# Optimization of Various Parameters for Enhancing the Performance of Multi-compressor Refrigeration System


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Under the guidance of
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# CERTIFICATE

**DELHI TECHNOLOGICAL UNIVERSITY**

(Formerly DELHI COLLEGE OF ENGINEERING) Date:-_____

This is to certify that report entitled **"Optimization of various parameters for enhancing the performance of multi-compressor refrigeration system "** by **Mr. Dharmendra Pratap Singh is** the requirement of the partial fulfillment for the award of Degree of **Master of Technology (M.Tech)** in **Thermal Engineering** at **Delhi Technological University, New Delhi.** This work was completed under my supervision and guidance. He has completed his work with utmost sincerity and diligence.
The work embodied in this project has not been submitted for the award of any other degree to the best of my knowledge.


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# Abstract

The present optimization has been done on multi-screw compressor system. Refrigeration Cycle requires a lot of improvements for better performance to fulfill the desired condition with high performance in multi-compression system. Many of the parts run during the cooling effect and the running condition of system may vary as per the condition like ambient condition, part load, full load etc has also been changing which create an effect on the suction pressure, evaporator temperature, the rate of heat exchange, refrigerant flow and many other factors which are continuously changing.

For efficient cooling process, the noise parameter of the system has to be analyze and its affecting level for the designed system. These noise parameter has controlled (minimize) by some controlled parameter. In this project, the optimization parameter is the Power and the affecting parameters are temperature, discharge pressure and slide valve positioning.

The variation in these parameter with different working condition has analyze. The deviation in the effective cooling process can be obtained by its optimization. For the optimization process, taguchi approach with four factor and three level has selected which is best suitable with these conditions. This approach provides the best suited result for the different working conditions as per the requirement of the company. So the main motive of this work is to analyze the noise parameter and provide the best suited parameters in different working conditions so that effective and efficient cooling may be provided for the organization.

# Chapter-1

## 1. Introduction

In this chapter, the required basics of refrigeration and its analysis has been discussed which is based on reverse rankine cycle . Working fluid (refrigerant) is circulate through the system for making the cooling effect which will vary with the required condition (parameter) .The increase / decrease in latent heat of vaporization of refrigerant is the only concept which is used to transfer large amount of thermal energy [01] and according to this , required pressure is also changes due to absorb / expel the energy during the phase changes of the working fluid .

However, This cycle has reservoir (hot) generally at room temperature and a cold reservoir , the refrigerant which is used as the working fluid must contain low boiling point . Due to this condition , some working fluid recognize as an practical refrigerants. Required basic principle of thermodynamics has also been discussed in detail [05]. Different stages of cycle and every changes of the working fluid which are referring by the certain plots for example pressure and enthalpy variation etc.

### Related theory about the refrigeration system.

Cooling is the process which is done by transfer of heat to system / surroundings in a controlled working conditions . Work of heat transmission generally driven by mechanical work and also be driven by magnetism, electricity, laser, heat, work etc . Many application which required cooling based on diffrent refrigeration cycle [05] for example , cooling used for household forpuse , air conditioning, cooling in industrial stations (freezers), cryogenics etc. Heat pump has also been used for the heat output in the refrigeration process which may be designed to be a reversible but it is quite similar with the air conditioning .

Many applications are based on the refrigeration system for example industrial freezers, cryogenic, air conditioning system , heat pumps, and medical purpose etc . As per the second law of thermodynamics , some required work must be perform for happening this things , eg. If we put the hot coffee in the cup and leaving it to the atmosphere [05], it temperature decreases and shows cooling effect and after sometime this coffee comes to the surrounding temperature because heat is transfer from the coffee to the surrounding , which makes it cooling . This is the principle which is used in refrigeration system that is by removing the heat from the product (system) and transfer to the outside (surroundings).

### 1.1 Components of the refrigeration system

Refrigeration System has five components:

1. Compressor.
2. Expansion Valve.
3. Condensor .
4. An Evaporator .
5. Working Fluid (Refrigerant) .

For functioning the refrigeration cycle, the component of the refrigeration system has been described below .

## 1.1.1 A compressor.

Function of the compressor is to increase the pressure by providing energy and reached in saturation temperature of the working fluid which is at high level and can condense by rejecting certain energy trough the condenser. In a compressor, the working fluid reaches at high energy level (temperature) . This high energy working fluid enters to the condenser and rejecting the extra energy higher than the ambient temperature to the surroundings. By this method, the cooling effect has been produced from the working fluid (refrigerant). This process allows to condense the high pressure fluid into an ambient temperature.

Compressors generally used in the refrigeration system are rotary, reciprocating and centrifugal compressors. Reciprocating compressors is the most popularly used compressors for domestic or MSME industries, large industries and automobile industries. Reciprocating compressor is similar compressor which is used in automobile engine. Piston is driven by a motor which compress refrigerant in cylinder. As per the movement of cylinder which is used for the suction/expel the refrigerant. The suction process of the refrigerant is done from the evaporator. The intake valve closes when the pressure of refrigerant reaches to the pressure in the evaporator inside the cylinder. Exhaust valve has been used for pushed out the refrigerant from the condenser. The designing of intake and exhaust valves in such a manner the refrigerant flow in a unidirectional manner.

## Compressor and its details

Types of Compressors which is used for the refrigeration and the air conditioning system. Various types of compressor used in the refrigeration and air condining machineries :

a) **Reciprocating compressor :** One of the most widely used compressor of the refrigeration system. Like the automotive engine, it has piston and cylinder arrangement.

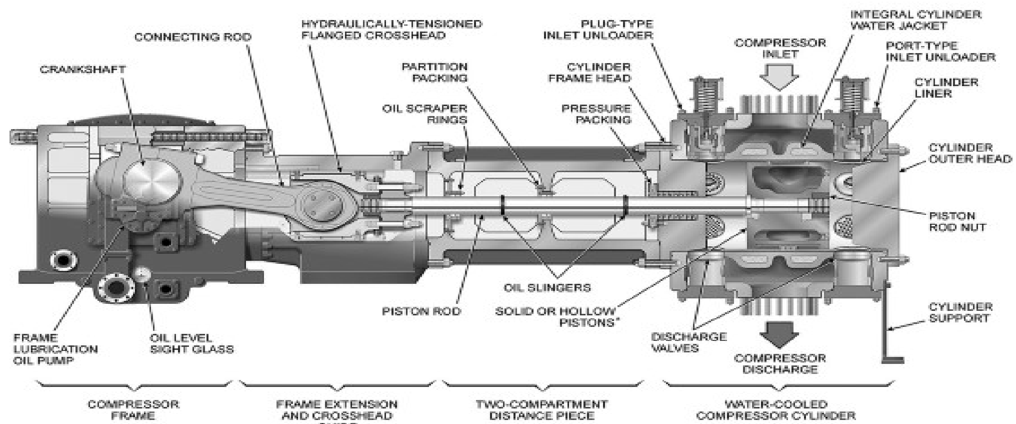


**Fig-1.1 Components of the reciprocating compressor [01]**

b) **Screw Compressor :** This compressor contain the pair of meshing screws between which the compression of refrigerant takes place. This compressor generally produce high pressure for small quantity of fluids. This compressor consume less input power than reciprocating compressors. Refrigerants used in this compressor are generally R12, R22 etc.

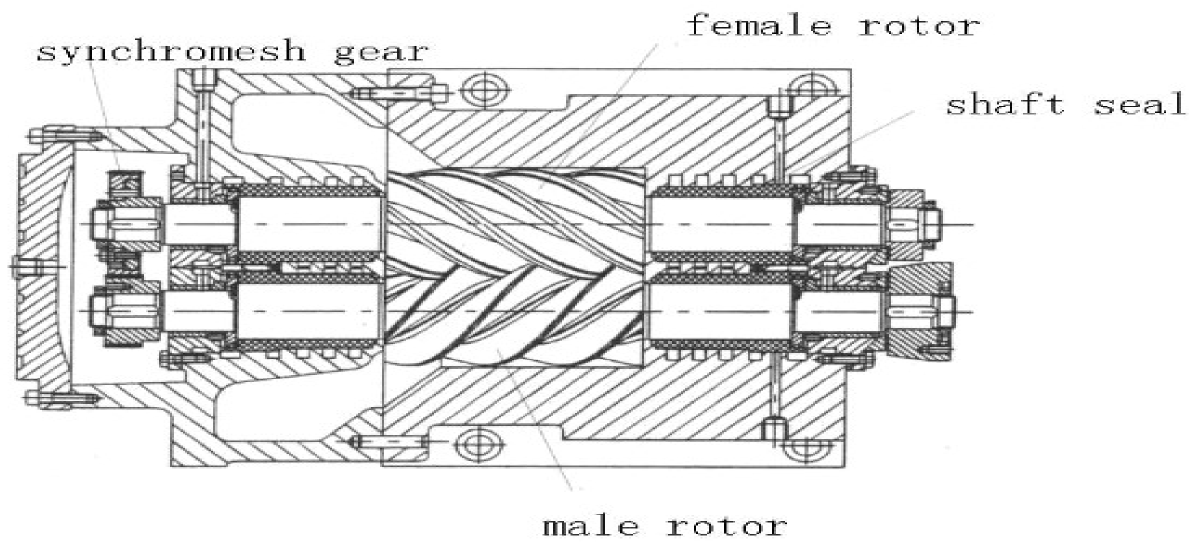


**Figure1.2 Components of the screw compressor [01]**

**c) Rotary Compressor :** This compressor has two rotating elements (gears) in which compression of refrigerant takes place. Pumping of refrigerant which is at lower or moderate condensing pressures also can be achieve in this compressors . This compressor generally used for the small volume of fluid at lesser pressure so this compressor has fewer applications.

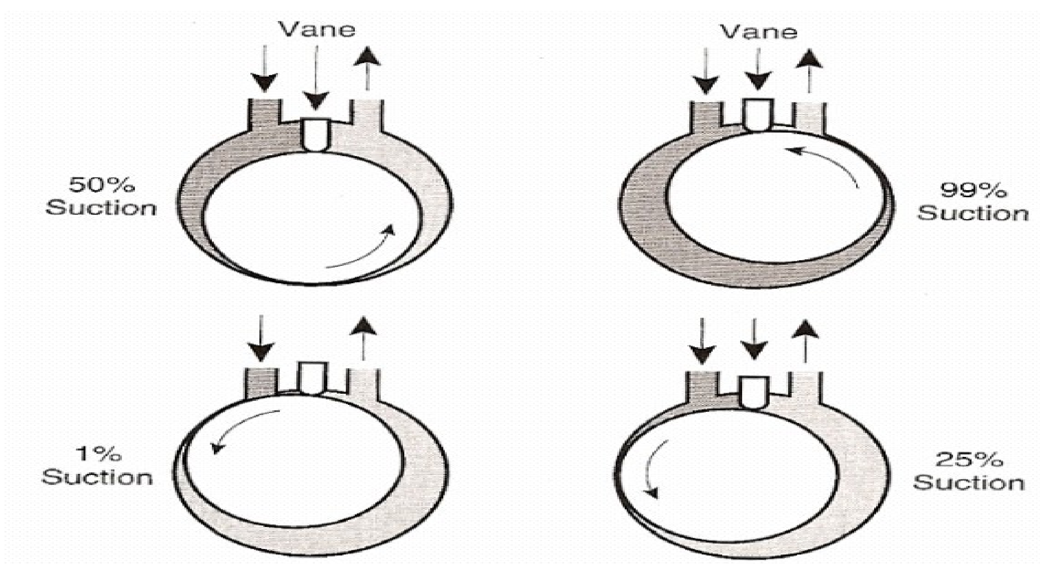


**Fig-1.3 Working of rotary compressor [05]**

**d) Centrifugal compressor :** This compressors contains blower which can handle large quantities of fluid with a lower condensing pressure and impeller. Best compressor for the working refrigerants like R113,R11 etc.

