# DESIGNING AN AIR RECIPROCATING COMPRESSOR WITH CAPACITY 1000LIT/MIN AT 6 BARS

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# DESIGNING AN AIR RECIPROCATING COMPRESSOR WITH CAPACITY 1000LIT/MIN AT 6 BARS

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**YOUSEF ABDALRZAQ MOHMMED HUSN**

**IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR**
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**MECHANICAL ENGINEERING**

**April 2017**

I certify that in my opinion the thesis submitted by Yousef HUSN titled "DESIGNING AN AIR RECIPROCATING COMPRESSOR WITH CAPACITY 1000 LIT/MIN AT 6 BARS" is fully adequate in scope and in quality as a thesis for the degree of Master of Science.

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This thesis is accepted by the examining committee with a unanimous vote in the Department of Mechanical Engineering as a master thesis. April 7, 2017

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The degree of Master of Science by the thesis submitted is approved by the Administrative Board of the Graduate School of Natural and Applied Sciences, Karabük University.

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Head of Graduate School of Natural and Applied Sciences

*"I declare that all the information within this thesis has been gathered and presented in accordance with academic regulations and ethical principles and I have according to the requirements of these regulations and principles cited all those which do not originate in this work as well."*

YOUSEF HUSN

## ABSTRACT

**M. Sc. Thesis**

**DESIGNING AN AIR RECIPROCATING COMPRESSOR WITH CAPACITY 1000L/M AT 6 BARS**



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Compressed air is a basic requirement for many establishments. It drives all pneumatic based systems that can run power machinery and used as a standalone device in many service stations, hospitals, research establishments, transporting and gaseous material require a compressor and air compressors is used almost everywhere.

Compressor technology is historic, but today numerous types of compressors exist, with distinct advantage and disadvantage of each, however, he oldest mechanical compressor, a reciprocating compressor is still widely used and has the distinct the advantage of having power and the ability to provide high flow rates, through proper design and to high pressures by using multiple stages of the compressor. The generic thermodynamic understanding of the compressor explained here, Moreover, the entire system, from piping, receivers, dryers, and the ventilation, all play an important role for housing and running a commercial compressor, this project takes the topic of an exhaustive design of a compressor for a particular flow, the entire theory of each section and the detailed work is carried out to establish, Thermodynamic states, the dimension and size of each equipment, the engineering drawing of each part and an implementable model with each part specified is created to clearly establish the underlying principles of designing a reciprocating compressor. The current report takes a compressor design for delivering 1000 lit/min of compressed air at 6-bar pressure, the entire process is shown iteratively to explain the assumptions and subsequent relaxations to make a more practical system design.

**Key Words** **:** Compressor, thermodynamics, adiabatic, work done, receiver volume, stroke, piston, cylinder.

**Science Code** **:** 914.1.131

## ÖZET

**Yüksek Lisans Tezi**

**6 BAR 1000 L / D KAPASİTELİ PİSTONLU HAVA KOMPRESÖRÜ TASARIMI**

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Basınçlı hava birçok pratik mühendislik uygulaması için temel bir gerekliliktir. Güç makineleri çalıştıran ve birçok servis istasyonunda, hastanelerde, araştırma kuruluşlarında bağımsız bir cihaz olarak kullanılabilen tüm pnömatik esaslı sistemler için kullanılır. İlaveten toz veya granül halindeki malzemelerin pnomatik taşınması için bir kompresör gerekli olup, hava kompresörleri yaygın olarak kullanılır.

Kompresör teknolojisi zamanla gelişmekte olup, günümüzde sayısız kompresör tipi mevcuttur, her birinin avantaj ve dezavantajı vardır, ancak bir pistonlu kompresör kompresör halen yaygın olarak kullanılmaktadır. İstenilen güç ve kapasite ve basınçt için kompresörün çoklu aşamalarını kullanarak uygun tasarımla ve yüksek basınçlı ve yüksek akış oranları sunma kabiliyeti bulunmaktadır. Bu çalışmada kompresörün çalışmasındaki genel termo dinamik ve mekanik prensipler kullanılarak, bir 1000 l/dak kapasiteli bir kompresörün, bütün ekipmanları tasarlanmış ve mühendislik hesapları yapılmıştır. Çalışmada kompresörün kapsamlı bir tasarımı konusunu ele alınarak, her kısmın detaylı çalışma teorisi, termodinamik ilişkileri, mekanik tasarım ve boyutlandırılmaları, mühendislik çizimleri oluşturulmuştur. Çalışma pistonlu

kompresörler alanında çalışmak isteyen mühendisler için konuyu bütün yönleriyle ele alan spesifik bir örnek olarak faydalı olacaktır.

**Anahtar Kelimeler** : Kompresör, termodinamik, adyabatik, yapılan iş, alıcı, hacim, kurs, piston, silindir.

**Bilim Kodu** : 914.1.131

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## CONTENTS

**Page**

**Page**

## LIST OF FIGURES

**Page**

**Page**

## LIST OF TABLES

**Page**

## SYMBOLS AND ABBREVIATIONS INDEX

**SYMBOLS**

$\rho$ : Density of fluid

$\gamma$ : The specific heat ration

n : The polytropic ratio

$\eta$ : Volumetric efficiency

$\mu$ : Viscosity

$C_P$ : Specific heat in a constant pressure process

$\omega$ : The rotational speed of crankshaft

R : Ideal gas constant

**ABBREVIATIONS**

FAD : Free Air Deliver

LP : Lower pressure

HP : High pressure

RMF : Rotating Magnetic Field

# PART 1

## INTRODUCTION

Compressed air is required to pressurize air and subsequently, use it for transporting air and/ or delivering air at high pressure; Botha is needed widely and form the essence of many services and pneumatic systems. In real life depending on the application, an engineer made aware of two things, the pressure at which the air needs to be delivered and the rate of which it estimated to be needed.

The work of the engineer is to design, the entire compressor and the compressor network, to ensure that both the requirement are met, However, with the limited amount of initial data provided, the problem solving is iterative in nature and we need to stick to some assumptions or thumb rules to be able to satisfy, This is because there are numerous designs that can fulfill the requirement and each is correct as it addresses the needs.

Generally, some information regarding the quality of air required if provided, aid in assessing some extra equipment that might be required to meet the quality standard (in terms of moisture, dust or other entrainments). However, as no such data it provided here, it again becomes an engineering decision to make proper and relevant assumptions such that the systems can be widely applicable.

Hence, to sum up, the problem is three-fold:

- Design a preliminary thermodynamic analysis of the compressor to estimate the stages, number of equipment, speeds etc.

- The actual design of the compressor by building upon the preliminary design, in order to make the design more practical and closer to one by dissolving some assumptions.
- Mechanically design each component, part of the compressor assembly after determining the equipment to be used.

## 1.1. THESIS AIM AND PURPOSE

The widespread dependence of many research units, hospitals, service stations on the supply of proper compressed air is essential to their working. Some of the specific areas that employ compressed air roughly mentioned as.

- Oxygen, Nitrogen at hospitals, which require high purity compressed air whose need can be dynamic.
- Assembly lines, workshops where compressed air commonly used to blow away metal chips and dust particles.
- Service stations and repair stations where most of the power tools (drill, fasteners, nail guns) pneumatically operated.
- Argon, nitrogen, oxygen are common utilities required in research facilities worldwide. Each posing a unique challenge for the compressor design.
- Filling of numerous pressure cylinders requires such compressors widely.
- As it has seen, the compressor need is largely varying across various fields and the design of compressor is a dynamic process that caters to the widespread areas in which it is required. Proper compressor design serves this purpose. In this project, we aim to design one such compressor that specified to require compressed air at 6-bars pressure with a flow rate of 1000 lit/min.

With no additional information, the idea of the project is to design a compressor, such that, it is largely free of moisture, dust, oil, grease and other particulate matters. Furthermore, care needs to be took to ensure:

- Proper control over the flow rate.
- Conservative design such that the pressure at the point of delivery does not fall below 6bars.
- Proper storage vessel design according to engineering methods such that, the flow of 1000 lit/min met.
- A dryer design to bring the compressed air to the desired moisture level.
- Elaborate engineering diagram that uses exact dimensions and parameters to illustrate the real-time installation of the compressor system.

The theory behind each portion extensively reported in order to make sure that the report not only serves as an example document but also use to apply to various compressor design methods.

## PART 2

## LITERATURE REVIEW

The invention of air compressor was a fundamental step in the field of engineering. Today, compressors used extensively in areas such as marine, power production, aerospace among another field. While designing an air compressor, numerous experimental and systemic processes employed, scientific optimization techniques invested in establishing design constraints in the compressor phase, these machines extensively applied in manufacturing industries. There are various types of compressors; thus, the right choice needed to accomplish the requirement of a particular industry, the steadiness of the air inside the compressor is prominent in the chamber with rotating blades or reciprocating pistons. The rotary device applied for high volume stream where the required pressure is considerable. When the high release pressure is required, a reciprocating compressor is used [1].

The human lung is actually the earliest air compressor. Since the human lungs can exhale air, people first used their breath when stoking the fire. The mechanism of using human lungs to stoke fire died when the invention of metals began, the demand for stronger air compressors facilitated more innovations. Despite the fact that healthy human lungs can generate about 0.08 bar or air pressure, the air contains carbon dioxide, which is not helpful in sustaining fires. Around 1500 B.C. an air compressor referred as Bellow invented and was a hand-held later foot controlled – flexible that generated a blast of air enough to achieve high temperatures fires [2].

Another invention referred as water organ invented in 300 B.C; the father of water organ, Ctesibius from Alexandria, Egypt later invented similar machines positive displacement cylinder and piston that were used to pump water.

Ctesibius water organ consisted of a water pump, a compartment to accommodate water's air and an array of pipes of varying diameters and sizes on top, and valves. Working with the water organ involved driving water into the water/air chamber to displace the air inside, the displaced air converted into a compressed shape. Another engineer of that age, Hero from Alexandria also described the theory of expanding steam to convert steam power to shaft power used this idea to expand this innovation [3].

John Smeaton (1762), a professional engineer invented the water wheel powered blowing cylinder that came to replace the bellows. John Wilkinson (1776) who invented the blasting machine improved Smeaton's device, The Wilkinson's blasting machine later became the prototype for the later air compressors [4].

John Dumball (1808) came up with an idea of the multistage axial compressor. In his proposal, John believed that rotating blades deprived of motionless airfoils would turn the motion into a successive phase. Unfortunately, Dumball's proposal was not impractical; however, the idea was valid.

Philander and Francis Roots (the 1850s) came up with a famous Roots blower, the pair discovered the blower as they were looking for a spare part of a broken water wheel. The Roots Blower comprised of numerous figure-eight impeller that was revolving in opposite directions , The Roots brothers completed their design which was tested by Europeans and was credited as practical [5].

Dr Franz Stolze (1872), combined the efforts of John Dumball and John Barber to enhance the axial compressor that was operated by an axial turbine, Due to lack of money to implement his project, Stolze's was not successful. Stolze opted to construct a multistage axial flow compressor, an incinerator compartment, a multistage axial turbine and a regenerator that used gasses to heat the compressor and release the compression gas [6].

In Apr 2016 from International Research Journal of Engineering and Technology, talking about reciprocating air compressor is the most widely used on a large scale type of compressor found in many industrial applications, a crucial instrument in gas pipelines, petrochemical plants, oil refineries, lines, etc. Due to the requirements of a high-pressure ratio, exchange is commonly used air compressor in the locomotives, after a period of unexpected failures of life of the internal components for a variety of reasons occurs, which affects the performance of the operating system? This paper presents a case study for reciprocating air compressor locomotive shed light on the problems associated with the diagnosis and effective strategies and solutions backed by proper maintenance for the repair and restoration arising because of repeated failures of parts.

It is often necessary to put permits recommended due to the different parts of the compressor. Based on measuring the dimensions of the parts of the compressor choice of reform and it becomes easy to replace and it is better economic terms [7].

Through it has been selected all previous studies of this thesis, which was adopted on designing an air reciprocating compressor with capacity 1000 lit/min at 6 bars.

# PART 3

# COMPRESSOR THEORY

## 3.1. AIR PREPARATION

A pneumatic control system, according to with the capacity of the system, must be available in sufficient quantity and pressure, works with a supply of compressed air. Operational reliability and service life of the pneumatic system is largely dependent on the production of compressed air not only the impurities of the compressed air, but others such as rust, scale, dust, and the liquid component of the deposits in the air can cause a lot of damage to the pneumatic system as condensate. These contaminants affect the life of the functions and services of the adverse effects pneumatic equipment, and a sliding surface and the elements to accelerate the wear of the seal. On and off the compressor switching of the results of, resulting in pressure fluctuations have an adverse effect on the function of the system. In order to eliminate these effects, one should prepare for the compressed air giving it high importance [11].

Stage 1 Intake air filter removes the particles, which are likely to damage the air compressor.

- Location**:** For better inhalation of the compressor, that is to provide a better quality of air, can be either outdoor or indoor. The relative rise of the compressor to the sea level is needed to determine the pressure and density of the intake air. The quality of the air, the temperature, humidity determined by the cleanliness. You need to make sure that air in this stage is free of moisture and contamination.

- The intake temperature: Density of the air change in inverse proportion to the temperature. Lowering of about -20 degree Centigrade obtained for an air temperature deliveries intake increase for 1%.

- The intake pipe material: the inside of the suction pipe is smooth; it must be not subject to rust and oxidation. The intake air piping material to flake off the rust, cause damage by entering the compressor. Material that is acceptable, will include plastic, Cooper, stainless steel, aluminum or galvanized steel. With a metal pipe, a mechanical coupling is used. Weld bead, can enter the compressor as it gets free and because there is a possibility of damage, the pipe must avoid any welded joints in it.

- Critical pipe length: the resonance of the intake pipe will be prevented by avoiding the particular length of a pipe for the air compressor, these called important pipe length and it has been a function of the speed of the Compressor at the temperature and revolutions per minute of air. The length of the critical pipe needs to be make sure by the equipment manufacturer.

- Intake air filter: selection of the filter type is dependent on whether or not the air compressor lubricated, and on the ambient air quality.

- Viscosity collision filter, it has around 85% to 90% of the efficiency of the larger particle size than 10 microns. This type of filter allowed for lubricating reciprocating compressors operating in the normal state.

- Oil bath filter has a larger 10-micron size efficiency of 96% to 98% of the particles. This type of filter is more expensive, but no longer recommended by the manufacturer of the compressor. It can, however, be taken into account for the operation lubrication reciprocating compressor in heavy-duty conditions.

- The Dry filter has 99% of the efficiency of the particles larger than 10 microns. Because of this, these filters are the favored choice for the rotation and the reciprocating compressor; they applied every time you must use a non-lubricated compressor and every time you must maintain air to be free of oil.

- Two-stage dry filter, in order to provide 99% of the efficiency of particles larger than 0.3 microns used in a centrifugal separation device.

- A means of monitoring and all types of filter, the pressure drop of the air passing through the elements, shows the contamination of elements, it must provide.

Stage 2 at this stage, air compressed by the compressor. It does not mean that this is a complete analysis of every type of air compression system possible to design. Instead, it in most cases will focus on what the industry likes from what and the thermodynamic analysis in those systems are a very detailed analysis of positive displacement type and concise analysis of dynamic compressor.

Stage 3 in this stage, the outlet temperature of the compressor reduced, usually larger than 100 microns solid contaminants removed and the air dried to reduce its moisture, the units used in the first step of cooler dryer and in the main line filter.

Stage 4 moisture and fine dust at this stage will removed. In order to support the lubrication it adjusted to suit the requirements of individual machines at this stage pressure, it introduces a fine of oil mist in the compressed air, and the units used in the treatment of secondary air, the filter (called FRL or service units) regulator and lubricator [14].

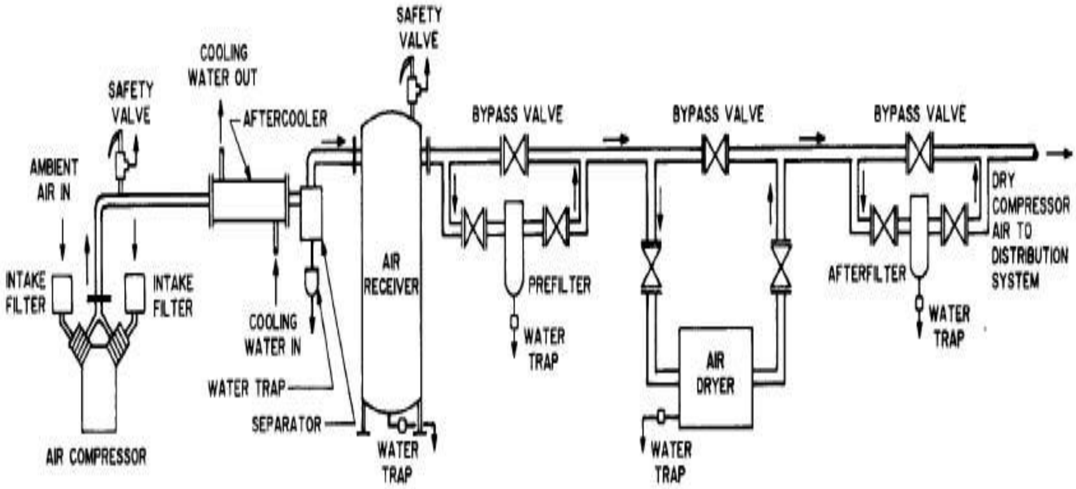



Figure 3.1. Preparation stages of air [13].

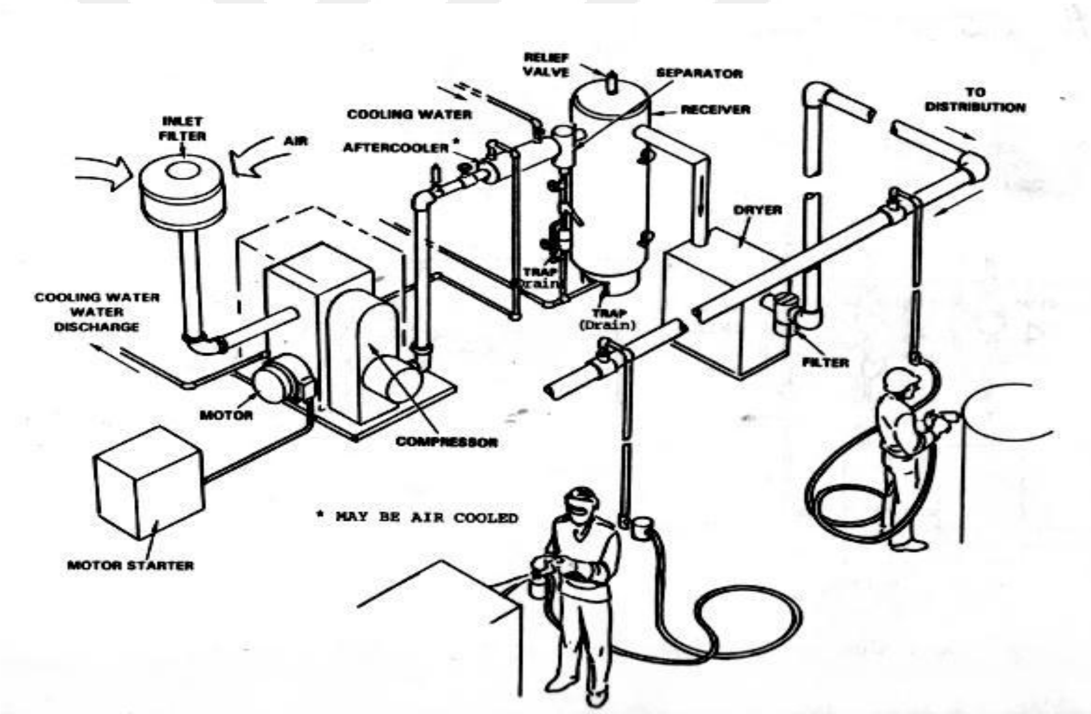



Figure 3.2. An industrial compressed air system diagram [13].

## 3.2. AIR COMPRESSOR

### 3.2.1. History And Its Classification

One of the oldest recorded use of compresses gas (air) found dating back to 3$^{rd}$ century BC and is a water organ, the credit for designing the "water organ" goes to Ctesibius of Alexandria. Ctesibius has also developed a positive displacement cylinder, the piston to move the water organ made up from the connecting tube and the water pump and valve plus various pipe string (organ pipes) and a partial air, and water filled chamber. Air will compress the water in the pump to the water/air chamber. This concept furthered with the improvement from Alexandrian Hero (also note in order to explain the principle of expanding the steam to convert steam power the power to the shaft).

When you try to find the water wheel of an alternative woolen mill, the first mechanical compression of the family of bellows operated by hand, appeared in 1500 BC. In the 1850s, Philander and Francis Roots devised Roots blower. Its design had been a pair of figure number of the impeller to be rotate in the opposite direction. Some of the Europeans had been at the same time, trying this design, but with Roots brothers completing the design; they placed it in the large-scale production [8].

John Dumball, in 1808 has assumed a multi-stage axial flow compressor. Unfortunately his ideas were made from the moving blade without a fixed airfoils required in order to turn the flow to each subsequent stage, Dr. Franz Stolze, in 1872 formed a combination of the idea of John Barber, and John Dumball in order to develop a first axial flow compressor which is driven by the axial direction of the turbine, the design of Dr. Stolze, consisting of a multi-stage axial compressor, a single combustion chamber, the regenerator using the exhaust gas to heat compressor discharge gas, multi-stage, axial-flow turbine, and heat was not constructed until 1900 for lack of funds.

Compressors create a higher desired pressure level in the air or other gasses with a low inlet pressure (usually atmospheric pressure). Compressors, by reducing its volume, increases the pressure of the air work necessary for the increase of the pressure of the air received from the engine driving the compressor. In general, the electric motor, the internal combustion engine or steam engine, will used as a prime mover such as a turbine. The compressor is similar to the fan or blower but differs in terms of the pressure ratio. Fan is said to have up to 1.1 pressure ratio, while compressors have a pressure ratio above 4, and blower has a pressure ratio between 1.1 and 4 [15].

The compressor can be classified in the following a variety of ways.

- Based on the operating principle: You can classify compressors based on the principle of operation.

    - Compressor with a displacement, which is positive.
    - Compressor with a displacement, which is not positive.

The compression of the positive displacement compressor, which realized by preventing the fluid by solid boundaries flowing in the direction of the displacement and the backpressure gradient of the solid boundary of. Due to the displacement of the solid wall, these will be able to provide a very large pressure ratio. Positive displacement compressor further classified based on the type of mechanism that is used for compression. These can be.

