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# Improving Energy Efficiency of an Industrial Refrigeration System Through Model-Based Sequencing of Compressors and Condensers

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# IMPROVING ENERGY EFFICIENCY OF AN INDUSTRIAL REFRIGERATION SYSTEM THROUGH MODEL-BASED SEQUENCING OF COMPRESSORS AND CONDENSERS

by

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# Abstract

The refrigeration cycle is highly dynamic and changing. From the time the compressor commences to run until it stops, the suction pressure, evaporator temperature, the rate of heat exchange, refrigerant flow and many other factors are continuously changing. The total efficiency of the system changes through the entire life cycle. The load on refrigeration system is also not constant; it varies based on ambient conditions and production activities.

Two control algorithms are developed to predict the load on system based on predicted ambient wet-bulb temperatures and to control the capacity of compressors while improving the compressor energy efficiency. The approach here is to create a database of different operating parameters and utilize that database to create model of system energy. The capacity control algorithm provides the energy efficient capacity variation and sequencing of compressors based on compressor models.

Main conclusions derived from the work are as follow: due to many practical reasons compressors may consume different amount of energy even if they are rated same. In high power systems this difference can be significant. In multi compressor system, whenever there is a redundancy in compressors, it is advantageous to evaluate compressor energy consumption in different operating conditions and then selecting the most efficient compressor to run. In case of multi screw compressor system, rather than applying specific rules for loading and unloading of compressors via slide valve, a model based approach, which uses database of different compressor parameters and minimizes energy according to each compressors' performance characteristics, is more efficient.

# 1. Introduction of Refrigeration System and the Vapor Compression Refrigeration Cycle

In this chapter basic building blocks of the refrigeration system are discussed. The chapter explains working of an evaporator, compressor, condenser, expansion valve and refrigerant. Basic thermodynamic principles related to the vapor compression refrigeration cycle are also discussed in brief. We will also discuss vapor compression refrigeration cycle. Different stages of the cycle are explained by referring to the pressure vs. enthalpy diagram. The last part of this chapter discusses motivation behind the work, literature survey and outline of the thesis.

## 1.1 What is Refrigeration?

The most widely used explanation of this question is "Refrigeration is a process in which work is done to remove the heat from one location to another". Let's understand this by one day-to-day example - If you were to place a hot cup of coffee on a table and leave it for a while, the heat in the coffee would be transferred to the materials in contact with the coffee, i.e. the cup, the table and the surrounding air. As the heat is transferred, the coffee cools. Using the same principle, refrigeration works by removing heat from a product and transferring that heat to the outside air. Refrigeration has many applications including but not limited to; household refrigerators, industrial freezers, cryogenics, air conditioning, and heat pumps. In order to satisfy the Second Law of Thermodynamics, some form of work must be performed to accomplish this. The work is traditionally done by mechanical work but can also be done by magnetism, laser or other means.

## 1.2 Refrigeration System Components

There are five basic components of a refrigeration system are evaporator, compressor, condenser, expansion valve and refrigerant. In order for the refrigeration cycle to operate successfully each component must be present within the refrigeration system. A description of each of the components of a refrigeration system is given below.

### 1.2.1 The Evaporator

Purpose of the evaporator is to remove unwanted heat from the product, via liquid refrigerant. Liquid refrigerant contained within the evaporator is boiling at a low-pressure. Level of this pressure is determined by two factors: The rate at which the heat is absorbed from the product to the liquid refrigerant in the evaporator and rate at which the low-pressure vapor is removed from the evaporator by the compressor. To enable heat transfer, the temperature of the liquid refrigerant must be lower than the temperature of the product being cooled. Once transferred, the liquid refrigerant is drawn from the evaporator by the compressor via the suction line. When leaving the evaporator coil the liquid refrigerant is in vapor form.

Evaporators that operate with a refrigerant temperature below the freezing point of water and dew point of the conditioned space will build frost on the coils during operation. Frost accumulation degrades the performance of an evaporator by reducing the

UA and impeding the airflow. Periodically the frost must be removed from the coil surface. Several defrosting methods are commonly used. Hot gas, hot water, electric heat, and warm air can all be used to melt the frost off of evaporator coils. Figure 1.1 shows evaporator coils with fins to increase the rate of heat transfer. In industrial and commercial systems fans are used to accelerate the heat transfer by forced convection.

Figure 1.1 Evaporative Coils

### 1.2.2 The Compressor

The refrigeration industry widely uses screw and reciprocating compressors. Compared to reciprocating compressors screw compressors are compact and create less noise. At full load they are more efficient. But at part load reciprocating once are more efficient. For this thesis we have worked with screw compressors and we will discuss them in briefly. The rotary screw compressor is designed for low pressure applications with inlet pressures ranging from vacuum pressure up to 100 psig and discharge pressures up to 350 psig. There are some screw machines available capable of operating at higher pressures by using cast steel casings but these are not yet commonly used in the natural gas industry due to capital cost and availability. The purpose of the compressor is to draw the low-temperature, low-pressure vapor from the evaporator via the suction line. Once drawn, the vapor is compressed and rises in temperature. Therefore, the compressor transforms the vapor from a low-temperature vapor to a high-temperature vapor, in turn increasing the pressure. The vapor is then released from the compressor in to the discharge line. The rotary screw compressor is a positive displacement machine that operates without the need for suction or discharge valves. It has the ability to vary suction volume internally while reducing part load power consumption. Screws provide a much wider operating range and lower maintenance costs than conventional reciprocating machines. The machines are much smaller and create much lower vibration levels than piston machines as well. Some of the major components include one set of male and female helically grooved rotors, a set of axial and radial bearings and a slide valve, all encased in a common housing (Reindl et al., 2002).

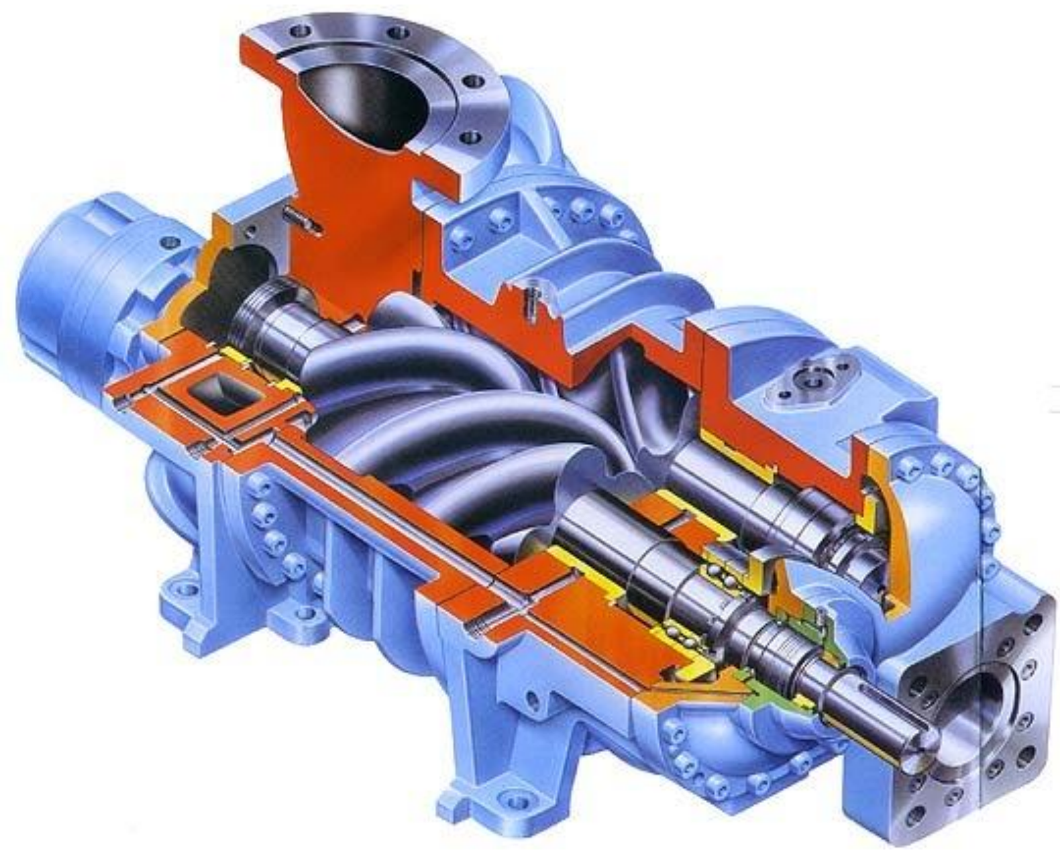

Figure 1.2 Industrial Screw Compressors

### 1.2.3 The Condenser

The purpose of condenser is to extract heat from refrigerant to outside air. Condenser is usually installed on the reinforced roof of building, which enables the transfer of heat. Fans mounted above the condenser unit are used to draw air through the condenser coils.

The temperature of high-pressure vapor determines temperature at which the condensation begins. As heat has to flow from condenser to the air, condensation temperature must be higher than that of the air; usually between - 12°C and -1°C. The high-pressure vapor within the condenser is then cooled to the point where it becomes a liquid refrigerant once more, whilst retaining some heat. The liquid refrigerant then flows from the condenser in to the liquid line. Chapter 3 discusses evaporative condenser in more details.

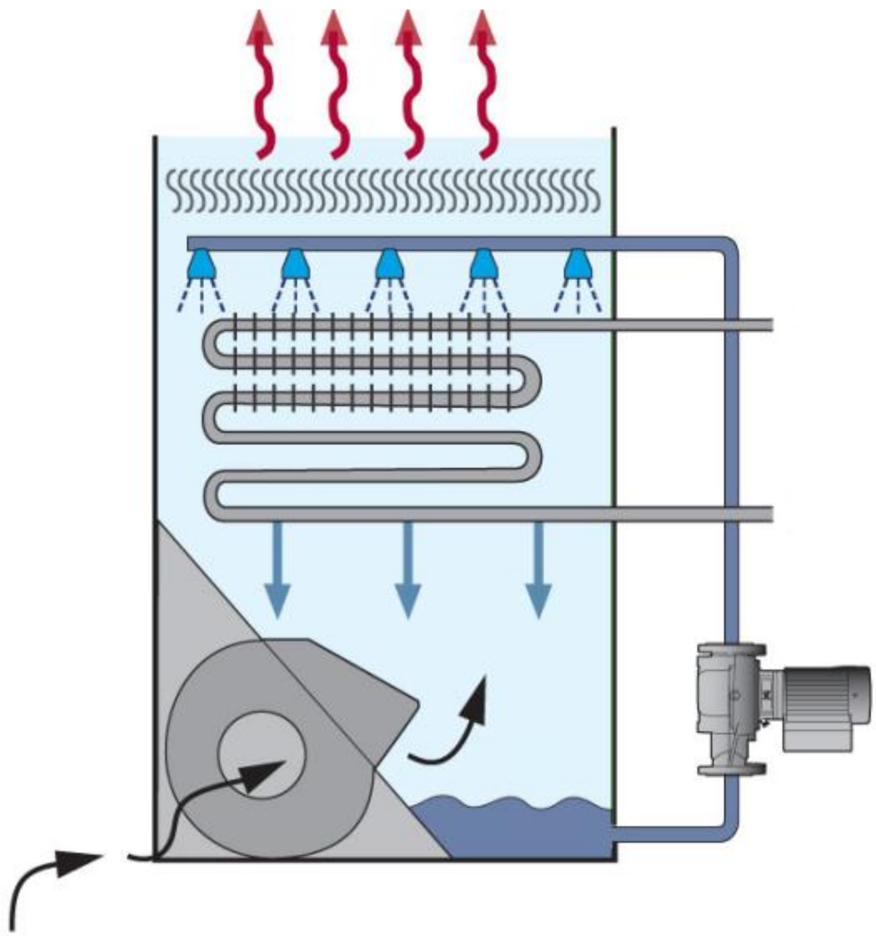

Figure 1.3 Evaporative Condensers

### 1.2.4 The Expansion Valve

Within the refrigeration system, the expansion valve is located at the end of the liquid line, before the evaporator. The high-pressure liquid reaches the expansion valve, having come from the condenser. The valve then reduces the pressure of the refrigerant as it passes through the orifice, which is located inside the valve.

On reducing the pressure, the temperature of the refrigerant also decreases to a level below the surrounding air. This low-pressure, low-temperature liquid is then pumped in to the evaporator. Figure 1.4 shows orifice tubes that reduces the pressure and intern temperature of refrigerant.

Figure 1.4 Thermal Expansion Valve

### 1.2.5 The Refrigerant

The refrigerant is a medium for heat transfer. The type of refrigerant used will depend on the pressure capabilities of the system and the temperatures that have to be achieved during refrigeration. Common refrigerants used in various applications are ammonia, sulfur dioxide, and non-halogenated hydrocarbons such as methane. Ammonia is widely used in industrial set up because of low cost, strong odor and environment friendly properties.

## 1.3 Fundamentals of Thermodynamics

Before we turn to describe the vapor-compression refrigeration cycle some fundamental thermodynamically relations are briefly recalled.

**The First Law of Thermodynamics**

The first law of thermodynamics describes conservation of energy that is for an insulated system, the change of internal energy $\Delta U$ equals the sum of the applied work $W$and heat $Q$.

$$\Delta U = W + Q \qquad (1.1)$$

**The Specific Enthalpy**

The specific enthalpy ($h$) is a refrigerant specific property that only depends on the state of the refrigerant, i.e. pressure, temperature and quality. The specific enthalpy is defined as:

$$h \equiv u + Pv \qquad (1.2)$$

Where, $u$ is the specific internal energy, $P$ is the pressure and $v$ is the specific volume.

The enthalpy of a refrigerant can be interpreted as the quantity of energy supplied to the refrigerant to bring it from a certain initial reference state to its current state. By applying first law of thermodynamics on a finite volume with an entering ($\dot{m}_i$) and exiting ($\dot{m}_o$) mass flow, the internal energy increase of the volume can be written as (neglecting the potential and kinetic energy (Sonntag et al., 1998)):

$$\frac{dU}{dt} = \dot{Q} + \dot{W} + (\dot{m}_i h_i - \dot{m}_o h_o) \qquad (1.3)$$

For a steady state and steady flow process this gives:

$$\dot{Q} + \dot{W} = \dot{m}\,(h_o - h_i) \qquad (1.4)$$

**The Second Law of Thermodynamics and Entropy**

The second law of thermodynamic basically states that energy stored as heat, cannot be converted to the equivalent amount of work. This means that the efficiency of a process that involves transforming heat to work cannot under even ideal conditions become 1. If we focus on the compression process which among others takes place in the vapor compression cycle process in a refrigeration system, then the theoretically best one can do is to perform a reversible process, such that the increase of the involved entropy is 0. The specific entropy ($s$) is as the enthalpy a refrigerant specific property only dependent on the state of the refrigerant. By using the definition introduced in Sonntag *et al.* (1998), the entropy can be defined as:

$$dS = \left(\frac{\partial Q}{T}\right)_{rev} \qquad (1.5)$$

Where $T$ is the temperature and the sub subscript *rev* means it is defined in terms of a reversible process.

To get a true measure on how close the compression process is to the theoretical most efficient, that is to a reversible isentropic process, the isentropic efficiency ($\eta_{is}$) is introduced.

The isentropic efficiency is defined as (for a process where work is added (Sonntag et al., 1998):

$$\eta_{is} = \frac{W_{is}}{W_{real}} \tag{1.6}$$

Where $W_{is}$ is the required work for performing an isentropic compression process and $W_{real}$ is the real work added.

## 1.4 The Vapor Compression Refrigeration Cycle

The purpose of the vapor-compression cycle process is basically to remove heat from a cold reservoir (e.g. a cold storage room) and transfer it to a hot reservoir, normally the surroundings. The main idea is to let a refrigerant circulate between two heat exchangers, which are an evaporator and a condenser, see Figure 1.5.

In the evaporator the refrigerant "absorbs" heat from the cold reservoir by evaporation and "rejects" it in the condenser to the hot reservoir by condensation. In order to establish the required heat transfer, the evaporation temperature ($T_e$) has to be lower than the temperature in the cold reservoir ($T_{cr}$) and the condensation temperature ($T_c$) has to be higher than the temperature in the hot reservoir (normally the surroundings $T_a$), i.e. $T_e < T_{cr}$ and $T_c > T_a$. The refrigerant has the property (along with other fluids and gasses) that the saturation temperature ($T_{sat}$) uniquely depends on the pressure. At low pressure the corresponding saturation temperature is low and vice versa at high pressure.