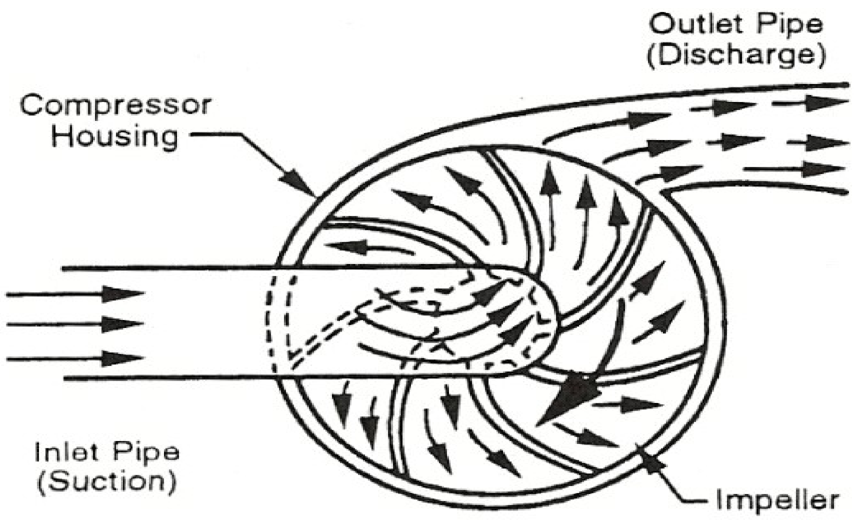


**Fig-1.4 Centrifugal Compressor [01]**

**e) Scroll Compressor :** This compressor contain two interleaved scrolls in which one is orbit eccentrically without rotating and another is fixed. During its working, certain small gaps are produced between scrolls where the compression of refrigerant takes place. These scrolls may have some different shapes for example involute , hybrid curve and Archimedean spiral.

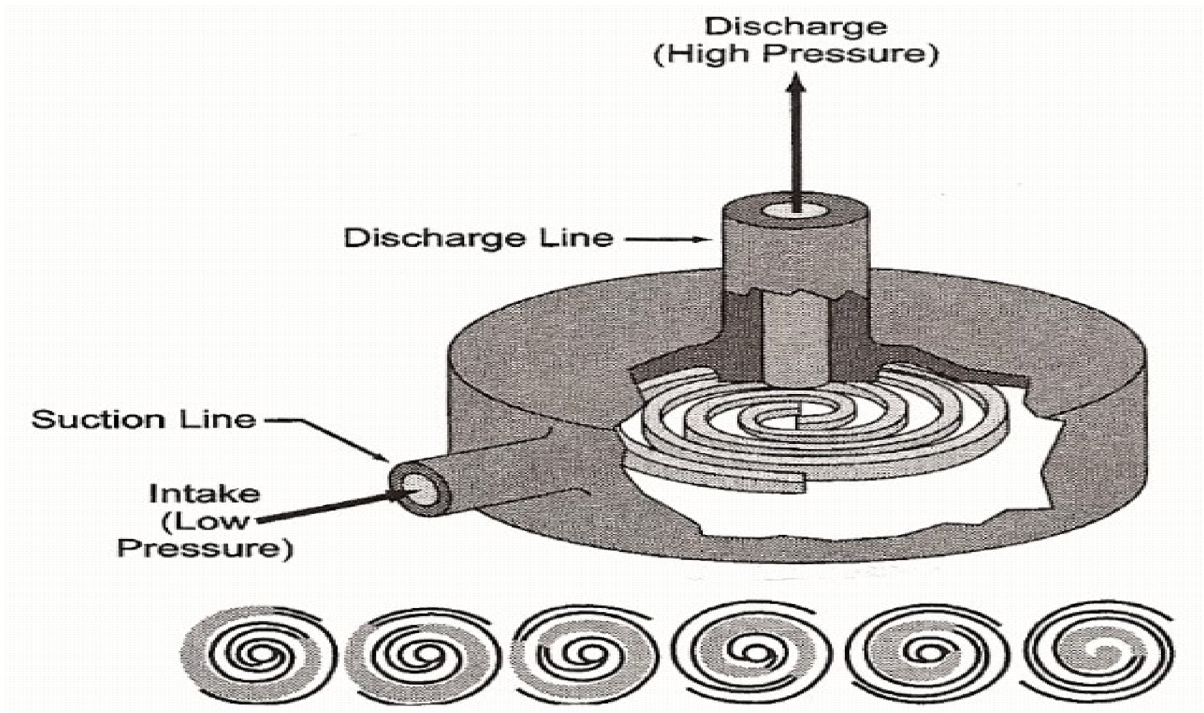


**Fig- 1.5 Scroll Compressors [05]**

### 1.1.2 Expansion Valve

**Expansion valves and its components :** Four types of the expansion valves which are used for the refrigeration system and also for the metering devices .

(a) Thermostatic Valve
(b) Automatic Valve
(c) Capillary Tube
(d) Float Valve

**(A)** Thermostatic valve: This device contains valve system for controlling the flow of refrigerant into the evaporator coil. This movement of refrigerant is controlled by the pressure available in evaporator. This device is able to function well during the fluctuation of load hence best for the air conditioning system.

**(B)** Automatic Expansion Valve : This device contains the valve system used to control the movement of refrigerant from the liquid line to the evaporator with the help of pressure actuated diaphragm which maintain the constant pressure inside the evaporator.

**(C)** Capillary Tube : This tube contains small inner diameter which can be coiled for part of its length. This tube is fixed in the suction line . Due to the small diameter , the working fluid used un this tube is properly calibrated with the standard at factory level(manufacturing stage ) Generally used diameter of the capillary tube is 0.031” to 0.065” mm and outside diameter are commonly range between 0.083” to 0.130”.

**(D)** Float Valve : This device is actuated by float which is immersed in the working fluid. High side float and low side float are used for controlling the movement of working fluid. High side float is fixed at the side of higher pressure which maintain a constant level of working fluid in side condenser. Low side float is used to maintain a constant level of working fluid in side the evaporator.

### 1.1.3 Condenser and its types

**(A)** Air cooled condenser : This device is used for small units for cooling system mainly in deep freezers, split air conditioners, window air conditioners household refrigerators etc.This system are generally used where load is less and quantity of working fluid is also less in a refrigeration cycle. Another name of air condensers is coil condensers because it is usually made of aluminum of copper coil.

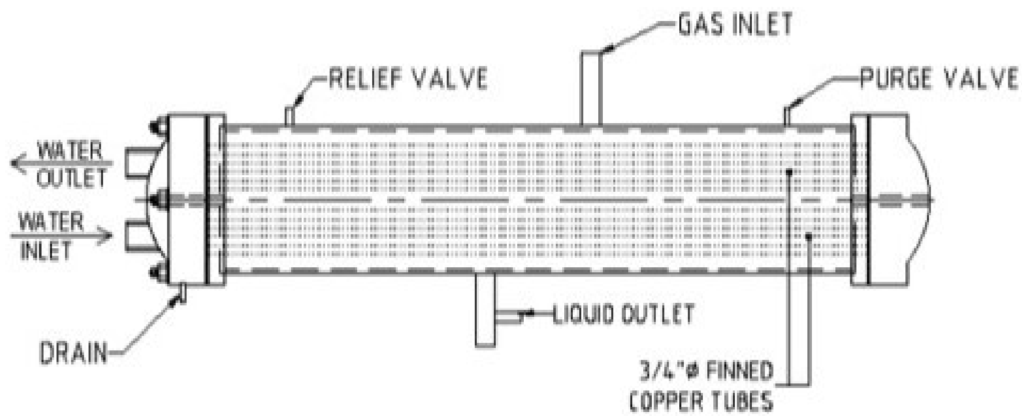


**Fig-1.6 Component of the air cooled condenser [05]**

**(B)** Water cooled condenser : This system is used for large cooling system which contains high load of cooling like central air conditioning plants etc . Water cooled condenser are of three types (a) shell and coil type , tube in tube or double pipe type and tube type . This system used for high load which required high quantity of movement of working fluid .

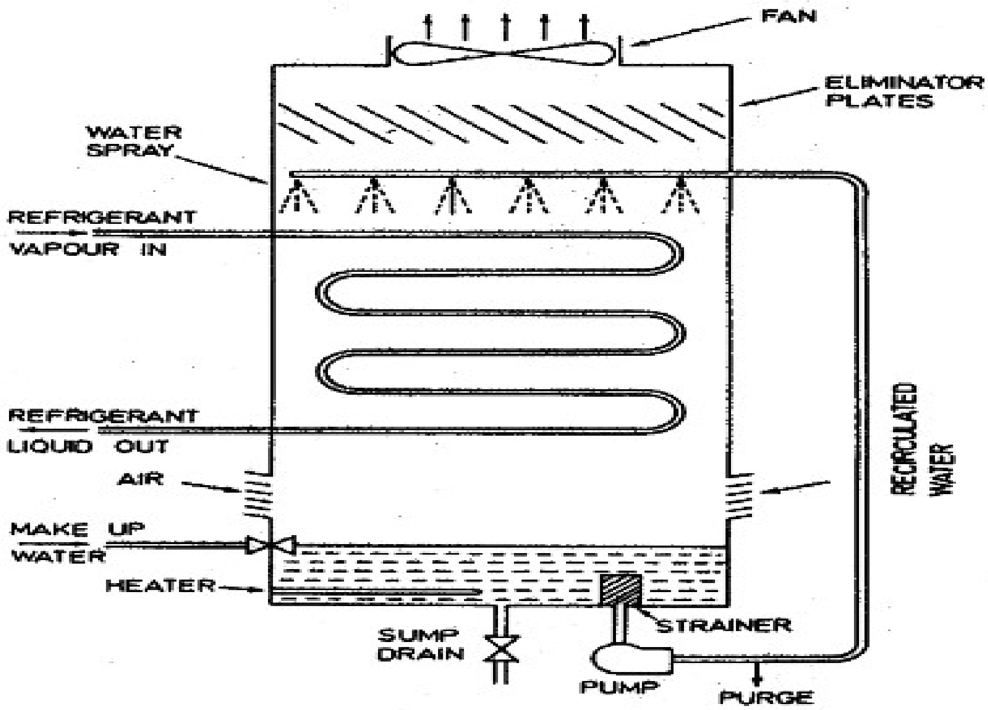


**Fig-1.7 Water cooled condenser [05]**

**(C)** Evaporative Condensers : This system is generally used in ice plants/cold storage . This system is a combination of air and water cooled condensers . High temperature working fluid flows inside the coil and at the same time water is sprayed on these coils and fan draws air from the down side to the up side .

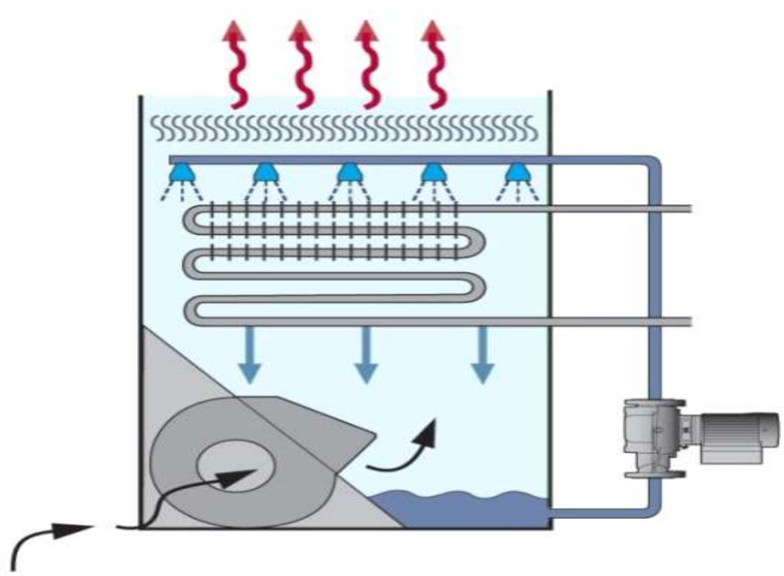

**Fig-1.8 Evaporative condenser [05]**

**1.1.4** An Evaporator : Working fluid enters in the evaporator at very low temperature and low pressure through the expansion valve . Working fluid absorb the heat from the substance that is to be cooled and refrigerant gets cooled by rejecting the heat from the substance and it gets cooled.

**(A) Evaporators and its type :** For high cooling load required in cooling plants , the evaporators are used for chilling / cooling purpose . Chillers are two types which is :

**(1) Flooded type evaporators**

**(2) Dry expansion type evaporator.**

For the dry expansion chiller type of evaporator, movement of refrigerant in the evaporator is regulate by expansion valve. This valve used to flow of refrigerant depending on the load of the plants.

Evaporator are classified on the basis of its construction:

**(1) Finned type .**
**(2) Bare Tube .**
(3) **Plate Surface** .

**1.1.5 Refrigerant (Working Fluid) :**

Refrigerant is a fluid which is used for cooling purpose in a refrigeration cycle. Refrigerant generally goes to the phase change that is liquid to gas and vice versa. Some of the working fluids are ammonia , chloroflouro carbons , sulphur dioxide, non halogenated hydrocarbon like propane etc .

