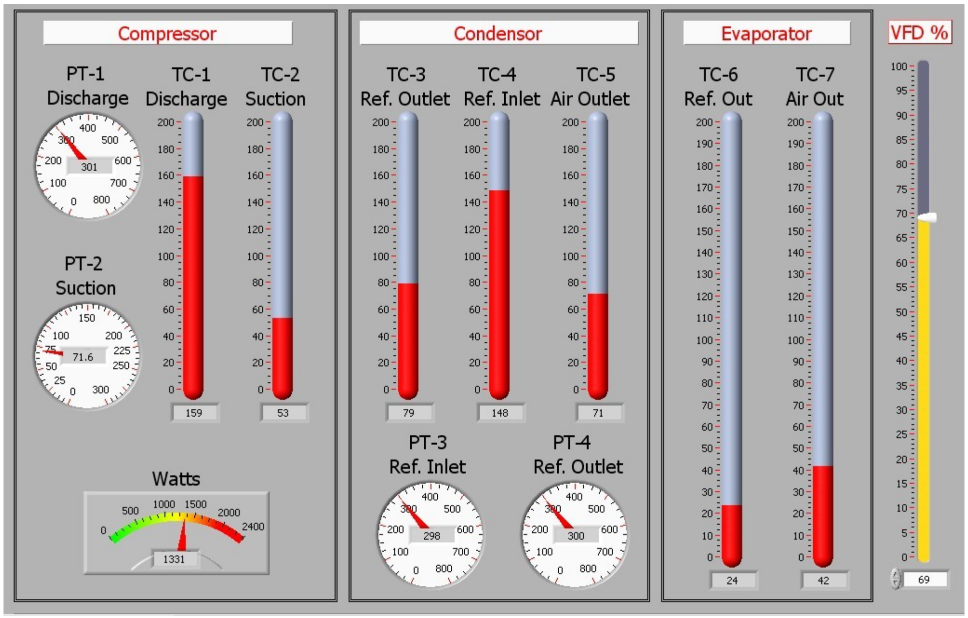


Figure 5.1 Variable frequency driver panel when operating

Figure 5.2 shows the way how to find the air flow of the condenser. Since the fan of the condenser can not give a steady flow directly, we can use this tube to get the ideal data. After the air going through the tube, we can get a steady data from every different point in the cycle.

Figure 5.2 Air flow measurement: (a) System layoff (b) Air flow tube placement

5.1.2 Data explanation

The pressure of refrigerant inlet from condenser is taken at B in Figure 5.3. This is the service valve at the condenser.

The temperature of outlet refrigerant from condenser is taken at E. This is the liquid line going to the expansion valve.

Temperature E to F indicates the metering device pressure and temperature drop.

The line from F to G is the evaporator that absorbs heat.

The line crossing the liquid curve G to A is the superheat.

The ambient temperature comes from the thermocouple placed at the air inlet of the condenser.

The power is read for the whole system.

The compressor frequency comes from the LOCAL control panel.

The compressor's voltage, current and speed are read from the LOCAL control panel.

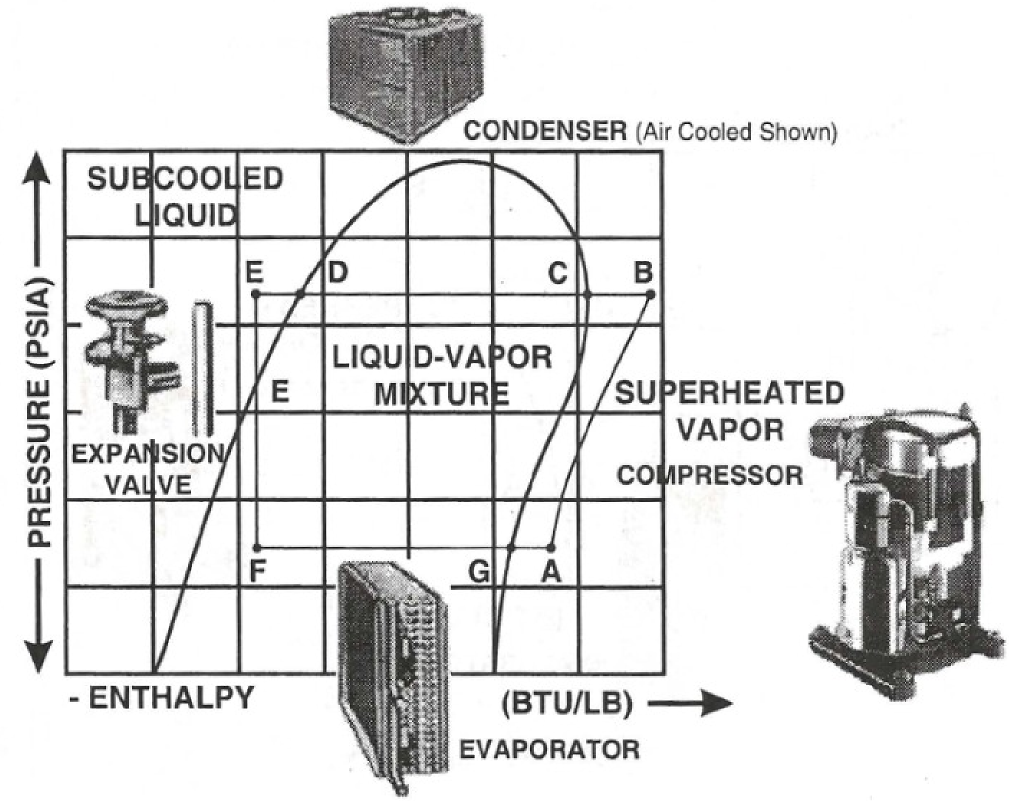


Figure 5.3 Pressure- enthalpy chart

5.2 Data Analysis

5.2.1 Basic theoretic analysis

To determine the accurate enthalpy for each point, we can find the enthalpy the nearest upper limit and under limit of the measured data from the chart. For example, in Table 5.2, for the point of inlet condenser, the pressure and temperature are 344 psig, 160 °F. After this, we can find the enthalpy of 350 psig, 160 °F first, and then find the point at 340 psig, 160 °F. After we have two limits enthalpy, we can find the point of 344 psig, 160 °F by subtracting from low enthalpy to high enthalpy point then divided by ten. If the temperature is not 160 °F but 158 °F, we can make the same procedure based on temperature.

After we get the enthalpy, the coefficient of a simple vapor-compression heat pump with states is

$$COP = \frac{h_2 - h_3}{h_2 - h_1} \tag{5.1}$$

For our system, the COP comes from the equation as below:

$$COP = \frac{h_2' - h_3}{h_2 - h_1} \tag{5.2}$$

Where the h_2' is the enthalpy from the condenser inlet which is different from the h_2, the enthalpy from compressor outlet. Because there is around half meter tube connecting from compressor outlet to the condenser inlet, the enthalpy will drop and this two points' data are different.

Then we can get the COP 5.61

Then we can get the COP for this condition.

There are two ways to find the change of condenser's refrigerant enthalpy. First one is from the air flow of the condenser. Second one is from the refrigerant flow.

From the air:

$$\Delta h = \int_{T_1}^{T_2} C_p dT = C_p \Delta T \tag{5.3}$$

Where the C_ρ is constant which is 1.008 $J/Kg \cdot K$

We can get the Δh which is 13.18 Btu/lb.

From the refrigerant:

$$\Delta h = h_3 - h_2 \tag{5.4}$$

Which is 13.65 Btu/lb.

Similarly, we can find the Q_h for the condenser also in two ways.

From the air:

$$Q_h = 2 * \pi * (\frac{r}{2})^2 * V * \rho * \Delta h \tag{5.5}$$

The r is the radius of the condenser's fan and there are two fans for condenser. V is the velocity of the air flow. ρ is the density of the air. Then we can find the Q_h is 6276.17 watt.

From the refrigerant:

$$Q_h = w_h * COP \tag{5.6}$$

Where the w_h come from the measurements of compressor about the voltage and electricity which is W = E*I. Then we can get the Q_h is 6655.32 watt.

To find the $\Delta(T_{lm})_h$, from the theory as below

$$\Delta(T_{lm})_h = \frac{(T_\infty - T_o) - (T_\infty - T_i)}{\ln\left[\left(T_\infty - T_o\right)/\left(T_\infty - T_i\right)\right]} \tag{3.11}$$

We can get the equation for our system by substituting as below:

$$T_\infty = T_4, T_o = T_{e,o}, T_i = T_{e,i}$$

Where the T_4 is the suction of compressor; $T_{e,o}$ is the refrigerant outlet of evaporator; $T_{e,i}$ is the refrigerant inlet of the evaporator

Then, we can find the log mean temperature which is 77.01 K.

Known $\Delta(T_{lm})_h$, from the first law, we can find

$$\bar{U}A = \frac{Q_h}{\Delta T_{lm}} \tag{5.7}$$

Then we can find the $\bar{U}A_s$ which is 1389.88 W/K

To find the mass flow rate of the refrigerant went through the condenser, we can do the process as below:

$$\dot{m}_{ref} = \frac{w_n}{h_2 - h_1} \tag{5.8}$$

Which is 125.57 g/s.

After this, by making the same procedures for the evaporator, we can find the parameter for the evaporator.

Table 5.2 Analyzed data sheet

Enthalpy of h2 REF	142.08 Btu/lb
Entropy of h2 REF	0.2637 Btu/lb °R
Enthalpy of h1 REF	128.90 Btu/lb
Entropy of h1 REF	0.2713 Btu/lb °R
Enthalpy of h3 REF	46.4 Btu/lb
Entropy of h3 REF	0.0958Btu/lb °R
COP	5.61
Δh (Condenser from air)	13.2270 Btu/lb
Q_h (Condenser from air flow)	6276.16 watt
Δh (Evaporator from air)	5.2670 Btu/lb
Q_c (Evaporator from air flow)	4900.36 watt
Δh (Condenser from REF)	13.18 Btu/lb
Q_h (Condenser from COP)	6655.32 watt
$\Delta(T_{lm})_h$	4.79 K
$\left(\bar{U}A\right)_h$	1389.88 w/k
$\Delta\left(T_{lm}\right)_c$	61.87 K
$\left(\bar{U}A\right)_c$	61.8655
COP (Overall system)	5.5513

Table 5.2 shows the whole data of the analyzed result from the experimental data.

You can find the rest fifteen sets of the result in the appendix.

5.3 Infrared Data

5.3.1 Infrared Imaging

While running the data acquisition system, we run the infrared imaging system. We separate it to three states: Cooling mode, Heating mode and Shutting down. Figure 5.5 shows the whole system temperature changes when we turn on to the cooling mode. The color shows the degree of the temperature. When the temperature turning colder the color getting darker. Figure 5.5 shows the whole system temperature changes when we turn from cooling mode to heating mode. Figure 5.4 shows the system shutting down from heating mode. For the heating part, the parts get hotter the color turns brighter.

Figure 5.4 System window for infrared imaging

In the following infrared images, the part in the camera corresponds to Figure 5.4 which is the main part from the equipment including: refrigerant cycle, evaporator and condenser.

