MODELING AND FUZZY CONTROL OF MODERN REFREGERATION SYSTEM

by

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To my parents
Reza Afzali & Maryam Barat Zadeh
To whom I owe everything I am today
Abstract

Current typical refrigeration systems work with on-off controllers in each of their components and their expansion valves are regulated mechanically which is quite inefficient. This issue resides in the core design of the system which is based on the maximum possible load. This leads to high power consumption, low performance and an inability to precisely maintain the required temperature. In addition, this leads to a high number of on-off cycles and a high maintenance cost. Refrigeration cycles are highly dynamic and non-linear and moreover difficult to control. Due to this reason, this thesis focuses on a new control design that leads to precise temperature control. This design reduces power consumption and saves energy using fuzzy-PID controllers and variable speed drives. Different components such as compressors and heat exchangers are controlled and mechanical expansion valves are replaced with electronic units. The simplified model has been derived for refrigeration systems with vapor compression cycle, semi hermetic reciprocating compressor, direct expansion (DX) evaporator and air-cooled condenser. The refrigerant fluid used in the model is R-22. This model was validated and tested using experimental data acquired from a refrigeration system running in the field with the same proposed components. The on-off and fuzzy-PID controllers are simulated and the results are compared to demonstrate the efficiency and power-saving capability of the proposed system using a fuzzy-PID Controller.

Keywords: Refrigeration system, fuzzy controller, PID controller, variable speed drive, on-off controller, vapor compression cycle model
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Chapter 1: Introduction

Refrigeration is the process of transferring heat from a low temperature environment to a higher temperature environment [1]. Refrigeration has many applications in different industries. This has led to many different refrigeration cycles which we will review briefly in this chapter.

1.1 Background

Artificial refrigeration has existed for more than two centuries, from the days that natural ice was shipped from the New England states throughout the Western world to the current date [2]. Nowadays, the refrigeration industry has expanded in many industries and has numerous applications such as cooling of electronic devices, air conditioning, and food cooling. Refrigeration also has a multi-disciplinary character involving a combination of disciplines such as civil, mechanical, food, and chemical engineering [3, 4].

Refrigeration itself consists of many different types, namely [5, 6]:

- Vapor-compression refrigeration system
- Absorption refrigeration system
- Air-standard refrigeration system
- Thermoelectric refrigeration and
- Magnetic refrigeration

In a Vapor-Compression Refrigeration System, there are several main components which are:

- compressor
- condenser
- evaporator and
- throttling device

There are some other auxiliary pieces of equipment which are associated with these major components such as refrigeration line, piping, refrigeration capacity control, receiver and accumulators. Each of these components have different types,
but the most common refrigeration systems consist of a semi-hermetic reciprocating compressor, an expansion valve as a throttling device, an air cooled condenser and a fin-type evaporator. This setup is mainly used in environments with a high degree of humidity [6].

Early controls and refrigerant flow capacity use a mechanical device such as a pressure switch used to turn on-off compressor and condenser, mechanical thermostat used for maintaining a cold room temperature in an upper and lower preset limits and a mechanical expansion valve used to control refrigerant flow through the refrigeration system. Through the passage of time, these mechanical control systems evolved into an electrical and electronic system and then into an embedded electronic system in order to increase the accuracy, reliability and flexibility of the system [7]. In the latest research, variable speed drives with different types of controllers were implemented on major components of refrigeration systems and tested on experimental setups, which are discussed more in-depth in the literature review.

1.2 Thesis Contribution

The main contribution of this thesis can be summarized as follows:

- Designing and validating of the complete mathematical model of the vapor compression cycle
- Implementing a conventional on-off controller of the designed refrigeration model and simulating it on Matlab/Simulink
- Designing a fuzzy-PID controller for all major components of the refrigeration process and simulating each controller on Matlab/ Simulink
- Analyzing the effect of adding the fuzzy-PID controller to each component of the vapor compression cycle on the overall improvement of the refrigeration system
- Comparing the performance of a fuzzy PID and on-off controller in all aspects of set-point tracking, power consumption, and system efficiency.
1.3 Thesis Outline

In Chapter 1, an introduction of refrigeration systems and their control history is presented. Some of the current system deficiencies are outlined, and reasons are given for why a new controller with variable speed drives is proposed.

In Chapter 2, a literature review related to different types of controllers on different components of the refrigeration system is presented. The controllers’ contribution to improving each of these components is presented.

In Chapter 3, models of refrigeration system components are presented. These models were coupled together to enable the simulation of the dynamic behavior of the system on MATLAB Simulink, and in the end the results gathered from this simulation were validated.

In Chapter 4, a conventional on-off control is designed and implemented on the simulated model. The results are presented after the design completion and validated using experimental results.

In Chapter 5, an introduction to the fuzzy PID controller is given. The design details are presented and the controller is implemented. The results after the simulation are outlined.

In Chapter 6, a comparison is made between conventional on-off controllers and the proposed fuzzy PID controller. Improvements that could be made on refrigeration set point tracking, performance, and lowering power consumption are discussed.

In Chapter 7, conclusions are drawn and a summary is presented.
Chapter 2: Literature Review

There are many studies on control methods of the vapor compression cycle. In this chapter, control methods for each subsystem and a combination of them are reviewed.

2.1 Compressor Controller

There have been many studies focused on reciprocating compressor performance. In compressor-varying frequencies, the operating frequency should be equal to or above 30Hz. This is to avoid lubrication failure due to oil splash. The result obtained from this paper is that the lower the frequency, the higher the coefficient of performance (COP). When the compressor was operated at 30Hz, the best performance was obtained and power consumption was reduced to 12% considering 95% efficiency for a variable speed drive [8]. This is due to higher mechanical efficiency at low revolution speed caused by lower friction. In addition, due to the positive trend of the volumetric and isentropic efficiencies and the smaller decrease of the electric motor efficiency (until 5%), as the speed decreased, the global compressor efficiency increased [9, 10].

There are some detailed studies on controlling the vapor compression cycle through a variable speed drive, some of which will be outlined here. The main reasons for implementing VFD and varying the speed of the compressor were to better track the temperature of the cold chamber, lower the power consumption, and increase the life cycle of compressor. The control method used was a proportional controller [11, 12] and a fuzzy PD controller [13, 14] on the experimental setup. Even in some situations, they had great success in saving energy and reducing power consumption about 13% [13]. In mathematical model analysis for multiple inputs single output (MISO), inputs were delivery pressure and compressor speed and mass flow rate for output. A fuzzy controller was used to maintain the mass flow rate on a preset value. The results showed that a fuzzy controlled MISO system is expected to perform well in a real vapor compression cycle [15].
2.2 Expansion Valve Controller

A metering device or expansion valve is one of the devices responsible for regulating mass flow in a refrigeration system. In a typical vapor compression cycle, regulation is done through mechanical expansion valves. The regulation happens according to the compressor suction pressure which has a direct relation to super heat temperature. In recent research, it has been found that performance of the system can be improved by replacing the mechanical expansion valves with electronic expansion valves (EEVs) and implementing a new controller method to improve the transient response of set point tracking. For this reason, fuzzy controllers were used to control the opening function of the EEV, along with a neural network to enable the online tuning of fuzzification and defuzzication of the fuzzy controller [16]. In other research, a Fuzzy-PID controller was used to control the function of opening the EEV. The fuzzy-PD controller was used to decrease the overshoot and rise time during the transient resonance period. A fuzzy-I controller was used to get the disturbance rejection and zero steady state error. This fuzzy-PID controller was compared with a conventional FLC and PID control and the results suggest that the fuzzy-PID controller performs much better than the conventional controller in all the aspects of control parameter and set point tracking [17].

2.3 Heat Exchanger Controller

Controlling heat exchangers (evaporators & condensers) are quite important in a refrigeration system, in addition to the controlling compressor and expansion valves. A significant performance gain can be achieved by simply controlling the fan speed of the heat exchangers. This achievement is due to maintaining the constant temperature difference between ambient and condenser coil temperature. Variable speed fans enable the system to run more flexibly and increase the efficiency of the entire system. In the case of condensers, regulating the condenser fan helps in achieving lower power consumption and steady state response. In the case of evaporators, controlling the speed of their fan can create another degree of freedom which can improve the transient response of the system [18]. In other research, heat exchanger fans were controlled by a simple linear model. The result showed that as the fan speed decreased, power consumption also decreased greatly. This happened because the power consumed by the fans is the cubic function of the fan speed. This concept
greatly affects the evaporators since the generated heat from the evaporator fan to the room is reduced. This heat reduction saves energy and lowers power consumption. Also, by decreasing the evaporator fans, less dehydration will occur which leads to less ice formation in the evaporator coils [19].

2.4 Combined Controller

In some research, controllers were used on more than one component. Most of the researchers believe that the best combination is using a controller on compressor speed and an expansion valve opening function. Implementing both controllers has a great impact on having an effective controller for heat exchanger fans. So, it is suggested to have heat exchanger fans for the speed controller in addition to the compressor and EEVs [17, 18]. Some of the research that has studied the combination of compressor and EEV controllers are reviewed [20-23]. In most cases, the speed of the compressor was varied in order to maintain the room temperature. For EEVs, the percentage of opening was varied to maintain the super heat temperature at the desired value [20-22]. An MIMO controller based on the linear quadratic Gaussian (LQG) method was implemented on a mathematical model and the results gathered were validated through experimental data. What it proved was that the controller performed well on reference tracking and disturbance rejection [20]. In another study, an independent PI controller was implemented to control the compressor and EEV. Findings showed more precise set point tracking and higher COP in comparison to the conventional on-off controller [21]. In comparing the conventional PID controller, fuzzy Controller and Artificial Neural Network (ANN) against each other, it was found that all of them were able to achieve satisfactory performance. PID was found to be more prominent and stable in the steady state response. Fuzzy control had lower power consumption compared to PID. ANN was the best controller in achieving minimum power consumption, a stable transient response, and a robust response for disturbance. However, ANN was also the most expensive controller to implement both in terms of hardware and software. It could be reasonable to implement if a high rate of disturbance were present in the system [22]. All mentioned controllers had a focus on room temperature reference tracking. In terms of evaporating pressure control, a fuzzy and ANN controller with an adaptive network based fuzzy Inference system (ANFIS) architecture and learning process were used. This controller excels in
performing online tuning, can maintain set points on desired values with a very quick response speed, small overshoot, and has a small steady state error [23].

Another method to gain control of the EEV opening, compressor speed, and evaporator fan input power is to implement feed forward single input single output (SISO) decentralized control design. This control method can maintain average pressure, differential pressure and average evaporator refrigerant temperature to achieve system coupling. This method was merged with a PI feedback controller in order to improve reference tracking and decrease RMS error, which leads to lower power consumption [24]. The last control method reviewed was an extremum seeking controller, which provides control of the entire vapor compression system and is designed for energy optimization of the vapor compression system. This is achieved by having different sets of inputs. The input in the system was the evaporator’s fan speed, the condenser’s fan speed, compressor frequency and EEV opening position. The outputs were electrical power consumption, room temperature, and state of the vapor compression machine. This controller managed to regulate the room temperature and at the same reduced the power consumption of the system.
Chapter 3: Modeling of Refrigeration Process

3.1 Introduction

One of the most common methods of refrigeration is the vapor compression system which transfers heat from the desired area to the outside environment by four major thermal processes [6]:

- Compression
- Condensation
- Expansion
- Evaporation

Figure 1 shows a simple single-stage vapor compression refrigeration system and its entropy and enthalpy diagram.

