# MODELING OF INDUSTRIAL AIR COMPRESSOR SYSTEM ENERGY CONSUMPTION AND EFFECTIVENESS OF VARIOUS ENERGY SAVING ON THE SYSTEM

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I want to thank God for guiding me to the finish line. I am grateful to my parents and family for all the endless love and support that they showed me throughout this journey. Special thanks to my friends who believed in me the most when I doubted myself. None of this would have been possible without every single one of you.

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# SYMBOLS

| | |
|---|---|
| $D$ | pipe diameter, mm |
| $\dot{E}$ | air power, J/s |
| $L$ | pipe length, m |
| $P$ | absolute pressure of compressed air, Pa, psi |
| $Q$ | volumetric flow rate of compressed air, $m^3/s$ , cfm |
| $T$ | temperature, $K,^{\circ} R$ |
| $TI$ | average temperature of inside air, K |
| $TO$ | annual average outside air temperature, K, $^{\circ}R$ |
| $V$ | volume of compressed air, $m^3$ |
| $\dot{W}$ | theoretical adiabatic compression power, J/s |
| $WI$ | work of compressor with inside air, hp |
| $WO$ | work of compressor with outside air, hp |
| $WR$ | work reduction |

**Subscripts**

a: the atmospheric state

1: the input of air power

2: the output of air power

**Greek symbols**

| | |
|---|---|
| $\rho$ | air density, $kg/m^3$ (1.225 $kg/m^3$) |
| $\kappa$ | airs adiabatic index |
| $\varepsilon$ | effectiveness |
| $\eta$ | efficiency |

# ABSTRACT

Ayoub, Abdul Hadi Mahmoud. M.S.M.E., Purdue University, December 2018. Modeling of Industrial Air Compressor System Energy Consumption and Effectiveness of Various Energy Saving on the System. Major Professor: Ali Razban.

The purpose of this research is to analyze the overall energy consumption of an industrial compressed air system, and identify the impact of various energy saving of individual subsystem on the overall system. Two parameters are introduced for energy consumption evaluation and potential energy saving: energy efficiency ($\eta$) and process effectiveness ($\epsilon$). An analytical energy model for air compression of the overall system was created taking into consideration the modeling of individual sub-system components: air compressor, after-cooler, filter, dryer and receiver. The analytical energy model for each subsystem included energy consumption evolution using the theoretical thermodynamic approach. Furthermore, pressure loss models of individual components along with pipe friction loss were included in the system overall efficiency calculation.

The efficiency analysis methods and effectiveness approach discussed in this study were used to optimize energy consumption and quantify energy savings. The method was tested through a case study on a plant of a die-casting manufacturing company. The experimental system efficiency was 76.2% vs. 89.3% theoretical efficiency. This showed model uncertainty at $\sim 15\%$. The effectiveness of reducing the set pressure increases as the difference in pressure increase. The effectiveness of using outside air for compressors intake is close to the compressors work reduction percentage. However, it becomes more effective when the temperature difference increase. This is mainly due to extra heat loss. There is potential room of improvement of the various component using the efficiency and effectiveness methods. These components include compres-

sor, intercooler and dryer. Temperature is a crucial parameter that determines the energy consumption applied by these components. If optimum temperature can be determined, plenty of energy savings will be realized.

# 1. INTRODUCTION

## 1.1 Problem Overview

Compressed air systems are very popular and widely used in industrial facilities. They are used for providing compressed air to the systems, such as machines, pneumatics, tools, and transportation. Compressed air usage varies depending on the type of industry. Compressed air systems are known to be one of the biggest energy consumer systems in a facility, usually 10% - 30% of the electricity consumption [1]. Thus, to lower the energy consumption of the plant, it is important to understand the energy consumption of this Significant Energy Use (SEU) equipment, which is critical for compressor optimization for increasing its efficiency. Figure 1.1 below shows the typical lifetime compressed air cost, 76% of the total cost is on energy consumption, while 12% for both implementation and maintenance each.

Many factors, such as equipment age, dirt, and component conditions, may lower the compressor system efficiency [3]. To quantify the effects, models are needed to relate the factors to the energy consumptions. These models are used to estimate the efficiency of each individual component in the system, and how air is reacting between stages, which affects energy consumption. This will help in identifying problems within the system and thus improving the overall system efficiency. Another important aspect that needs to be discussed is the effectiveness of various energy saving recommendations on the system. There is a difference between how much energy you can save, and what the impact on each component is. [4]

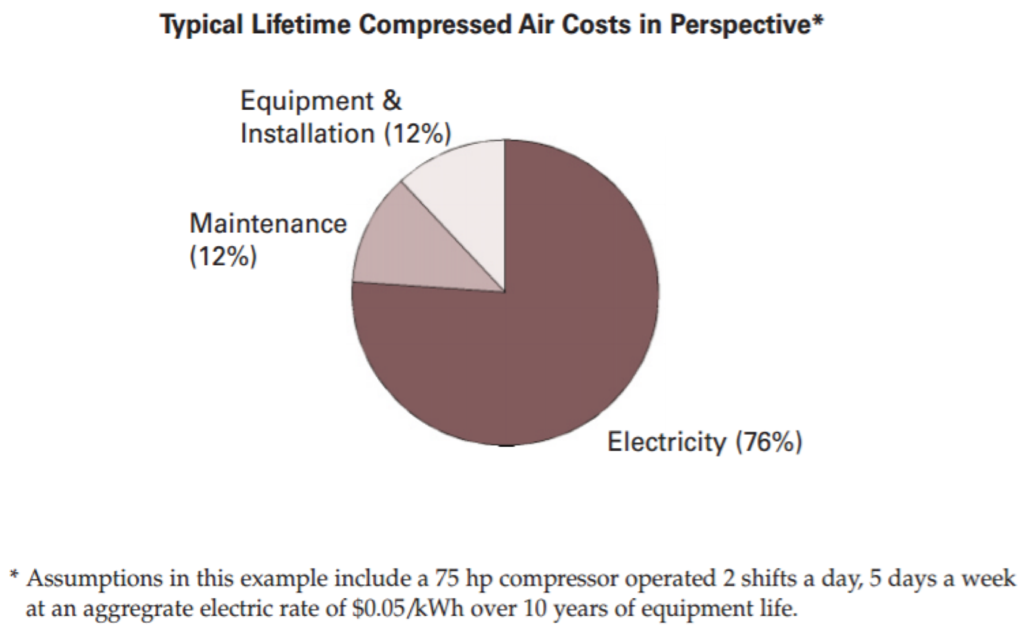


Fig. 1.1. Typical Lifetime Compressed Air Costs [2].

## 1.2 System Diagram

Figure 1.2 shows a typical air compressor system diagram. The fresh air intake flows into a filter before entering the compressor. Many of the industrial air compressors come in a closure that will include multiple components such as compressor, motor, aftercooler and a separator. Then, air flows into a wet receiving tank. The wet receiving tank helps store the air and drain any liquid and solid particulate that was not removed by the separator. A common practice is to place a filter between the receiver and air dryer. Refrigerant air dryers come in closures similar to air compressors. These closures include the main component of a refrigerant cycle, and that would be a heat exchanger, evaporator, condenser, and a compressor. Finally, air will flow through an optional filter before being stored in a dryer receiver tank. Pressure Flow Controller plays an important role in distributing the air to the plant side based on the required air demand. It is discussed furthermore in section 1.3.6.

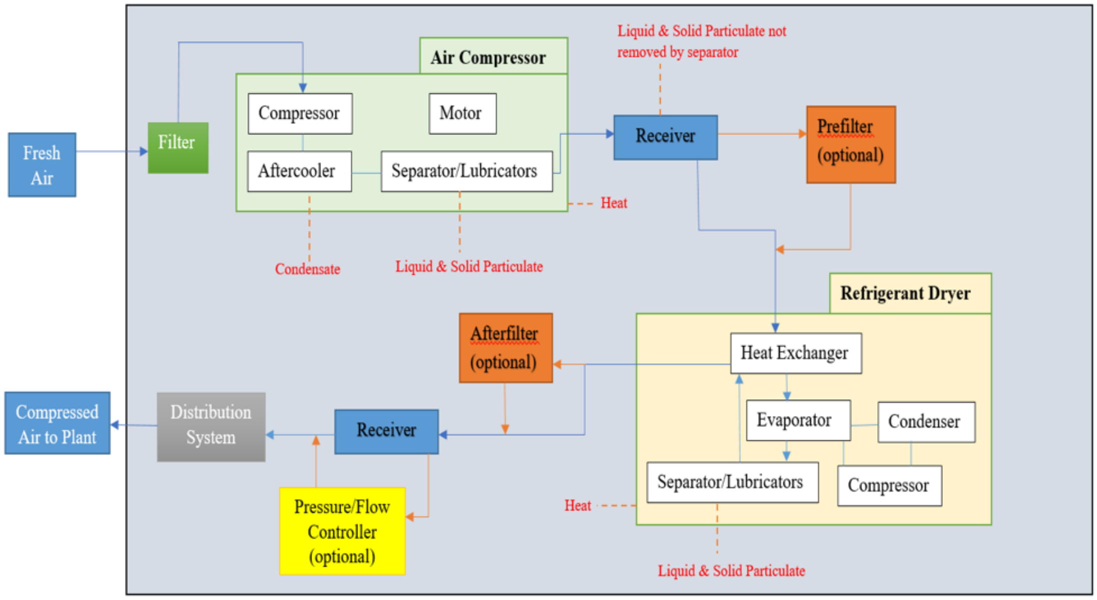


Fig. 1.2. Compressed Air System Diagram.

## 1.3 Components

### 1.3.1 Air Compressors

The air compressor is the basic element that makes up a compressed air system. Air compressors come in many different types and sizes. Figure 1.3 below shows the categories and subcategories of different types of air compressors.

Some are reciprocating and rotary which falls under the positive displacement compressors category. Other types are centrifugal and axial which falls under the dynamic compressors category. However, the most common type that is used in the industry is the rotary screw air compressor.

A rotary-screw compressor is a type positive displacement machines. It uses two rotating helical screws, which captures the air between them. Figure 1.4 below shows different compressor parts, and how air is flowing from one part to the other. The space in which the air is trapped becomes smaller as it moves down the axis of the screw. The compressed air is then discharged from the opposite end. This type of

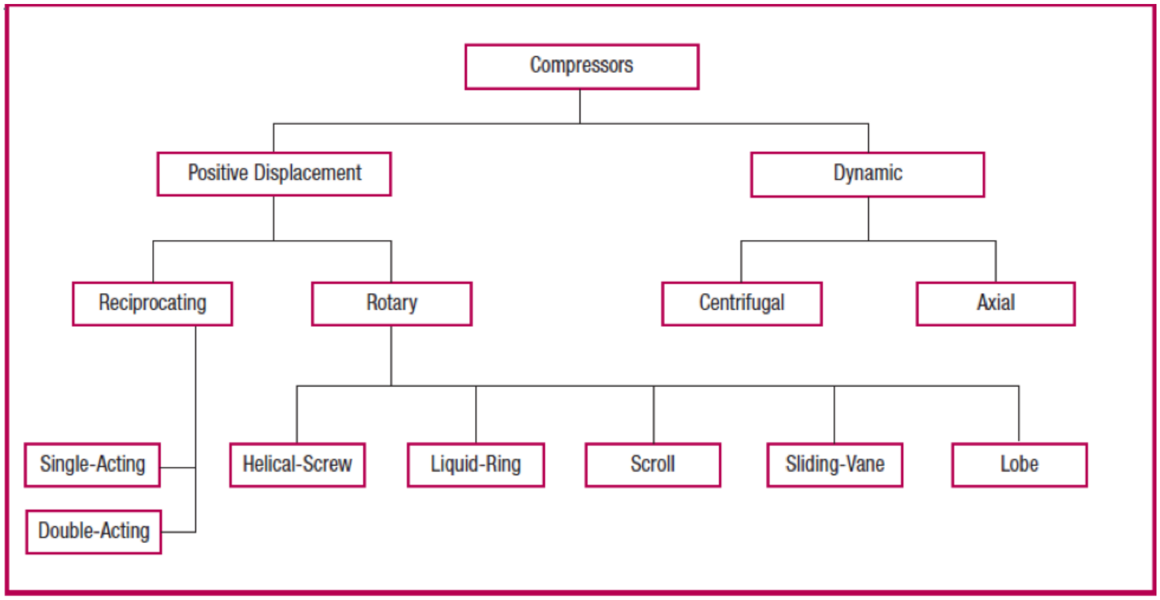


Fig. 1.3. Types of Air Compressors [1].