    - A reciprocating positive displacement compressors
    - A rotary type positive displacement compressor.

In general, the piston displacement in a reciprocating cylinder causes the compressor pressure increase using a piston-cylinder mechanism. With a reciprocating compressor, it is possible to provide a large pressure ratio, but a lot of processing power is limited. The reciprocating compressor can be either a single acting compressor or a double-acting compressor. In a single acting, the compressor has a one-delivery stroke per rotation and there is two-delivery stroke per rotation of the double-acting crankshaft. Rotary compressor using a positive displacement, the boundary fluid, thereby have a rotating portion, which causes positive displacement compression. This type of rotary compressors are available in a name, such as the following [15].

- Roots blower.
- Vane compressor.

The above type rotary compressor can run faster, than reciprocating motion of the positive type displacement compressors can handle a large mass flow rate.

In addition, non-positive displacement compressor, or the steady flow compressor, realize the pressure rise by using the dynamic action of the solid boundary. Here, the fluid volume is not definite and post reduction in volume does not take place just like in constant volume and positive displacement compressors. Non-positive-displacement compressors, dependent on the type of flow of the compressor might be axial or it can be centrifugal.

- Based on the number of stages: the compressor also classified based on the number of stages. In general, the number of stages depends on the maximum discharged pressure. The compressor may be having a single stage or multiple stages. Normally the maximum compression ratio of 5 is achieved by the single-stage compressor and consequently, for the compression ratio of 5 or more of the multi-stage compressor is used.

Different types of compressor, with the value of the general maximum discharge pressure available, are:

- The single-stage compressors for discharge pressure till 5 bar.
- The two-stage compressor, for the discharge pressure of 5 to 35 bar.
- The three-stage compressor, during the ejection pressure of 35-85 bar.
- Four-stage compressor for the discharge pressure, more than 85 bar.

- Based on the capacity of the compressor: the compressor also classified according to the air to deliver per volume and per unit time of the compressor. General values of the capacitance of the different compressors given as follows.

    - Low capacity compressor has a 0.15 $m^3$ / s or less of the air delivery capacity Medium-capacity compressor has the air delivery capacity between 5 m3 per second and 0.15 $m^3$ per second.

    - High-capacity compressor has a 5 $m^3/$ s or more, having an air supply capacity.

- Based on the maximum pressure**:** Based on the maximum pressure available from the compressor, it classified as low pressure, medium pressure, the high pressure, and ultra-high pressure compressor. General values for the maximum pressure that has developed for the different compressor is as below.

    - Low-pressure compressor, which has the highest pressure up to 1 bar
    - The intermediate pressure compressor, which has the highest pressure from one bar to 8 bars.
    - A high-pressure compressor, which has the highest pressure of 8-10 bars
    - Ultra-high pressure compressor, which has the highest pressure than 10 bars.

A detailed description shown in Figure 3.3. Air compressor is generally, any of the positive displacement portion and the reciprocating piston type or rotary screw or rotary vane type. These three types described in detail. It is a commonly used piston type for the industrial compressor.

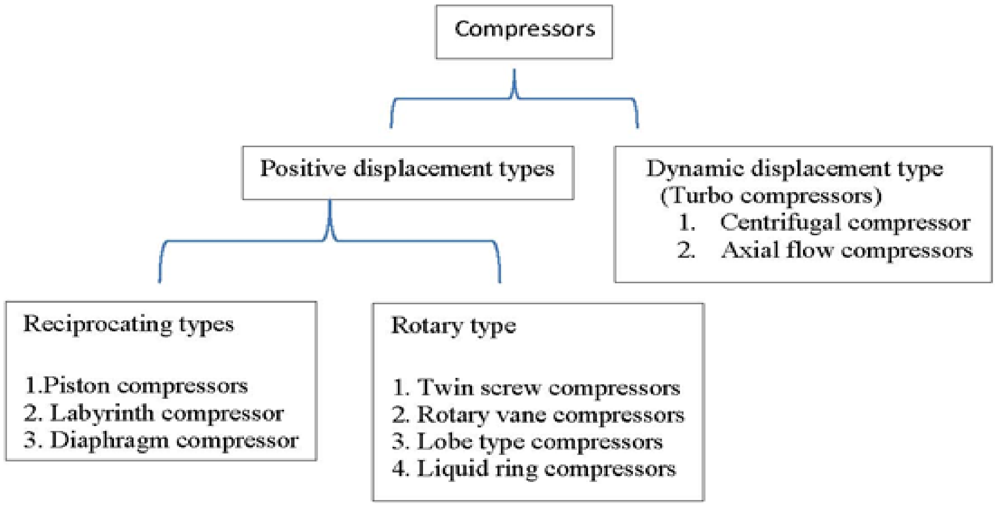



Figure 3.3. Compressor types and classification [13].

### 3.2.2. Reciprocating Compressors

The reciprocating compressor hugely used for the air system in the industrial plant. The two major types are double acting, and the single-action, which is a two-stage or one-stage compressor, the single-acting cylinder is set in one direction for the power stroke and does the compression on one side of the piston. Two-stage compression is a two separate compression cycle in series or stepwise to reach final output pressure.

As double acting compressor, piston move in both direction and that provides a compression stroke, this in turn is achieved by placing the cross head to a crank arm connected to a double acting piston with a piston rod, distance piece connects the cylinder to the crankcase, these have sealed to prevent contamination of the lubricating oil in the crankshaft and the air, but it also evacuated to prevent the build-up pressure [12].

#### 3.2.2.1. Piston Compressors

Piston type compressor is the oldest in the air industries owing to its flexibility, high-pressure performance ability to dissipate compression rapidly and oil-free heat, they built for services that are either fixed or mobile.

- Single Cylinder Compressor, Piston compressors and single or double acting, is available as free of oil or in oil lubrication and in different configurations except for a small compressor in a vertical cylinder. V configuration is the most common small compressor. In the case of double-acting, large compressors with the vertical low-pressure cylinder and horizontal high-pressure cylinder, provide a huge profit and is thus the most common design, The structure and operation of the reciprocating compressor of piston type are very similar to that of the internal combustion engine.

    - Structure Piston compressors, piston rings, inlet and outlet of the spring-loaded valve, connecting rod, crankshaft and the bearing, the cylinder, the cylinder head, and the piston is what is consisted of the Piston Compressor.

    - Operation of Compression is achieved by the reciprocating motion of the piston in the cylinder, the movement after filling the cylinder compresses air Connecting rod then converts rotational movement of crankshaft into reciprocating motion of the piston in the cylinder, depending on the application, rotation crank (or eccentric) has been driven at a constant speed by a suitable prime mover (usually an electric motor).

Inlet Stroke: It begins with a piston (or position to provide a minimum clearance volume) in the top dead center of suction or intake stroke. During the downward stroke, the piston movements will reduce the pressure in the cylinder at atmospheric pressure. Inlet valve opens against the pressure of the spring it allows the air to flow into the cylinder. Piston air is drawn into the cylinder until it reaches the maximum volume position in stroke (bottom dead center) while the discharge valve is closed.

Outlet stroke the piston of the compression stroke move in the opposite direction to reduce the amount of air in the movement, (at the top dead center from the bottom dead center). As the piston starts moving upwards, the inlet valve be closed, the pressure begins to continuously increase until above the pressure of the discharge side pressure is connected to the receiver in the cylinder, Then outlet valve released and air reaches the upper part of the receiver during the movement of the piston [13].

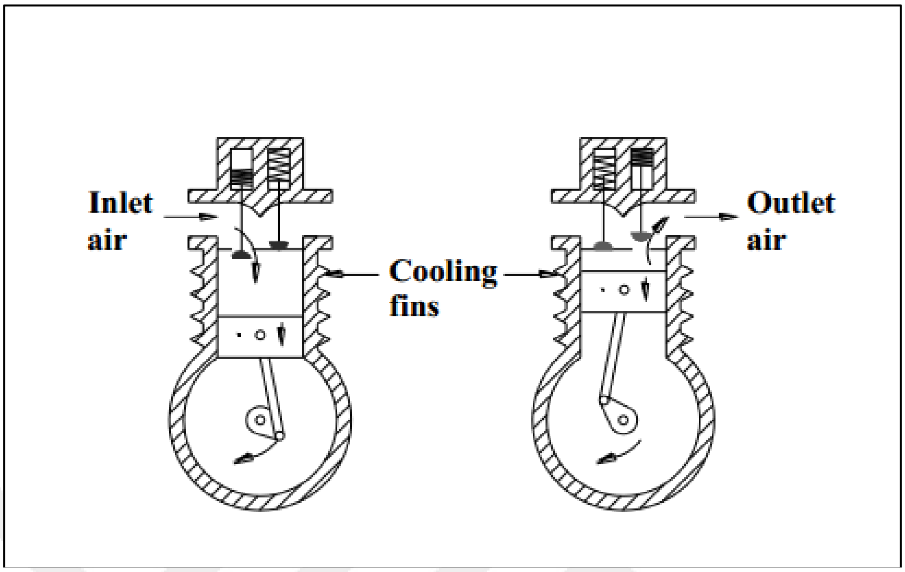



Figure 3.4. Compressor with single stroke [13].

- Analysis of the Air Compressor of Single-Cylinder Single-Stage. Representative indicator diagram for the reciprocating compressor in three different types of compression shown in Figure 3.5. Clearance volume will be ignore.

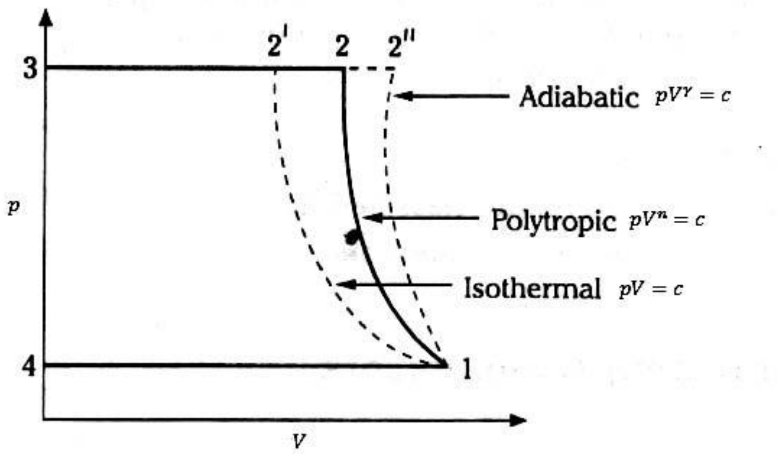



Figure 3.5. Compression diagram [13].

Constant pressure line reflects the suction stroke, Air adiabatically compressed followed by thrown out at constant pressure, and the enclosed area with outermost dotted lines reflects the work, If the compression performed in isothermal then the innermost curve played which has a smaller gradient than the processes both isentropic and polytrophic, The area with the innermost curve and the vertical axis reflects that the task is significantly less than above task due to an adiabatic form of compression. If compression is according to the isothermal process, the compressor will have a higher efficiency, The compressor needs to slowly run, so in order to achieve the isothermal process, which is virtually impossible. In fact, the compressor runs at high speed leading to the polytropic process, Cold-water spray and the multi-stage compression used to reach the isothermal compression while a compressor runs at high speed.

- Work of Single Stage Compressor Ignored Clearance Figure 3.6 reflects the air PV in the air compressor cylinder. Constant pressure line reflects the suction stroke, the air then goes via adiabatic compression, and then pushed out from the cylinder at a constant pressure. The enclosed area reflects the work

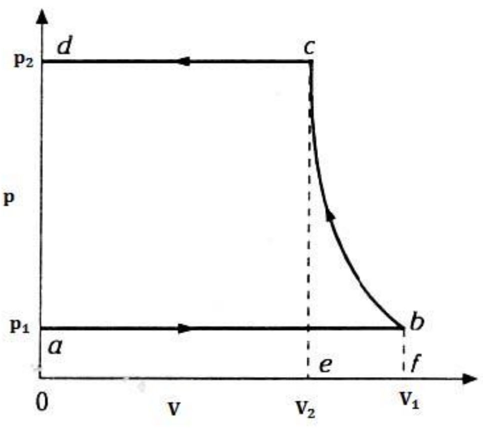



Figure 3.6. Pressure-volume diagram [13].

Three types of compression process can performed in the compressor, which is as follows:

Isothermal compression. Air Compression has done at a constant temperature

$$W = P_1 V_1 \, ln\left(\frac{P_2}{P_1}\right) \tag{3.1}$$

Where

$W =$ The compression work performed

$P_1 =$ The pressure at the inlet and

$P_2 =$ The pressure at the outlet.

$V_1 =$ The initial volume

Adiabatic compression. At the time of expansion or compression, it does not have the thermal energy flow in or out of gas.

$$W = \frac{\gamma}{\gamma - 1}(P_2V_2 - P_1V_1) \tag{3.2}$$

$$W = \frac{\gamma P_1 V_1}{\gamma - 1}[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \tag{3.3}$$

γ Is the specific heat ratio = 1.4 in the case of air

$$C_P = 1.005 \left(\frac{K_j}{K_g}\right) k, C_v = 0.718 \left(\frac{K_j}{K_g}\right) k$$

Polytropic compression: This is located between the isothermal and adiabatic processes. The air pressure compression and expansion are neither adiabatic nor fast nor is it isothermal or slow. It is polytrophic.

$$W = \frac{n}{n-1}(P_2 V_2 - P_1 V_1) \tag{3.4}$$

$$W = \frac{n\, P_1\, V_1}{n-1} \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \tag{3.5}$$

n is the polytropic ratio = 1.4 in the case of air

The efficiency of compression work according to the isothermal process in a reciprocating compressor is minimal. Isothermal efficiency is the ratio of the isothermal work and the actual work.

$$\eta\, isothermal \; = \frac{Isothermal\ Work}{Actual\ Work}$$

- Single Stage Work done in compressor considering Clearance, The actual design of the compressor, some distance between the cylinder and the piston is essential to prevent the piston to blown to the cylinder of the crown, Figure 3.7 shows the PV diagram of a single-stage compressor having a clearance.

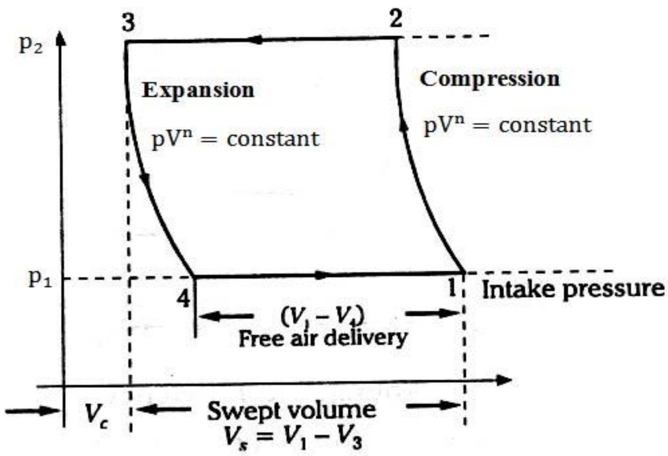



Figure 3.7. PV Clearance diagram [13].

Thus while the delivery stroke; a certain portion of compressed air corresponding to the gap amount $V_3$ at a specified pressure of $P_2$ left in the cylinder. During the next intake, stroke air deployed back to the initial pressure and volume. This way, prior to the fresh air that enters the cylinder some amount of air by volume will already be present in the cylinder. This is the inhalation stroke volume during the intake stroke is given by

$$V_1 - V_4$$

The work was done in the air released is not affected by volume clearance as it is to be recovered from the theoretical expansion.

$$W = \frac{n\,P_1\,(V_1 - V_4)}{n-1}[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \tag{3.6}$$

- Volumetric efficiency. The amount of air that can be sucked by the free air delivery This compressor inhalation or inlet conditions of the compressor at 1 atmospheric, located at 20.100% dry air compressor motor is running at 100% of the rated value you have. FAD is an important purchasing parameter; it processed, and then measures the ability of the compressor in terms of airflow, FAD used to compare the different compressor, Induced by delivery volume will be different, but the mass that has been the induction on a cycle-by-cycle basis, in accordance with the mass of the law of conservation, it is important to note must be equal to the delivery mass of each cycle.

$$\eta\,volumetric = \frac{P_1(T_a)}{P_a\,T_1}\;1 + k - k\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \tag{3.7}$$

$$Where\ k = \left(\frac{V_c}{V_s}\right) = Ratio\ of\ Clearance$$

A implies the free air or ambient conditions index and 1 imply the state before compression.

- Compressor Air Capacity Rating Analysis Air compressor was defined as the air in the actual atmospheric conditions, it is evaluated in terms of $m^3$/ min of free air. Standard atmospheric conditions of 101000 Pa (absolute) and 20-degree centigrade are used.

We still use the British system for the evaluation of air compressors in the industry. Air compressor defined as the air in the actual atmospheric conditions; it evaluated in terms of CFM in free air and CFM free air, when the inlet air in the compressor is at the standard atmospheric conditions of 17.7 (PSIA) 1 bar and 68 degrees Fahrenheit, have been referred to as the SCFM.

$Q_1$ And $Q_2$ = Air Flow Rate by Volume at the inlet and outlet of Compressor($m^3$/min).
$P_1$ and $P_2$ = Absolute Air Temperature at the inlet and outlet of CompressorkPa(abs).
$T_1$ and $T_2$ =Absolute Air Temperature at the inlet and outlet of Compressor(k).

According to regular practice.

$$Q_1 = Q_2 \left(\frac{p2}{p1}\right)\left(\frac{T1}{T2}\right) \tag{3.8}$$

G. Piston Compressor with Multi-Stage, According to the general gas principle, there is an increase in the temperature as the pressure is increased, For example, if the outlet pressure of the compressor is 5 bar in the single acting compressor, the temperature of the compressor air can rise to over 200-degree centigrade required to drive the rise of the compressor, Therefore for high-pressure usage single-stage compressor is not very popular, Better cooling between stages effectively reduce input power requirements when higher pressure is needed increasing efficiency and thus multi-stage compressor is used.

With a single-stage machine air is compressed to a pressure of about 6 bars, and in exceptional cases up to 10 bars, The two-stage machine typically compresses air pressure up to 15 bars and discharge pressure in the range of 250 bars can be achieved with a reciprocating compressor involving 3 or 4 stages.

In the single-stage compressor leads to the compression of the entire air with a single piston stroke. In the multi-stage compressor, compression takes the step by step. Maximum compression efficiency achieved by using the inter-stage cooler, in a two-stage compressor initial compression has done in the low-pressure cylinder. Air from this stage passes through the intercooler in order to reduce the temperature, then the cooled air compressed in a high-pressure cylinder.

- Working Figure. 3.8 shows the air compressor reciprocating in two stages (in-line type), When the crankshaft-attached prime mover rotates, the crank rotation occurs and the piston will reciprocate in the first stage, This will suck the air through the suction filter and the inlet valve, To some extent the compressed air passes from the left cylinder into the right cylinder via an intercooler, The compression ratio of the first stage is dependent on the degree of cooling required

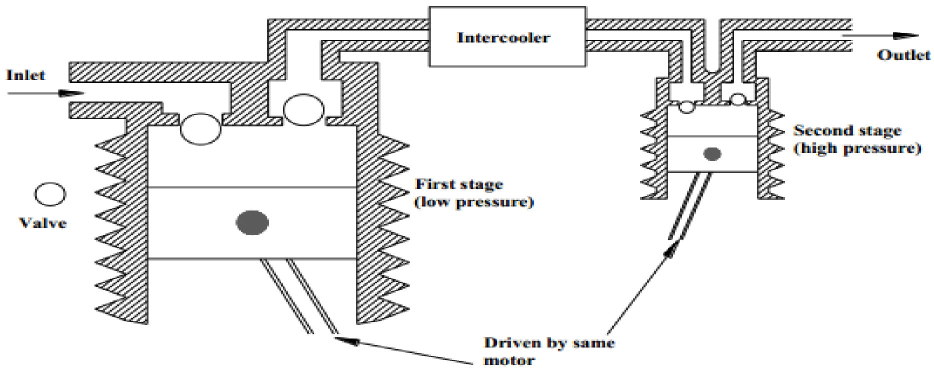



Figure 3.8. The intercooler with multi-stage piston air compressor [16].

Figure. 1.12, enumerates the different components of the three-stage (V-type) round trip air compressor and the receiver air tank, The pressure switch connected to the electric motor upon reaching a desired pressure in the air tank, it stops the motor and compressor, The safety valve opens when the pressure in the air tank exceeds the safety pressure level.

### 3.2.2.2. The Merits Of The Piston Type Compressor

- It is widely available in different types of capacity and pressures.
- It is possible to achieve high pressure up to 250 bars and air flow per volume using multiple staged compressors
- The multi-stage compressor achieves good mechanical balance by employing appropriate cylinder adjustments.
- It has a very level of efficiency in comparison to other compressors available.

### 3.2.2.3. The Demerits Of The Piston Type Compressor

- Reciprocating piston compressor raises the inertial force, which leads to the rattling of the machine, Hence, a strong frame and foundation are essential to hold it.
- It leads to a flow of the air, which is pulsating in nature and thus damping Chambers, or receiver tank used to combat the same.
- They used for smaller volumes of air and that too at pressure levels that are high.

### 3.2.3. Analysis Of Air Compressor Multiple Single Stages

The ratio of clearance, pressure, and expansion index together determine the efficiency of volume in a reciprocating compressor, The efficiency of volume with a fixed level of clearance decreases as with the increase in the ratio of pressure and it can even decrease to a level where the efficiency of volume is 0 as it can be viewed with the below given diagram.

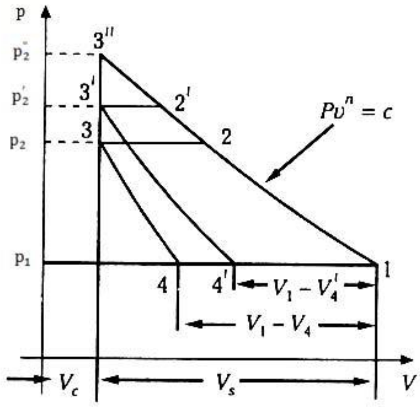



Figure 3.9. Compression at multiple stage [13].

With the intake pressure, it fixed the amount of air to be sucked into the compressor cylinder size can be decreased with an increase in the discharge pressure, Some of the compression line at a given supply pressure will intersect with the amount of clearance indicating the absence of air delivery.

Such compression and re-expansion are continuous at this level without letting any release of air. As a result, the achievement of maximum pressure ratio in reciprocating compressors inversely proportional to the amount of clearance compressor, which cannot reduced beyond a certain limit in this case, and in order to reduce it further one has to opt for multiple staged compressor [14].