![Figure 1](image.png)

Figure 1 a) A basic vapor compression refrigeration system, b) its t-h diagram, and c) its log p-h diagram [5]

3.2 Compressor

The heart of a vapor compression refrigeration system is the compressor. Its purpose is to raise the pressure of the refrigerant to circulate it in the refrigeration cycle. The performance and reliability of the compressor are significant to the overall performance of the refrigeration system.

In general, compressors can be classified as positive displacement compressors or non–positive displacement compressors [5]. Non-positive displacement or dynamic compressors convert angular momentum of refrigerant vapor into pressure. Positive displacement compressors, like reciprocating compressors, increase the pressure of the refrigerant vapor by means of volume reduction. In this study, the reciprocating type compressor, which dominates the cold room industry, will be considered.
Reciprocating compressors can be further classified as hermetic, semi-hermetic and open. In a hermetic compressor, the motor and the compressor are sealed in a common housing. Hermetic compressors minimize leakage of the refrigerant, which is advantageous to the durability of the compressor. Cooling of the motor windings is accomplished using the suction vapor flow; as a result, the compressor will be more economical.

Semi-hermetic compressors are similar to the hermetic compressor in many aspects unless they are accessible for repair. In medium capacity refrigeration, semi-hermetic compressors are most popular. Figure 2 shows a drawing of a semi-hermetic variable speed reciprocating compressor, which will be considered in this study. Last of all, an open compressor includes two separate housings for the motor and for the compressor itself. Refrigerant leakage must be seriously considered in open compressors.

![Figure 2 Drawing of a semi-hermetic variable speed reciprocating compressor](image)

A simplified model of a reciprocating compressor is shown in Figure 3. Refrigerant vapor enters the compressor via a suction port. Due to the energy losses of the compressor components, the temperature of flowing refrigerant to the compressor increases. The refrigerant isentropically compressed to state. Finally, the refrigerant discharges by a throttling process from the compressor discharge port. The throttling valve is a simplified approach to modeling the known losses within the compressor due to pressure drops across the suction and discharge valves.
The work per unit mass of refrigerant, specific work, done by an ideal compressor is expressed by eq. 3-1 as follows:

\[ W_s = h_2 - h_1 \]  (3-1)

where \( h_1 \) is the refrigerant enthalpy [26].

Friction loss and turbulent loss are mainly responsible for the difference between work input and work delivered to the vapor. In reality, the required amount of work depends on the efficiency [1]. The actual compression processes for most compressors are irreversible polytropic processes. For simplicity, an isentropic analysis is often used. The actual power input to the compressor is usually calculated by eq. 3-2:

\[ W = \frac{W_{ideal}}{\eta_c} = h_2 - h_1 \]  (3-2)

where \( \eta_c \) is the compressor thermal efficiency.

In Figure 4 a refrigeration cycle is presented on the pressure-enthalpy diagram. The process in the compressor can be modeled as three stages: heating due to the losses, isentropic compression and throttling valve. The relevant equations that are required in modeling the compressor are provided in the following paragraphs.
When the vapor is not in the saturated state, it is assumed to behave like an ideal gas. In an ideal gas, the below equation 3-3 is valid:

\[ v = \frac{RT}{P} \]  

(3-3)

where \( v \) is the specific volume, \( T \) is absolute temperature, \( R \) is the universal constant of gases and \( P \) is the pressure.

When the refrigerant is not in the saturated state, vapor or liquid, the enthalpy change can be calculated from equation 3-4 [26]:

\[ \Delta h = C_p \Delta T \]  

(3-4)

where \( \Delta h \) is enthalpy change and \( C_p \) is the constant specific heat.

The entropy change of an ideal gas can be calculated from equation 3-5 [26]:

\[ \Delta s = \int_1^2 C_p \frac{dT}{T} - R \ln \frac{P_2}{P_1} \]  

(3-5)

With constant specific heats, the above equation simplifies to equation 3-6:

\[ \Delta s = C_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \]  

(3-6)

where \( \Delta s \) is the entropy change.
Another important parameter of the compressor is the volumetric efficiency. The volumetric efficiency is the ratio of the mass of vapor that is compressed to the mass of vapor that could be compressed if the intake volume were equal to the compressor piston displacement.

In order to determine the refrigerant flow rate in the refrigeration system, volumetric flow must be calculated. Volumetric flow depends on the pressure difference across the compressor. For a given compressor, size mass flow rate can be calculated from equation 3-7 [7]:

$$m = \frac{\eta \cdot PD}{v}$$

where PD is the piston displacement, \( \eta \) is the compressor volumetric efficiency, and v is the specific volume.

For a simple reciprocating cylinder, PD is calculated from equation 3-8 [27]:

$$PD = A_{HE} \times \frac{S}{12} \times N_s$$

where \( A_{HE} \) = area of head end of the piston, S = stroke length, and \( N_s \) = revolutions speed.

In this study, the compressor is a variable speed compressor. The speed of the compressor is controlled by the frequency of the input power. The magnetic field rotation speed can be determined from equation 3-9[28].

$$N_s = \frac{120f}{P}$$

where \( N_s \) is the magnetic field rotation speed, F is the frequency in Hertz and P is the number of the double poles. For the sake of simplification it is assumed that the slip velocity, relative velocity between rotor and magnetic field, is negligible.

The volumetric isentropic efficiencies are a function of the compressor speed. As the compressor speed increases, the volumetric efficiency improves. On the other hand, the isentropic efficiency decreases. Lower speeds in the compressor result in a smaller pressure drop in the valves and to the reduced heating at the cylinder. This explains the volumetric efficiency variation with compressor speed: the lower the
compressor speed, the higher the volumetric efficiency. The variation of volumetric efficiency with compressor speed is shown in Figure 5. The improvement in the isentropic efficiency happens because of the reduction in discharge superheat at higher compressor speeds. The variation of isentropic efficiency with compressor speed is shown in Figure 6.

The thermodynamic properties of the refrigerant R22 are derived from the ASHRAE graphs [4]. Properties of the refrigerant at the saturated liquid and saturated vapor states were fitted into the polynomial equations as a function of temperature.
In order to model the compressor given the amount of the superheat, the entropy of the vapor at the inlet of the compressor was determined from equation 3-6. As the compressor is supposed to be isentropic, the entropy of the refrigerant vapor at the compressor outlet should be equal to the entropy at the inlet. Using equation 6 in reverse manner the temperature of the vapor at the outlet will be calculated. The compressor work can be calculated by equation 3-1. Given the isentropic efficiency the actual work can be calculated from equation 3-2. To avoid complexity, the throttling effect was not considered in this study. In most applications, this assumption is reasonable [5]. Mass flow rate of the compressor is calculated by equations 3-7 and 3-8. To determine the specific volume, the ideal gas relation was used, given by equation 3-3.

3.3 Expansion Valve

The metering device controls the refrigerant flow through the refrigeration system. Due to the incompressibility of the liquids, if a liquid refrigerant enters to the compressor, it can cause mechanical damage in the compressor. A metering device adjusts the refrigerant flow in order to keep a fixed amount of superheat in the outlet of the evaporator. Another task of the metering device is to provide sufficient refrigerant flow to produce the required cooling load.

Expansion valves are usually used as metering devices in industrial refrigeration. In the condenser, the vapor refrigerant backs up behind the expansion valve and the pressure increases until the refrigerant is fully condensed. Low superheat does not provide sufficient margins of safety. On the other hand, superheat that is too high reduces the evaporator cooling effect and efficiency of the refrigeration cycle.

Different types of expansion valves are being used in refrigeration. For many years, thermal expansion valves were the dominant type, but recently electric expansion valves (EEVs) have begun to be used more frequently. EEVs use a microprocessor to control refrigerant flow. Three types of electric expansion valves are being used: step motor, pulse width modulated (PWM) and analog valves. In Figure 7 a refrigeration cycle with an EEV is shown. Two sensors sense the pressure and the temperature of the evaporator outlet. Using stored refrigerant data in the controller memory, the controller can calculate the superheat for a particular refrigerant.
In a step motor, the rotational movement of the electrical motor is transformed into linear movement to open or close the valve. The PWM valves are either fully opened or fully closed. The flow rate of refrigerant is determined by the duration of valve opening time, which follows from the pulse width. In the analog valve, instead of fully opening or closing, the opening can vary at multiple intermediate positions.

Electric expansion valves provide precise temperature control, consistent superheat control under variable condensing pressure, can be operated at lower ambient air temperature, and can save more energy [7].

Figure 3 shows that the enthalpy is constant across the expansion valve. This is given by eq. 3-10.

\[ h_3 = h_4 \quad (3-10) \]

Expansion valves are not only used to reduce the pressure but also to adjust the mass flow rate of refrigerant. The mass flow rate can be estimated by equation 3-11 [30].

\[ \dot{m} = C_d A \sqrt{2 \rho (P_1 - P_2)} \quad (3-11) \]

where \( \dot{m} \) is the mass flow rate, \( A \) is the flow area, and \( \rho \) is the refrigerant density at the inlet. \( P_1 \) and \( P_2 \) are the inlet and the outlet pressures, respectively. \( C_d \) is the mass
flow rate coefficient. For incompressible single-phase fluids, $C_d$ is a constant. However, for a two-phase mixture it must be determined by experiments.

### 3.4 Heat Exchangers

Heat exchangers are responsible for exchanging heat with their surrounding environment. Two different types of heat exchangers are used in vapor compression refrigeration systems in order to extract heat from cold environments and reject it to the outside environment [31].

#### 3.4.1 Condenser. A condenser is used to turn refrigerant from the vapor state to the liquid state. In the condenser, high pressure, high temperature refrigerant vapor flows through the tubes. The air-cooled condenser rejects heat from the refrigerant to the ambient air. Finned tube heat exchangers are the most commonly used type for small to medium capacity cold room applications. In larger capacity applications, evaporative condensers or cooling tower are widely used; however, in hot and dry climates the best option is the air-cooled condenser. Finned tube heat exchangers are indirect contact heat exchangers. Figure 8 shows a schematic of an air-cooled heat exchanger. Air flows on the outer surface, while refrigerant flows inside tubes. The air side heat transfer coefficient is much lower than the tube side heat transfer coefficient. Therefore, fins are used on the air side in order to compensate for air side heat transfer. The fin is a thin metal sheet, bonded to tubes, mainly mechanically, in order to increase the heat transfer area. The fin material is normally aluminum or copper. Air side fins can be plain or corrugated. Corrugated fins provide better improvement in the heat transfer. However, corrugated fins cause the pressure drop to increase.

![Figure 8 Schematic of an air cooled heat exchanger [32]](image)

The compressor discharges the superheated refrigerant vapor to the condenser. Superheated vapor rejects heat to the ambient air, and as a result, the superheated
vapor temperature decreases. Lastly, the vapor reaches the saturation state. As the saturation starts, the vapor will change phases from saturated vapor to liquid in a constant temperature. Finally, refrigerant exits as a sub-cooled liquid. Therefore, the condenser can be divided into three sections: desuperheating, condensing and subcooling. The amount of heat per unit mass of refrigerant rejected from each section can be expressed as the difference between the refrigerant enthalpy at the inlet and at the outlet of each section. The different sections of the condenser are shown in Figure 9.

![Figure 9 Three different sections of the condenser [7]](image)

To calculate the heat transfer capacity of a heat exchanger that works between two streams of fluids, both heat transfer coefficients must be known. In the case of air-cooled condensers, fin efficiency must also be known.