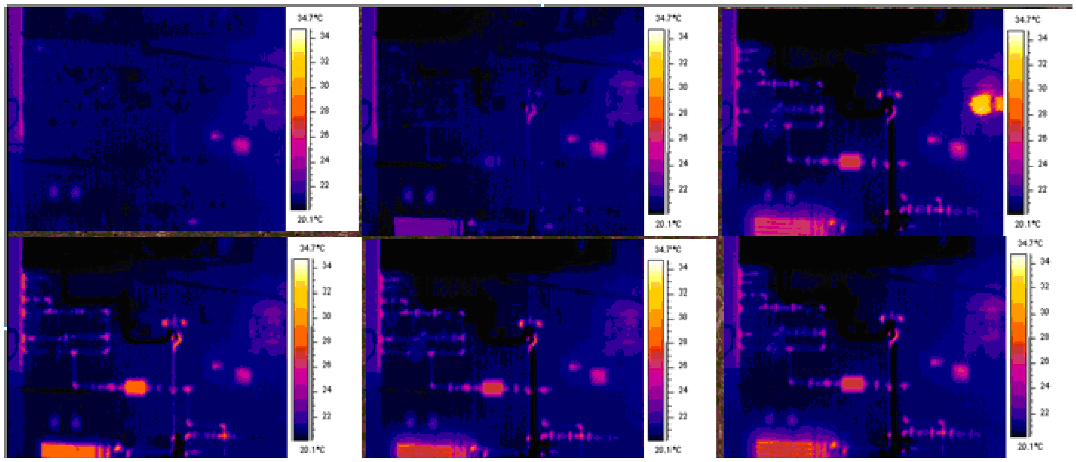


Figure 5.5 Infrared imaging from start to cooling mode for five minutes

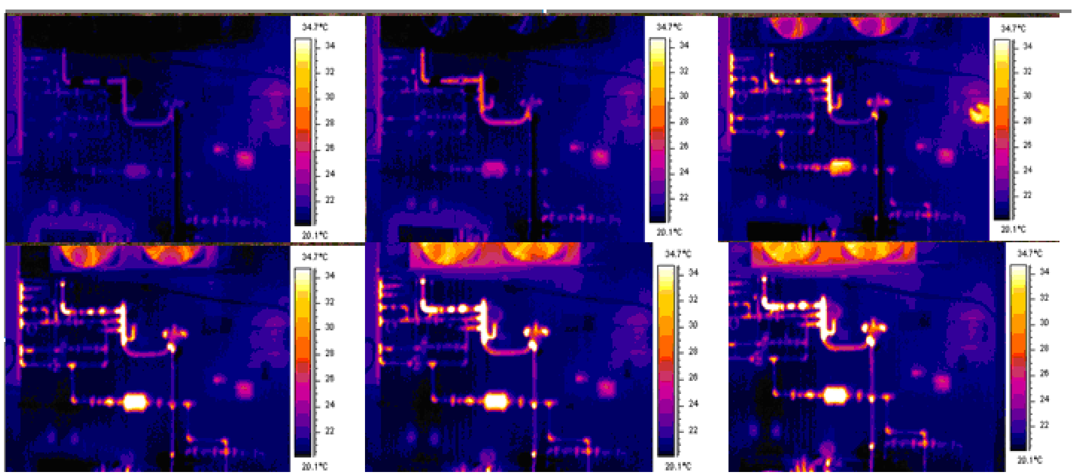


Figure 5.6 Infrared imaging from cooling mode to heating mode for ten minutes

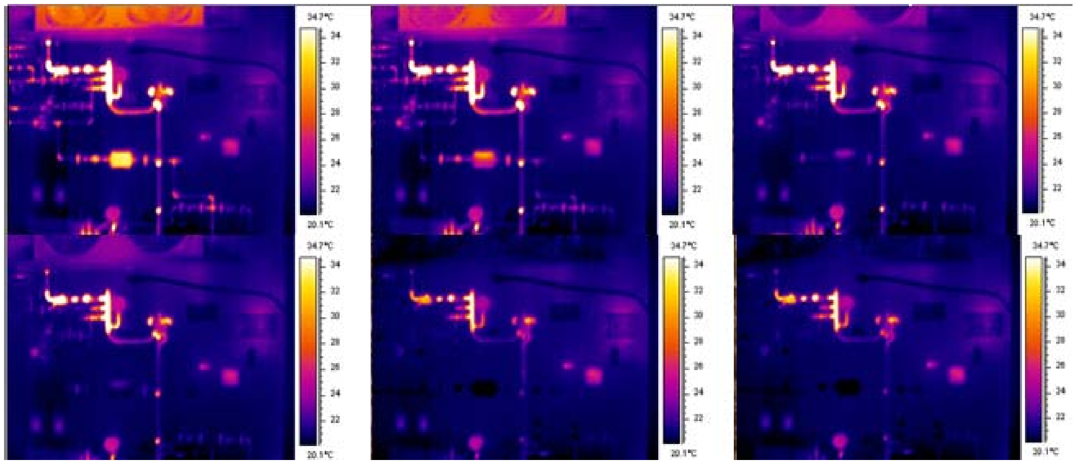


Figure 5.7 Infrared imaging from heating mode to shutting down for five minutes

We can see the state all in the P-H diagram in Figure 4.19.

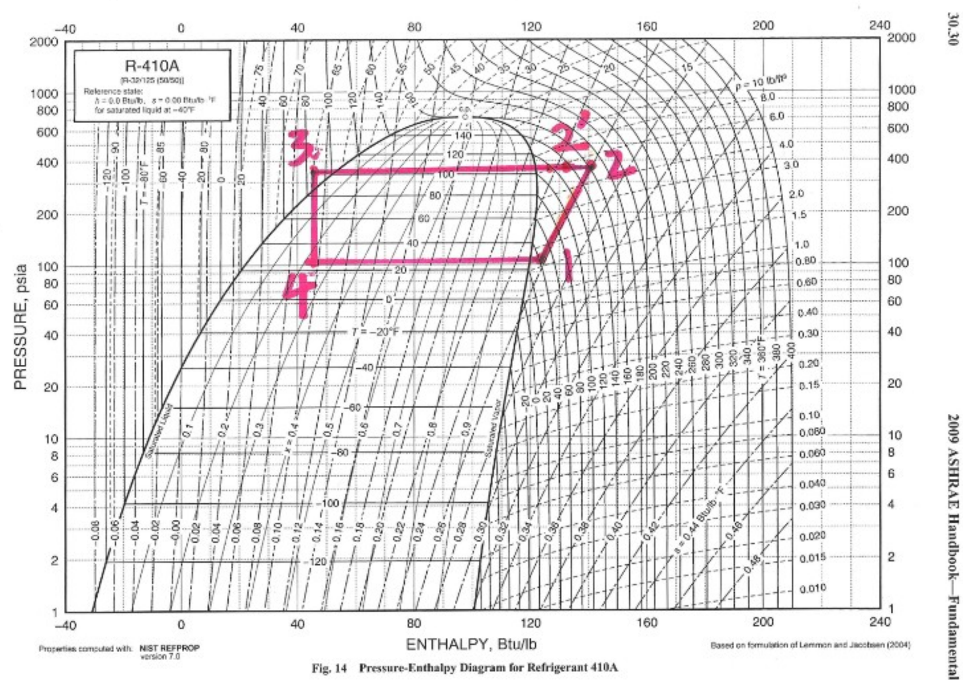


Figure 5.8 P-h diagram based on the experimental data

In figure 5.8, we can understand the whole cycle easily. From 1 to 2, the refrigerant goes through the compressor which makes the temperature and pressure booth up. From 2 to 3, the refrigerant goes through the evaporator with temperature drop and blow out hot air. Then the refrigerant goes through the evaporator from 4 to 1 blowing out cold air. From 2, through 2 to 3, it is the part for condenser which corresponds to figure 5.6. From 4 to 1, it is the evaporator part corresponding to figure 5.1.

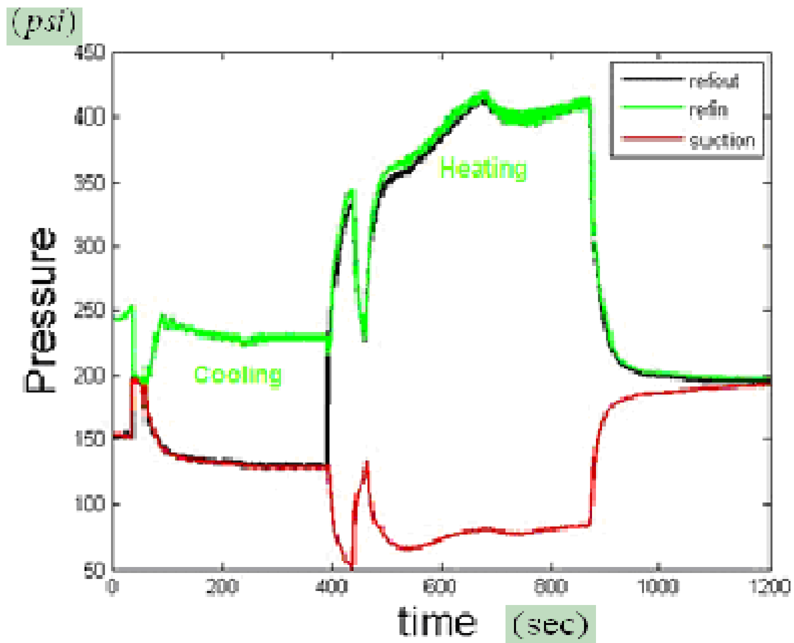


Figure 5.9 Pressure versus time for 20 minutes

For the air temperature both from the condenser and evaporator, we find the graph in figure 5.9. In the three different states, the condenser and evaporator exchange once in the whole process. So, from the graph we can see that, the air temperature of upper fans goes down first and goes up when we change to the heating mode. For the under fans near the compressor, the air temperature goes to the opposite direction.

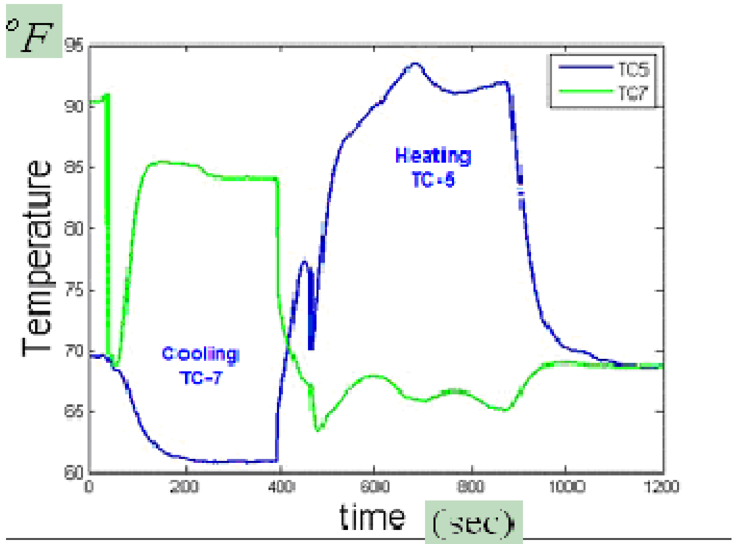


Figure 5.10 Air temperatures versus time for 20 minutes

For the temperature from different point of the cycle, the trends are all match Figure 1.2 cycles. Under the reversing cycle, the cooling parts turn to hot. We can find this from figure 5.10 and figure 5.11.

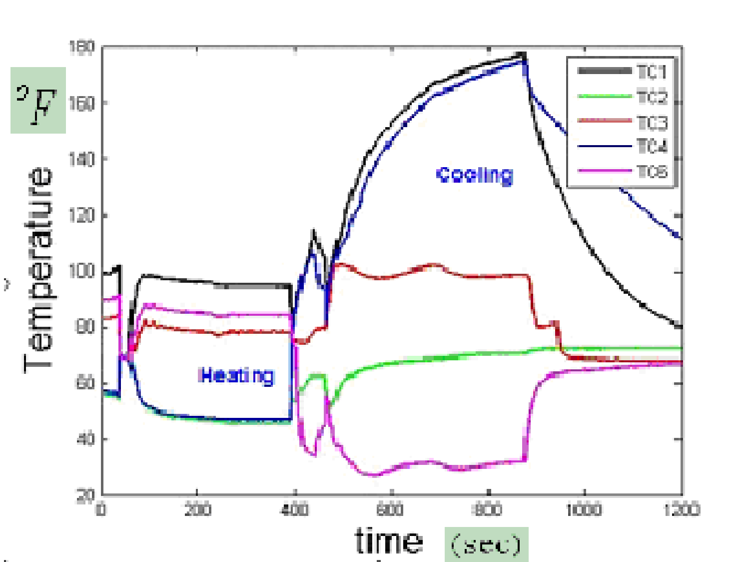


Figure 5.11 Refrigerant temperature versus time for 20 minutes

From the data acquisition system, we can find the trend for each point. There are three states in both figure 5.9 and figure 5.10. The three states are system in cooling mode, heating mode and shutting down. From above three ways, we can decide that we have to keep the system working at least 15 minutes before we record the data, because the system will go to the steady state in 15 minutes.

5.3.2 Overall data analyze

From the sixteen sets of the data, we can choose some of them to analyze the whole system in different ways.

Table 5.3 shows the connection between COP and the ambient temperature under four different compressor frequencies.

