Study and Design of Compressed Air System for Optimum Operation

by

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Dissertation submitted in partial fulfilment of
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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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Approved by,

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(Dr. Zainal Ambri Bin Abdul Karim)

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CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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MUHAMMAD FARID BIN ABDULLAH

ABSTRACT

This is a report on the study and design of compressed air system installed at a plant for manufacturing processes. The plant suffers high electrical energy consumption after modifications were done to increase the production capacity of the original system. This provides an opportunity to study the reasons for such situation. The project will study both the original and modified systems and compare the performances of the systems. Consequently a new optimized system will be designed. The project is divided into two phases namely phase 1 and phase 2. The first phase focuses on the study of the original and modified systems and the second phase will be on design of a new system. The second phase involves design of the compressed air system by matching supply to demand of compressed air. Phase one study is performed by evaluating parameter that affects the usage of the air compressor which is pressure drop. The findings indicated that pressure drops were mainly contributed by pipe friction and losses due to pipe fittings. However other reasons which contributed significantly to pressure drop was high velocity flow in small diameter pipe, mainly in 15mm diameter pipe. Analysis result showed that the flow velocity in 15mm diameter pipe increased by 13 times the flow velocity in the mainline. Thus higher flow velocity increased the pressure drops greatly when the changes in all other parameters were relatively small. Other sources of pressure drops was also identified as leakage at the connection of piping and flexible hoses. Phase 2 requires evaluation of design parameters required to ensure efficient operation. The design parameters are unique to each system therefore each compressed air system has to be evaluated exclusively. Demand of compressed air is calculated by summing of required volumetric flowrate during operation of the plant and this is matched by designing supply system which able to deliver the required flowrate. At the end of the report, recommendations are given to optimize the system.

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND STUDY

In industrial plants, in addition to common utilities such as water and electricity, compressed air system is also utilized in manufacturing process. Compressed air system is used for many applications either involves directly in processes or to power up tools. Nevertheless, like all concerns about usage of energy, compressed air system potential energy also plagues with energy consumption issues such as efficiency. High energy consumption related to inefficient use of compressed air system always become hot topics in the industries as compressed air system derived its potential energy from electrical energy. Thus inefficiencies in compressed air system would mean higher operating cost and decrease industries' productivity.

1.2 PROBLEM STATEMENT

A pharmaceutical plant is suffering high electrical energy consumption which is believed to be originated, among others, from its compressed air system. This reportedly occurred after modifications were done to their compressed air system. Therefore it is imperative that a study is conducted to find the problems and an optimal designed is proposed.

1.3 OBJECTIVES

The objectives of the project are:

1. To study the original and modified compressed air system at a plant.
2. To compare the performance of the systems.
3. To design a system with optimum energy efficiency.

1.4 SCOPE OF STUDY

The scope of study of the project is limited to as specified below:

- Theoretical analysis of original compressed air system of a plant as provided in the plant’s original compressed air system layout by evaluating pressure drop and power requirement of the system.
- Theoretical analysis of modified compressed air system as provided in the plant’s modified compressed air system layout by evaluating pressure drop and power requirement of the system.
- Design of a compressed air system to optimize the current system at the plant by evaluating supply and demand of the system and proposing an optimized system.

CHAPTER 2

LITERATURE REVIEW

Compressed air system is widely used in industry to supply air for manufacturing processes or to power equipments such as conveyers, grinders, jack hammers, drills, aerators, and paint sprayers. According to United States of America's Department of Energy [1], compressed air is often called as the fourth utility due to its widespread use in industry and is the most expensive industrial utilities as 10% of electrical energy used by industrial is employed in compressing air. Furthermore, in many industrial plants, air compressors use more electricity than any other equipment in the plant. Electricity consumption of an industrial plant should be able to be reduced to a range of 20 to 50 percent by improving the systems. Thus, inefficiencies in the compressed air systems could be significant to the plant's energy consumption.

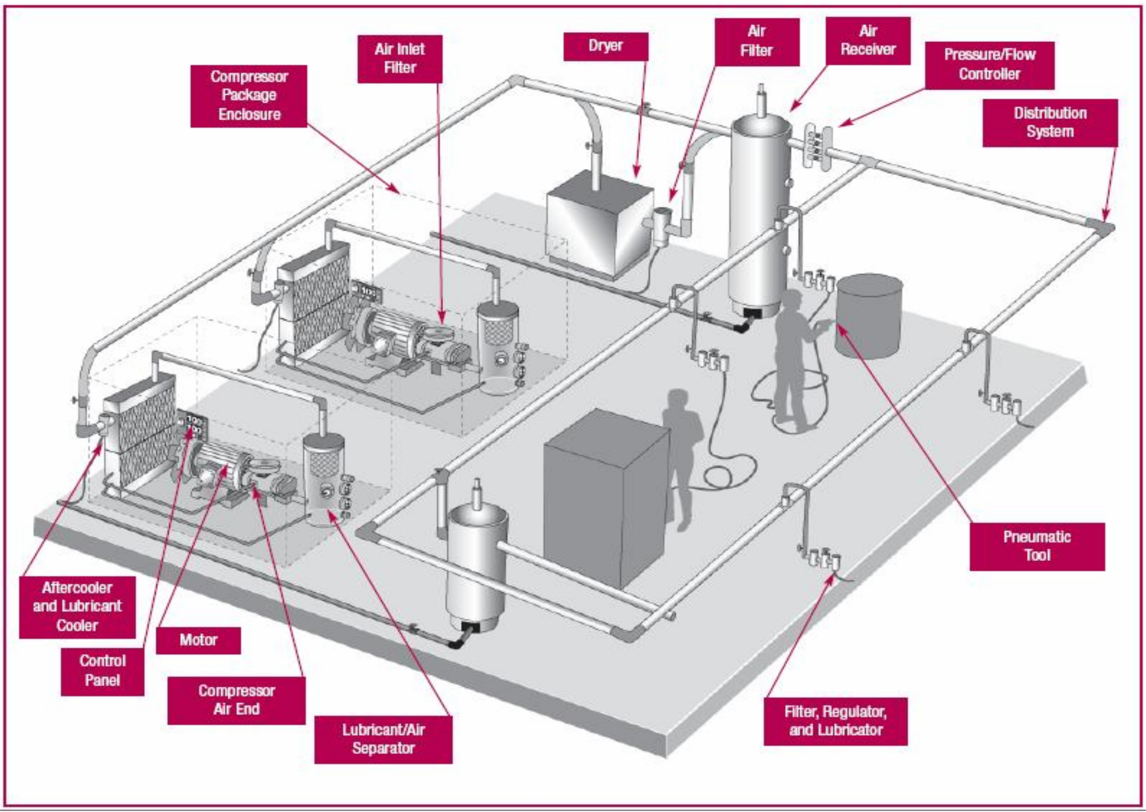

Figure 1: Typical layout of industry compressed air system.[1]

Compressed air system in an industrial plant generally consists of a supply side, which includes compressors and air treatment, and a demand side, which includes distribution and storage system, and end-use equipment as shown in Figure 1. In designing a compressed air system, there are certain elements that need to be considered which are; type of compressor, size of compressor, required pressure, air quality and load factor. All these elements form the specifications of the compressed air system installed in a plant and should be finely chosen in order to get the most efficient system running. Nevertheless, the same system does not apply to all industries because depending on the industry, the requirement of the compressed air might be different and so does the equipments. For example, for a machine shop, the compressed air needs to meet the requirements of removing solids measuring 3 microns and larger, 99% of water droplets, and 40% of oil aerosols [3]. Meanwhile for food and dairy industries, the compressed air requirements are to remove moisture and produce a -1° to 10°C pressure dew point, 99.999% of oil aerosols, all solid particles measuring 0.025 microns and larger, oily vapor, oily smell and oily tastes [3]. As for pharmaceutical industry, the compressed air system should meet requirements such as removes moisture and produce a -40° to -65.6°C pressure dew points, removes 99.999% of oil aerosols, and all particles 0.025 micron and larger [3]. Therefore it is evident that different industries necessitate different compressed air requirements.