Thermal conductance and fouling are less important and can be neglected in many refrigeration applications without causing major inaccuracies.

The total rate of heat rejection from the high temperature refrigerant to the ambient air depends on the heat exchanger effectiveness, the heat capacity of each fluid and the available temperature difference (eq. 3-12).

\[
\dot{Q} = \varepsilon (\dot{m}c_p)_{\text{min}} (T_{h1} - T_{c1})
\]

(3-12)

where \( \varepsilon \) is the heat exchanger effectiveness, \( \dot{m} \) is the mass flow rate of fluid, \( c_p \) is the specific heat of the fluid, \( T_{h1} \) is the inlet temperature of the hot fluid and \( T_{c1} \) is the inlet temperature of the cold fluid.
The heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the maximum possible amount of heat transfer. The heat exchanger effectiveness depends on the temperature distribution within each fluid and on the configuration of the heat exchanger. In general, the refrigerant mass flow is distributed into a number of separate tubes in condensers and evaporators, i.e. the fluids are unmixed. In addition, the fin sheets of the heat exchanger prevent mixing of the air flowing over the heat exchanger. In a heat exchanger with a cross flow configuration with both fluids unmixed, the effectiveness can be related to the number of transfer units (NTU) as expressed by eq. 3-13 [33]:

$$
\varepsilon = 1 - exp \left\{ \left( \frac{1}{C_r} \right) (NTU)^{0.22} \left[ exp \left( -C_r(NTU)^{0.78} \right) - 1 \right] \right\}
$$

(3-13)

where $C_r$ is the heat capacity ratio given by eq. 3-14

$$
C_r = \frac{C_{\text{min}}}{C_{\text{max}}}
$$

(3-14)

where $C$ is the heat capacity, and $C_{\text{min}}$ and $C_{\text{max}}$ are the smaller and larger of the heat capacities between the hot and cold fluids, respectively. Heat capacity is defined as the product of heat capacity and mass flow rate. Heat capacity shows the amount of heat transfer in a substance as the temperature changes. In a given heat exchanger in the saturated section of the condenser, the heat capacity on the refrigerant side approaches infinity and the heat capacity ratio, $C_r$ approaches zero. When $C_r$ is zero, the effectiveness for any heat exchanger configuration is expressed as eq. 3-15.

$$
\varepsilon = 1 - exp(-NTU)
$$

(3-15)

The NTU is a function of the overall heat transfer coefficient, $U$, and is defined by eq. 3.16.

$$
NTU = \frac{UA}{C_{\text{min}}}
$$

(3-16)

where $A$ is the heat transfer area, $U$ is the overall heat transfer coefficient, it is expressed as eq. 3-17 Note that $A$ and $U$ are defined for a specific heat transfer
surface. For example, if the air side area is chosen, U will be calculated for the air side area and cannot be used with tube side area, and vice versa.

\[
\frac{1}{U} = \frac{1}{\eta h_a A_a} + \frac{1}{\bar{h}_r A_r} \tag{3-17}
\]

where \( \eta \) is the surface efficiency, and \( h_a \) is the air side heat transfer coefficient. It is assumed that there are no fins on the refrigerant side of the condensing tubes. In deriving eq. 3-17, wall thermal resistance and the fouling factors are neglected.

To calculate the heat transfer rate in a finned tube heat exchanger, the heat transfer coefficient inside tube, air side heat transfer coefficient and air side surface efficiency need to be known. The average values of the heat transfer coefficient are extracted from [5]. Air side surface efficiency is defined by eq. 3-18:

\[
\eta = 1 - \frac{A_f}{A_o} \left( 1 - \eta_f \right) \tag{3-18}
\]

where \( A_f \) is the fin area, \( A_o \) is the total heat exchanger area and \( \eta_f \) is the fin efficiency defined by equation 3-19.

\[
\eta_f = \frac{actual \ heat \ transferred}{Ideal \ heat \ transfer} \tag{3-19}
\]

Ideal heat transfer is heat that would be transferred if the entire fin area were at base temperature. The fin efficiency in case of a circular fin can be calculated as eq. 3-20.

\[
\eta_f = \frac{\tanh(mr_c \phi)}{mr_c \phi} \tag{3-20}
\]

where

\[
m = \left( \frac{2h_a}{kt} \right)^{1/2} \tag{3-21}
\]

\[
\phi = \left( \frac{T_e}{T_r} - 1 \right) \left( 1 + .35 \ln \left( \frac{T_e}{T_r} \right) \right) \tag{3-22}
\]
where $h_a$ is the air side heat transfer coefficient, $k$ is the thermal conductivity of the fin and $t$ is the thickness of the fin.

Eq. 3-23 is an empirical equation that can be used to calculate equal circular fins for the hexagonal plate [34].

$$\frac{r_e}{r} = 1.27\psi(\beta - 0.3)^{1/2} \quad (3-23)$$

where

$$\psi = \frac{X_t}{2r} \quad (3-24)$$

$$\beta = \frac{1}{X_t} \left( \frac{X_L^2 + X_T^2}{4} \right)^{1/2} \quad (3-25)$$

where $r$ is the tube radius, $X_L$ is the tube spacing in the direction parallel to the direction of air flow, and $X_T$ is the tube spacing normal to the direction of air flow, as depicted in Figure 8.

---

Figure 8. Definition of transverse and longitudinal spacing

In this study, a simplified but valid estimation of the condenser is satisfactory. To avoid unnecessary complexity of two phase flow in the condenser, the efficiency of the heat exchanger is supposed to be constant over the condenser sections. The
efficiency of the condenser will be calculated from eq. 3-15. The heat rejection rate can be calculated from eq. 3-12.

A fan must be employed to maintain the airflow at a sufficient rate over the fins and the tubes of the heat exchanger. Insufficient airflow increases the size of the condenser. On the other hand, excessive air flow causes elevated electrical consumption in the fan motor and higher noise level. Therefore a good design should maintain a balance between initial cost and operation cost. While it is true that the compressor consumes much of the power consumed by the total refrigeration system, the condenser fan requires a considerable amount of power as well. The power required by the fan is directly related to the air side pressure drop across the condenser and to the velocity of air across the condenser:

\[ W_{fan} = \frac{V_a \Delta P A_{fr}}{\eta_{fan}} \]  

(3-26)

where \( V_a \) is the air velocity over the face of the condenser, \( \Delta P \) is the air side pressure drop over the condenser, \( A_{fr} \) is the frontal area of the condenser, and \( \eta_{fan} \) is the condenser fan efficiency.

The pressure drop over the condenser is a function of the air velocity, air properties and the heat exchanger characteristics. A pressure drop in a compact heat exchanger can be calculated from equation 16. Since the temperature difference is not very high in the condenser (5~15 K), the density variation was neglected in equation 3-27. A more detailed form of this equation can be found in [35].

\[ \Delta P = \frac{G^2}{2 \rho} \left[ f \frac{A_0}{A_{min}} \right] \]  

(3-27)

where \( G \) is the mass flux defined in equation 3-28, \( \rho \) is the average air density, \( f \) is the friction factor and \( A_{min} \) is the minimum flow area. The friction factor varies with flow conditions; however, in this study an averaged value used.

\[ G = \frac{\rho V_a A_{fr}}{A_{min}} \]  

(3-28)

Using equations 3-26 to 3-28 and given the electrical power, the air flow over the condenser can be estimated.
3.4.2 Evaporator. The evaporator absorbs the heat from the cold room in order to keep the required temperature. The refrigerant enters the evaporator as a saturated two-phase mixture and exits as a superheated vapor. Several types of evaporators are used in the industry, but in this study a dry expansion or direct expansion (DX) evaporator is considered. In DX evaporators, liquid refrigerant is fed through an expansion valve and a distributor. The refrigerant flows inside the tubes in a finned coil, and it is completely vaporized and superheated to a certain degree before reaching the exit of the evaporator. Finally the superheated refrigerant vapor enters the compressor suction port. Figure 11 shows a DX finned coil evaporator.

![Figure 11 Direct expansion (DX) evaporator](image)

Evaporators and condensers have a similar construction. Geometric parameters defined for the condenser are used unchanged for the evaporators. Table 1 presents some illustrative data for finned tube coil construction parameters [7].
Thermal analysis of the evaporator and the condenser is essentially similar. However, in the evaporator some modifications must be made to account for the dehumidification process in the air side. In order to simplify the evaporator model, the evaporator coil is assumed to operate in a dry manner. However, since the air is being dehumidified as it flows over the evaporator, part of the total heat causes water content in the air to condense. There is no temperature change due to this part of heat transfer, which is called latent heat. On the other hand, the part of heat transfer that causes a temperature difference is called sensible heat. To account for the latent heat, the specific heat must be modified. The total enthalpy change of the air is the sum of the enthalpy change due to sensible heat, and the enthalpy change due to latent heat. This is given in equation 3-30.

$$\Delta h_{total} = \Delta h_{sensible} + \Delta h_{latent}$$  \hspace{1cm} (3-30)

In order to account both the latent heat and the sensible heat, an effective specific heat is utilized. Employing an effective specific heat will result in a more accurate determination of the evaporator exit temperature without the complications associated with using the standard equations for air-water mixtures. Since the complete modeling of the evaporator is not the focus of this study, this simplification is justified.

The specific heat ($c_p$) is defined as the ratio of the sensible heat enthalpy change to the temperature change. The effective specific heat is defined as equation 3-31 [36].

### Table 1 Finned tube coil construction parameters [4]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter of copper tube $D_o$</td>
<td>0.528 in.</td>
</tr>
<tr>
<td>Inner diameter of copper tube $D_i$</td>
<td>0.496 in.</td>
</tr>
<tr>
<td>Aluminum fin thickness $F_t$</td>
<td>0.006 in.</td>
</tr>
<tr>
<td>Longitudinal tube spacing</td>
<td>1.083 in.</td>
</tr>
<tr>
<td>Transverse tube spacing</td>
<td>1.25 in.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fin spacing $S_f$</th>
<th>Fins/in.</th>
<th>$A_p/A_p^+$</th>
<th>$A_o/A_i$</th>
<th>$A_f/A_{f}^{+}$</th>
<th>$F_s$</th>
<th>$S_f/F_t$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>0.125</td>
<td>7.85</td>
<td>7.95</td>
<td>0.873</td>
<td>9.91</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>0.100</td>
<td>9.68</td>
<td>9.68</td>
<td>0.896</td>
<td>12.07</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>0.0833</td>
<td>11.54</td>
<td>11.40</td>
<td>0.913</td>
<td>14.21</td>
</tr>
<tr>
<td></td>
<td>14</td>
<td>0.0714</td>
<td>13.46</td>
<td>13.17</td>
<td>0.925</td>
<td>16.37</td>
</tr>
<tr>
<td></td>
<td>15</td>
<td>0.0667</td>
<td>14.47</td>
<td>14.03</td>
<td>0.928</td>
<td>17.48</td>
</tr>
</tbody>
</table>

$^1A_p$ is area of the primary surfaces (outside surface of the copper tubes).

$^2A_f$ is area of fins.

Note: For corrugated plate fins, $A_o/A_p$, $A_f/A_{f}^{+}$ and $F_s$ should be multiplied by a factor of 1.10 to 1.20.
where \( C_p \) is the specific heat for dry air and \( C_{p.e.} \) is the effective specific heat. The ratio of the latent heat to the total heat depends on the application and the environment. For example, if the latent heat accounts for roughly 1/4 of the total enthalpy change of the air as it flows over an evaporator, by equation 3-31, the effective heat capacity is calculated to be 1.33 times the \( C_p \) of the dry air. Higher \( C_p \) means lower temperature change in the air, which is justified by the fact that a part of the heat transfer in the evaporator is used to change the phase of the moisture at the constant temperature.