Work has done in air compressor in two stages

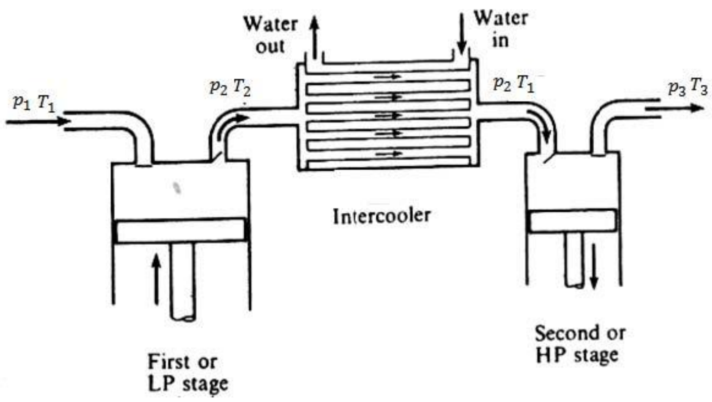



Figure 3.10. Multi-stage compression [13].

The schematic diagram of an air compressor of two stages with intercoolers enumerated hereunder, LP cylinder compresses air at pressure $P_2$ then compressed in HP cylinder after it passes through the intercooler for cooling the air at a constant pressure, It said that the cooling was apt in case the air cooled back to the initial stage.

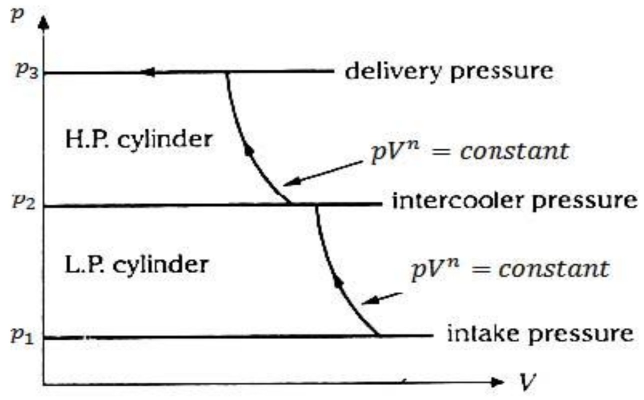



Figure 3.11. PV diagram for a two-stage air compressor [13].

Completion of work per LP cylinder cycle:

$$W(LP) = \frac{n\,P_1\,V_1}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] \tag{3.9}$$

Completion of work per HP cylinder cycle:

$$W(HP) = \frac{n\,P_2\,V_2}{n-1}\left[\left(\frac{P_3}{P_2}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] \tag{3.10}$$

In case of an optimal intercooler phase;

$$P_1V_1 = P_2V_2$$

And thus,

$$W = \frac{n\, P_1\, V_1}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} + \left(\frac{P_3}{P_2}\right)^{\left(\frac{n-1}{n}\right)} - 2\right] \tag{3.11}$$

Minimum power conditions in the two-staged compressor with optimal intermediate cooling is enumerated by.

$$P_2 = \sqrt{P_1\, P_3} \tag{3.12}$$

For multiple N-staged compressors having an optimum intermediate cooling of the compressed air moving in the direction of $P_1$ from P (N+1) enumerated by.

$$W = \frac{N\, n\, P_1\, V_1}{n-1}\left[\left(\frac{P(N+1)}{P_1}\right)^{\left(\frac{n-1}{nN}\right)} - 1\right] \tag{3.13}$$

## 3.3. COMPARISON BETWEEN SEVERAL TYPES OF COMPRESSORS

Factors affecting the performance measurement of compressors and its ultimate selection are dependent on the rate of flow, the rise of pressure within the compressor and its efficiency, Centrifugal and axial flow compressors used during medium and large rates of flow while compressor with a positive displacement that is applied to slower rate flows, the centrifugal compressor has several merits, which include good resistance to external damage, high reliability, being compact and strong, and is fouled.

Less. When compared with other compression types, positive displacement machine has a wider operating range, Centrifugal compressor at a flow rate of 30 $m^3$ per minute to almost 3000 $m^3$ per minute it used in petrochemical and process industries, A typical analysis shown in Table 3.1.

Table 3.1. Comparison between several types of compressors [13].

| **item** | **reciprocating** | **rotary vane** | **rotary screw** | **centrifugal** |
|---|---|---|---|---|
| Efficiency at full load | High | Medium-high | High | High |
| | | | | |
| Efficiency at part load | High due to staging | Poor: below 60% of full load | Poor: below 60% of full load | Poor: below 60% of full load |
| Efficiency at no load(power as % of full load) | High (10%-25%) | Medium (30%-40%) | High Poor (255-60%) | High-medium (20%-30%) |
| Noise level | Noisy | Quiet | Quiet it enclosed | Quiet |
| Size | Large | Compact | Compact | Compact |
| Oil carry over | Moderate | Low-medium | Low | Low |
| Vibration | High | Less | Less | Less |
| Maintenance | Many wearing parts | Few wearing parts | Very few wearing parts | Sensitive to dust in air |
| Capacity | Low-high | Low-medium | Low-high | Medium-high |
| Pressure | Medium- very high | Low-medium | Medium-high | Medium-high |

## 3.4. CONTROL OF THE COMPRESSOR

### 3.4.1. Compressor Drive

Mostly motors operating on alternating currents or AC used as drivers in air compressors, Induction motor and synchronous motor are types of large electric almost all of the industrial motors will be supplied with three-phase AC power, Induction and synchronous motors are both dependent on the rotating magnetic field (RMF) being produced in the winding field.

Induction motors are either of the wire wind rotors type or of the squirrel caged type and used in 90% of industrial applications, these motors differ in terms of power, slip amount and the current used and it is the squirrel caged type ones that is more widely applied.

Typically, gas turbine, diesel or Otto cycle engine power, except for the special situation are not feasible to install in cases requiring compressor of continuous service [13].

### 3.4.2. Compressor Controls

There is a growth in the availability of control system in cases of the installation for compressed air, which are most dedicated to controlling of electric drivers and the control of the compressor itself. Such controls described hereunder.

The plant air's compressor system usually designed to provide a variable volume capacity while operating at a constant pressure, the compressor customized to provide the maximum amount and at the same time, the control system used to decrease the output of the compressor to meet the requirements of the system.

Compressor to coincide with the demand can incorporate a number of different control systems to generate different compression volume and pressure as required, Such controls provide an evaluation and quick monitoring of the pressure in the system as a balance between the output and the demand, normally the control system recognizes and designed to deliver pressure air between the design's minimum value and the design's maximum value of pressure in the damped system, Such damped systems help negate the effects of pulses found in almost all types of compressors, The difference of pressure ranging between 0.1-0.5 bars, which referred to as the range of control usually specified but at the same, it changed as per the needs of the system and the users.

Compressed air costs are composed of costs on energy consumption up to 80%. Therefore, you must choose a prudent regulatory system, which classified as the following two segments [15].

- Regulator with continuous capacity, as shown in Fig. 3.12, this method requires continuous control of the drive motor and the valve in response to changes in pressure. As a result, according to the amplification and the speed of the control system, there are small pressure fluctuations ranging between 0.1 bars to 0.5 bar.

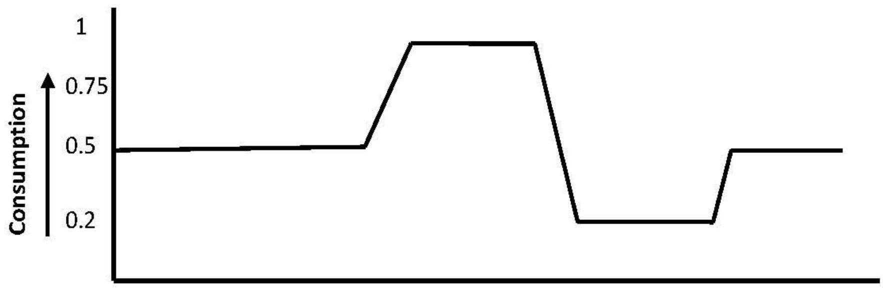



Figure 3.12. Continuous capacity adjustment [16].

- Regulation of Load / unload: As shown in Fig. 3.13, the method is about the acceptance of change in pressure between the two numbers, This is completely done by cutting the high-pressure flow during unloading and restarting it while loading when pressure is lowest. Variation in pressure is usually ranging in between 0.3 bars to 1 bar and is variable according to the number of cycles of loading and unloading.

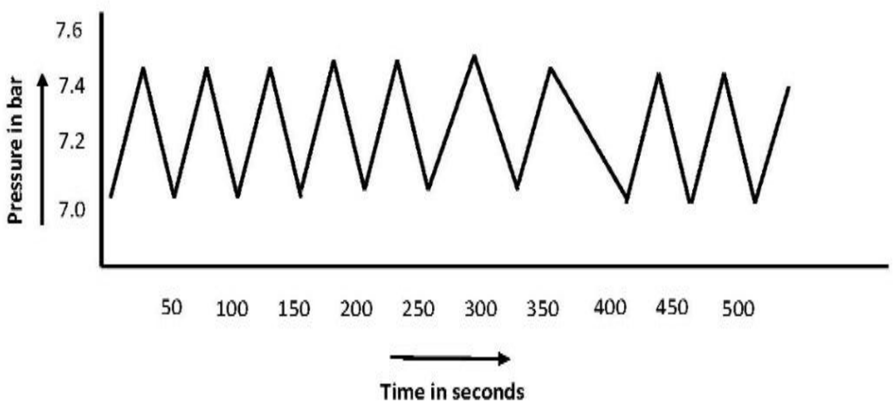



Figure 3.13. Load/unload regulation [16].

## PART 4

## COMPRESSOR DESIGN

We are given that we need to design a reciprocating compressor, which delivers 1000 lit/m at a pressure of 6 bars, Now we will first review what a reciprocating compressor is and how does it work, then we will proceed to thermodynamic design and finally to mechanical design.

### 4.1. RECIPROCATING COMPRESSORS

A compressor is a device that increases the pressure of a gas and provides this compressed gas at the required rate, now reciprocating compressors use a piston-cylinder arrangement (shown in Fig 4.1) to compress the air while the cylinder moves towards the piston surface, thus reducing the volume and compressing the gas. This, power is generally delivered to the piston, by a crankshaft, which is in turn attached to an electric motor.

Thermodynamically the compression takes place from the inlet pressure to the final discharge pressure as shown in Fig. ii. We can explain each step with the help of images as shown from Fig. 4.1 to Fig. 4.7. [13].


Figure 4.1. A general piston cylinder [13].

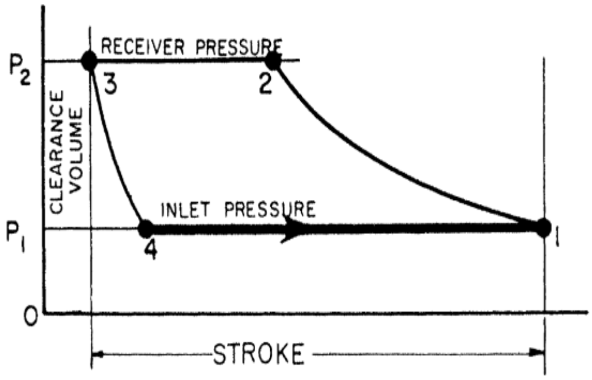



Figure 4.2. A complete Thermodynamic cycle for a compressor (1-2: Isothermal Compression, 2-3 Discharge, 3-4: Isothermal expansion, 4-5 Intake) [13].

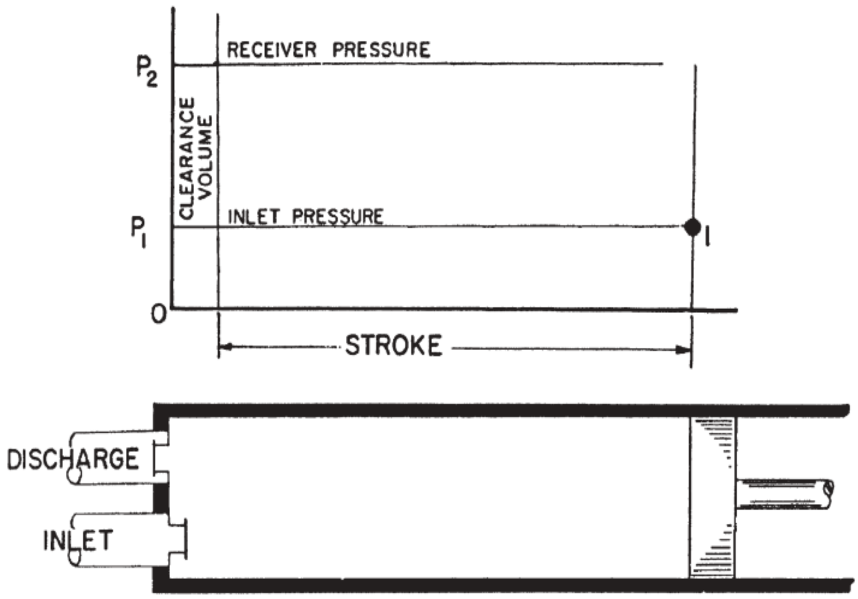
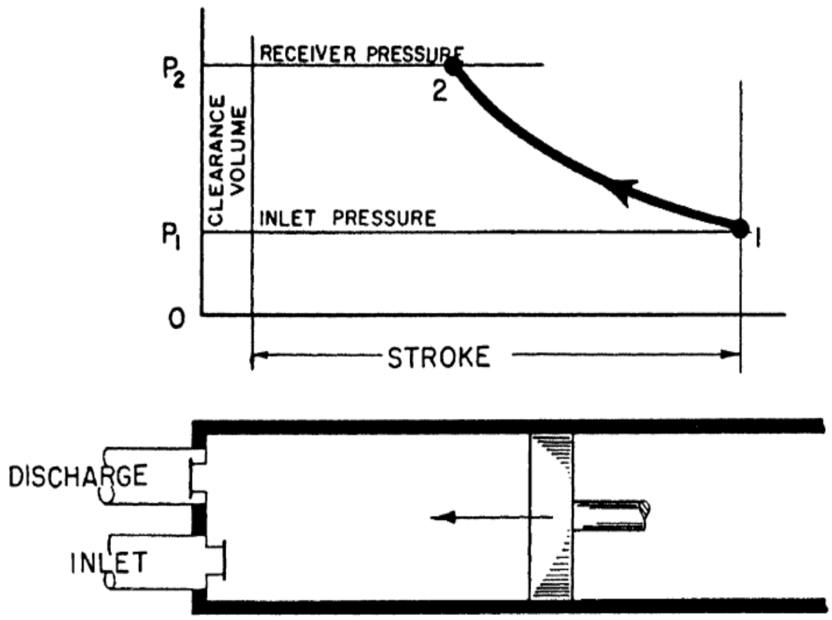



Figure 4.3. Basic elements of the compressor with the gas-filled cylinder Point 1 is the start of compression in the PV diagram both valves are considered closed [13].




Figue 4.4. Compression stroke. The piston shifted towards the left and led to a rise in pressure with a decrease original as volume. Valves are still closed. The pressure from point 1 to 2 reflected in PV diagram and the cylinder pressure is as much as that in the receiver [13].

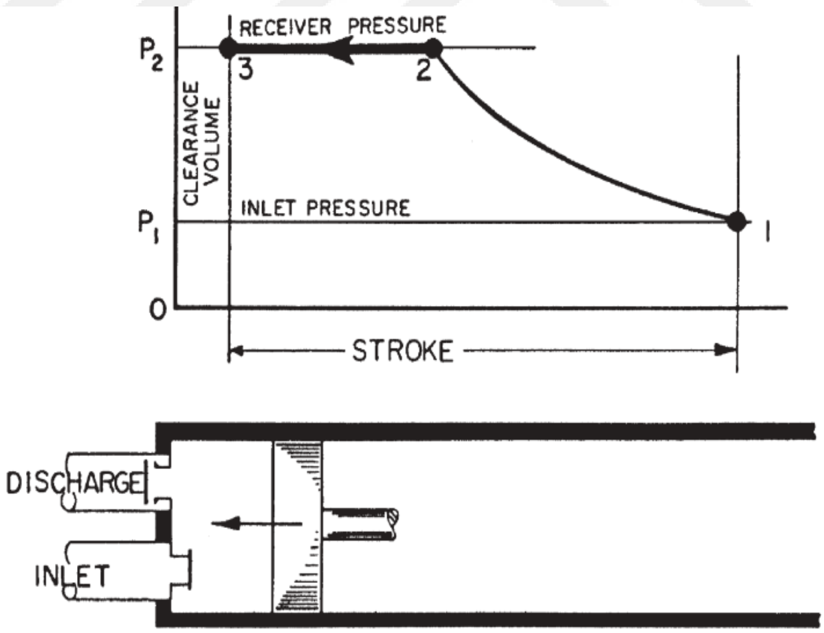



Figure 4.5. Piston completes delivery stroke. Beyond point 2, the discharge valve opened and compressed air rushes to the receiver [13].

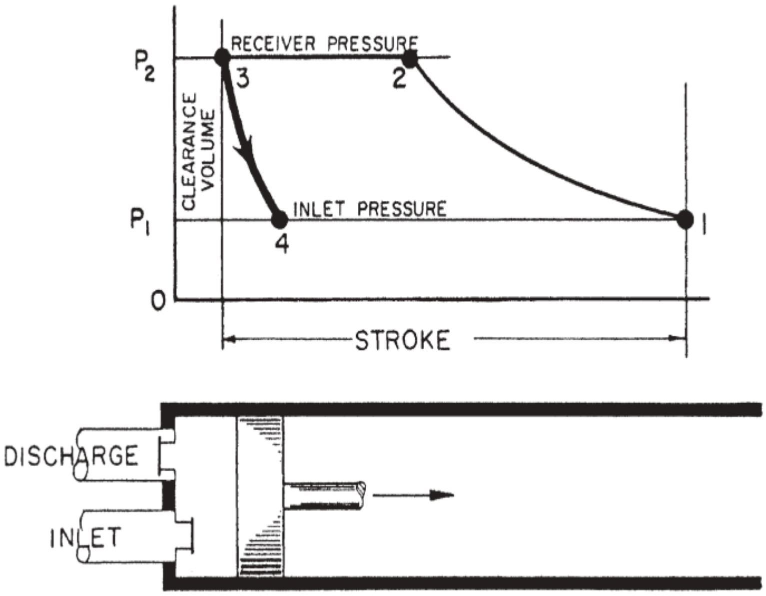



Figure 4.6. Inlet and the discharge valves closed during the expansion stroke In the Clearance space gas trapped, volume increase and pressure decrease [13].

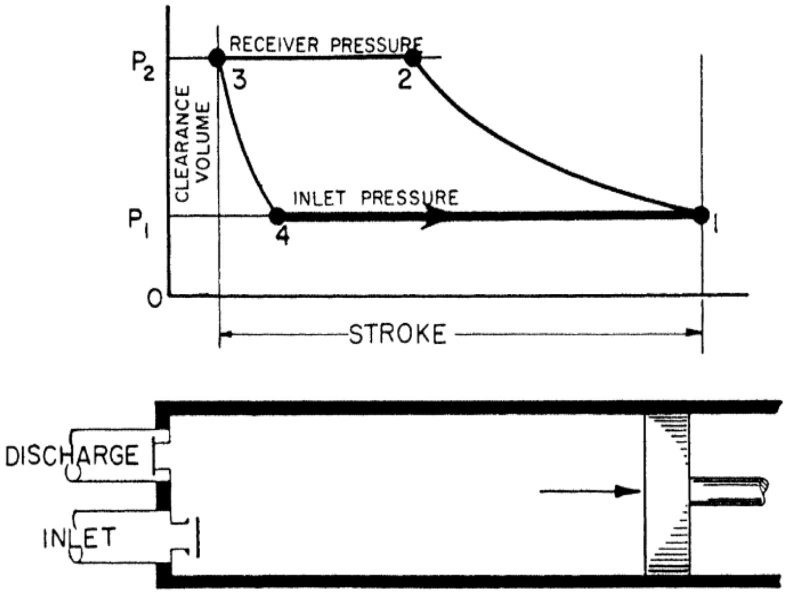



Figure 4.7. Inlet valve open at point 4 and gas flows into cylinder until the point 1 reverse stroke [13].

Thus, we understand the basic working of a reciprocating compressor (positive displacement type).

## 4.2. AIR TREATMENT THEORY

The compressed air needs to be of desired quality, as contaminants can quickly become a problem and lead to high cost of replacement and rejection, the required quality varies with application and purified using different methods to different level, to remove moisture, oil, dust or other contaminants to the desired level [8].

### 4.2.1. Contaminants

- Water vapor, Air contains moisture and some vapor is included in the compressed air and can cause many problems, such as high maintenance cost, high rate of rejection with spray painting and injection methods, shortened life and impaired performance, increased leakage, shorter life of pipes, and disturbed control system. Thus removing the water from the compressed air is essential and  be done by using after-coolers, refrigerant dryers, condensation separators and adsorption dryers. The degree of removal of water determines the type of equipment used.

- Oil, The amount of oil entering the compressed air depends on the type of machine used, service condition and age, Today, we have an option of choosing from lubricant type compressors that employ lubricant during the compression process and has a higher quantity of lubricant in the compressed air, and the other, which employs minimal to no lubricant. The cost involvement of the latter is higher and the choice is hence on a case-to-case basis. In either case, an oil filter is generally included in the design to prevent any oil from reaching the application area.

- Microorganisms in compressed air, Most of the contaminants in the air stream (around 80%) are smaller than 2 mm, and thus easily passes through the inlet filter, These contaminants can mix with water and oil to create microorganism growth, this eliminated through a filter directly after the compressor and any growth after this filter must controlled as microorganisms are small, it can pass through into the system and the control becomes a challenge, However, it becomes exceedingly important as the application sector becomes more sensitive to such organisms like medical application’s oil acts as nutrients, water presence can create an atmosphere suitable for such growth, and sterile filters employed at various spots to reduce the microorganism concentration, Sterilization of air is a good way to maintain quality.

#### 4.2.2. Equipment For Treatment

Filters with advancing technology, filters have high efficiency in removing oil, as filters are standalone equipment; there is no control on the amount of oil be removed. Moreover, the performance is subject to the working temperature, water content and oil concentration, and thus no precise assessment of the degree of removal can be made, Hence, in order to maximize the utility, the air must be dried before passing it through the filter, A combination of fiber filter and activated carbon filter used to ensure that both oil aerosol and vapor removed, moreover, to ensure the better working of the filter in the system, the filter should be replace periodically, Proper selection of filter size is also important to ensure best results.