This property is exploited in the refrigeration cycle to obtain a low temperature in the evaporator and a high temperature in the condenser simply by controlling respectively the evaporating pressure ($P_e$) and the condensing pressure ($P_c$). Between the evaporator and the condenser is a compressor. The compressor compresses the low pressure refrigerant ($P_e$) from the outlet of the evaporator to a high pressure ($P_c$) at the inlet of the condenser, hereby circulating the refrigerant between the evaporator and the condenser.

To uphold the pressure difference ($P_c > P_e$) an expansion valve is installed at the outlet of the condenser. The expansion valve is basically an adjustable nozzle that helps upholding a pressure difference (Larsen, 2005).

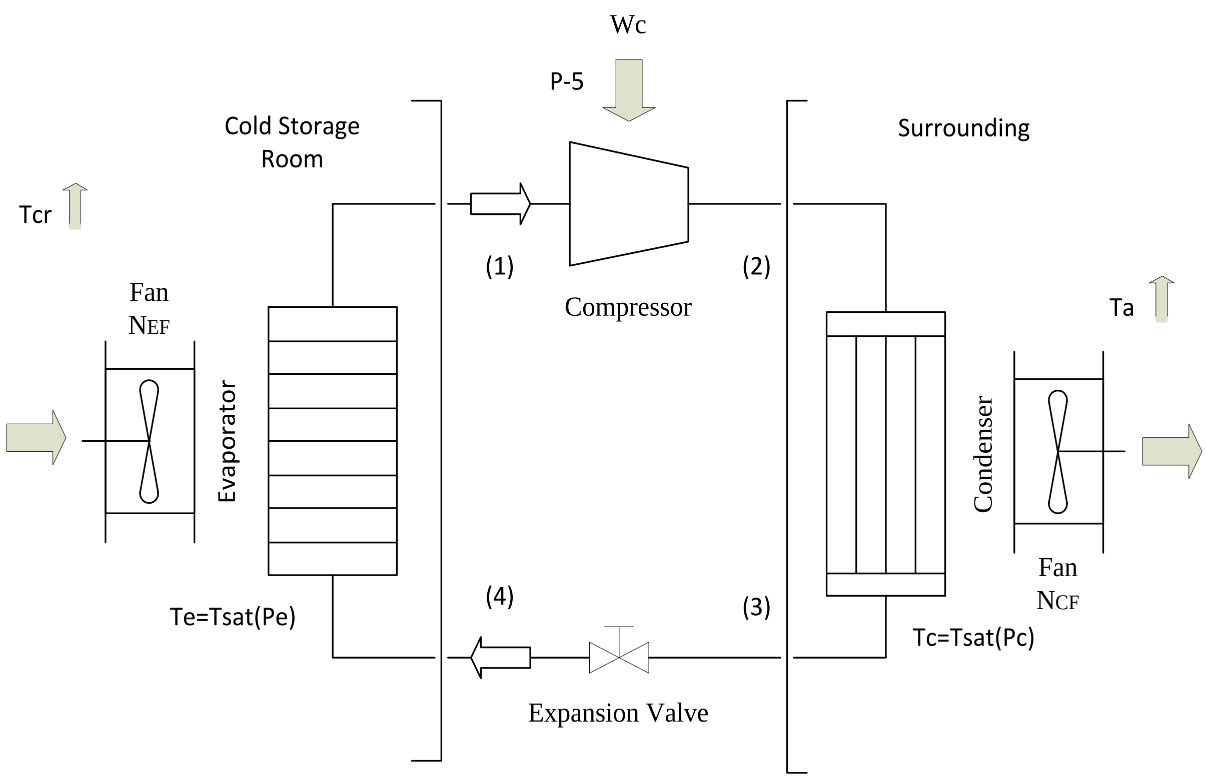


Figure 1.5 Schematic Diagram of Typical Vapor Compression Refrigeration Cycle

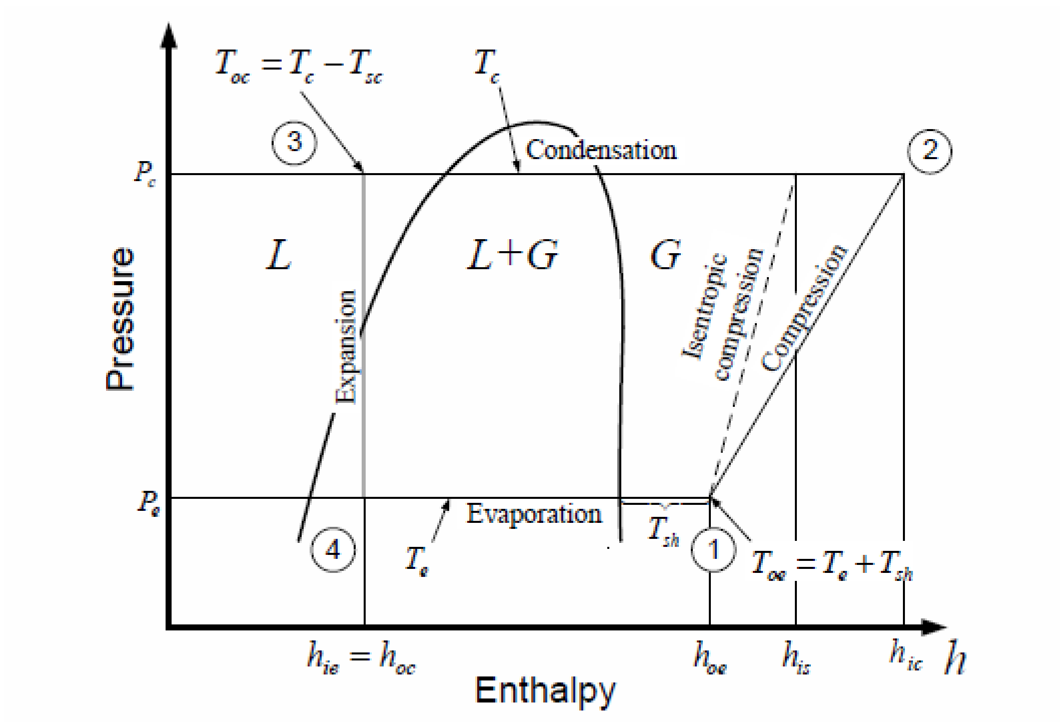


Figure 1.6 The vapor compression cycle in an *h* - *log*(*p*)-diagram (Larsen, 2005)

In Figure 1.6 the vapor compression cycle is plotted in a *h*-*log*(*P*)-diagram. The diagram is specific for a given refrigerant and gives a general view of the process cycle and the phase changes that takes place. In the *h*-*log*(*P*)-diagram, Figure 1.6, the specific enthalpy at the different state points of the refrigeration cycle is denoted by the sub-indices *i* and *o* for "inlet" and "outlet", plus *e* and *c* denoting "evaporator" and "condenser", i.e. $h_{oe}$ is the specific enthalpy at the outlet of the evaporator. The vapor compression cycle consists of 4 connected sub processes namely a compression, a condensation, an expansion and evaporation. We will go through each the processes following the numbers depicted in Figure 1.5 and 1.6 (Larsen, 2005).

**Compression; State Point 1-2**

At the inlet of the compressor the refrigerant is in a gas phase at low pressure and temperature. By compressing the refrigerant, the temperature as well as the pressure increases.

The required work for the compression can be found by forming a control volume around the compressor, assuming it is insulated (adiabatic compression) and using Eq. 1.4:

$$\dot{W}_C = \dot{m}(h_{ad} - h_{oe}) \qquad (1.7)$$

Where, $\dot{m}$ is the mass flow of refrigerant. If the compression is not adiabatic, then $h_{ic} \neq h_{ad}$ *i.e.* some heat is transmitted to the surroundings during the compression. In that case it is common to introduce a heat loss factor $f_q$, to compensate the measurements at the outlet of the compressor for the heat losses. $f_q$ is normally defined as the heat fraction of the applied compressor work that is transmitted to the surroundings (Sonntag et al.,1998):

$$f_q = \frac{(h_{ad} - h_{ic})}{(h_{ad} - h_{oe})} \qquad (1.8)$$

The work applied to the compressor ($\dot{W}_C$) can then using Eq. (1.7) and Eq. (1.8) is written as:

$$\dot{W}_C = \frac{1}{1-f_q}\ \dot{m}(h_{ic} - h_{oe}) \qquad (1.9)$$

**Condensation; State Point 2-3**

Form the outlet of the compressor the refrigerant flows into the condenser. The condenser enables a heat transfer ($\dot{Q}_C$) from the hot gaseous refrigerant to the surroundings. Because of the high pressure (the condensing pressure $P_c$) the refrigerant starts to condense at constant pressure changing its phase from gas into liquid (state point 2-3). A fan blowing air across the condenser helps increasing the heat transfer. Through the last part of the condenser, the refrigerant temperature is pulled down below the condensing temperature ($T_c$), creating a so-called sub cooling ($T_{sc}$). This sub cooling ensures that the entire refrigerant is fully condensed when it passes on to the expansion valve (state point 3). This is important because even a small number of gas bubbles in the liquid refrigerant would lower the mass flow through the valve dramatically, causing a major drop in the cooling capacity ($\dot{Q}_e$). The heat rejected in the condenser ($\dot{Q}_C$) can be computed forming a control volume around the condenser and using Eq. (1.4):

$$\dot{Q}_C = \dot{m}(h_{ic} - h_{oc}) \qquad (1.10)$$

Using the first law of thermodynamics on the system the heat rejected in the condenser can also be computed as:

$$\dot{Q}_C = \dot{Q}_e + \dot{W}_C - \dot{W}_C.f_q \qquad (1.11)$$

**Expansion; State Point 3-4**

The expansion valve separates the high pressure side from the low pressure side. When the refrigerant passes the valve (state point 3-4), it is therefore exposed to a large pressure drop causing some of the refrigerant to evaporate. In Figure 1.6 this can be seen as the process moves from the liquid phase ($L$) in state point 3, into the two-phase region in state point 4. This partial phase change causes the temperature to drop down to the evaporation temperature ($T_e$), determined by the low pressure ($P_e$). From the expansion valve the refrigerant flows to the evaporator. Since no work is done when the refrigerant passes the expansion valve ($\dot{W}= 0$) and the expansion valve is assumed insulated ($\dot{Q}= 0$) then according to Eq.(1.4) the inlet enthalpy ($h_{oc}$) to the valve equals the outlet enthalpy ($h_{ie}$), i.e. $hoc = h_{ie}$.

**Evaporation; State Point 4-1**

In the evaporator the low inlet temperature ($T_e$) enables a heat transfer from the cold reservoir (the cold storage room) to the refrigerant. Hereby the remaining part of the liquid refrigerant evaporates at a constant temperature ($T_e$) under heat "absorption" from the cold storage room. Like in the condenser a fan helps increasing the heat transfer by blowing air across the evaporator. At the outlet of the evaporator (state point 1) the entire refrigerant has evaporated and the temperature has increased slightly above the evaporating temperature ($T_e$). This small temperature increase is called the superheat ($T_{sh}$). The superheat is important to maintain, as it ensures that the entire refrigerant has evaporated, such that no liquid gets into the compressor (state point 1). The liquid could otherwise cause a breakdown of the compressor. The cooling capacity ($\dot{Q}_e$) can be computed forming a control volume around the evaporator and using Eq. (1.4).

$$\dot{Q}_e = \dot{m}(h_{oe} - h_{oc}) \qquad (1.12)$$

The refrigerant has now completed the vapor compression cycle and returns to the inlet of the compressor, state point 1 (Larsen, 2005).

## 1.5 Background and Motivation

Refrigeration and Air Conditioning systems can be found in many types of applications ranging from food and chemical preservation to space cooling to process cooling. Generally residential units are standard single compressor systems while the commercial systems are either custom built or standard off the shelf systems. In industrial scenario refrigeration systems are custom built. Custom systems are generally an assembly of components that have been tested and rated individually under design operating conditions but not necessarily rated as an integral, coordinated system. Therefore, in the assembled refrigeration plant, there is a possibility that the individual components may not provide optimum operation based on the deviation between the different components in the system. Consequently the final refrigeration plant may not operate optimally, resulting in lower operating efficiency and higher operating cost.

Table 1 presents information on the manufacturing industries that are intensive in process cooling and refrigeration (PC&R) energy consumption as monitored by the US Energy Information Administration (EIA) in 2006. The information in Table 1 indicates that in the Food and Beverage industries, process cooling and refrigeration represent one of the largest electrical end uses, while the Food and Chemical industries use are the largest energy users of process cooling and refrigeration in the manufacturing industries (D'Antonio et al., 2006).

The factors that influence the refrigeration system energy use are the inherent efficiency of design and refrigerant, the condition of the equipment, the control strategy and the load profile of the system (deviation of the operating cooling loads from the design cooling loads). In many cases, the installed refrigeration plants can benefit in subsequent control adjustments and or system operational changes based on assessed data.

The refrigeration cycle is dynamic and changing. From the time the compressor commences to run until it stops, the suction pressure, evaporator temperature, the rate of heat exchange, refrigerant flow and many other factors are continuously changing. The total efficiency of the system changes through the entire life cycle.

Due to the high cost of energy and growing consumer awareness about the energy saving there is a growing need for energy efficiency in the field of industrial refrigeration. The present work provides various techniques of reducing energy consumption of the refrigeration system.

| Industries | NAICS Code | Energy Consumption (Millions of kWh) | | PC&R % of Total (A/B) | % of Total US PC&R (A/D) |
|---|---|---|---|---|---|
| | | Process Cooling & Refrigeration (PC&R) (A) | Total (B) | | |
| Food | 311 | 17,679 | 67,390 | 26.2% | 28.6% |
| Beverage and Tobacco Products | 312 | 2,349 | 8,242 | 28.5% | 3.8% |
| Chemicals | 325 | 16,109 | 215,008 | 7.5% | 26.1% |
| All Manufacturing Industries | 311-339 | 61,763 (D) | 1,025,149 | 6% | 100.0% |

Table 1: Energy Consumed as a Fuel by End Use in Manufacturing Installations

(D'Antonio et al., 2006)

## 1.6 Literature

Numerous literatures can be found in the fields related to this study, but literature related to the specific aspect of industrial refrigeration control and operation strategies is rather sparse. Manske (1999) did a study on performance optimization of industrial refrigeration systems. He developed a detailed model of a vapor compression refrigeration system, including subcomponents. Primozic et al. (2003) discussed about staging of reciprocating compressors by following the rate of rise and fall of the suction pressure. Aprea and Renno (2004) performed an experimental analysis of a thermodynamic model of a vapor compression refrigeration plant on varying the compressor speed. Larsen (2005) developed a model based control of refrigeration system. In that work model based set point optimization control method was developed. The optimizing control is divided into two layers, where the system oriented top layers deals with set-point optimizing control and the lower layer deals with dynamical optimizing control in the subsystems. The objective was to derive a general applicable set-point optimization method for refrigeration system that can drive the set-points towards optimal energy efficiency, while respecting the system limitations. Widell and Eikevik (2008) did a study on reducing power load in multi compressor refrigeration systems by limiting part load operation. An experimental analysis of compressor operation in a large refrigeration system was undertaken and a model for optimal compressor operation for energy efficiency was developed. The system used 5 screw compressors and ammonia as the refrigerant, with slide valves to regulate the compressors and match their refrigeration capacity with product freezing loads. Optimized operation was made both with and without a variable frequency drive. Reindle et al. (2003) have suggested load sharing strategies in multi compressor refrigeration systems. In that study they have showed that when two identical screw compressors are operating in parallel, there exists an optimum point at which it is best to switch from each compressor equally sharing the load to one compressor operating at full load and the other unloaded to match the remaining system load When two screw compressors of different sizes are operating, an optimal compressor control map can be developed which maximizes the efficiency of the entire system over the entire range of loads. These optimum operating maps are shown to depend on the characteristics of the individual compressor's unloading performance and the relative sizes of compressors. An optimum control strategy for systems having multiple compressors, screw and/or reciprocating can be implemented using the concept of crossover points introduced.