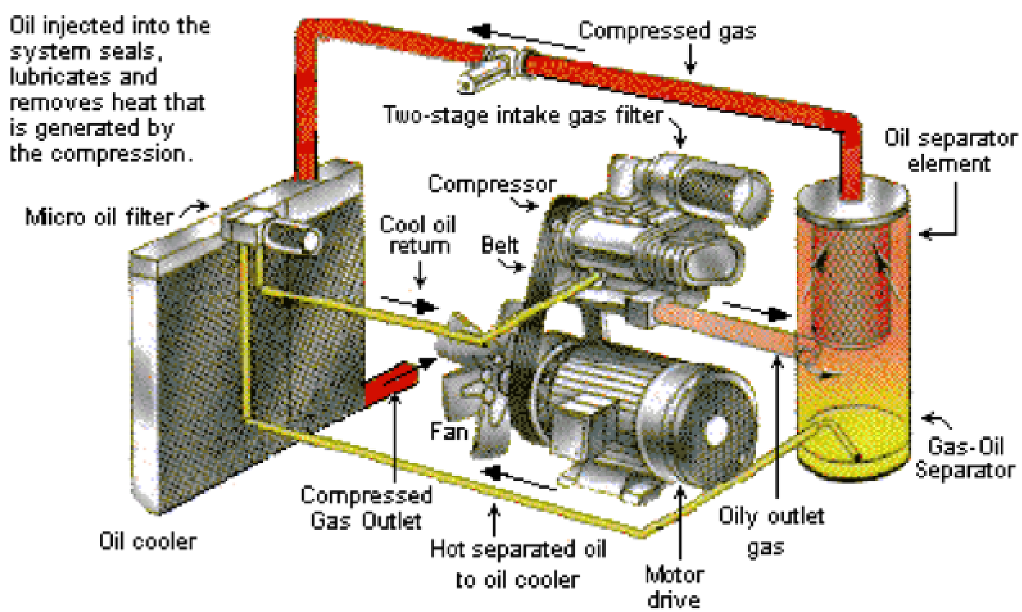


Fig. 1.4. Rotary-Screw Air Compressor [5].

compressor can be usually used to power air tools, pneumatics, and transportation in industrial applications. Sizes may vary from 5 five horsepower to 600 horsepower.

Table 1.1 below shows, on average, rotary air compressors produce a ratio of 4 CFM to 1 HP based on data extracted from CAGI data sheets collected for different air compressor models and sizes.

Table 1.1.
Compressors Data Sheets Information [6].

| Brand | Model | hp | Capacity (acfm) | Specific Power (kW/100 acfm) | cfm/hp | Capacity (m3/min) | Specific Power (kW/m³/min) | m³/min /hp |
|---|---|---|---|---|---|---|---|---|
| Sullair | TS20C-200HAC | 200 | 768 | 20.64 | 3.84 | 21.5 | 7.4 | 5.6 |
| Sullair | 3007PV | 40 | 144 | 18.51 | 3.60 | 4.0 | 6.6 | 1.12 |
| Sullair | V320TS-400LAC | 400 | 1,579 | 17.56 | 3.95 | 44.2 | 6.3 | 11.2 |
| Sullair | VCC200S-200HAC | 200 | 718 | 21.98 | 3.59 | 20.1 | 7.9 | 5.6 |
| Sullair | 1812 | 25 | 85 | 26.7 | 3.40 | 2.4 | 9.5 | 0.7 |
| Kaeser | CSD100S | 100 | 417 | 18.47 | 4.17 | 11.7 | 6.6 | 2.8 |
| Kaeser | DSD150 | 150 | 574 | 20.19 | 3.83 | 16.1 | 7.2 | 4.2 |
| Ingersoll Rand | HPE250 | 250 | 1,167 | 19.5 | 4.67 | 32.7 | 7.0 | 7 |
| Ingersoll Rand | R55I-A125 | 75 | 328 | 20 | 4.37 | 9.2 | 7.1 | 2.1 |
| Quincy | QSI245i | 60 | 240 | 20.5 | 4.00 | 6.7 | 7.3 | 1.68 |
| Quincy | QSI600 | 150 | 615 | 19.4 | 4.10 | 17.2 | 6.9 | 4.2 |

### 1.3.2 Air Dryers

Air dryers are one of the most common methods used to modify air quality. Their main purpose is to convert air into a form that can be used for different applications. Compressed air must be kept free of moisture. Without air dryers, wet compressed air running through the system would negatively impact the machines. Parts of the machines would rust and tear due to the wetness of the air passing through. However, excessively dry compressed air would result in unnecessary energy and costs. Table 1.2 lists some advantages and disadvantages for various types of air dyers. There is no best air dryer type. Application and setting help determine compatibility and choosing the ideal components [1].

Table 1.2.
Advantages and Disadvantages for Various Types of Air Dryers.

| Dryer Type | Advantage | Disadvantage |
|---|---|---|
| Refrigerant (cycling and non-cycling) | Energy efficient | Does not offer low dew point (~35F) |
| | Low cost (capital, operation) | Not recommended in sub-freezing ambient temperatures |
| | Low maintenance | |
| | not damaged by oil | |
| | low/no pressure drop | |
| Regenerative-desiccant | Low dew point | high capital cost |
| | heatless, can operate in hazardous locations | periodic replacement required (3-5 years) |
| | | purge air required |
| | | oil can damage desiccant material |
| Deliquescent | no electricity required for operation | Deliquescent material must be added to or replaced as it absorbs and melts. |
| | low pressure drop | high material cost |
| | easy to maintain | poor performance, deliquescent material could solidify in bed |
| | | Ecological problem |
| Membrane | no electricity required for operation | limited to low capacity systems |
| | Low dew point (~-40F) | high air loss |
| | silent operation | oil can damage membrane |
| | low maintenance | |

Various types of dryers are:

1. Refrigerant (cycling and non-cycling): does not offer a very low dew point as other types and it is most commonly used in industry. However, it is not recommended for operation in sub-freezing ambient temperatures. Filtration is recommended even though the oil in air stream does not damage the dryer. This type has many advantages such as low initial capital cost, operating cost, and maintenance costs.

2. Regenerative-desiccant: uses a porous desiccant that adsorbs the moisture by collecting it in its myriad pores, allowing large quantities of water to be retained by a relatively small quantity of desiccant. The moisture can be reduced with

regeneration process by applying purge air and heat. It has very low dew point and moderate cost of operation. However, the initial cost is relatively high.

3. Deliquescent: use a drying medium that absorbs, rather than adsorbs, the moisture in the compressed air. This type of dryer does not require electricity for operation.

4. Membrane: water vapor passes through hollow fibers of specially designed membranes faster than air. This process reduces the amount of water vapor in the air stream at the outlet of the dryer, suppressing the dew point.

### 1.3.3 Air Receivers

Air receivers, or compressed air storage tanks, is a crucial part in compressed air systems. They are used as temporary storage for compressed air to meet peak demand from systems and optimizes the efficiency of the plant. It also helps control the system pressure by controlling the rate of change in a system. Receivers should be properly sized to meet the required demand. There are several factors that should be considered when sizing a receiver. These factors are: minimizing pressure fluctuations/drops, meeting short term peak air demands, energy considerations, and safety considerations.

There are two types of storage tanks, wet and dry. A receiver before a dryer is filled with saturated air. A sudden demand for air exceeding the dryer rating, may result in overloading the dryer and a higher downstream dew point. A receiver placed after the dryer will be filled with air already dried and can satisfy a sudden demand without impacting the dryer performance and pressure dew point.

After defining air receivers and discussing their importance/functionality in a compressed air system, it will just be reasonable to utilize in every system. Air receivers are commonly used in industrial sectors, especially intensive compressed-air user facilities. However, there are still plenty of industrial plants that still do not take advantage of this great component. This could be due to lack of knowledge, poor design capabilities, and cutting capital cost.

Compressed air storage is an essential component of compressed air systems. Abel's and Kissock's study discussed methods to properly size compressed air storage tank to reduce system energy use. Mathematical equations were created using thermodynamic equations. These equations addressed storage, pressure, and air flow. Next, these relations were used to calculate the relationship between the volume of storage and cycle time in load-unload compressors. The study explained the effect of pressure drop between the compressor system and storage tank on cycle time. The authors developed guidelines for compressed air storage that minimize energy consumption. [7]

### 1.3.4 Pressure/Flow Controller

A pressure/flow controller is a device that serves to separate the supply side of a compressed air system from the demand side. Most plant air systems have an ever changing fluctuation of demand, and changes in plant air pressure. Peak air demands will draw from the air system, and tend to disrupt the air capacity. These changes inevitably lead to inconsistent production, poor product quality, and wasted energy. The system will utilize the supply-side storage tank installed with the compressors operating at an appropriately set control pressure, and the controller will deliver the desired plant air pressure set point. Peak air consumptions will be drawn from storage, and less compressor horsepower is required for peak events.

A pressure/flow controller is used to:

- Stabilize of the plant production quality by stabilizing the compressed air.
- Optimize operations of the Air Compressors.
- Make constant adjustments to stabilize the system air pressure in responses to the ever-changing fluctuations of demand.

- Minimize waste through air leak reduction.
- Reduce compressor energy consumption.
- Improve compressor controls and response to changing air requirements.

### 1.3.5 Air Inlet Filters

An air inlet filter protects the compressor from any atmospheric particles such as dust and dirt. It is the most important filter on the compressor. It will prevent the dust from getting sucked into the oil, oil filter, and oil separator. Any of these issues could lower the compressors performance and increase both maintenance costs and energy bills. Filtration only to the level required by each compressed air application will minimize pressure drop and resultant energy consumption.

### 1.3.6 Aftercoolers

Aftercoolers are installed to cool the discharge from the compressor. They also reduce the moisture and water vapor in the compressed air system. Aftercoolers are placed after the final stage of the air compression. The air is cooled below its dew point converting the water vapor into liquid and drained from the system.

## 1.4 Literature survey

Air compressor systems may come in various designs. However, the systems usually consists of six sub-system which are crucial in establishing the proper design since the system revolves around its operating principles. The air compressor's key function is to produce required air capacity at a desired pressure. These are various compressor types, such as reciprocating compressors, centrifugal compressors and finally the packaged rotary air compressor. Rotary air compressor is widely utilized in the small/mid-sized industrial sector, and thus will be used in the modeling of the air compressor system in this research. One of the main component of the system is the air cooler, which is found in two different locations serving generally the same purpose. One is the intercooler is inside the compressor and the other being the after-cooler, located right after outlet of the air compressor. The air cooler just likes the filter and dryer is used to treat the air as it flows through the system to acquire preferred characteristics of the air before reaching the end users. However, rather than cooling like the air coolers, the filters and dryers are used to remove impurities and monitor after compression. To assist in achieving the required cooled compressed air, the system is equipped with an air receiver (storage) tank or a tank reservoir that exclusively functions to store and cool the excess compressed air and releasing it at times of need. To achieve the desired flow, the distribution system utilizes pipes and valves that will coordinate the flow of compressed air. In the distribution, parameters like isolation and ventilation are a major concern.

Industrial Air Compressors provide a mean of delivering enough compressed air at a demanded pressure to drive tools, machines and processes. Demand on such an operating system is high and often found but not exclusively at Automotive, Furniture, General Manufacturing, Metals Fabrication, Petroleum and Rubber Industries. Functions of air compressor systems aim to achieve clamping, tool powering, controls and actuators, blending, stamping, spraying, forming, Assembly station powering or many more. Since the demand on air compressors is high, the mathematical modeling would help in process optimization for improving the overall efficiency [8].

Many machine applications require compressed air to operate. This process consumes a large amount of energy. Heat loss generates as a by-product. The amount of heat loss indicates whether energy-consuming cooling is required or not. Zust presented a theoretical approach to quantify the energy equivalent of compressed air and its by-products. A model-based method is set up to describe the required physical relationships for the compressor and its peripheral components. On-site measurements were collected and used to validate the results of the theoretical approach. This study resulted in a general method for the theoretical energy equivalent calculation, including the compressor and treatment of heat loss, is possible. [9]

In 2007, China's compressor electricity consumption reached 200 billion kWh which amounted to approximately 9.4% of the national total electricity consumption [10]. Taking into consideration that compressors amounted to a high percentage of the electric consumption, an elevated concern with research in improving compressor efficiency embarked. A practice to analyze the overall energy of compressed air systems by the school of automation science and electrical engineering in Beijing, China describes efficiency analysis methods used for the lubricant-injected rotary screw compressor and the after-cooler. The research aims to assist in the development of a more precise data collection of the system to find its efficiency, allowing a better energy saving equipment selection to go with the system design. Such a selection is achieved by comparing the output pressure of the compressed air versus the overall efficiency. The method of improvements was initiated by factoring out the losses in the compressor and the after-cooler. Taking into consideration the efficiency of the electric motor powering the compressor and then improving it by VFDs. The motor and VFD efficiencies were factored in alongside the efficiency of the compressor to compare with ideal systems and overall energy efficiency of the air compressor system [10].