Table 5.3 COP versus temperature

45 Hz				
Temperature	44.2 °F	50.2 °F	53.8 °F	61.5 °F
COP	6.904	9.429	12.487	6.636
50 Hz				
Temperature	44.6 °F	50.2 °F	53.9 °F	62.6 °F
COP	8.316	8.77	10.73	6.219
55 Hz				
Temperature	44.8 °F	50.3 °F	55.4 °F	62.6 °F
COP	7.902	10.588	7.009	5.598
60 Hz				
Temperature	44.5 °F	51.4 °F	55.7 °F	62.6 °F
COP	6.525	8.256	7.255	5.21

By plotting Table 5.3, we can get Figure 4.10 as below:

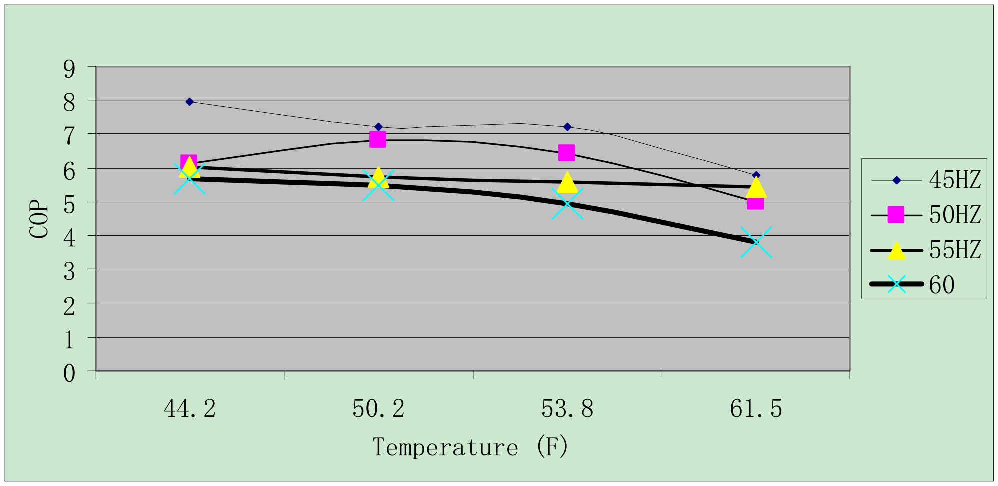


Figure 5.12 COP versus temperature under different frequencies

Table 5.12 shows the relationship between the COP and compressor frequency. From the figure, we can find that while the temperature increase, the COP decreases. Also, the COP will drop when the compressor frequency goes up. From here, we can find that, the more power we input, the lower COP we get which means we lost more energy. After we did the uncertainty analysis, it is ± 0.23 for COP.

Table 5.4 COP versus frequency

44 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
COP	6.904	8.316	7.902	6.525
50 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
COP	9.429	8.77	10.588	8.256
55 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
COP	12.487	10.730	7.009	7.255
60 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
COP	6.636	6.219	5.598	5.21

By plotting Table 5.4, we can get Figure 5.13 as below:

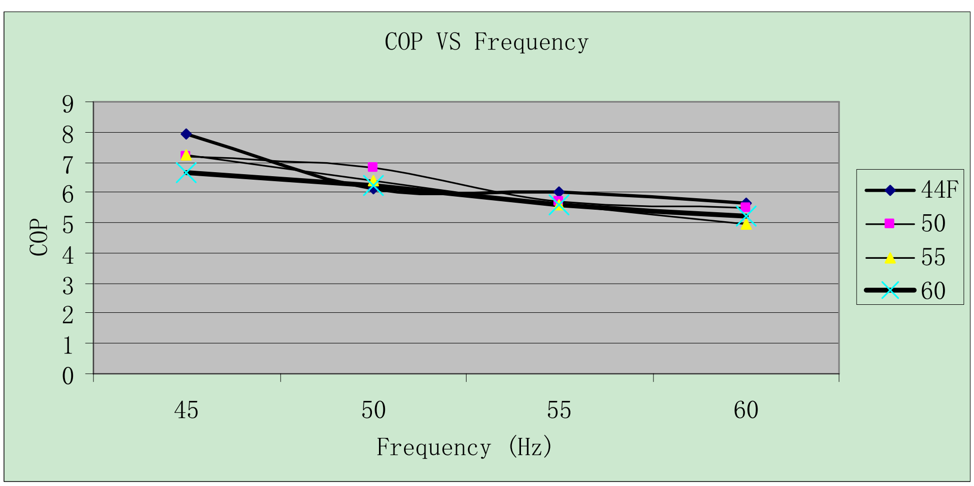


Figure 5.13 COP versus frequency under different temperatures

Table 5.5 shows the relationship between the Q_h and temperature

Table 5.5 Q_h versus temperature

45 Hz				
Temperature	44.2 °F	50.2 °F	53.8 °F	61.5 °F
Q_h (Condenser)	5788.88 W	7789.608 W	8288.026 W	7574.866 W
50 Hz				
Temperature	44.6 °F	50.2 °F	53.9 °F	62.6 °F
Q_h (Condenser)	7906.031 W	8573.256 W	11341.69 W	10220.60 W
55 Hz				
Temperature	44.8 °F	50.3 °F	55.4 °F	62.6 °F
Q_h (Condenser)	4672.009 W	11169.69 W	8315.867 W	10039.59 W
60 Hz				
Temperature	44.5 °F	51.4 °F	55.7 °F	62.6 °F
Q_h (Condenser)	6190.379 W	6608.843 W	6773.611 W	6773.61 W

By plotting Table 5.14, we can get Figure 5.6 as below:

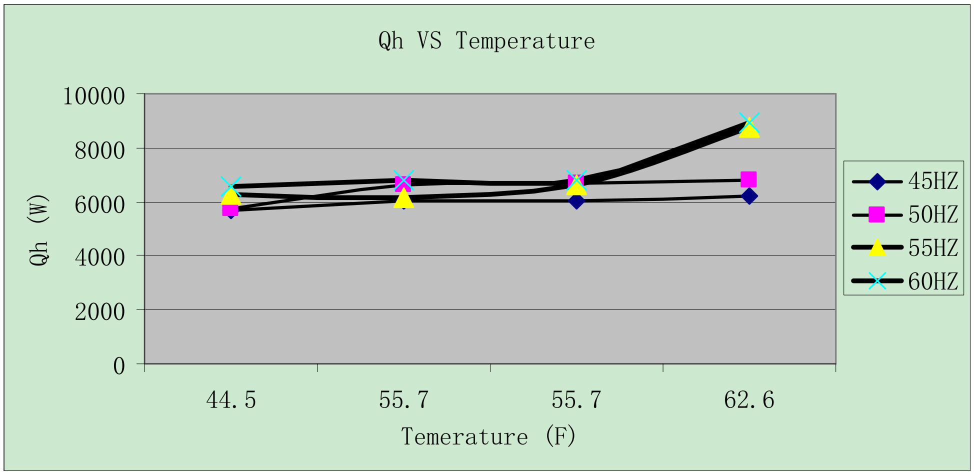


Figure 5.14 Q_h versus temperature under different frequencies from condenser

On the other hand, we can also find the relationship between the $Q_{h,air}$ and the temperature. $Q_{h,air}$ comes from the fan of condenser. There are a few differences from the way above, since there are some heat loss and leakage around the fans in the system. From here, we can see that, even the lower COP we get, however, the larger capability of the system when we increase the compressor frequency.

Table 5.6 shows the data.

Table 5.6 $Q_{h,air}$ versus temperature from condenser

45 Hz				
Temperature	44.2 °F	50.2 °F	53.8 °F	61.5 °F
Q_h (Condenser)	4203.26 W	4253.26 W	5639.45 W	4653.26 W
50 Hz				
Temperature	44.6 °F	50.2 °F	53.9 °F	62.6 °F
Q_h (Condenser)	6223.50 W	5246.24 W	6142.11 W	5160.72 W
55 Hz				
Temperature	44.8 °F	50.3 °F	55.4 °F	62.6 °F
Q_h (Condenser)	6702.23 W	5505.40 W	6276.16 W	5692.11 W
60 Hz				
Temperature	44.5 °F	51.4 °F	55.7 °F	62.6 °F
Q_h (Condenser)	7051.70 W	5744.77 W	6726.17 W	5955.41 W

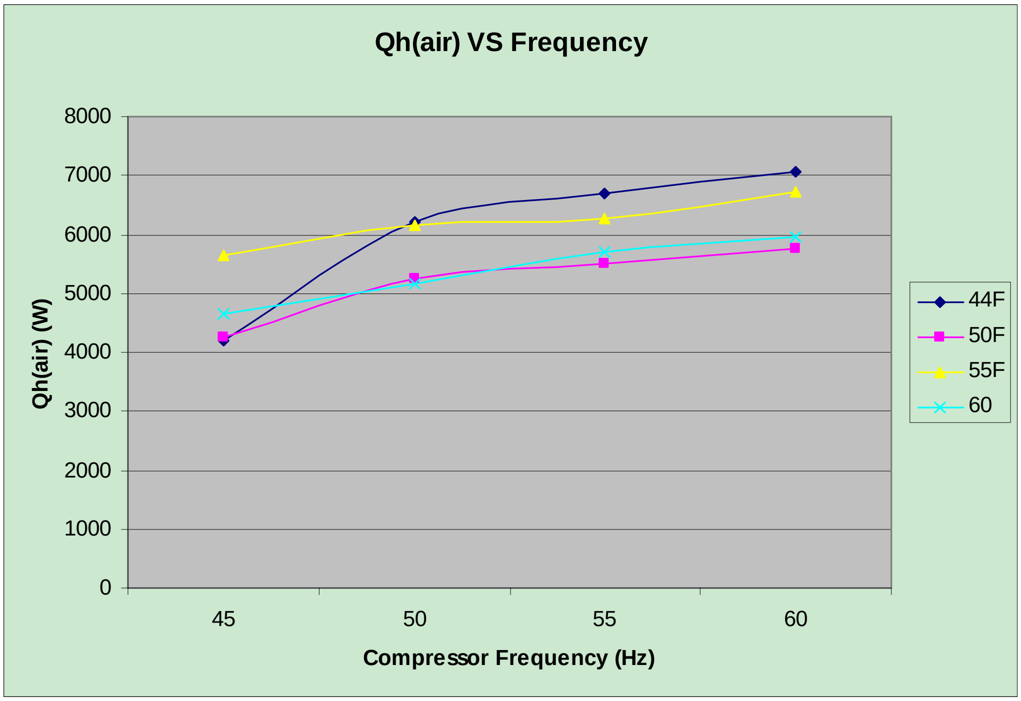


Figure 5.15 $Q_{h,air}$ versus compressor frequency under different outdoor temperatures from condenser

From Figure 5.15, we can find that the $Q_{h,air}$ increase when the compressor frequency goes up. It says that we can get better capability of this system when we input more power.

Table 5.7 Log mean temperature difference versus Compressor frequency under different outdoor temperatures from condenser

44 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	14.85 K	18.7 K	19.38 K	21.24 K
50 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	14.74 K	17.93 K	19.07 K	20.7 K
55 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	16.47 K	18.98 K	20.01 K	21.43 K
62 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	15.46 K	17.22 K	18.71 K	19.97K

Table 5.7 shows the relationship between the log mean temperature differences and frequency of compressor for the condenser. Figure 5.16 shows the trend of this.

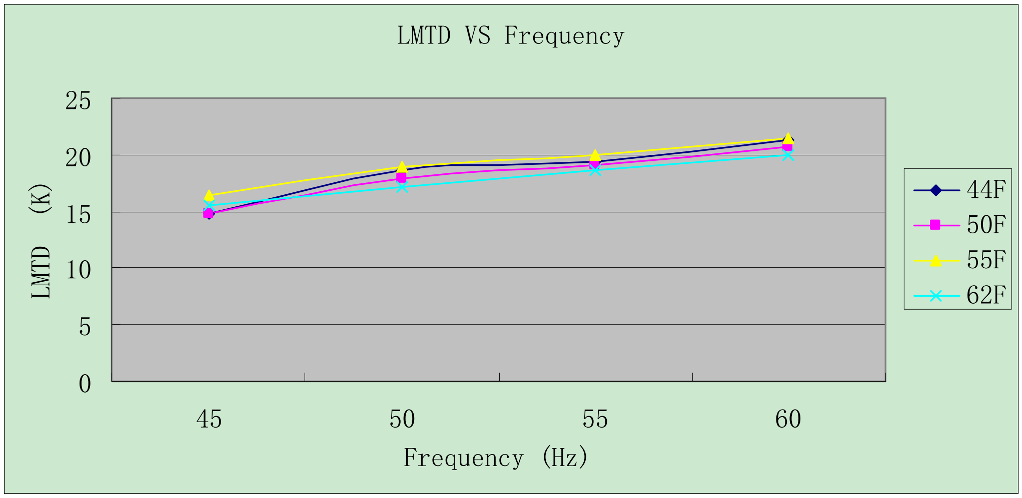


Figure 5.16 Log mean temperature difference versus compressor frequency under different outdoor temperatures from condenser

We can find that when the frequency increases, the log meant temperature difference increases too which means that the temperature increases more when we input more power.