## 1.7 Objective of Thesis

The main objective of this thesis is to create novel control algorithms and operating procedures that can reduce the energy consumption of dynamic industrial refrigeration system. The approach here is to create a database of different operating parameters and utilize that database to create model of system energy consumption. Compared to the physical model, the data driven model will account for dynamic behavior of the refrigeration system. The data driven model will be used to predict the performance of compressors and condensers and to provide the energy efficient capacity control. Control algorithms developed in this study will also identify maintenance and operational problems up to certain extent in future and indicate the operator about those possible problems before hand. An algorithm is developed for changing the discharge pressure set-point (floating the discharge pressure) for reducing the energy consumption of the whole system.

The objectives that identify the scope of this project are deciding floating discharge pressure set point for energy optimization, compressor performance prediction and selecting most efficient compressor(s) in predicted conditions and compressor capacity control by real time model based energy optimization in multi-compressor system

## 1.8 Thesis Outline

The chapter 1 is an introduction to refrigeration system. Basic building blocks of the refrigeration system are discussed and principles of thermodynamics are presented that drives the refrigeration cycle. The vapor compression refrigeration cycle is also discussed which is the area of study of this work. The chapter ends with background and motivation. The chapter 2 provides a detailed description of the particular system that was selected for the study. It also discusses the analysis of gathered data and observations. The chapter 3 discusses Manske (1999) approach of finding discharge pressure with energy optimization.  The chapter 4 discusses a novel method of designing the sequencer, algorithm and simulation results.

# 2. The Refrigeration Plant

In this chapter we will describe the refrigeration plant examined for the study. The second part of the chapter will discuss analysis of the data obtained from the plant and describe the findings. These will form the bases for chapters 3 and 4.

## 2.1 System Overview

The refrigeration plant examined in this project is a food processing facility. The facility uses single stage refrigeration system for cold storage as well as for process cooling. The facility maintains 19°F suction temperature and the discharge temperature floats between 70°F to 90°F i.e. 115 psig to 165 psig discharge pressure. The facility has seven single stage screw compressors installed. All the compressors have liquid injection cooling system. Our analysis shows that the liquid injection cooling consumes 7% of cooling capacity of compressors. Out of seven six compressors have fixed volume index (fixed Vi) and one has variable volume index (Vi). The refrigeration system uses ammonia (R717) as refrigerant. On a pick summer day with full production activities the facility operates maximum of four compressors to satisfy 2000 to 2200 tonnage load. Loads on the system can be classified as, plant and office air conditioning load, process cooling load and cold storage load. The table 2 shows rated compressor capacity, motor voltage and Full Load Ampere (FLA) data for all seven compressors.

| Compressor Name | Type | Load Varying Mechanism | Make / Year | Cooling Capacity (Tonnage) | Motor HP | Motor Voltage and FLA |
|---|---|---|---|---|---|---|
| Compressor #1 | Screw | Slide Valve | FES / 1998 | 426 | 500 HP | 4160V / 60A |
| Compressor #2 | Screw | Slide Valve | FES / 1971 | 512 | 600 HP | 4160V / 74A |
| Compressor #3 | Screw | Slide Valve | FES / 1971 | 512 | 600 HP | 4160V / 74A |
| Compressor #4 | Screw | Slide Valve | FES / 1971 | 512 | 600 HP | 4160V / 74A |
| Compressor #5 | Screw | Slide Valve | FES / 1971 | 512 | 600 HP | 4160V / 74A |
| Compressor #6 | Screw | Slide Valve | FES / 1984 | 512 | 600 HP | 4160V / 77A |
| Compressor #7 | Screw | Slide Valve | FES / 2002 | 390 | 450 HP | 4160V / 55A |

Table 2 Capacity, motor voltage and Full Load Ampere (FLA) data for Compressors

Evaporators and heat exchangers in the plant are either top fed or pumped liquid overfeed. A greater quantity of liquid is pumped through the evaporator than the amount of refrigerant that is actually evaporated in pumped liquid overfeed evaporators. All pumped liquid overfeed systems require receiver vessels. Receiver vessels are large tanks that hold two phase refrigerant. They have four main purposes: Separate the liquid and vapor components of a two-phase flow, maintain a liquid level with suitable static head for the liquid pumps, store a reserve of refrigerant to smooth transient load fluctuations in the system, prevent liquid refrigerant surges in the system from damaging compressors (since gravity will separate the liquid from the vapor inside the vessel).

Three evaporative type condensers are placed for converting the refrigerant vapor into liquid phase. The sprayed water evaporates during the refrigerant phase transfer process. Each condenser has six 7.5 HP axial fans to provide forced convection. Two step speed control strategy is applied (50% speed and 100% speed) for condenser fans. Condensers are designed with nominal heat rejection capacity of 18,746.5 KBTU/hr (or also referred as MBH) at 95ºF dry bulb / 74ºF wet bulb outside air temperature. Two types of expansion valves are used, hand expansion type (HEX) and thermal expansion type (TXV). Process cooling is performed by shell and tube and plate and frame type heat exchangers. Due to safety issues many processes use glycol-water mix to transfer the heat. Evaporator coils in the coolers are defrosted with hot gas twice a day on a time scheduled basis. Evaporator defrost cycles are staggered so that there is usually no more than one evaporator in defrost at a time. Hot gas is superheated refrigerant vapor. It is piped directly from the discharge headers of the compressors.

## 2.2 Data Analysis and Conclusions

Discharge pressure reading, suction pressure reading, and current draw by each running compressor and slide valve readings are taken three times a day for daily round sheets. Also a software called RSEnergy Matrix is also installed in the plant to log the current draw data of each compressor automatically on a computer server. In this study the daily round sheet database as well as the computer database is used. One year long data of different operating parameters were gathered and analyzed. Figures 2.1 to 2.4 show second-order polynomial trend lines created from the data for each compressor. The whole discharge pressure range is divided in four sub ranges

An analysis of figures provides us following observations. In low discharge pressure range compressors #3 and #6 draw 15 to 20 ampere more current than compressors #2, #4 and #5 at 100% slide valve positions. Near 85% slide valve position, compressor #3 becomes more efficient than compressor #6. Compressor #4 is only efficient in high slide valve values. In the 130-134 psig pressure range also compressor #3 and #6 are less efficient than Compressor #2, #4 and #5 for 100% slide valve. Second order polynomial model for Compressor #1 is different in this pressure range compared to the previous pressure range. It draws less amount of current in less than 130 psig pressure range compared to 130 to 134 pressure ranges. Among all 600 HP compressors Compressor #5 becomes most efficient below 50% slide valve. In this pressure range the Compressor #4 is the most efficient compressor to run in all slide valve positions. There are many cross over points between compressors that makes one compressor more efficient than other.

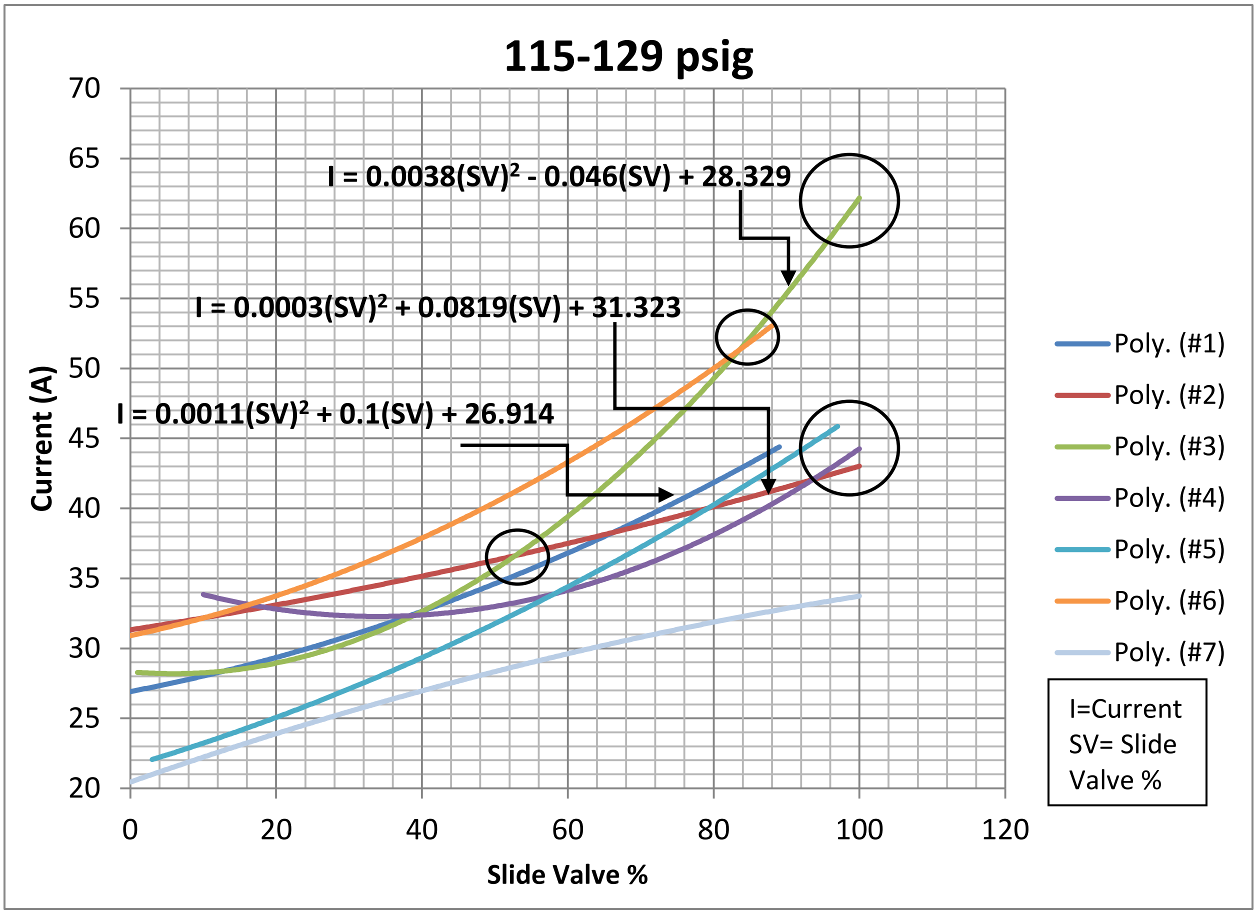


Figure 2.1 Current vs. % Slide valve graph for 100 to 129 psig discharge pressure

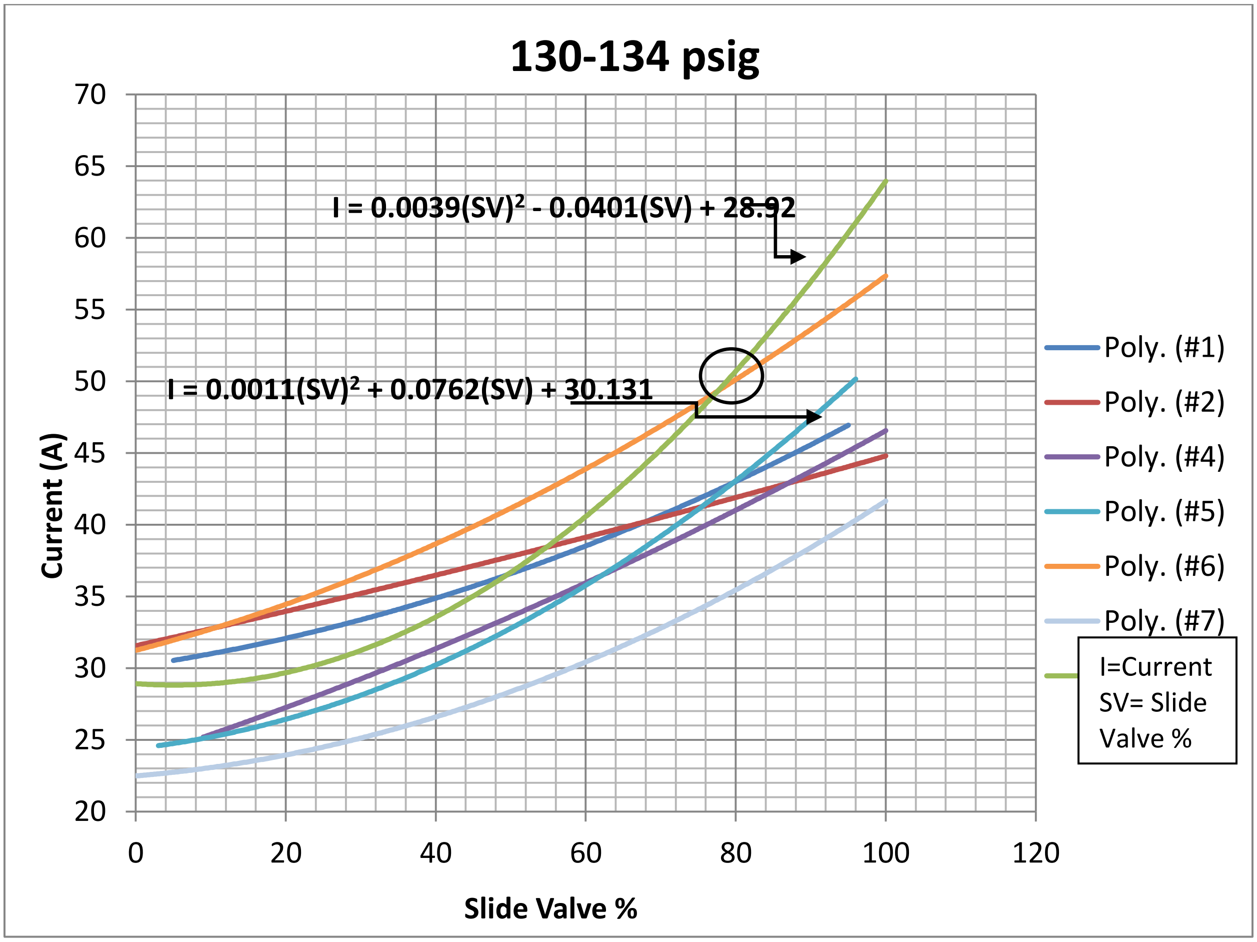


Figure 2.2 Current vs. % Slide valve graph for 130 to 134 psig discharge pressure

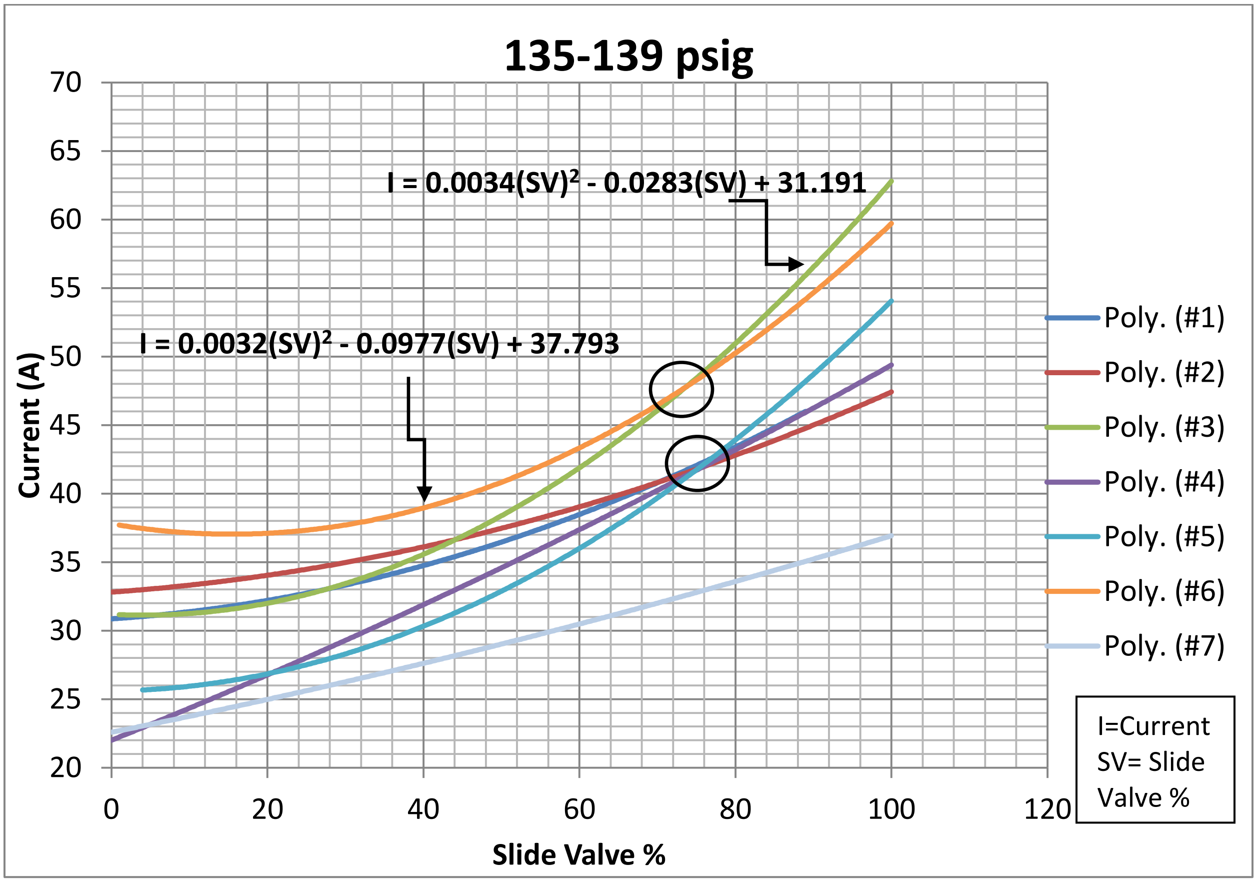


Figure 2.3 Current vs. % Slide valve graph for 135 to 139 psig discharge pressure