Regardless whatever air compressors were installed the compressor system will cause energy loss of more than 80%. The study found that the energy loss was due to unoptimized air compressor system operation and air leakage. The paper also suggested investing in air compressor systems as the investment could be cost-effective. If companies invest $84,000 in air compressors to improve efficiency, the payback period is about six months. [11]

The paper presents a new control method for the compressed air system. The control method helps to improve energy management of electrically driven compressors through Real-Time Physical System. The System allows a real-time scheduling algorithm to generate the sequence of opening/closing action. The result, as presented in the paper, shows about 6% energy savings. [12]

Mousavi et al. Air compressor systems have a significant impact on the energy consumption and efficiency of industrial operations. Systems can consist of one or more air compressors networked together. Compressors with constant or variable speed systems have dynamic energy consumption profiles. Also, they require an algorithm to control multiple compressors. The control system reacts to the air flow rate and pressure parameters that determine the load/unload sequence. The study provided an overview of techniques to model energy consumption as well as various approaches used to control the air compressor system. A state-based modeling technique is used to develop a simulation model that includes both fixed and variable speed drive compressors. Mousavi's method was applied to analyze different control system strategies. An industrial case study was used to validate the model. [13]

This study reviewed common types of compressor control for reciprocating and rotary air compressors. Relations were derived for estimating compressed air output as a function of the type of control and motor loading. A method was developed to estimate the ratio of compressed air output to energy input. This method was based on input power to the compressor. [14]

Saidur et al. defined a comprehensive literature review about compressed air energy use, savings, and the payback period of energy efficient strategies. Previous studies in this area showed that for an electric motor used in an air compressor system, a large amount of energy consumption and utility cost could be saved using high-efficiency motors and applying VSDs in matching speed requirements. In addition, notable amounts of energy and emission are reducible through various energy-saving strategies. Payback periods for different energy savings measures have been identified and found to be economically viable in most cases. [9]

Gopalakrishnan et al. conducted an experimental study study was done on an automotive facility type. Several dianostic tools and methods were reviewed to detemine energy saving measures on the air compressor system. These measures included air leak reduction, reducing set pressure and heat recover. The goal was to have a data driven approach to monitor and improve compressed air systems in the manufacturing facilities using different tools and equipment. It was possible that total compressed air energy consumption can be significantly reduced. [15]

To fully comprehend and analyze the performance of an engineered air compressor system, one must not rely solely on the scientific standards and engineering formulations that make up the system. Compressed air process description is the best portrayed through the use of mathematical modeling and so is the case in the air compressor system. The resulting mathematical modeling of a system proves dividend while simulating the system process. The biggest benefit is the ease of descriptive information transfer. This information, once simulated, will unfold the required results of the variables in interest [16].

The use of computer modeling to analyze the performance of rotary screw compressors has been discussed [17]. The authors start by stating the importance of computer modeling and the factors that will decide the preferred approach. The authors used a simulation software to analyze the performance data from the model since experimental analysis is expensive and impractical. The research aimed to create a computer model based analysis program which can effectively analyze general types of compressors. In the complex shape of the compressor volume curve calculations were simplified by the principle of virtual work. The screw rotors were designed according to the volumes. Finally, they applied the compression cycle equations to evaluate energy consumption. Inlet velocities, Temperature, Pressure Drop across the inlet port and heat flow from gas and oil were kept constant. The modeling method's key concern was the leakage flow rate of the screw compressor. Leakage flow rate values were used to calculate the efficiency of the system. In comparison to the ideal efficiency of the software calculated air compressed system efficiency and performance ratings [17]. This process is further described in Figure 1.5:

Maxwell et al. presented a study on a dynamic compressed air system simulation model that was developed using MATLAB/SIMULINK. The model considered ther-

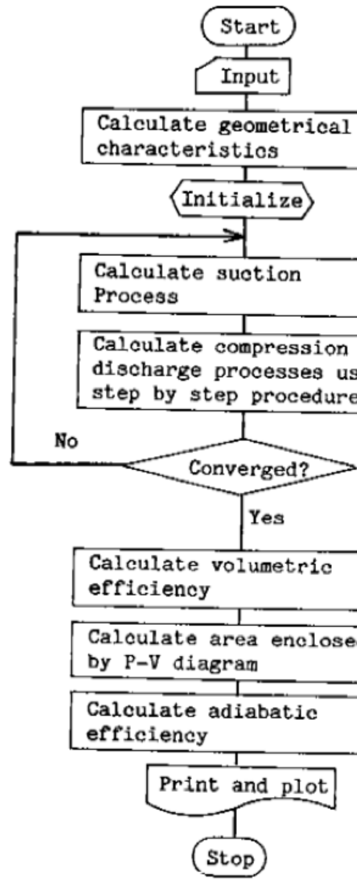


Fig. 1.5. Computer Analysis Flow Diagram Example [17].

modynamic and fluid dynamic interactions within the compressed air system under a variety of operating conditions and control strategies. The system model is composed of component models that are linked to form the compressed air system. Models focused on two screw air compressors, an auxiliary air cooler, a receiver, the system piping, and both regulated and unregulated air demand. Dynamic system modeling provided an analytical tool for evaluating compressed air system performance under a variety of operating conditions and control strategies. [4]

Benedetti et al. (2017) performed an analysis of energy audits on CAS used by energy-intensive enterprises found that CAS use an average of 7% of these industries energy expenditures [18]. In an effort to identify markers for calculating the use and efficiency of CAS, they found that the kWhe CAS/m3 can be used as a cross-sectional indicator of energy efficiency in the production of compressed air. Dindorf (2012) highlighted that several other specific basic markers for calculating the performance of CAS are specific power, annual energy cost, cost of compressed air, compressed air leaks, and pressure drops in the system [19]. Qin and McKane from Shanghai Energy Conservation Service Center compiled over 50 energy audits completed and found the potential for energy saving was up to 50% [20]. They illustrated that this saving can be accomplished by limiting the use of compressed air to only applications that do not have more economical source, minimizing leaks as they account for up to 30% of output, pressure control to control rapid drops thus saving energy expenditure, and optimizing the settings and configuration of each individual CAS.

This paper introduces the modeling and control of a new type of compressed air energy storage system (CAES) for wind turbines. The system captures excess power before it is generated, allowing it to reduce the size of electrical components based on demand rather than supply. Energy is stored in a high-pressure two-chamber liquid compressed air storage vessel. It utilizes the power density and aerodynamic energy density of the hydraulic system in the "open accumulator" architecture. In order to realize accumulator pressure regulation and generator power tracking, a nonlinear controller based on energy based Lyapunov function is designed. The nonlinear controller is then modified to distribute control between hydraulic and pneumatic components according to its bandwidth capability. As a result, the liquid-piston air compressor/expander will loosely maintain the accumulator pressure ratio while the down-tower hydraulic pump/motor accurately tracks the required generator power. The control scheme also allows the accumulator to be used as a damper of the storage system by absorbing the power disturbance in the hydraulic path generated by the gusting wind. A set of simulation case studies demonstrate the operation of the combined system with nonlinear controllers and demonstrate how the system can be used for load balancing, minimizing electrical systems and maximizing benefits. [21]

### 1.5 Research Needs

In general, every industrial facility is unique in its way. Whether it is the facility type, size, and the type of equipment it has. However, majority of these facilities include a compressed air system. This shows how much the system is utilized, and that any potential improvement could have a high impact on the entire sector. To have a better understanding of this system, more in-depth analysis needs to be done. It is important to know what makes up the air compressor system, what components are involved and are dominant energy consumers. The other thing to consider is the type of parameters that affect the energy system energy consumption. By answering these questions, one would be able to identify potential improvement opportunities.

Multiple energy saving measures are identified to address air compressor systems, and provide means to cut back the energy usage. For example, reducing pressure set point and utilizing outside air intake in colder temperatures are two common recommendations. Facilities tend to set the air compressor higher than what the plant demand requirement is. This could be due safety factors, lack of knowledge, or not realizing the set pressure impact on the system. Utilizing outside air intake in colder temperatures is a great of way of reducing the compressor work. Since colder air is denser, it will require less work to compress. These two recommendations are common, but they are currently addressed in a way that only consider the work reduction by the air compressor. What if they have more impact on the rest of the system? Will this be realizing more energy savings? How is it affecting the other system components? And is it worth looking into?

Many papers and textbooks have been published in the area of compressed air modeling. This includes modeling of air power, air consumption, and energy consumption. However, none of these focus on the individual components of the system. This is an important gap because different system losses are happening at these components such as pressure loss and heat loss. By knowing such information, one would be able to see what component is a major energy contributor. This will draw a better picture on how the system is operating, what parameter inputs are essential. The other area that will need to be investigated is the impact on different energy saving recommendations on the air compressor system as a whole, including all of its components. This could result in realizing higher energy savings.

In this research, modeling of the individual component is done, which would help to optimize overall system energy consumption. The model evaluates the system from the moment ambient air is entered, till it gets stored in a receiver tank. By identifying and investigating four process parameters, the energy efficiency of individual components, along with the overall system, were calculated. Also, effectiveness method was created. This targeted the two energy savings measures discussed earlier, with the ability to explore more measures.

## 1.6 Research Goals

The goal of this research is to optimize the energy consumption of an air compressor system by using models of various elements of the compression system. The models are based on various parameters such as pressure, temperature, and flow rate. It focuses on rotary screw compressors and refrigerant dryers since these are the most common types used in industrial sector. The system considered only one air compressor, with a capability of future expansion. The model was created using Excel file that will allow the user to input various data and show the total energy consumption of the system. The models considered both ideal and real cases. Ideal case was defined as the energy consumption of the system without considering any system losses such as pressure loss. Real case was defined as the energy consumption of the system including pressure and heat loss. Four process parameters including Air Power, Pressure, Temperature and Flow Rate were addressed in each case. The value of these parameter was calculated at each component stage.

List of research goals:

- Create overall air compressor energy model. This model investigate four process parameters (Flow Rate, Air Power, Pressure and Temperature).
- Create pressure loss models.
- Create effectiveness methods to determine the accuracy of the current energy savings measures calculations.
- Calculate efficiency of individual system components along with the overall system efficiency.

## 1.7 Research Boundaries

The research defined compressed air system from the time air enters the compressor all the way until it is stored in the receiver tank. It will also consider the piping layout for pressure drop due to friction loss. Fluctuation of demand from the plant side was not considered in this study. This study assumed no pressure drop from leakage within the system. The control parameters of the system are addressed in chapter 2. Figure 2.1 shows parameter inputs and outputs in each component.

# 2. METHODOLOGY

Figure 2.1 displays the various components in the compressed air system that were considered in the model, including inputs and outputs. The models focused on four main process parameters: Volumetric Flow Rate ($Q$), Pressure ($P$), Temperature ($T$), and Air Power ($\dot{E}$). These are the main four parameters that impact the energy consumption and identifies air quality. The input parameters were labeled with a number $(1, 2, 3, 4, 5)$, and the output were labeled with a number prime $(1', 2', 3', 4')$ for each component. The compressor and refrigerant dryer required work inputs $W1, W2.H1, H2, H3$ which represented the heat loss in the process due to temperature changes. Tables 2.1 and 2.2 list the parameter changes in each component, whether it increases, decrease, or stays constant.