# Chapter-2

## Literature Review

Many literatures may be found in this fields but the literature related with the specific aspect of industrial cooling regulate and its operation strategies is rather spase :

- Manske [13] perform a study on performance on optimization of induatrial refrigeration system . He had develop a detailed model of performance of vapour compression refrigeraton system including its subcomponent.
- Aprea and renno [16] had performed an experimental analysis of thermodynamic model of vapour compression refrigeration plant on varying the compressor speed .
- Primozic et al. [09] had discussed about staging in reciprocating system of compressor by following the rate of rise and fall of suction pressure .
- Larsen [02] had developed a model based control of refrigeration system . Work model based on based on set point optimizing control method which had been developed further. The optimizing control is divided into two layers where system oriented top layers deals with set point optimizing control and the lower layer deals with dynamical optimizing control in the systems. The objective was to derive a general applicable set point optimization method for the refrigeration system that can drive the set points towards optimal energy efficiency while respecting the system limitations.
- Widell and Eikevik [13] did study on reducing power load in multicompressor refrigeration systems by limiting the part load operation . An experimental analysis of compressor operation in large refrigeration system was undertaken and model for optimal compressor operation for energy efficiency was developed . The system used five screw compressor operation for the energy efficiency was developed . The system used five screw compressor and ammonia as the refrigeration, with slide valves to regulate the compressors and the match their refrigeration capacity with product freezing loads . Optimising operation was made both with and without variable frequency drive .
- Reindle et al [07] had suggested load sharing strategies in the multicompressor refrigeration system. In that study they had showed that when two identical screw compressor are operating in parallel there exist 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 other unloaded to match the remaining system load . When two screw compressor of different sizes are operating , an optimal compressor control map can be developed which maximizes efficiency of the entire system over the range of the loads .
- Zhu et al [12] had achieved new constructional solution which increases the refrigeration efficiency . On the basis , theoretical analysis which had a second input at the hot end of tube which connected to generator . Experimental and numerical analysis confirm that the double inlet tube has improved performance over the normal cooling system.
- Ju et al [19] had developed an improved model for simulating the fluid flow and seen its effect on the refrigeration.
- Linde Group [26] had developed the world’s first air separation plant for the production of the oxygen in 1902 , the linde group installed the largest pressure swing absorption

plant in Europe with vacuum regeneration for production of oxygen from air . Line north America is the one of the world's largest helium supplier and since 1994 linde has been the sole provider of helium.

• Herold [28] presented two papers to show the effect of Prandtl no. on heat transfer and pressure drop in OSF array. Experimental study was carried out in the first paper to study the effect for which they used the seven OSFs having different geometries and three working fluids with different Prandtl number. At the same time the effect of changing the Prandtl number of fluid with temperature was also investigated. The study was carried out in the range of Reynolds number varying from 10 to 2000 .

• Orifice pulse tube refrigerator was introduced in 1984 by Mikulin et al [32] The invention of the simple OPTR which is regarding as the mils stone of the developments of pulse tube cooler for an industry based refrigeration.

•Starch and Radebaugh [33] had developed an analytical model for OPTR with a simple expression for the gross refrigeration power which agrees with an experiments.

•Wu et al [28] had done the numerical analysis for an OPTR having a valves less compressor and described the process occurring in the pulse tube.

•Richardson [34] had reported that the valved pulse tube otherwise known as the orifice pulse tube and is able to reach much lower temperatures which is required for an industry.

•From the studies of few researchers like Ray and London [35] it is easy to get information regarding the effects of OSFs on heat transfer and pressure drop.

•Dugan et al [04] made experiments and analysis considering the above factors to get better thermal and hydraulic performance from the OSFs. Sixteen types of OSFs and flat tube heat exchangers were used to make the experimental studies on heat transmission.

•Gifford and lonsworth from Syracuse university [23] ,pioneered introduction the pulse tube refrigeration for an industry like yogurt , milk etc which gives a new method of achieving cryogenic temperature for the industry . the development of this model for research purpose started in year 1963. First paper regarding improvement of the refrigeration system of an industry which had multicompressor system was published which giving a brief account of progress made . The method was described as
" pressurization and depressurization of any closed volume from a point on its periphery sets up temperature gradients in that volume". Thus the temperature gradients obtained dependent upon geometry of closed volume and its operating conditions.

## Chapter-3

### Equation used for Calculation i.e Basic fundamental of thermodynamics

Some fumendamental of the thermodynamics which is used directly or indirectly in the refrigeration process .

### First law of thermodynamics

The 1st law of thermodynamics deals with the conservation of energy that is valid for an insulated system which is the change of internal energy ΔU equals the sum of applied work W and heat Q .

$$\Delta U = Q + W \tag{3.1}$$

### Specific Enthalpy

It is defined as the property that only depends on the state of the refrigerant like pressure , temperature and quality which is given as :

$$h = u + Pv \tag{3.2}$$

from the above expression ,

u-internal energy of the system

p-pressure

v-volume of the refrigerant

$(m_i)$- mass entered in the control volume

$(m_o)$- mass exit from the control volume.

Rate of change of internal energy can be calculate as :

$$\frac{dU}{dT} = \dot{Q} + \dot{W} + (m_i h_i - m_o h_o) \tag{3.3}$$

And for the steady state condition , the flow process is :

$$\dot{Q} + \dot{W} = \dot{m}(h_i - h_o) \tag{3.4}$$

## Second law of Thermodynamics and Entropy

The 2nd law of thermodynamics states that energy stored in the form of heat which cannot be converted in the equivalent amount of the work . This means the efficiency involves in the process of conversion of energy from heat to work will never be 100% . For a reversible cycle, the efficiency of the vapour compression refrigeration cycle is one theoretically and the assumption for this is that the change in the entropy for this process is zero . The specific entropy is same as the enthalpy of a refrigerant specific property only dependent on the state of the refrigerant .

The specific entropy ($s$) is same as the enthalpy of a refrigerant specific property only dependent on the state of the refrigerant. By using the above definition , the entropty of the system can be defined as :

$$dS = \left( \frac{\partial Q}{T} \right)rev \qquad (3.5)$$

from the above expression , the terms used in the equation is S – entropy , T- temperature , and rev means reversible , for an isentropic process the change in entropy is zero .

## 3.1 Vapour compression refrigeration cycle

The basic purpose of vapor compression cycle is to remove heat from one reservoir to another reservoir which is at different temperature. In the process of cooling, the refrigerant absorbs energy from evaporator (system) and then passes to the compressor for changing the phase with help of input energy which increases the pressure and then refrigerant reaches to the saturated state . After compressor , refrigerant goes to the condenser where it releases energy and getting cold with the help of heat exchanger (condenser) . After the condenser , the refrigerant goes to the isentropic process where enthalpy of the refrigerant remains same , during this process the refrigerant changes its phase and comes at low temperature and pressure . At low pressure condition , the corresponding temperature is low and vice versa . The condenser pressure be ($P_c$) and evaporator pressure be ($P_e$) , then the pressure difference ($P_c > P_e$) , an expansion valve is installed at the outlet of condenser . The expansion valve is generally has an adjustable nozzle which helps in upholding a pressure difference .

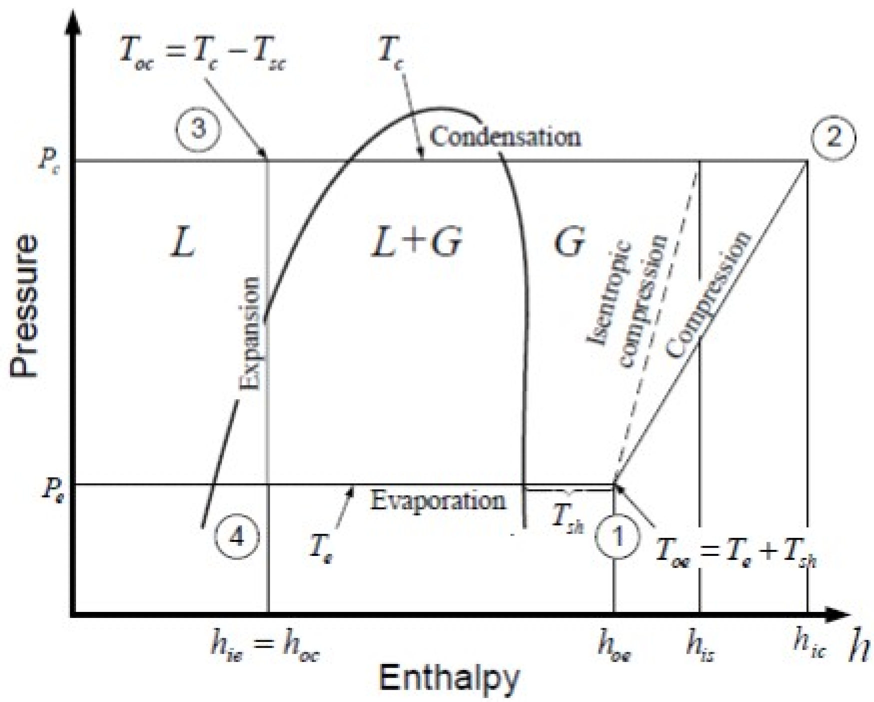


Fig 3.1 shows the variation between pressure and enthalpy (h-log(p)) [05]

The vapour compression cycle is plot between enthalpy (h) and pressure logarithimic (log(p)) . this diagram is only for the specific refrigerant which gives the general view of the different stages/phases of the process . At inlet and outlet condition , the specific enthalpy and other property denotes with "*i"and "o"* respectively .

**3.2At compression stage : Point 1-2 :**

When the working fluid (refrigerant) goes to the inlet of the compressor which is low temperature and pressure and at saturated condition also . During the compression process , electrical energy is used to convert the pressure and temperature of the refrigerant . The work which is required in compression process can be calculate by forming a control volume around the compressor , assuming it is insulated (adiabatic compression) and using equation 3.1 ,

In the case of heat loss during the process , the heat loss factor $f_q$ can be introduced for calculating the remaining energy after loss ,

$$f_q = \frac{(h_{ad} - h_{ic})}{(h_{ad} - h_{oe})} \qquad (3.6)$$

The work applied to compressor ($W_c$) as: can Eq. (3.7) is written

$$\dot{W}_c = \frac{1}{1-f_q} \dot{m}(h_{ic} - h_{ie}) \qquad (3.7)$$

### 3.3 At Condensation stage ; State Point 2-3:

After exiting of refrigerant from the compressor , the refrigerant is ready to enter in the heat exchanger (condenser) . At the entry of the condenser , the refrigerant has high temperature and pressure and at the saturated condition . The condenser enable the refrigerant for losing the heat which converts its phase from saturated gas to saturated liquid because of high pressure . Condenser has many types, according to the cooling capacity required , the type of condenser is used for better efficient process . The heat which is removed from the refrigerant goes to the fluid(generally water) which is again ready for the cooling for another circulation . This cooling of the water is done by blown of air through spray of the water (increased surface area) and helps the water to reduce the temperature for further cooling of the refrigerant . The amount of heat rejected from the condenser may be computed as :

$$Q_c = \dot{m}(h_{ic} \dot{-} h_{oc}) \quad (3.8)$$

From the above expression , the terms used are $Q_c$ - heat from condenser , $h_{ic}$ - enthalpy of the refrigerant at the inlet of the condenser and $h_{oc}$ - enthalpy at the exit of the condenser.

If heat loss occurs during the process by other means , then the $Q_c$ (heat from condenser) can be calculated as :

$$\dot{Q}_c = \dot{Q}_e + \dot{W}_c - \dot{W}_c.f_q \quad (3.9)$$

### 3.4 At expansion stage : State point 3-4 :

After condensation , expansion of the refrigerant takes place at constant enthalpy (also known as isenthalpic process) . At the inlet of the expansion tube (3) , the refrigerant is at the high pressure and in saturated liquid state . During the expansion process , refrigerant loses the pressure as well as temperature by keeping the enthalpy of the refrigerant remains same . At the end of the expansion process(4) , the refrigerant is in two phases that is in liquid pahse and as in gaseous phase . During the expansion process ,the enthalpy remains same which can be represented as i.e. $h_{ic} = h_{ie}$.

### 3.5 At evaporator stage : State point 4-1:

After the expansion process , the refrigerant has low temperature and low pressure which is ready for absorbing the heat at constant pressure (phase changing(single phase)) . The system's heat can be absorb by the refrigerant and go to the single phase that is gaseous phase . During the evaporator process , the refrigerant goes to absorb heat at constant pressure , due to which the phase of the refrigerant changes . This changes in phase of the refrigerant helps it for the next process that is compression. The heat absorb by the refrigerant can be calculated by forming a control volume around evaporator and using the equation (3.8)

$$Qe = \dot{m}(h_{oe} \dot{-} h_{oc}) \qquad (3.10)$$

From the above expression the terms used has already discloses , Now the refrigerant which is at saturated gaseous state at low pressure is ready to go to the compressor process

**Chapter-4**

## Background and Motivation

In the practical world , the use of the refrigeration reaches to a general life either individual , MSME industries or large industries . Usually residential units has standard single compressor system while in the commercial system , they has either custom built or standard off the shelf system , in the scenario for the industry custom in built unit is used . So for the large or medium industry which has around 1000 tonnage to 3000 tonnage load , every process involves in the cost analysis for the industry . So every individual part of the industry should be optimize so that the cost analysis for producing the high quality goods is less and process required for these goods are more efficient .