In Table 5.8, it shows the trend of overall heat transfer coefficient based on the compressor frequency for the condenser. Figure 5.17 correspond to this.

Table 5.8 Overall heat transfer coefficient versus compressor frequency from condenser

44 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
UA	283.05 W/K	332.75 W/K	345.75 W/K	331.99 W/K
50 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
UA	285.22 W/K	308.6W/K	288.76 W/K	277.48 W/K
55 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
UA	342.33 W/K	323.65 W/K	313.58 W/K	313.82 W/K
62 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
UA	300.91 W/K	299.67 W/K	304.24 W/K	298.24 W/K

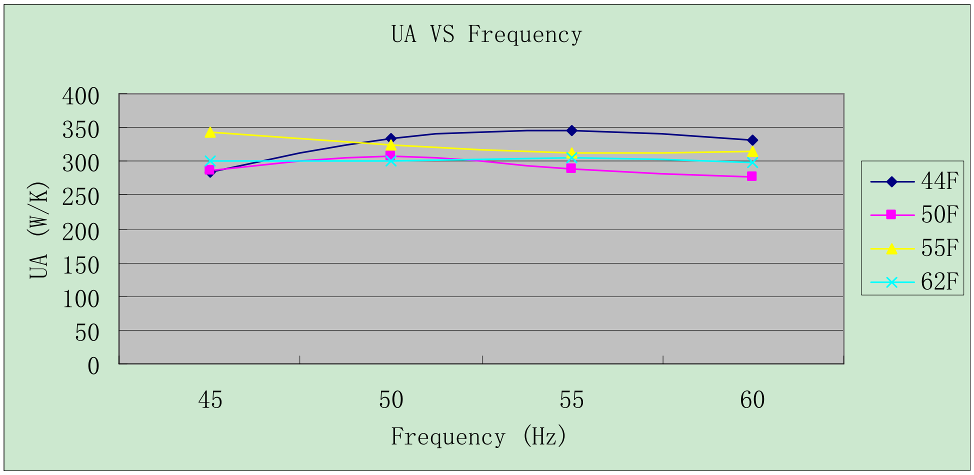


Figure 5.17 Overall heat transfer coefficient versus compressor frequency under different outdoor temperatures from condenser

In Table 5.9, it shows the data of the mass flow rate of the refrigerant based on the different compressor frequency at different temperature.

Table 5.9 Mass flow rate versus compressor frequency

44 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
$\dot{m}$	0.119 kg/s	0.108 kg/s	0.118 kg/s	0.124 kg/s
50 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
$\dot{m}$	0.113 kg/s	0.125 kg/s	0.115 kg/s	0.127 kg/s
55 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
$\dot{m}$	0.114 kg/s	0.125 kg/s	0.125 kg/s	0.128 kg/s
62 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
$\dot{m}$	0.102 kg/s	0.100 kg/s	0.165 kg/s	0.100 kg/s

Figure 5.18 shows the trend of the data.

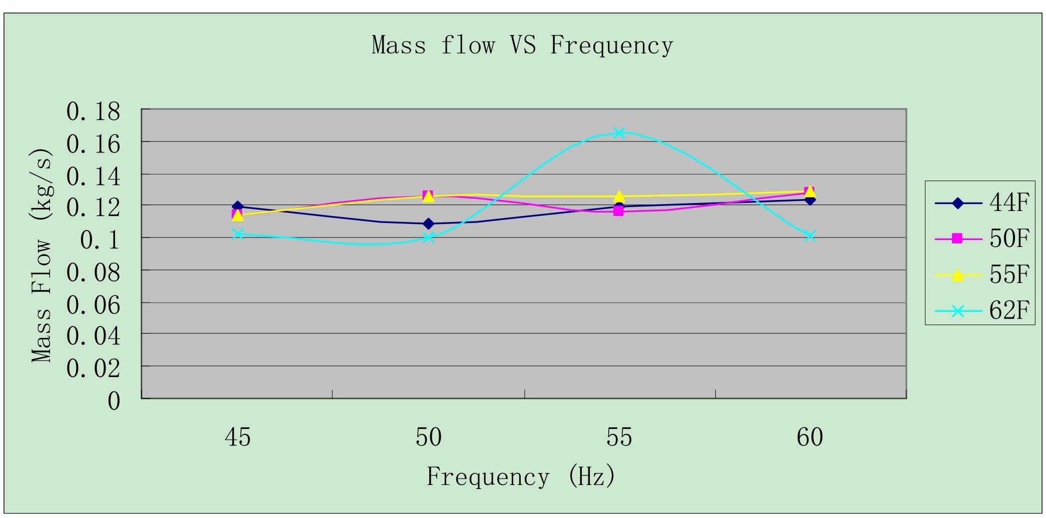


Figure 5.18 Mass flow rate versus Compressor frequency

To compare the heating mode, the below data will show the situation of the evaporator, the cooling part of the system.

Table 5.10 Log mean temperature difference versus compressor frequency under different outdoor temperatures from evaporator

44 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	13.73 K	13.95 K	13.96 K	15.2 K
50 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	14.78 K	14.27 K	16.7 K	16.87 K
55 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	14.58 K	16.27 K	17.15 K	16.83 K
62 °F				
Frequency	45 Hz	50 Hz	55 Hz	60 Hz
LMTD	14.57 K	16 K	16.63 K	17.61 K

Figure 5.19 shows the trend of this set of data.

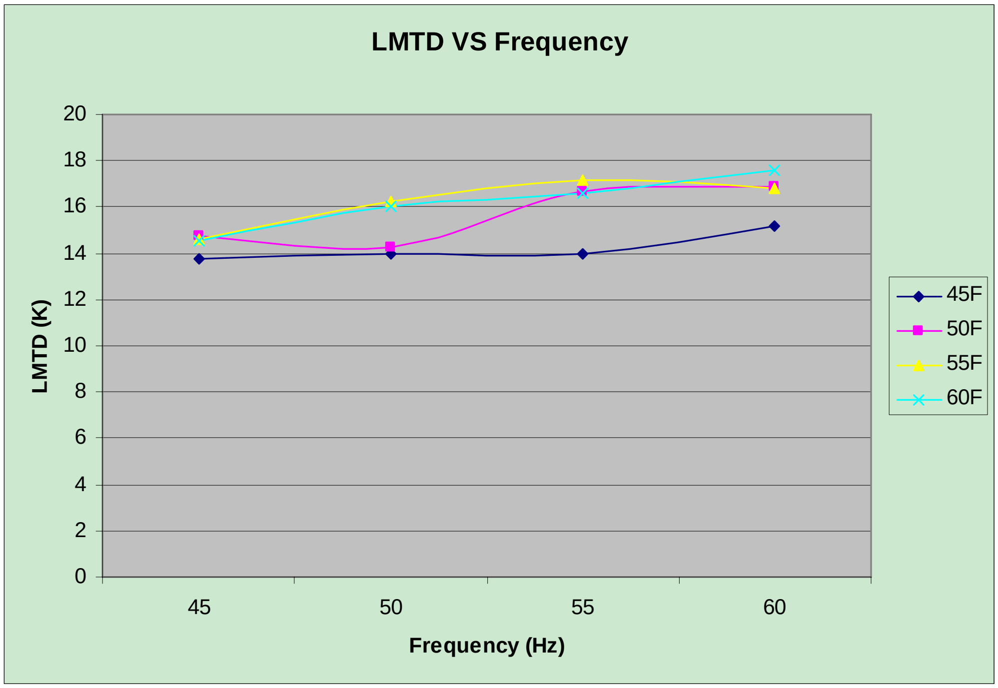


Figure 5.16 Log mean temperature difference versus compressor frequency under different outdoor temperatures from evaporator

From this, we can find that, the trend of LMTD from heating and cooling are the same. However, LMTD from cooling mode is a little lower than heating mode which corresponds to the first law.

5.4 P-H diagram analysis

Figure 5.19 shows that the refrigerant flow rate for the given compressor speeds tends to decrease as the outdoor temperature decreased. When the compressor frequency increases, the capability of the system increases.

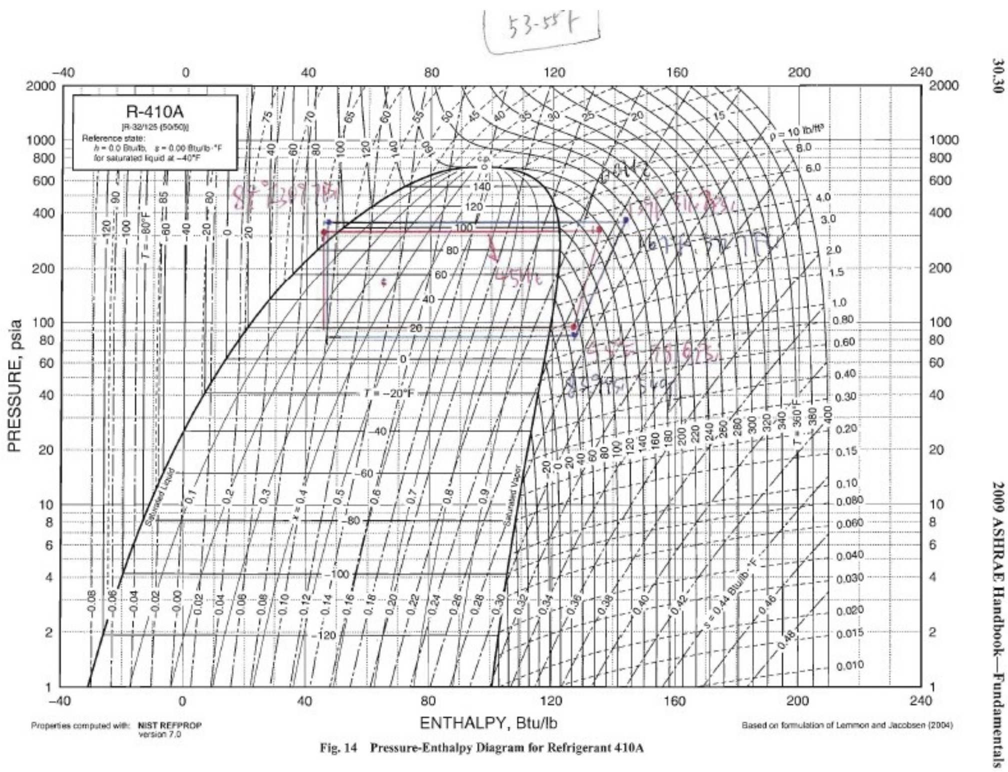


Figure 5.19 Pressure- enthalpy diagram for different frequencies

Figure 5.19 shows that the refrigerant flow rate for the given compressor speeds tended to decrease as the outdoor temperature decreased. When the compressor speed was 45 Hz, the temperature went through the evaporator 44°F and 143°F for the condenser. The temperatures shift to 29°F and 130°F when the outdoor temperature was 45°F to maintain the equation balance. This caused a drastic reduction in refrigerant flow in addition to the decrease in the decrease in compressor power consumption because of the lower evaporating temperature. Consequently, heating capacity decreased. (Bong H. Kim, 2005)