After-cooler as the air from the compressor is hot, it needs to be cooled, This also reduces the water content in the compressed air by condensing it out this is then coupled with a water separator that has automatic drainage, to continuously remove a lot of water (around 80-90%) before storage.

Water separator this is the device used after the after-cooler and separated around 90% of the water, the remaining water enters the air receiver as mist.

Oil water separation now along with the water coming out of the water separator, we also get droplets of oil that also separated out of the compressed air, this results in the drainage of the separator becoming an oil-water emulsion that not directly let off into the environment as it considered polluting, Thus, the oil and water must be separate from this stream before eliminating them this has done by installing an oil-water separator, with maybe a diaphragm filter filtering out oil in a cross filtration process [13].

## 4.3. COOLING SYSTEM

### 4.3.1. Compressor Cooled By Water

The efficiency and amount of water vapor produced directly linked to the amount of air compressed within an intercooler and aftercooler of a compressor, almost 90% of the heat required by the electric motor comes from the cooling water of a water-cooled compressor and thus such compressors are light on the compressor rooms.

Water cooled compressors can be built using three systems which include systems using external water sources and not using circulation techniques, systems using cooling towers as water sources and using circulation techniques, and systems which are closed and are using external radiators and heat exchangers with circulation techniques.

### 4.3.2. An Open System Without Circulating Water

Compressors, which do not employ circulation techniques, use water sources that are external as if lakes, rivers, like for use within the compressor, and ultimately release them as waste, not only to maintain the desired air temperature but also in order to manage the consumption of water, the system should regulated via using a thermostat.

In general, open systems have cheap and are simple installation but it might be a bit costly to run such systems, this is so as the cooling water taken from the tap water mains, lakes or stream might be freely available but the same needs to purify and processed before it is ready to use in systems. Otherwise, excess lime, salt, or impurities in the water can lead to system and boiler corrosion and scale, which would be very inconvenient and even more expensive to repair and replace [16].

### 4.3.3. An Open System With Circulating Water

In open systems employing circulating water technique, water again cooled in the cooling tower after be compressed by sprinkling it in the downward direction as it is simultaneously cooled by air blown from the sides, as a result a portion of the water turns into vapor and the remaining water cools down to a temperature, which is 2 degrees centigrade, lower than the ambient temperature, However, this temperature is a variant on the humidity levels as well.

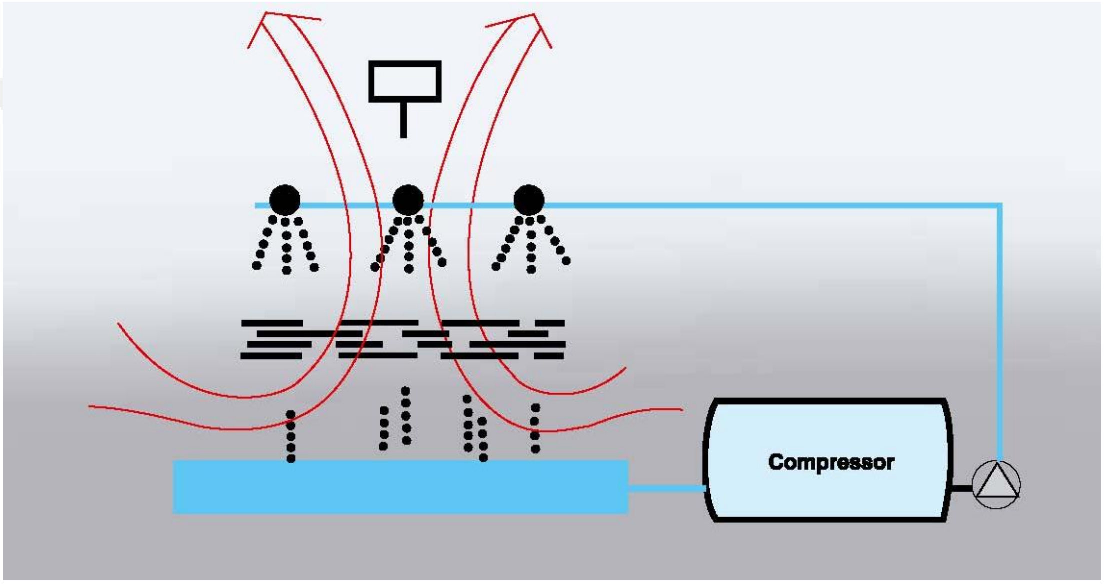



Figure 4.8. Open cooling system with circulating cooling water [16].

When the accessibility of the external water supply is constricted open system, using circulating techniques are used, However, this system is subject to certain demerits as the water used constantly exposed to dirt, algae growth, contamination, and evaporation, it is required to be regularly diluted, changed and chemically treated to keep it effective, also the salt deposition on hot machine parts leads to a reduction of heating efficiency, in order to avoid these issues, the cooling tower water should be drain when not in use.

### 4.3.4. Compressor Cooled By Air

These days compressor cooling is also done via air cooling systems where in the energy consumption of the electric motor is contained in the package of the air compressor.

In this case, care should take to avoid the water frozen. In order to achieve this glycol mixture, which has a lower thermal ability as compared with pure water has taken into consideration.

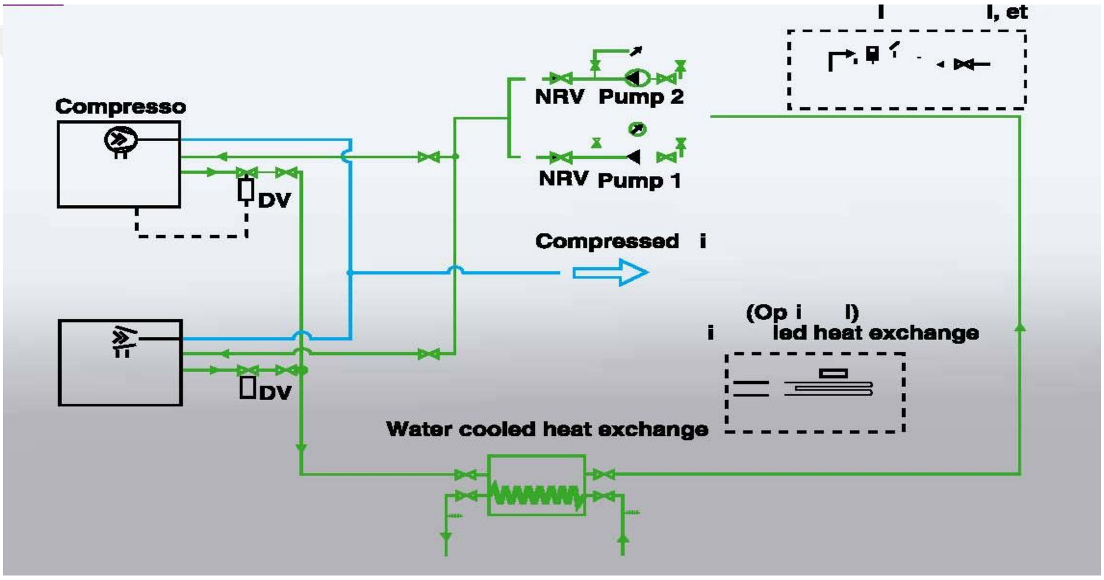



Figure 4.9. Closed cooling system where heat exchanger is either water or air-cooled [16].

## 4.4. COMPRESSOR ROOM

Until a few years back, people were required to buy all parts separately which included buying an electric motor, starter, cooler, filters and like, In addition, customers were required to consider the capacity, and quality of all the components incorporated and their compatibility with the entire system, These issues have been dealt with in today's times and nowadays consumers, which are a fully functional system and are applied as holistic solutions, purchase the entire system, This system is composed of a box frame, which contains all the different components in which all connections are made in the factory to decrease the level of noise that is produced. Such an idea has hugely helped customers as it has reduced many hassles and have made the buying and installation of compressors very easy.

One such example of an operational air compressor is the compressor built on the worksite and which includes all parts like the cooler dryer etc. that completely integrated, Such compressors are very effective and have low levels of noise, Such compressors are installed either along the already present distribution systems of air or along future extensions being added but such installations need to be monitored carefully or else it would negatively impact the efficiency and dependability.

While installing a compressor, should be noted that the compressor placed separately to increase efficiency, cost effectiveness, design and usage ease, safety, safeguard, control and to decrease noise [16].

On the other hand, it should also note that another area of the premises used for other matters could made to use of for installing the compressor, This has done to combat issues related to installation such as noise reduction, dealing with toxins and hazards, ventilation issues, drainage ease, reduction of damage risks, dirt and impurities to avoided, area required for further expansion in the future, ease of use, recovery of energy among several other issues to look into.

In the case of absence of space required the compressor could also installed at outside premises under a covered area. This, however, comes with certain prerequisites such as dealing with the risk of freezing condensates and discharges, protection from the environment, seepage from inlets and ventilators, protection from unauthorized personnel, hyperactive substances, inflammable articles, dirt and the presence of a firm concrete foundation.

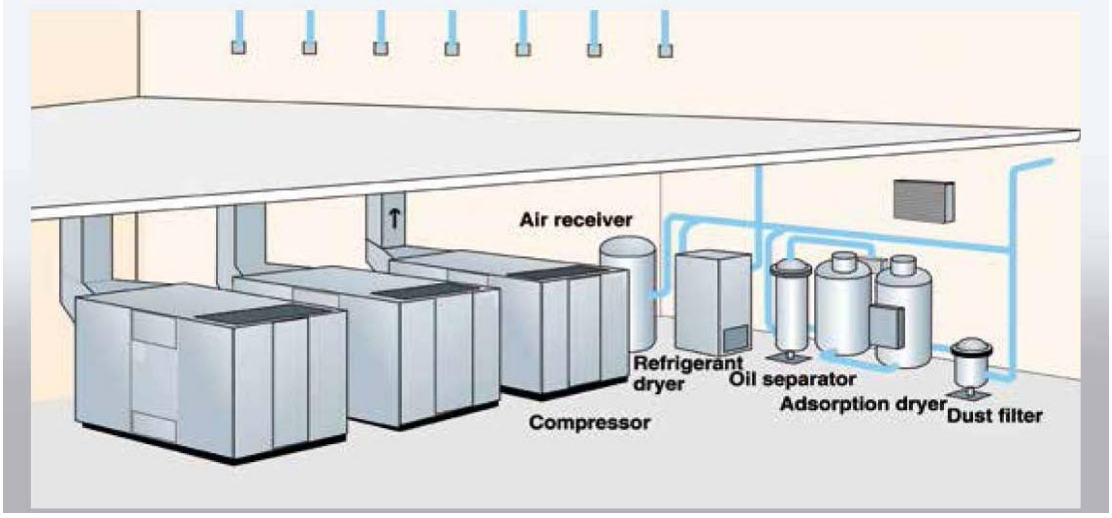



Figure 4.10. Compressor room installation is a ready to install package, which is Readily conncected to auxiliary equipment [16].

### 4.5. DESIGN AND PLACE

Air Compressor plant can used to help in distributions with the help of pipes throughout the facility and the same has done by strategically placing the compressor at a centralized area or close to boilers or pumps and coolers.

The premises containing the compressor should be laden with lifts or forklift trucks to help with heavy items such as the electric motors. To facilitate the same the height of the premises should be optimal to facilitate the lifting of motors. In addition, provision for further expansion should made by leaving enough area on the floor. The system should also facilitate the drainage of condensate and the same should done according to the prevalent rules and regulations of the territory.

### 4.6. FOUNDATION

In general, the compressor needs a flat foundation with a certain minimum weight bearing capacity. Such foundations should designed to bear any vibrations and the compressors should installed on a plinth to allow the area to clean regularly and properly.

It is essential to form a concrete foundation to help the centrifugal compressors and large pistons to operate efficiently. The foundation needs to attach to the underlying soil or rock surface. Such foundations can greatly reduce the vibrations in modern compressor plants and in the case of centrifugal compressors, in order to decrease the vibrations, it would be imperative to create a dampened surface.

### 4.7. AIR INTAKE

The air intake within the compressor needs to be is clean and completely free of dust, dirt, solid and gaseous pollution, Otherwise, particulates can lead to damage and corrosion of the compressor plant.

The inlet for the air for the compressor is generally found at the opening where there is less noise however as the air needs to be clean thus I can alternatively also be placed in an area which can provide the same even if it is a distant place.

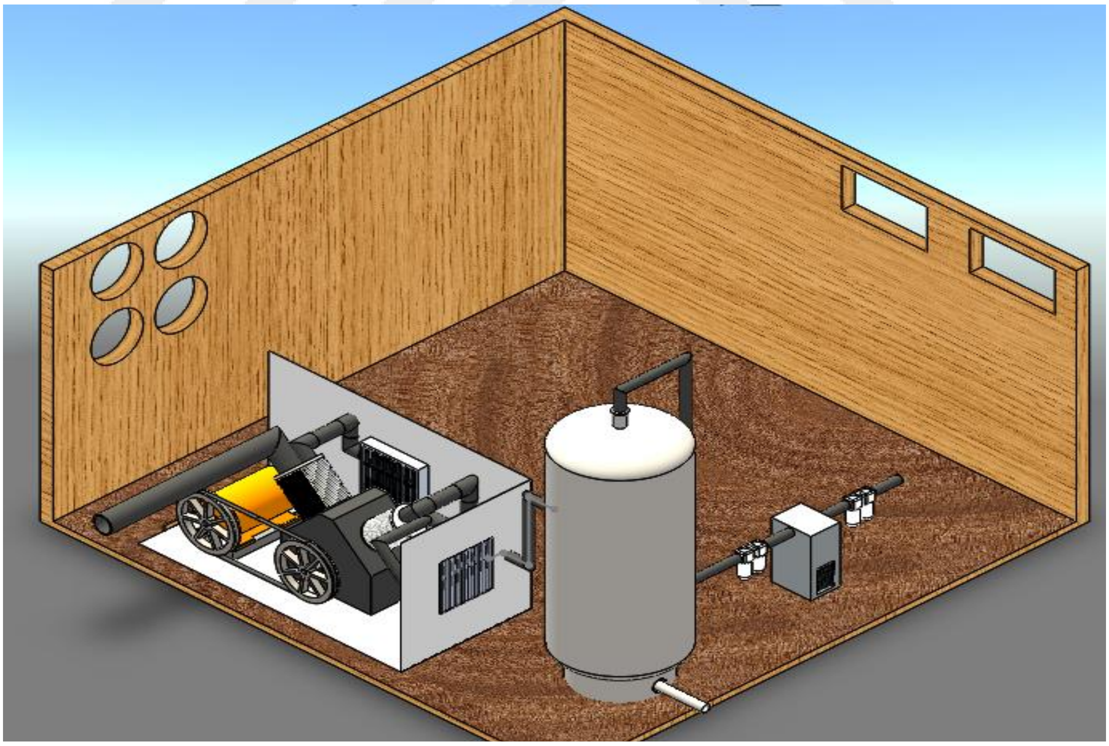


Figure 4.11. Compressor room installation should be user-friendly and expandable There should be 1200 mm area in front of machine electric cabinets at service points.

Air pollution and particulates therein due to vehicles can be detrimental for the compressors and in order to avoid corrosion of plants, filters should installed and the decrease in pressure due to their installation should adjusted for while designing the plant itself. Such filters can be of the rotary band, panel, and cyclone type.

Taking in cold air can also be good and for the same reason, air routed into the plant using a different pipe outside of the plant.

It is imperative that inlet pipes be large to account for pressure drops and that they be corrosion resistant and that they contain filters to avoid damage from rain and snow, which could enter the pipe.

The inlet tube pipes should properly designed to combat vibrations and pulsating waves from the compressors cycles, and the noise from the surroundings. Otherwise, it can lead to damage of the pipes and the entire compressor itself.

## 4.8. VENTILATION OF THE COMPRESSOR ROOM

The number of compressors in the compressor room, its size, and type (whether it cooled by air or water) determines the total heat generated in there and in order to ventilate the same, ventilators should be installed.

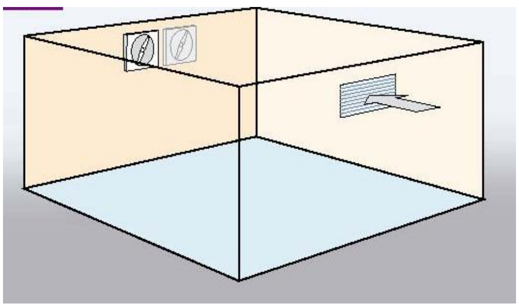


Figure 4.12. Solution for basic type ventilation [16].

Almost all of the energy used by the electric motors in the form of heat is contained in the air ventilation of the compressors. This same ratio of energy containment reduces to almost 10% in cases of water-cooled compressors. Thus in order to regulate the compressor room temperature the heat needed to reduces in there. While construction, the flow of ventilation needs to be calculated which has done as follows:

$$q_v = \frac{Pv}{1.21 * \Delta T} \tag{4.1}$$

Where,

$q_v$= Air ventilation quantity in m3 per second

$P_v$= flow of heat in kW

$\Delta T$ = permissible rise in temperature given in degree centigrade

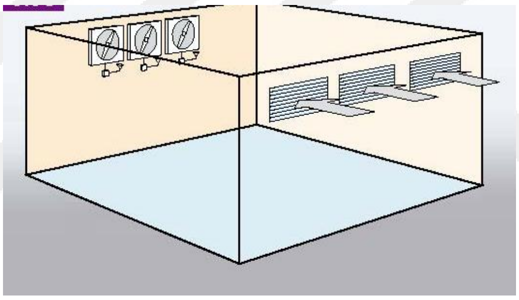


Figure 4.13. Multi-stage thermostat system [16].

The already used heat can reused to deal with the heating issues. Outdoor ventilation using smaller pipes should be beneficial. At the same time, the inlets should place at a lower altitude but should be free of rain and snow damage as well. Air pollution and other particulate corrosion should also care for the fans in the ventilation area would be most beneficial if placed at a higher place on the wall and the intake air should come from the opposite side.

Fans that regulated via thermostat would also be apt for such uses, it can designed to tackle the issues of pressure drops at the outer wall, and the volume of intake air can regulate other places, the rise in temperature, which would be restricted to almost 7 to 10 degrees.  In the case of abundant ventilation options, water-cooled compressors can also use [16].

## 4.9. AIR RECEIVER

Each compressor installations consists of a single or multiple air receivers. The size of the compressor is dependent on the capacity of the Compressor, the system of regulation and the air patterns demand by the users. For the compressed air, the air receiver creates a buffer storage area. The pulsations from the compressor balanced as well. It also makes the air cool and then collects the same. As a result, you need to install a condensate drainage system in the air receiver.

For compressors with offloading or unloading systems, the following equations would be applicable.

$$V = \frac{0.25 * q_c * p_1 * T_0}{f_{max} * (p_U - p_L) * T_1} \tag{4.2}$$

$V$ = receiver volume of air (liter)
$q_v$= compressor FAD (L / S)
$P_1$ = compressor inlet pressure (bar (a))
$T_1$= Maximum inlet temperature in compressor (K)
$T_0$ = Compressor temperature of air receiver (K)
$(p_U - p_L)$ = Load and Unload difference in set pressure
$f_{max}$ = maximum loading frequency (30 seconds is applied per 1 cycle To Atlas Copco compressor)

The volume of air receiver for the variable speed control compressor will reduced substantially. If you want to use the above equation, q(c) must be taken as FAD at the lowest rate.

If demand for the air compression varies frequently in the short span of time, it considered feasible to will not change the dimension and pipe network. It considered more prudent to install a different air receiver near the consumer outlet and adjust it as the requirements.

In order to meet the air requirements of high capacity, a small compressor with higher-pressure capacity is used. The average consumption met with the adjustments in the dimension of the compressor. These types of receivers follow the following relation

$$V = \frac{q * t}{(P_1 - P_2)} = \frac{L}{(P_1 - P_2)} \tag{4.3}$$

$V$ = air receiver volume (L)

$q$ = discharge phase air flow (L / S)

$t$ = discharge phase time (s)

$P_1$ = Network normal work pressure (bar)

$P_2$ = consumer function minimum pressure (bar)

$L$ = air requirements in the filling stage (1 / work cycle)

In the above relation, it is considered that the compressor cannot supply air during the discharge/emptying phase. It is generally applicable to large ship engines, which have a 30 bar receiver pressure at filling.

## 4.10. COMPRESSED AIR NETWORK DESIGN

The compressed air network design and dimensioning contains details of all the consumers of the compressed air and shows their individual positions. These consumers then segregated into logical units, and the same distribution pipe used to supply to them, which supplied by the risers in the compressor plant. Compressed air is transferred from the compressor plant to the consumption are with the use of these risers. The compressed air then distributed throughout the distribution area with the help of distribution pipes and from the distribution pipes; the air distributed to the workplaces using service pipes [13].

## PART 5

## PRELIMINARY THERMODYNAMIC DESIGN

Now, we can design a compressor, according to the given data, a volumetric rate, and a delivery pressure. Generally, the compression ratio is determined from the discharge temperature, we are not giving much data about the working gas, or the temperature range, so we will make assumptions. Similarly, we are giving a discharge rate, but that is not enough to make the entire design, so we will make an important assumption about the motor rpm we are using. As in normal air compressors, we can select from some commonly used rpm. We can choose 2000 rpm, as that is easily available on many electric motors.

Assumptions**:**

- working gas is air
- The air at the intake is at 25 deg. C and 1 atmosphere
- The compression process is adiabatic
- Motor rotates at 2000 rpm
- For simplicity, we assume the compression/ expansion to be isothermal
- Other assumptions will be make as needed during the analysis.

Now we can use the volumetric rate and the rpm to get the volume the cylinder discharges in each rotation, which will in turn mean the cylinder volume (or stroke volume = Max volume – Min Volume).

$$Stroke\ Volume = \frac{V}{\omega} = \frac{1000\,\frac{lit}{min}}{2000\,\frac{rot}{min}} = 0.5\,\frac{lit}{rot}$$

Thus, our cylinder will need to discharge 0.5 liters in each rotation, which gives us our stroke volume.

Now we again reference our thermodynamic cycle:

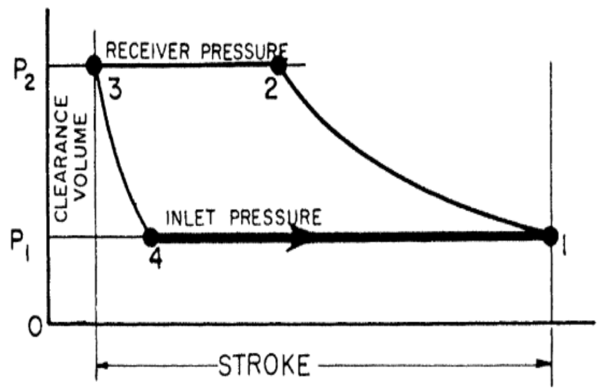



Figure 5.1. Schematic of the compression cycle [13].