Generally industry goes for the optimization process which mainly involves direct capital investment. The cooling process required an input energy like in the form of electricity, maintenance cost, rent , refrigerant and storage (inventory) . So for the optimize/less investment of the capital by the industry ,the cooling process should also be optimize which will definitely help the industry for the efficient process during peak hours with less input energy by the industry . Usually single standard compressor system which is commercially used are custom built or standard off shelf system. If final cooling system/plant is not work optimally which decrease the efficiency of the plant/system. Due to this, optimization has been done at individual part as well as combination , so that system will perform efficient .

For this analysis , the factors which are affecting the cooling process are important so that the noise parameter should be minimize and makes process more efficient . All the refrigeration process are generally adopt the reversible process (theoretically) for the minimum loss of the energy and makes the process best efficient . This kind of model is theoretical which helps to know the further improvement in the process .

So at the process level , it is difficult to make it optimize . so there may be another method for the multi-compressor system to make efficient . Generally multi-compressor system involves the fluctuation of the parameter for providing the different temperature at the inlet and exit side of the chain of manufacturing of the product or may be more than two stations, the requirement of different temperatures at the different stations and may be even fluctuations of temperature at this station after certain period of time. This will help to optimize the process of the cooling at different station with the help of particular set of parameter for the different working hours (peak hours or in idle hours). After analyzing these parameter, noise parameter can be minimize by the controlled parameter which will help to reduce the extra expenditure of the cost .

| Industry (Stony feild form yogurt) | NAICS | 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 | 67,763 | 1,025,149 | 6% | 100% |

Table 4.1- Consumption of energy at end use of manufacturing installation [18]

## 4.1 Objective of Thesis

The main objective of the thesis is to analyze the process in the working hour in the industry which involves the cooling process during the manufacturing of the product . In case of multi compressor refrigeration system , the cooling process can be achieve by fluctuation of parameter in different working hours . Some process may be efficient and definitely some are not due to fluctuation and providing different temperatures at the different stations . So first step to analyze the process which has noise parameter and certainly dependent on some controllable parameter .Second step is to understand the requirement of the industry regarding quality of product and boundary up to which the company wants to invest for the cooling system .

Requirement of companies mainly is to produce high quality product with less manufacturing cost . This manufacturing cost has some percentage of cooling process at the factory level which can be minimize .

After analyzing the affected data for the cooling process of the organization. Noise parameter (uncontrolled parameter) can be minimized by the controlled parameter of the process which can minimize the input of the system . This can be done with the help of DOE (Design of Experiment) software and L9 array with Taguchi optimization technique has been used .

### 4.2 System Overview

For this project , food processing facility refrigeration plant's(stonyfield form yogurt) is to be examined which produce milk based product . This plant maintain 19°F suction temperature and the the discharge temperature floats between 70°F to 90°F i.e. 115 psig to 166 psig discharge pressure . The compression process has liquid cooling injection system . This plant has the facility of single stage screw compressor . A RSEnergy software has been installed in this industry which gives the data at each and every point of stage with regression if required . The data required for the calculation has been taken for this software and then certain calculation and analysis has been performed . The refrigerant which is used in this process is ammonia (R717) . This industry has multi-compressor system in which 6 compressor has same fixed volume index and one compressor has variable index . The requirement load of the organization for cooling purpose is in between 2000 tonnage load and 2200 tonnage load which may vary during the working hours in the organization . The specification of the compressors which is used in the industry are given below :

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

Table 4.2 Specification of the compressors [18]

Heat exchanger and evaporators which are used in this plant are either top feed or pumped liquid overfeed because larger quantity of the refrigerant has been pumped by the compressors . These pumped liquid overfeed system needs a receiver vessels for storage the refrigerant which are in two phase mostly. The main purpose of these vessel to separate the phases of the refrigerant so that pumping can be performed efficiently and maintain the static head of the liquid.
In condenser , sprayed water uses which evaporates during the refrigerant phase transfer process Each condenser of the multi-compressor system has six 7.5 HP axial fans to provide forced convection for effective cooling . these condensing fans has two step speed control strategy (50% speed and 100% speed)

## Chapter-5 : Data analysis and Conclusions

Pressure readings (suction) , slide valve readings and current draws readings are taken four times in a day . A software called RSEnergy Matrix is also installed which helps to provide the data in the regression form or in digital form and is directly connected to the computer server for further command and assistance . Figure 5.1 and 5.2 clearly shows the $2^{nd}$ order polynomial trend lines which is created from the data for each compressor provided by the installed software. Subranges are also provide bt the same software if required .

After the calculation , it comes to know that the compressor-3 and compressor-6 draws 15A to 20A extra power than compressor 2, compressor , 4 and compressor 5 at the 100% slide valve condition i.e. at peak hours of the industry . In case of 85% slide valve condition , compressor -3 becomes more efficient than compressor-6 . Compressor – 4 is the more efficient compressor in high slide valve values .

In the pressure range 130 psig to 134 psig , compressor-3 and compressor-6 are less efficient than compressor-2, compressor-4 and compressor-5 for 100% slide valve condition. If we focus on the compressor-2 , the polynomial obtained in the previous experiment is different from the current experiment . Compressor-2 draw less power in pressure less than 130 psig as compared to 130 psig to 134 psig . If we are talking about the other six compressor , compressor-5 become most efficient compressors in the 50% slide valve condition.Compressor-4 is the more efficient compressor among all the compressor in the range of 130 psig to 134 psig pressure and all slide valve variation .

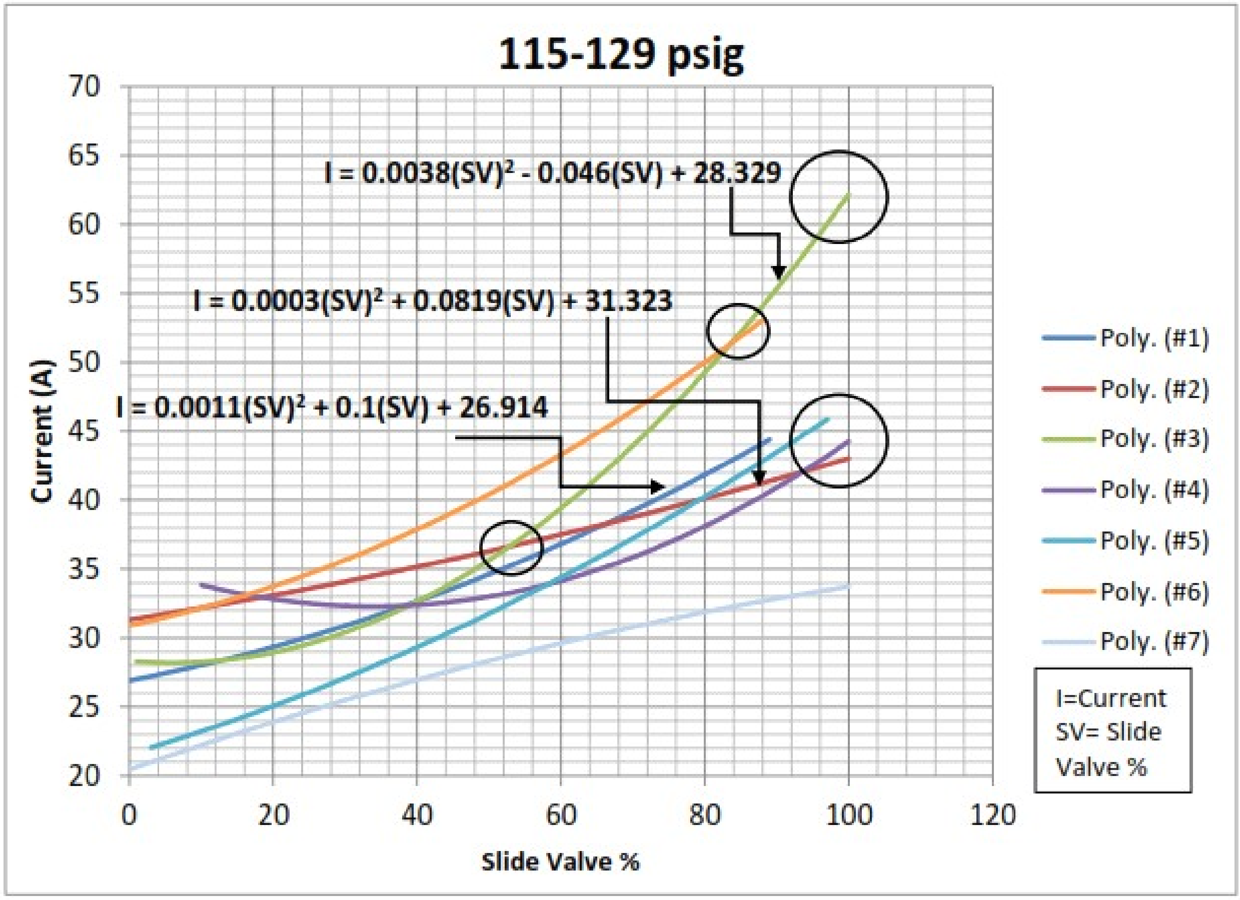


Figure 5.1 shows the plot between the % of slide valve and the current draws by the different compressor in the pressure range of 100 psig to 129 psig [18]

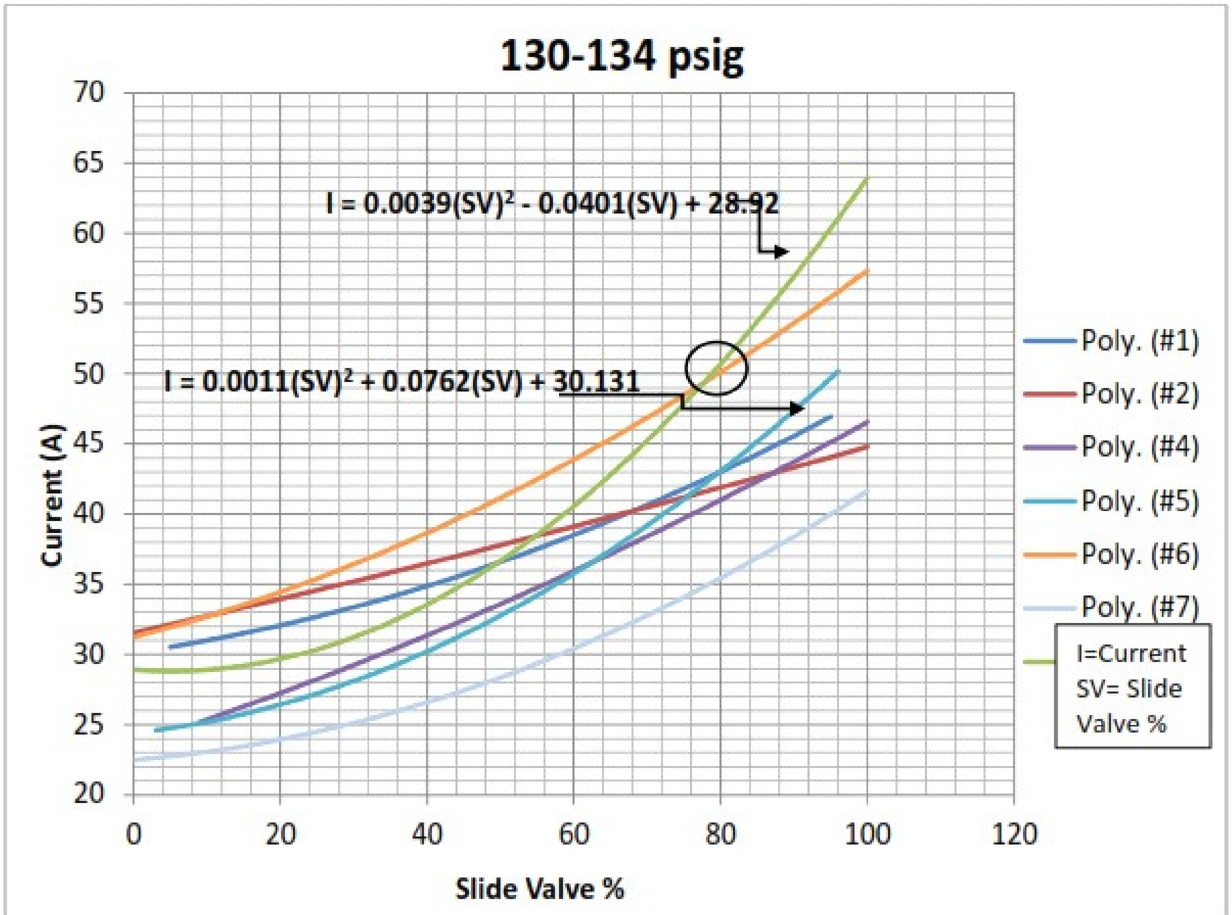


Figure 5.2 Shows the plot between Slide valve % and current draws in the pressure range of 130 psig to 134 psig [18]