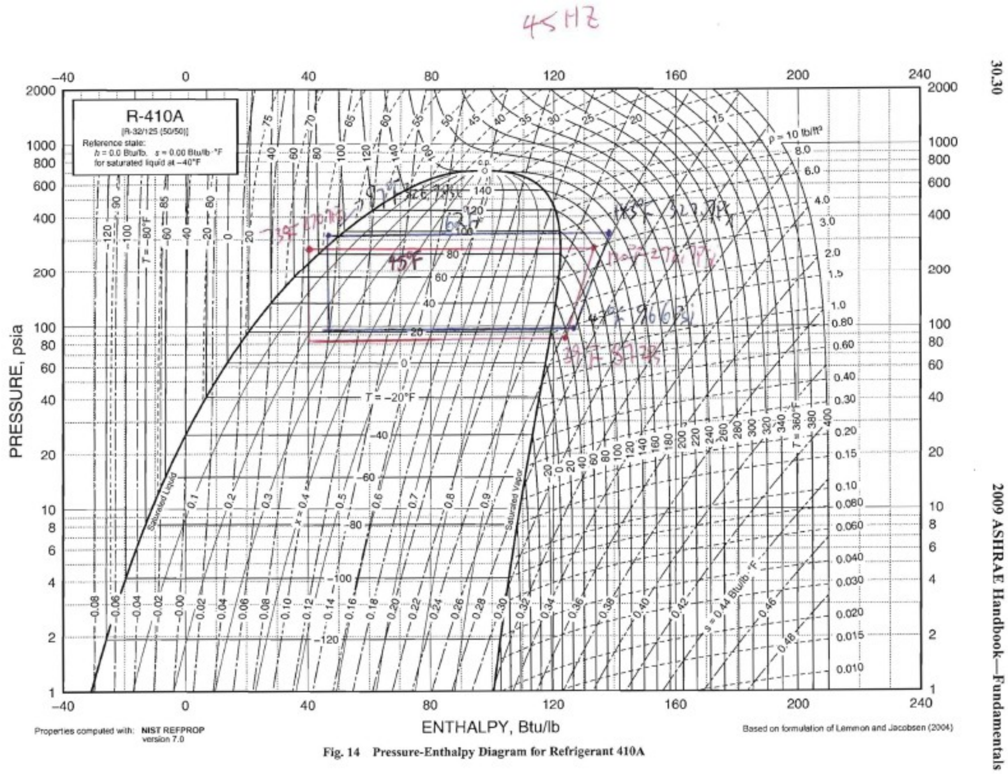


Figure 5.20 Pressure-enthalpy diagram for different outdoor temperatures

From above two P-h diagram, we can find that there are some limits for our system. If we want to keep our system working, the outdoor temperature has to be between the two horizontal lines. We can increase the capability of the system by input more power to the compressor. However, when the difference between the outdoor temperature and the condenser refrigerant temperature become close, the COP will decrease. That is the reason people do not use the heat pump in most of the cold states but gas.

CHAPTER 6

6 CONCLUSIONS AND RECOMMENDATIONS

The objective of this project was to investigate the effects of various power input to the variable speed compressor based on different ambient temperatures. Experimental and analytical studies were carried out by parametric studies to show the best coefficient of performance for different ambient temperatures. We analyzed the system performance with the first law and second law of thermodynamics as well as the heat exchanger theory.

The heat pump and air conditioning system based on a variable speed compressor will help to save more energy than traditional constant speed system. The variable speed compressor system will provide a more comfortable and steady indoor temperature than the traditional system which is controlled by on-off switch.

6.1 Conclusion

An experimental investigation was conducted to examine the effect of electronic flow control on the heating performance of a variable speed heat pump. Performance enhancement was analyzed in view of superheat, heating capacity, and EER. (C. Aprea,2009)

The main objective of this paper is to identify for variable speed compressors the current frequency that optimizes the energy and economy aspects. It is important to identify it because it is not sure that the compressor speed decrease results in an energy saving when compared to the thermostatic control. The optimum frequency corresponds to a defined cooling/heating load has been determined.

To avoid the unacceptable uncertain data of the enthalpy and entropy, we choose to find them by calculating from upper and under limits of the experimental data. For the data from LOCAL and REMOTE, after we compared them to the measurement from third part such as thermocouple, the REMOTE data is more accurate.

Since the size and the structure of the fans are not regular, the data of the air flow rate is not ideal which cause the small differences between the air and refrigerant. We decide to make the solution by refrigerant.

Theoretically, the COP will keep going up when the compressor's frequency increases. However, in the experiment, we find that when the compress increase to some point, and then the COP decrease. There are two reasons causing this situation. First, since the TXV and TEX control the mass flow rate of the refrigerant which has large influence on the system? Second, the limit of the system is one ton which limits the COP. An experimental analysis has been realized varying the frequency and the cooling load to determine the compressor performances when its speed is continuously regulated.

The experimental data have been elaborated to obtain the trend of refrigerant mass flow rate, input power and cooling capacity in terms of evaporation temperatures for fixed compressor frequencies (45, 50, 55 and 60 Hz). Moreover, from the experimental analysis it has been observed that for different working conditions the ratios of the refrigerant mass flow rate, the input power and the cooling capacity values at any frequency with respect to the values at the basic frequency are practically constant. In other words these ratios are independent from the evaporation and condensation temperatures and depend only on the compressor frequency, and so can be represented by means of second-order functions of the frequency. These equations allow determining the optimum frequency capable of obtaining the highest energy saving in comparison with the working at the basic frequency for each working condition, once the evaporation and condensation temperatures and then the cooling load are fixed.

6.2 Recommendation

Since this equipment is not ideal for research, the data is not as accurate as desired, especially the air flow rate. The power supply unstable electricity since it is for civil use. The steady state can be in a very small range but not enough. The ideal setup is to put the system in a stable power supply space and put the evaporator outside the window.

We need a mass flow rate meter for the refrigerant which can make a better collation for the data in a different way. By using this, people can easily understand the refrigerant’s state change for the different mode in each component.

To avoid the damage to the compressor, the frequency can not go below 45 Hz which maybe cause the refrigerant phase as mixture. For the industrial, people need a large scale of the compressor such as 100 Hz or 200Hz. People can change the capability of the compressor for different use.

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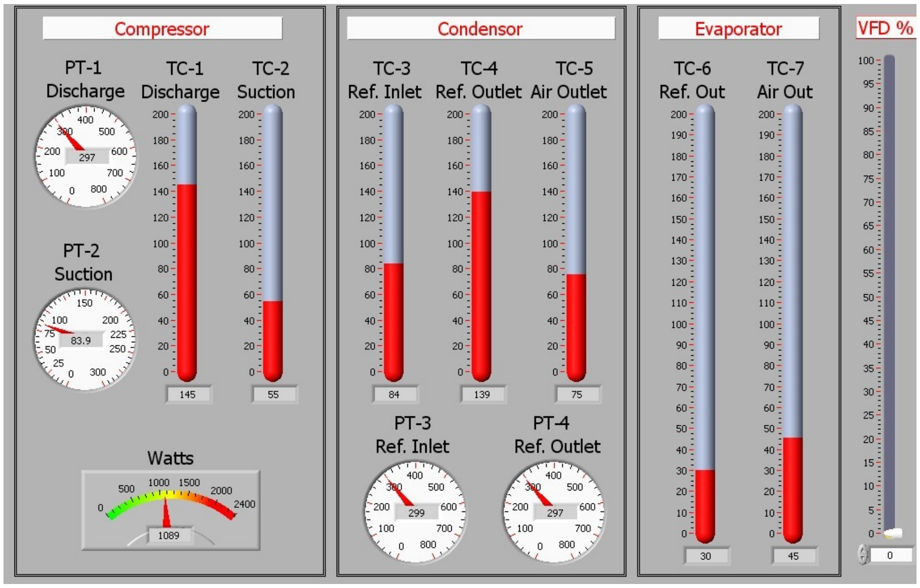
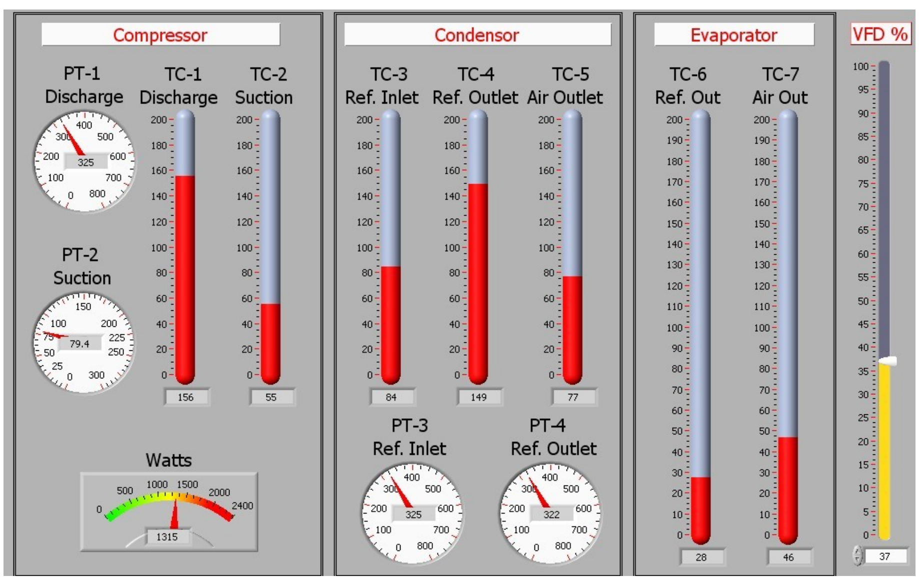
APPENDIX A: SIXTEEN SETS OF THE EXPERIMENTAL DATA

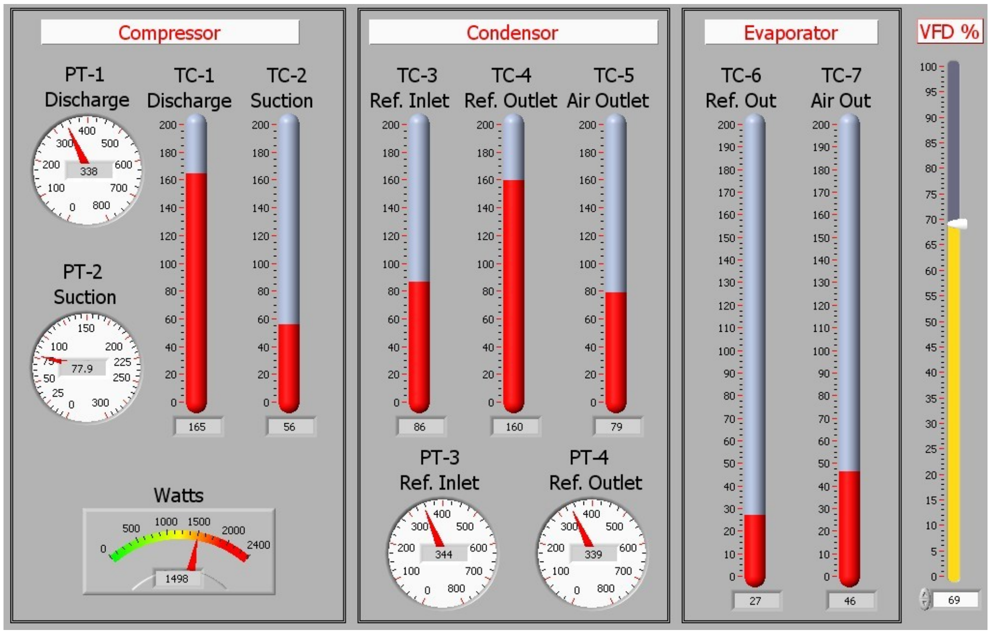
Room temperature: 53.8 °F, Compressor Frequency: 45 HZ



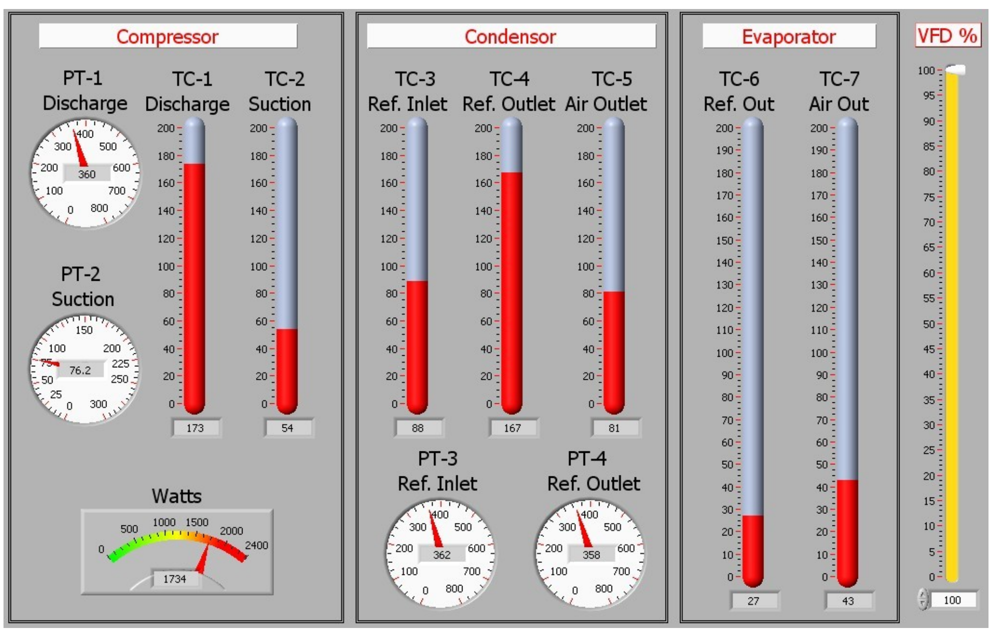
Room temperature: 53.9 °F, Compressor Frequency: 50.4 HZ