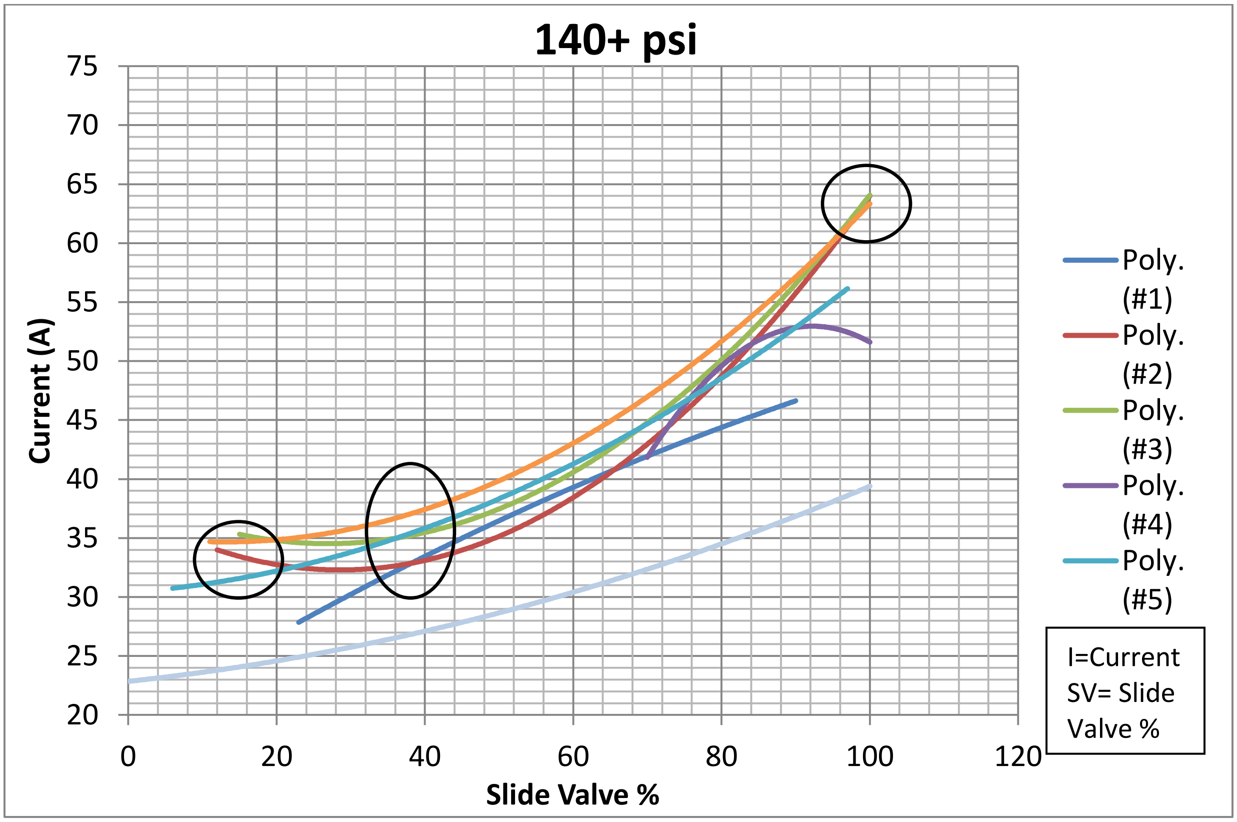


Figure 2.4 Current vs. % Slide valve graph for 140 and higher discharge pressure

In 135 to 139 psig pressure range the Compressor #4 is most efficient in all slide valve conditions. Also cross over points are different compared to previous plots. Cross-over points are circled in all plots. The figure 2.4 shows Current vs. % Slide valve graph for 140 and higher discharge pressure. In that pressure range compressors #2, #3 and #6 draws same amount of current in full load condition. But as these compressors start unloading, the Compressor #2 becomes more efficient than #3 and #6. At low slide valve values compressors #2, #3 and #6 have almost same current draw. Compressor #1 performs better in this range compared to other pressure ranges. In all figures compressor #7 draws least amount of power, as it is rated with lowest power. But its specific power (kW/Tonnage) indicates that it is efficient in only a few conditions (see chapter 3).

Based on the above analysis we concluded that a model based approach of selecting and sequencing compressors would save significant amount of energy in such high power systems and so a new design of sequencer algorithm is required to integrate the data driven modeling capabilities.

# 3. Evaporative Condenser Part Load Operation with Energy Optimization

In this chapter we will describe the evaporative condenser in more details and discuss advantages of floating discharge pressure. We will also discuss Manske's (1999) method of finding discharge pressure with energy optimization. The approach followed in this chapter was developed by Manske (1999). Here we are following the approach to simulate the data gathered from the plant in consideration. The results obtained and lessons learnt during the simulation, described in this chapter are very much useful in the next chapter where Manske's work is extended to predict the discharge pressure for the model based sequencer.

## 3.1 Evaporative Condenser Description

The task of an evaporative condenser is to reject the heat gained during evaporation action and gas compression action, from the refrigerant into the atmosphere. As energy is removed from the hot refrigerant a change in state from vapor to liquid occurs. Evaporative condensers reject energy from the high pressure, hot compressor discharge refrigerant to the ambient air. A diagram of an evaporative condenser is shown in Figure 3.1.

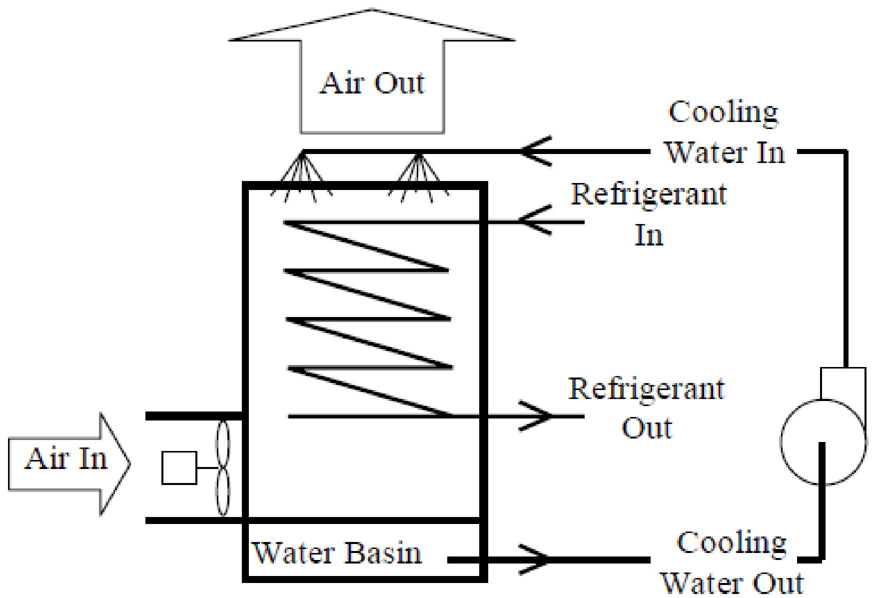


Figure 3.1 Evaporative condenser descriptions

Superheated high pressure refrigerant vapor coming out of the compressor, enters the coils of the evaporative condenser at the top of the unit. In order to have better heat transfer, water from a basin is pumped up to the top of the unit and sprayed down over the outside of the coils as outside air is drawn or blown through the unit by fans. As the water pours over the coils and evaporates into the air stream, the exterior heat exchanger surface tends to approach the outside air wet-bulb temperature. Also energy is transferred from the high temperature refrigerant to the cold water resulting in a phase change and condensing the refrigerant into a liquid (still at high pressure). Nearly saturated air leaves the top of the condenser at a temperature near the refrigerant's saturated condensing temperature (SCT). The SCT is the refrigerant's saturation temperature corresponding to the pressure inside the condenser. The refrigerant then leaves the condenser as a saturated or perhaps slightly sub cooled liquid. An evaporative condenser rejects energy by both heat and mass transfer on the outside surface of the condenser tubes. The main component of energy rejected by the condenser comes from evaporating the water, so an evaporative condenser is mainly a wet-bulb sensitive device (Manske, 1999). The heat transfer depends on the condensing pressure of the refrigerant. Higher the wet bulb temperature, higher refrigerant condensing pressure is required to maintain the temperature gradient.

## 3.2 Fixed vs. Floating Head Pressure Control

System condensing pressure, also referred to as head pressure, is typically controlled in one of two ways. Fixed head pressure control has the simplest control strategy; however this control strategy results in unnecessary compressor power due to the compressors operating with higher pressure lifts than required. The fixed level head pressure control strategy maintains the head pressure at a constant preset level regardless of system load and outside air conditions. This level could be adjusted several times a year in northern climates to improve performance during months of colder outdoor temperatures.

The second type of control strategy is termed “floating head pressure”. In this control strategy, the head pressure is allowed to “float” down to a minimum set value which is normally determined by system defrost pressure, expansion valve pressure drop requirements, or oil pressure requirements from oil injected screw compressor cooling. As the load on the system or the outdoor dry bulb (wet bulb in the case of evaporative condensers) temperature increases, the head pressure will rise, thereby, allowing the system to reject energy to the environment as needed to balance system heat rejection requirements. Although the condenser energy is reduced using fixed head pressure control due to less fan run-time, the compressor energy consumption increased significantly (Manske, 1999). The current system uses a floating head pressure control strategy with a minimum head pressure set point of 115 psig.

## 3.3 Manufacturer’s Ratings

Typically, manufacturers provide the nominal volumetric air flow, the nominal heat rejection capacity, and a variable load multiplier that is referred to as the heat rejection factor (HRF). The HRF is a function of the outside air wet bulb and the refrigerant SCT. The rated heat rejected by the evaporator is calculated by dividing the nominal heat rejection capacity by the HRF. (Equation (3.1))

$$Capacity = \frac{Nominal\ Capacity}{HRF\ (T_{wb}, SCT)} \qquad (3.1)$$

Evapco provided HRF's for wet bulb temperatures between 50°F and 86°F and for saturated condensing temperatures (SCT) between 70°F and 90°F. Dry bulb temperature was not specified and can be assumed to have negligible effects on evaporative condenser performance (Manske, 1999). Varying inlet conditions are adjusted by mixing outside air with some of the moisture laden exhaust air exiting out the top of the evaporator until the desired inlet air wet bulb temperature is reached.

## 3.4 Evaporative Condenser Part-Load Operation

Condenser rejects the system energy to the environment. Condensing pressure/temperature is controlled with the fans to transfer the required amount of energy to the environment. If the load on the system increases, the head pressure needs to increase. As the head pressure increases, the refrigerant condensing temperature increases which increases the condensers heat rejection capacity. If the condenser is operating in a mode where it is rejecting too much energy, the condensing pressure will decrease along with the condensing temperature of the refrigerant until a head pressure which balances the heat rejection needs of the system is reached. Allowing too low a condensation pressure can cause operational problems with other components in the system such as expansion valves, back pressure regulators, and hot gas defrost capacities. When the amount of energy that needs to be rejected from the system is less than the full load capacity of the evaporative condenser, which is most of the time, the capacity of the condenser must be reduced. An evaporative condenser's capacity can be reduced in two ways: Head pressure control by altering the airflow through the unit with fan speed control or fan cycling and dry operation by shutting the cooling water off.

The capacity of a condenser can be changed by modulating the mass flow of air through the unit by controlling the speed or cycling the fans. Equation (3.2) is used to de-rate the performance of the evaporator using fan speed control (Manske, 1999).

$$Capacity_{actual} = Capacity_{rated} * \left(\frac{Fan\ Speed_{actual}}{Fan\ Speed_{rated}}\right)^{N} \tag{3.2}$$

The coefficient N is expected to vary between 0.5 for laminar flow and 0.8 for turbulent flow (Mitchell and Braun, 1998). A manufacturer's representative from Evapco suggested a value of 0.76 for their evaporative condenser units (Manske, 1999).

As air mass flow through the condenser is increased by increased fan run time or faster fan speeds, more energy is rejected from the warmer refrigerant and the SCT of the refrigerant is reduced. With a lower SCT to drive heat and mass transfer mechanisms, the capacity of the condenser will be reduced. With this capacity reduction scheme, the system sees an additional benefit in the reduction of the head pressure on the compressors which cuts down on the power consumption of the compressors.

The other method used to reduce the capacity of an evaporative condenser is to simply shut the cooling water off. Without a wetted exterior surface, the only mechanism of energy rejection is by sensible heat transfer between the refrigerant and outside air. With reduced loads and relatively cool outside air dry bulb temperatures, as commonly found during winter months in cooler climates, this is a perfectly acceptable method of capacity control. This type of control also has the advantage of reduced power consumption because the water circulating pumps are no longer energized. For outside air temperatures below 32°F, a manufacturer's representative suggested an evaporative condenser running dry operates between 30 and 35 percent of its wet capacity (Manske, 1999). Thirty-five percent was used in the present model. Due to the drastic change in capacity between wetted and dry operation (100% to 35%), condenser water cycling should never be used for capacity control.

Typically a combination of both capacity reduction schemes mentioned above are used to maintain the desired condensing pressure in the system throughout the year. Mass flow of air through the condenser is controlled by the fan motors. Widely used control schemes are: On\Off motor cycling, (Current evaporator control strategy), half-speed motor cycling (High speed, low speed, off) and Variable Frequency Drive (VFD) controllers on the motors.

With the on-off motor control strategy, the condenser fans are run at full speed until the condenser's pressure falls below an acceptable limit and then the motors are shut off.

Half-speed control first cycles the fans to half-speed and then to full speed if the condenser's head pressure is still too high. VFD control runs the fans at a speed just fast enough to maintain a constant head pressure at a defined set point. The advantage of using the half-speed and VFD control can be explained by the fan laws. Fan power is related to the cube of fan speed. If the speed is cut in half, half the mass flow is achieved at only one-eighth of the design fan power. Depending on the size and arrangement of the condenser, there may be more than one motor driving any number of fans. Of course each individual motor can be sized differently. The evaporator modeled in this study has two motors. One 7.5 horsepower motor drives one fan and one 15 horsepower motor drives two fans. The condenser has an internal baffle that prevents internal recirculation of the air when only one of the motors is on. The internal baffle combined with two separate fan motors splits the large condenser into two smaller ones; one with 33.3 percent of the total capacity and the other with 66.7 percent. When one section is active the other will still reject approximately 10 percent of its nominal capacity due to natural convection effects. When this arrangement exists, there are several different control strategies to choose from. Each control strategy dictates a different order by which "parts" of the condenser are activated or deactivated to build up to its full capacity. When motors are purchased with a half-speed option, the number of possible control strategies increases further. Several control schemes were selected and compared. Figure 3.2 shows what percent of the full load power each fan control scheme would use given the percent of full load that the condenser must operate at to satisfy the energy rejection requirements of the system. The fan power drops to zero at ten percent capacity because some natural convection effects were assumed.