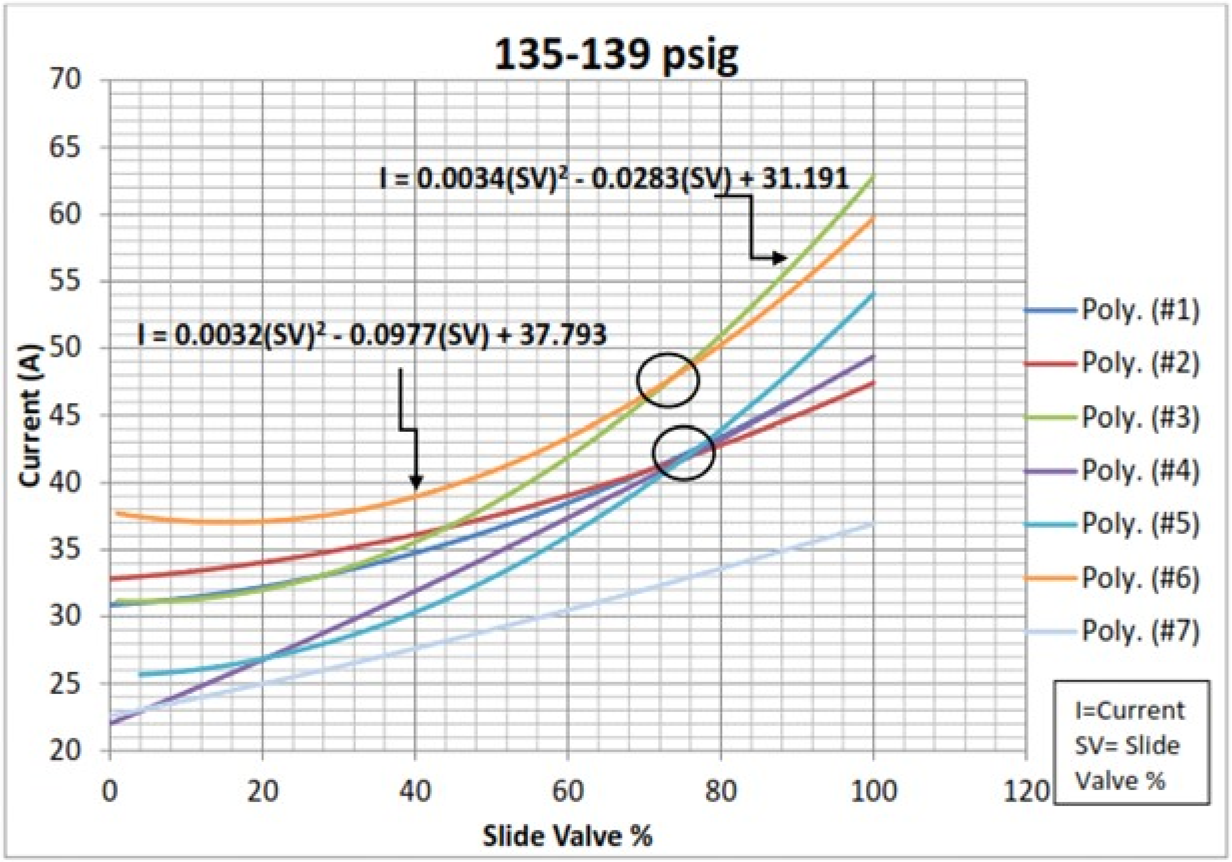


Fig: 5.3 Shows the variation of SV% and current draws for 135psig to 139psig [18]

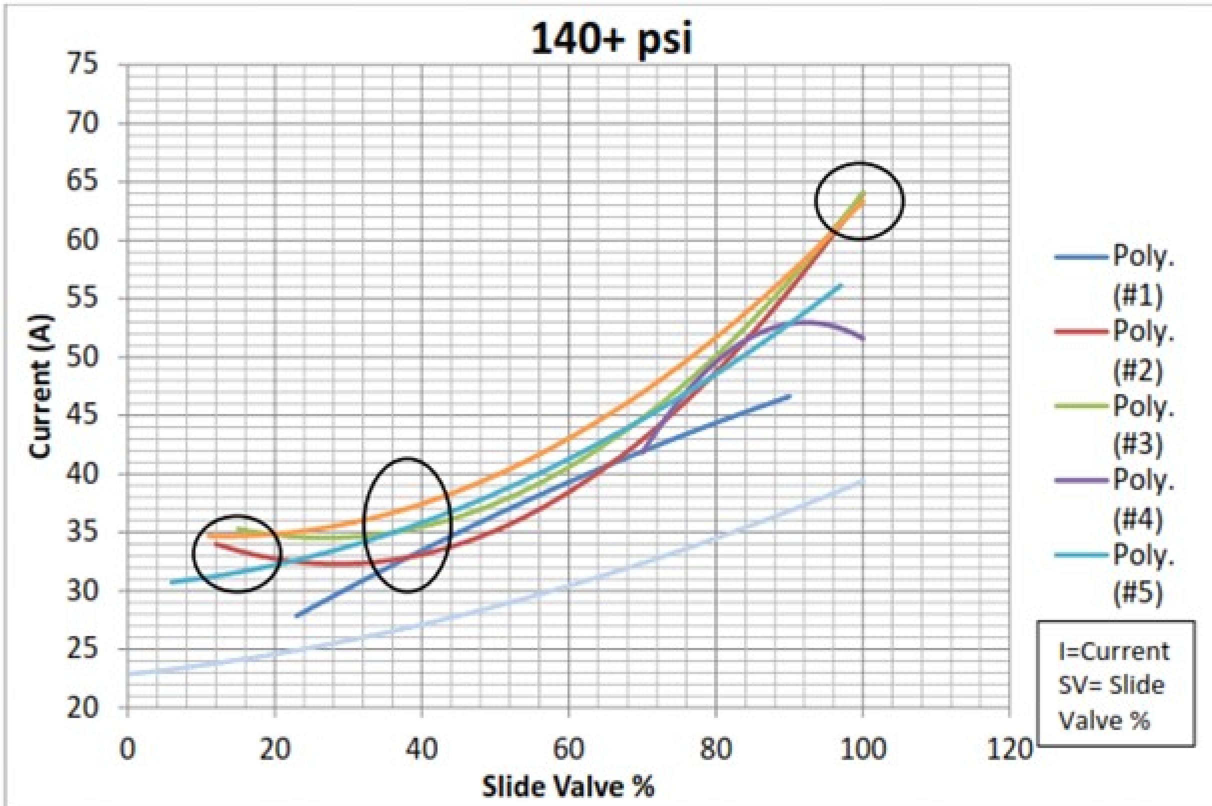


Fig: 5.4 shows the variation in SV% and current draws in 140psig and higher discharge pressure .

In Compressor-4 is the most efficient compressor in the condition of 135 to 139 psig and all slide valve condition . Figure 5.4 shows the variation between sv% and current draws for 140psig and higher discharge pressure condition , in this graph , compressor-2 ,3 and 6 draws the same amount of current in full load condition . In case of unloading, compressor-2 becomes highly efficient than the compressor-3 and compressor-6 .
Compressor-2,3,6 draws the same current in the low slide valve values .Compressor-1 shows good result in this range as compared to the rest of the range and compressor-7 is the least power draws in the pressure range but the power is mention in th kw/tonnage which makes compressor-7 efficient in few working condition .

From the above analysis , it is clear that due to fluctuations in the compressor's power as per the pressure range changing . So optimization can be done at the technical level which may minimize the noise parameter for the compressor to make the cooling effective and efficient. For this kind of system , a new design and a new approach is required which help toptimize the power draws and this reduction shows some accountability in the cost analysis.

| Condenser+ Compressor Power | Systeam Full load average specific power | Saturated Condenser Pressure |
|---|---|---|
| 1 | 1.55 | 130 |
| 2 | 1.1 | 140 |
| 3 | 0.9 | 150 |
| 4 | 0.8 | 160 |
| 5 | 0.75 | 170 |

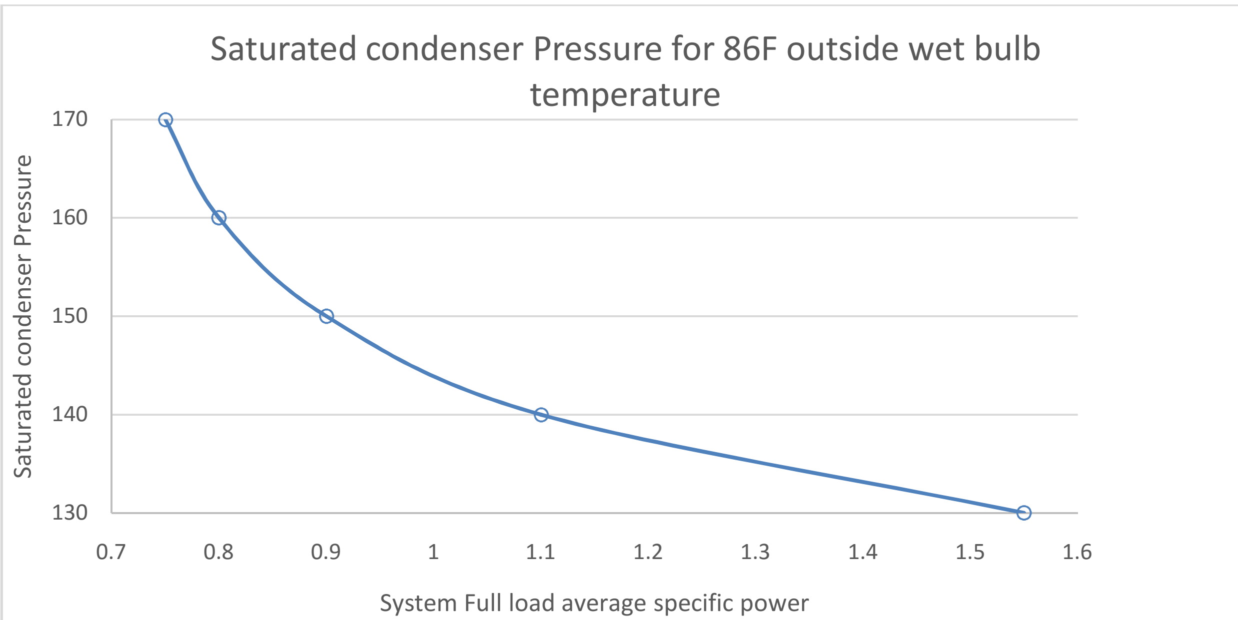


Figure 5.5 Saturated condenser Pressure vs full load for 86°F outside wet bulb temperature

On eight working hours , the program predicts the behavior of discharge pressure with the condition of (a) ambient condition ,(b) avg full load specific power condition and (c) part load specific power condition. This data of specific power will be use in optimization process . Fig. 5.7 and 5.8 represent the zero day (for an eight hours period) part load and full load specific power for all compressors. This analysis has taken 60°F to 70°F and RH(relative humidity) is 50% to 70% .

The observe discharge pressure in ambient conditions are in the range of 133psig to 142 psig. As per the fig. 5.8 , compressor-2 and compressor-4 has minimum consumption of power at full load condition . and as per the fig.5.8 the compressor-5 will consume minimum power at part load operating condition.