### 3.5 Cold Room

The purpose of any refrigeration system is to provide the required cooling in the cold room. The cooling effect of the refrigeration system must encompass the heat gains of the cold room and the enthalpy changes of the freight. To estimate the heat load of a cold room several factors are considered. The most important factors are: field heat, heat of respiration, conductive heat gain, air infiltration, air exchange, equipment heat load and human energy load.

Reception heat, sometimes called field heat, is the required heat transfer needed to reduce the product temperature at reception down to the storage temperature. The type of product and quantity of produce, the initial temperature and the final cold storage temperature affect the reception heat load.

Heat of respiration is the energy released by the product. It is generated as a result of chemical reactions inside the product. Respiration heat production decreases during the cool down period after loading and then stabilizes after the storage temperature is reached.

Conductive heat gain is heat transfer through the building floor, walls and ceiling by conduction. Conduction heat transfer depends on the conductivity of the walls and the temperature differences between the inside and outside of the cold room walls. In practice, conduction heat transfer can be calculated from equation 3-32 [37]:

\[
Q = UA\Delta T
\]
where Q is the heat transfer rate, U is the overall heat transfer coefficient, A is the heat transfer area and $\Delta T$ is the temperature difference.

The overall heat transfer coefficient in a multi-layer wall can be calculated from equation 3-33 [37]:

$$U = \frac{1}{h_e + \sum_{i=1}^{n} \frac{x_i}{k_i} + \frac{1}{h_i}}$$  \hspace{1cm} (3-33)

where $x_i$ and $k_i$ are the thickness and conductivity of each layer, respectively. The inside and outside air convection heat transfer coefficients are represented by $h_i$ and $h_e$, respectively.

Figure 12 shows a multi layered wall of a cold room.

Air change heat transfer is due to the introduction of outside air with the cold inside air. The air change heat load depends on the construction quality, size of the room and frequency of the opening and closing of the structure.

Equipment heat load is the energy generated by the electrical equipment operating inside the cold room. Fan motors, lights, defrost equipment and other electrical equipment reject some of the input power to the cold room. The higher the efficiency of the electrical machines, the less heat will be introduced to the cold room. The human energy load is the amount of energy expended by the presence of the workers in the room.
It is not a good practice for the refrigeration unit to operate for 24 hours continuously. A time interval must be dedicated for the evaporators to be defrosted. A normal time of 16 hours out of 24 hours is standard practice. As a result in most application the refrigeration capacity is calculated on the basis of 16 hours of operation. The operation hours can be described as a safety factor in the design and selection of the refrigeration unit.

Calculation of the heat load in cold rooms is very cumbersome and uncertain. Since the aim of this study is to optimize the control process of the refrigeration system, a simplified model of the cold room will be used. Heat losses from the cold room are supposed to be only via conduction through the walls. Other means of heat transfer such as respiration, ventilation and radiation are neglected. The thermal inertia of the structure is also neglected. The thermal inertia of the system is assumed to come completely from the freight in the cold room.

The cold room building can be constructed from steel, masonry or a wooden frame. The construction of the building for refrigerated cold storages is similar to conventional construction. To reduce the heat losses from the exterior of the cold room, insulation is required. Polyurethane, styrofoam, fiberglass, and other good insulation material or a combination of the above are used as insulation. The foundations and floors, walls and ceilings should be well-insulated.

The thermal inertia of the system is assumed to come completely from the freight in the cold room. Equation 3-34 [33] can be used to calculate the required heat transfer to change the temperature of the freight from an initial temperature to the final temperature.

$$Q = mc \Delta T$$

$$\Delta Q = mC \Delta t$$

(3-34)

where $Q$ is the heat transfer, $m$ is the mass of the freight, $C$ is the specific heat and $\Delta T$ is the temperature change. Given the amount of heat transfer, the temperature change can be calculated.

3.6 Simplified Model of Refrigeration System and Cold Room

The refrigeration system and the cold room were modeled in order to be used in computer simulation. The ambient conditions and those of the cold room, with the parameters of the refrigeration system components and freight thermo-physical...
properties were the input to the computer model. Based on the input, the performance of the vapor compression reciprocating refrigeration system was predicted. An iterative procedure was used to modify the unknown parameters. If the estimated performance did not meet the cooling requirement, the controlling parameters of the system were adjusted to compensate for the heat transfer requirement. The parameters were repeatedly modified as long as the different governing equations were satisfied.

The results of computer simulation should at least be stable, rapid and with reasonable accuracy. In reality, to obtain more accurate results, more time is required and vice versa. Numerous techniques to improve the stability, speed and accuracy have been investigated. But the results are not good enough in many cases and more research is necessary [39]. A schematic of the refrigeration cycle is given in Figure 13, which shows the important temperatures in the system.

![Figure 13: A schematic of the refrigeration cycle](image)

Initial condensing and evaporating temperatures were estimated by the addition and subtraction of a fixed amount of temperature difference from ambient air and initial room temperatures, respectively. A temperature difference of about 10 K (18 F) is reasonable in most applications. To avoid complicating the two-phase flow in the evaporator, a constant super heat was assumed.

\[
t_{\text{evap}} = t_{\text{room}} - dt_{\text{evap}}
\]

\[
t_{\text{cond}} = t_{\text{air}} + dt_{\text{cond}}
\]

For the sake of simplicity, the overall heat transfer coefficient in the evaporator and the condenser were assumed to be constant. The heat transfer rate was calculated from equation 3-32.
It can be seen that the only parameter that can alter the heat transfer rate is the temperature difference. To estimate the actual temperature difference, the heat transfer calculated from the refrigeration cycle model must be the same as the heat transfer computed from the heat exchanger model. Heat transfer in a given heat exchanger only varies with the temperature difference. Therefore, in both the condenser and evaporator, the new temperature difference can be calculated by multiplying the old temperature difference by the heat transfer ratios from the refrigeration cycle and the heat exchanger calculations. [40]

\[
d t_{evap}^{new} = d t_{evap}^{old} \times \left( \frac{q_c}{q_{evap}} \right) \\

\]

(3-37)

\[
d t_{cond}^{new} = d t_{cond}^{old} \times \left( \frac{(q_c + W_c)}{q_{cond}} \right) \\

\]

(3-38)

The modified condensing and evaporating temperatures can then be calculated from equation 3-35 and 3-36.

The expansion valve adds a higher limit to the refrigerant mass flow in the refrigeration system. Closing the expansion valve causes the refrigeration system to operate at a lower evaporating temperature and higher superheat. On the other hand, opening the expansion valve more than the capacity of the compressor does not change the mass flow rate. A linear relation is assumed between the expansion valve opening and the mass flow rate. [30]

\[
mass\_flow = c_d \times A_{eev} \times \left( \frac{2}{\sqrt{vf^2a}} \right) \times (p2a - p4a) \times 10^6 \times eev\_op \\

\]

(3-39)

After the higher limit of mass flow rate were calculated from the expansion valve equation, the input parameters of the compressor were adjusted to comply with the higher limit of mass flow. If the calculated mass flow in compressor was less than the higher limit, nothing happened. On the other hand, if there was a higher mass flow in the compressor compared to the mass flow calculated from the expansion valve, the evaporating temperatures were set lower iteratively until both components (compressor and expansion valve) generated the same mass flow. Finally, the temperature change of the cold room can be calculated by equating the heat absorbed by the refrigeration cycle, heat loss to the surroundings, and the thermal inertia of the cold room. This is shown schematically in Figure 14.
Figure 14. An algorithm of the model calculations
3.7 Simulation Result

The modeled refrigeration system and cold room were simulated for 3 hours with a zero degree initial room temperature in Matlab. Similar specifications and parameters with the real system were used. Figures 15-17 show the cold room temperature, evaporator coil temperature and condenser coil temperature.
Figure 17 Condenser coil temperature
Chapter 4: Design and Implementation of On-off Controller

4.1 Introduction

An on-off controller or thermostat maintains the cold room temperature by turning the compressor, condenser fan and evaporator fan on and off. All of these major components are on full load capacity during the “on” status. A simple commercial thermostat for a cold room is shown in figure 18.

Figure 18 (a) front view of cold room thermostat (b) wiring of thermostat [41]

A refrigeration thermostat has some other functions in order to protect the compressor such as an on and off time delay for the compressor to prevent it from turning on or off suddenly, which can damage the compressor. Also, a thermostat is used to open and close the solenoid to prevent refrigerant liquid from entering the compressor.

An algorithm of the on-off controller is provided in Figure 19.

To control refrigerant flow in the refrigeration cycle, a mechanical expansion valve is used in order to proportionally regulate the opening of the expansion valve to maintain the superheat temperature on the preset value. Figure 20 shows a refrigeration system equipped with a thermal expansion valve.
Figure 19 Flow chart for algorithm of On-Off controller
4.2 Simulation

A simulation of refrigeration system with an on-off controller and thermal expansion valve was done using Matlab software. Matlab code was written to simulate logic as mentioned in Figure 19. A fuzzy controller was used to simulate the thermal expansion valve, but because of the complexity of superheat modeling (it cannot be modeled by lumped parameter modeling), the temperature difference between the cold room and the evaporator coil was maintained on a pre-set value instead of using superheat temperature.

A Simulink block, which is designed for an on-off controller and mechanical expansion valve, is given in Figure 21.
Figure 21 Simulink block diagram for on-off controller
4.3 Controller Validation with Experimental Data

In order to validate the simulated on-off controller, experimental data was extracted from a running system in the field with the specifications mentioned in Appendix C.

The room temperature was recorded for 167 minutes and compared with the results of the simulated model. The results of both the experimental and simulated models are presented in Figure 22. It can be seen that the results of the simulated data are satisfactory for our purpose, which is analyzing and comparing different controllers on a simulated model.

![Figure 22 Simulation Validation](image)

4.4 Simulation Result

The results for the inputs and outputs of the refrigeration system using the on-off controller are presented in this section. Inputs to the system were compressor frequency, condenser fan input power, evaporator fan input power and expansion valve opening.

The response of compressor frequency is given in Figure 23 and fan input power is presented in Figures 24 and 25. All of these three major components were turned on to operate at full load in order to decrease the room temperature and turned off to increase the room temperature.
Figure 23 Frequency Response of compressor using on-off controller

Figure 24 Input Power Response of condenser using on-off Controller

Figure 25 Input Power Response of evaporator using on-off Controller
Figure 26 shows the opening of the expansion valve.

![Figure 26 Opening of thermal expansion valve](image)

The outputs of the system were room temperature and the room-evaporator coil temperature difference. The response of room temperature is shown in Figure 27. The room temperature kept oscillating between set point + hysteresis and set point – hysteresis which led to an almost average value equal to the set-point.

![Figure 27 Room Temperature using on-off controller](image)

Figure 28 shows the difference between cold room temperature and evaporator coil temperature. The set point for the temperature difference should be at the lowest possible value because reducing the efficiency leads to increasing the system efficiency.
Figure 28 Difference of room temperature and evaporator coil temperature using on-off Controller

Figure 29 shows the difference between the ambient temperature (45 °C) and condenser temperature.