Given that: $P_2 = P_3 = 6\ bar$

In addition $P_4 = P_1 = 1atm = 1.01325\ bar; T_1 = T_4 = 25\ ℃ = 298\ K$ (4 is intake).

Similarly, we know that the volume delivered should be $= V_2 - V_3 = 0.5\ lit = 5 * 10^{-4} m^3$.

Now, we also assume the clearance volume ratio to be 10, i.e.

$$\frac{V_1}{V_3} = 10$$

We note that 1-2 is an adiabatic process so (With gamma =1.4 as air is diatomic. [9].

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

$$=> V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} = V_1\left(\frac{1.01325}{6}\right)^{\frac{1}{\gamma}} = 0.2807V_1$$

Similarly,

$$P_3V_3^{\gamma} = P_4V_4^{\gamma}$$

$$=> V_4 = V_3\left(\frac{P_3}{P_4}\right)^{\frac{1}{\gamma}} = V_3\left(\frac{6}{1.01325}\right)^{\frac{1}{\gamma}} = 3.567V_3$$

Now as we know that $V_1 = 10V_3$ and as

$$V_2 - V_3 = 0.5 \;\; => 0.2807V_1 - 0.1V_1 = 0.5$$

$$=> 0.1807V_1 = 0.5 => V_1 = 2.766\; lit.$$

Hence, we can calculate:

$$V_2 = 0.776\; lit$$

$$V_3 = 0.2766\; lit, and$$

$$V_4 = 0.9866\; lit$$

Now again for 1-2, we can find $T_2$ as:

$$T_2P_2^{\frac{1-\gamma}{\gamma}} = T_1P_1^{\frac{1-\gamma}{\gamma}}$$

$$=> T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 298\left(\frac{6}{1.01325}\right)^{\frac{0.4}{1.4}} = 495.35K = 222\;℃$$

As in 2-3, the pressure is same and the temperature is also same, we get

$$T_3 = T_2 = 495.35\ K$$

This discharge temperature is a bit high, and we may want to consider a dual stage compressor.

### 5.1. PRELIMINARY DUAL STAGE DESIGN

Now if we plan to compress the air in two stages then we will need a dual stage compressor. The pressure at the outlet of the first compressor is then to be determined so that the other factors can be determined likewise [13].

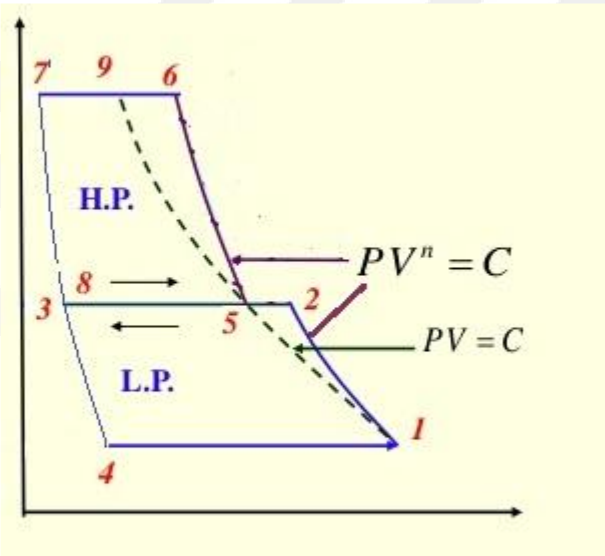



Figure 5.2. Schematic of dual stage compression [13].

Now, for determining the intermediate pressure, we will use the minimization of work principle, which determines that the intermediate pressure should be the geometric mean of the extreme pressures, thus:

$$P_{int} = \sqrt{P_{in}P_{out}} = \sqrt{(1.01325)(6)} = 2.4657\ bar$$

Now using this as the discharge pressure of the first stage, we can find the parameters of the first cylinder.

First Cylinder (LP)

As fond out: $P_2 = P_3 = 2.47\ bar$

And $P_4 = P_1 = 1atm = 1.01325\ bar; T_1 = T_4 = 25\ C = 298\ K$ (4 is intake)
We note that 1-2 is an adiabatic process so (With gamma =1.4 as air is diatomic).

$$P_1V_1^\gamma = P_2V_2^\gamma$$

$$=> V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} = V_1\left(\frac{1.01325}{2.47}\right)^{\frac{1}{1.4}} = 0.5312V_1$$

Similarly,

$$P_3V_3^\gamma = P_4V_4^\gamma$$

$$=> V_4 = V_3\left(\frac{P_3}{P_4}\right)^{\frac{1}{\gamma}} = V_3\left(\frac{2.47}{1.01325}\right)^{\frac{1}{1.4}} = 1.8825V_3$$

Also as the clearance ratio still chosen at 10% we get

$$V_1 = 10V_3$$

Now again for 1-2, we can find T2 as:

$$T_2P_2^{\frac{\gamma-1}{\gamma}} = T_1P_1^{\frac{\gamma-1}{\gamma}}$$

$$=> T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 298\left(\frac{2.47}{1.01325}\right)^{\frac{0.4}{1.4}} = 384.2\ K = 111.2\ ℃$$

As in 2-3, the pressure is same and the temperature is also same, we get

$$T_3 = \mathrm{T}_2 = 384.2\ \mathrm{K}$$

Intercooler 2 to 5 now after the exit from the first compressor we will need to cool the exit from 384.2 K to 298 K. For doing this we will use a copper coil (as copper is non-corrosive and has a high thermal conductivity), The length of the intercooler tube can be determined or a standard length of 1 m can be used It is to be noted here that we are assuming a perfect intercooler right to the ambient temperature, which is an assumption made for simplicity, For practical purpose no matter how the intercooler is designed the temperature will not reach ambient (unless some cooling utility is used other than an air radiator). Here, as the cooling takes place at the same pressure, we can calculate the relation between volumes at 5 and 2 [11].

$$\frac{V_5}{T_5} = \frac{V_2}{T_2} => V_5 = V_2\left(\frac{T_5}{T_2}\right) = V_2\left(\frac{298}{384.2}\right) = 0.7756V_2$$

<u>Second Cylinder (HP)</u>

Now as the second cylinder will also work on similar thermodynamics we can see.

As we found that: $P_5 = P_8 = 2.47\ bar$

And $P_6 = P_7 = 6\ bar;\quad T_5 = T_8 = 25\ C = 298\ K$ (after intercooler)

Similarly, we know that the volume delivered should be=

$$V_6 - V_7 = 0.5\ lit = 5 * 10^{-4}m^3$$

Now, we also assume the volume ratio to be 10, i.e.

$$\frac{V_7}{V_1} = 10.$$

We note that 5-6 is an adiabatic process so (With gamma =1.4 as air is diatomic).

$$P_5V_5^{\gamma} = P_6V_6^{\gamma}$$

$$=> V_6 = V_5 \left(\frac{P_5}{P_6}\right)^{\frac{1}{\gamma}} = V_5 \left(\frac{2.47}{6}\right)^{\frac{1}{1.4}} = 0.5312V_5$$

Similarly,

$$P_7V_7^{\gamma} = P_8V_8^{\gamma}$$

$$=> V_8 = V_7 \left(\frac{P_7}{P_8}\right)^{\frac{1}{\gamma}} = V_7 \left(\frac{6}{2.47}\right)^{\frac{1}{1.4}} = 1.8825V_7$$

Now as we know that $V_5 = 10V_7$ and as

$$V_6 - V_7 = 0.5 \ => 0.5312V_5 - 0.1V_5 = 0.5$$

$$=> 0.4312V_5 = 0.5 => V_5 = 1.159\ lit.$$

Hence, we can calculate:

$$V_6 = 0.5312V_5 = 0.6157\ lit$$

$$V_7 = 0.1V_5 = 0.1159\ lit V_8 = 1.8825V_7 = 0.2182\ lit$$

Also for the first cylinder, we can use the relation:

$$V_2 = \frac{V_5}{0.7756} = 1.4943\ lit$$

Using this we can find:

$$V_1 = \frac{V_2}{0.5312} = 2.813\ lit$$

$$V_3 = 0.1V_1 = 0.2813\ lit, and$$

$$V_4 = 1.8825V_3 = 0.5296\ lit$$

Now again for 5-6, we can find T2 as:

$$T_5 P_5^{\frac{1-\gamma}{\gamma}} = T_6 P_6^{\frac{1-\gamma}{\gamma}}$$

$$=> T_6 = T_5 \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}} = 298 \left(\frac{6}{2.47}\right)^{\frac{0.4}{1.4}} = 384.2\ K = 111.2\ ℃$$

As in 6-7, the pressure is same and the temperature is also same, we get

$$T_7 = T_6 = 384.2\ K$$

This temperature as the output is acceptable and we can proceed further. Also, we can note that the design volume of compressor 1 is 2.813 lit and for compressor 2 is 1.159 lit.

Hence, we now have our volume, pressures, and temperatures, we can now move to mechanical design.

## 5.2. ACTUAL THERMODYNAMIC DESIGN

### 5.2.1. Dual Stage Compressor

Now, we will use relax some constraints and try to implement the compressor in a real-life scenario. Under this case, we will consider pressure drops, heat losses, and leakage during operation to make a better assessment of the thermodynamic cycle [14].

Practical cycle can be implemented with consideration given too many other equipment and systems that will be added.

- Pressure drop at the valves, and in the system components will be considered

- Air requirement specified at 1000 lit/ min, but we will design for a higher flow (15% extra), as there will be some leakage, air dryer will be used [16].
- Air filters will be provided.
- The intercooler and the aftercooler will be designed according to the actual specimen.
- The power consumption will be determined.
- The heat loss to the ambient is assessed and the air cooling need is determined.
- The room ventilation is thus designed and specified.
- With all the inputs, the proper engineering 3D drawing created.

Now, the outlet pressure for which the system is being designed is P = 6 bar

In practice, the outlet from the second cylinder will pass through an after-cooler, such that it can help to reduce the temperature to desired levels [12].

Ideally, the temperature will be reduce to the ambient, but as mentioned earlier, this is not feasible, and considering an air-cooled system, and approach temperature of 10-11 ℃ is generally reached, this means that the temperature from the outlet of the aftercooler will be around 10 ℃ higher than the ambient. Considering an ambient temperature as 298.15 K, we can determine the exit temperature from the after-cooler as 308.15 K. Now, this is important because this will reduce the temperature and the volume (as the pressure across after-cooler is constant). Thus, the design flow rate of the two-stage compressor will increase, to be able to deliver the required rate of 1000lit/min after the after-cooler. Now, as we have already performed a rough ideal analysis earlier, we can estimate the design flow rate that we can use to calculate the columns, in the actual design we perform, this can simply be written as.

$$FAD = 1000\frac{lit}{min}$$

$$Discharge\ volume = \frac{FAD}{Storage\ temperature}(Discharge\ temperature)$$

$$Discharge\ rate = \frac{100}{308.15}(384.2) = 1433.81\ {}^{l}/_{min}$$

$Total\ Discharge = 23.90\ {}^{l}/_{s}$

Considering a margin of safety 10-20%, we can use a dimensioned flow rate of 23.9*1.15. Thus, for design purpose, we will use a flow rate of around

$$Discharge\ flow = 23.90 \times 1.15 = 27.48\ {}^{l}/_{s} = 1648.89\ {}^{l}/_{min}$$

Now we assume that the pressure drop in the entire system does not exceed 1.5 bars, the design pressure considered as:

Design Pressure = 7.5 bars.

Now, this pressure drop verified later to make sure that the system pressure drop does not exceed this value.

As our preliminary design also showed that, a two-stage compressor will be better suited as.

- It will take lesser work input
- The outlet temperature better controlled.

The suction valve pressure drop can be considered as = 0.2 bars [11].

The discharge valve pressure drop can be considered as = 0.4 bars [11].

The pulsating nature of the discharge pressure will be taken care of while selecting the valves and thus not considered here, this pressure mentioned to determine the actual suction and discharge pressure that now calculated as.

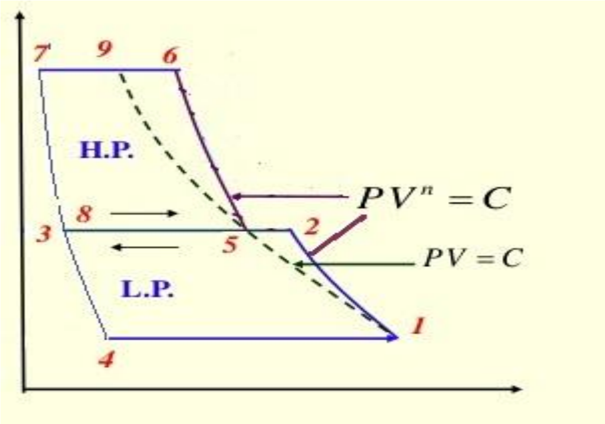



Figure 5.3. Schematic of dual stage compression [13].

As the ambient conditions considered as:

Pressure = 1.01325 bar

Temperature =25 C = 298.15 K

Thus, for the suction process 4-1, the pressure found, after adjusting for the suction valve loss as.

$$P_1 = P_4 = P_{amb} - P_{loss} = 1.01325 - 0.2 = 0.81325\ bar$$

Also, the inlet temperature considered as.

$$T_1 = T_4 = T_{amb} = 298.15K$$

The discharge pressure is written as:

$$P_6 = P_7 = P_{des} + P_{loss} = 7.5 + 0.4 = 7.9bar$$

Now, we can assess our cylinder volume to be able to determine other properties. After further study, we can re-fix our constraint on motor speed. As any speed higher than 1500-rpm falls under high-speed compression now from the starter motors available at Crompton Greaves seen as below to choose an industrially available speed [11].

Table 5.1. Performance data for aluminum motors [15].

| RATED POWER | | FRAME | FULL LOAD CURRENT | | | FL | FLT | EFFICIENCY | | | POWER FACTOR | | | D.O.L STARTING | | PULLOUT | $GD^2$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | In | | | SPEED | | | | | | | | STT | SSC | POT | |
| kW | hp | | 380V | 400V | 415V | RPM | kg.m | FL | 3/4L | 1/2L | FL | 3/4L | 1/2L | %FLT | %FLA | %FLT | $kg.m^2$ |
| **2 POLE - 3000 Synchronous rpm, 50Hz** | | | | | | | | | | | | | | | | | |
| 0.18 | 0.25 | GD63 | 0.62 | 0.59 | 0.57 | 2810 | 0.06 | 64.0 | 61.0 | 56.0 | 0.69 | 0.60 | 0.49 | 190 | 550 | 240 | 0.0010 |
| 0.25 | 0.33 | GD63 | 0.73 | 0.69 | 0.67 | 2810 | 0.09 | 66.0 | 64.0 | 60.0 | 0.79 | 0.72 | 0.61 | 230 | 450 | 250 | 0.0016 |
| 0.37 | 0.50 | GD71 | 0.93 | 0.88 | 0.85 | 2815 | 0.13 | 72.0 | 73.0 | 72.0 | 0.84 | 0.78 | 0.66 | 180 | 500 | 210 | 0.0013 |
| 0.55 | 0.75 | GD71 | 1.30 | 1.23 | 1.19 | 2820 | 0.19 | 74.0 | 75.0 | 73.0 | 0.87 | 0.83 | 0.73 | 190 | 500 | 210 | 0.0018 |
| 0.75 | 1.00 | GD80 | 1.78 | 1.69 | 1.63 | 2850 | 0.26 | 77.0 | 77.0 | 73.0 | 0.83 | 0.79 | 0.68 | 220 | 550 | 250 | 0.0034 |
| 1.1 | 1.50 | GD80 | 2.52 | 2.39 | 2.30 | 2870 | 0.37 | 81.0 | 81.0 | 79.0 | 0.82 | 0.77 | 0.66 | 270 | 670 | 270 | 0.0047 |
| 1.5 | 2.00 | GD90S | 3.39 | 3.22 | 3.10 | 2870 | 0.51 | 82.0 | 82.0 | 80.0 | 0.82 | 0.77 | 0.66 | 250 | 600 | 300 | 0.0047 |
| 2.2 | 3.00 | GD90L | 4.63 | 4.40 | 4.24 | 2850 | 0.75 | 84.0 | 84.0 | 82.0 | 0.86 | 0.83 | 0.73 | 250 | 630 | 300 | 0.0067 |
| 3.0 | 4.00 | GD100L | 6.0 | 5.7 | 5.5 | 2875 | 1.02 | 86.0 | 86.0 | 84.0 | 0.88 | 0.83 | 0.75 | 300 | 700 | 310 | 0.0192 |
| 3.7 | 5.00 | GD100L | 7.4 | 7.0 | 6.7 | 2875 | 1.25 | 86.5 | 87.5 | 86.5 | 0.88 | 0.83 | 0.75 | 300 | 700 | 310 | 0.0192 |
| 4.0 | 5.50 | GD112M | 8.0 | 7.6 | 7.3 | 2870 | 1.36 | 86.5 | 86.5 | 86.0 | 0.88 | 0.83 | 0.75 | 275 | 780 | 310 | 0.0240 |
| 5.5 | 7.50 | GD132S | 10.8 | 10.3 | 9.8 | 2910 | 1.84 | 86.0 | 86.0 | 84.0 | 0.90 | 0.85 | 0.73 | 270 | 820 | 300 | 0.0579 |
| 7.5 | 10.00 | GD132S | 15.0 | 14.3 | 13.7 | 2900 | 2.52 | 87.0 | 89.0 | 88.0 | 0.87 | 0.87 | 0.82 | 250 | 820 | 300 | 0.0752 |
| **4 POLE - 1500 Synchronous rpm, 50Hz** | | | | | | | | | | | | | | | | | |
| 0.18 | 0.25 | GD63 | 0.67 | 0.63 | 0.61 | 1370 | 0.13 | 64.0 | 63.0 | 58.0 | 0.64 | 0.58 | 0.47 | 220 | 500 | 250 | 0.002 |
| 0.25 | 0.33 | GD71 | 0.85 | 0.80 | 0.77 | 1400 | 0.17 | 68.0 | 68.0 | 63.0 | 0.66 | 0.59 | 0.49 | 180 | 400 | 220 | 0.003 |
| 0.37 | 0.50 | GD71 | 1.16 | 1.11 | 1.07 | 1410 | 0.26 | 71.0 | 69.5 | 64.5 | 0.68 | 0.61 | 0.50 | 180 | 400 | 220 | 0.003 |
| 0.55 | 0.75 | GD80 | 1.66 | 1.58 | 1.52 | 1400 | 0.38 | 75.0 | 75.0 | 73.0 | 0.67 | 0.61 | 0.51 | 200 | 420 | 250 | 0.006 |
| 0.75 | 1.00 | GD80 | 1.97 | 1.88 | 1.81 | 1410 | 0.52 | 78.0 | 78.0 | 76.0 | 0.74 | 0.68 | 0.54 | 200 | 440 | 250 | 0.007 |
| 1.1 | 1.50 | GD90S | 2.78 | 2.64 | 2.55 | 1410 | 0.76 | 79.0 | 79.0 | 77.0 | 0.76 | 0.69 | 0.56 | 220 | 510 | 250 | 0.010 |
| 1.5 | 2.00 | GD90L | 3.70 | 3.52 | 3.39 | 1420 | 1.03 | 81.0 | 82.0 | 81.0 | 0.76 | 0.71 | 0.59 | 250 | 560 | 280 | 0.014 |
| 2.2 | 3.00 | GD100L | 5.0 | 4.7 | 4.6 | 1415 | 1.51 | 83.0 | 82.0 | 81.0 | 0.80 | 0.76 | 0.64 | 200 | 550 | 250 | 0.023 |
| 3.0 | 4.00 | GD100L | 6.9 | 6.5 | 6.3 | 1435 | 2.04 | 84.0 | 84.0 | 82.0 | 0.79 | 0.75 | 0.68 | 200 | 600 | 250 | 0.054 |
| 3.7 | 5.00 | GD112M | 8.4 | 7.9 | 7.6 | 1435 | 2.51 | 85.0 | 85.0 | 84.0 | 0.79 | 0.75 | 0.68 | 200 | 550 | 250 | 0.054 |
| 4.0 | 5.50 | GD112M | 9.2 | 8.7 | 8.4 | 1435 | 2.71 | 85.0 | 85.0 | 84.0 | 0.78 | 0.72 | 0.62 | 200 | 550 | 250 | 0.054 |
| 5.5 | 7.50 | GD132S | 12.0 | 11.4 | 11.0 | 1445 | 3.71 | 85.7 | 86.0 | 85.0 | 0.81 | 0.74 | 0.62 | 225 | 700 | 300 | 0.098 |
| 7.5 | 10.00 | GD132M | 15.9 | 15.1 | 14.6 | 1445 | 5.06 | 87.0 | 87.0 | 86.0 | 0.82 | 0.78 | 0.68 | 240 | 720 | 300 | 0.131 |

As seen, a standard speed lower than 1500 rpm would be 1400 rpm, and this is widely accepted in compressors [11].

Thus, the motor speed = 1400 rpm

Now, for the clearance ratio, $C = V3/V1 = 0.05$

This lower clearance value will make the volumetric efficiency and the commonly accepted design does support this design parameter.

Now, as we have the flow rate needed (discharge volume rate) and the crank rotation rate, we can determine the discharge volume as.

$$\text{Discharge volume} = V_d = \frac{Discharge\ rate}{Rotation\ rate} = \frac{1648.89\ {}^{lit}/_{min}}{1400\ {}^{rot}/_{min}} = 1778\ {}^{lit}/_{rot}$$

This volume is the final delivery volume and represented as:

$$V_6 - V_7 = 1.778\ lit$$

Intermediate pressure:

The optimal intermediate pressure for compressors in series given as.

$$P_{int} = \sqrt{P_{in}.P_{out}} = \sqrt{(0.81325)(7.9)} = 2.535\ bar$$

Now, again we will have a pressure loss of around 0.4 bars at the delivery (so for the first cylinder) and 0.2 bars for the suction (so in the second cylinder). Thus, we can write.

$$P_2 = P_3 = P_{int} + P_{loss} = 2.535 + 0.4$$

$$= 2.935\ bar \quad \text{(First Cylinder discharge)}$$

Similarly, for the second cylinder we add suction loss.

$$P_5 = P_8 = P_{int} - P_{loss} = 2.535 - 0.4$$

$$= 2.335\ bar \quad \text{(Second Cylinder discharge)}$$

Thus after relaxing a few assumptions to bring the system closer to real, we are finally keeping the following assumptions.