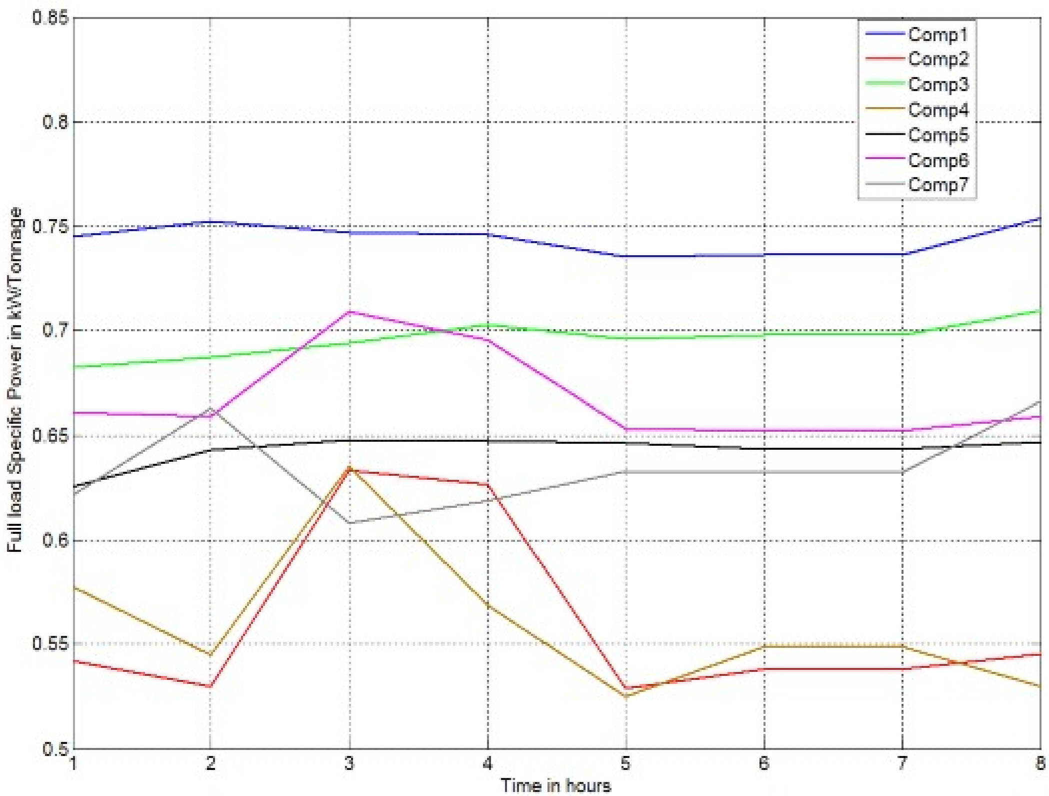


Figure 5.6 Average full load specific power vs time of operation (8 hour predicted period) [18]

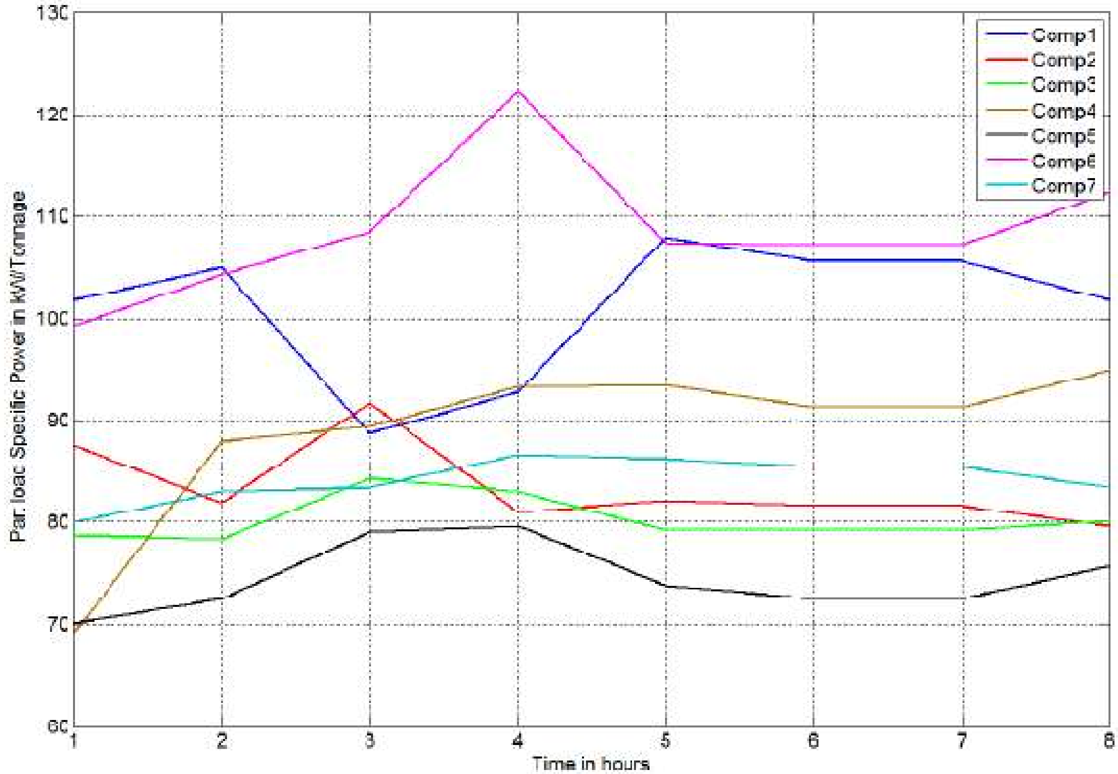

Figure 5.7 shows the plot between total part load specific power vs time (an 8 hour) [18]

### 5.1 RESULTS AND DISCUSSIONS

From fig.-5.6, the consumption of electrical power at 115 psig pressure by each compressor , for 1 tonnage cooling load with varying slide valve conditions . At this discharge pressure , compressor-1 has poor performance for every slide valve conditions . Compressor-2 is good at performance above 50% slide valve . Compressor-3 at 100 % slide valve condition , draws 30% extra power than compressor-4 but it draws 10% to 20% less power at 50% slide valve condition than compressor-4.
The variation of the specific power at 115 psig discharge pressure condition for higher slide valve has shown in figure-5.4. For the range of 100psig to 129psig discharge pressure, compressor-3 draws 61.7A at base load condition and 100% slide valve as compared to compreesor-2 which draws 42.5A current. Each compressor's power variation at 135psig discharge pressure and for 1 tonnage cooling load at different slide valve conditions .

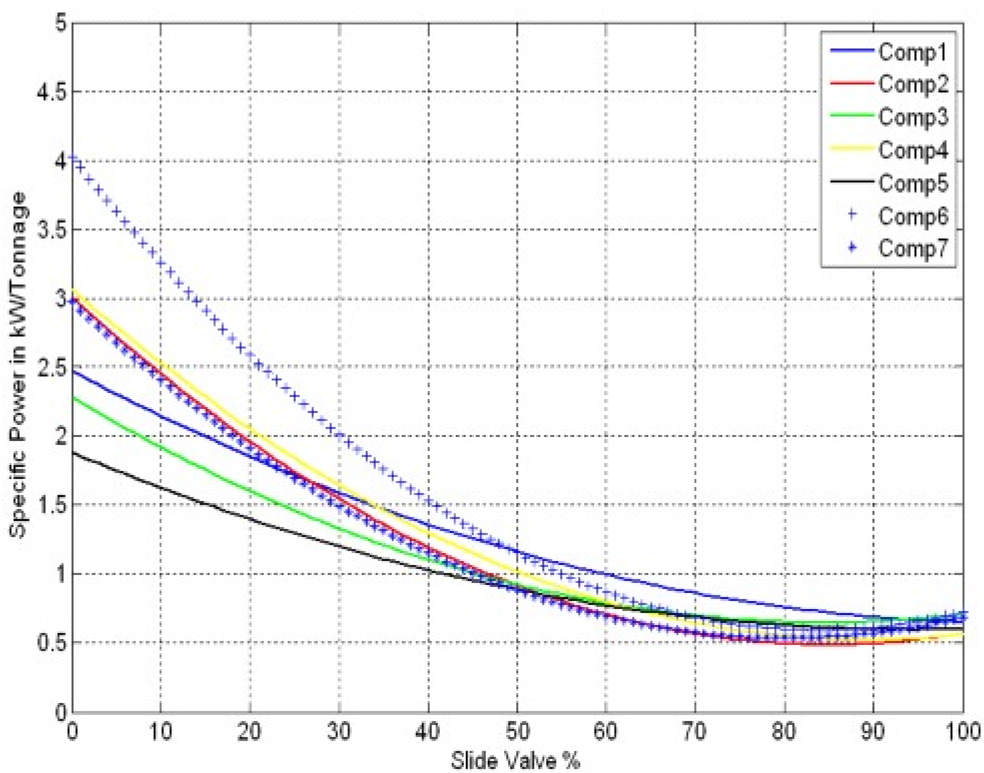


Figure 5.8 Shows the plot between specific Power vs Slide Valve plot at 115 psig pressure [18]

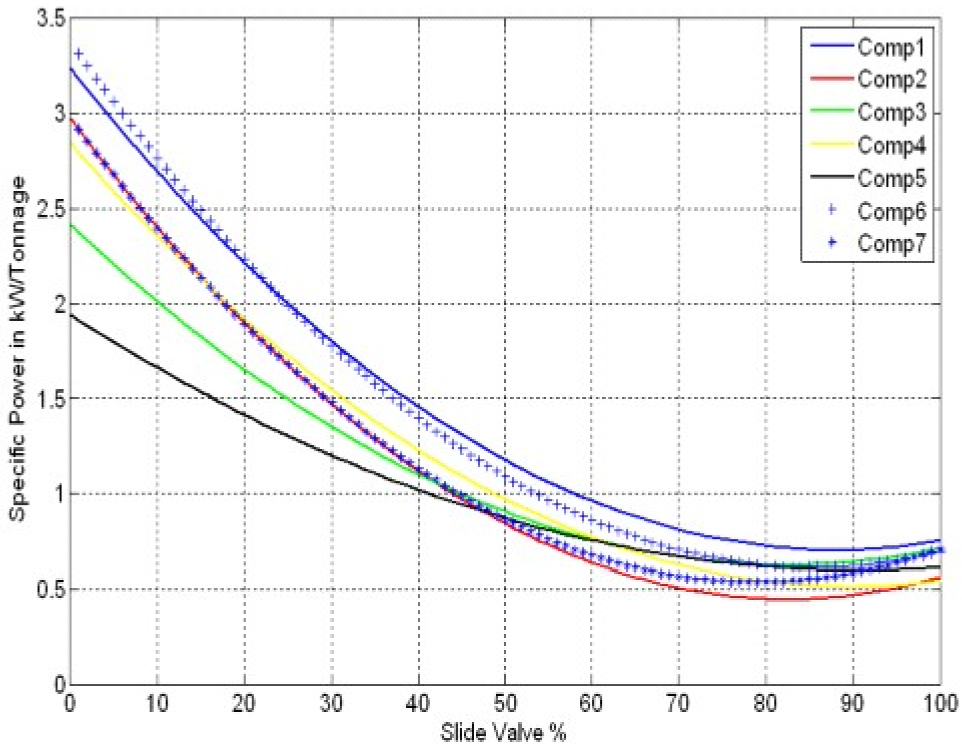


Figure 5.9 shows the plot between specific Power vs Slide Valve plot at 135 psig pressure [18]

For more than 60% slide valve condition , compressor-2 is more efficient . On the other hand , below 60% slide valve condition , compressor-5 is more efficient . For the same discharge pressure condition (140 psig) , compressor-6 draws 32% to 46% more power for the 1 tonnage heat than the compressor-5 which is working below 50% slide valve condition , on the other hand , compressor-7 is inefficient in all the slide valve conditions at the discharge pressure of 140 psig but same compressor is more efficient when the operating slide valve condition is in the range of 50% to 80% . This variation shows , compressor-6 and 1 draws more power for 1 tonnage of refrigeration in all the discharge pressure values and for every slide valve positions .
The poor efficiency of the system under part load condition is mainly due to the two factors (a)friction and (b) volume ratio change .