Table 2.1.
Parameter Changes in Components for Ideal Model.

| **IDEAL** | ***Parameter*** | | | | Heat Loss (H) |
|---|---|---|---|---|---|
| ***Component*** | Flow Rate (Q) | Pressure (P) | Temperature (T) | Air Power ($\dot{E}$) | |
| Compressor | decrease | increase | increase | constant | constant |
| Intercooler | decrease | constant | decrease | decrease | yes |
| Aftercooler | decrease | constant | decrease | decrease | yes |
| Filter | constant | constant | constant | constant | constant |
| Dryer | decrease | constant | decrease | decrease | yes |

Air compression in the lubricant-injected rotary screw compressor is actually rapid and this makes it near an adiabatic process [10]. The theoretical work by the compressor shaft ($W1$) was calculated using Equation 2.1 [10]:

$$\dot{W}1 = \frac{(i+1)k}{(k-1)} \times \dot{Q}mRT_a((\frac{P}{P_a})^{\frac{k-1}{(i+1)k}} - 1) \tag{2.1}$$

Table 2.2.
Parameter Changes in Components for Real Model.

| **REAL** | ***Parameter*** | | | | Heat Loss (H) |
|---|---|---|---|---|---|
| ***Component*** | Flow Rate (Q) | Pressure (P) | Temperature (T) | Air Power ($\dot{E}$) | |
| Compressor | decrease | increase | increase | constant | constant |
| Intercooler | decrease | decrease | decrease | decrease | yes |
| Aftercooler | decrease | decrease | decrease | decrease | yes |
| Filter | constant | decrease | constant | decrease | yes |
| Dryer | decrease | decrease | decrease | decrease | yes |

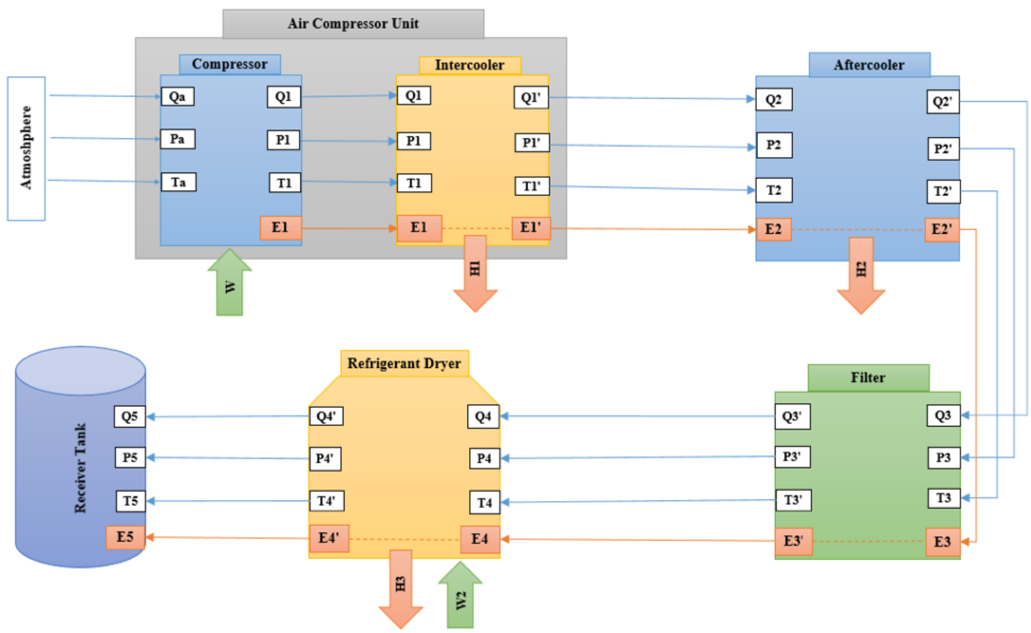


Fig. 2.1. Energy Flow Diagram of a Compressed Air System.

R-22 was considered as refrigerant dryer work is calculated using Equation 2.2.

$$\dot{W}2 = \dot{Q}m(h_2 - h_1) \tag{2.2}$$

Air power is the available energy rate of compressed air, which is a relative energy compared with the energy of uncompressed air at the atmospheric state. When the temperature of compressed air is not equal to the atmospheric temperature, air power was calculated using Equation 2.3, [10].

$$\dot{E} = P_a Q_a (ln\frac{P}{P_a} + \frac{k}{k-1}(\frac{T-T_a}{T_a} - ln\frac{T}{T_a})) \tag{2.3}$$

Considering air as an ideal gas ($PV = mRT$, $P_a Q_a = \dot{Q} mRT_a$) the Equation 2.3 can be written in terms of mass flow rate as shown in Equation 2.4 [22],

$$\dot{E} = \dot{Q} mRT_a (ln\frac{P}{P_a} + \frac{k}{k-1}(\frac{T-T_a}{T_a} - ln\frac{T}{T_a}))) \tag{2.4}$$

In the adiabatic process there is no heat loss and the isotonic equation [10] was used for volumetric flow rate calculation as shown in Equation 2.5:

$$\frac{Q_{out}}{T_{out}} = \frac{Q_{in}}{T_{in}} \tag{2.5}$$

# 3. TERMINOLOGY

Efficiency and effectiveness parameters were used to evaluate the effect of the individual subsystem on overall system energy performance. Efficiency ($\eta$) is the overall efficiency of a system and is defined in Equation 3.1, which is equal to the energy consumption of the entire system (real) divide by the total energy consumption of the system (ideal).

$$\eta = \frac{E_{real}}{E_{ideal}} \tag{3.1}$$

"$E_{real}$" is the total system energy consumption based on data which is summation of ideal case plus energy losses through system. This value was obtained using theoretical energy equations for each component to find the total energy output, plus considering efficiency of the motors and the pressure drops of the different components. "$E_{ideal}$" is the total system energy consumption based on rated data and using theoretical energy equations for each component to find the total energy output.

A new parameter called Effectiveness ($\varepsilon$) was introduced for evaluation of the system. Effectiveness is the potential amount of energy saving for each recommendation of the system and it is given in Equation 3.2. For example, reducing set pressure, this will calculate the proposed total energy consumption considering the pressure losses. The larger the effectiveness value, more impact it has on the overall system.

$$\varepsilon = 1 - \frac{E_{proposed}}{E_{real}} \tag{3.2}$$

where "$E_{proposed}$" is proposed energy consumption of the energy saving recommendation following "$E_{real}$". The effect of potential recommendations using thermodynamics modeling are discussed in the following sections.

## 3.1 Pressure Loss Models

The pressure drop models were created using experimental data from the components manufacturers provided in the data sheets. Specific type of after-cooler, air dryer, and filter were chosen to complete the models.

### 3.1.1 Aftercooler

Figure 3.1 shows the pressure drop model for an after-cooler. These are experimental data points from the after-coolers manufacturer [23]. Equations 3.3 and 3.4 shows the linear regression equation of this set of points for SI and English units respectively.

$$dP = 8 \times 10^{-5} Q + 1.586 \tag{3.3}$$

$$dP = 1163.5 \times Q + 10935 \tag{3.4}$$

### 3.1.2 Refrigerant Air Dryer

Figure 3.2 shows the pressure drop model for a refrigerant air dryer. These are experimental data points from the refrigerant air dryers manufacturer [24]. Equations 3.5 and 3.6 shows the linear equation of this set of points for SI and English units respectively.

$$dP = 0.0004 \times Q + 2.336 \tag{3.5}$$

$$dP = 5637.15 \times Q + 16106 \tag{3.6}$$

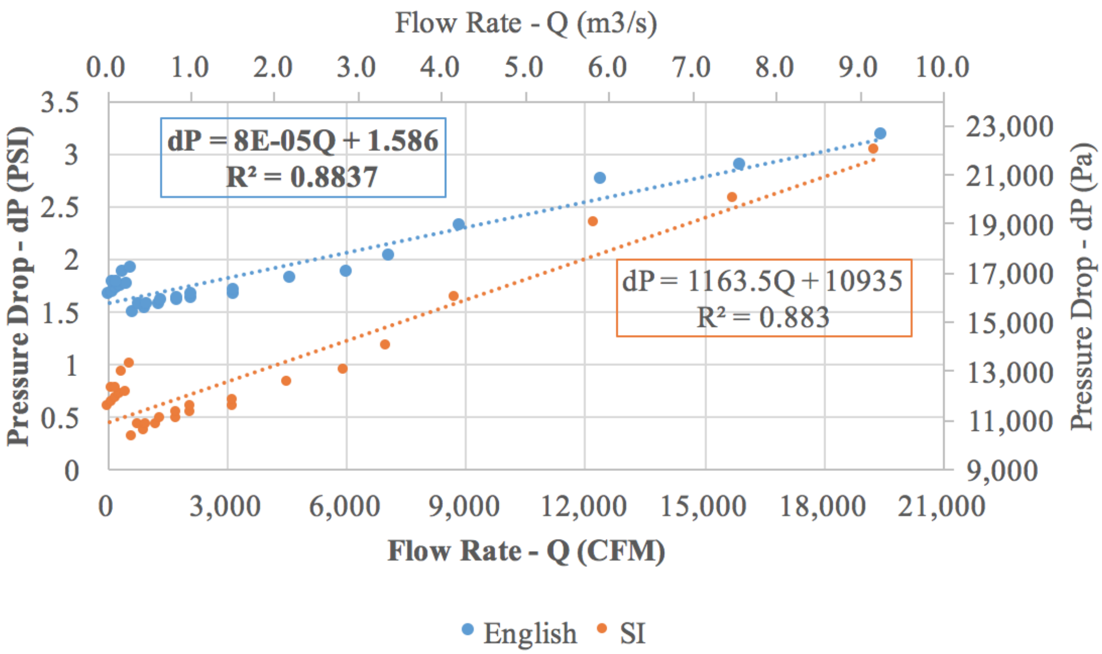


Fig. 3.1. Pressure Drop Model for Aftercooler.

### 3.1.3 Filter

Table 3.1 shows various grades of air filters with the estimated dry and wet pressure drop in each [25].

Table 3.1.
Estimated Pressure Drop in an Air Filter.

| Element Grade | Element Type | Dry dP (psi) | Dry dP (Pa) | Wet dP (psi) | Wet dP (Pa) |
|---|---|---|---|---|---|
| Grade C | Coarse | 0.44 | 3,034 | 0.73 | 5,033 |
| Grade F | Fine | 0.73 | 5,033 | 2.17 | 14,962 |
| Grade S | Superfine | 0.87 | 5,998 | 2.9 | 19,995 |

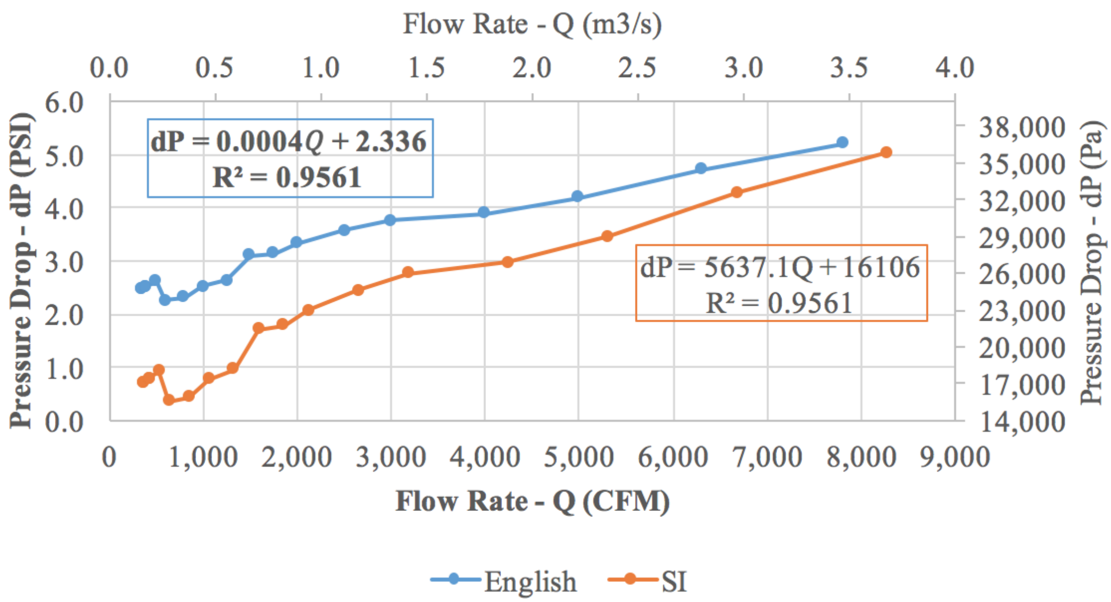


Fig. 3.2. Pressure Drop Model for Refrigerant Air Dryer.