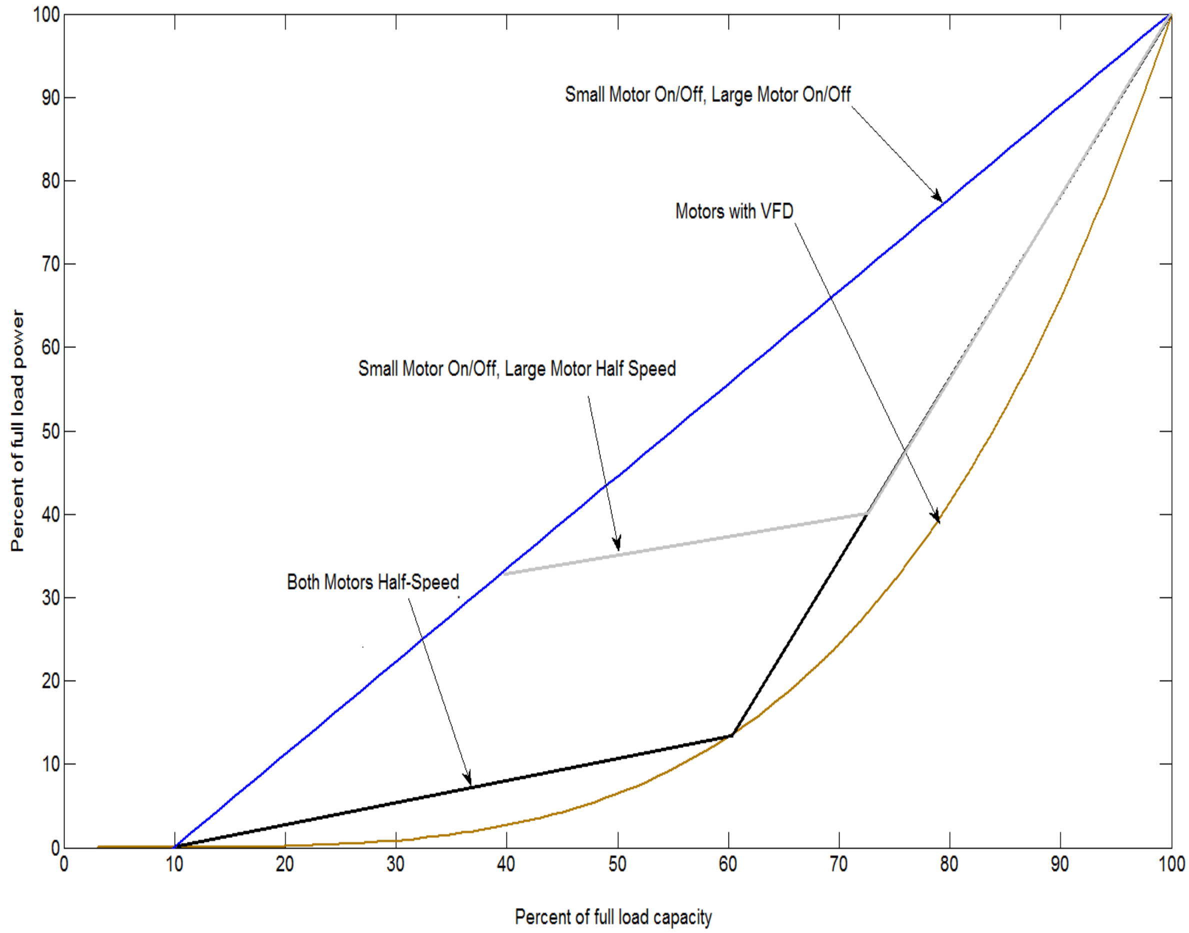


Figure 3.2 Part load evaporative condenser operation

Fan control for control strategy 1 would be as follows. Given a saturated condensing temperature and outdoor wet bulb temperature the nominal capacity of the condenser is 100 [MBH]. If the actual amount of heat that needed to be rejected was below 40 [MBH] (33.3 from the small fan side plus (0.1*66.6) = 6.7 from the natural convection of the large fan side), then only the small fan would have to be operated. For example if the load was 25 [MBH] the small fan would have to cycle on (25/40) = 62.5% of the time. The large fan would remain off. If the load was 50 [MBH] the large fan would cycle on (50/70) = 71.4% of the time.

Fan motors can be designed to operate at single speed, multi-speed, or be controlled by variable frequency drives (VFD). Advantages of using multi-speed and VFD motors appear when a system is operating at part-load. Condenser capacity control is accomplished by modulating or reducing the airflow through the unit with fan control. As shown in Figure 3.2, given a specific system operating point, VFD motor control requires significantly less condenser fan power than simple on/off control when the condenser is operated between 30 and 90 percent of its full load capacity. Figure 3.2 also showed that a half-speed motor option also realizes significant power savings in that same operating range.

## 3.5 Deciding the Head Pressure with Energy Optimization

As we discussed earlier, that the total system power depends on saturated condenser pressure. If the pressure is allowed to increase, the condenser fans have to run less often or at lower speeds and a savings in condenser fan energy results. Secondly, high head pressure requires increased amounts of compressor energy to produce the extra pressure lift. Figure 3.3 is a plot of the combined compressor and condenser energy requirements per tonnage of refrigeration (Specific Power) as a function of head pressure. The point furthest to the left on the curves in Figure 3.3 represents the pressure at which the condenser is operating at 100 percent capacity. Any further decrease in condensing pressure would prevent the condenser from rejecting the required amount of energy from the system. Simulations were performed to identify optimum head pressure for the system in discussion.

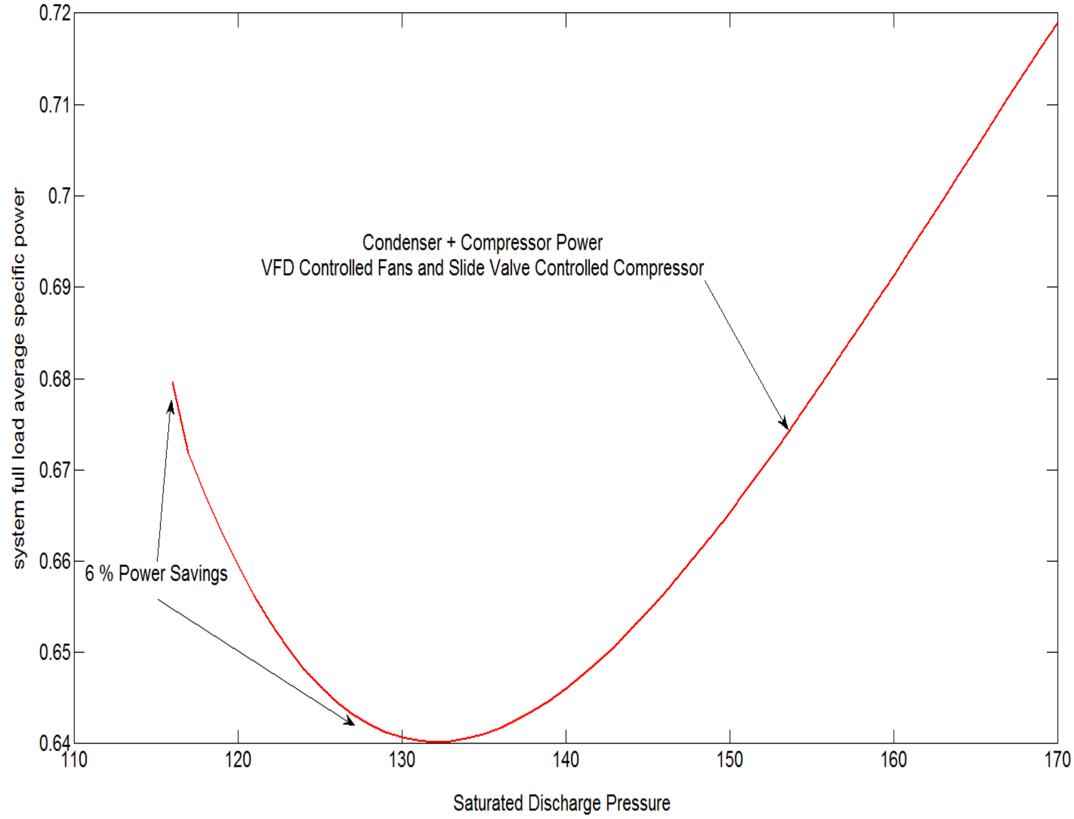


Figure 3.3 Evaporative Condenser Fan Motor Control Strategies

Figure 3.3 shows optimum system head pressure at ambient wet bulb temperature of 72°F. It also demonstrates that VFD fan control could save the system nearly 6% in combined compressor and condenser energy requirements if the head pressure were raised to 116 psig. The optimum pressure is dependent upon both the system load and size of condenser. VFD fan control loses its advantages at low head pressures because the fans must run at near full speed most of the time anyway. Simulations were created for all different wet bulb temperatures for an imaginary two compressor system. Figure below shows optimum saturated condenser pressure for 86°F outside wet bulb temperature.

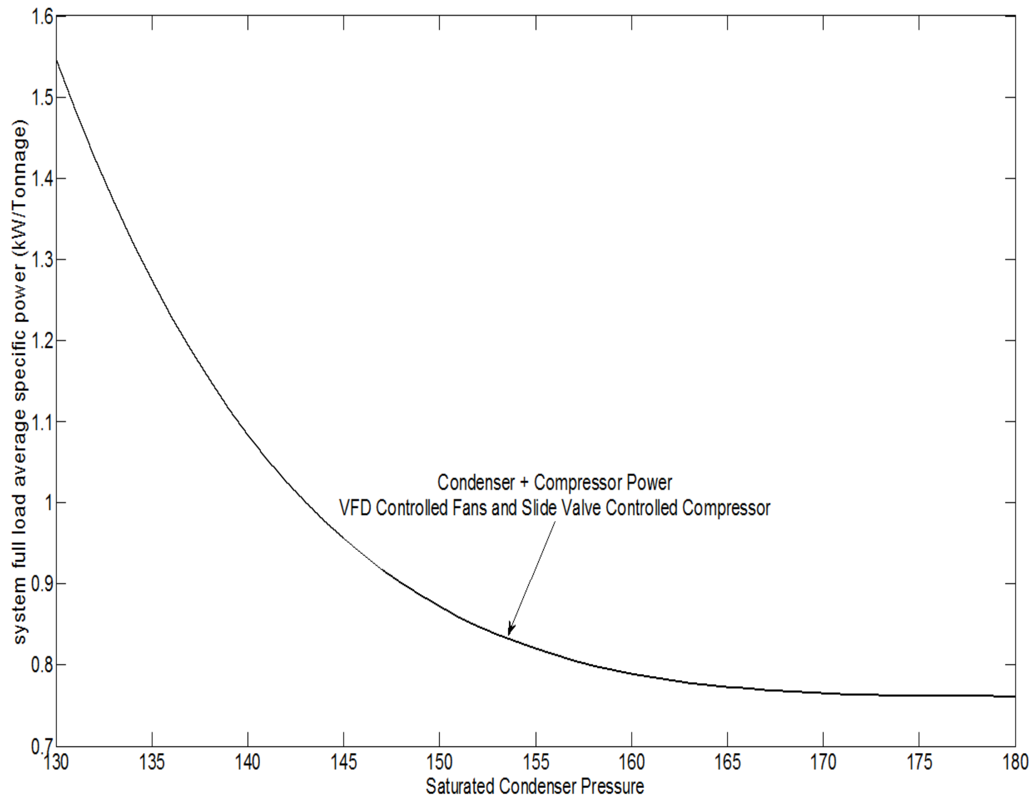


Figure 3.4 Saturated condenser pressures for 86°F outside wet bulb temperature

As the ambient wet bulb temperature changes, the point of optimum pressure changes, with higher wet bulb temperature the optimum head pressure increases. An algorithm was written (Chapter 4) to change system head pressure in real time while optimizing the system energy. The new method of designing sequencer described in next chapter includes this optimization algorithm to predict the discharge pressure for compressors.

# 4. Model based Sequencer for Energy Optimization

In this chapter first we will discuss present methods of designing sequencers for compressors than we will discuss the model based sequencer and its algorithm. Based on the plant described in chapter 2 sequencer software was created and data of different operating conditions were fed to obtain the results. These results are also discussed in this chapter. Finally we will conclude the study.

## 4.1 Present Methods of Sequencing Compressors

In most application areas refrigeration is a part of critical processes and so in majority of cases companies install redundant equipment as a backup. In some cases, over a period of time, product range changes, because of which extra compression capacity exists. Also the systems are designed based on maximum cooling load which in many cases exist during small period of time in a year. Due to all these reasons extra compression capacity exists in a refrigeration plant.

When there is an extra compression power the question arises as to which compressor to operate during a given time period. Refrigeration plants apply different methods of selecting compressor(s) to run for a given period of time. These methods vary from simple manual selection to semi-automated selection by user configurable hardware/software to fully automated selection by a PLC or Sequencer.

The load on refrigeration system is not constant; it varies based on ambient conditions and production schedules. Sequencers are also designed to vary the capacity of compressors to meet the varying load in real time.

### 4.1.1 Introduction to Sequencers

Dugan (2010) explained present day sequencers ar “Sequencers are control systems that sequentially stage multiple compressor systems, running only the minimum number required, based on one suction pressure signal, usually with only one running in a part-load mode (“trim”) and the rest either fully loaded (“base-load”) or off. Here we are describing three basic types of sequencers based on their algorithm, “cascade”, “target”, and “custom”. The first two are for “discrete” control only, using binary or relay interface, best suited for load-unload screw or reciprocating compressors. Custom sequencers can be applied to proportional control, which includes variable speed (VS) compressors”.

### 4.1.2 Cascade Sequencers

The simplest sequencers use a “cascade algorithm”. It is the sequential starting and loading of compressors based on rising pressure, and the reverse for falling pressure. This algorithm comes from a time before the age of the computer. Sequencers started their life as a mechanically-driven pressure switch selectors, using relays, cams and timers. They function as follows: as the pressure rises, the next compressor starts and loads, and then the next starts and loads if the pressure rises further. As the pressure drops, the reverse occurs. The last ON will load and unload once the number of compressors running stabilizes. The sequencer swaps the order around to even out wear. This was coded into simple programmed logic when programmable logic controllers (PLCs) and embedded controllers were introduced to industry. The cascade algorithm is best suited for positive displacement and reciprocating compressors. Cascade sequencers have a wide operating pressure differential (Dugan, 2010).

### 4.1.3 Target Sequencer

Dugan (2010) explained target sequencer as follows “With the advent of PLC and embedded controller technology, different algorithms have been designed. One common alternative to the cascade algorithm is the “target” algorithm. There are variants, but the simplest uses one pressure band for the trim compressor and a wider pressure band to trigger base-load compressors. The sequencer manages the number of base-load compressors running without having to wait for pressure to continue to rise again. The first time the pressure rises to the lower “base-load” point, the trim compressor is already fully loaded and the #1 base starts. The second time it hits the same base-load point, the #2 starts, and so on. The reverse happens at the high limit of the wider pressure band, but in reverse. Another way is to use timers instead of the wider pressure band to determine if the next compressor needs to start. Target sequencers have a narrow operating pressure differential. These common designs seem simple. In a “perfect” world, implementation would be simple also. However, simple sequencers described above assume the following system characteristics for smooth implementation: All of the compressors are the same vintage, make, type, and size, all can run load-unload and be remotely started and stopped, all are plumbed to a common header, generously sized and there is adequate storage”.

### 4.1.4 Custom Sequencers

We will comment briefly on two algorithms: Flow-based and Load-sharing.

In "Flow-based” sequencer the optimal number and size of base-load compressors are run at any time based on total flow, not strict sequential order. A more advanced processor and algorithm is required. Significantly different size or type of base-load compressors make this algorithm a good fit.

In “Load-sharing” multiple proportional compressors are run at the same pressure and percent load. The management system “bumps” the local settings to make this happen. This expands the effective range of efficient trim operation; making a system more stable (Dugan, 2010).

### 4.1.5 Review

These methods try to equalize the running hours, stabilize the pressure set points and sometimes try to minimize the energy consumption. But in performing these tasks they do not account compressors’ relative energy efficiency. Considering that, compressors rated with same specifications draw same amount of power, is incorrect. Due to many practical reasons like wearing of parts, age of the compressors, type of compressors, compressors’ motor efficiencies, effectiveness of the cooling system, fixed volume ratio vs. variable volume ratio, discharge pressure range, method of capacity control, compressor shaft alignment etc. energy consumed by each compressor is different than other compressor even if they are rated same. In this section we will discuss about integrating the computer database of compressor(s) energy consumption with compressor sequencers and utilizing the database to predict the operating discharge pressure for each compressor. We will also discuss the computer program created for a refrigeration plant and energy saving analysis for that plant.