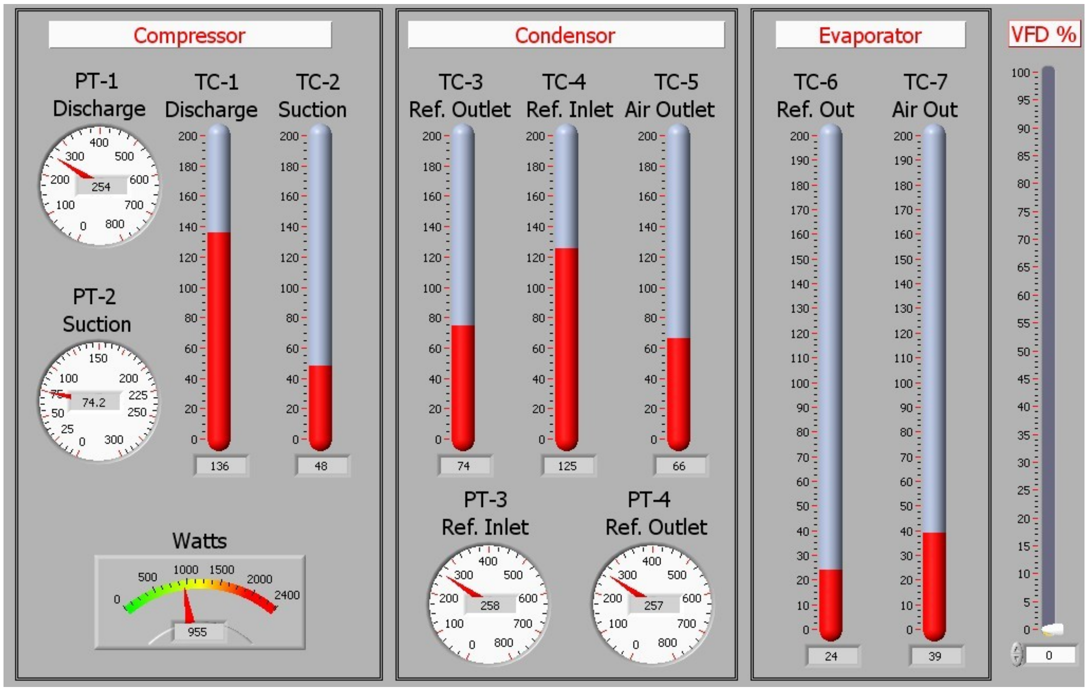
Room temperature: 55.4 °F, Compressor Frequency: 55.2 HZ



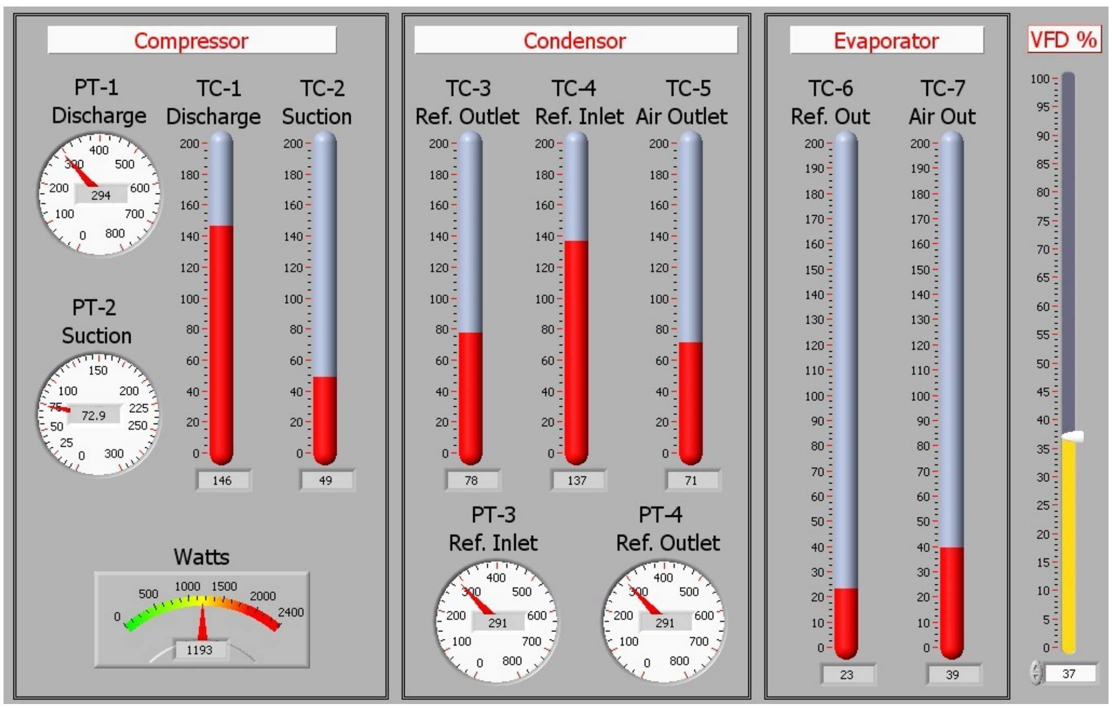
Room temperature: 55.7 °F,   Compressor Frequency: 59.8 HZ



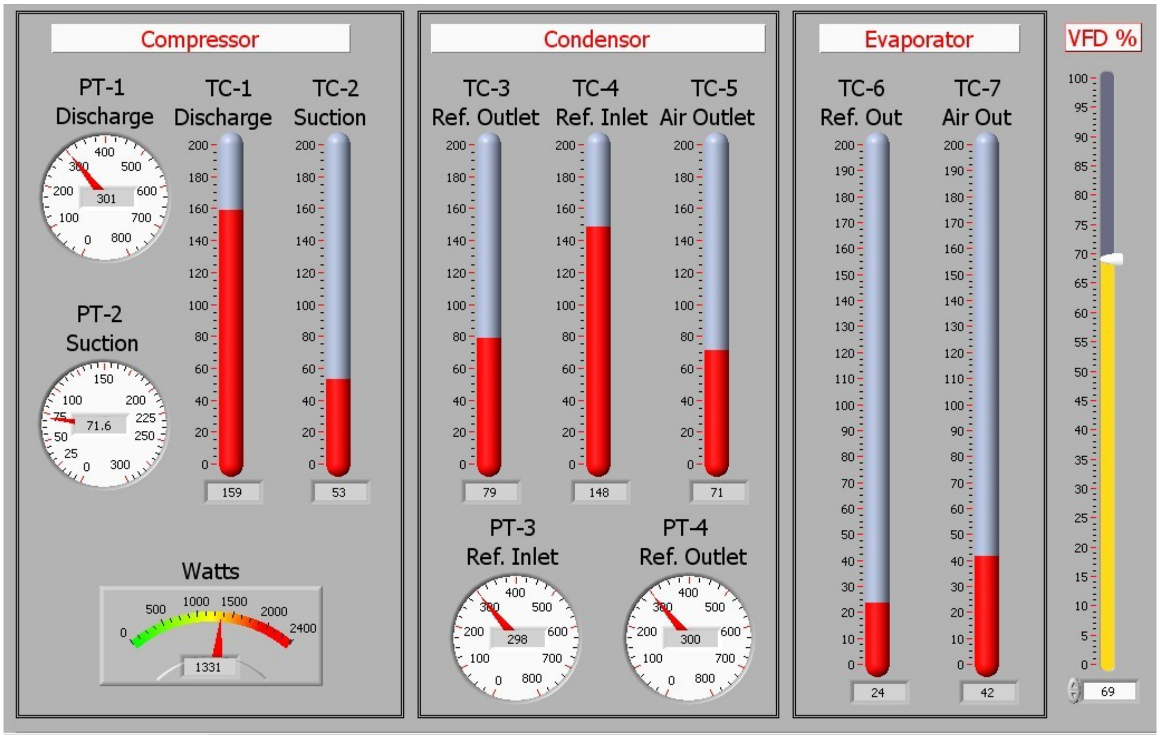
Room temperature: 50.2 °F,   Compressor Frequency: 45 HZ



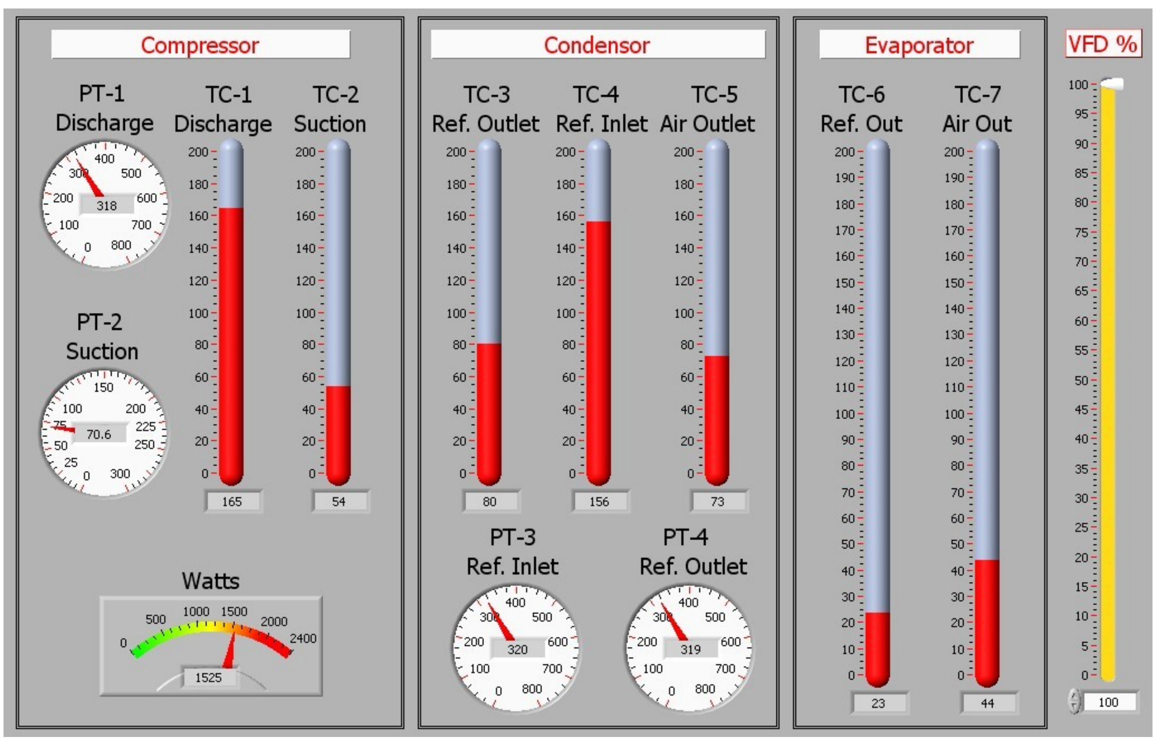
Room temperature: 50.7 °F,   Compressor Frequency: 50.4 HZ