As compressed air is readily available in a system, clean and easy to use, it is often used for application which would be economical when running using other alternatives sources such as electricity. This is one of many inefficient way of using compressed air, which consequently consumes high energy. Besides, high energy consumption could be related to other causes as well such as unregulated consumption, open blowing, leaks, point of use with regulators adjusted to their maximum setting, and overpressure.

The primary components of a compressed air system are:

- Compressor
- Prime mover
- Controls
- Air treatment and accessories
- Distribution system

- Storage Tank/ Air Receiver
- Compressed air powered equipment

The compressor is the mechanical device that takes in ambient air and increases its pressure. The main power source powers the compressor (typically an electric motor). Controls function to regulate the amount of compressed air being produced. The treatment equipment removes contaminants and water from the compressed air, and accessories keep the system operating properly. A distribution system consists of the piping that is analogous to wiring in the electrical world - they transport compressed air to where it is needed.

2.1 COMPRESSOR

Many modern industrial air compressors are sold “packaged” with the compressor, drive motor, and many of the accessories mounted on a skid for ease of installation. There are two basic compressor types: positive-displacement and dynamic [5]. In the positive-displacement type, a given quantity of air or gas is trapped in a compression chamber and the volume which it occupies is mechanically reduced, causing a corresponding rise in pressure prior to discharge. Dynamic compressors impart velocity energy to continuously flowing air or gas by means of impellers rotating at very high speeds. Compressor used in the system is oil free rotary screw compressor which falls under the type rotary positive-displacement. The power rating of the compressor is 30kW. Two distinct types are available for lubricant free rotary screw: the dry-type and the water-injected type.

In the dry-type, the intermeshing rotors are not allowed to touch and their relative positions are maintained by means of lubricated timing gears external to the compression chamber. Dry-type, lubricant-free rotary screw compressors have a power capacity range from 18.6 to 2983 kW or flowrate range from 2.5 to 566 m^3/min. Single-stage units operate up to 344.7 kPa, while two-stage can achieve up to 1034 kPa [1].

In the water-injected type, similar timing gear construction is used, but water is injected into the compression chamber to act as a seal in internal clearances and to remove the heat of compression. This allows pressures in the 689.5 to 1034 kPa range to be accomplished with only one stage.

2.2 PRIME MOVER

The prime mover is the main power source providing energy to drive the compressor. The prime mover must provide enough power to start the compressor, accelerate it to full speed, and keep the unit operating under various design conditions. This power can be provided by any one of the following sources: electric motors, diesel or natural gas engines, steam turbines and combustion turbines. Electric motors are by far the most common type of prime mover.

2.3 CONTROLS

Compressed air system controls serve to match compressor supply with system demand. Proper compressor control is essential to efficient operation and high performance. Because compressor systems are typically sized to meet a system's maximum demand, a contr/ol system is almost always needed to reduce the output of the compressor during times of lower demand. Compressor controls are typically included in the compressor package, and many manufacturers offer more than one type of control technology.

2.4 AIR TREATMENT AND ACCESORIES

Accessories are the various types of equipment used to treat compressed air by removing contaminants such as dirt, lubricant, and water; to keep compressed air systems running smoothly; and to deliver the proper pressure and quantity of air throughout the system. Accessories include compressor aftercoolers, filters, separators, dryers, heat recovery equipment, lubricators, pressure regulators, air receivers, traps, and automatic drains. Besides friction and vertical elevation, these accessories are also sources of pressure drop.

2.4.1 Air Inlet Filters.

An air inlet filter protects the compressor from atmospheric airborne particles. Further filtration is typically needed to protect equipment downstream of the compressor [6].

2.4.2 Aftercoolers.

As mechanical energy is applied to a gas during compression, the temperature of the gas increases. Aftercoolers are installed after the final stage of compression to reduce the air temperature. As the air temperature is reduced, water vapor in the air is condensed, separated, collected, and drained from the system. [7]

2.4.3 Dryers.

When air leaves an aftercooler and moisture separator, it is typically saturated. Any further radiant cooling as it passes through the distribution piping, which may be exposed to colder temperatures, will cause further condensation of moisture with detrimental effects, such as corrosion and contamination of point-of-use processes. This problem can be avoided by the proper use of compressed air dryers.

2.4.4 Compressed Air Filters.

Depending on the level of air purity required, different levels of filtration and types of filters are used. These include particulate filters to remove solid particles, coalescing filters to remove lubricant and moisture, and adsorbent filters for tastes and odors. A particulate filter is recommended after a desiccant-type dryer to remove desiccant “fines.” A coalescing-type filter is recommended before a desiccant type dryer to prevent fouling of the desiccant bed. Additional filtration may also be needed to meet requirements for specific end uses.

2.4.5 Air Regulators

A pressure regulator is used so that a constant pressure is available for a given pneumatic system. The regulator contains an adjustable upper spring, which allows the valve to hold a given pressure on the downstream side. The fore of the spring is set for the required downstream pressure [5].

2.5 DISTRIBUTION SYSTEM

Distribution system consists of piping lines connecting various components to deliver air to point-of-use. The length of the piping lines should be kept to a minimum and air distribution piping should be large enough in diameter to minimize pressure drop. There are three different approaches to laying out compressed air distribution systems [12]. The first is a collection of pipes and hoses that have been added as required, with no planning. The second approach is the use of a central spine that feeds the necessary drops. Figure 2 below shows a typical spine distribution system. This method of distribution is an excellent choice for smaller facilities such as auto repair garage.

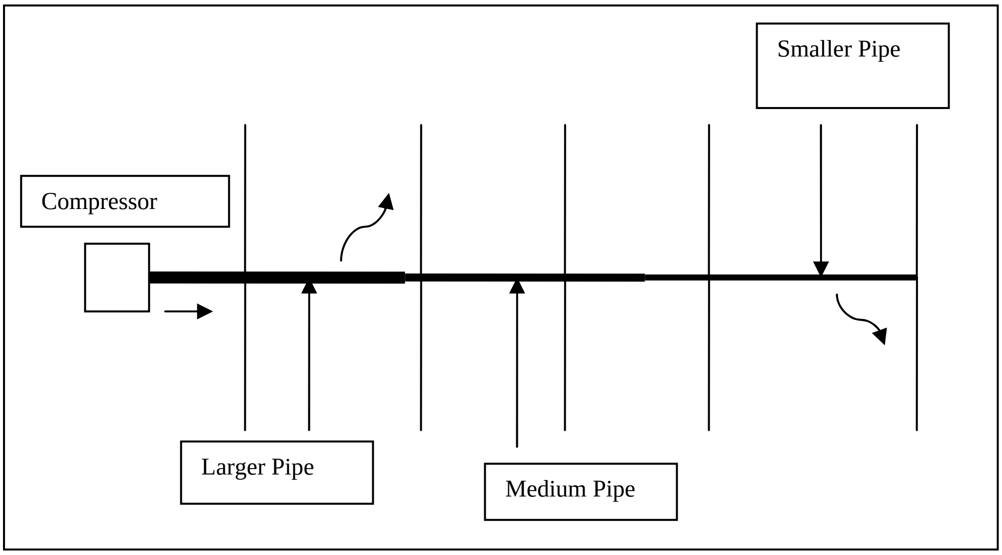

Figure 2: Spine Distribution System

The third method is the loop distribution system as illustrated in Figure 3 below. This method has the benefit of equalizing flow in the system and widely used in large facilities. The system in the plant understudy is a loop distribution in order to ensure overall economic use of the compressed air.