Figure 29 Difference between condenser temperature and ambient temperature using on-off Controller

The coefficient of performance of the refrigeration system is shown in Figure 30.
The calculated power consumption of the compressor, evaporator fan and condenser fan is presented in figure 31.
Chapter 5: Design and implementation of Fuzzy PID Controller

5.1 Introduction

The introduction of fuzzy concepts or multi-valued logic was presented in the 1930s by Jan Lukasiewicz. Current fuzzy logic in the form systems of mathematical logic with natural language terms was introduced by Lotfi Zadeh, professor and head of the electrical engineering department at the University of California at Berkeley.

Zadeh defines the term “fuzzy logic” as follows [42]:

“Fuzzy logic is determined as a set of mathematical principles for knowledge representation based on degree of membership rather than on the crisp membership of classical binary logic.”

The logic system contains a fuzzifier, rules, inference engine and deuzzifier, as shown in Figure 32:

5.2 Design Procedure

In this thesis, a fuzzy PID controller with a Mamdani inference engine using a minimum/maximum method for rule evaluation was used [42]. Centroid defuzzification was also used. This fuzzy PID controller behaves approximately similar to a parameter-varying PID controller [43].

A fuzzy PID controller is constructed from a combination of a fuzzy PD controller and fuzzy PI controller. A fuzzy PD controller is rule-based and has two inputs, which are error \( e \) and the change rate of error \( \dot{e} \). Inputs and output of fuzzy PD controller are presented in figure 33.
There are many types of membership functions but the most popular types are triangular and Gaussian bell shape functions [42].

The inputs and outputs of the controller are normalized and each input and output has five membership functions in the form of a triangular and Gaussian bell shape.

Figures 34-36 show the different types of inputs and output membership functions.

Figure 34 Membership functions in form of triangular shape a) input variable “error”, b) input variable “change rate of error”, c) output variable
Rules of fuzzy control are given in the form of:

\[ R_k: \text{if } e \text{ is } A_i \text{ and } \dot{e} \text{ is } B_j, \text{ then } u \text{ is } u_{ij} \]

where \( R_k \) is the fuzzy control rule, \( A_i \) is the fuzzy sets of error, \( B_j \) is the fuzzy set of change rate of error and \( u_{ij} \) is the set of output.

The “min” method was used to calculate and in rule evaluation, which is shown in equation 5-1[42]:

\[

\text{\textsc{Figure 35}} \text{ Membership functions in form of Gaussian bell shape a) input variable “error”, b) input variable “change rate of error”, c) output variable}

\text{\textsc{Figure 36}} \text{ Membership functions in form of mixture of Triangular and Gaussian bell shape a) input variable “error”, b) input variable “change rate of error”, c) output variable}
According to inputs and output membership functions, the rules of the controller are presented in Table 2.

**Table 2 Rules of fuzzy controller**

<table>
<thead>
<tr>
<th>e</th>
<th>NS</th>
<th>NB</th>
<th>Z</th>
<th>PS</th>
<th>PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>PB</td>
<td>M</td>
<td>LB</td>
<td>LB</td>
<td>LS</td>
<td>LS</td>
</tr>
<tr>
<td>PS</td>
<td>HS</td>
<td>M</td>
<td>LB</td>
<td>LS</td>
<td>LS</td>
</tr>
<tr>
<td>Z</td>
<td>HS</td>
<td>HS</td>
<td>M</td>
<td>LB</td>
<td>LB</td>
</tr>
<tr>
<td>NB</td>
<td>HB</td>
<td>HS</td>
<td>HS</td>
<td>M</td>
<td>LB</td>
</tr>
<tr>
<td>NS</td>
<td>HB</td>
<td>HB</td>
<td>HS</td>
<td>HS</td>
<td>M</td>
</tr>
</tbody>
</table>

A graphical surface presentation of the rules is presented in Figure 37.

**Figure 37 input and output relation surface**

\[
\mu_{uij}(u) = \mu_{A_i \cap B_j}(u) = \min[\mu_{A_i}(e), \mu_{B_j}(\dot{e})]
\]
The defuzzification was done using the centroid method. This method is also called center of gravity (COG). A mathematical representation of COG can be expressed as follows [41]:

\[
COG = \frac{\int_{a}^{b} \mu(x) dx}{\int_{a}^{b} \mu(x) dx}
\]  

(5-2)

where \( A \) is the fuzzy set and \( ab \) is the fuzzy set’s interval.

For a discrete system, COG can be obtained by calculating it over \( a \) of sample points. In this case, the following formula is applied [41]:

\[
COG = \frac{\sum_{x=a}^{b} \mu(x) dx}{\sum_{x=a}^{b} \mu(x) dx}
\]  

(5-3)

Implementation of the fuzzy PD controller is given in Figure 38.

![Figure 38 fuzzy PD control block diagram [43]](image)

Equation 5-4 and Figure 39 show how a fuzzy PI controller can be constructed from a fuzzy PD controller by adding an integrator.

![Figure 39 fuzzy PI controller block diagram [43]](image)
\[ u_t = \alpha \int (A + PK_pe + DK_d \dot{e}) \, dt = \alpha At + \alpha K_p De + \alpha K_d P \int e \, dt \] (5-4)

where \( K_p, K_d \) are scaling factors for \( e, \dot{e} \) and \( \alpha \) is the integral constant.

A fuzzy PID is constructed by adding a fuzzy PI and fuzzy PD controller. A controller block diagram for the discrete system is presented in Figure 40.

![Fuzzy PID control block diagram](image)

A refrigeration process with a fuzzy PID controller was simulated in Simulink. Figure 41 shows a Simulink block diagram of the simulated system.

### 5.3 Results

The response of the inputs and outputs to the refrigeration process using a fuzzy PID controller are given in this section.

Compressor frequency can vary between 30 to 60 Hz to maintain room temperature on a set-point. Compressor frequency cannot be less than 30 Hz due to lubrication failure in reciprocating compressors and it is preferable to not be more than 60 Hz, although efficiency decreases when the frequency increases. The frequency response of the compressor is shown in Figure 42.
Figure 41: Simulink implementation for fuzzy controller
Compressor frequency in a steady state response is between 30 to 35 Hz. This range of frequency results in high energy savings according to compressor performance with R-22 refrigerant [8].

Varying evaporator and condenser fan speed gives more degrees of freedom for controlling the room temperature. Fan speeds are controlled by changing the input power of the fans. The response of the condenser and evaporator input power using a fuzzy PID controller are presented in Figures 43 and 44.
Figure 44 Evaporator Input Power Using fuzzy PID Controller

As shown in Figures 34-36, different types of membership functions have been used for simulation. The following figure shows the room temperature transient response and disturbance rejection of a fuzzy PID controller using different types of membership functions when a ramp disturbance of 2 °C/sec is added to it.
As can be observed from Figure 46, the transient response of a fuzzy PID controller using all 3 types of membership functions are very similar but the disturbance rejection of controller with mixture of Triangular and Gaussian bell shape function is slightly better than other 2 types.

The response of room temperature is presented in Figure 47.

The difference between room temperature and evaporator coil temperature is presented in figure 48.
Figure 48 Difference between room temperature and evaporator coil temperature

Figure 49 shows the difference between ambient temperature (45 °C) and condenser coil temperature.

Figure 49 Difference of room temperature and condenser coil temperature using fuzzy PID controller

The coefficient of performance of the refrigeration system is shown in Figure 50.
Figure 50 COP of refrigeration system using fuzzy PID controller

Figure 51 shows the calculated power consumption of the compressor, evaporator fan and condenser fan for 3 hours.

Figure 51 Power Consumption of refrigeration system using fuzzy PID controller for 3 Hours
Chapter 6: Comparison between Controllers

In this chapter, the classical on-off controller and the proposed fuzzy PID controller are compared in different aspects from subsystem performance to overall system performance.

6.1 Compressor Controller

In this part, the responses of a refrigeration system using a fuzzy PID controller for the compressor are compared with the responses of a system using on-off controllers for all components.

As the compressor speed decreases, less refrigerant is circulated in the refrigeration system component. Lower refrigerant mass flow requires less heat transfer in the evaporator and condenser. However, heat exchangers are designed to operate adequately at full capacity operation of the compressor. In short, the heat exchangers are oversized when the compressor operates in reduced load mode. Supposing a nearly constant heat transfer coefficient, since the area of the heat exchanger is constant, it is clear from equation 3-12 that a smaller heat transfer rate results in a smaller temperature difference in the heat exchangers. It is known that the COP improves up to 4 percent for each °C when the evaporating temperature is raised or the condensing temperature is lowered [26]. This explains a major part of the improvement in the retrofit from a conventional on-off controller compared to the variable speed compressor. Also, as mentioned in section 3.2, as the compressor speed decreases, the volumetric efficiency also decreases. Meanwhile, the isentropic efficiency improves. The trend of improving isentropic efficiency is higher than decreasing volumetric efficiency which causes improvement of the system’s COP.

Figure 52 shows the response of the compressor’s frequency for both the on-off and fuzzy PID controller.
As shown in Figure 53, a fuzzy PID controller on the compressor can perform much better than an on-off controller for room temperature reference tracking.

Figure 54 shows the difference between ambient temperature (45 °C) and condenser coil temperature for both the classical and fuzzy controller implemented on the compressor.
The coefficients of performance of both controllers are presented in Figure 55:

The COP of the fuzzy PID controllers is 19-21% higher than the COP of the on-off controller during “on” times, which causes high energy savings for compressor.

Figure 56 shows the power consumption of the compressor, condenser fan and evaporator fan of both controllers for one day.

The power consumption for one day of the main components of the refrigeration system using the fuzzy PID controller was about 50 Kw less than for the system using the on-off controller. This means a power savings of 11.9 %. 
6.2 Heat Exchanger Controller

In this part, the responses of a refrigeration system with a variable speed compressor and heat exchangers using a fuzzy PID controller are compared with responses of a system with a variable speed compressor using fuzzy PID controllers and fixed-speed heat exchanger fans.

The condenser fan and evaporator fan power input are presented in Figures 57 and 58.
Figure 58 Evaporator fan input power for both systems using variable speed and fixed speed fan

In order to maintain the room temperature at the desired set-point, the fuzzy PID controller decreases the input power of both fans which causes a lower fan speed.

Figure 59 shows the responses of cold room temperature for both systems with variable and fixed speed fans.

Figure 59 Room temperature responses for both systems with variable and fixed speed fans

A system with variable speed fans controlled by a fuzzy PID controller has a lower overshoot compared to a system with fixed speed fans.

The difference between condenser coil temperature and ambient temperature (45 °C) for both systems is shown in Figure 60.
Figure 60 Difference between condenser and ambient temperature for both systems using variable and fixed speed fans

Figure 61 shows the coefficient of performance of the refrigeration system for both systems.

It is observed from Figure 61 that lowering the speeds of fans causes a decrease in the COP of the refrigeration cycle because a lower air mass flow rate facilitates heat exchangers to operate less refrigeration performance. It is clear from equation 3-12 that lower mass rate results in a bigger temperature difference.

The power consumption of both systems using variable speed fans and fixed speed fans is presented in Figure 62.
Figure 62 Daily power consumption for both systems with variable and fixed speed fans

It is observed that using a variable speed fan reduces overall power consumption of the refrigeration cycle around 5%. As can be seen from equations 3-(27, 28, 29), lower fan speed requires less power input. In short, up to a point, decreasing the fan speed reduces the overall power consumption of the refrigeration cycle because the reduction of electrical power consumed by the fan motor is justified the energy lost from decreasing of COP of the compressor. In constant speed fans, it is not possible to adjust the speed to the optimum one, while it is possible in variable speed fans.