### 3.1.4 Pipe Pressure Loss

A pressure drop theoretical analysis was created considering Blasius and Prandtl formulations, Equation 3.7 [26]. A pressure drop study was conducted for several pipe types including Schedule 40 Steel, Type K Copper, Type L Copper, and Schedule 10 Stainless Steel. Table 3.2, 3.3, and 3.4 show the ranges of pressure loss of various pipe material for common pipe sizes using Equation 3.7. Since pipe diameter size vary among systems, nine diameter sizes were considered. The pipe length is set at 100 ft (30.5 m) and nominal pressure at 50 psi (344,738 Pa), 100 psi (689,476 Pa) and 150 psi (1,034,214 Pa) which are common in industry. In these tables, a pressure drop above 15 psi (103,421 Pa) is in general not relevant in industry and the formula may not be valid.

$$dP = 7.57 \times Q^{1.85} \times \frac{L \times 10^4}{D^5 \times 0.01P} \tag{3.7}$$

Table 3.2.
Pressure Loss for Different Pipe Types at Different Pipe Sizes at a Pressure of 50 psi (344,738 Pa) and Pipe Length of 100 ft (30.5 m).

| at 50 psi, 100 ft | | | Pipe Type | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Schedule 40 Steel | | Type K Copper | | Type L Copper | | Schedule 10 Stainless Steel | |
| Pipe Size | Flow Rate (cfm) | Flow Rate (m3/min) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) |
| 1/2" | 4 - 42 | 0.1 - 1.2 | 0.13 - 13.4 | 896 - 92,390 | 0.31 - 30.62 | 2,137 - 211,118 | 0.26 - 25.89 | 1,793 - 178,505 | 0.13 - 13.37 | 896 - 92,183 |
| 3/4" | 7 - 92 | 0.2 - 2.6 | 0.12 - 13.7 | 827 - 91,011 | 0.2 - 22.68 | 1,379 - 156,373 | 0.15 - 17.46 | 1,034 - 120,383 | 0.14 - 16.36 | 965 - 112,798 |
| 1" | 11 - 177 | 0.3 - 5 | 0.08 - 13.73 | 552 - 94,665 | 0.1 - 17.89 | 689 - 123,347 | 0.08 - 15.42 | 552 - 106,317 | 0.08 - 14.94 | 552 - 103,008 |
| 1 1/4" | 25 - 353 | 0.7 - 10 | 0.09 - 12.57 | 621 - 86,667 | 0.08 - 21.03 | 552 - 144,997 | 0.08 - 19.42 | 552 - 133,896 | NA | NA |
| 1 1/2" | 35 - 530 | 1 - 15 | 0.08 - 12.31 | 552 - 84,875 | 0.08 - 18.69 | 552 - 128,863 | 0.08 - 17.25 | 552 - 118,935 | 0.08 - 12.15 | 552 - 83,771 |
| 2" | 71 - 1,059 | 2 - 30 | 0.08 - 12.72 | 552 - 87,701 | 0.07 - 16.64 | 483 - 114,729 | 0.09 - 15.58 | 621 - 107,420 | 0.08 - 13.88 | 552 - 95,699 |
| 2 1/2" | 106 - 1,765 | 3 - 50 | 0.07 - 13.46 | 483 - 92,804 | 0.08 - 14.43 | 552 - 99,491 | 0.07 - 13.57 | 483 - 93,562 | NA | NA |
| 3" | 194 - 3,354 | 5.5 - 95 | 0.08 - 14.9 | 552 - 102,732 | 0.08 - 19.51 | 552 - 134,517 | 0.08 - 18.28 | 552 - 126,036 | 0.08 - 19.29 | 552 - 133,000 |
| 4" | 530 - 5,295 | 15 - 150 | 0.13 - 8.9 | 896 - 61,363 | 0.07 - 11.04 | 483 - 76,118 | 0.15 - 10.38 | 1,034 - 71,568 | 0.09 - 6.19 | 621 - 42,679 |

Table 3.3.
Pressure Loss for Different Pipe Types at Different Pipe Sizes at a Pressure of 100 psi (689,476 Pa) and Pipe Length of 100 ft (30.5 m).

| at 100 psi, 100 ft | | | Pipe Type | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Schedule 40 Steel | | Type K Copper | | Type L Copper | | Schedule 10 Stainless Steel | |
| Pipe Size | Flow Rate (cfm) | Flow Rate (m3/min) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) |
| 1/2" | 4 - 64 | 0.2 - 1.8 | 0.24 - 14.16 | 1,655 - 97,630 | 0.15 - 32.42 | 1,034 - 223,528 | 0.13 - 27.41 | 896 - 188,985 | 0.24 - 14.15 | 1,655 - 97,561 |
| 3/4" | 11 - 124 | 0.3 - 3.5 | 0.13 - 11.87 | 896 - 81,841 | 0.1 - 19.65 | 689 - 135,482 | 0.08 - 15.13 | 552 - 104,318 | 0.15 - 14.18 | 1,034 - 97,768 |
| 1" | 18 - 265 | 0.5 - 7.5 | 0.1 - 14.54 | 689 - 100,250 | 0.08 - 18.94 | 552 - 130,587 | 0.07 - 16.33 | 483 - 112,591 | 0.11 - 15.82 | 758 - 109,075 |
| 1 1/4" | 32 - 530 | 0.9 - 15 | 0.07 - 13.3 | 483 - 91,700 | 0.08 - 22.26 | 552 - 153,477 | 0.09 - 20.56 | 621 - 141,756 | NA | NA |
| 1 1/2" | 49 - 706 | 1.4 - 20 | 0.08 - 10.48 | 552 - 72,257 | 0.09 - 15.91 | 621 - 109,696 | 0.08 - 14.68 | 552 - 101,215 | 0.08 - 10.35 | 552 - 71,361 |
| 2" | 99 - 1,589 | 2.8 - 45 | 0.08 - 13.47 | 552 - 92,872 | 0.08 - 17.61 | 552 - 121,417 | 0.07 - 16.49 | 483 - 113,695 | 0.08 - 14.69 | 552 - 101,284 |
| 2 1/2" | 159 - 2,648 | 4.5 - 75 | 0.08 - 14.25 | 552 - 98,260 | 0.08 - 15.28 | 552 - 105,352 | 0.08 - 14.37 | 552 - 99,078 | NA | NA |
| 3" | 282 - 3,530 | 8 - 100 | 0.08 - 8.19 | 552 - 56,468 | 0.08 - 10.72 | 552 - 73,912 | 0.07 - 10.05 | 483 - 69,292 | 0.08 - 10.61 | 552 - 73,153 |
| 4" | 706 - 8,825 | 20 - 250 | 0.11 - 11.47 | 758 - 79,083 | 0.08 - 14.21 | 552 - 97,975 | 0.07 - 13.36 | 483 - 92,114 | 0.07 - 7.96 | 483 - 54,882 |

## 3.2 Efficiency

In compressed air system energy analysis, usually just the motor efficiency of the air compressor is considered and the other components of the system are neglected. So, to be able to find out the efficiency of the entire system, one needs to examine each individual component and find out how the other sub-system affecting the work

Table 3.4.
Pressure Loss for Different Pipe Types at Different Pipe Sizes at a Pressure of 150 psi (1,034,214 Pa) and Pipe Length of 100 ft (30.5 m).

| at 150 psi, 100 ft | | | Pipe Type | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Schedule 40 Steel | | Type K Copper | | Type L Copper | | Schedule 10 Stainless Steel | |
| Pipe Size | Flow Rate (cfm) | Flow Rate (m3/min) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) | Pressure Drop (psi) | Pressure Drop (Pa) |
| 1/2" | 7 - 106 | 0.2 - 3 | 0.08 - 12.14 | 552 - 83,702 | 0.19 - 27.8 | 1,310 - 191,674 | 0.16 - 23.51 | 1,103 - 162,096 | 0.08 - 12.14 | 552 - 83,702 |
| 3/4" | 14 - 247 | 0.4 - 7 | 0.07 - 14.26 | 483 - 98,319 | 0.12 - 23.61 | 827 - 162,785 | 0.09 - 18.18 | 621 - 125,347 | 0.09 - 17.03 | 621 - 117,418 |
| 1" | 28 - 353 | 0.8 - 10 | 0.08 - 8.25 | 552 - 56,881 | 0.08 - 10.75 | 552 - 74,119 | 0.09 - 9.27 | 621 - 63,914 | 0.08 - 8.98 | 552 - 61,915 |
| 1 1/4" | 64 - 1,059 | 1.8 - 30 | 0.09 - 15.99 | 621 - 110,247 | 0.09 - 26.75 | 621 - 184,435 | 0.09 - 24.7 | 621 - 170,301 | NA | NA |
| 1 1/2" | 92 - 1,412 | 2.6 - 40 | 0.08 - 12.59 | 552 - 86,805 | 0.07 - 19.12 | 483 - 131,828 | 0.08 - 17.64 | 552 - 121,624 | 0.08 - 12.43 | 552 - 85,702 |
| 2" | 177 - 3,001 | 5 - 85 | 0.08 - 14.56 | 552 - 100,388 | 0.08 - 19.04 | 552 - 131,276 | 0.08 - 17.83 | 552 - 122,934 | 0.08 - 15.88 | 552 - 109,489 |
| 2 1/2" | 282 - 3,530 | 8 - 100 | 0.08 - 8.09 | 552 - 55,779 | 0.07 - 8.67 | 483 - 59,778 | 0.08 - 8.15 | 552 - 56,192 | NA | NA |
| 3" | 530 - 8,825 | 10 - 250 | 0.08 - 14.87 | 552 - 102,525 | 0.11 - 19.47 | 758 - 134,241 | 0.1 - 18.25 | 689 - 125,829 | 0.11 - 19.26 | 758 - 132,793 |
| 4" | 1,059 - 8,8825 | 30 - 250 | 0.08 - 13.78 | 552 - 95,010 | 0.09 - 17.07 | 621 - 117,694 | 0.09 - 16.05 | 621 - 110,661 | 0.09 - 9.57 | 621 - 65,983 |

done by the system. Using the modeling Equations 2.1-2.5, and assumption listed in Table 3.5, Table 3.6 was generated for ideal model using $k = 1.4$. It is the ideal model in which no losses were taken into account. Every variable is calculated at different states (refer to Figure 2.1). The set pressure for this model was 105 psi (729,950 Pa).

Table 3.5.
Assumptions for Calculation of Ideal and Real Models.

| Assumptions | Volumetric Flow Rate ($m^3$/s) | Temperature (°K) | Pressure (Pa) | Enthalpy (kJ/kg) |
|---|---|---|---|---|
| | $Q$ | T | P | h |
| a: atmospheric | 0.413 | 293 | 101,325 | |
| 1: Compressor Output | | 513.9[a] | 723,950 | |
| 1': Intercooler Output | | 328[b] | | |
| 2': Aftercooler Output | | 301.3[c] | | |
| 4: Dryer Input | | | | 412.46 |
| 4': Dryer Output | | 293 | | 413.46 |

$^a : T1 = 513.9K$ using $T = T_a(\frac{P}{P_a})^{(\frac{k-1}{k}}$ derived from adiabatic equation

$^b : T1' = 328K$ For the isotonic process, the air is typically cooled by $50 \sim 60°C$ from the atmospheric temperature ($0°C$). For this application, $55°C$ is assumed (55+273 = 328 K) [27]

$^c : T2' = 301.3K$ [28]

Table 3.6.
Ideal Model Results at a Set Pressure 105 psi (723,950 Pa).

| State | Work Rate (J/s) | Air Power (J/s) | Volumetric Flow Rate ($m^3$/s) | Temperature (°K) | Pressure (Pa) | Heat Loss (J/s) |
|---|---|---|---|---|---|---|
| | $\dot{W}$ | $\dot{E}$ | $Q$ | T | P | H |
| | 112,254 | | | | | |
| a: atmospheric | | | 0.4130 | 293 | 101,325 | |
| 1: Compressor Output | | 112,254 | 0.1014 | 513.9 | 723,950 | |
| 1': Intercooler Output | | 84,642 | 0.0647 | 328.0 | 723,950 | 27,612 |
| 2: Aftercooler Input | | 84,642 | 0.0647 | 328.0 | 723,950 | |
| 2': Aftercooler Output | | 83,716 | 0.0594 | 301.3 | 723,950 | 926 |
| 3: Filter Input | | 83,716 | 0.0594 | 301.3 | 723,950 | |
| 3': Filter Output | | 83,716 | 0.0594 | 301.3 | 723,950 | |
| 4: Dryer Input | 505.9 | 83,716 | 0.0594 | 301.3 | 723,950 | |
| 4': Dryer Output | | 83,658 | 0.0578 | 293.0 | 723,950 | 58.6 |

In this model, the atmospheric temperature was calculated using experimental data, while flow rate used compressor data sheet. For the other states, the rest values were calculated using thermodynamic equations.

To calculate the Real case, similar approach was made including the pressure drop models for after-cooler, filter, dryer along with the pressure drop due to pipe friction loss and $k = 1.383$ [28] as shown in Table 3.7. The total pressure drop is the sum of all the pressure drops in the system, which is equal to 5.3 psi (1.6+0.7+2.4+0.6) (36542 Pa). The pressure drop in the pipe was considered to be 0.6 psi which is for 500 ft pipe (152.4 m) with diameter of 2″. Therefore, is equal to 120,891 J/s when considering the total pressure drop and motor efficiency. The rate of work, $\dot{W}_2$ (dryer) is equal to 532.6 J/s by using the enthalpy values and the mass flow rate of 0.0578 $m^3/s$, and a motor efficiency of 95%.