- Air is ideal gas - A fair assumption as the pressure range is not dramatic to call for introduction of Z
- The compression process is adiabatic Fair as a higher polytrophic index is not widely used.

Thus, ϒ = 1.41

First cylinder (Lower pressure -LP) design

$$P_2 = P_3 = 2.935\ bar$$

And as determined,

$$P_1 = P_4 = 0.81325\ bar$$

$$\text{and } T_1 = T_4 = 298.15\ K$$

Also as C = 0.05, we can write:

$$Discharge\ rate = \frac{V_3}{V_1} = 0.05$$

$$\mathrm{V_3} = 0.05\mathrm{V_1} \tag{5.1}$$

Thus for process 1-2 (adiabatic)

$$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$$

$$V_2 = V_1 (\frac{P_1}{P_2})^{\frac{1}{\gamma}} = V_1 (\frac{0.81325}{2.935})^{\frac{1}{1.41}} = 0.4025 V_1 \tag{5.2}$$

Similarly, for process 3-4:

$$P_3 V_3^{\gamma} = P_4 V_4^{\gamma}$$

$$V_4 = V_3\left(\frac{P_3}{P_4}\right)^{\frac{1}{\gamma}} = V_3\left(\frac{2.935}{0.81325}\right)^{\frac{1}{1.41}} = 2.485V_3 \quad (5.3)$$

Similarly, for determining the temperatures in process 1-2:

$$T_2 P_2^{\frac{1-\gamma}{\gamma}} = T_1 P_1^{\frac{1-\gamma}{\gamma}}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 298\left(\frac{2.935}{0.81325}\right)^{\frac{0.41}{1.41}} = 433.01K = 159.86\ ℃$$

As in 2-3, the pressure is same and the temperature is also same, we get

$$T_3 = T_2 = 433.01\ K$$

Intercooler (2 to 5)

Now after the exit from the first compressor we will need to cool the exit from 433.01 K to the inlet temperature of the second compressor. For doing this we will use a copper coil (as copper is non-corrosive and has a high thermal conductivity). The length of the intercooler tube can be determined or a standard length of 1 m can be used [11].

We are not assuming a perfect intercooler and an approach temperature of 10 ℃ is used. i.e. the intercooler is able to cool to Ambient +10 C = 298+10 = 308 K. This is the present case of an air-cooled system, as the discharge rates do not justify using a refrigerant. Here, as the cooling takes place at the same pressure, we can calculate the relation between volumes at 5 and 2.

$$\frac{V_5 - V_8}{T_2} = \frac{V_2 - V_3}{T_2} \ => V_5 - V_8 = (V_2 - V_3)\left(\frac{T_5}{T_2}\right)$$

$$= (V_2 - V_3)\left(\frac{308.15}{433.01}\right) = 0.7116 * (V_2 - V_3) \quad (5.4)$$

The intercooler's mechanical design has shown and dealt with later.

Second cylinder (High pressure - HP)

$$P_6 = P_7 = 7.900\ bar$$

And as determined,

$$P_5 = P_8 = 2.335\ bar$$

$$T_5 = T_8 = 308.15\ bar$$

Also as C = 0.05, we can write

$$\frac{V_7}{V_5} = 0.05 \quad => V_7 = 0.05V_5 \quad (5.5)$$

Now, as determined for discharge volume,

$$V_6 - V_7 = 1.778\ lit \quad (5.6)$$

Thus for process 5.6 (adiabatic):

$$P_5 V_5^{\gamma} = P_6 V_6^{\gamma}$$

$$V_6 = V_5 \left(\frac{P_5}{P_6}\right)^{\frac{1}{\gamma}} = V_5 \left(\frac{0.81325}{2.935}\right)^{\frac{1}{1.41}} = 0.4213V_5 \quad (5.7)$$

Similarly, for process 7.8

$$P_7V_7^{\gamma} = P_8V_8^{\gamma}$$

$$V_8 = V_7(\frac{P_7}{P_8})^{\frac{1}{\gamma}} = V_7(\frac{0.81325}{2.935})^{\frac{1}{1.41}} = 2.374V_7 \qquad (5.8)$$

Similarly, for determining the Temperatures, in process 5-6:

$$T_5P_5^{\frac{1-\gamma}{\gamma}} = T_6P_6^{\frac{1-\gamma}{\gamma}}$$

$$T_6 = T_5(\frac{P_6}{P_5})^{\frac{\gamma-1}{\gamma}} = 298(\frac{2.935}{0.81325})^{\frac{0.41}{1.41}} = 439.24\ K = 166.09\ C$$

As in 2-3, the pressure is same and the temperature is also same, we get

$$T_7 = T_6 = 439.24\ bar$$

Now, this temperature will be lowered in an after-cooler before entering the air receiver for storage.

We can compute the volumes as follows, using the 8 equations with us:

Using equation 5.6

$$V_6 - V_7 = 1.1778$$

$$0.4213V_5 - 0.05V_5 = 1.1778 \quad \text{(Substituting eqn. 5.7 \& 5.5)}$$

$$V_5 = 3.172\ lit$$

Using equation 5.7

$$V_6 = 0.4213V_5 = 1.336\ lit$$

Using equation 5.5

$$V_7 = 0.05V_5 = 0.159\ lit$$

Using equation 5.8

$$V_8 = 2.374 \times 0.159 = 0.377\ lit$$

Similarly, using the intercooler eq. 5.4

$$V_2 - V_3 = \frac{V_5 - V_8}{0.7116} = 3.929 lit$$

$$0.4025V_1 - 0.05V_1 = 3.929$$

$$V_1 = 11.1467\ lit$$

Using equation 5.2

$$V_2 = 0.4025 \times 11.1467 = 4.48611\ lit$$

Using equation 5.1

$$V_3 = 0.05V_1 = 0.5571\ lit$$

Using equation 5.3

$$V_4 = 2.485V_3 = 1.385\ lit$$

Thus, we have all the thermodynamic points and can write them as:

Table 5.2. Summary of thermodynamic properties at various points.

| Cycle | Point | P (Bar)- | V (Lit) | T (K) |
|---|---|---|---|---|
| LP | 1 | 0.81325 | 11.147 | 298.15 |
| | 2 | 2.935 | 4.486 | 433.01 |
| | 3 | 2.935 | 0.557 | 433.01 |
| | 4 | 0.81325 | 1.385 | 298.15 |
| HP | 5 | 2.335 | 3.172 | 308.15 |
| | 6 | 7.900 | 1.336 | 339.24 |
| | 7 | 7.900 | 0.159 | 339.24 |
| | 8 | 2.335 | 0.337 | 308.15 |

Thus, the two compressors have the volume:

Low Pressure: 11.15 liter

High Pressure: 3.17 liter

After-cooler

After air leaves the

$$\frac{FAD}{T_{dis}} = \frac{V_6 - V_7}{T_7} = FAD = (V_6 - V_7)\left(\frac{T_{dis}}{T_7}\right)$$

$$= (1.336 - 0.159)\left(\frac{308.15}{433.01}\right) = 0.82628\,\frac{lit}{rot} = 1156.79 lit/min$$

Thus, the free air delivery to the storage tank is around 1150 lit/m. This is acceptable as the 15% margin is to secure against any small leakages or losses, to be able to deliver the required 1000 lit/m still.

Thus, the final flow rate obtained from the system = 1156.79 lit/m

### 5.2.2. Air Receiver Volume Design

$$V = \frac{0.25 q_c T_o}{f_{max}(P_{u-} P_L) T_1} \tag{5.9}$$

Where

$q_c$ = compressor capacity = 1156.79 lit/min

$P_1$= compressor inlet pressure = 0.81325 bar

$P_2$= compressor capacity = 298.15 K

$f_{max}$= maximum cycle frequency = 1 cycle/min

$(P_{U-} P_L)$= Pressure difference between loaded and unloaded compressor = 0.4

$T_0$ = Compressed air temperature out of the selected compressor = 308.15 K

The range $P_U$-$P_L$ has been estimated at a 5% difference from the max discharge pressure of 7.9 bars of the compressor, so $P_U$-$P_L$ = 0.05 (7.9) = 0.395 bars, Thus, it is approximately taken as 0.4 bars to give a regulation range of 7.5 bars to 7.9 bars.

Thus, V: 607.693 lit

= 160.535 gallon

Now, as it has seen from the next standard vessel size is 200 gallon.

The dimensions of the receiver are 30"x72". This vertical receiver should be able to handle the flow rate desired; this obtained from industrial design of Manchester tanks [18].

## 5.3. DRYER DESIGN

As the compression, process compresses the air but not the accompanying moisture, the amount of water that produced is very high, the after-cooler does cause a lot of this water to condense and if a dew point of +5 C desired, as it is a standard for my applications [13].

We will need a dryer for the required dew point and the flow rates, a small refrigerant dryer will be a perfect choice as there will be no need for regeneration or replacement and the dryer will reduce the humidity to the required level and the water will be removed from the system, This is a standard device and can be easily purchased from many sources. The operating pressure of around 7.5 bars selected. Similarly, a flow of 1000 lit/min gas taken as the parameters to select the refrigerant dryer.

## 5.4. FILTERS

An oil filter added after the receiver and before the dryer. This will help in removing any oil/grease from the air before it goes into the dryer so that the dryer performance increases. Similarly, a final dust filter added to ensure that the supply air is of proper quality. If the desired quality is even more stringent then further set of filters added right before delivery. Now, for the filters, the pressure drop will increase over time and these will need to replace after a certain time to ensure proper flow.

## 5.5. PRESSURE DROPS CALCULATIONS

First, we will consider the following pressure drops in the following equipment [16]

- Oil Filter 0.09 bar
- Refrigerant dryer 0.08 bar
- Dust Filter 0.08 bar
- Total 0.25 bar

In addition, with the subsequent increase of pressure drops in the filters, we consider that this pressure drop doubles over time, for making an estimate on the piping possible.

Thus, pressure drop in equipment outside compressor units = 0.5 bar

Thus, as we have considered that the pressure varies from 7.5 bars to 7.9 bars, we will base our calculations on the lower 7.5 bar, to be able to determine the piping length that will help in keeping the pressure of final delivery at 6 bars. That means that we can have a pressure drop off (7.5-6) = 1.5 bars overall. Subtracting the pressure drop in the equipment will give us the pressure drop that we can sustain in the pipes.

Thus the pressure drop that can take place in pipes and coolers is:

$$\Delta_p = 1.5 - 0.5 = 1\ bar = 100000\ Pa$$

Now, if we want to calculate the equivalent length (including skin friction (pipes) and from friction (fittings and bends)), then we can use the formula [19].

$$\Delta_p = f_D \frac{L}{D} \frac{\rho v^2}{2} \qquad (5.10)$$

Where

$\Delta p$= pressure loss

$f$ =the Darcy friction factor

$L$ =length of pipe or pipe part

$D$= inner diameter of the pipe

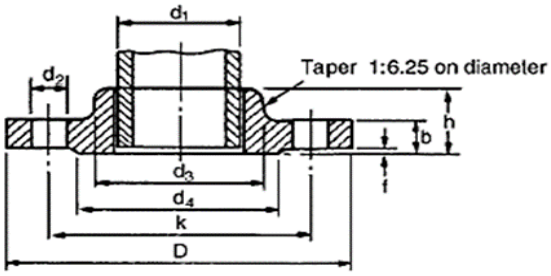
$\rho$= density of fluid

$V$ =flow velocity

Now for determining the D, we can use the nominal pipe sizing formula as:

$$d_{optimal} = 260 G^{0.52} \rho^{-0.37} \qquad (5.11)$$

$$\text{and } \rho = \frac{PM}{RT} \qquad (5.12)$$




Figuer 5.4. Steel slip on boss flange for welding.

Table 5.3. General Flange design (in mm) [10].

| **Nominal Size** | **Pipe** | **Flange** | | | **Raised face** | | **Bolting** | **Drilling** | | | **Boss** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | $d_1 \approx$ | D | b | h | $d_4$ | f | | No. | $d_2$ | k | $d_3$ |
| 10 | 17.2 | 75 | 12 | 20 | 35 | 2 | M10 | 4 | 11 | 50 | 25 |
| 15 | 21.3 | 80 | 12 | 20 | 40 | 2 | M10 | 4 | 11 | 55 | 30 |
| 20 | 26.9 | 90 | 14 | 24 | 50 | 2 | M10 | 4 | 11 | 65 | 40 |
| 25 | 33.7 | 100 | 14 | 24 | 60 | 2 | M10 | 4 | 11 | 75 | 50 |
| 32 | 42.4 | 120 | 14 | 26 | 70 | 2 | M12 | 4 | 14 | 90 | 60 |
| 40 | 48.3 | 130 | 14 | 26 | 80 | 3 | M12 | 4 | 14 | 100 | 70 |
| 50 | 60.3 | 140 | 14 | 28 | 90 | 3 | M12 | 4 | 14 | 110 | 80 |
| 65 | 76.1 | 160 | 14 | 32 | 110 | 3 | M12 | 4 | 14 | 130 | 100 |
| 80 | 88.9 | 190 | 16 | 34 | 128 | 3 | M16 | 4 | 18 | 150 | 110 |
| 100 | 114.3 | 210 | 16 | 40 | 148 | 3 | M16 | 4 | 18 | 170 | 130 |
| 125 | 139.7 | 240 | 18 | 44 | 178 | 3 | M16 | 8 | 18 | 200 | 160 |
| 150 | 168.3 | 265 | 18 | 44 | 202 | 3 | M16 | 8 | 18 | 225 | 185 |
| 200 | 219.1 | 320 | 20 | 44 | 258 | 3 | M16 | 8 | 18 | 280 | 240 |
| 250 | 273 | 375 | 22 | 44 | 312 | 3 | M16 | 12 | 18 | 335 | 295 |
| 300 | 323.9 | 440 | 22 | 44 | 365 | 4 | M20 | 12 | 22 | 395 | 355 |

Table 5.4. Properties at inlet and outlet of equipment.

| **Cycle** | **Point** | **P (bar)** | **T (K)** | **M (g/moll)** | **R** | **ρ** |
|---|---|---|---|---|---|---|
| LP | In(4-1) | 0.81235 | 298.15 | 28.82 | 8.314 | 0.94553 |
| | Out(2-3) | 2.93469 | 433.01 | 28.82 | 8.314 | 2.34936 |
| HP | In(8-5) | 2.33469 | 308.15 | 28.82 | 8.314 | 2.62635 |
| | Out(6-7) | 7.9 | 439.238 | 28.82 | 8.314 | 6.23463 |
| After cooler | Out(9) | 7.9 | 308.15 | 28.82 | 8.314 | 8.88687 |

As the flow out of the after-cooler is obtained as =19.28 lit/s [10]

Thus, the mass flow rate

$$G = \rho Q = (8.8868)(19.28)/1000 \;=>\; G = 0.171336676\, kg/s$$

Using the formula for optimal diameter, we can find the pipe sizes by finding the optimal diameter, and then choosing the nominal dia., with the upper size from the charts.

Table 5.5. Inlet and outlet nominal diameter selection.

| **Cycle** | **Point** | **P** | **G** | $\mathbf{D_{OPT}}$ **(MM)** | **Chosen D** |
|---|---|---|---|---|---|
| LP | In(4-1) | 0.94553 | 0.17134 | 106.066 | 125 |
| | Out(2-3) | 2.34936 | 0.17134 | 75.7398 | 80 |
| HP | In(8-5) | 2.62635 | 0.17134 | 72.6801 | 80 |
| | Out(6-7) | 6.23463 | 0.17134 | 52.7832 | 65 |
| After cooler | Out(9) | 8.88687 | 0.17134 | 46.2955 | 50 |

Thus, the following pipe sections are determined:

Table 5.6. Intermediate pipe connection diameters.

| **Pipe No** | **From** | **To** | **Diameter (Mm)** | **Discussion** |
|---|---|---|---|---|
| 1 | open | LP in | 125 | This intake line and the pressure drop here can be neglected |
| 2 | LP out | HP in | 80 | This line has intercooler but the piping will be considered to have negligible pressure drop |
| 3 | HP out | Cooler in | 65 | This line can have a pressure drop but as will be very small the pressure drop can be neglected |
| 4 | Cooler out | Everywhere | 50 | This is the main piping line and needs to be sized to keep the decided pressure drops in limits |

Now, again as the flow Q in this pipe (After after-cooler) is = 0.01928 $m^3$/s

Now, with the diameter of 50 mm = 0.05 m, we can calculate the velocity as:

$$v = \frac{Q}{A} = \frac{Q}{\frac{\pi d^2}{4}} = 9.82\ m/s$$

The pressure drop:

$$\Delta_p = f_D \frac{L}{D} \frac{\rho V^2}{2}$$

Where,

$\Delta_p =$ Pressure loss

$L =$ Length of pipe or pipe part

$f$ = The Darcy friction factor

$D$ = Inner diameter of the pipe

$v =$ flow velocity

$\rho$ = Density of the the fluid

Now, the Reynolds number Re can be determined as

$$Re = \frac{\rho v d}{\mu} \tag{5.13}$$

Where

$\rho$ = Density = 8.98 kg/m$^3$

$v$ = Velosty = 9.82 m/s

$D$ = Diamter = 0.05 m

$\mu$ = Viscocity = 1.89E-05 kg/ms (at 308.15 K)

Thus,

$$Re = \frac{\rho v d}{\mu} = 2.31E + 05$$

As Re>2000, it is in the turbulent regime, thus the friction factor can be written as [19].

$$\frac{1}{\sqrt{fD}} = -2\, log\left(\frac{2.51}{Re\sqrt{fD}}\right) \tag{5.14}$$

In which

$f$ = The Darcy friction factor
$e$ = Roughness of the pipe
$D$ = Inner diameter of the pipe
$Re$ = The Reynolds number

We can start with an initial guess of $f_D$ =0.5, and use the RHS to compute FD repeatedly until convergence:

Table 5.7. Convergence calculation of friction factor.

| **FD** | **FD'** |
|---|---|
| 0.5 | 0.01079 |
| 0.01079 | 0.01578 |
| 0.01578 | 0.01515 |
| 0.01515 | 0.01521 |
| 0.01521 | 0.01521 |
| 0.01521 | 0.01521 |

Thus $f_D$ = *0.015206568*

Now, in the pressure drop equation, we know all the terms and can determine the equivalent length.

$$\Delta_p = f_D \frac{L}{D} \frac{\rho v^2}{2}$$

$$L_{eq} = \frac{2D\Delta_p}{f_D \rho v^2} = \frac{2(0.05)(10^5)}{0.015(8.89)(9.8)^2} = 767.5\ m$$

Now the equivalent length of each fitting summarized as,

Table 5.8. Equivalent length of pressure drop of various pipe fitting [16].

| EQUIVALENT LENGTH IN METERS | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **INNER PIPE DIAMETER IN MM (D)** | | | | | | | | | | | |
| **Component** | 25 | 40 | 50 | 80 | 100 | 125 | 200 | 250 | 250 | 300 | 400 |
| **Ball Valve (Full Flow)** | 0.3 5 | 0.5 8 | 0.6 10 | 1.0 16 | 1.3 20 | 1.6 25 | 1.9 30 | 2.6 40 | 3.2 50 | 3.9 60 | 5.2 80 |
| **Diaphragm Valve Fully Open** | 1.5 | 2.5 | 3.0 | 4.5 | 6 | 8 | 10 | - | - | - | - |
| **Angle Valve Fully Open** | 4 | 6 | 7 | 12 | 15 | 18 | 22 | 30 | 36 | - | - |
| **Poppet Valve** | 7.5 | 12 | 15 | 24 | 30 | 38 | 45 | 60 | - | - | - |
| **Flap Check Valve** | 2.0 | 3.2 | 4.0 | 6.4 | 8.0 | 10 | 12 | 16 | 20 | 24 | 32 |
| **Elbow R=2d** | 0.3 | 0.5 | 0.6 | 1.0 | 1.2 | 1.5 | 1.8 | 2.4 | 3.0 | 3.6 | 4.8 |
| **Elbow R=D** | 0.4 | 0.6 | 0.8 | 1.3 | 1.6 | 2.0 | 2.4 | 3.2 | 4.0 | 4.8 | 6.4 |
| **$90^o$ Angle** | 1.5 | 2.4 | 3.0 | 4.5 | 6.0 | 7.5 | 9 | 12 | 15 | 18 | 24 |
| **Tee Through-Flow** | 0.3 | 0.4 | 1.0 | 1.6 | 2.0 | 2.5 | 3 | 4 | 5 | 6 | 8 |
| **Tee Side-Flow** | 1.5 | 2.4 | 3.0 | 4.8 | 6.0 | 7.5 | 9 | 12 | 15 | 18 | 24 |
| **Reducing Nipple** | 0.5 | 0.7 | 1.0 | 2.0 | 2.5 | 3.1 | 3.6 | 4.8 | 6.0 | 7.2 | 9.6 |

As an estimate, we can consider that each equipment has 2 90' angle bends (1 at inlet and 1 at exit). This assumption verified during the drawings. This will give us 4*2 = 8 bands, as there are 4 types of equipment (compressor, filter1, dryer, filter 2). Now a further number of bands will be present to align with the room and to reach the service point. We can assume that 12 bands are present apart from the equipment, thus we have 20 90' bends. Now for valves, we will choose gate (poppet) valves as it offers greater flow control and is easier to install and maintain. These can be assumed to be associated 1 per equipment and 6 in extra, so a total of 10 gate poppet valves), thus we can summarize the fittings and the corresponding pressure drop equivalent length for a 50 mm diameter pipe as follows.

Table 5.9. Equivalent pressure drop of calculated pipe fittings.

| Sr.No | Fitting | Nos. = N | Eq. Length | Total Loss (M) |
|---|---|---|---|---|
| 1 | 90' Bends | 20 | 3 | 60 |
| 2 | Gate Valves | 10 | 15 | 150 |
| | | | **Total** | 210 |

Thus, from the allowable length, we will reduce that equivalent to form (fittings) friction to get the length of pipe that used.

$$L_{total} = L_{skin} + l_{form}$$

as $L_{form} = 210\ m$

and $L_{total} = 747\ m$

We get $L_{skin} = 557.5\ m$

Now allowing for a 20% margin of error, so they any loss in flow still makes the desired pressure level, we can estimate the installable pipe length as:

$$L_{pipe} = 0.8L_{skin} = 0.8(557.5) = 446.0\ m$$

Thus, a total of around 450 m of piping installed without affecting the pressure ratings, and thus the assumed design pressure drop is easily suiting the need and is safe from a delivery pressure point of view.