## 4.2 Model based Sequencer

As we discussed earlier, the refrigeration cycle is highly dynamic and changing. From the time the compressor commences to run until it stops, the suction pressure, evaporator temperature, the rate of heat exchange, refrigerant flow and many other factors are continuously changing. The total efficiency of the system changes through the entire life cycle. In order to account for the dynamic behavior of the refrigeration system and provide energy efficient controls it is necessary to keep track of the system's behavior by gathering operational data and model the compressors' energy consumption based on the created database.

We have developed an adaptive control algorithm that creates a database of important system parameters and uses that to predict performance of each compressor. Here the ratio of compressor's energy consumption to cooling capacity is calculated for an 8 hour time period and based on that the sequencer selects compressors to run for a given period of time. In order to maintain the stable suction pressure, the sequencer creates energy models of each compressor in real time and feeds the cooling capacity requirement data in the model to optimize the energy. Slide valves of different running compressors are varied in order to meet the load requirements while minimizing the energy. The algorithm keeps updating the database and works accordingly. The program uses the database of the refrigeration plant discussed in chapter 2. The sequencer program was simulated for all different discharge pressure conditions and different operating conditions for that plant.

### 4.2.1 Correlations and Compressor Models

In most industrial refrigeration applications, compressors consume the majority of the system total energy requirements. The three parameters that are of most interest to a refrigeration system designer or operator are the power required by the compressor, the amount of useful cooling (capacity) it provides, and the amount of oil cooling it needs. Most compressor manufacturers provide tables for each of their compressor models that list these three requirements (brake horsepower, capacity [tons], and oil cooling load [kBTU/Hr]) given saturated suction and saturated discharge temperature/pressure. Saturation temperature is defined as the temperature corresponding to the saturated vapor state of the refrigerant given a particular pressure (Manske, 1999). But as we discussed in topics 2.2 and 4.1.5 that compressor's actual power consumption changes based on different reasons including total running hours put on them, the present system uses data base of past ampere, slide valve and saturated pressure data to create compressor models.

In order to select compressors for full load and part load operations, considering different size of compressors, it is necessary to calculate compressors' cooling capacity with respect to the electrical power input. The ratio of electrical power input to the cooling capacity is called Specific Power (Manske et al). Its unit is kW/Tonnage. Polynomial correlations of steady state compressor power, capacity and Specific Power in the form given by equation 4.1-4.4 were developed as a function of slide valve movement. Power and capacity are also function of saturated suction pressure and saturated discharge pressure. Here saturated suction pressure is taken constant as the facility has a constant suction pressure set point. Equation 5 and 6 were developed to calculate maximum compressor power and capacity for different saturated discharge pressures. The capacity data were provided by manufacturer. Equations 4.1 to 4.6 are totally empirical.

For a discharge pressure, power consumed by each compressor at various slide valve positions can be described as

$$POW_{electrical} = P1 * (\%\, SV)^2 + P2 * (\%SV) + P3 \tag{4.1}$$

For a discharge pressure, cooling capacity of each compressor at various slide valve positions can be described as

$$CAP_{thermal} = C1 * (\%SV)^2 + C2 * (\%SV) + C3 \tag{4.2}$$

For a discharge pressure Specific Power of each compressor can be described as

$$SP_i = \frac{POW_{(electrical)i}}{CAP_{(thermal)i}} \tag{4.3}$$

For a discharge pressure Specific Power of each compressor at various slide valve positions can be described as

$$SP = S1 * (\%SV)^2 + S2 * (\%SV) + S3 \tag{4.4}$$

The equation below decides maximum power consumed when the compressor is 100% loaded at each saturated discharge pressure values.

$$POW_{full\,load} = DP1 * (SDP)^2 + DP2 * (SDP) + DP3 \tag{4.5}$$

The equation below decides maximum capacity consumed when the compressor is 100% loaded at each saturated discharge pressure values.

$$CAP_{thermal} = D1 * (SDP)^2 + D2 * (SDP) + D3 \qquad (4.6)$$

Where, P1-P3, S1-S3, DP1-DP3 are empirical coefficients and calculated every time. C1-C3 and D1-D3 was calculated based on data provided by manufacturer.

### 4.2.2 Compressor performance prediction

Compressor models were used to find optimum power consuming compressor(s) for full load and low load operations based on predicted discharge pressures for the day. The equations below are used to select full load and part load compressors for sequencer.

$$Full\ Load\ Compressor = \min((\frac{1}{time}\sum_{t=1}^{time}\frac{1}{20}\sum_{SV=80}^{100}SP(SV))_{All\ Compressors} \qquad (4.7)$$

$$Part\ Load\ Compressor = \min((\frac{1}{time}\sum_{t=1}^{time}\sum_{SV=10}^{80}SP(SV))_{All\ Compressors} \qquad (4.8)$$

Note here the part load compressor is selected based on total specific power (adding specific powers at all slide valves (10% to 80%)).

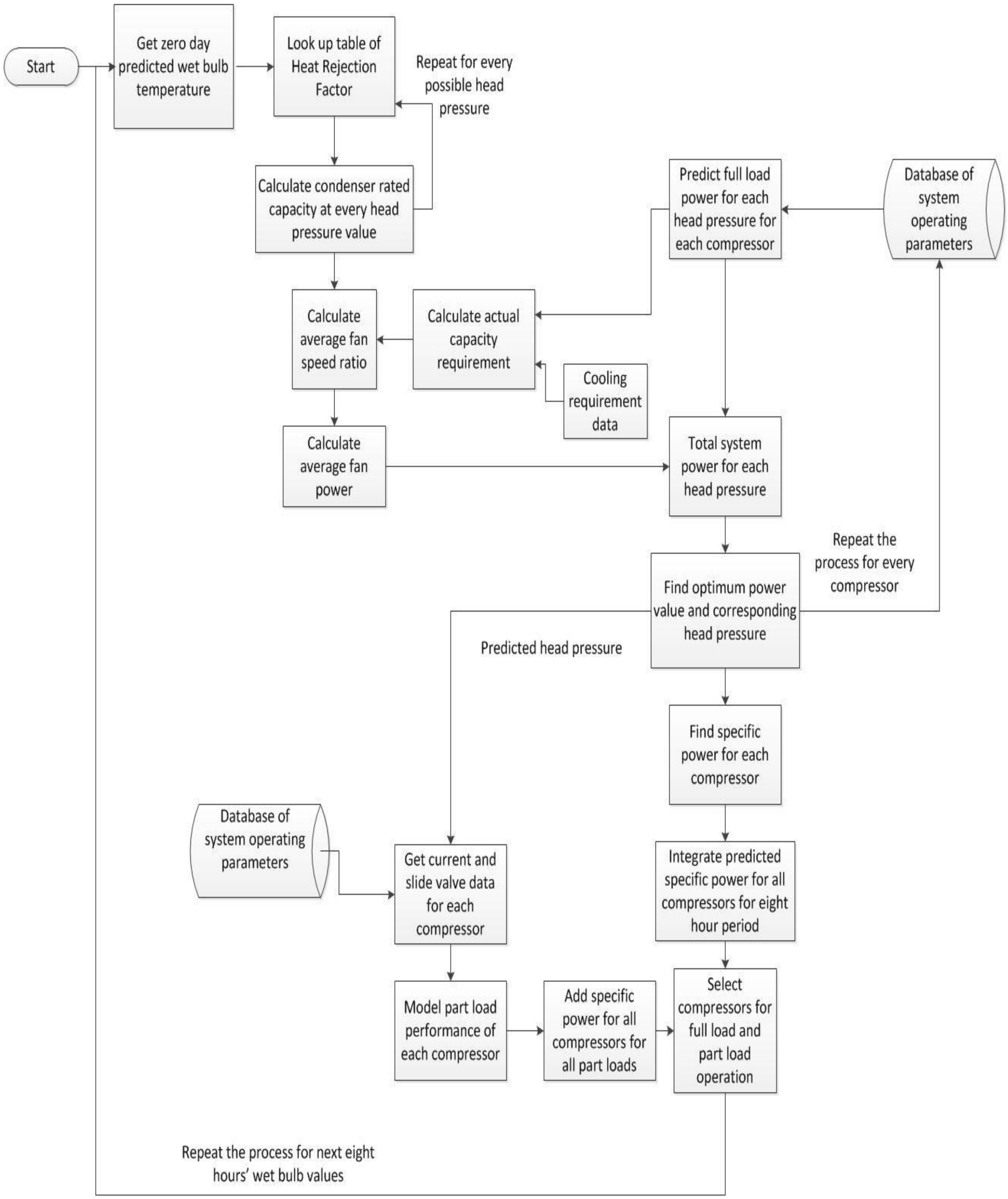


Figure 4.1 Algorithm of Compressor Selection Program

Every eight hour the program predicts discharge pressure based on the ambient conditions and for those discharge pressures the average full load specific powers and total part load specific powers are predicted. These specific power values are fed into the optimization program. The optimization program integrates the average full load specific power and total part load specific power over the time period to find the least power consuming compressor combination from the available options.

Figures 4.2 and 4.3 show predicted zero day (for an 8 hour period) full load and part load specific powers for all compressors. In this simulation dry bulb temperatures are taken between 60°F to 70°F and the relative humidities are taken between 60% to 65%. In these atmospheric conditions the predicted discharge pressures are between 133 psig to 142 psig. Integration of specific powers for the time period gives us predicted total specific power of each compressor. This prediction is used to select the least power consuming compressor. Accoring to figure 4.2, compressors #2 and #4 are least power consuming compresors at full load operation. According to figure 4.3, the compressor #5 will consume least amount of power at part load.

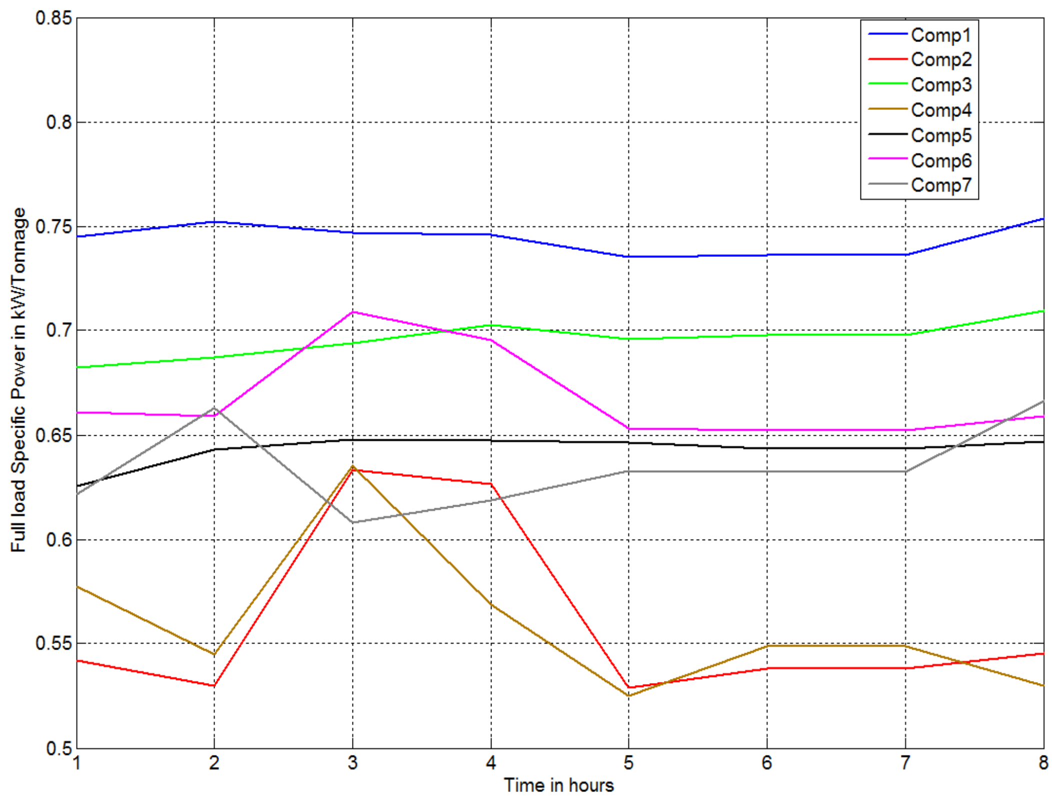


Figure 4.2 Average full load specific power for an 8 hour predicted period

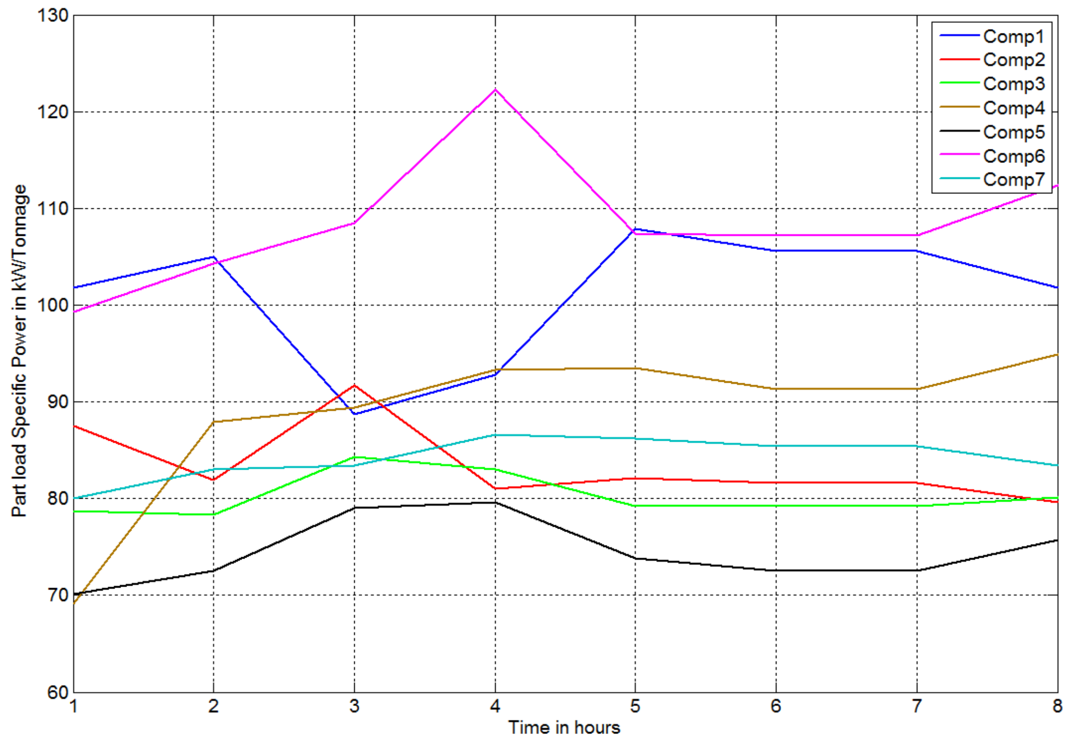


Figure 4.3 Total part load specific power for an 8 hour predicted period

### 4.2.3 Simulation Results

For each discharge pressure between 115 psig to 165 psig the program was simulated and Specific Power curves were obtained. The results show that compressors' efficiency varies based on discharge pressure and % slide valve values. The Figure 4.2 shows electrical power consumed by each compressor at 115 psig discharge pressure, for one tonnage cooling capacity at different slide valve positions. Compressor #1 is poor performing in all slide valve conditions, while Compressor #2 is best above 50% slide valve. Compressor #3 draws 30% more power at 100% slide valve than Compressor #4, but it draws 10 to 20% less power below 50% slide valve than Compressor #4. The Figure 4.4 shows Specific Power curves for higher slide valve positions at 115 psig discharge pressure. Here the cross over points between different compressors is clearly visible. In this pressure range (100-129 psig) Compressor # 3 draws 61.7A current at full load (100% slide valve) compared to 42.5A drawn by Compressor #2. The Figure 4.5 and 3.6 shows electrical power consumed by each compressor at 135 psig discharge pressure, for one tonnage cooling capacity at different slide valve positions.