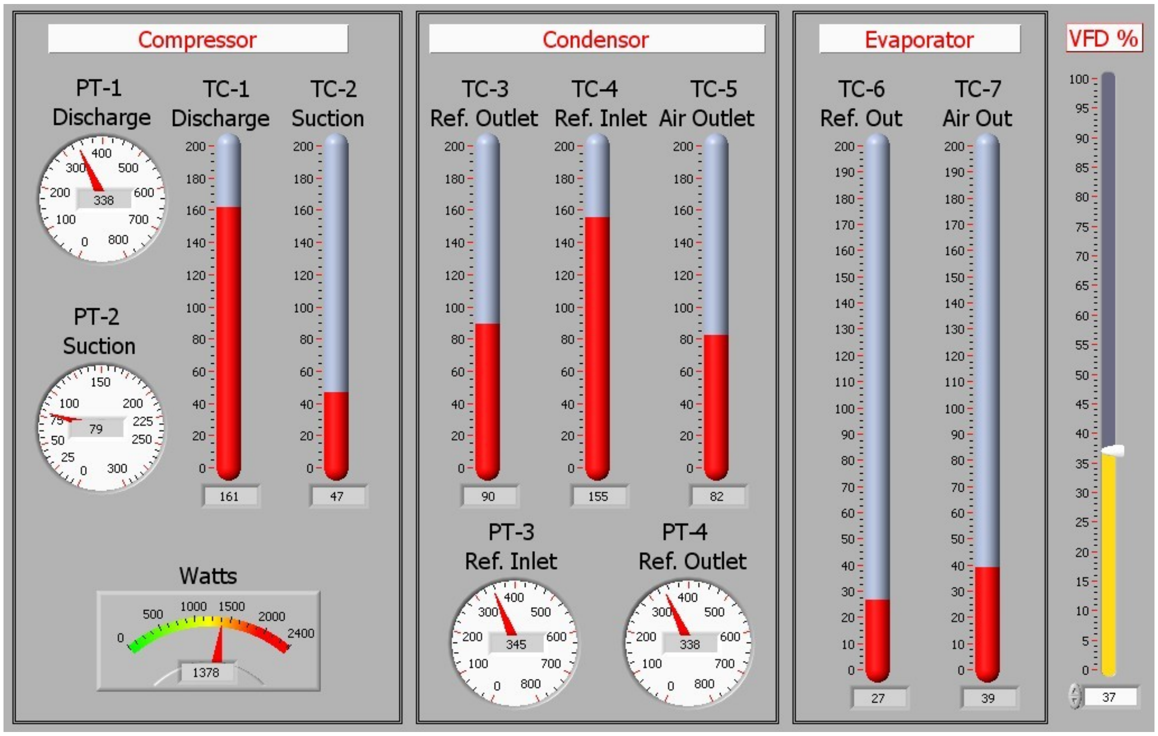
Room temperature: 50.3 °F,   Compressor Frequency: 55.2 HZ



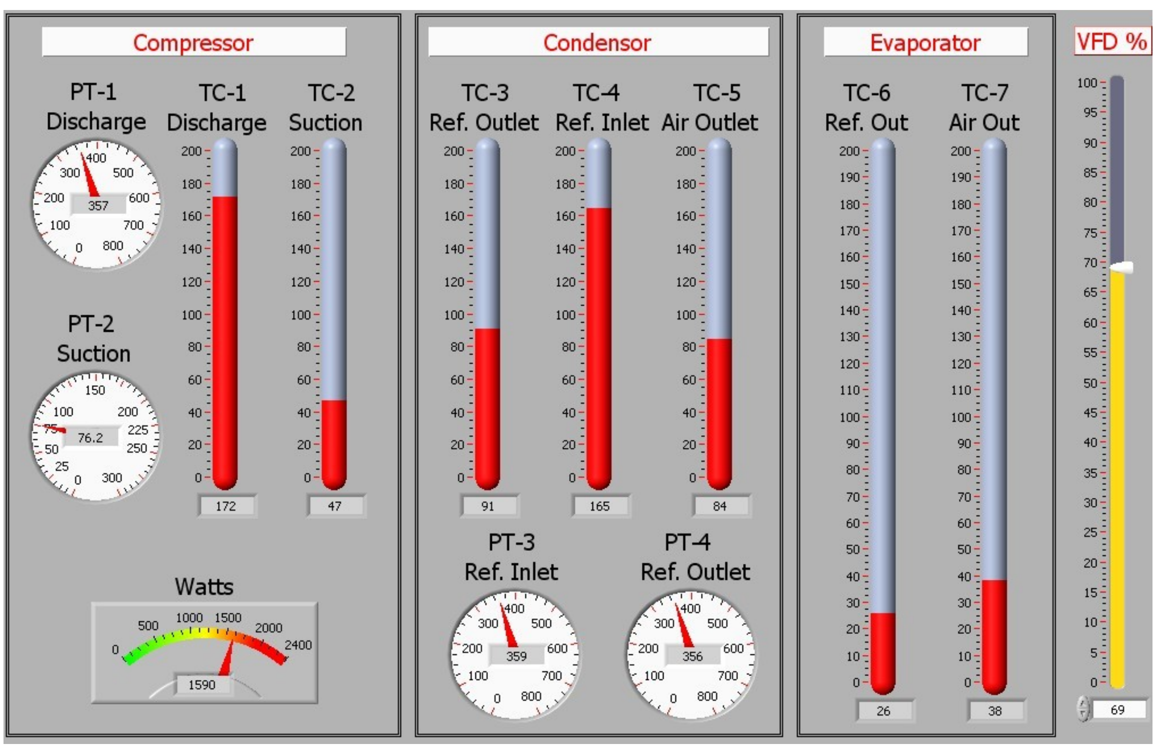
Room temperature: 51.4 °F,   Compressor Frequency: 59.9 HZ



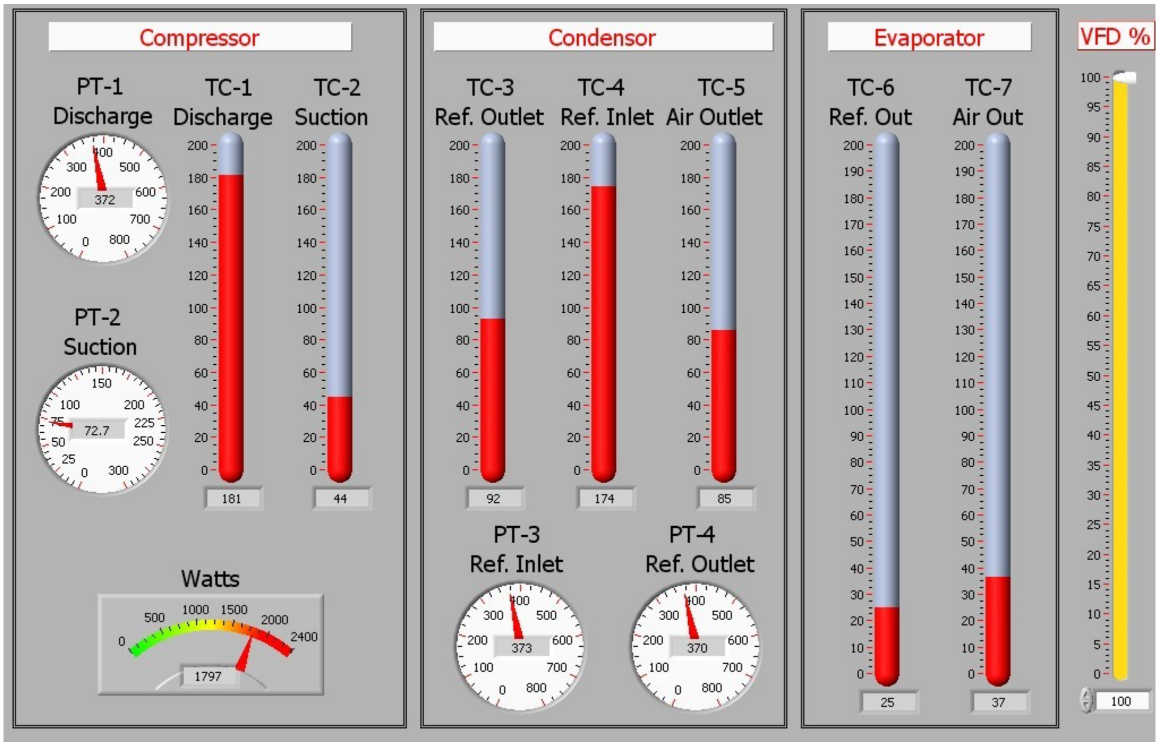
Room temperature: 61.5 °F,   Compressor Frequency: 45 HZ



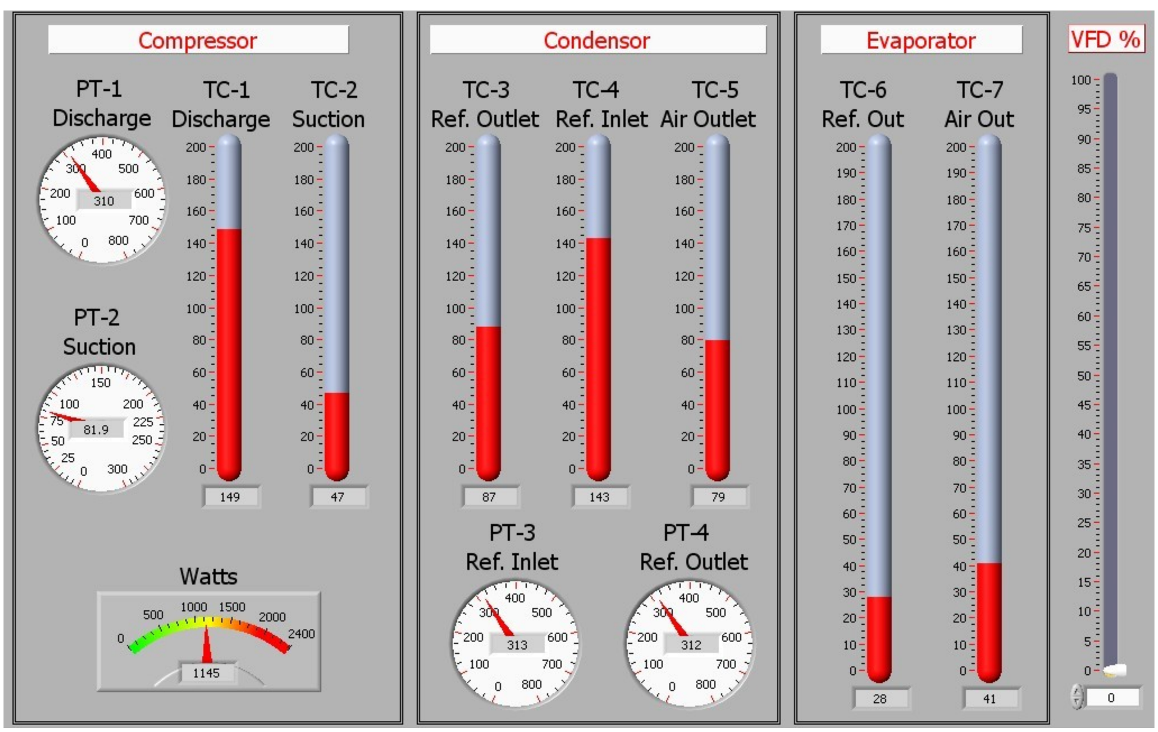
Room temperature: 62.6 °F,   Compressor Frequency: 55.3 HZ



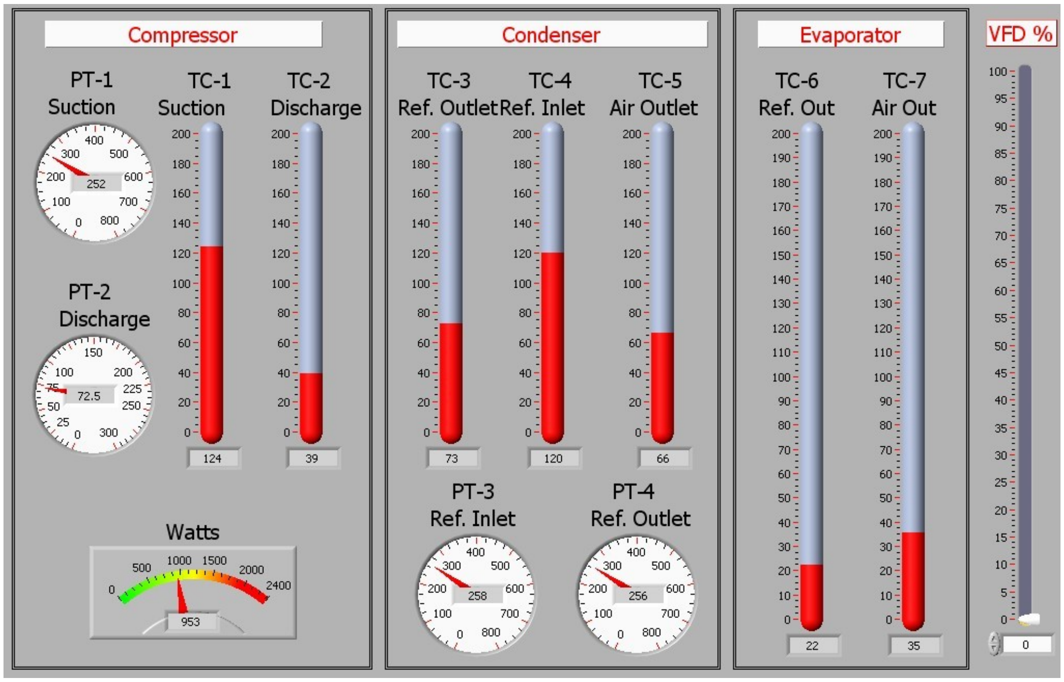
Room temperature: 62.6 °F,   Compressor Frequency: 55.3 HZ