## 5.6. POWER REQUIREMENT OF THE COMPRESSOR

$$W_1 = \frac{RT_1\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 3855.92\,J/mol$$

$$So, P = \frac{Wm}{M} = \frac{(3855.9)(171.3)}{28.82} = 22923.7\,W$$

$$W_2 = \frac{RT_5\gamma}{\gamma - 1}\left[\left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 3748.09\,\frac{J}{mol}$$

$$So, P = \frac{Wm}{M} = \frac{(3748.09)(171.3)}{28.82} = 22282.6\,W$$

This work supplied in the form of electricity, the total power required to operate the compressor.

$$W = W_1 + W_2 \;= 45205.3W \quad = 45.2K$$

Assuming a 95% efficiency the supply power will be

$$P = \frac{W}{Efficiency} = \frac{45.2}{0.95} \;= 47.59kW$$

Thus, 47.59 $kW$ energy is required to operate the designed compressors.

### 5.7. HEAT LOADS

#### 5.7.1. Intercooler

The intercooler operates at the pressure of that at pt. 2 of LP - pressure loss of the discharge valve or in effect, the pressure at the intercooler = Pint = 2.535 bar at this pressure, the temperature of the air in the intercooler changes from 433.01 to 308.15 *K*.

The mean temperature of this change is = 370.5799702 *K*
At this temperature the specific heat at constant pressure is.

$C_p$ = *1010 J/kg* [13].

Thus the heat loss from the intercooler is:

$$q_{intercooler} = \dot{m}C_P\Delta T = (0.171)(1020)(433.01 - 308.15) = 21607W$$

$$q_{intercooler} = 21.6kW$$

#### 5.7.2. After-Cooler

Similarly, the after-cooler reduces the temperature from 439.24 to 308.15 K, This has done at a pressure of 7.9 bars; again, we can find the $C_p$ value for this condition as:

$C_p$= 1020 J/kg (At mean temp of = 373.69 K)

Thus the heat loss from the after-cooler is:

$$q_{aftercooler} = \dot{m}C_P\Delta T = (0.171)(1020)(439.24 - 308.15) = 22909.5W$$

$$q_{aftercooler} = 22.9kW$$

### 5.7.3. Refrigerant Dryer

For simplicity, we will consider only the cooling part of the refrigerant and assume that the refrigerant cools all the way to the dew point of 5 C. This will help us estimate the heat radiated from the refrigeration radiator. Again, the pressure here is around 7.9 bar, and the temperature lowered from 308.15K to 278.15K. This will give us the $C_p$ value as:

$C_p$= 1020 J/kgK (At mean temp of = 373.69 K)

Thus the heat loss from the after-cooler is:

$$q_{dryer} = \dot{m}C_P\Delta T = (0.171)(1020)(308.15 - 278.15) = 34895.27W$$

$$q_{dryer} = 3.5kW$$

Thus the total heat loss that too made up by the air-cooling is:

$$q_{heat} = q_{intercooler} + q_{aftercooler} + q_{dryer} = 48.0kW$$

## 5.8. VENTILATION REQUIREMENT OF THE ROOM

The compressor housing and the allied equipment are all air-cooled and thus, we will need to determine the amount of air that needs to circulate to enable this total cooling. The heat that need to cool and summarized as:

The difference between total energy to compressor - energy transmitted to shaft: As determined earlier a 95% transmission efficiency was consider for the compressor. Thus, the power lost to the atmosphere is the difference between the power supplied and the power transmitted as:

$$q_{loss1} = 47.59 - 45.2 = 2.38kW$$

The losses determined at the coolers.

$$q_{loss2} = q_{heat} = 48kW$$

Thus, the total loss comes to around

$$q_{total} = q_{loss1} + q_{loss2} = 50.39kW$$

Assuming a 5% error in calculation, we can design the ventilation for a heat loss need

$$q = \frac{50.39}{0.96} = 53.04$$

Now this power needs to take away by the circulating air this power has written as.

$$q = \dot{m} C_p \Delta T$$

$$\dot{m} = \frac{q}{C_p \Delta T}$$

Now, here the change in temperature has taken as 10 K (the allowable rise in the ventilation temperature).

$C_p = 1.007\, k_J/k_g K$ (At 1 bar and 25C)

Thus,

$$\dot{m} = \frac{\rho}{C_p \Delta T} = \frac{53.04}{(1.007)(10)} = 5.27\, kg/s$$

For a density of 1.2 kg/m3, the flow rate of this ventilation air is.

$$Q = \frac{\dot{m}}{\rho} = \frac{5.27}{1.2} = 4.39\, m^3/s$$

Thus, the ventilation flow for covering the cooling needs is around 4.4 $m^3$/s. This air will need to be circulate in the room.

## 5.9. MECHANICAL DESIGN

For actually making the compressor we will need to consider factors and design [12].

- Crankcase – Material
- Crankshaft – Material
- Main Bearings
- Piston
- Valves
- Tank sizing
- Lubrication
- Protective parts.
- Motor

### 5.9.1. Crankcase

It is generally made of cast iron and houses the crankshaft. Aluminum is also used, but cast iron will be preferred here as the cylinder size is not so weight is not a great consideration. The crankcase has the gear and an oil sump, which has all the oil circulating around.

### 5.9.2. Crankshaft

They are made of forged steel and need to be extremely reliable. Surface finishing is also high for the material.  The crankshaft drives the cylinder in the piston, so has to be structurally strong and as it is a rotating part, it generally balanced by counterweights.

### 5.9.3. Main Bearings

This is the bearing on the crank and the drive. They are made of steel-backed Babbitt, bronze, or aluminum. This is the main bearing is designed to have a long life against fatigue loading.

### 5.9.4. Piston And Cylinder

They are usually made of cast iron or aluminum. In our case, we are dealing with air, which will not result in a lot of corrosion (some amount of the moisture is ok), has a temperature of 499K maximum and the pressure rating is high at 6 bars. High-grade Aluminum will be a better material for this condition. 3 piston rings can be used to support the piston and prevent leakage and lubricant infiltration. Now we can dimension the internal of the piston as the length of the cylinder is increased the mean speed of the piston increases. This causes faster wear and increases chances of failure. On the other hand, a higher diameter causes a greater amount of force on the piston and thus causes material limitations. As a tradeoff, the L/D ratio generally selected for a particular setup and the design completed while keeping the mean speed limitations in mind. Now, mentions and analysis a cylinder with 1.1 L/D ratio. This value is acceptable and is widely used as it also has seen from [13].
Thus,

$$\frac{L}{D} = \frac{L_1}{D_1} = \frac{L_2}{D_2} = 1.1$$

Also, as before N = 1400 rpm. Now as,

$$D_1 = \sqrt{\frac{4(V_1 - V_3)}{\pi L_1}}$$

$$\frac{\pi D_1^2}{4} L_1 = (V_1 - V_3)$$

$$\frac{\pi D_1^3}{4}\left(\frac{L_1}{D_1}\right) = (V_1 - V_3)$$

$$D_1 = \left(\frac{4(V_1 - V_3)}{\pi\left(\frac{L_1}{D_1}\right)}\right)^{\frac{1}{3}} = 0.231\ m = 230.57\ mm$$

Similarly,

$$D_2 = \left(\frac{4(V_5 - V_7)}{\pi\left(\frac{L_2}{D_2}\right)}\right)^{\frac{1}{3}} = 0.152\ m = 151.66\ mm$$

$$L_1 = 1.1D_1 = 0.254\ m = 253.62$$

$$L_2 = 1.1D_2 = 0.167\ m = 166.83$$

Thus, the piston-cylinder dimensions summarized as

Table 5.10. Piston cylinder dimension.

| | LP | HP |
|---|---|---|
| **Length (L)** | 253.62 | 166.83 |
| **Dia (D)** | 230.57 | 151.66 |

Here we also note that the LP and the HP cylinder will both have mounted on the same shaft. This means that the driving shaft needs to be designed such that the two lengths are matched. This should not be a problem as the variation in length is not very high, and the variation in the driving crank radius, will only be half this value, so the driving shaft should not be a tough task to manufacture, and thus the design is acceptable.

### 5.9.5. Valves

The suction and the discharge valves will need to be designed to enable proper function for the life of the system. Many standard valves has chosen, keeping in mind that standard velocities through the valve are between 45 m/s to 60 m/s. nowhere we should also discuss the connection of the valves. At the inlet of the first compressor, we have an air filter through which we take in the air. The outlet of the first compressor connects to the intercooler, which then continues into the second compressor. Then the output of the second compressor connects to the after-cooler and then the tank. For sizing the valves, we can use the sizes of the pipes determined earlier.

Table 5.11. Valve sizes at various inlet and outlet.

| Cycle | Valves | ρ | G | Mm | Chosen D |
|---|---|---|---|---|---|
| **LP** | In | 0.94553 | 0.17134 | 106.066 | 125 |
| | Out | 2.34936 | 0.17134 | 75.7398 | 80 |
| **HP** | In | 2.62635 | 0.17134 | 72.6801 | 80 |
| | Out | 6.23463 | 0.17134 | 52.7832 | 65 |
| **After Cooler** | In | 2.62635 | 0.17134 | 72.6801 | 65 |
| | Out | 8.88687 | 0.17134 | 46.2955 | 50 |

### 5.9.6. Tank sizing

As determined earlier will use a pressure tank of 200 gal capacity and dimensions of 30” x 72”. The drawing illustrates the receive [18].

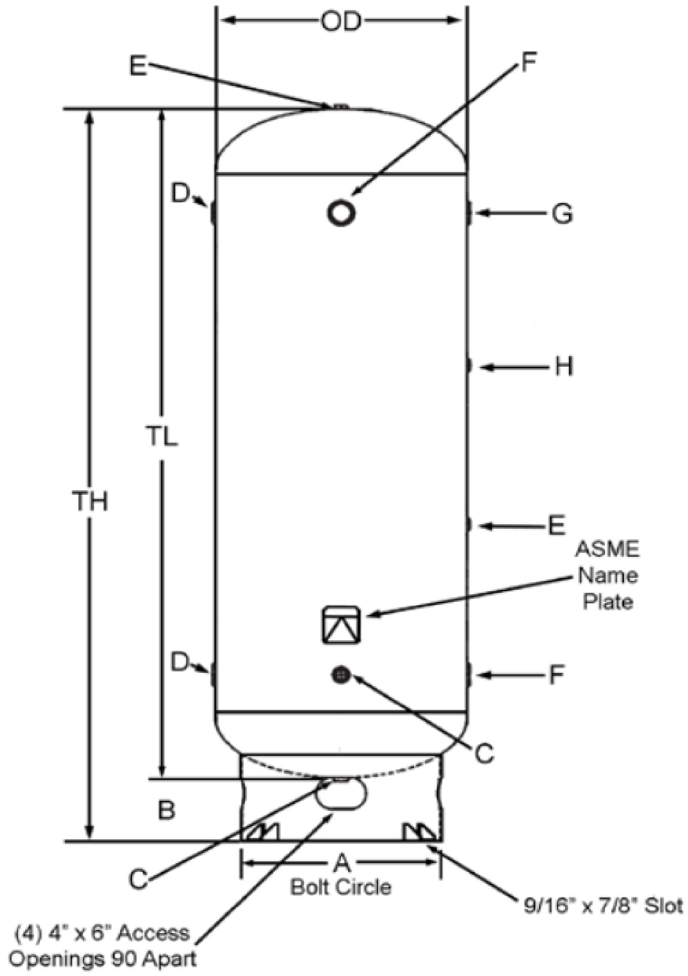



Figure 5.5. Dimension of tank receiver [18].

For the pressure relief safety, we can use a rupture disc rated at a pressure of 1.2 times that of the storage pressure i.e. at 9.5 bars. This will ensure that as the pressure reaches this value the rupture disc bursts. We can also use a relief valve, by using the standard device from any vendor rated at this pressure of 9.5 bars.

### 5.9.7. Lubricants

Proper lubricant needs to be use that can withstand such high pressure, also such that it does not oxidize and is stable in contact with moisture. An extreme pressure grease or a silicone type lubricant can be used.

### 5.9.8. Protective parts

Pressure relief system or leak proofing can be incorporate, as we are not dealing with any condensation or liquefaction after the compressor, we will not need to be careful about flooding and fluid flow back. Therefore, a pressure relief system in case of over-pressurization will be good as discussed in the tank sizing.

Protection against dust or foreign particle is necessary as it will otherwise corrode the internal of the piston-cylinder, and make the oil and lubricants dirty, and is very undesirable. So a good filter system at the inlet advised.

### 5.9.9. Motors

Many standard motors can be use, as determined the motor must transmit around 48 kW at 1400 rpm. A higher speed motor can be use, and the wheels can be size such that the speed reduced to the required 1400 rpm.

## PART 6

## DRAWING

All original drawings presented here; the files implemented in Solid works

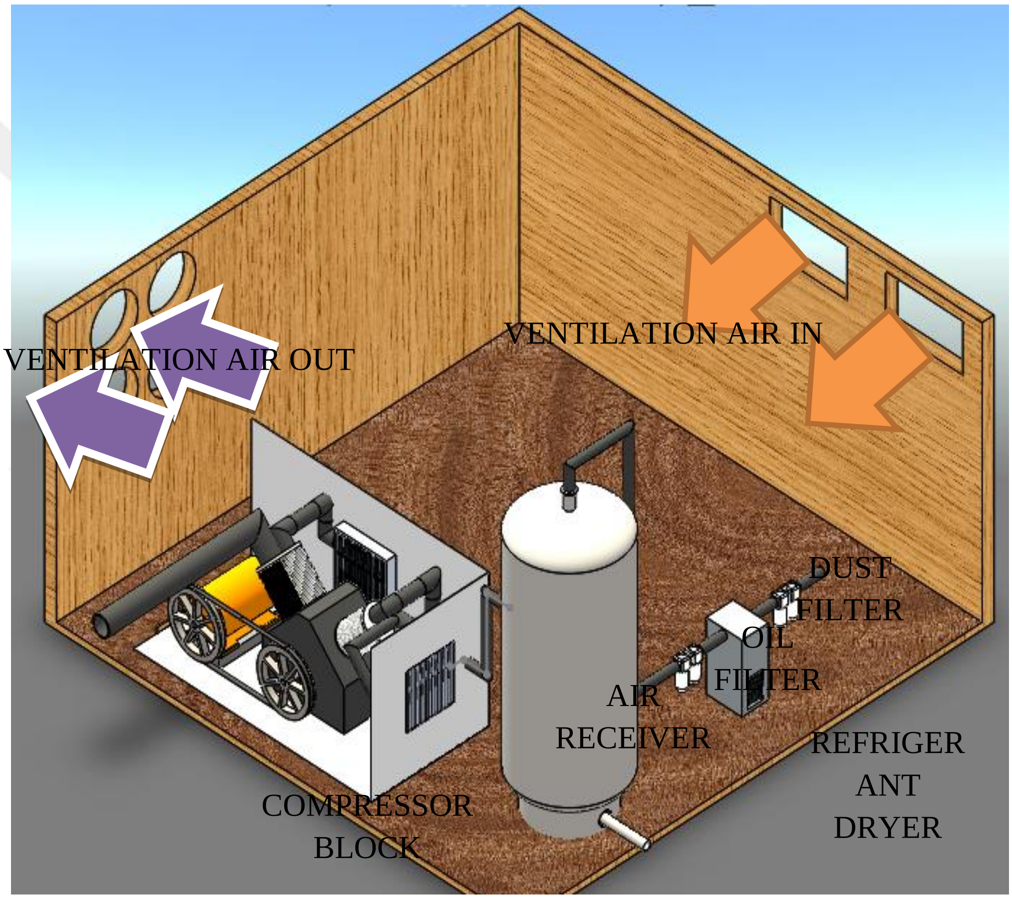



Figure 6.1. Compressor room with all parts.

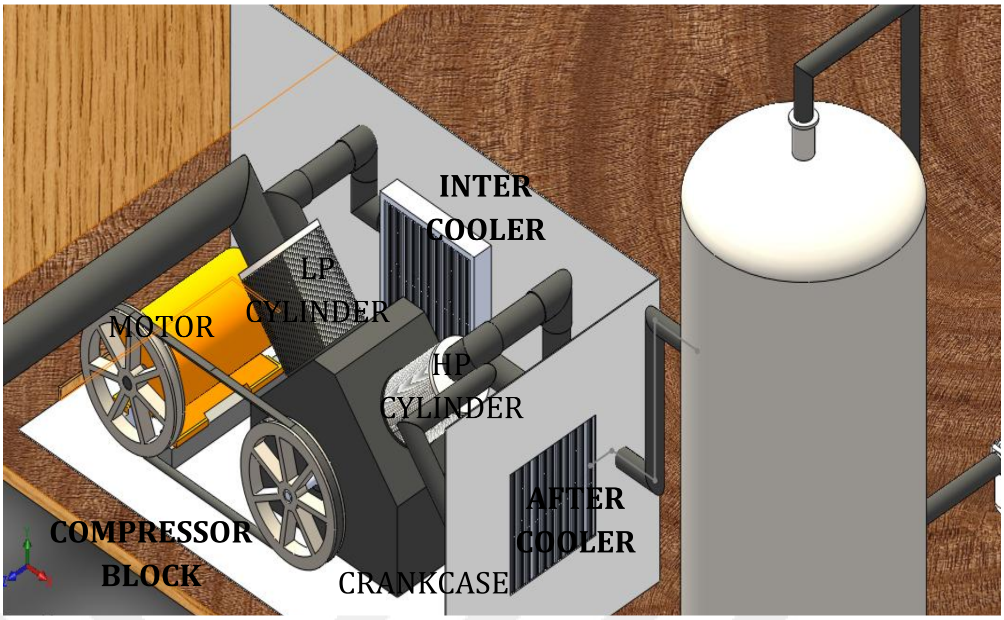



Figure 6.2. Compressor block.

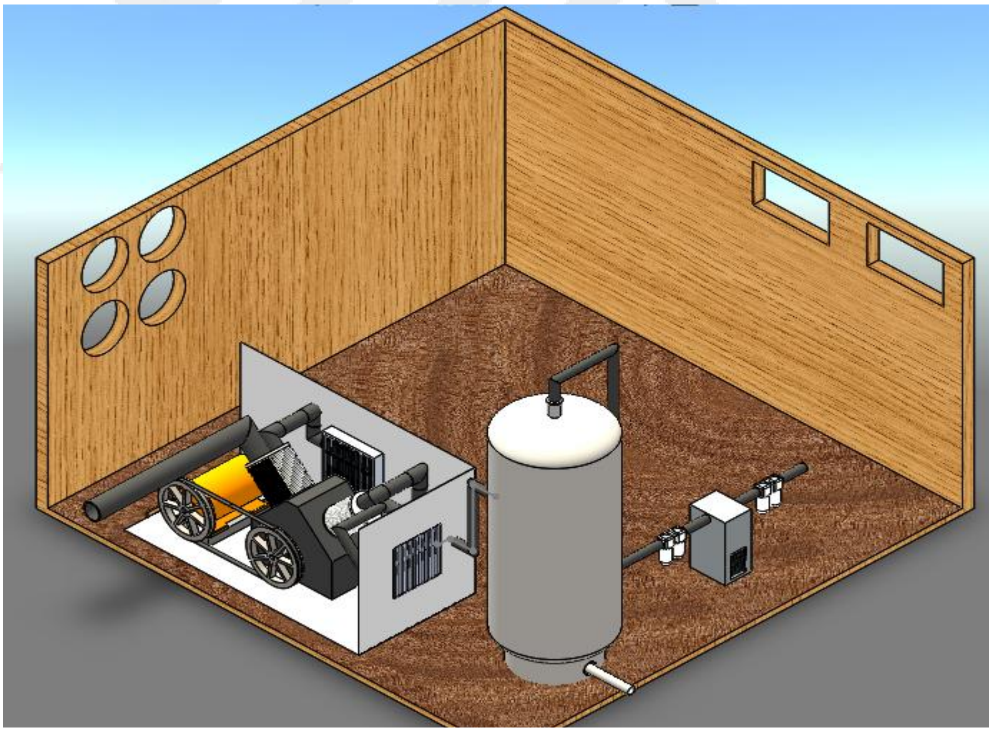


Figure 6.3. Compressor room.

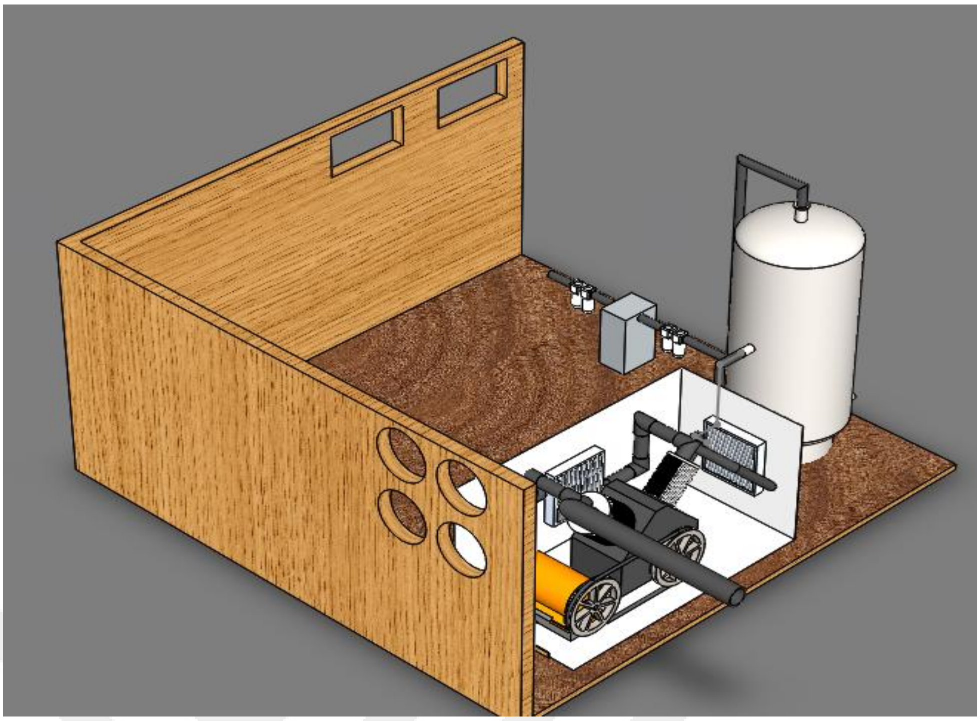


Figure 6.4. Another view of compressor room.