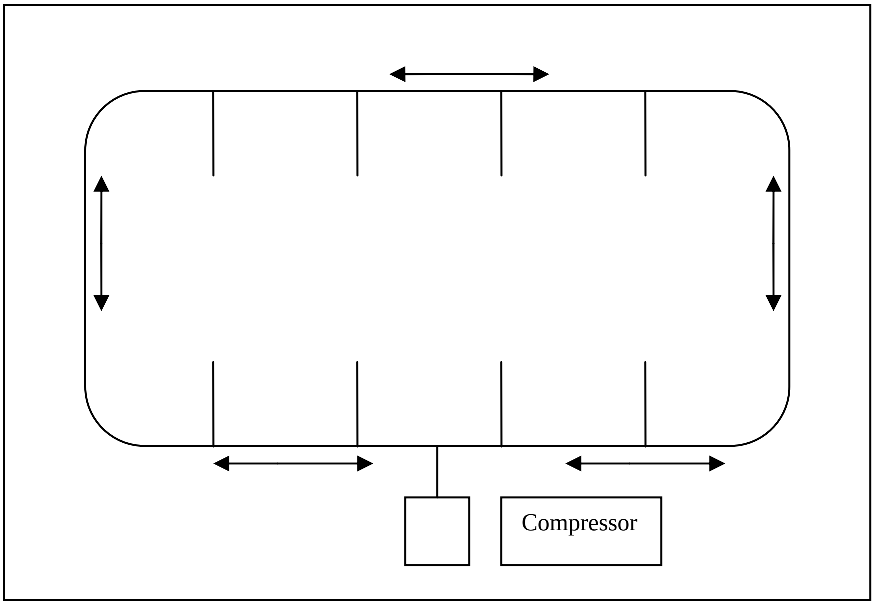

Figure 3: Loop Distribution System

Good design of distribution system would have headers with a slight slope to allow drainage of condensate and drop legs from the bottom side of the header should be provided to allow collection and drainage of the condensate. The direction of the slope should be away from the compressor.

2.6 STORAGE TANKS/AIR RECEIVER

Air receivers are used to serve the following purposes [13]:

1. Provide compressed air storage capacity to meet peak demand events
2. Help control system pressure by controlling the rate of pressure change in a system.
3. Collect residual condensate.

The air receiver is known depending on their location; wet air if it is located before the air dryer and dry air if it is located after. Receivers are especially effective for systems with widely varying compressed air flow requirements. Where peaks are intermittent, a large air receiver may allow a smaller air compressor to be used and can allow the capacity control system to operate more effectively and improve system efficiency. Demand-side control will optimize the benefit of the air receiver storage volume by stabilizing system header pressure and “flattening” the load peaks. Air receivers also play a crucial role in orchestrating system controls, providing the time needed to start or avoid starting standby air compressors.

In sizing the air receiver, it should at least large enough to hold all the air delivered by the compressor in 1 minute [3]. Determining the proper size of receiver could be a little complicated. Each end point application must be assessed on its own. The receiver size depends on [3] :

i. Delivery volume of compressor
ii. Air consumption
iii. Pipeline network
iv. Type and nature of on-off regulation
v. Permissible pressure difference in pipelines

All air receivers should be equipped with pressure relief valve that is set to open at a pressure no higher than the maximum pressure rating on the receiver's name plate and a flow rate that is 20 percent higher than the maximum delivery rate. Most importantly, the receiver must satisfy international standards for pressure vessel namely American Society of Mechanical Engineers (ASME) boiler and pressure vessel code.

2.7 THE SYSTEM AT THE PHARMACEUTICAL PLANT

The plant is producing pharmaceutical products and utilizing compressed air in some of their production machines such as in coating machine. Compressed air system in the plant was initially designed and constructed with adequate supply of compressed air for the production machines, which, in this study, is labeled as the original system. However, as the production capacity increased and renovations were done in the production floor, new demands for compressed air exist thus end-use points were added without considering the system capacity. By additions of the new end-use points, the system is now regarded as the modified system. Besides for machine use, the plant uses compressed air for variety of other unregulated purposes such as vacuuming and washing. Thus it leads to excessive consumption of compressed air which in turns cause high electrical energy consumption.

The compressed air system in the plant is using Hitachi DSP-22A5 II oil free rotary screw compressor with 22 kW of power. The system pressure is 600 kPa however the compressor is pressurizing atmosphere air to a higher discharge pressure of 680 kPa at volumetric flowrate of 3 m^3 per minute. This is to compensate for pressure drop due to various causes such as piping and air treatment accessories. A 400 m^3 air receiver is installed to provide capacity to meet peak demand. The allowable pressure difference in the system is 55 kPa in which the compressor cut-in pressure is 620.5 kPa and the cut-off pressure is 675.7 kPa.

The piping layout of the distribution system is the loop system with dropper at certain points along the mainline connecting to the end-use points (Figure 4). Originally, there are 15 end-use points as drawn in the original plant's layout, however the end-use points have been later installed to a total of 24 points. The points provide compressed air connections to many production machines through quick coupling and flexible hose.

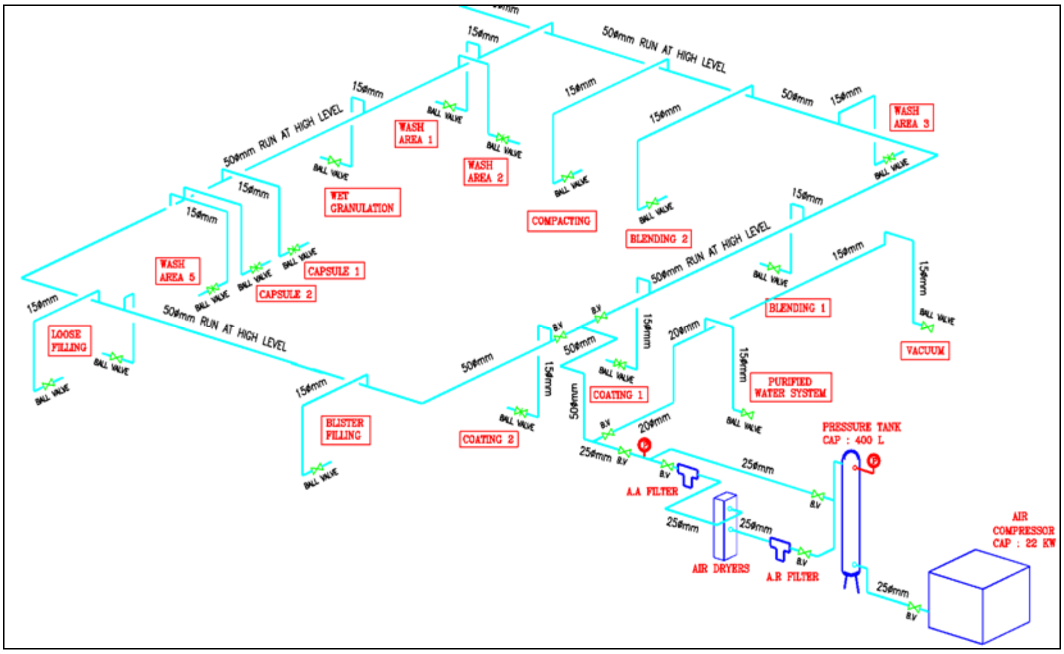

Figure 4: The plant's loop distribution system.

In terms of demand, the plants compressed air system is used by production machines, analytical and micro laboratory and also other pneumatic applications such as airlock doors. Each application requires different amount of compressed air pressure and flowrate, therefore the running compressed air system has to be able to satisfy all the application's requirements. Some of the production machines that are available at the plants are as follow:

1. RAMA COTA FC 39 Coating machines.
2. BOSCH Capsulating machine.
3. Pharma matrix Mixer 300/2G Blending machine.
4. NR Fluid Bed Dryer.