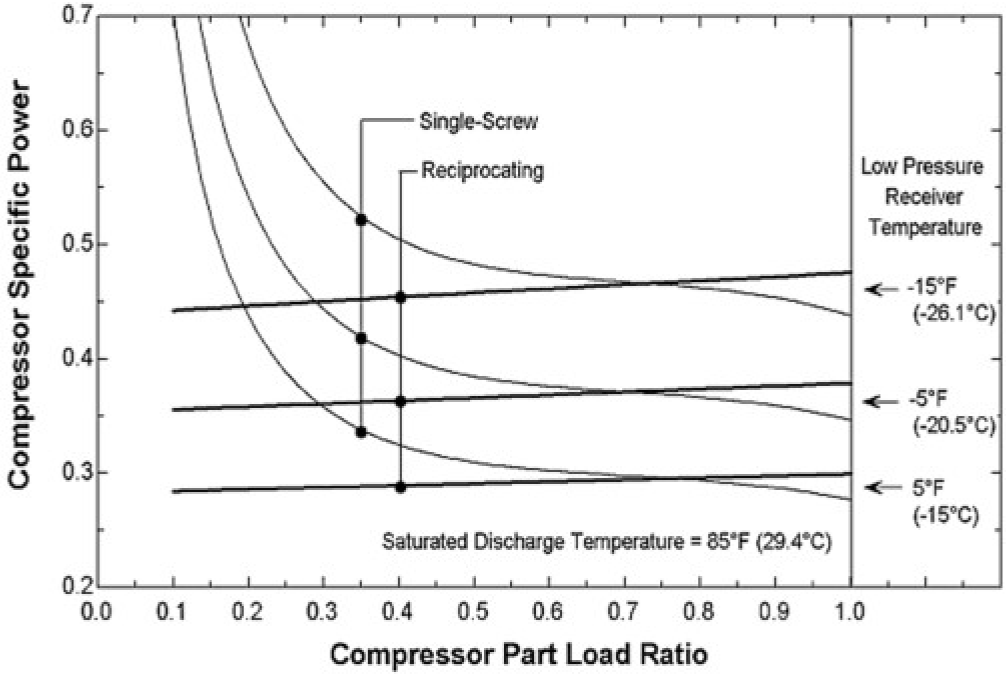


Fig:5.10 shows the plot between the comparison of performance of single screw and reciprocating compressors (suction and discharge-side) refrigerant pressure drop [05]

## 5.2 Design of Experiment

Although, in the Taguchi software there is a different type of Array but I have used L-9 Array that is suitable for four parameter and three level which is used in this thesis. The row denotes the individual experiments and column of the orthogonal array shows the factors which has to studied . This analysis relates with the 4 parameter with each at 3 levels. Generally orthogonal array used to find the effects of related parameter (here 4 parameter) namely the Power, Temperature (suction side and discharge side), Pressure and Slide Valve variation. These 4 factors at the 3 levels that is L9 experimentally studied which is shown in the table-1. Orthogonal array details has been shown with the 4 parameter at 3 levels in the table-5.2.In this analysis , for automatic design , taguchi approach which is a software used for analyze the results and optimize(minimize and maximize) the experimental conditions for setting the control variables.

$$\frac{S}{N_{(Smaller)}} = -10 \log\left(\frac{\sum y_i^2}{n}\right)$$

If smaller is better then used this formula.

Where n is no. of experiment perform for one solution, in this case it is equal to 1 and y is the S/N ratio for the each set L-9 array.

| Parameter | Level | | |
|---|---|---|---|
| | 1 | 2 | 3 |
| Discharge pressure(psig) | 115 | 135 | 140 |
| Slide valve | 78 | 75 | 70 |
| Temp (°C) | -15 | -20.5 | -26.1 |

**Table 5.1 Three parameter and three variables for design of experiment**

| **Exp.no.** | **Discharge pressure(psig)** | **Slide valve** | **Temp(°C)** |
|---|---|---|---|
| **1** | 1 | 1 | 1 |
| **2** | 1 | 2 | 2 |
| **3** | 1 | 3 | 3 |
| **4** | 2 | 1 | 2 |
| **5** | 2 | 2 | 3 |
| **6** | 2 | 3 | 1 |
| **7** | 3 | 1 | 3 |
| **8** | 3 | 2 | 1 |
| **9** | 3 | 3 | 2 |

**Table 5.2 Standard form of L9 Array**

### 5.3 Optimisation technique for minimum specific power

The objectives of the present work was to evaluate the optimum process variable for the minimum specific power consumption.

#### 5.3.1 S/N plot

From the figure it is observed that the optimum level of discharge pressure, slide valve, and temperature are 135(psig), 0.78, -0.26°C. Hence we can conclude that it can give the lowest specific power may be in the range of 0.63 kw/tonnage to 0.65kw/tonnage.

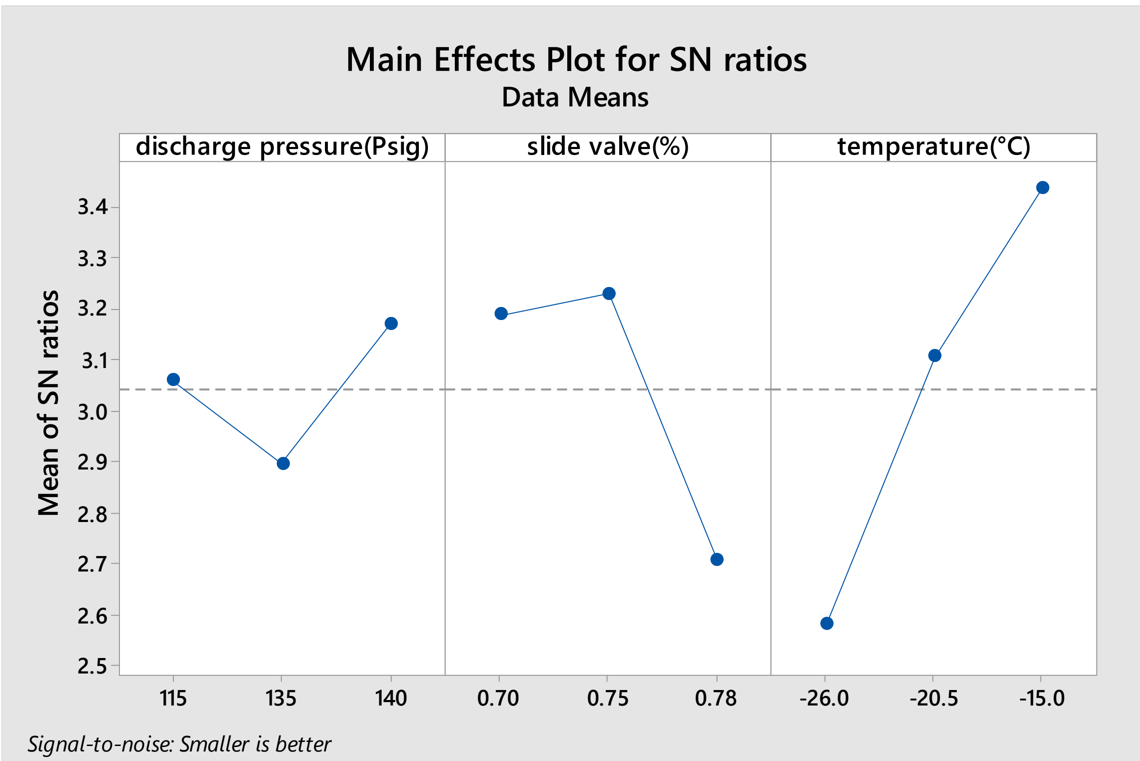


Figure 5.11 Main effects plot of the control parameters

### 5.3.2 Anova table

During cooling of the system, there are various parameter which affect the power consumption of the compressor with a different degree. The parameter which effects the power along with their magnitude are mention in the table. Best result can be obtained of the relative effect by the various factors , obtained by the splitting in variance , which is usually called as the analysis of variance (ANOVA). This result is obtained by the formula given below.

Total sum of squares = Sum of square due to parameter A [(total number of experiments at level A1) x (mA1-m)] + [(total number of experiments at level A2) x (mA2-m)] + [(total number of experiments at level A3) x (mA3-m)].

Now all these sum of squares are listed in Table 5.2, which is known as the ANOVA table.

### 5.3.3 Analysis of Variance

The larger the contribution of a particular parameter to the total sum of squares, the larger the ability is of that factor to affect the S/N ratio. More ever, the lower P- value, the larger will be the factor contribution in the reduction of specific power.

## Factor Information

| Factor | Levels | Values |
|---|---|---|
| discharge pressure(Psig) | 3 | 115, 135, 140 |

## Analysis of Variance

| Source | DF | Adj SS | Adj MS | F-Value | P-Value |
|---|---|---|---|---|---|
| discharge pressure(Psig) | 2 | 0.000622 | 0.000311 | 0.14 | 0.869 |
| Error | 6 | 0.013000 | 0.002167 | | |
| Total | 8 | 0.013622 | | | |

## Factor Information

| Factor | Levels | Values |
|---|---|---|

| | | |
|---|---|---|
| slide valve(%) | 3 | 0.70, 0.75, 0.78 |

## Analysis of Variance

| Source | DF | Adj SS | Adj MS | F-Value | P-Value |
|---|---|---|---|---|---|
| slide valve(%) | 2 | 0.003489 | 0.001744 | 1.03 | 0.412 |
| Error | 6 | 0.010133 | 0.001689 | | |
| Total | 8 | 0.013622 | | | |

## Factor Information

| Factor | Levels | Values |
|---|---|---|
| temperature(°C) | 3 | -26.0, -20.5, -15.0 |

## Analysis of Variance

| Source | DF | Adj SS | Adj MS | F-Value | P-Value |
|---|---|---|---|---|---|
| temperature(°C) | 2 | 0.007489 | 0.003744 | 3.66 | 0.091 |
| Error | 6 | 0.006133 | 0.001022 | | |
| Total | 8 | 0.013622 | | | |

For the temperature P-value is very small; hence it has more contribution in the reduction of specific power .

### 5.3.4 One-way ANOVA: discharge pressure (psi) versus slide valve (%)

Method

| | |
|---|---|
| Null hypothesis | All means are equal |
| Alternative hypothesis | At least one mean is different |
| Significance level | α = 0.05 |

Equal variances were assumed for the analysis.

Factor Information

| Factor | Levels | Values |
|---|---|---|
| slide valve(%) | 3 | 1, 2, 3 |

Analysis of Variance

| Source | DF | Adj SS | AdjMS | F-Value | P-Value |
|---|---|---|---|---|---|
| slide valve(%) | 2 | 0.00000 | 0.00000 | 0.00 | 1.000 |
| Error | 6 | 6.00000 | 1.00000 | | |
| Total | 8 | 6.00000 | | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 1 | 0.00% | 0.00% | 0.00% |

Means

| slide valve(%) | N | Mean | StDev | 95% CI |
|---|---|---|---|---|
| 1 | 3 | 2.000 | 1.000 | (0.587, 3.413) |
| 2 | 3 | 2.000 | 1.000 | (0.587, 3.413) |
| 3 | 3 | 2.000 | 1.000 | (0.587, 3.413) |

Pooled StDev = 1

### 5.3.5 One-way ANOVA: discharge pressure(psi) versus temp(°C)

Method

| | |
|---|---|
| Null hypothesis | All means are equal |
| Alternative hypothesis | At least one mean is different |
| Significance level | $\alpha = 0.05$ |

Equal variances were assumed for the analysis.

Factor Information

| Factor | Levels | Values |
|---|---|---|
| temp(°C) | 3 | 1, 2, 3 |

Analysis of Variance

| Source | DF | Adj SS | AdjMS | F-Value | P-Value |
|---|---|---|---|---|---|
| temp(°C) | 2 | 0.00000 | 0.00000 | 0.00 | 1.000 |
| Error | 6 | 6.00000 | 1.00000 | | |
| Total | 8 | 6.00000 | | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 1 | 0.00% | 0.00% | 0.00% |

Means

| temp(°C) | N | Mean | StDev | 95% CI |
|---|---|---|---|---|
| 1 | 3 | 2.000 | 1.000 | (0.587, 3.413) |
| 2 | 3 | 2.000 | 1.000 | (0.587, 3.413) |
| 3 | 3 | 2.000 | 1.000 | (0.587, 3.413) |

Pooled StDev = 1

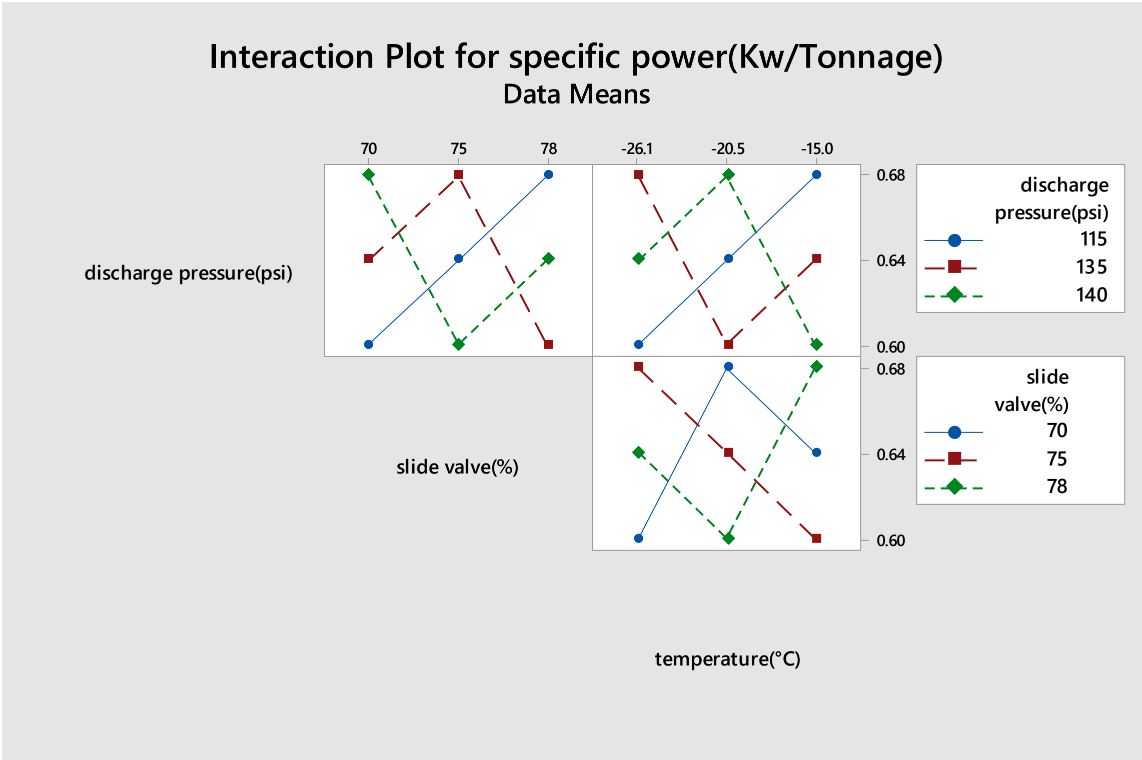


Figure 5.12 Main effects plot between discharge pressure, temperature and slide valve percentage .