### 6.3 Comparison between fuzzy PID and on-off Controller

From the previous two parts of this chapter, it can be concluded that a variable speed compressor, condenser fan and evaporator fan using a fuzzy PID controller can improve overall performance of the refrigeration cycle. In this chapter a refrigeration system with the proposed fuzzy PID controller for all sub systems is compared with a system using the classical on-off controller for the entire system.

Figure 63 shows the responses of both controllers for cold room temperature.

Figure 64 shows the response of cold room temperature for both controllers when a ramp disturbance of 2°C/sec is added to it. Disturbance rejection can be seen, as the fuzzy PID controller is faster and deviation is less than in the on-off Controller.
The difference between condenser coil temperature and ambient temperature for both controllers is presented in figure 65.
The temperature difference in a system using a fuzzy PID controller is lower than in a system using an on-off controller. A lower temperature difference in the condenser and evaporator causes higher efficiency of a refrigeration cycle.

The coefficient of performance and daily power consumption of both systems are shown in Figures 66 and 67.

**Figure 66** Difference between room temperature and evaporator temperature for both controllers

**Figure 67** Coefficient of performance for both controllers
Figure 68 shows the power consumption of both systems.

![Figure 68 Power consumption of both controllers](image)

Figure 68 Power consumption of both controllers

Figure 69 shows the power consumption of both systems in UAE currency.

![Figure 69 Power usage of both systems in UAE currency](image)

Figure 69 Power usage of both systems in UAE currency

It is observed from Figures 68 and 69 that a system using a fuzzy PID controller has about 17% savings compared to a system using a classical controller. Moreover, an analysis of the initial costs considered including inverters for the electric power of the compressor, evaporator fan and condenser fan. Further additional costs linked to the application of fuzzy PID controllers for all equipment have resulted in a pay-back period of less than a year.
A comparison between different control scenarios is presented in table 3.

Table 3 Summary of comparison

<table>
<thead>
<tr>
<th>Tests</th>
<th>On-Off Controller</th>
<th>Fuzzy PID (Fixed Speed Fans)</th>
<th>Fuzzy PID (Variable Speed Fans)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature Reference Tracking</td>
<td>Weak</td>
<td>Good</td>
<td>Best</td>
</tr>
<tr>
<td>Disturbance rejection</td>
<td>Good</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Coefficient of Performance (COP)</td>
<td>Low</td>
<td>Highest</td>
<td>High</td>
</tr>
<tr>
<td>Power Consumption (one day)</td>
<td>421.2 Kw 126 AED</td>
<td>370.1 Kw 111 AED (12% Saving)</td>
<td>350.9 Kw 105 AED (16.1% Saving)</td>
</tr>
<tr>
<td>Life Time of Equipment</td>
<td>Low</td>
<td>High</td>
<td>Best</td>
</tr>
</tbody>
</table>
Chapter 7: Conclusion

This thesis has proposed a fuzzy PID controller for the main subsystems of a vapor compression cycle to improve the overall performance of a refrigeration system. The main achievements of the proposed controller are a reduction in power consumption and improvement of reference tracking of cold room temperature in comparison to a classical on/off controller. Fuzzy PID controllers have a high potential of being implemented in real world refrigeration systems. Chapter 3 of this thesis discussed the modeling of a vapor compression cycle. The model was validated with data extracted from an actual real refrigeration plant in the field. Chapter 4 describes the structure of a classical thermostatic controller operating through an On/Off cycle for all major components at the nominal frequency. The room temperature response obtained is satisfactory but with the disadvantage of not being precise. Also, a large number of On/Off cycle reduces the lifetime of the components. In Chapter 5, a fuzzy PID controller was developed for the major components of the refrigeration system. Cold room temperature was controlled with high precision by varying the frequency of compressor and input power of the condenser and evaporator fan. Also, On/Off cycles were eliminated from the entire system. Chapter 6 describes a comparison between the proposed fuzzy PID controller and classical On/Off controller for subsystem performance and overall system performance. It has been shown that overall power consumption of the system is significantly reduced and the temperature of the cold room is controlled with higher precision.
References


[29] Emerson Climate Control, product manual.


[41] Fullgauge co. product manuals. www.fullgauge.com


Appendix A

R-22 Gas Properties

To model the refrigeration cycle, refrigerant thermo-dynamic properties must be provided as a function of temperature and pressure. Thermo-dynamic tables provide refrigerant thermo-dynamic properties [32, 43]. To utilize the table data, a polynomial in the form of equation 1 was fitted on the data from the tables using a curve-fitting technique.

\[
\varphi = p_n \cdot x^n + p_{n-1} \cdot x^{n-1} + \cdots + p_3 \cdot x^3 + p_2 \cdot x^2 + p_1 \cdot x + p_0
\]

where \(\varphi\) is the thermo-dynamic property, \(x\) is the temperature and \(P_i\) must be determined from the data. The MATLAB toolbox was used to fit the curves. Order of the polynomial is selected to provide desired accuracy for each property (less than 5% in the range of -100 C to 95 C).

The following properties were considered:

1. \(p\): pressure
2. \(v_f\): specific volume of the liquid
3. \(v_g\): specific volume of the vapor
4. \(h_f\): enthalpy of the liquid
5. \(h_g\): enthalpy of the vapor
6. \(c_{p_f}\): specific heat of the liquid
7. \(c_{p_g}\): specific heat of the vapor
8. \(\mu_f\): viscosity of the liquid
9. \(\mu_g\): viscosity of the vapor
10. \(k_f\): heat conduction coefficient of liquid
11. \(k_g\): heat conduction coefficient of vapor
12. \(\text{prf}\): prandtl number of liquid
13. \(\text{prg}\): prandtl number of vapor
14. \(s_f\): entropy of liquid
15. \(s_g\): entropy of the vapor

In Figure 1, a pressure-enthalpy diagram of R22 is provided. Other required properties were extracted from [43].
Figure 70. Pressure-enthalpy diagram of R22

Table 4. R22 properties

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Pressure (MPa)</th>
<th>Liquid Density (kg/m³)</th>
<th>Vapor Volume (m³/kg)</th>
<th>Enthalpy (kJ/kg)</th>
<th>Entropy (kJ/kgK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-100</td>
<td>0.00200</td>
<td>1571.7</td>
<td>8.2980</td>
<td>90.24</td>
<td>358.93</td>
</tr>
<tr>
<td>-90</td>
<td>0.00480</td>
<td>1545.1</td>
<td>3.6548</td>
<td>100.95</td>
<td>363.82</td>
</tr>
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<td>-80</td>
<td>0.01035</td>
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<td>1.7816</td>
<td>111.66</td>
<td>368.75</td>
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<td>133.11</td>
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</tr>
<tr>
<td>-50</td>
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<td>0.32405</td>
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</tr>
<tr>
<td>-48</td>
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<td>0.29469</td>
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</tr>
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<td>148.25</td>
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</tr>
<tr>
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<td>0.24507</td>
<td>150.43</td>
<td>386.23</td>
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<td>1412.6</td>
<td>0.22410</td>
<td>152.61</td>
<td>387.17</td>
</tr>
<tr>
<td>Temperature (^{\circ}\mathrm{C})</td>
<td>Pressure (MPa)</td>
<td>Liquid Density ((\text{kg/m}^3))</td>
<td>Vapor Volume ((\text{m}^3/\text{kg}))</td>
<td>Enthalpy ((\text{kJ/kg}))</td>
<td>Entropy ((\text{kJ/kgK}))</td>
</tr>
<tr>
<td>------------------------------</td>
<td>----------------</td>
<td>-------------------------------</td>
<td>------------------------</td>
<td>-----------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>(-40)^{b)}</td>
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<td>0.21256</td>
<td>153.93</td>
<td>387.72</td>
</tr>
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<td>388.09</td>
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<td>389.01</td>
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<td>Entropy (kJ/kgK)</td>
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<td>Temperature (°C)</td>
<td>Pressure (MPa)</td>
<td>Liquid Density (kg/m³)</td>
<td>Vapor Volume (m³/kg)</td>
<td>Enthalpy (kJ/kg)</td>
<td>Entropy (kJ/kgK)</td>
</tr>
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<td>Temperature ($^\circ$C)</td>
<td>Pressure (MPa)</td>
<td>Liquid Density (kg/m$^3$)</td>
<td>Vapor Volume (m$^3$/kg)</td>
<td>Enthalpy (kJ/kg)</td>
<td>Entropy (kJ/kgK)</td>
</tr>
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<tr>
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<td>Vapor</td>
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<td>Vapor</td>
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<td>523.8</td>
<td>0.00191</td>
<td>366.59</td>
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</table>
function [t_room, inttemp1, work, intevap1, intcond1, mass, kg] = Cycle(inttemp, freq, t_air, c_freight, m_freight, perimeter, height, t_wall, area_piston, stroke, w_f_evap, w_f_cond, eev_op, intevap, intcond)
%inputs
frq=freq;
dt_inc = .25; % resolution of temperature differential
toler = .05; % tolerance
tim_int=5; % time step for calculation in seconds
t_room=inttemp;
dt_super = 10;
dt_sub = 5;
etas_isen = -0.0036*freq+.836;
etas_vol = 0.002*freq+0.7;
poles=2;
c_d=0.2;
A_eev = .0011;

% initial values
dt_evap=intevap;
dt_cond=intcond;


t_evap= t_room - dt_evap;
t_cond = t_air + dt_cond;

[dummy q_c w_c kgs_r] = compressor
{t_evap,t_cond,dt_super,dt_sub,etas_isen,etas_vol,area_piston, stroke,
frq,poles);
quv_evap = evaporator
(0.00635,0.0005,2,400,10,.2,2,400,8,0.025,0.0325,120,1500,7000,0.0001,0
.03,0.9,w_f_evap,t_evap,t_room,0.25);
dt_evap = dt_evap*(q_c/q_evap) ;
qu_cond = condenser2
(0.00635,0.0005,3,400,20,.2,6,600,8,0.025,0.0325,120,1500,7000,0.0001,1
,0.03,0.9,w_f_cond,t_air - dt_cond,t_air);
dt_cond= dt_cond*(q_c+ w_c)/q_cond ;


t_evap= t_room - dt_evap;
t_cond = t_air + dt_cond;

[p4a, vf4a, vg4a, hf4a, hg4a,
cpf4a,cpg4a,muf4a,mug4a,kf4a,kg4a,prf4a,prg4a,sf4a,sq4a] = r_22
(t_evap) ;
[p2a, vf2a, vg2a, hf2a, hg2a,
cpf2a,cpg2a,muf2a,mug2a,kf2a,kg2a,prf2a,prg2a,sf2a,sq2a] = r_22
(t_cond) ;

mass_flow = c_d*A_eev*((2/vf2a)*(p2a-p4a)*10^6)^(.5)*eev_op;
kg=kgs_r;
while mass_flow < kgs_r

end
mass=mass_flow;

dt_evap = t_room - t_evap;