Amongst the production machines used in the plant, there are only 3 machines that are using significant amount of compressed air. These machines are two coating machines and a capsulating machine. It is undeniably that other machines also require compressed air for operation; however, the amount and duration of consumption and is insignificant to be considered.

CHAPTER 3

METHODOLOGY

The project is divided into 2 phase for ease of project management and execution namely phase 1 and phase 2. The tasks in each phase is structured according to works that should be accomplished in each semester of two semesters final year project actual duration.

3.1 PHASE 1

Phase 1 involves research into industrial compressed air system and evaluation of the original compressed air system installed at the plant prior to modification. Then it is followed by evaluation of the modified system which will be compared to the original system. The following are the procedures of the project for phase 1 [Refer Appendix for Gantt Chart] :

1. Industrial compressed air system designs together with all the components and accessories required for running such a system are researched and studied through all possible resources like internet, books, and industrial guidelines.

2. The compressed air system installed at a plant is studied in great details which would also include all the fittings in the distribution section of the system.

3. Compressed air pressure drop and power required across the distribution section are analyzed. This will also be compared with the capacity of the air compressor unit and of the distribution section.

4. Modifications of the plant's compressed air system will then be analyzed.

5. Compressed air pressure drop and power required will be recalculated in the analysis and again compared with the compressor and distribution section capacity. The modified and the original system performance will be compared.

3.2 PHASE 2

Phase 2 of the project requires a new compressed air system of the plant to be designed for optimum operation efficiency. New design does not necessarily mean total revamp of the system, however, new design may include reconfiguring of the compressed air system or upgrading of ineffective equipments or implementing effective system's operation control. For this project, design approach will be to increase efficiency by evaluating the reasons for costly operation of the system and rectifying using the least cost as possible. The following are the procedures for phase 2 [Refer Appendix for Gantt Chart]:

1. Analyzing the supply side of the current system design which includes the compressors and air treatments.

2. Analyzing the demand side of the current system design which includes the distributions and storage system.

3. Determining compressed air needs. Compressed air needs are defined by the air quality and quantity needed by the point-of-use.

4. A new compressed air system for optimum operation efficiency will be designed. The newly designed system should be able to rectify any efficiency and operating problems.

3.3 ANALYZING PARAMETERS

In analyzing the compressed air system, firstly, background studies were performed on the governing engineering laws and equations that would best describe the system. When compressed air flowing through pipes, certain matter that need to pay attention to is friction which in turn would affect the pressure drop and head loss during the flow. Besides friction, pressure drop is also affected by vertical pipe difference and changes of kinetic energy. Consequently, pressure drop is then used to determine the compressor discharge pressure that is needed to maintain pressure and air flowrate in the system.

3.3.1 Phase 1 Analysis

Pressure drop is calculated by first analyzing the flow regime using Reynolds number, surface roughness, and Darcy friction factor. Besides, minor losses are also contributing to the pressure loss in a system.

3.3.1.1 Reynolds Number

Reynolds number is the ratio of inertial forces to viscous forces in the fluid. It is used to determine the flow regime of the fluid, either the flow is laminar or turbulent. According to Cengel et al. [8], at high Reynolds number, the inertial forces are larger relative to viscous forces, thus the viscous forces cannot prevent the random and rapid fluctuation of the fluid. At small Reynolds number, viscous forces are large enough to suppress fluctuations due to inertial forces. Therefore, the flow is turbulent in the first case and laminar in the second case. The ratio is expressed for internal flow in a circular pipe as:

$$\mathrm{Re} = \frac{Inertial.Force}{Viscous.Force} = \frac{V_{avg}D}{\nu} = \frac{\rho V_{avg}D}{\mu}$$

Where, Re : Reynolds Number

V_{avg} : Fluid Velocity

D : Pipe Diameter

ν : Kinematic Viscosity

ρ : Density

μ Dynamic Viscosity

Flow at first is laminar and transform to turbulent as is flowing over the length of medium. The transition phase in which flow become turbulent is at critical Reynolds number which is Re = 2300. Transitions from laminar to turbulent flow, however is also affected by other conditions such as surface roughness, pipe vibrations and fluctuation in the flow. In compressed air system, the flow regime would be in turbulent regime as indicated by Reynolds number which higher than 2300 [8].

3.3.1.2 Surface Roughness

Surface roughness for each commercial pipe is different due to the different materials used in fabricating them. In practical, relative roughness is used to determine friction factor. Apart from surface roughness, another measure of roughness is relative roughness ε / D, which is defined as the ratio of the mean height of roughness of the pipe to the pipe diameter. Pipes in the compressed air system are made from stainless steel, thus the surface roughness is predetermined as 0.002 mm [8]. Therefore relative roughness could easily be determined by dividing above value with the pipe's diameter.

3.3.1.3 Darcy Friction Factor

Darcy friction factor together with Reynolds number and relative roughness are correlated in an equation to determine the friction factor known as Colebrook equation [8]:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

Where, f : Darcy Friction Factor

ε / D : Relative Roughness

Calculations for friction factor are done by iterative method of Colebrook equation. Moody chart is also used as first rough estimate of friction factor and then substituted into the equation above to obtain an accurate value. Each pipe size dimensions' friction factors are calculated independently by above method.

3.3.1.4 Pressure Drop

Pressure drop is vital in calculating power requirement of air compressor to maintain flow. It is calculated using the equation below [8]:

$$\Delta P = f\,\frac{L}{D}\,\frac{\rho V_{avg}{}^{2}}{2}$$

Where, ΔP : Pressure Drop

L : Length of pipe

By substituting all the values aforementioned, pressure drop at certain distance from the compressor could be determined. There will also be pressure drop due to different elevation as the end use of the compressed air system is located at certain height below the main line which should be accounted for by using the equation below [4]:

$$\Delta P = \rho \times g \times \Delta H$$

Where, ΔH : Elevation difference

g : Gravitational acceleration

3.3.1.5 Minor Losses

The air in the system will flow through typical piping system which has various fittings, valves, bends, elbows, and tees. These components will disrupt fluid flow and cause additional losses because of flow separation and mixing induced. Minor losses are determined by loss coefficient K_L, as in the equation below [8]:

$$h_L = K_L \frac{V^2{}_{avg}}{2g}$$

Where, K_L : Loss coefficient

Minor losses are also expressed in terms of equivalent length L_{equiv}, defined as [8]:

$$L_{equiv} = \frac{D}{f} K_L$$

3.3.1.6 Power

Power required to drive an air compressor to meet system pressure and flow rate requirements is another important factor in designing and evaluation of compressed air system. Theoretical power is governed by equation [5]:

$$Power_{theo} = \frac{P_{in} Q}{17.1}\left[\left(\frac{P_{out}}{P_{in}}\right)^{0.286} - 1\right]$$

Where, P_{in} : inlet atmospheric pressure

P_{out} : outlet pressure

Q : flowrate

To overcome frictional losses in the flow due to viscosity, the power and pressure drop relationship is given as [8]:

$$Power = Q \times \Delta P$$

3.3.2 Phase 2 Analysis

Supply and demand analysis of the system should produce both qualitative and quantitative results. The system is divided into supply and demand side. Supply side analysis involves acquiring parameters such as discharge pressure, air flowrate, and allowable pressure difference. While demand side analysis involves defining the air quality, quantity, and pressure level requires by the end-use points.

3.3.2.1 Supply Side

In analyzing supply side, one of the important things to consider is air receiver size. Air receiver helps supply compressed air to meet demand during peak load. The size of receiver for compressor with on-off regulation could be calculated using the equation below [3]:

$$Vr = \frac{15 \times Q \times P_{atm}}{\Delta P \times N}$$

Where,	V_r	: Receiver Volume
	Q	: Delivery Volume
	ΔP	: Pressure difference
	N	: Switching cycle/hour of compressor running.