In table 3.8 represent the ideal and real models with efficiency at each stage. The Real part has lower air power than the Ideal, which means there was less energy stored in the air. This is logical since the Real part considers pressure drops, motor efficiency, and lower air index. This is true for all components except between stages 1′ and 2, where air exits the first inter-cooler. The reason for higher air power in the Real part for these two stages (1′ and 2) is lower air index (k) values which are 1.383 and 1.4 respectively. Furthermore, real part efficiency is 95.9% due to considering motor efficiency and a lower air index (k). When considering losses due to pressure drop, the compressors work efficiency drops down to 92.9%.

Table 3.7.
Real Model Results at a Set Pressure 105 psi (723,950 Pa).

| State | Work Rate (J/s) | Air Power (J/s) | Volumetric Flow Rate ($m^3$/s) | Temperature (°K) | Pressure (Pa) | Heat Loss (J/s) | Pressure Loss (psi) |
|---|---|---|---|---|---|---|---|
| | $\dot{W}$ | $\dot{E}$ | $Q$ | T | P | H | ΔP |
| | 117,053 | | | | | | |
| **with losses** | 120,891 | | | | | | |
| | | | 0.4130 | 293 | 101,325 | | |
| **a: atmospheric** | | 111,201 | 0.0996 | 505.1 | 723,950 | | |
| **1: Compressor Output** | | 84,673 | 0.0647 | 328.0 | 723,950 | 26,527 | |
| **1': Intercooler Output** | | 84,673 | 0.0647 | 328.0 | 723,950 | | 0 |
| **2: Aftercooler Input** | | 83,066 | 0.0594 | 301.3 | 712,939 | 1,607 | 1.6 |
| **2': Aftercooler Output** | | 83,066 | 0.0594 | 301.3 | 712,939 | | 0 |
| **3: Filter Input** | | 82,765 | 0.0594 | 301.3 | 707,906 | 301.4 | 0.7 |
| **3': Filter Output** | | 82,765 | 0.0594 | 301.3 | 707,906 | | 0 |
| **4: Dryer Input** | 532.6 | 81,704 | 0.0578 | 293 | 691,452 | 1,061 | 2.4 |
| **4': Dryer Output** | | 81,704 | 0.0578 | 293 | 691,452 | | 0 |

The total theoretical efficiency of the system is equal to:

$\eta elec$=95.9%

$\eta_{mech} = \eta 1 \times \eta 1' \times \eta 2 \times \eta 2' \times \eta 3 \times \eta 3' \times \eta 4 \times \eta 4' = 93.2\%$

$\eta_{total} = \eta_{elec} \times \eta_{mech} = 89.3\%$

Figure 3.3 shows a block diagram of the compressing air sub-systems and the estimated energy consumption of each component. The ambient air enters the compressor where the most of energy is being consumed. Then, it cools down using an

Table 3.8.
Ideal vs Real Models and Each Stage Efficiency.

| Ideal | | | Real | | | |
|---|---|---|---|---|---|---|
| State | Work Rate (J/s) | Air Power (J/s) | State | Work Rate (J/s) | Air Power (J/s) | Efficiency (%) |
| | $\dot{W}$ | $\dot{E}$ | | $\dot{W}$ | $\dot{E}$ | η |
| | 112,254 | | | 117,053 | | 95.9% |
| | | | with losses | 120,891 | | 92.9% |
| 1: Compressor Output | | 112,254 | 1: Compressor Output | | 111,201 | 99.1% |
| 1': Intercooler Output | | 84,642 | 1': Intercooler Output | | 84,673 | 100% |
| 2: Aftercooler Input | | 84,642 | 2: Aftercooler Input | | 84,673 | 100% |
| 2': Aftercooler Output | | 83,716 | 2': Aftercooler Output | | 83,066 | 99.2% |
| 3: Filter Input | | 83,716 | 3: Filter Input | | 83,066 | 99.2% |
| 3': Filter Output | | 83,716 | 3': Filter Output | | 82,765 | 98.9% |
| 4: Dryer Input | 505.9 | 83,716 | 4: Dryer Input | 532.6 | 82,765 | 98.9% |
| 4': Dryer Output | | 83,658 | 4': Dryer Output | | 81,704 | 97.7% |

inter-cooler. As we can see, a big portion of energy is being transferred as heat loss due to temperature drop. Air continues to flow through a second cooling medium (after-cooler). Then, after being filtered, it flows through a refrigerant dryer to ensure a specific dew point. The final stage is the receiver tank, where it gets stored for future use. Figure 3.3 states the estimated energy consumption at each component. As shown, majority of the energy consumption is happening at the compressor and intercooler. At the compressor, the pressure and temperature are increasing drastically, thus high energy consumption is required. In the after cooler, there is a temperature drop of $50°C$ to $60°C$. The rest of the components affect the energy stored in the air minimally, and thus might not have a big room of potential improvement.

## 3.3 Effectiveness

Effectiveness is another parameter used for evaluation of air compressor sub system energy consumption. Each energy saving recommendation has an effectiveness on the compressed air system, not just the air compressor, but each individual component of the entire air compressor system. This will be explored further in the following

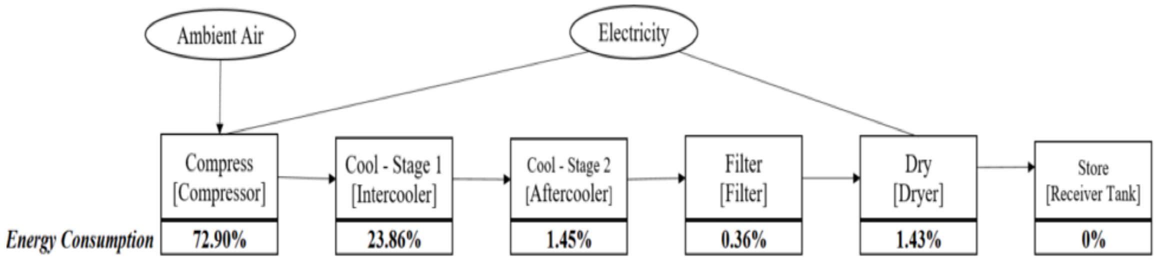


Fig. 3.3. Compressed Air Sub-System Block Diagram and Estimated Energy Consumption Percentage.

section, by looking at various energy saving recommendations including: reducing set pressure and using outside air temperature for compressor intake.

### 3.3.1 Reduce Compressor Set Pressure

Having a set pressure higher than the required demand pressure is very common in facilities. This action usually happens to avoid any issues with not having enough pressure for the machines, or simply because the plant is not aware of its actual requirements. Thus, reducing the air compressors set pressure based on the required demand is a typical energy saving recommendation. The reduction of the compressors operating pressure reduces its load and operating horsepower, otherwise known as brake horse power. The cost penalty for operating the system at a high pressure was found using fractional savings.

The fractional savings for operating the compressor at a reduced upper activation pressure, $P_{l2}$, compared to a high upper activation pressure, $P_{h2}$, when the inlet air pressure is $P_1$, can be calculated using Equation 3.8 [29]:

$$\text{Fractional Savings} = \frac{(\frac{P_{h2}}{P_1})^{0.286} - (\frac{P_{l2}}{P_1})^{0.286}}{(\frac{P_{h2}}{P_1})^{0.286} - 1} \qquad (3.8)$$

In this case, we will be consider a set pressure of 105 psi, and a proposed pressure of 90 psi. Fractional savings after reducing the air compressor discharge pressure from 105 psi to 90 psi would be:

$$\text{Fractional Savings} = \frac{(\frac{105+14.7}{14.7})^{0.286} - (\frac{90+14.7}{14.7})^{0.286}}{(\frac{105+14.7}{14.7})^{0.286} - 1} = 8.33\%$$

So, there is a saving of 8.33% in current energy consumption by lowering the set pressure to desired value. However, this only considers work done by the air compressor. In fact, the effectiveness will be higher than these savings if we consider the Real model created.

Table 3.9 is similar to table 8 which is generated for real model at pressure of 90 psi (620,528Pa). Work proposed is equal to 109,689 J/s when considering the pressure drop in components. If the flow rate is increased, the pressure drop would also increase in the after-cooler and dryer since the pressure drop models for these two components is the function of flow rate. Therefore, effectiveness is equal to:

$$\varepsilon = 1 - \frac{\dot{E}_{Proposed}}{\dot{E}_{Real}} = 1 - \frac{\dot{W}_{Proposed}}{\dot{W}_{Real}} = 1 - \frac{109,689}{120,891} = 9.3\%$$

This shows that the effectiveness of this recommendation is higher on the system than just calculating the energy savings when looking at each individual component. This was tested for multiple set pressures drop as shown in Figure 3.4.

Based on Figure 3.4, taking this specific case, Figure 3.5 shows the effectiveness of reducing the set pressure and impact on annual energy consumption by the air compressor system.

### 3.3.2 Using Outside Air for Air Compressor Intake

Outside air is cooler and denser than indoor air during cooling degree days (CDD). By utilizing the outside air as the air compressor intake, the energy requirement can be reduced. Thus, the work required to reach a certain operating pressure decreases, i.e. less energy to produce the same operating pressure. Due to the excess heat

Table 3.9.
Model Results for a Set Pressure 90 psi (620,528Pa)

| State | Work Rate (J/s) | Air Power (J/s) | Volumetric Flow Rate ($m^3/s$) | Temperature (K) | Pressure (Pa) | Heat Loss (J/s) | Pressure Loss (psi) |
|---|---|---|---|---|---|---|---|
| | $\dot{W}$ | $\dot{E}$ | $Q$ | T | P | H | ΔP |
| | 105,404 | | | | | | |
| with losses | 109,689 | | | | | | |
| a: atmospheric | | | 0.4130 | 293 | 101,325 | | |
| 1: Compressor Output | | 100,133 | 0.1114 | 484.0 | 620,528 | | |
| 1': Intercooler Output | | 78,115 | 0.0755 | 328.0 | 620,528 | 22,018 | |
| 2: Aftercooler Input | | 78,115 | 0.0755 | 328.0 | 620,528 | | 0 |
| 2': Aftercooler Output | | 76,397 | 0.0693 | 301.3 | 609,505 | 1,718 | 1.6 |
| 3: Filter Input | | 76,397 | 0.0693 | 301.3 | 609,505 | | 0 |
| 3': Filter Output | | 76,045 | 0.0693 | 301.3 | 604,472 | 352.8 | 0.73 |
| 4: Dryer Input | 532.6 | 76,045 | 0.0693 | 301.3 | 604,472 | | 0 |
| 4': Dryer Output | | 74,806 | 0.0674 | 293.0 | 587,960 | 1,238.8 | 2.4 |
| 5: Receiver Input | | 74,806 | 0.0674 | 293 | 587,960 | | 0 |

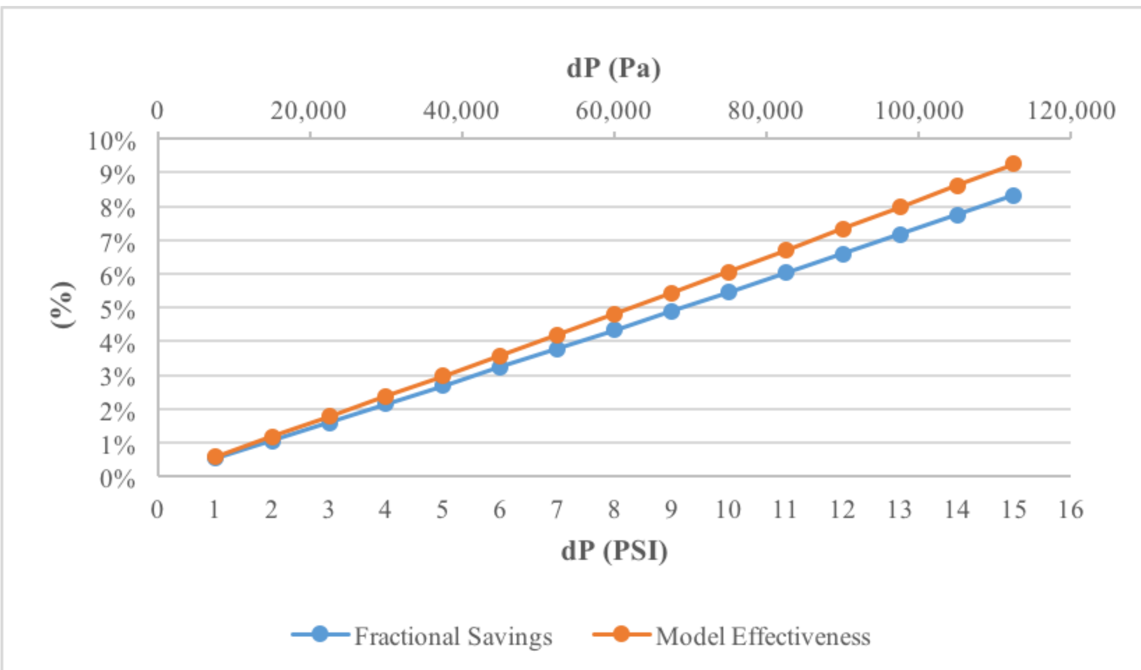


Fig. 3.4. Reducing Set Pressure Effectiveness.