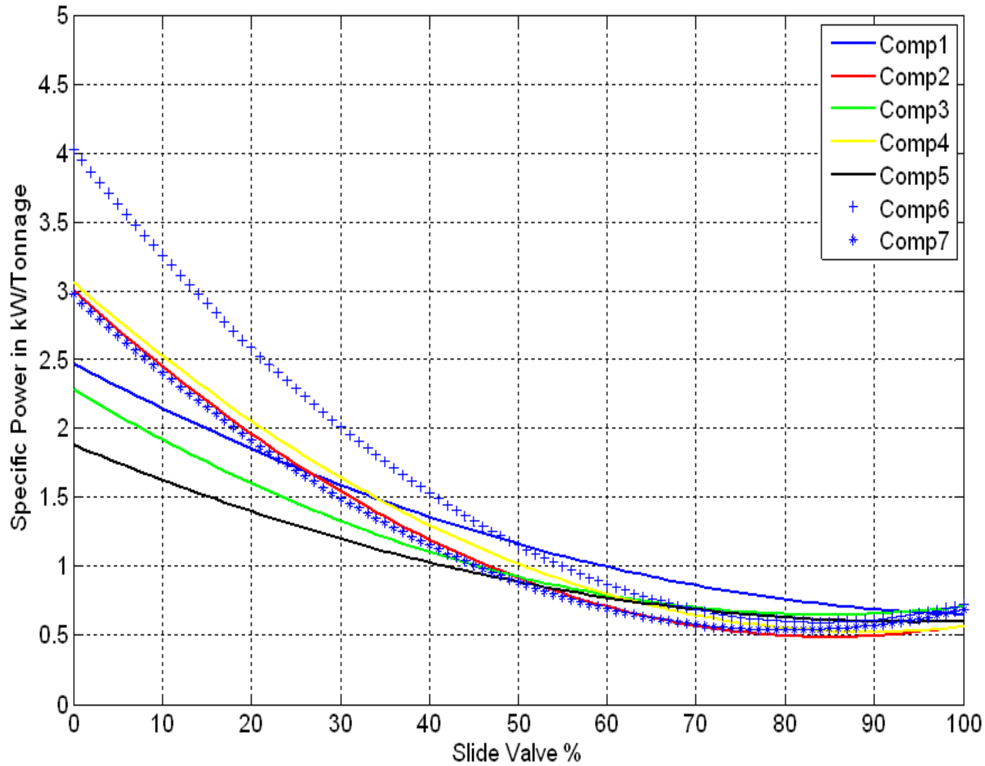


Figure 4.4 Specific Power vs. Slide Valve plot for 115 psig pressure

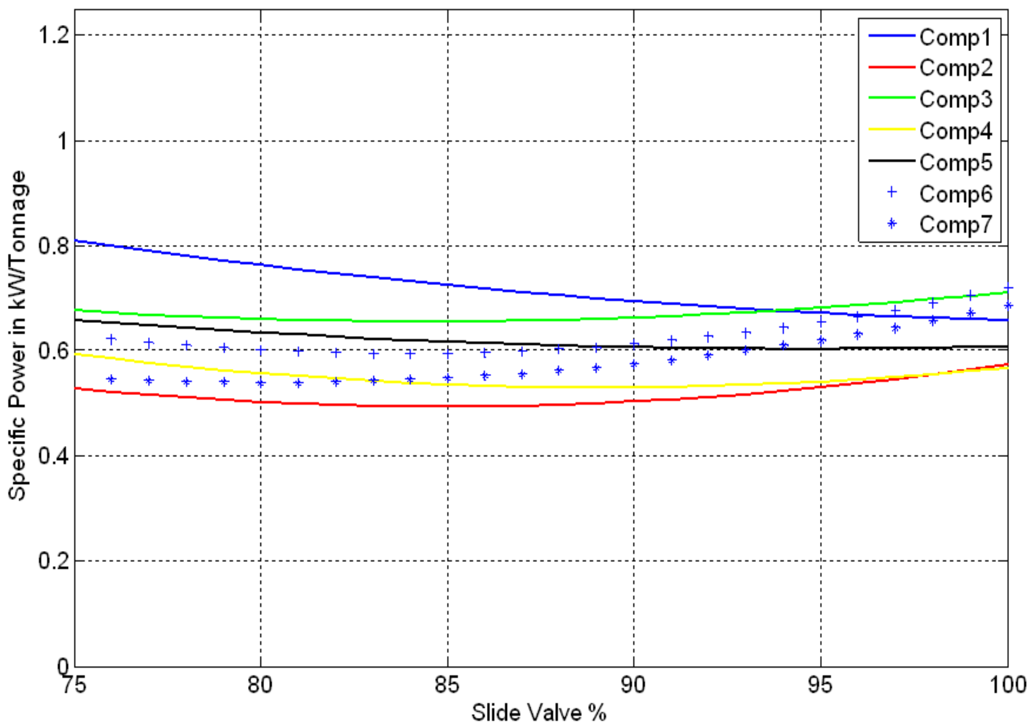


Figure 4.5 Specific Power vs. Slide Valve plot for 115 psig pressure

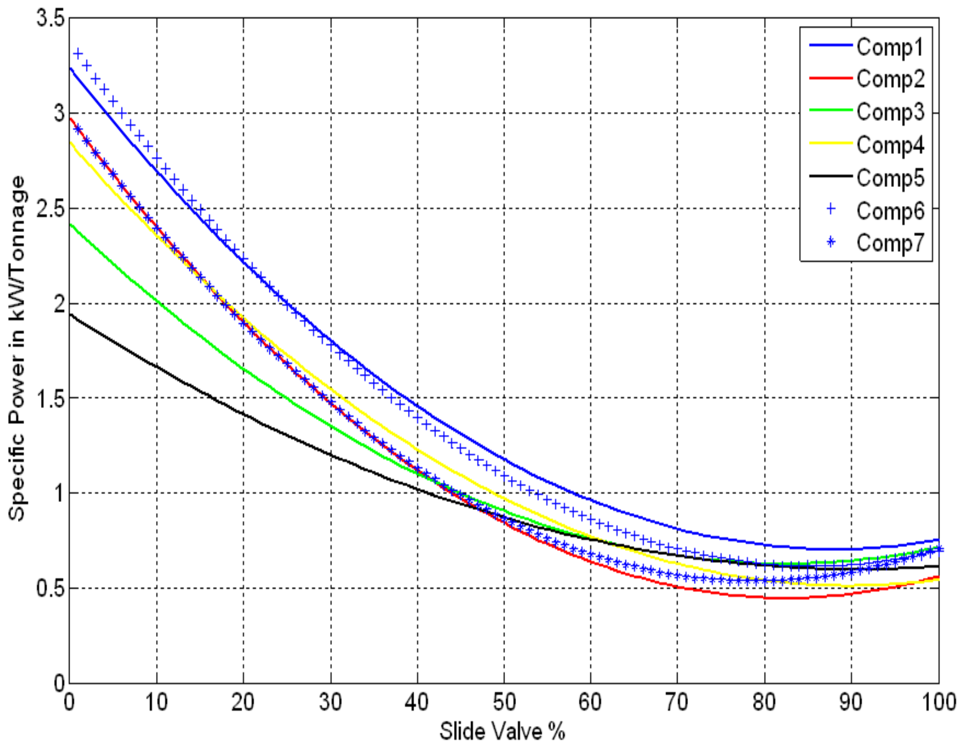


Figure 4.6 Specific Power vs. Slide Valve plot for 135 psig pressure

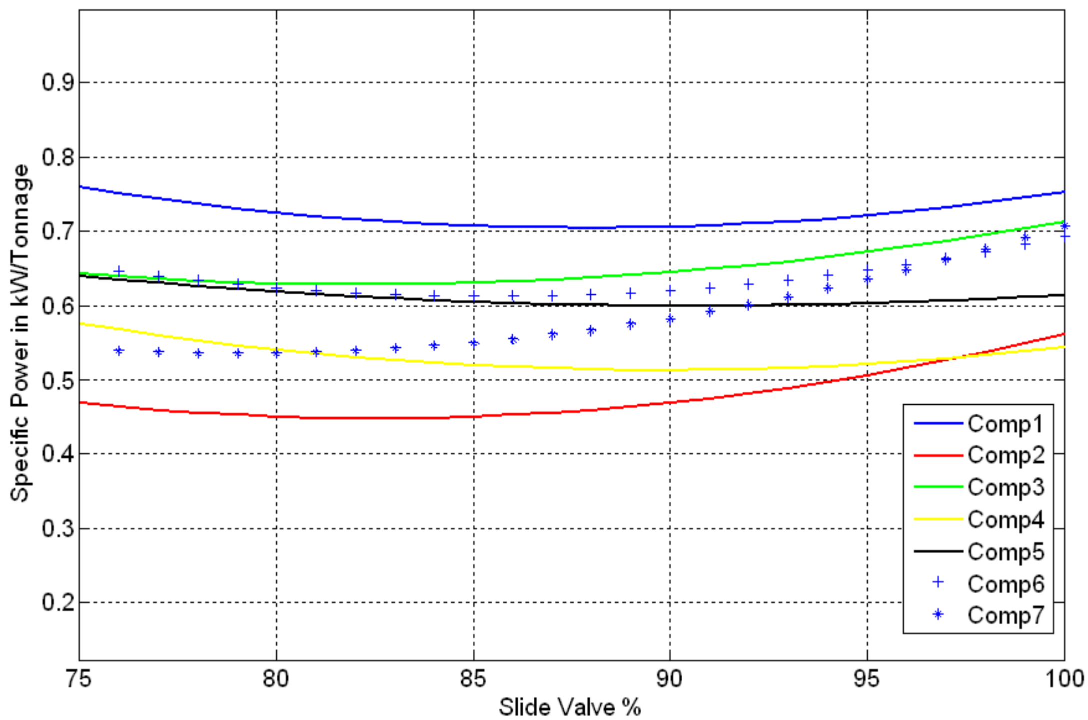


Figure 4.7 Specific Power vs. Slide Valve plot for 135 psig pressure

Compressor #2 is efficient for more than 60% slide valve, while Compressor #5 is efficient for less than 50% slide valve. Compressor #6 draws 30% to 45% more power to remove one tonnage of heat than Compressor #5 below 50% slide valve at this discharge pressure value. Compressor #7 is inefficient in all slide valve values. While in 140 psig pressure Compressor #7 is most efficient between 50% to 80% slide valve. The simulation shows that Compressor #6 and #1 draw more power per tonnage of refrigeration in all discharge pressure conditions and for all slide valve values. The poor efficiency of part-load operation is mainly related to two factors, friction and volume ratio change. When the slide valve is in use, a slot opens for the refrigerant to vent back to the suction side. This leads to friction in the gas and a change in the volume ratio of compressor (Stoecker, 1998).

### 4.2.4 Real Time Capacity Control in Multi Compressor System

If multiple compressors are used to meet refrigeration loads, it is desirable to operate the compressors at the lowest combined power while still meeting the system loads. In refrigeration systems with variable loads, the delivered capacity of the compressors must be modulated by unloading the compressors in order to balance the compressor(s) capacity with the refrigeration demands of the system. As we concluded in section 3.2.2, each compressor depending upon type and manufacturer, may have a different unloading characteristic. The following results are strictly valid only for the particular screw compressors investigated in this study; however, the general concepts can be applied to all refrigeration systems with multiple compressors.

Manske et al have investigated a performance comparison, in terms of specific power, between the screw and reciprocating compressors for several different saturated suction temperatures over a range of part load conditions assuming a fixed saturated discharge temperature of 85 °F (Figure 4.8). The performance maps include effects of refrigerant pressure drop in both the suction-side and discharge-side of the compressor. The figure shows that reciprocating compressors unload nearly linearly and their performance curve is nearly flat for a fixed suction temperature. The slight increasing trend (corresponding to a decrease in compressor performance) from left to right in Figure 4.6 for the reciprocating compressor is a result of increasing pressure drop in the dry suction line due to increasing refrigerant mass flow rate. The additional (3%) increase in total compressor power discussed above also contributes to the increasing trend in specific power. Several observations can be made about compressor operation from Figure 4.8. A single screw compressor unloaded to 25 percent of its full load capacity has nearly a 50 percent increase in specific power when compared to a reciprocating compressor. Screw compressors perform better than reciprocating compressors when operated near full load; the screw compressor's full load performance advantage increases as the suction pressure drops (i.e. as compression ratio increases). Reciprocating compressors are better suited in refrigeration systems where significant unloading, i.e. load following, is required. From energy standpoint, is more important to size screw compressors correctly as compared to multi cylinder reciprocating compressors.

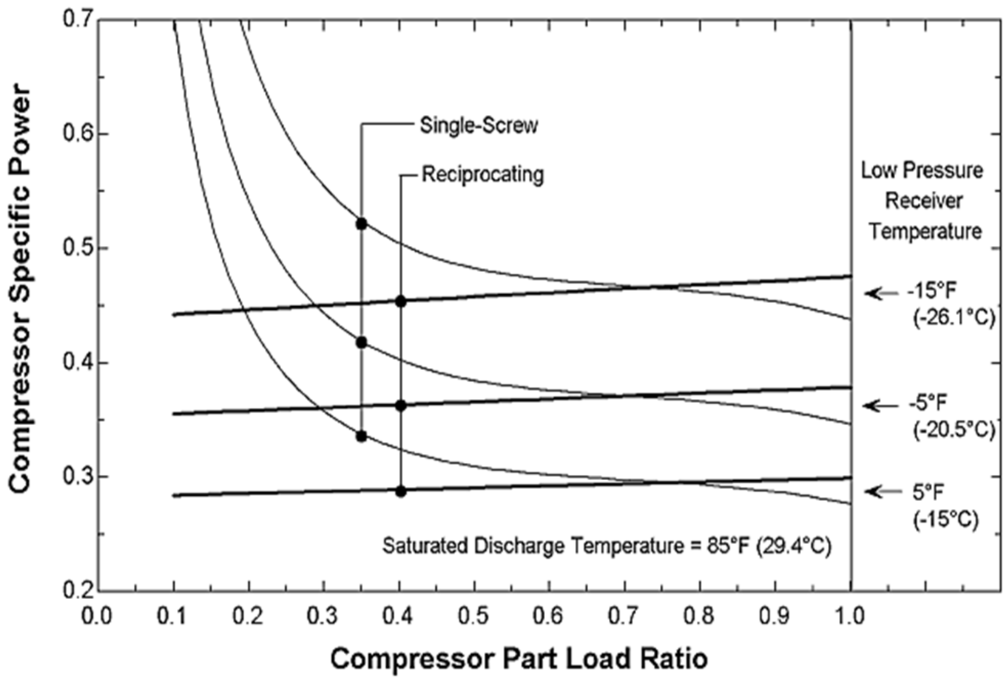


Figure 4.8 Comparison of the performance of single stage screw and reciprocating compressors including suction and discharge-side refrigerant pressure drop (Manske, 1999)

### 4.2.5 Load Sharing with Reciprocating Compressors

If two similar reciprocating compressors in parallel operation are sharing a load, the load should be split to equalize suction line pressure drop to each compressor. This operating strategy minimizes the dominant compressor performance penalty source – suction line pressure drop. This conclusion, evident from the results of simulations performed by Manske (1999), is based on several factors. First, the unloading characteristic of the reciprocating compressor exhibits minimal performance degradation when unloaded. Second, the pressure loss in the suction line is roughly proportional to the square of the refrigerant mass flow due to line frictional losses whereas the power per unit mass increases as the suction pressure is reduced in the manner shown in Equation 3.7.

The work per unit mass required compressing refrigerant from the suction conditions (state 1) to the discharge conditions (state 2) in a polytrophic process (i.e., *Pvn* = constant) is given by Equation 3.7 (Kuehn et al., 1998).

$$W_{12} = \frac{n}{(n-1)} * P_1 v_1 [\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1] \qquad (3.7)$$

Where,

$W_{12}$ is the compressor work per unit mass

$n$ is the polytrophic coefficient

$P_1$ and $P_2$ are the suction and discharge pressures (corresponding to the saturated suction temperature and saturated discharge temperatures, respectively)

$v_1$ is suction specific volume of the suction gas (which is affected by superheat)

Since the total mass flow rate for both compressors is fixed in order to provide the required refrigeration capacity, splitting the load to equalize pressure drop yields optimum combined compressor performance. For the reciprocating compressor, the combination of lower suction line pressure drop and minimal compressor unloading penalty lead to the improved compressor specific power as shown in Figure 4.8, (Manske et al.).

### 4.2.6 Load Sharing with Screw Compressors

Screw compressors unload non-linearly and their parallel operation must be treated quite differently compared to reciprocating compressors. We have developed an algorithm that creates model of specific power vs. slide valve for each running compressor in real time and performs optimization to minimize the total power required per tonnage of cooling. The model is created by gathering past performance data for each compressor for particular operating discharge pressure (Figure 4.9).