Before the above equation could be used, the switching cycle needs to be determined using actual data acquired from the plant. Switching cycle is determined by the time compressor is working under full load and no load condition. Full load condition is when the compressor filling up the air receiver. No load condition is when the compressor stop compressor air and the air receiver is being depleted by machine demands. Below is the equation used to determine time taken for recovery and depletion of air receiver [12]:

$$T = \frac{V \times (P_1 - P_2)}{q_s \times P_{atm}}$$

Where,	T	: Time to recover
	V	: Volume of receiver
	P_1-P_2	: Allowable Pressure difference
	q_s	: Supply air flow or discharge flow
	P_{atm}	: Atmosphere pressure

3.3.2.2 Demand Side

Analyzing demand side is quite straightforward. There is no equation involves in the analysis. Demand side analysis requires pressure and air flowrate to be obtained from the machines used in the plant. The data is then analyzed to determine the minimum required pressure of the machines and the total volumetric flowrate required by the production machines.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 RESULT AND DISCUSSION OF PHASE 1

In order to compare the parameters of both original and modified systems, analysis was performed on both systems using data and system layout acquired from the plant [14]. Both systems were analyzed using actual working data of the plant's compressed air system such as volumetric flowrate, system pressure and pipe length. The analysis was performed according to the pipe's diameter as different diameters would yield different flow velocities, given a constant pressure such as what required in the system.

In the calculation, pipe parameters were calculated for each of the three pipe dimensions which were 50 mm, 20 mm and 15 mm for mainline, extended branch and dropper respectively. For each of the dimensions, the flow area and relative roughness were calculated to provide parameters value for future calculations. The pipes used in the plant are made of stainless steel of SCH 80 with surface roughness of 0.000002. Table 1 below shows the pipe's specifications used in the plant:

Table 1: Pipe's specifications

50 mm Pipe		20 mm Pipe		15 mm Pipe	
OD (m)	0.06034	OD (m)	0.02667	OD (m)	0.02134
ID (m)	0.049251	ID (m)	0.01885	ID (m)	0.01387
Surface Roughness	0.000002	Surface Roughness	0.000002	Surface Roughness	0.000002
Relative Roughness	4.06E-05	Relative Roughness	0.000106	Relative Roughness	0.000144
Area, m^2	0.001904	Area, m^2	0.000279	Area, m^2	0.000151

Once the piping system specifications had been established, properties of air were acquired next. These properties are for air flowing in the system, which is the compressed air. Properties of air were evaluated at temperature of 20 °C which was the assumed air temperature. The air volumetric flowrate was evaluated at the demand side flowrate which is 0.0135 m^3/s. Air properties were extracted from references [7] and listed together with several parameters for evaluating fluid flow in the Table 2 below:

Table 2: Air properties and evaluated parameters of compressed air.

Properties	50 mm	20 mm	15 mm
Density, (kg/m^3)	1.204		
Temperature, T (°C)	20		
Dynamic Viscosity kg/m.s	0.0000183		
Kinematic Viscosity m^2/s	0.0000152		
Flowrate, Q, (m^3/s)	0.0135		
Velocity, m/s	7.09	48.39	89.4
Reynolds Number	2.30E+04	1.57E+05	2.90E+05
Friction Factor	0.0254	0.0171	0.0159

From air properties gathered, calculations were performed to obtain velocity, Reynolds numbers and friction factors for each of the pipe's dimension. Velocity was obtained by dividing flowrate by cross sectional area of each pipe. Then by substituting into Reynolds number equation such as depicted in Chapter 3, Reynolds number for each pipe dimension was determined to be in the range of 23000 to 290000. Consequently, the air flow in the compressed air system was determined to be in the turbulent regime. Friction factor for each pipe's dimension was calculated using Colebrook equation as mentioned in Chapter 3. Results revealed that friction factors were varied with pipe's diameter. The largest diameter of pipe will have the largest friction factor and followed by other smaller diameters as shown in Table 2 above.

In the original's system analysis, parameters were acquired from the plant's layout of the compressed air system and it was regarded as the designed system which was installed together with the plant's construction. Therefore there were some end-use points which were not exist at the time however exist at current time. In calculating pressure drop, total pipe length was determined by the addition of pipe's length and also fittings' equivalent length for each pipe dimension. Moreover, pressure drop due to elevation difference was also taken into account by measuring the height difference between the mainline and end-use point. The difference will reduce pressure drop in the whole system because the mainline was located at a height above the end-use point thus a column of compressed air in the dropper will increase the end-use points pressure. Table 3 below shows the results of analysis on the original and modified compressed air system:

Table 3: Result on analysis of both system for each end-uses points

No	End uses	L 50 (m)	Leq 50 (m)	L 15 (m)	Leq 15 (m)	L 20 (m)	Ele Drp (Pa)	PD 50 (Pa)	PD 25 (Pa)	PD 15 (Pa)	Tot PD (kPa)	Power kW
1	Coating 2	3.81	13.95	0	1.94	0	-34.25	273.05	0.00	9882.60	10.12	0.14
2	Blister Filling	15.64	16.45	2.18	1.94	0	-34.25	493.36	0.00	20987.78	21.45	0.29
3	Loose Filling 1	21.95	18.45	2.18	1.94	0	-34.25	621.12	0.00	20987.78	21.57	0.29
4	Loose Filling 2	25.12	19.45	1.18	1.94	0	-34.25	685.23	0.00	15893.66	16.54	0.22
5	Wash Area 5	37.6	22.95	4.22	1.94	0	-34.25	930.91	0.00	31379.79	32.28	0.44
6	Capsule 2	37.6	22.95	4.22	1.94	0	-34.25	930.91	0.00	31379.79	32.28	0.44
7	Capsule 1	41.6	23.95	4.22	1.94	0	-34.25	1007.78	0.00	31379.79	32.35	0.44
8	Wet Granulation	46.12	24.95	3.16	1.94	0	-34.25	1092.65	0.00	25980.02	27.04	0.37
9	Coating 1	2.89	13.95	0	1.94	0	-34.25	258.90	0.00	9882.60	10.11	0.14
10	Blending 1	14.32	14.95	0.67	1.94	0	-34.25	450.00	0.00	13295.66	13.71	0.19
11	Wash Area 3	21.52	17.45	1.65	1.94	0	-34.25	599.13	0.00	18287.90	18.85	0.25
12	Blending 2	26.35	18.45	7.67	1.94	0	-34.25	688.77	0.00	48954.52	49.61	0.67
13	Compacting	31.34	19.45	7.67	1.94	0	-34.25	780.86	0.00	48954.52	49.70	0.67
14	Wash Area 1	39.69	24.95	3.16	1.94	0	-34.25	993.79	0.00	25980.02	26.94	0.36
15	Wash Area 2	39.69	24.95	2.67	1.94	0	-34.25	993.79	0.00	23483.90	24.44	0.33
16	Purified Water	0	6.6	1	1.94	6.59	-34.25	101.47	7941.58	14976.72	22.99	0.31
17	Vacuum	0	7.6	1	1.94	13.17	-34.25	116.84	15871.12	14976.72	30.93	0.42
18	Strip Filling 2	21.95	18.45	2.18	1.94	0	-34.25	621.12	0.00	20987.78	21.57	0.29
19	Strip Filling 1	15.64	16.45	2.18	1.94	0	-34.25	493.36	0.00	20987.78	21.45	0.29
20	Dry Syrup 1	25.12	19.45	1.18	1.94	0	-34.25	685.23	0.00	15893.66	16.54	0.22
21	Dry Syrup 2	28.27	20.45	2.18	1.94	0	-34.25	749.03	0.00	20987.78	21.70	0.29
22	Analytical Lab	14.36	18.1	0	1.94	27.33	-34.25	499.05	32935.28	9882.60	43.28	0.58
23	Micro Lab	14.36	18.1	0	1.94	27.33	-34.25	499.05	32935.28	9882.60	43.28	0.58
24	Capsule 3	46.12	24.95	2.13	1.94	0	-34.25	1092.65	0.00	20733.08	21.79	0.29

Modified System | Points with consumption

Table 3 shows the result of analysis on the original and modified systems on each end-use points. Modifications were done on the original system by adding several new end-use points which depicted in the table as shaded item number 16 to 24. From the table, the maximum pressure drop occurred at compacting and blending 2 end-use points with pressure drop value of 49.7 kPa and 49.6 kPa respectively. Figure 5 below shows the pressure drop occurred at each end-use point.