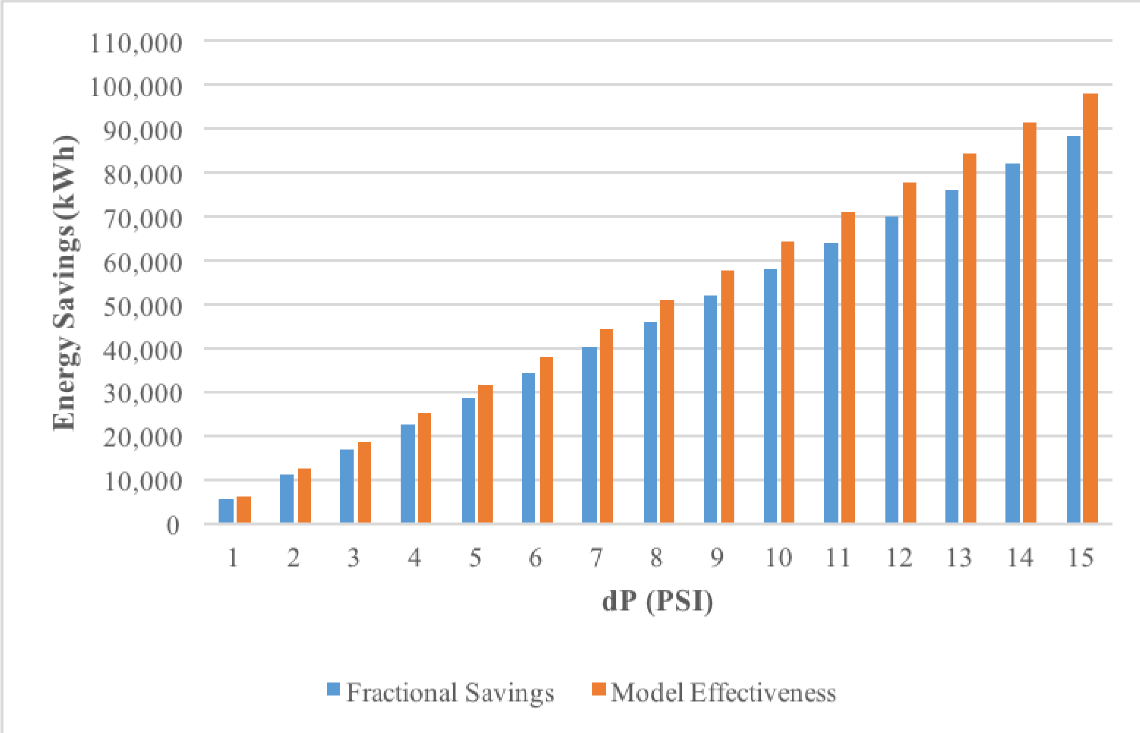


Fig. 3.5. Energy Savings (kWh) vs Pressure Drop (PSI).

produced from within the plant, it would be beneficial to use outside during CDD. The compressor work for the usual operating conditions in manufacturing plants is proportional to the absolute temperature of the intake air. Thus, the fractional reduction in compressor work, WR, resulting from lowering the intake air temperature is estimated using equations (14-15) [1]:

$$WR = WI - \frac{WO}{WI} \tag{3.9}$$

$$WR = \frac{TI - TO}{TI} \tag{3.10}$$

The range of air intake temperature was tested for both work reduction, and compared to the model which the results were close since this action mainly affects the work done by the air compressor.

Table 3.10.
Results for Different Air Intake Temperatures with a Base Temperature of 293 K

| Intake Temp. (°K) | Intake Temp. (°R) | Discharge Temp. (°K) | Work Rate (J/s) | Air Power - $\dot{E}$ (J/s) | Model Effectiveness (%) | Work Reduction (%) | Difference (%) |
|---|---|---|---|---|---|---|---|
| 293 | 527.4 | 505.1 | 120,891 | 81,704 | | | |
| 292 | 525.6 | 503.4 | 120,478 | 81,426 | 0.34% | 0.34% | 0.00% |
| 291 | 523.8 | 501.6 | 120,066 | 81,149 | 0.68% | 0.68% | 0.00% |
| 290 | 522 | 499.9 | 119,653 | 80,875 | 1.01% | 1.02% | 0.01% |
| 289 | 520.2 | 498.2 | 119,241 | 80,602 | 1.35% | 1.36% | 0.02% |
| 288 | 518.4 | 496.5 | 118,828 | 80,331 | 1.68% | 1.71% | 0.03% |
| 287 | 516.6 | 494.7 | 118,416 | 80,062 | 2.01% | 2.05% | 0.04% |
| 286 | 514.8 | 493.0 | 118,003 | 79,795 | 2.34% | 2.39% | 0.05% |
| 285 | 513 | 491.3 | 117,591 | 79,530 | 2.66% | 2.73% | 0.07% |
| 284 | 511.2 | 489.6 | 117,179 | 79,266 | 2.98% | 3.07% | 0.09% |
| 283 | 509.4 | 487.8 | 116,766 | 79,005 | 3.30% | 3.41% | 0.11% |
| 282 | 507.6 | 486.1 | 116,354 | 78,745 | 3.62% | 3.75% | 0.13% |
| 281 | 505.8 | 484.4 | 115,941 | 78,487 | 3.94% | 4.09% | 0.16% |
| 280 | 504 | 482.7 | 115,529 | 78,231 | 4.25% | 4.43% | 0.18% |

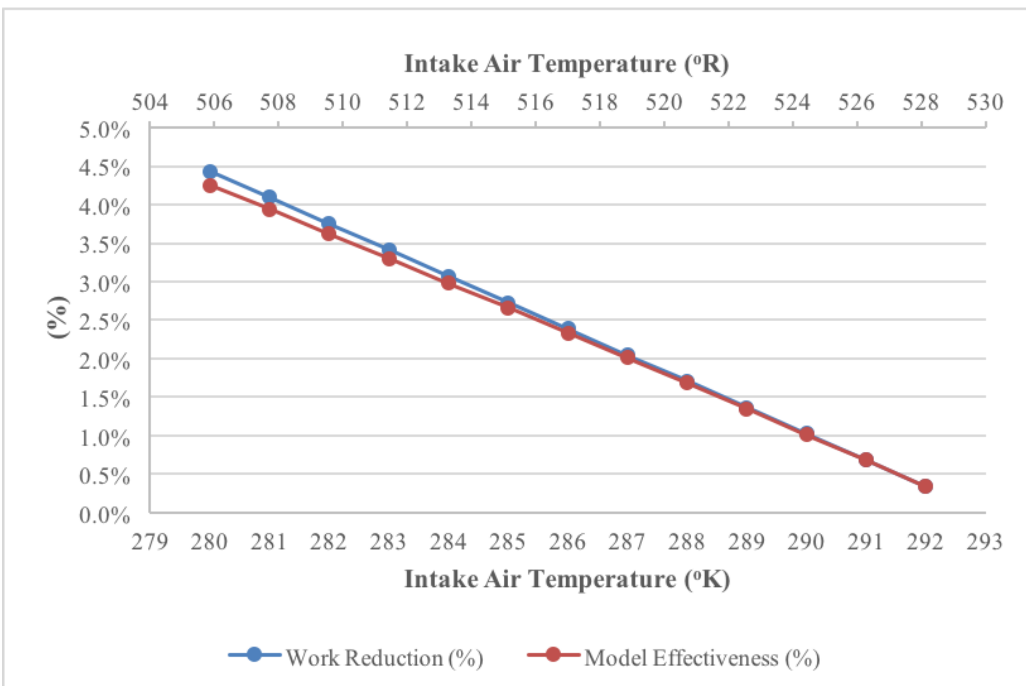


Fig. 3.6. Outside Air Compressor Air Intake Effectiveness.

This will determine the air discharge temperature from the air compressor. However, the slight difference in the model is due to the heat loss in each component. The effectiveness was calculated using air power, $\dot{E}$ in the final stage at the receiver tank. Table 3.10 and Figure 3.6 show the results of each proposed intake air temperature, and the difference between the work reduction and model effectiveness. Thus, higher the temperature difference would result to higher effectiveness due to heat losses among components.

# 4. RESULTS VALIDATION

To validate the model, the Ideal case was compared to the Real case for three different case studies. The goal was to collect experimental data that will to validate the model and determine model accuracy. The temperature and pressure parameters were addressed. The expectation was to have model parameters match these measurements. Data was collected using current, temperature and pressure sensors. Current (amps) data was used to calculate the Demand Draw, while temperature and pressure values were used as inputs in the model. The actual air compressors work was compared to the Real case model. This identified the experimental efficiency of the system. Theoretical and experimental efficiencies were calculated to find the uncertainty. The uncertainty indicated model accuracy.

## 4.1 First Case Study

The first case study was performed at a die casting company. The current and pressure were measured for a 200 HP air compressor. Current sensors were connected to the air compressor and air dryer to calculate Demand Draw. Pressure sensors were connected at the receiver tank and down the line to collect pressure data. Figure 4.1 shows the compressed air system layout at a die casting company. It also shows exactly where the different sensors are connected. The temperature and relative humidity sensor (T, RH) was placed next to the compressors air intake. Current transducers (CT) were connected to the air compressor and refrigerant dryer. Pressure transducers (PT) were connected at the line before the air entered the first receiver tank, and down the line next to a second receiver tank.

Table 4.1 states the data sheet information of the air compressor. This was obtained using CAGI data sheets. The data is included to give a better understanding

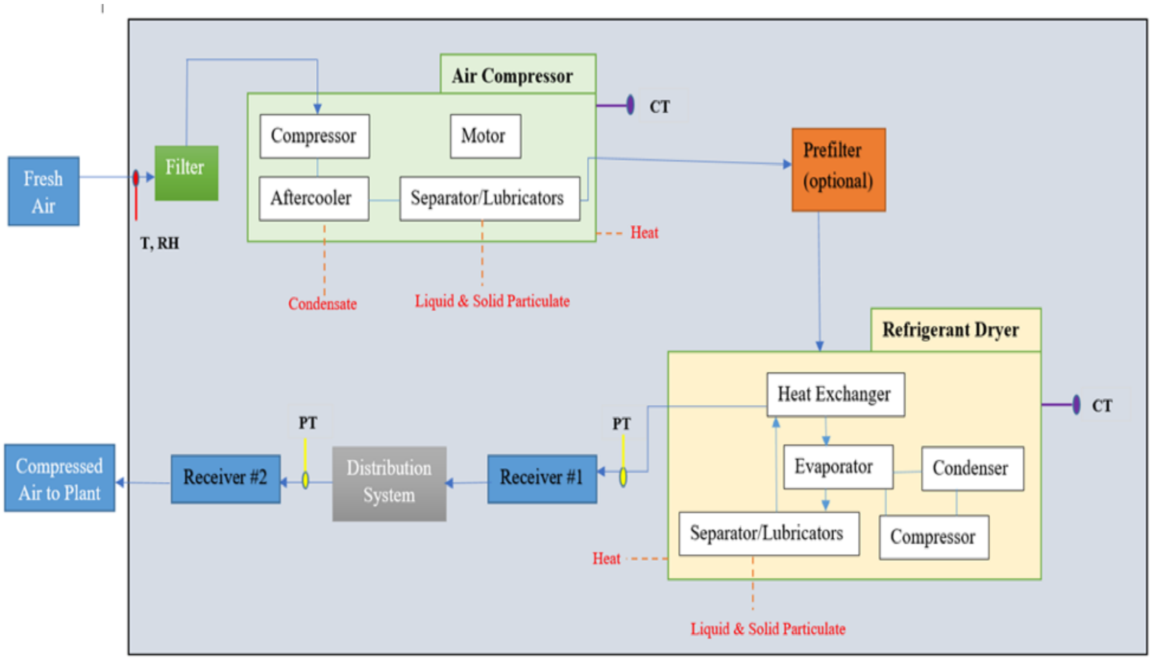


Fig. 4.1. Compressed Air System Layout.

on the compressor ratings. Figure 4.2 is the actual nameplate of the compressor, to prove the values in Table 4.1. Finally, Figure 4.3 is the nameplate associated with the air dryer.