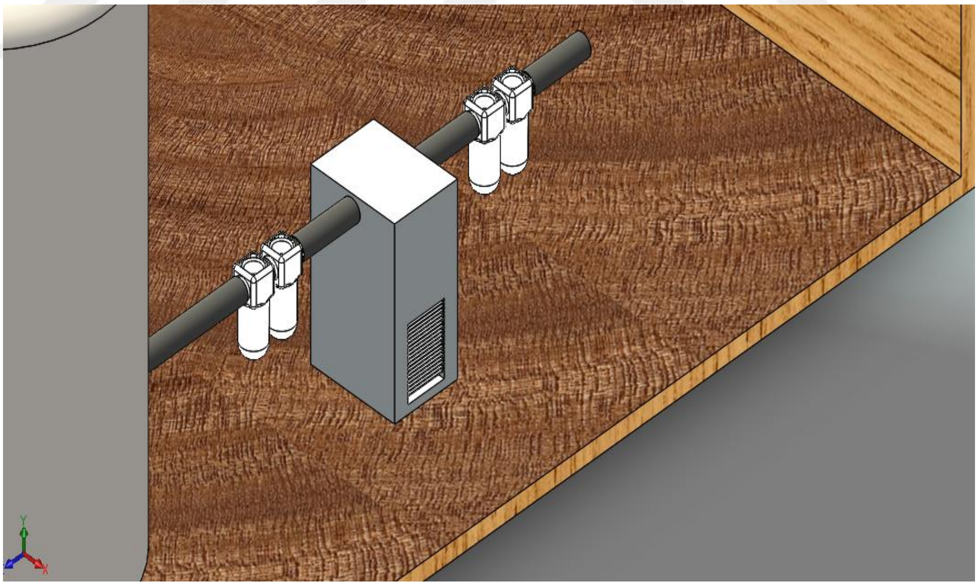


Figure 6.5. Oil filter, refrigerant dryer and dust filter.

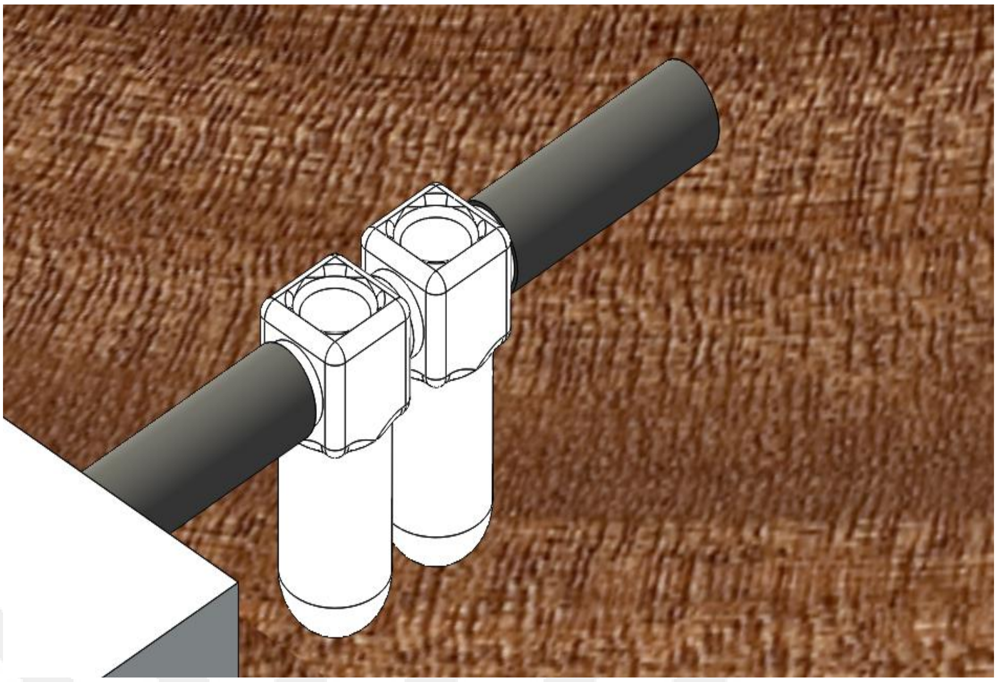


Figure 6.6. Dust filter and exit pipe.

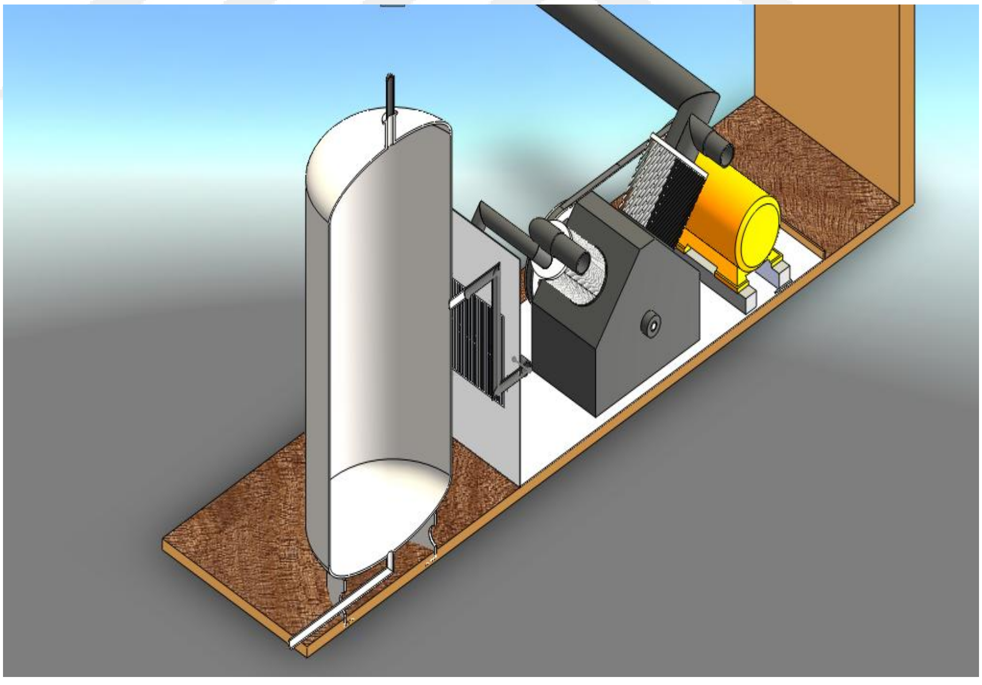


Figure 6.7. Cross section view of the receiver tank.

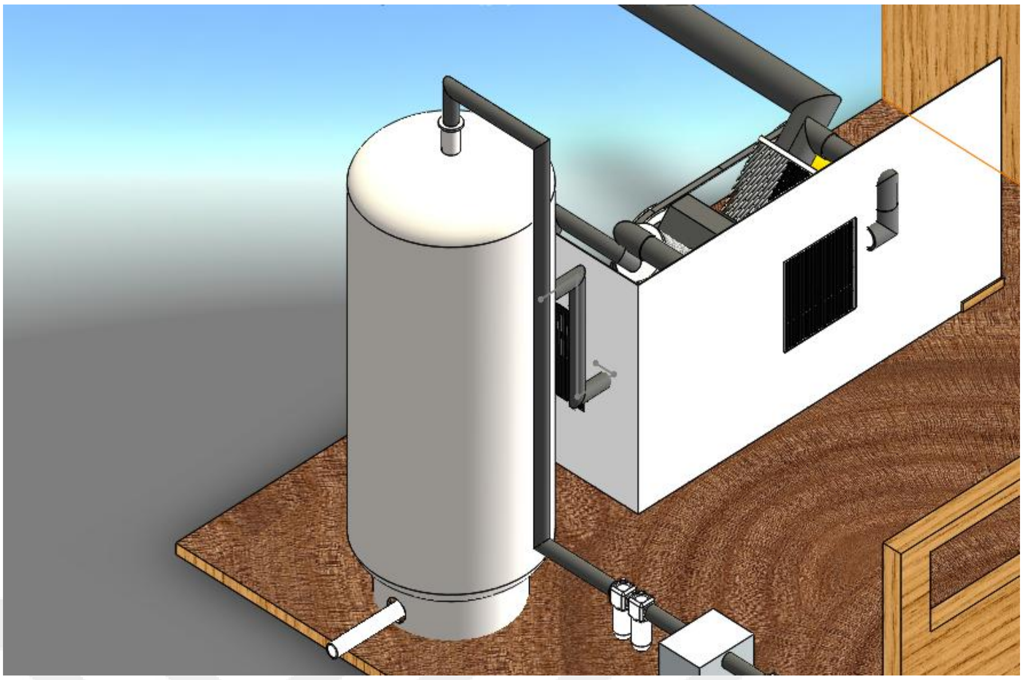


Figure 6.8. Receiver tank.

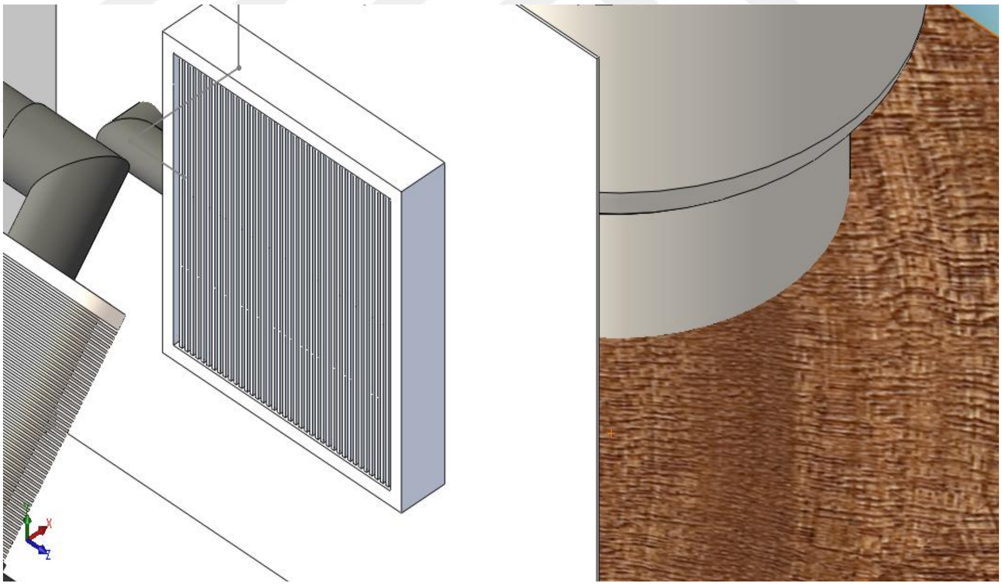


Figure 6.9. After-cooler schematic.

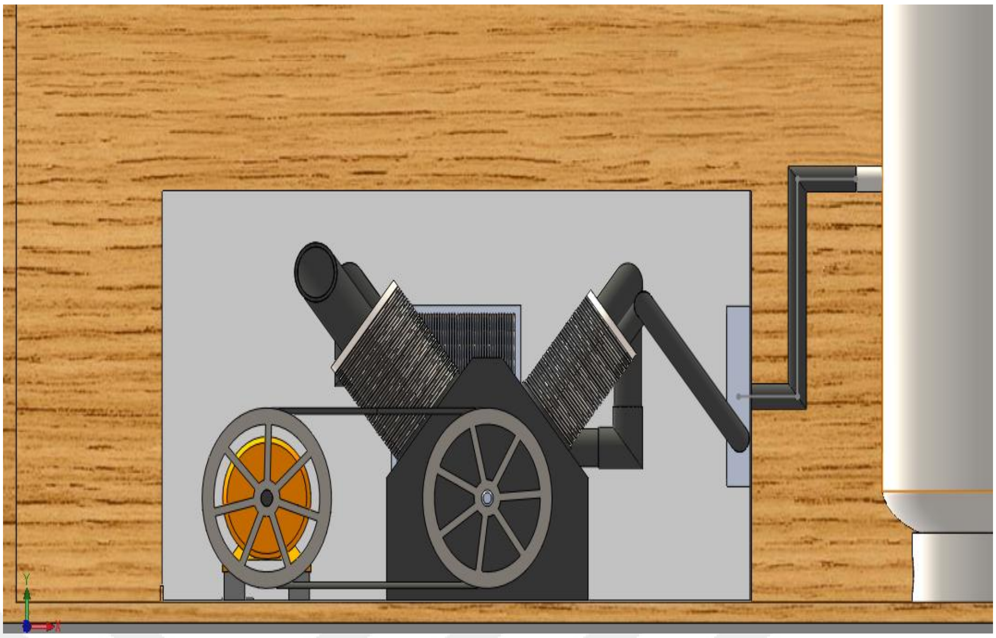


Figure 6.10. Compressor block front view.

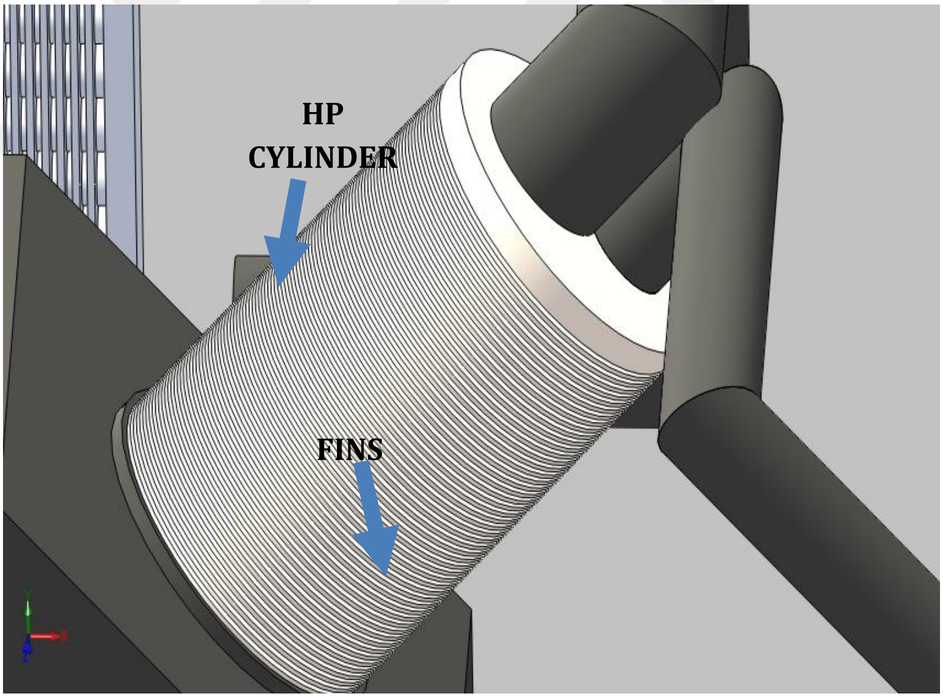



Figure 6.11. HP cylinder with fins.

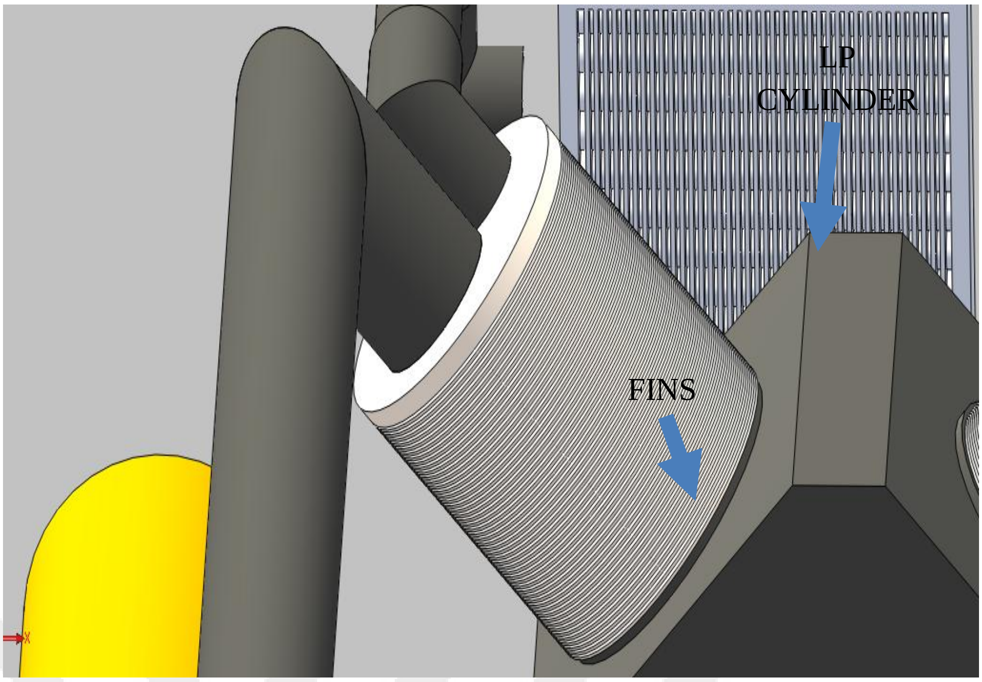



Figure 6.12. LP cylinder with fins.

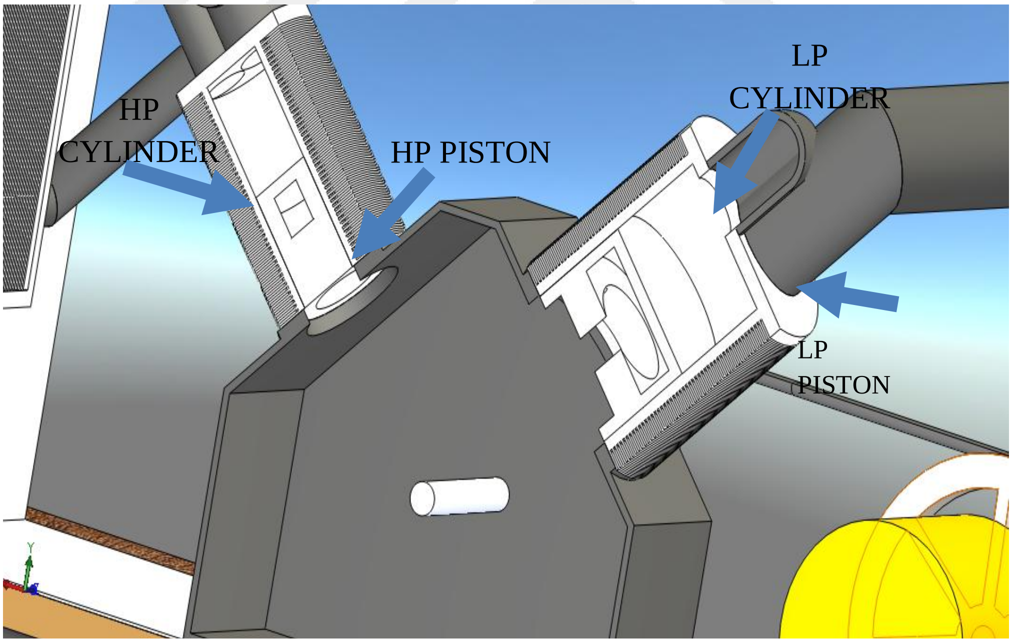



Figure 6.13. Cross-Sectional view of cylinder and crank case.

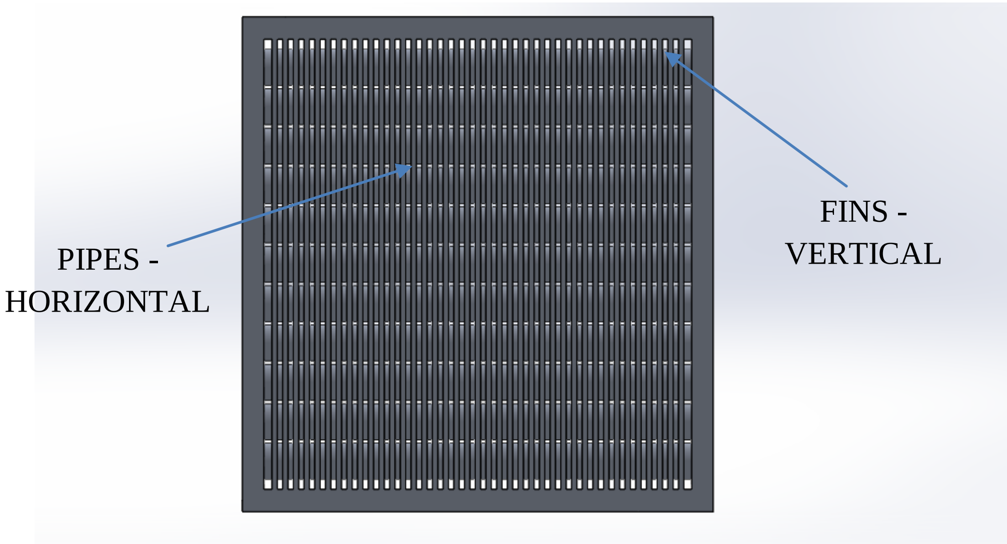



Figure 6.14. Intercooler and after-cooler schematic with pipes and fins.

# PART 7

# CONCLUSION

The compressor is largely used and the term refers to a large variety of device types that can be used for compression, of all, the reciprocating compressor is probably the easiest to understand and is employed the most where ‘muscle’ is required.

The thermodynamics of a compressor, could be understand and so can the mechanical design and the entire assembly. However, practical implementation of a compressor device is slightly different, mainly in that the information provided, as the starting point is very less from a literary add academic point of view. This is what reflects in the process of design and mechanical engineering, and this is where assumption, approximations, and thumb rules, play a large role.

The report tries to not only describe, each part of thermodynamics and design principals, mostly taken from current industrial practices, but also implements a complete design of the compressor and the allied systems, with the help of iterative designing principals.

The assumptions on flow, stages, types and pressures first made, and a very primitive design (single stage) made. By understanding, the temperatures that will be involved an engineering decision made to use a dual stage compressor and its design principal and intercooler design carried out. With a rough outline of the properties derived and everything being acceptable in the range, for the dual stage design, a final design made by relaxing all the assumptions to the extent possible. Proper clearance ratios, pressure drops, work principals and efficiencies have taken into consideration. Furthermore, and aftercooler is integrated to bring the temperature to desired levels.

Finally, with the compressor thermodynamics designed, the thermodynamic properties is used to make a final design of the air receiver, refrigerant dryer, connecting pipe sizes, valve sizes and other parts are sized and designed.

For the mechanical design, each equipment sized and standard industrially available sizes incorporated to make a real like model. With all the cylinders sized and the type of tanks, pipes that will be used a final layout is prepared with each equipment drawn in detail, This is presented to clearly illustrate what a real as if compressor room would look like and how it will function.

The report addresses the compressor design topic in detail and presents the process clearly to aid a person in designing a compressor system.

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## RESUME

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