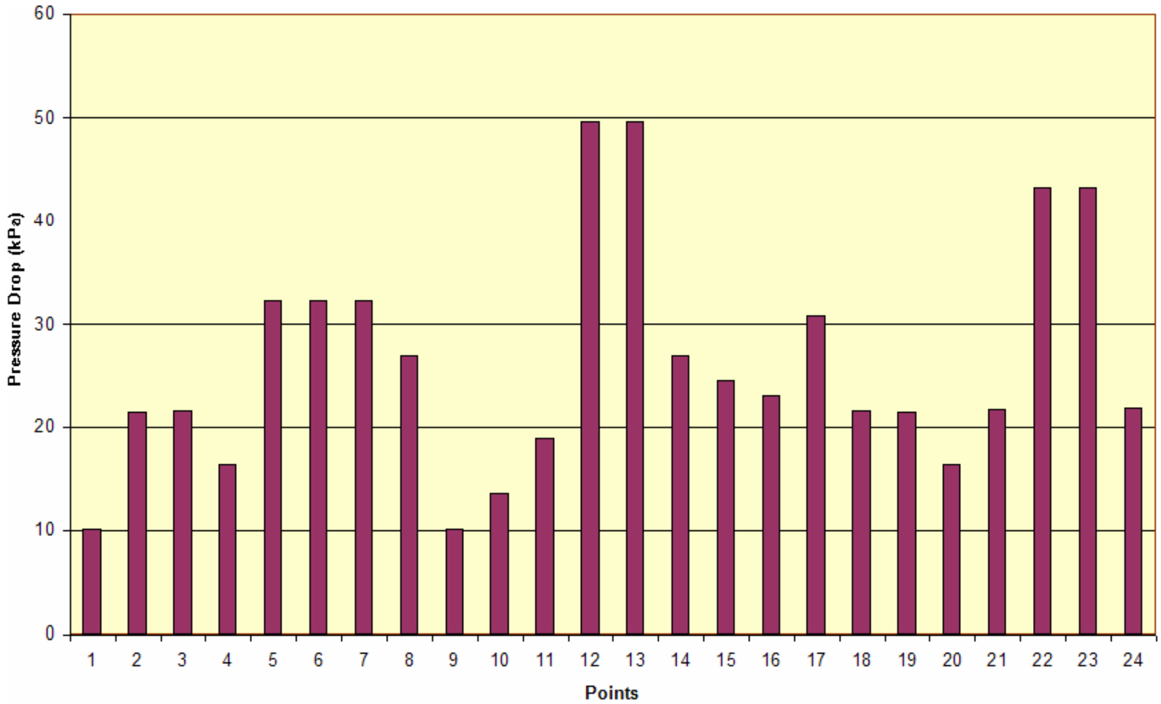

Figure 5: Pressure drop at each end-use points

The pressure drop is considered reasonable as Majumdar states that pressure drop in a system should be less than 100 kPa. The reason for the maximum pressure drop occurred at these 2 points was largely contributed by losses due to pipe frictions and fittings in 15 mm diameter piping system. Those points are having the largest length of 15 mm line, thus resulting in high pressure drop. This is due to the fact that in order to maintain system pressure at 600 kPa, the air was required to flow at 13 times higher velocity in the 15 mm diameter pipe, which value at 89.4 m/s, than in the 50 mm main line, which value at 7.09 m/s. Consequently, higher flow velocity increased the pressure drop. This is also true for analytical and micro lab end-use points which ranked as the second largest pressure drop to occur at end-use points. The 20 mm diameter pipes used to supply compressed air to the end-use points caused velocity to increase about 2 times greater than velocity in the main line.

As for power analysis, there are two types of power considered which are theoretical power to compressed air from atmosphere pressure to system pressure and power to overcome pressure drop. Both power were calculated using equation provided in the section 3.1.1.6 . Theoretical power to compressed air was calculated to be 13 kW. While power to overcome maximum pressure drop was 0.67 kW.

Comparing both systems, it is evident that with additional end-use points in the modified system, compressed air volume and pressure drop increased. Hence, the additional pressure drop contributed to increase in power requirement for the system to deliver constant pressure at each end-use point. There are several new end-use points were sharing the same branch of the original system, thus the points will have the same pressure drop as the original end-use point. For instance, strip filling 1 end-use point were sharing the same branch as blister filling end-use, hence experienced the same pressure drop as blister filling. However, there were several new points which were quite far from the main line and a new piping line had to be constructed such as for analytical and micro lab end-use points. These new points actually increased the power requirement significantly.

4.2 RESULT AND DISCUSSION OF PHASE 2

The supply and demand analyses of the compressed air system were done by utilizing the system's data acquired from the plant. The supply side refers to how the compressed air is generated and treated. Table 4 below shows the supply side data acquired from the plant:

Table 4 : Supply Side Data of Compressed Air System

System Pressure	600 kPa
Compressor Cut in	620.5 kPa
Compressor Cut off	675.7 kPa
Discharge Pressure	680 kPa
Max Full Flow	0.05 m^3/s
No Load Pressure	80 kPa

The compressed air system is operated at pressure of 6 bar which is about 600 kPa, therefore the compressor must be able to supply at least the minimum pressure. As it is found out, the compressor is supplying compressed air at discharge pressure of 680 kPa. The discharge pressure supplied by the compressor is slightly higher than the system pressure in order to compensate for pressure drop caused by air treatments accessories such as air filters and dryers. Discharge pressure in excess of system pressure is an acceptable practice in industry and does not translate to waste. At maximum operating load, the compressor is supplying 0.05 m^3/s of compressed air to the system.

Demand side refers to how the compressed air is actually used in the plant. Table 5 below shows the demand side of the compressed air system.

Table 5 : Demand Side Data of Compressed Air System

Coating Machine	
Brand	RAMA COTA 39"
Quantity	2
Required Pressure	> 0.049 kPa
Required Air Flow	> 48 m^3/ hr
Capsulating Machine	
Brand	BOSCH
Quantity	1
Required Pressure	500 kPa
Required Air Flow	0.5 m^3/ hour

The table above shows production machines in the plant that use compressed air for operation. Coating machines require pressure of at least 0.049 kPa and air flow of at least 48 m^3/hour. Comparing the requirement with supply data, result revealed that the supplied compressed air pressure and flow are more than enough for coating machine. This is also true for capsulating machine which required compressed air pressure of 500 kPa and flowrate of 0.5 m^3/hour. Other machines such as blending and dryer machines which also used in the plant's production are reportedly using very minimal amount of air volume and operating within system supplied pressure. Moreover, both machines are used occasionally and should not be considered as normal demand.

Air receiver was also analyzed to determine whether the receiver capacity was sufficient to accommodate generated compressed air and satisfying demand at peak load. Figure 6 below shows the schematic diagram of supply and demand with actual usage data of compressed air.