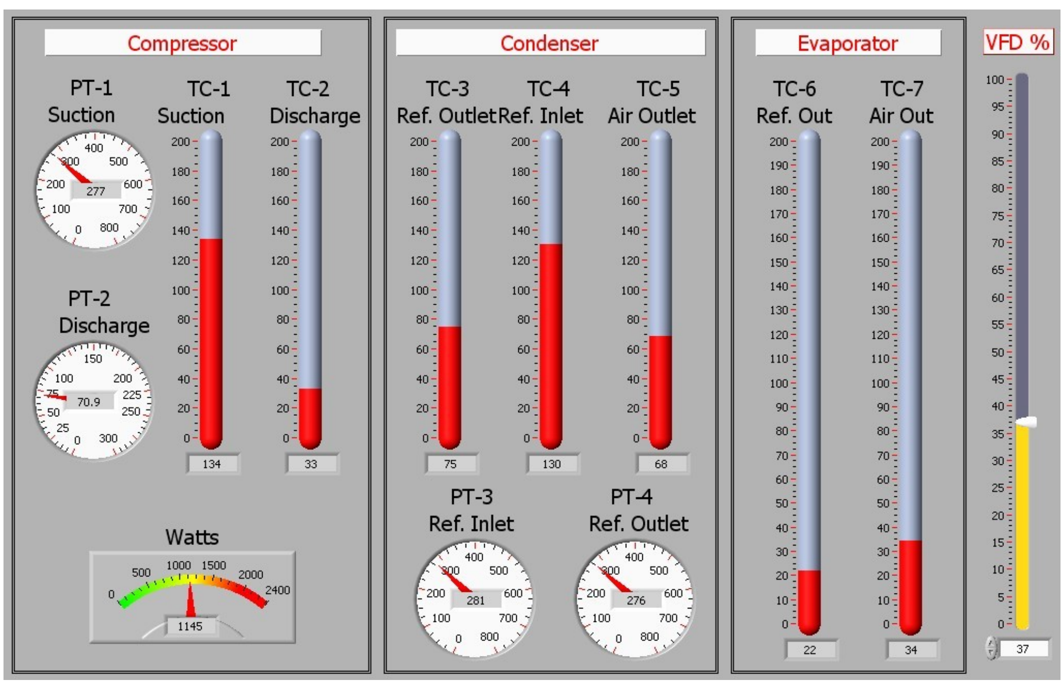
Room temperature: 62.6 °F,   Compressor Frequency: 59.9 HZ



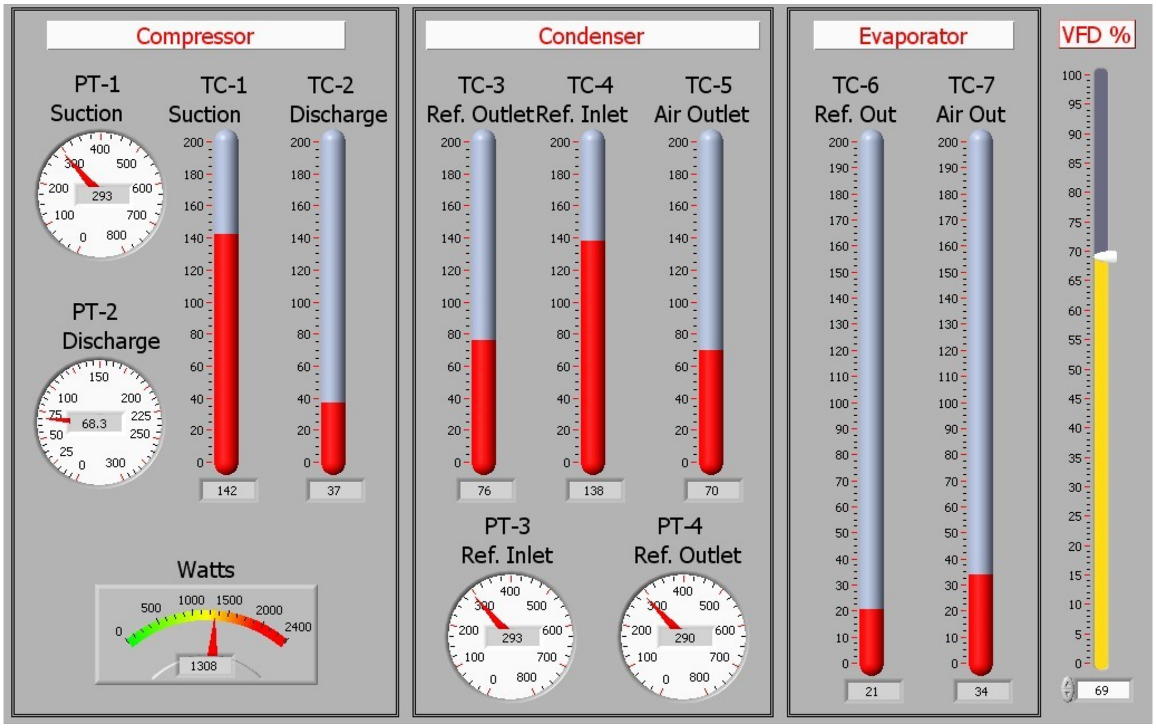
Room temperature: 44.2 °F,   Compressor Frequency: 45 HZ



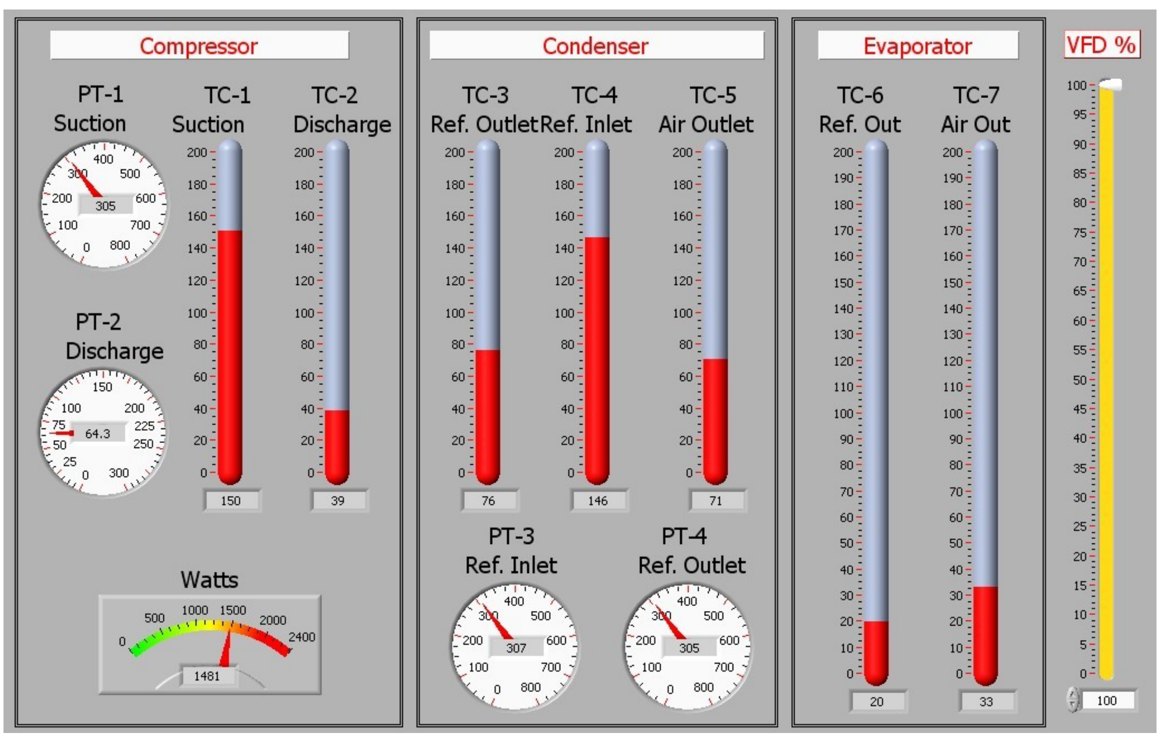
Room temperature: 44.6 °F,   Compressor Frequency: 50.3 HZ



Room temperature: 44.8 °F,   Compressor Frequency: 55.2 HZ



Room temperature: 44.5 °F,   Compressor Frequency: 59.8 HZ



APPENDIX B: 410-A REFREGERANT THERMAL PROPERTIES

	R-22	R-407C	R-410A
Critical Temperature (°C)	96.2	86.1	72.0
Critical Pressure (Bar a)	49.9	46.3	47.7
Saturation Pressure at 50°C (bar a)	19.4	22.1	30.6

	R-22	R-410A
Density (kg/cu.m.)	1247	1130
Viscosity (μPa.S)	196	147
Thermal Conductivity (W/m.K)	0.090	0.108

	R-22	R-410A
Density (kg/cu.m.)	28.8	41.8
Viscosity (μPa.S)	12.0	12.9
Thermal Conductivity (W/m.K)	0.0101	0.0136

APPENDIX C: MATLAB CODE

```
refom = xlsread('outlet');
refim = xlsread('inlet');
 sucm = xlsread('suction');
 tc1m = xlsread('tc1');
 tc2m = xlsread('tc2');
 tc3m = xlsread('tc3');
 tc4m = xlsread('tc4');
 tc5m = xlsread('tc5');
 tc6m = xlsread('tc6');
 tc7m = xlsread('tc7');

 refo = refom(:,2);
   to = linspace(0, 20*60,2462)';
 refi = refim(:,2);
   ti = linspace(0, 20*60,2461)';
  suc = sucm(:,2);
   ts = linspace(0, 20*60, 2460)';
  tc1 = tc1m(:,2);
   t1 = linspace(0, 20*60,2461)';
  tc2 = tc2m(:,2);
   t2 = linspace(0, 20*60,2462)';
  tc3 = tc3m(:,2);
   t3 = linspace(0, 20*60,2463)';
  tc4 = tc4m(:,2);
   t4 = linspace(0, 20*60,2463)';
  tc5 = tc5m(:,2);
   t5 = linspace(0, 20*60,2464)';
  tc6 = tc6m(:,2);
   t6 = linspace(0, 20*60,2464)';
  tc7 = tc7m(:,2);
   t7 = linspace(0, 20*60,2464)';

   figure(1);
   plot(to,refo,'black','LineWidth',2);
   hold on;

   plot(ti,refi,'green','LineWidth',2);
   hold on;

   plot(ts,suc, 'red','LineWidth',2);
   hold on;
   legend('refout','refin','suction');
   xlabel('time','FontSize',20)
   ylabel('Pressure','fontsize',20)


   figure(2)
   plot(t1,tc1,'black','LineWidth',2);
   hold on;

   plot(t2,tc2,'green','LineWidth',2);
   hold on;
```

```
plot(t3,tc3,'red','LineWidth',2);
hold on;

plot(t4,tc4,'blue','LineWidth',2);
hold on;

plot(t6,tc6,'m','LineWidth',2);
hold on;
legend('TC1','TC2','TC3','TC4','TC6')
xlabel('time','FontSize',20)
ylabel('Temperature','fontsize',20)

figure(3)
plot(t5,tc5,'blue','LineWidth',2);
hold on;

plot(t7,tc7,'green','LineWidth',2);
hold on;
legend('TC5','TC7')
xlabel('time','FontSize',20)
ylabel('Temperature','fontsize',20)
```