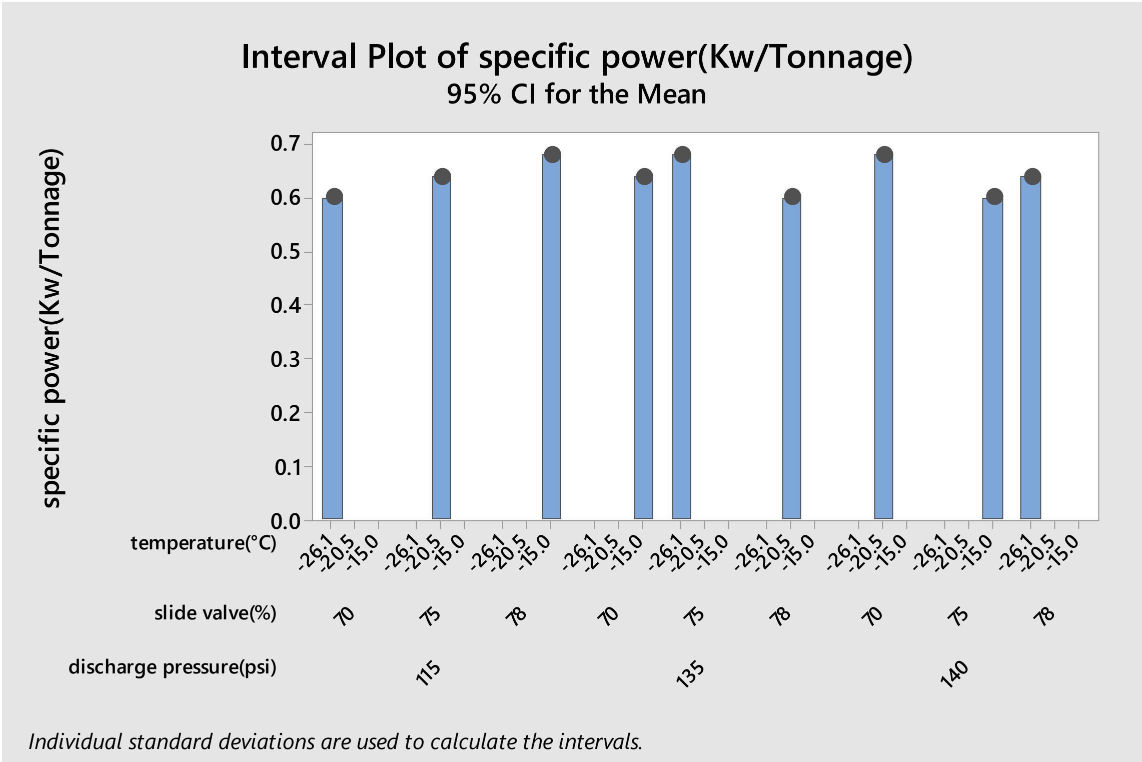


Figure 5.13 specific power(kw/tonnage) variation with temperature , slide valve percentage and temperature

## 5.4 Conclusions

After applying Taguchi optimization technique in the received data from RSEnergy Matrix software which is installed in the organization, the figure is observed that the optimum level of discharge pressure, slide valve, and temperature are 135 (psig), 0.78, -0.26°C. Hence we can conclude that it can give the lowest specific power in the range of 0.63kw/tonnage to 0.67kw/tonnage which means for more efficient refrigeration on 8 hours working, the organization should maintain this much of temperature and pressure condition at their inlet then the minimum consumption of power is required for the last 8 hours working with an efficient manner.

Milk based company mostly uses the refrigeration system (multicompressor refrigeration system) for making the different inlet and outlet temperature. For example

**(1) AMUL –** Many optimization technique used by the industry to their process efficient , some of them are :

- **(a)** Analyze the factors which affects the consumer's purchasing character during product purchasing .
- **(b)** Swot analysis .
- **(c)** Cost optimization by different technique.

AMUL had performed the cost optimization/swot analysis in their capital in terms of expenditure and revenue but this analysis may be require more optimize at the technical level (refrigeration system) which finally make the production cost lower and make process efficient [35] .

**(2). APNI DIARY - Key function/Techniques uses by the company for efficient production.**

- Enhance quality, quantity of the milk with reduction of landed cost.
- Enhance quality of milk products and service with quality systems.
- Enhance maintenance system to reduce downtime and its related cost .
- Enhance accountability status, authority and responsibility.
- Maximize the results by improving team spirit and work culture.
- Enhance revenue of sales via customer oriented innovations
- Enhance power management for reducing consumption and its cost.
- Minimize utilization of the required resources and recovery of material inputs.

From the above techniques which is uses by the **APNI DIARY** company for producing milk and its product in the efficient manner , cost optimization had been done in the capital expenditure/revenue basis but the minimum cost obtained by the company may be more optimize (minimum expenditure) by optimizing the cooling parameter which certainly affects the input (investment) by the company [37].

From the above data analysis and graphs between reciprocating compressor and screw compressor at different loading conditions below the total available capacity(2000kw/tonnage to 2200kw/tonnage) the screw compressor perform good in the full loading condition and reciprocating compressor perform good in the part load conditions .As per the figure 5.12 , screw compressor efficiency decreases as the loading shifted from the full loading to part loading condition . Screw compressor performs well in the base load condition where this compressor may be run at full load in all the working hours of the industry. Same rated compressor may perform different (draws different power) due to many practical reasons. If value of drawn power is more , these difference can be seen clearly .

In case of multi-compressor system, the best way to optimize that select all the compressor separately and then perform the cooling process . This will help to draw the best usage of the compressor whether this compressor best perform in unloading condition , base loading condition or in part loading condition rather than applying specific rules for the compressor in the different working condition .

**References**

1.American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) Handbook, Refrigeration Volume, Georgia, USA, 1990.

2.Lars Finn Sloth Larsen “Model Based Control of Refrigeration Systems”, Central, R&D, Danfoss A/S, DK-6430 Nordborg, Denmark, 2005.

3.D'Antonio M., Moray S., McCowan B. and Epstein G., "Optimization of Industrial Refrigeration Plants: Including a Case Study at Stony field, Farm Yogurt", Energy & Resource Solutions, Inc., New Hampshire, USA, 2005

4.Dugan T., "Compressor Sequencer Problems and Solutions", Compression Engineering Corporation, Oregon, USA.

5.Incropera, F. P. and D. P. DeWitt, "Fundamentals of Heat and Mass Transfer (2nd edition)", John Wiley and Sons, 1985.

6.Sonntag, R. E., C. Borgnakke and G. J. Van Wylen, "Fundamentals of Thermodynamics (5th edn)". John Wiley and Sons, 1998.

7.Reindl, D. T., and Jekel, T. B., Selection of Screw Compressors for Energy Efficient Operation, University of Wisconsin-Madison, Wisconsin, USA, 2005

8.Evapco, PMC-E Condenser Design and Construction Features, Maryland, USA, 2010

9.Mitchell, John W. and Braun, James E., Design, Analysis, and Control of Space Conditioning Equipment and Systems, University of Wisconsin-Madison, Wisconsin, USA, 1998.

10.Schmidt, D., Klein, S., Reindl, D., American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) Research Project RP-962.

11.Moyer R., Facilities Manager, URS Corporation, Pennsylvania, USA, private communication, 2010

12.Seebeck G., Engineer, URS Corporation, Pennsylvania, USA, private communication, 2010

13.Manske, Kyle, "Performance Optimization of Industrial Refrigeration Systems," M.S. Thesis, Mechanical Engineering, Solar Energy Laboratory, University of WisconsinMadison, 1999.

14.Bruce J. T., "Screw Compressors: A Comparison of Applications and Features to Conventional Types of Machines", Toromont Process Systems, Alberta, CanandaWidell K. N., Eikevik T., "Reducing Power Consumption in Multi-Compressor Refrigeration System, Norwegian University of Science and Technology, Dept of Energy and Process Engineering, Trondheim, Norway, 2009.

15.Soplop J., "An Algorithmic Approach to Enterprise Energy Management: Developing an Integrated Energy Solution Utilizing Real-Time Data Collection and Predictive Modeling Capabilities", Rockwell Automation, Wisconsin, USA, 2010

16. Aprea C., Mastrullo R., and Renno C. "Experimental Analysis of the Scroll Compressor Perfomances Varying its Speed", Applied Thermal Engineering, 2006

17. Stoecker W. F., Industrial Refrigeration Handbook, Mc Graw Hill, 1998 .

18. Makati, Tanuj B., "Improving Energy Efficiency of an Industrial Refrigeration System Through Model-Based Sequencing ofCompressors and Condensers" (2011).

19. Ju Y. L., Wang C. and Zhou Y. ,Numerical simulation and experimental verification of the oscillating flow in pulse tube refrigerator, Cryogenics, (1998).

20. Zhu Shaowei, Wu Peiyi and Chen Zhongqi, Double inlet pulse tube refrigerators: an important improvement, Cryogenics, (1990).

21. Gifford, W.E. and Longsworth, R.C. Pulse tube refrigeration, Trans ASME B J Eng Industry (1964).

22. Gifford, W.E. and Longsworth, R.C. Pulse tube refrigeration progress, Advances in cryogenic engineering  (1964).

23. Gifford, W.E. and Longs worth, R.C. Surface heat pumping, Advances in cryogenic engineering (1966).

24. Gifford, W.E. and Kyanka, G.H. Reversible pulse tube refrigerator, Advances in cryogenic engineering (1967).

25. Kittel, P., Lee, J.M., and Timmerhaus, K. D. Steady Secondary Momentum and Enthalpy Streaming in the Pulse Tube refrigerator. Cryocooler (1994).

26. Liang, J., Ravex, A. and Rolland, P., Study on pulse tube refrigeration Part 1: Thermodynamic nonsymmetry effect .Cryogenics, (1996).

27. Liang, J., Ravex, A. and Rolland, P., Study on pulse tube refrigeration Part 2: Theoretical modeling, Cryogenics, (1996).

28. Hoffmann A. and Pan H., Phase shifting in pulse tube refrigerators, Cryogenics, (1999).

29. Yuan, J. and Pfotenhauer, J. M., Thermodynamic analysis of active valve pulse tube refrigerators .Cryogenics, (1999).

30. Xu, M. Y., de Waele, A. T. A. M. and Ju Y. L., A pulse tube refrigerator below 2 K. Cryogenics, (1999).

31. de Boer P.C.T. “Characteristics of the double inlet pulse tube” Cryogenics,  (2003).

32. Mikulin, E.I., Tarasow, A.A. and Shkrebyonock, M.P. Low temperature expansion pulse tube, Advances in cryogenic engineering (1984).

33. Starch , Pulse tube refrigerator- an alternative cryocooler, Cryogenics (1986).

34. Richardson, R. N. Development of a practical pulse tube refrigerator: co-axial designs and the influence of viscosity. Cryogenics (1988).

35. A STUDY ON CONSUMER BUYING PATTERN TOWARDS AMUL MILK Dissertation Submitted to the Padmashree Dr. D.Y. Patil University in partial fulfilment of the requirements for the award of the Degree of MASTERS IN BUSINESS ADMINISTRATION Submitted by: ROHAN.P.NAIK) (Roll No.MBA-CORE-C-011179) Research Guide: Prof Vinod Dumbre Department of Business Management Padmashree Dr. D.Y. Patil University CBD Belapur, Navi Mumbai .

36. Radebaugh, Ray. Development of the pulse tube refrigerator as an efficient and reliable cryocooler, Proc. Institution of Refrigeration (London) , 2000.

37. http://www.apnidairy.com/professional-offer.html .