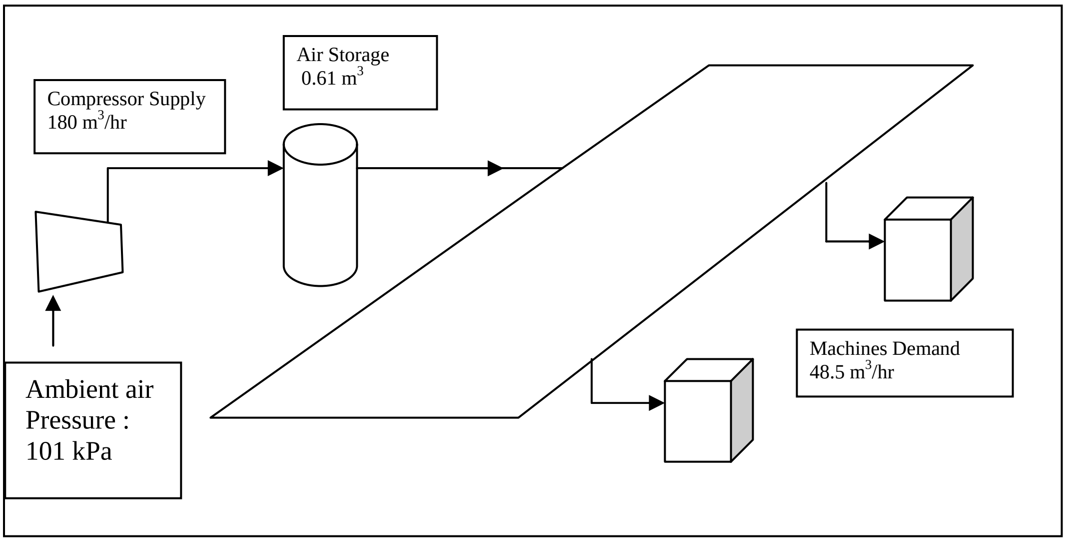

Figure 6: Schematic Diagram of Supply and Demand

The figure above shows that the supply of compressed air comes from the compressor at flowrate of 3 m^3/hr with discharge pressure of 681 kPa. The storage is summation of air receiver storage of 0.4 m^3 and distribution piping storage of 0.21 m^3 as shown in the Table 6 below:

Table 6 : Air Storage Volume of the System

Air Storage Volume					
Item					Total
Storage Tank					
Tank 1					0.4
Piping Volume					
	50 mm	20 mm	15 mm	15 mm Dropper	
Length m	96.138	47.12	42.63	49.3	
Inside Dia m	0.049251	0.01885	0.01387	0.01387	
Volume m^3	0.183	0.0132	0.006	0.0074	0.210
Total Volume, m^3					0.610

The 3 analyses performed on the air receiver using data obtained from the plant namely:

1. Time taken to refill from minimum pressure.
2. Time taken to deplete to minimum pressure.
3. Volume of receiver required given flowrate and allowable pressure difference.

Analysis 1: Time taken to refill from minimum pressure

$$T = \frac{V \times (P_1 - P_2)}{q_s \times P_{atm}}$$

$$T = \frac{21.5 \times (98 - 90)}{105.94 \times 14.7}$$

$$T = 0.11 \min \approx 6.6 \sec$$

Analysis 2: Time taken to deplete to minimum pressure

$$T = \frac{V \times (P_1 - P_2)}{q_s \times P_{atm}}$$

$$T = \frac{21.5 \times (98 - 90)}{28.6 \times 14.7}$$

$$T = 0.409 \min \approx 25.54 \sec$$

From both analysis, the compressor ‘on’ cycle is 6.6 seconds and its ‘off’ cycle is 25.54 seconds. Analysis 1 is using supply flowrate from the compressor while analysis 2 is using demand flowrate by the machines and applications. Total on-off cycle of the compressor is 31.14 seconds. This is deduced from the fact that during refilling the compressor is working on full load and during depletion, the compressor is in no-load mode. To determine the cycle per hour of compressor running:

$$N = 1cycle \div \frac{31.14}{3600}$$

$$N = 115.6cycle / hr$$

Analysis 3: Volume of receiver with on-off regulation

$$Vr = \frac{15 \times Q \times P_{atm}}{\Delta P \times N}$$

$$Vr = \frac{15 \times 3 \times 1.01}{0.5516 \times 115.6}$$

$$Vr = 0.713m^3$$

Table 7 below shows the results of the analysis on time taken to refill and deplete air receiver:

Table 7 : Result of analysis on the air receiver

Item	Time
Time to Refill, sec	6.6
Time to Deplete, sec	25.54
On-Off Cycle, sec	31.14
Volume, m^3	0.713

The table above shows that the time to refill is 6.6 seconds and the time to deplete is 25.54 seconds. The time was calculated using parameters acquired from the plant in which the allowable pressure difference for compressor cut-in and cut-off is 620.5 kPa and 675.7 kPa respectively. Both times then were added to determine the compressor on-off cycle. By using the cycle of compressor running parameters, the recommended volume of receiver for that particular supply and demand data was determined to be 0.713 m^3.

4.2.1 Design of an Optimized Compressed Air System

Optimization of the system requires the system supply and demand to function efficiently. The system supply should supply just enough compressed air to meet demand in terms of pressure to minimize cost. Additionally, the supply side should be able to meet the volume demand of the system. Table 8 below shows the comparison between the plant's requirements, current system specifications, and the proposed design system specifications.

Table 8 : Comparison of the system requirement, current system and proposed design specifications

Parameters	System Requirement	Current System	Proposed design
Pressure Drop, kPa	< 100	Max. 49.7	No changes
Pressure, kPa	500	600	No changes
Power, kW	0.67 and 13	22	No changes
Air receiver and storage volume, m^3	0.713	0.610	> 0.713

Table 8 above shows that there are no design changes needed in terms of parameters relating to pressure drop, pressure and power such as pipe diameter, system pressure and pipe materials respectively. This is due the fact that current system is already meeting the system requirements for those parameters.

However, analysis revealed that the air receiver volume for the current system, 0.61 m^3, is less than the calculated value, 0.713 m^3. This is believed to be the root cause of the system inefficiency as inadequate air receiver volume to meet demand caused the air compressor to work excessively. This is because inadequate air receiver volume leads to quick depletion and refill time of the air receiver which consequently leads to short on-off cycle of the air compressor, about 31.14 sec per cycle. In other words, the compressor would have to be turned on for about 25.54 sec and then turned off for about 6.6 sec in a cycle. This translates to the air compressor is working full time during the plant's operation thus increasing the electrical energy consumption.

As for the new design, the total air storage volume is increased to at least the calculated pressure of 0.713 m^3. However Elliot recommended that for air compressor of power 22 kW, the air storage volume should have been 0.931 m^3 [12]. It is suggested that, in order to meet both design constraints, to a new air receiver should be installed while maintaining current receiver. The new air receiver should be a remote air receiver with a suggested capacity of 1 m^3 and installed close to end-use point with large compressed air consumption. The point is nearby coating machines, preferably in between the two coating machines. By installing the receiver close to the end-use points with the largest consumption, demand from these points could be met by the new receiver and demand from any other applications will be met by the current air receiver.

The total storage after installation of the new air receiver will be 1.6 m^3. Evaluating using the equation to determine volume of air receiver with on-off regulation revealed that the cycle per hour is reduced from 115.6 to 51.5 cycle/hour. Further analysis revealed that on-off cycle time is increased from 31.14 seconds to 69.9 seconds. Therefore the total time taken for compressed air to deplete is increased to 1.08 minutes.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

From results obtained in the analysis of phase 1, it is evident that with modifications, the demand of compressed air in the system increased significantly. End-use points in the original design has a maximum pressure drop, 49.7 kPa, compared to end-use points in modified design, 43.3 kPa. The maximum pressure drop occurred at the compacting and blending end-use points which are in the original design. This is due to the fact that the piping lines for the end-use points are using the greatest length of 15 mm diameter pipe. The 15 mm diameter pipe, which is the smallest diameter pipe in the system contribute significantly to pressure drop because velocity in the line, 89.4 m/s, is very large compared to 50 mm diameter pipe velocity, 7.09 m/s. Other main sources of significant pressure drops are pipe fittings and air leakage at flexible hose connection.