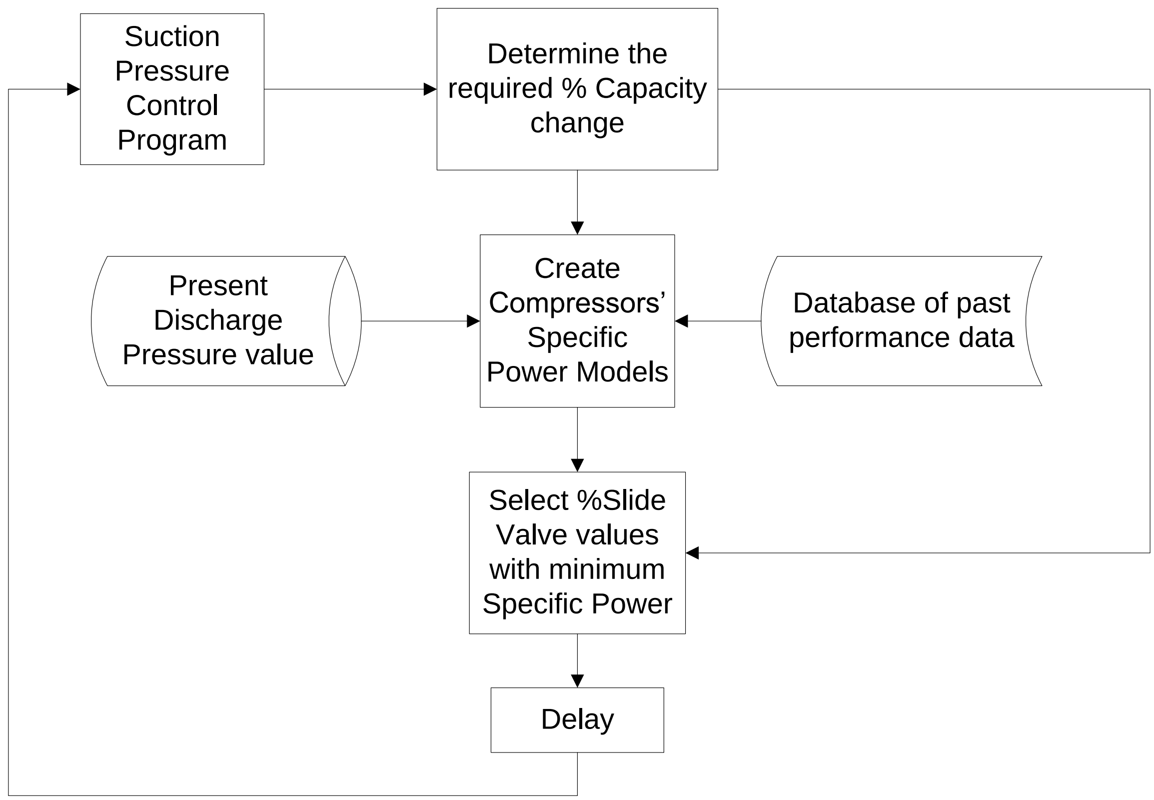


Figure 4.9 Algorithm of real time compressors' capacity control with energy optimization

## 4.3 Simulation Results

For the simulation a hypothetical two compressor system was chosen and Compressors #2 and #5 were selected to operate for some size compressor case. Compressors #2 and #7 were chosen in different size compressor case. Figure 4.8 to 4.10 show plots of the aggregate specific power for a system with two equally sized screw compressors operating in parallel. The abscissa is the compressor part load ratio which is a ratio of present compressor capacity to total available compressor capacity. The red curve shows aggregate specific power of Compressor #2 and #5, by keeping the Compressor #2 fully loaded and only unloading the Compressor #5. The system will consume more power if Compressor #2 is unloaded and Compressor #5 is kept fully loaded, refer figures 4.4 and 4.5.

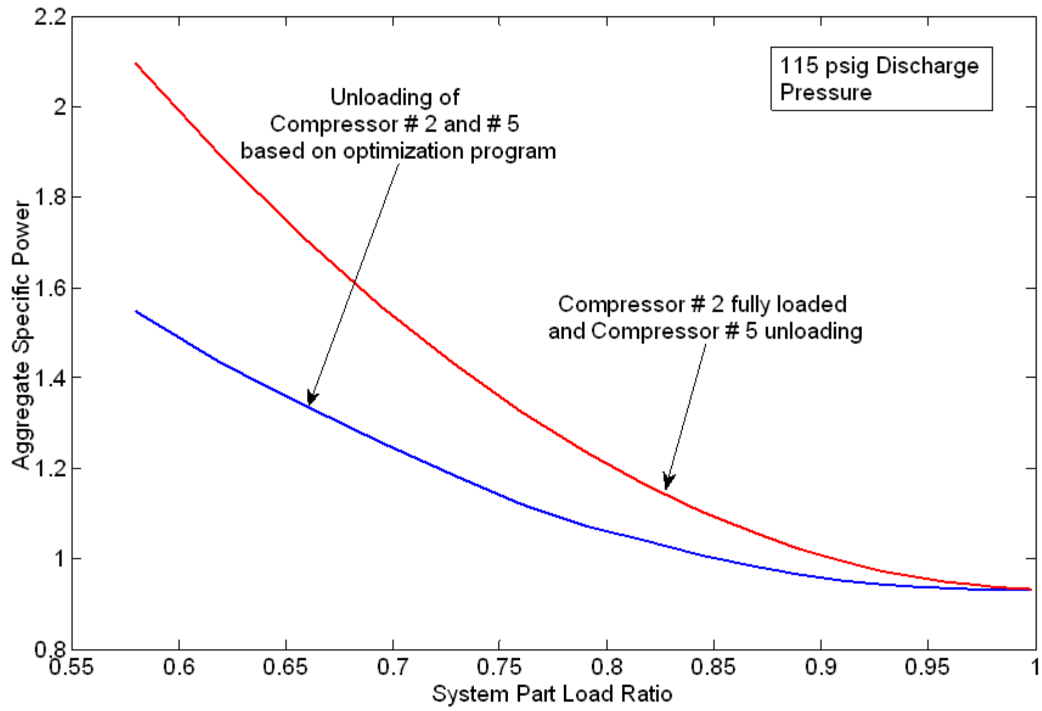


Figure 4.10 Aggregate specific powers vs. system part load ratio for constant discharge pressure

The blue curve represents aggregate specific powers of Compressors #2 and #5, by using model based optimization program. The program cycles compressors based on their different performance areas as discussed in section 3.2.2. In this case, for more than 90% system part load, aggregate specific power is almost same in both the control schemes, but as system part load decreases, single compressor unloading scheme becomes inefficient. The figure below shows slide valve movements of both compressors, decided by the optimization program. Note that slide valves of both the compressors are changed to meet the decreasing capacity requirement while minimizing the specific power. The figure 4.12 shows the comparison curves of two different schemes when the discharge pressure is not constant. The figure 4.13 shows aggregate specific power vs. system part load ratio curve for dissimilar sized compressors. The red curve shows aggregate specific power when the larger compressor is kept fully loaded and the smaller compressor is unloaded based on load requirements. The blue curve shows aggregate specific power when both the compressors are unloaded according to model based optimization program.

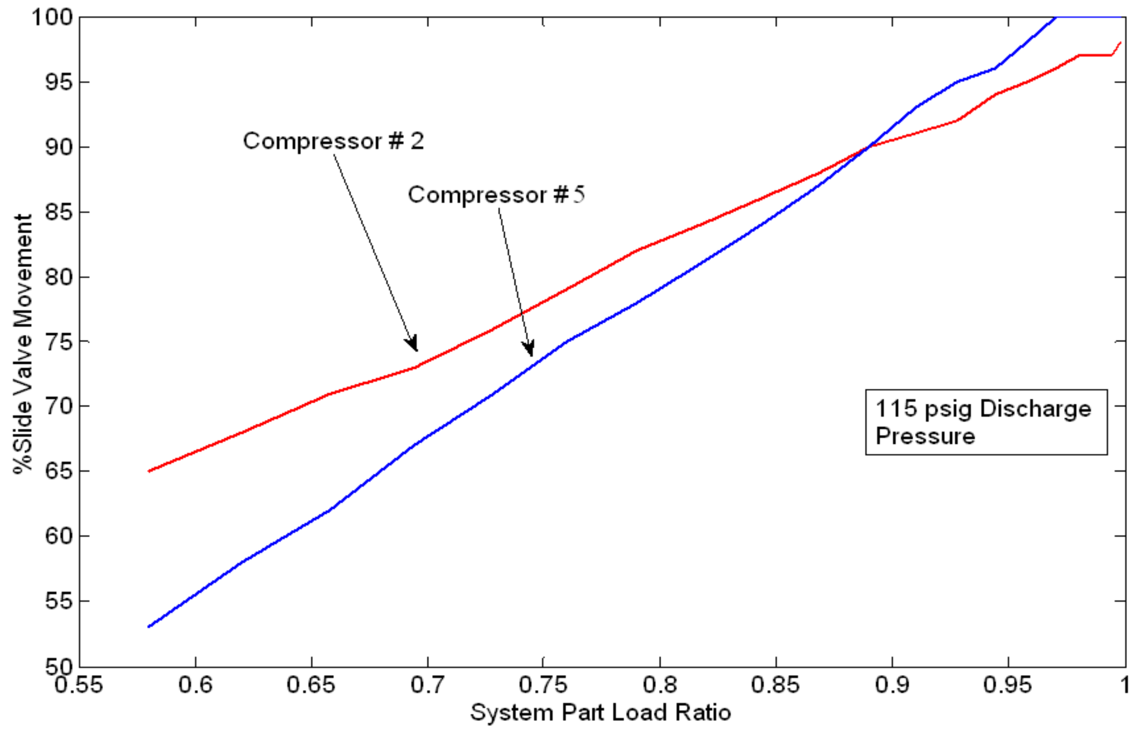


Figure 4.11 Unloading curves of compressors #2 and #5 for different system loads

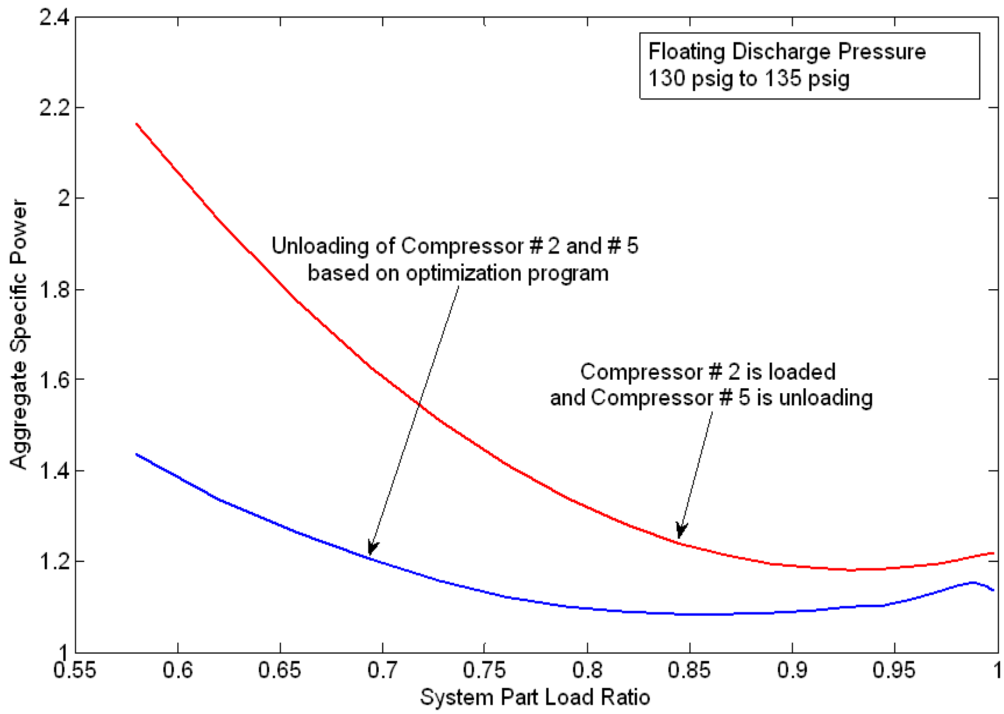


Figure 4.12 Aggregate specific powers vs. system part load ratio for floating discharge pressure

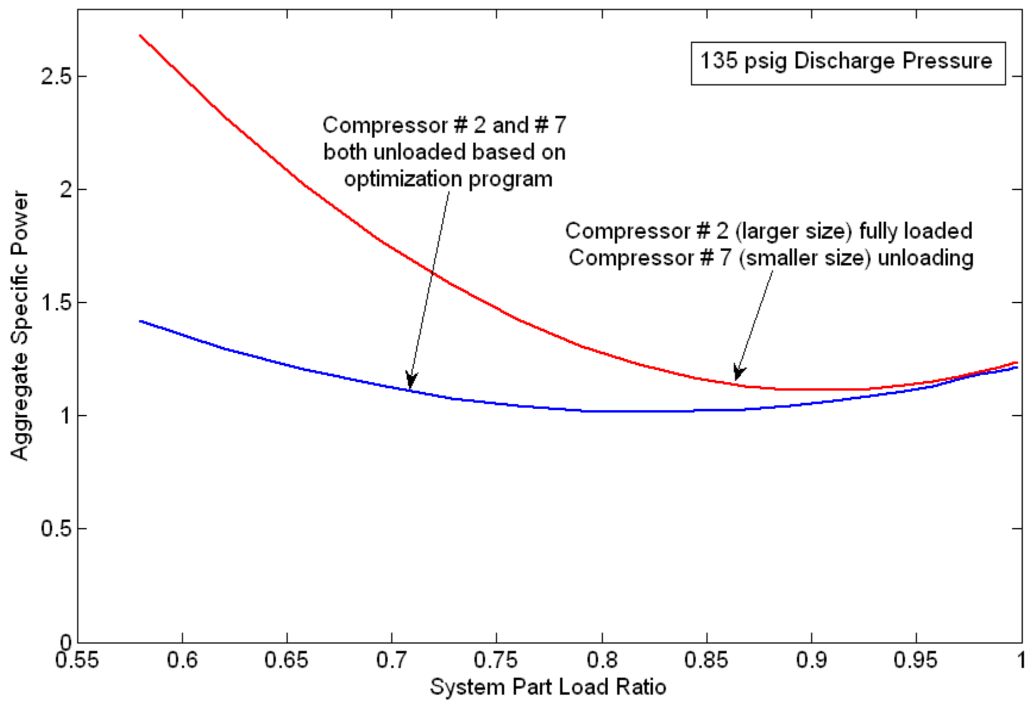


Figure 4.13 Aggregate specific powers vs. system part load ratio for constant discharge pressure system with dissimilar sized compressors

## 4.4 Conclusions

When a screw and reciprocating compressor are sharing a reducing load, that is below the total available capacity of the system, the screw compressor should be fully loaded and the reciprocating compressor used for load following. This conclusion is a result of the part load characteristic in Figure 4.6 which indicates that the efficiency of the screw compressor decreases as the percentage full load capacity is reduced.

When two screw compressors are sharing a load, control strategies should avoid operating any screw compressor below 50 percent of its full load capacity.

Screw compressors are better suited for base loading where they can be run at full load all the time.

Due to many practical reasons compressors may consume different amount of energy even if they are rated same. In high power systems this difference can be significant.

In multi compressor system, whenever there is a redundancy in compressors, it is advantageous to evaluate compressor energy consumption in different operating conditions and then selecting the most efficient compressor to run.

In case of multi screw compressor system, rather than applying specific rules for loading and unloading of compressors via slide valve, a model based approach, which uses database of different compressor parameters and minimizes energy according to each compressors' performance characteristics, is more efficient.

Data driven self-updating modeling of the system is more accurate than the fixed one time model.

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# Vita

The author was born in Vadodara, India on May 22, 1986 to Bhupendra and Manjula Makati. He earned his Bachelors of Engineering in Mechatronics Engineering from Sardar Patel University in July, 2008 with distinction. The author then worked as a Trainee Project Engineer at Hi-Mak Process Controls. In August 2009 he joined Lehigh University to pursue higher studies as well as start working as a Project Engineer at Lehigh University's Enterprise Systems Center. He will receive his Masters degree in Mechanical Engineering in May, 2011.