[dummy q_c w_c kgs_r]= compressor
(t_evap,t_cond,dt_super,dt_sub,eta_isen,eta_vol,area_piston, stroke,
frq, poles);
% t_room = coldroom(150,3,
7,1,1,0,0,15,15,t_room,t_air,q_c,20000,1600,tim_int);
% end
t_room = coldroom(perimeter,height,
7,1,1,t_wall,0,0,15,15,t_room,t_air,q_c,m_freight,c_freight,tim_int)
;
inttempl=t_room;
intevap=dt_evap;
intcond=dt_cond;
work=w_c;

function [COP, Cooling, Work, mass_flow] = compressor
(t_evap,t_cond,dt_super,dt_sub,eta_isen,eta_vol,area_piston, stroke,
frq, poles)

% OUTPUT
%COP: coefficient of performance [-]
% Cooling: cooling effect in evaporator [W]
%Work: work input to compressor [W]
%mass_flow: refrigerant mass flow rate [kg/s]

% INPUT
%t_evap: saturation temp. of the evaporator [C]
%t_cond: saturation temp. of the condenser [C]
%dt_super: amount of superheat at the suction [K]
%dt_sub: amount of subcool at the expansion valve [K]
%eta_isen: isentropic efficiency [-]
%eta_vol: volumetric efficiency [-]
%area_piston: area of the piston head [m^2]
%stroke: stroke of the piston [m]
%frq: frequency of the electrical power [1/s]
%poles: number of the double poles in the electrical motor [-]

% Code
[p4a, vf4a, vg4a, hf4a, hg4a,
cpf4a, cpg4a, mug4a, kf4a, kg4a, prf4a, prg4a, sf4a, sg4a] = r_22
(t_evap) ;
[p2a, vf2a, vg2a, hf2a, hg2a,
cpf2a, cpg2a, mug2a, kf2a, kg2a, prf2a, prg2a, sf2a, sg2a] = r_22
(t_cond) ;
t_eev = t_cond - dt_sub;
(t_eev) ;

t_suction = t_evap + dt_super; % discharge temp. from compressor
vg_suction = vg4a*(t_suction+273)/(t_evap+273); % specific volume at
the compressor suction

ds = (sg4a+cpg4a*log((t_suction+273.15)/(t_evap+273.15))- sg2a)   ;
t_discharge=(t_cond+273.15)*exp(ds/(cpg2a));
t_discharge = t_discharge - 273.15;
entp_1 = hg4a + cpg4a*(t_suction - t_evap);
entp_2 = hg2a + cpg2a*(t_discharge - t_cond);
comp_w = (entp_2 - entp_1);

dt = (1/eta_isen - 1)*(comp_w)/cpg2a; % actual temperature difference between saturated line and discharge from compressor

t_discharge = t_discharge + dt;
comp_w = comp_w/eta_isen; % compressor work input

ns = 60*frq/poles; % revolution speed
displacement = area_piston*stroke*ns;
mass_flow = displacement*eta_vol/vg_suction;

evap_q = (hg4a - hf3) + cpg4a*(t_suction - t_evap);

cond_q = (comp_w + evap_q);

COP = evap_q / comp_w; % coefficient of performance
Cooling = evap_q * mass_flow; % cooling effect
Work = comp_w * mass_flow;

%%%%%%%%%%%%%%%%%%%  R22 Properties  ##########################

function [p, vf, vg, hf, hg, cpf, cpg, muf, mug, kf, kg, prf, prg, sf, sg] = r_22(temp)
  x = temp;

  % p: pressure
  p1 = 1.144e-006;
p2 = 0.000206;
p3 = 0.01596;
p4 = 0.4933;
  p = p1*x^3 + p2*x^2 + p3*x + p4;

  % vf: specific volume of the liquid
  p1 = 8.582e-014;
p2 = 1.869e-013;
p3 = -4.129e-010;
p4 = 1.124e-008;
p5 = 2.58e-006;
p6 = 0.0007806;
  vf = p1*x^5 + p2*x^4 + p3*x^3 + p4*x^2 + p5*x + p6;

  % vg: specific volume of the vapor

91
\[ p1 = -1.008e-010 ; \]
\[ p2 =  1.538e-008 ; \]
\[ p3 = -5.888e-007 ; \]
\[ p4 =  9.589e-006 ; \]
\[ p5 =   -0.001215 ; \]
\[ p6 =      0.0521 ; \]

\[ vg = p1x^5 + p2x^4 + p3x^3 + p4x^2 + p5x + p6; \]

%hf: enthalpy of the liquid

\[ p1 =    0.002604 ; \]
\[ p2 =       1.206 ; \]
\[ p3 =       198.8 ; \]

\[ hf = p1x^2 + p2x + p3; \]
\[ hf=hf*1000; \]

%hg: enthalpy of the vapor

\[ p1 =  -5.83e-009 ; \]
\[ p2 = -4.656e-008 ; \]
\[ p3 =  2.166e-005 ; \]
\[ p4 =  -0.001713 ; \]
\[ p5 =     0.3314 ; \]
\[ p6 =     404.8 ; \]

\[ hg=p1x^5 + p2x^4 + p3x^3 + p4x^2 + p5x + p6; \]
\[ hg=hg*1000; \]

%cpf: specific heat of the liquid

\[ p1 =  2.305e-005 ; \]
\[ p2 =    0.001019 ; \]
\[ p3 =     -0.03044 ; \]
\[ p4 =     1.652 ; \]
\[ p5 =     1189 ; \]

\[ cpf = p1x^4 + p2x^3 + p3x^2 + p4x + p5; \]

%cpg: specific heat of the vapor

\[ p1 =  3.149e-005 ; \]
\[ p2 =    0.001121 ; \]
\[ p3 =   -0.06334 ; \]
\[ p4 =     2.763 ; \]
\[ p5 =     773.4 ; \]

\[ cpg = p1x^4 + p2x^3 + p3x^2 + p4x + p5; \]
%muf: viscosity of the liquid

\[ \text{muf} = p_1 x^2 + p_2 x + p_3; \]
\[ \text{muf} = \text{muf} \times 10^{-4}; \]

%mug: viscosity of the vapor

\[ \text{mug} = p_1 x^3 + p_2 x^2 + p_3 x + p_4; \]
\[ \text{mug} = \text{mug} \times 10^{-4}; \]

%kf: heat conduction coefficient of liquid

\[ \text{kf} = p_1 x^2 + p_2 x + p_3; \]

%kg: heat conduction coefficient of vapor

\[ \text{kg} = p_1 x + p_2; \]

%prf: prandtl number of liquid

\[ \text{prf} = p_1 x^4 + p_2 x^3 + p_3 x^2 + p_4 x + p_5; \]

%prg: prandtl number of vapor

\[ \text{prg} = p_1 x^3 + p_2 x^2 + p_3 x + p_4; \]

%sf: entropy of liquid

\[ p_1 = 0.004368; \]
\[
p2 = 0.9965;
\]

\[
sf = p1 \times x + p2;
sf = sf \times 1000;
\]

\% sg: entropy of the vapor

\[
p1 = -1.235e-007;
p2 = 6.16e-006;
p3 = -0.001341;
p4 = 1.751;
\]

\[
sg = p1 \times x^3 + p2 \times x^2 + p3 \times x + p4;
sg = sg \times 1000;
\]

\textit{function} q_c = \textit{condenser2} (r, ttube, tubelength, finpm, width, depth, tubeface, rows, xt, xl, ha, hr, k_fin, t_fin, cpg2a, f, eta_fan, W_fan, t_c, t_a)

\% condenser2

\% Input
\% r: tube radius [m]
\% ttube: tube thickness [m]
\% tubelength: tube length [m]
\% finpm: number of the fins per meter [-]
\% width: width of the heat exchanger [m]
\% depth: depth of the heat exchanger [m]
\% tubeface: number of tubes from front view [-]
\% rows: number of tube rows from upward view [-]
\% xt: transverse tube pitch [m]
\% xl: longitudinal tube pitch [m]
\% ha: air side convective heat transfer coefficient
\% hr: tube side convective heat transfer coefficient
\% k_fin: heat conduction coefficient in fin material
\% t_fin: thickness of the fin sheet [m]
\% kg_s_a: mass flow rate of the air [kg/s]
\% cpg2a: specific heat coefficient of the refrigerant vapor
\% f: friction factor
\% eta_fan: fan efficiency
\% W_fan: fan input power
\% t_c: condensing temperature
\% t_a: air temperature

\% Output
\% q: heat rejection rate [W]

\% Air propertis
\textit{cp} _\textit{air} = 1000; \% air specific heat
\textit{rou} = 1.2; \% air density

\% area
\textit{at} = 3.14 \times (r - ttube)^2; \% single tube area
\textit{afin} = 2 \times (tubelength \times finpm) \times width \times depth; \% fin area
\textit{atube} = 3.14 \times 2 \times (r) \times tubelength \times tubeface \times rows; \% total tube area
\textit{atotal} = afin + atube; \% total area
\textit{A} _\textit{fr} = width \times tubelength; \% frontal area
\textit{A} _\textit{min} = A _\textit{fr} - 2 \times r \times tubeface \times tubelength - finpm \times tubelength \times t_fin; \% minimum area
% fan

\[ G = \left(2\cdot\text{rou}\cdot W_{\text{fan}}\cdot\eta_{\text{fan}}\cdot\frac{1}{f}\right)\cdot\left(\frac{1}{\text{atotal}}\right)^{\frac{1}{3}}; \]  
\% mass flux
\[ \text{kgs}_a = G\cdot A_{\text{min}}; \]  
\% mass flow rate

% fin efficiency

\[ \text{si} = \frac{x\text{t}}{2\cdot r}; \]
\[ \text{beta} = \left(\frac{1}{x\text{t}}\right)\cdot\left(x_l^2 + \frac{(x_t^2)}{4}\right)^{0.5}; \]
\[ \text{req} = \frac{1.27\cdot\text{si}\cdot(\text{beta}-0.3)^{0.5}}{r}; \]  
\% equal radius
\[ \text{phi} = (\frac{\text{req}}{r} -1)\cdot(1+0.35\cdot\log(\text{req}/r)); \]
\[ m = \frac{2\cdot h\text{a}}{\left(k_{\text{fin}}\cdot t_{\text{fin}}\right)^{0.5}}; \]
\[ \text{nf} = \frac{\tanh(m\cdot\text{req}\cdot\text{phi})}{m\cdot\text{req}\cdot\text{phi}}; \]  
\% fin efficiency
\[ \text{ns} = 1 - \left(\frac{a_{\text{fin}}}{\text{atotal}}\right)\cdot(1-nf); \]  
\% area efficiency

% overall heat transfer coef.

\[ \text{ua} = \frac{1}{\left(\frac{1}{\text{ns}\cdot h\text{a}\cdot \text{atotal}}\right) + \frac{1}{(h\text{r}\cdot \text{atube})}}; \]  
\% overall heat transfer area

\[ \text{cair} = \text{kgs}_a\cdot c_{\text{p}\_\text{air}}; \]  
\% air heat capacity
\[ \text{cvap} = \left(kgs\_r\cdot c_{pg2a}\right); \]
\[ \text{cvap} = \text{inf}; \]

\[ \text{if } \text{cair} > \text{cvap} \]
\[ \quad \text{cr} = \frac{\text{cvap}}{\text{cair}}; \]
\[ \quad \text{cmin} = \text{cvap}; \]
\[ \text{else} \]
\[ \quad \text{cr} = \frac{\text{cair}}{\text{cvap}}; \]
\[ \quad \text{cmin} = \text{cair}; \]
\[ \text{end} \]

\[ \text{ntu} = \frac{\text{ua}}{\text{cmin}}; \]  
\% number of transfer units
\[ \text{epsilon} = 1 - \exp(-\text{ntu}); \]  
\% heat exchanger efficiency

\[ \text{q}_c = \epsilon\cdot\text{cair}\cdot(t\_a - t\_c); \]  
\% heat rejection rate