Table 4.1.
Air Compressors Data Sheet Information.

| Parameter | Value | Unit |
|---|---|---|
| Rated HP | 200 | HP |
| Rated kW | 149.2 | kW |
| Full Load Amps | 267 | A |
| Voltage | 460 | V |
| Motor % | 95% | |

The sensors were connected on December 8th, 2016. The data was extracted on Feb 3rd and July 20th of 2017. Results are described below.


Fig. 4.2. Air Compressors Name Plate Information.

**Results:**

Average Compressors Current: 234.2 A

Average Dryers Current: 3.6 A

Average Temperature: 68°F

Average Relative Humidity: 32%

First Average Pressure: 105 PSI

Second Average Pressure: 97 PSI

The set pressure is 105 psi (723,950 Pa) and the average current draw was 234 amps. The demand draw of the air compressor can be calculated using the Equation 4.1:

$$\text{Demand Draw} = \frac{I \times V \times \sqrt{3} \times PF}{1000} = 158.6kW \tag{4.1}$$

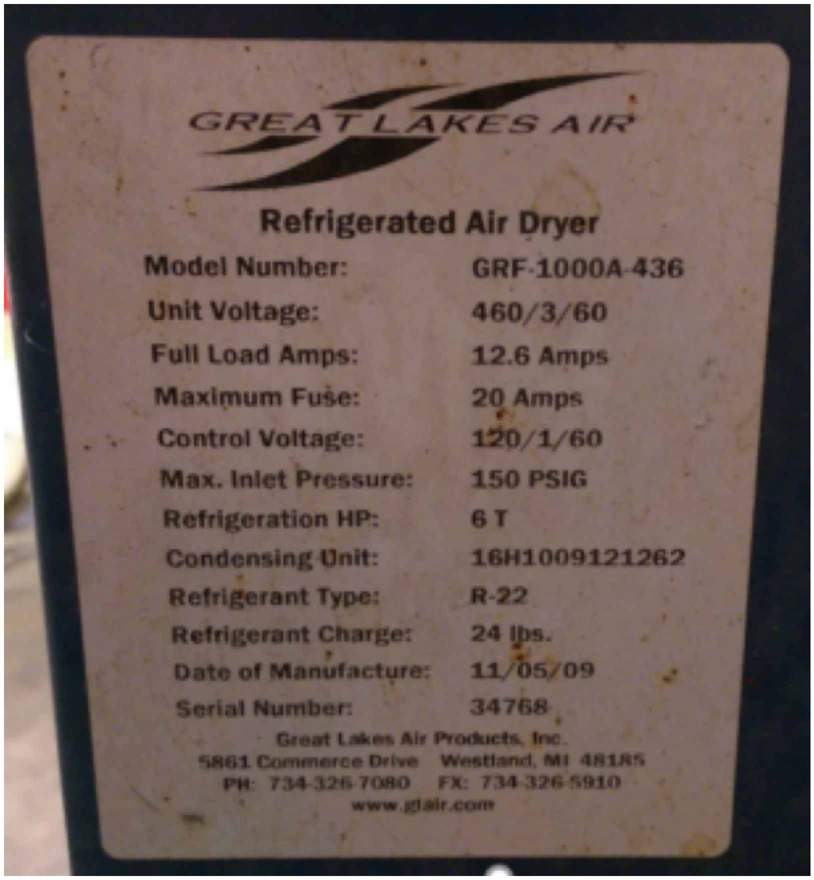


Fig. 4.3. Air Dryers Name Plate Information.

By comparing the collected data to the Real model results ($W_{real} = 120.9kW$), we get:

$$\text{Efficiency of the System} = \frac{120.9kW}{158.6kW} = 76.2\% \tag{4.2}$$

Using values from Table 3.8, the total theoretical efficiency of the system is equal to: $\eta elec$=95.9%

$\eta_{mech} = \eta 1 \times \eta 1' \times \eta 2 \times \eta 2' \times \eta 3 \times \eta 3' \times \eta 4 \times \eta 4' = 93.2\%$

$\eta_{total} = \eta_{elec} \times \eta_{mech} = 89.3\%$

When the theoretical efficiency (89.3%) and experimental efficiency (76.2%) are compared, the model accuracy is equal to the ratio of experimental efficiency and theoretical efficiency which is 85.3%. This means the model has uncertainty $\sim$ 15%.

## 4.2 Second Case Study

The second case study is a metal parts manufacturer. The current and pressure were measured for a 100 HP air compressor. The set pressure was 115 psi. The average current draw was 90.3 amps. The demand draw of the air compressor is 74.6 kW. By comparing the collected data to the Real model results ($W_{real}$ = 66.7 kW), the efficiency of the system was calculated.

$$\text{Efficiency of the System} = \frac{66.7kW}{74.6kW} = 89.4\% \tag{4.3}$$

When the theoretical efficiency (90.5%) and experimental efficiency (89.4%) are compared, the model accuracy is 98.7%. This means the model has uncertainty $\sim$ 1%.

## 4.3 Third Case Study

The third case study is a metal parts manufacturer. The current and pressure were measured for a 75 HP air compressor. The set pressure is 115 psi. The average current draw was 75.3 amps. The demand draw of the air compressor is 53.2 kW. By comparing the collected data to the Real model results ($Wreal$ = 49.5 kW), the efficiency of the system was calculated.

$$\text{Efficiency of the System} = \frac{49.5.7kW}{52.3kW} = 94.6\% \tag{4.4}$$

When the theoretical efficiency (90.5%) and experimental efficiency (94.6%) are compared, the model accuracy is 95.6%. This means the model has uncertainty $\sim$ 4%.

Table 4.2 show the results summary of the three different case studies. Demand Draw was calculated using the average current (amps) value that was collected using current sensor. This sensor was connected directly to the air compressor. The voltage and power factor values were set at 460V and 0.85 respectively. The model uncertainty ranged from $\sim$ 1% to $\sim$ 15%. Multiple reasons could have caused this wide range

Table 4.2.
Case Studies Summary.

| | Case 1 | Case 2 | Case 3 |
|---|---|---|---|
| **Set Pressure (PSI)** | 105 | 115 | 115 |
| **Capacity (HP)** | 200 | 100 | 75 |
| **Demand Draw (kW)** | 158.6 | 74.6 | 53.2 |
| **$W_{real}$ (kW)** | 120.9 | 66.7 | 49.5 |
| **Experimental Efficiency (%)** | 76.2 | 89.4 | 94.6 |
| **Theoretical Efficiency (%)** | 89.3 | 90.5 | 90.5 |
| **Uncertainty (%)** | ~15 | ~1 | ~4 |

such as compressor capacity, facility type, and the compressor load while the sensor data was collected.

# 5. DISCUSSION

The compressed air system is such a unique and complicated system. So many factors affect the quality and outcome of this system. It is difficult to track down what is causing system losses. The energy model developed in this research provides the tool to dissect each component and study the parameters that have the highest impact on the system. These process parameters include Air Power, Pressure, Temperature and Flow Rate. The model was created based on rotary screw compressor and refrigerant air dryer. These are common among small/mid-sized industrial sector. Process parameters (Air Power, Flow Rate, Temperature, and Pressure) are common across other types of compressors and dryers. However, model might have a higher uncertainty of other compressor and dryer types are used. By applying the Ideal and Real cases, efficiency was calculated for individual components as well as the overall system. Theoretical and experimental efficiency were looked into to identify model accuracy. Part of the energy model was based on the pressure loss models for different components including aftercooler, filter, dryer and pipe losses. The flow rate was the indicating variable to find out what are the system pressure losses in the components. There were no pipe pressure loss between the components due to the short pipe length. It will only be effective to consider pipe loss when looking into pipe length greater than 500 ft. Three different case studies were investigated to validate the models. The first case study was the main focus in this project. Some of the challenges regarding this case were:

1. Three air compressors and two air dryers connected in one system, and they alternated based on the air demand. This caused some fluctuation in the data. The focus was on just one air compressors and one air dryer to avoid complications.

2. It was impossible to implement pressure and temperature sensors after each component (aftercooler, filter, and air dryer), so this caused some inaccuracy in the model especially when calculating the output pressures and temperatures. Assumptions had to be made based on literature and alternative case studies. Assumptions included input atmospheric flow rate and change in temperature within the intercooler and aftercooler.

3. Some changes in the plant happened during the data collection phase.

4. The facility did not have any information regarding the system except an audit report that was done 5 years ago.

These challenges slowed down the data collection process and caused some mathematical errors. To improve results and model accuracy, experimental data need to be collected after each component by addressing main process parameters such as pressure, temperature and flow rate. Other parameter that could be explored would be humidity. Humidity plays role in air quality, the more humid the air flowing through the system is, and the higher the pressure drop will be occurring.

Results for the three case studies proved model uncertainty varying from 1% to 15%. This gave a close illustration of the system performance. However, more experimental data at specific system terminal would result and in a more accurate model.

Finally, effectiveness method created was applied on two major energy conservation measures. The first being reducing system set pressure. This proved that fractional savings methodology used by the Department of Energy to quantify such savings only account for compressor work savings. In fact, this measure impact the rest of the system and results in higher energy savings due pressure and heat loss. In this method, it was assumed that there was no system leakage. The second energy saving measure investigated was using outside air for compressor intake. The work reduction methodology used by the U.S. Department of Energy to quantify savings proved to be overestimating the savings. Effectiveness method provides more accuracy on quantifying energy savings, and thus more realistic. In future, the goal is to see if we can predict more accurate savings for different energy saving measures. This will help the industry build the confidence in investing in such energy saving techniques.

To sum up, it has been great working on this research since I was able to get a better understanding of the air compressor system as whole. Being able to investigate individual components and deciding what parameters are affecting the energy consumption. Inputs and outputs of each component illustrates a good picture of the change process inside the component. Looking at efficiency values helped realize what components should be focused on for potential room of improvement. Having the ideal case and experimental data was the best way to validate the model. Measurements reflect real scenarios. Real applications can widely vary from ideal cases. Real applications are case-specific. So models created can perform better from case to another. The most important thing is to truly understand the concept and basis of what this model was built in. This topic proved that there will always be future work that can be done to enhance the research. Identifying and realizing these opportunities is one step closer to an ideal application.

# 6. CONCLUSION

Compressed air systems are major energy consumers in industrial plants. It is considered to be a complicated system due to many outside factors that have a big effect on its efficiency. It is important to know how much energy this system is estimated to consume and how much it is actually being consumed. The model discussed in this thesis allows the user to calculate the Ideal energy consumption of the system, and compare it to the Real case taking into account pressure drops and heat loss within individual components. Having the air power change data due to different inputs and output within components helps in pointing out the potential problems and optimizing the overall system efficiency. The model results proved to be 85% - 97.8% accurate when tested for three case studies. Real system applications are usually less efficient due to factors such as equipment age, dirt, and heat loss. The best way to validate this model is to have sensors that would monitor all the parameters entering and exiting each component. The efficiency of each component is based on the input and output air power. In this study, the efficiency of the compressor, intercooler, aftercooler, filter, and air dryer were 95.9%, 100%, 99.2%, 98.9%, and 97.7% respectively. Potential improvement can be made on the compressor and dryer components. This is reasonable since these two components apply motor work.

The effectiveness of two energy saving recommendations was calculated to show the impact on the entire system not just the air compressor. Effectiveness showed that more energy could be saved than what is usually calculated in Reducing Set Pressure recommendation. Also, effectiveness showed that the Work Reduction method to calculate percent savings in Using Outside Air Intake recommendation is over estimating work reduction percentage. This means that there is actually less energy savings than what was calculated in the Word Reduction method.

## 6.1 Future Work

For the future, pressure and temperature sensors should be implemented after each component such as aftercooler and air dryer, to get more readings throughout the system that would result in a higher accuracy model.

Second, expanding project boundaries by including distribution line and individual end-users. This will lead into exploring pressure losses at the pipe level, fittings, and air leaks. Also, different compressor and air dryer types should be explored.

Finally, the effectiveness of other recommendations could be looked at such as:

1. Installing local air receivers for end-users with higher and frequent demands. Air volume could decrease and pressure could drop especially in the end-user is located towards the end of the pressure line. This will require the compressor to work more to make up for these losses and thus have higher energy consumption. Installing local air receiver will help eliminate this problem.

2. Providing additional compressed air receiving tank capacity to reduce frequent compressor cycling.

3. Providing wet compressed air receiving tank before the dry tank. This will help the improving air quality, removing moisture and lubricants, and thus reducing the work on air dryer. This will also help reducing pressure drop at different system components.

4. Upgrading to premium efficiency electric motors. Increasing motor efficiency should help reducing the compressor work.

5. Installing a dehumidifier at compressor intake. This will help improving air quality before entering the air compressor, and thus require less work by the compressor to reach the desired air pressure.

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