Theoretical power calculated to compressed atmosphere air to system pressure is 13 kW. As the compressor power is 22 kW and the maximum power to overcome pressure drop in the system is just 0.67 kW, it can be concluded that power of compressor is adequate to deliver compressed air to the plant’s system.

For the second phase, analysis was performed on the supply and demand of the compressed air system. The goal is to use demand data to determine supply of compressed air was at appropriate level. From demand analysis, it can be concluded that the pressure supplied for the system was at reasonable level and satisfied all the machines and applications requirements. This is because the maximum pressure required by a machine is 500 kPa while the system is pressurized at 600 kPa. Flowrate analysis revealed that for allowable pressure of 55 kPa, the compressed air receiver refill time is 6.6 seconds which is below the recommended refill hold time of 60 seconds [3]. Further analysis revealed that with on-off cycle of 31.14 seconds, the receiver volume should have been 0.719 m^3 instead of 0.61 m^3 as what the plant is currently running. Moreover, according to Elliot [12], the recommended volume of receiver for compressor of 22 kW power is 0.931 m^3. Judging by both recommended air receiver volume by Majumdar and Elliot, the plant's air receiver volume should have been increased.

Therefore it is proposed that a new air receiver with capacity larger than 0.931 m^3 is installed and at the same time retain the current air receiver. The new receiver should be installed nearby the points with largest compressed air consumption which is the coating machines. Installing the new high capacity air receiver would reduce the cycle time into half, from 115.6 to 51.5 cycle/hour. Besides, on-off cycle of the air compressor would be increased to 69.9 seconds, hence giving the air receiver more time to deplete to minimum allowable pressure before refilling. The new depletion time is therefore, increased to 1.08 minutes.

It is recommended that temporary connection such as flexible hoses are attached securely to the end-use points when they are in use. Observation at the plant showed that it was the main leakage source. It is also recommended that piping in the system using larger diameter pipe to reduce flow velocity and consequently pressure drop. Scheduled maintenance on the compressed air system accessories and air compressors are advised as preventive measures. This is to ensure that the equipments are operating at their maximum efficiency and be able to function with only minimal pressure drop.

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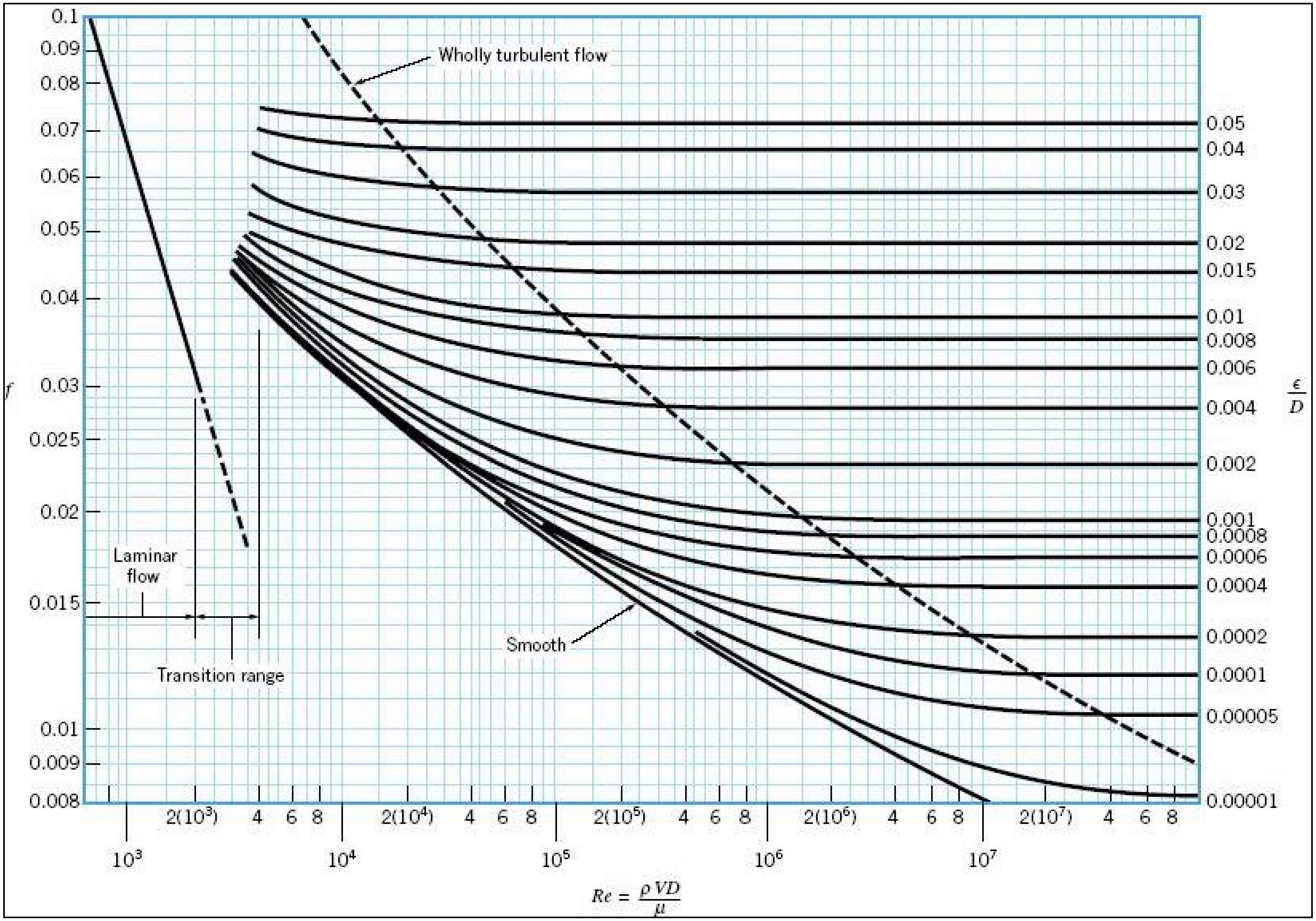
APPENDICES

Appendix I

Moody Chart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0.07
0.06
0.05
0.04
0.03
0.025
0.02
0.015
0.01
0.009
0.008
f
Wholly turbulent flow
Laminar flow
Transition range
Smooth
0.05
0.04
0.03
0.02
0.015
0.01
0.008
0.006
0.004
0.002
0.001
0.0008
0.0006
0.0004
0.0002
0.0001
0.00005
0.00001
ε/D
2(10³) 4 6 8 2(10⁴) 4 6 8 2(10⁵) 4 6 8 2(10⁶) 4 6 8 2(10⁷) 4 6 8
10³ 10⁴ 10⁵ 10⁶ 10⁷
Re = ρVD/μ

Appendix II
Equivalent Length of Pipe Fittings

	Nominal standard pipe size (nominal bore) - millimetres							
Fitting	8	10	15	20	25	32	40	50
Gate Valve (Full Open)	0.091	0.091	0.106	0.134	0.170	0.225	0.262	0.335
Tee (Straight Through)	0.152	0.152	0.213	0.335	0.457	0.549	0.671	0.915
Tee (Side Outlet)	0.762	0.762	1.006	1.281	1.616	2.135	2.470	3.172
90° Elbow	0.427	0.427	0.518	0.640	0.793	1.067	1.250	1.586
45° Elbow	0.152	0.152	0.237	0.295	0.375	0.488	0.579	0.732
Angle Valve (Full Open)	2.440	2.440	2.836	3.507	4.483	5.886	6.893	8.845
Globe Valve (Full Open)	4.270	4.270	5.673	7.045	8.967	8.723	13.786	17.690
Ball Valve (Self Exhausting, Full Open)	0.006	0.030	0.091	0.122	0.152	0.219	-	-

Appendix III
Plant's Compressed Air System Layout

Appendix IV
Gantt Chart for Final Year Project I & II