\textbf{function } \text{q} = \text{evaporator}(r, ttube, tubelength, finpm, width, depth, tubeface, rows, xt, xl, ha, hr, k\_fin, t\_fin, f, eta\_fan, W\_fan, t\_e, t\_a, lhr)

% evaporator
\{0.00635, 0.0005, 0.35, 500, 0.35, 0.35, 10, 3, 0.025, 0.025, 120, 400, 700, 0.000
1, 0.25, 0.03, 0.8, 220, 50, 35, 0.25\}
% Input
% r: tube radius [m]
% ttube: tube thickness [m]
% tubelength: tube length [m]
% finpm: number of the fins per meter [-]
% width: width of the heat exchanger [m]
% depth: depth of the heat exchanger [m]
% tubeface: number of tubes from front view [-]
% rows: number of tube rows from upward view [-]
% xt: transverse tube pitch [m]
%xl: longitudinal tube pitch [m]
%ha: air side convective heat transfer coefficient
%hr: tube side convective heat transfer coefficient
%k_fin: heat conduction coefficient in fin material
%t_fin: thickness of the fin sheet [m]
%kgs_a: mass flow rate of the air [kg/s]
%f: friction factor [-]
%eta_fan: fan efficiency [-]
%W_fan: fan input power [W]
%t_e: evaporating temperature [C]
%t_a: cold room air temperature [C]
%latent heat ration [-]

% Output
%q: heat absorption rate [W]

% Air properties
cp_air = (1 + lhr/(1-lhr))* 1000; % air specific heat
rou=1.2; % air density

%area
at=3.14* (r-ttube)^2 ; % single tube area
afin = 2*(tubelength*finpm) *width * depth ; % fin area
atube = 3.14*2*(r)*tubelength*tubeface*rows; % total tube area
atotal = afin + atube; % total area
A_fr = width * tubelength; % frontal area
A_min = A_fr - 2*r*tubeface*tubelength - finpm*tubelength*t_fin; % minimum area

%fan
G = (2*rou*W_fan*eta_fan*(1/f)* (1/atotal))^(1/3); % mass flux
kgs_a = G*A_min; % mass flow rate

% fin efficiency
si=xt/(2*r);
beta=(1/xt)*(xl^2 + (xt^2)/4)^0.5;
req=(1.27*si*(beta-0.3)^(0.5))*r;% equal radius
phi = (req/r -1)*(1+0.35*log(req/r));
m=(2*ha/(k_fin*t_fin))^0.5;
nf = tanh(m*req*phi)/(m*req*phi); % fin efficiency
ns = 1- (afin/atotal)*(1-nf); % area efficiency

% overall heat transfer coef.
ua= 1/(1/(ns*ha*atotal) + 1/(hr*atube)); % overall heat transfer . area

cair=kgs_a*cp_air; % air heat capacity
% cvap = kgs_r*cpg2a;
cvap=inf;

if cair>cvap
    cr=cvap/cair;
    cmin = cvap;
else
cr = cair/cvap;
cmin = cair;
end

ntu = ua/cmin; % number of transfer units
epsilon = 1-exp(-ntu); % heat exchanger efficiency
q = epsilon*cair*(t_a - t_e); % heat rejection rate

function ti2 = coldroom (P, h, k1, k2, k3, x1, x2, x3, hi, he, ti, to, q_c, m, c, tint)

%ti2 = coldroom(9,3,
1,1,1,.01,0,0,10,10,t_set,t_air,q_c,1000,4000,tint)
%Input
%P: perimeter of the cold room building [m]
%h: height of the cold room building [m]
%ki: conductivity of the layer []
%xi: thickness of the layer [m]
%ti: initial temperature [C]
%to: outside ambient temperature [C]
%q_c: cooling effect from the refrigeration cycle in the evaporator [W]
%m: mass of the freight [kg]
%c: specific heat of the freight [J/kg.k]
%tint: time interval [s]

%Output
% ti2: final temperature [C]

U = (1/hi) + (x1/k1 + x2/k2 + x3/k3) + (1/he) )^-1; % overall heat transfer coefficient
A = P*h; % exterior walls of the cold room
dT = to - ti; % temperature difference
q_l = U*A*dT; % heat loss from the wall by conduction
q = q_c - q_l;
ti2 = ti - tint*q/(m*c);
Appendix C

Specifications of the compressor, condenser, evaporator and cold room are presented in the following tables:

### Table 5 Compressor specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total area of piston head</td>
<td>0.03 m²</td>
<td>Number of motor poles</td>
<td>4</td>
</tr>
<tr>
<td>Total length of stroke</td>
<td>0.006 m</td>
<td>Rated frequency</td>
<td>60 Hz</td>
</tr>
</tbody>
</table>

### Table 6 Condenser specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube radius</td>
<td>0.00635 m</td>
<td>Number of tube from front view</td>
<td>600</td>
</tr>
<tr>
<td>Tube thickness</td>
<td>0.0005 m</td>
<td>Number of tube rows from upward view</td>
<td>8</td>
</tr>
<tr>
<td>Tube length</td>
<td>3 m</td>
<td>Transverse tube pitch</td>
<td>0.025 m</td>
</tr>
<tr>
<td>Number of fins per meter</td>
<td>400</td>
<td>Longitudinal tube pitch</td>
<td>0.0325 m</td>
</tr>
<tr>
<td>Width of heat exchanger</td>
<td>20 m</td>
<td>Thickness of fin sheet</td>
<td>0.0001</td>
</tr>
<tr>
<td>Depth of heat exchanger</td>
<td>.2 m</td>
<td>Rated power</td>
<td>3550 w</td>
</tr>
</tbody>
</table>
### Table 7 Evaporator specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube radius</td>
<td>0.00635 m</td>
<td>Number of tube from front view</td>
<td>400</td>
</tr>
<tr>
<td>Tube thickness</td>
<td>0.0005 m</td>
<td>Number of tube rows from upward view</td>
<td>8</td>
</tr>
<tr>
<td>Tube length</td>
<td>2 m</td>
<td>Transverse tube pitch</td>
<td>0.025 m</td>
</tr>
<tr>
<td>Number of fins per meter</td>
<td>400</td>
<td>Longitudinal tube pitch</td>
<td>0.0325 m</td>
</tr>
<tr>
<td>Width of heat exchanger</td>
<td>10 m</td>
<td>Thickness of fin sheet</td>
<td>0.0001</td>
</tr>
<tr>
<td>Depth of heat exchanger</td>
<td>.2 m</td>
<td>Rated power</td>
<td>2500 w</td>
</tr>
</tbody>
</table>

### Table 8 Cold room specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Perimeter</td>
<td>75 m</td>
<td>Thickness of layer</td>
<td>0.1</td>
</tr>
<tr>
<td>Height</td>
<td>6 m</td>
<td>Mass of freight</td>
<td>20000 Kg</td>
</tr>
</tbody>
</table>
Appendix D

Tables of Thermal Properties and Specific Heat

Thermal properties of selected insulating and building material are shown in Table 9.

Table 9 thermal properties of building material [37]

<table>
<thead>
<tr>
<th>Material</th>
<th>Insulating &quot;R&quot; Factor hr sq ft deg F BTU</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Insulation</strong></td>
<td></td>
</tr>
<tr>
<td>Fiberglass batts</td>
<td>3.22/inch</td>
</tr>
<tr>
<td>Fiberglass, loose</td>
<td>2.55/inch</td>
</tr>
<tr>
<td>Fiberglass, board</td>
<td>4.00/inch</td>
</tr>
<tr>
<td>Cellular glass (Foamglas)</td>
<td>2.86/inch</td>
</tr>
<tr>
<td>Styrofoam, extruded</td>
<td>5.26/inch</td>
</tr>
<tr>
<td>Styrofoam, bendboard</td>
<td>4.17/inch</td>
</tr>
<tr>
<td>Polyurethane, board</td>
<td>6.25/inch</td>
</tr>
<tr>
<td>Polyurethane, foamed-in-place</td>
<td>6.25/inch</td>
</tr>
<tr>
<td>Polyisocyanurate, board</td>
<td>7.04/inch</td>
</tr>
<tr>
<td><strong>Building Materials</strong></td>
<td></td>
</tr>
<tr>
<td>Fir plywood</td>
<td>1.25/inch</td>
</tr>
<tr>
<td>Fiberboard sheathing 1/2&quot;</td>
<td>1.32</td>
</tr>
<tr>
<td>Particle board 1/2&quot; (Aspenite)</td>
<td>0.92</td>
</tr>
<tr>
<td>Gypsum board (5/8&quot;)</td>
<td>0.56</td>
</tr>
<tr>
<td>Concrete, cast</td>
<td>0.11/inch</td>
</tr>
<tr>
<td>Concrete block 8&quot;</td>
<td>1.11</td>
</tr>
<tr>
<td>Concrete block 12&quot;</td>
<td>1.28</td>
</tr>
<tr>
<td>Glass, single pane</td>
<td>0.10</td>
</tr>
<tr>
<td><strong>Air Film and Air Gaps</strong></td>
<td></td>
</tr>
<tr>
<td>Air Film, outside or warm</td>
<td>0.28</td>
</tr>
<tr>
<td>Air Film, inside or cold</td>
<td>0.17</td>
</tr>
<tr>
<td>Air gap 1 inch or greater</td>
<td>0.72</td>
</tr>
</tbody>
</table>

Density of 1.5 lb/cu ft

Metric Conversion Factors: hr sq ft deg F / BTU x 0.276 = sq m deg K / W

Extracted from:
Agricultural Engineering Extension Bulletin 448
Cornell University, Ithaca, New York 14853
The specific heat and latent heat of several food products are presented in Table 10:

<table>
<thead>
<tr>
<th>Product</th>
<th>Water content % mass</th>
<th>High freezing point °C (°F)</th>
<th>Specific heat kJ/kg·K</th>
<th>Latent heat of freezing kJ/kg (Btu/lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Above freezing</td>
<td>Below freezing</td>
</tr>
<tr>
<td>Apples</td>
<td>84</td>
<td>−1.1 (30)</td>
<td>3.78 (0.902)</td>
<td>1.90 (0.453)</td>
</tr>
<tr>
<td>Chicken</td>
<td>74</td>
<td>−2.8 (27)</td>
<td>3.53 (0.843)</td>
<td>1.77 (0.423)</td>
</tr>
<tr>
<td>Peas</td>
<td>74</td>
<td>−0.6 (31)</td>
<td>3.53 (0.792)</td>
<td>1.77 (0.423)</td>
</tr>
<tr>
<td>Ham</td>
<td>56</td>
<td>−1.7 (29)</td>
<td>3.08 (0.735)</td>
<td>1.55 (0.368)</td>
</tr>
<tr>
<td>Salmon</td>
<td>64</td>
<td>−2.2 (28)</td>
<td>3.28 (0.783)</td>
<td>1.65 (0.392)</td>
</tr>
<tr>
<td>Sirloin beef</td>
<td>56</td>
<td>−0.8 (31)</td>
<td>3.93 (0.938)</td>
<td>1.97 (0.471)</td>
</tr>
<tr>
<td>Strawberries</td>
<td>90</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
VITA

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Mr. Afzali joined American University of Sharjah to pursue his Master in Mechatronics Engineering. He was a graduate teaching assistant at the American University of Sharjah for two years.