ABSTRACT

HUNT, DAVID GEORGE. An Experimental and Quantitative Analysis to Determine the Effect of Changing Inlet Air Temperatures on the Performance of an Oil injected Electrically Powered Twin Screw Air Compressor for Industrial Processes. (Under the direction of Dr. Stephen Terry.)

The objective of this thesis is to experimentally determine if compressing cooler outside air instead of warm plant air affects the performance of an oil flooded screw compressor. It is common knowledge that compressing cooler air requires less power than compressing warmer air, but the effects of compressing cool air in an oil injected compressor are not known. It is hypothesized that the effect of inlet air temperature on an oil injected screw compressor's performance will be small since the oil is maintained at specific temperatures.

To determine if this hypothesis is true an oil flooded screw compressor used for industrial processes was data logged. This compressor is located in western North Carolina in a manufacture's parking lot and effectively draws in outside air. Originally a year of data was desired but the compressor failed prematurely due to dirty oil cooler. Therefore 14 weeks of data were recorded. Fortunately, enough data existed to see temperature variances from 60ºF to 110ºF. After analyzing all the data two conclusions were drawn. One, using the isentropic compression process to predict energy savings due to lower inlet air temperatures in an oil flooded screw compressor is wrong. Two, inlet air temperatures do not have a significant effect on the oil injected screw compressor's performance analyzed in this experiment.

An Experimental and Quantitative Analysis to Determine the Effect of Changing Inlet Air Temperatures on the Performance of an Oil injected Electrically Powered Twin Screw Air Compressor for Industrial Processes

by
David George Hunt

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APPROVED BY:

Dr. Herbert M Eckerlin

Dr. James W Leach

Dr. Stephen D Terry
Committee Chair

DEDICATION

In particular I would like to dedicate this work to my parents, George and Caroline Hunt, for supporting me always. I would like thank and dedicate my work to Dr. Stephen Terry also. Dr. Terry never gave up on me and encouraged me to attend graduate school at NC State. Finally, I would like to dedicate my work to all my friends and family for the help and support they give me. I am very fortunate to have been blessed with a great group of family and friends.

BIOGRAPHY

David George Hunt was born March 19, 1987 in Oxford, NC. He was raised by two intelligent, inspiring, supportive, hardworking, and loving parents; George and Caroline Hunt. David was raised along with two intelligent and pretty sisters also, Carlyon and Martha Hunt. At the age of six David Hunt moved to Wilson, North Carolina. After high school David moved to Raleigh, North Carolina to study mechanical engineering at North Carolina State University (NCSU).

In college David Hunt quickly discovered earning a degree in mechanical engineering was going to be a rigorous task. Luckily, in college David found a Christian church group and a rugby team to help him with the frustrations of school. Through David's church group he met a true friend Michael Simon. Michael Simon found David a summer job at the Industrial Assessment Center (IAC). Here David actually enjoyed utilizing his course work to solve real world problems. After 5 years David graduated from NCSU with a bachelor's of science in mechanical engineering and completed NCSU's Co-op program.

After undergrad David decided to attend NCSU's graduate school to study mechanical engineering. At first David was very hesitant to attend graduate school, but after influence from Dr. Stephen Terry (IAC's principal auditor) and David Hughes (colleague) he decided to go for it. David did not enjoy his first year of graduate school. Fortunately, David has truly enjoyed his second year of graduate school and is looking forward to the future.

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Chapter 1 – Introduction

1.1 Motivation and Significance

The total amount of electrical energy consumed in the United States was 3,884 Billion kWh in 2010 (1). Experts, in the United States of America, estimate 26% of electricity generated is consumed by industry and 10% of the electricity utilized in industry is devoted to generating compressed air (2). Therefore 101 billion kWh of energy were devoted to generating compressed air in the year of 2010. To put this phenomenal amount of energy in prospective it would take all the nuclear power plants in North Carolina operating fully loaded nonstop for 2.32 years to generate this amount of energy. With substantial numbers like this it should be obvious generation of compressed air needs to be as efficient as possible.

What causes a compressed air system to consume so much energy? Why is compressed air needed? Why is compressed air so inefficient and how can it be improved? These are some of the questions that need to be answered. To answer these questions the history and why compressed air is used today is discussed followed by basic theory of compressing air and discussion of how to make a compressed air system more efficient. Topics such as air leaks, compressor waste heat, capacity control schemes, and inlet air temperatures are addressed. A

strong focus on inlet air temperatures is addressed to determine whether inlet air temperatures affect oil flooded screw compressors or not.

This study will consider if an oil flooded screw compressor consumes less energy using outside air rather than warmer plant air. Currently there is confusion on this topic because some energy engineers recommend using outside air in oil flooded screw compressors. If this recommendation is wrong then industrial facilities need to be educated so they will stop wasting money on useless energy savings projects and focus their time and money on proven applications or technologies.

1.2 History

Earth's atmosphere is composed of air. Air is very essential to life for humans because humans require oxygen to fuel their bodies. Air is around 20% oxygen, 78% nitrogen, and 2% miscellaneous elements. When most people hear compressed air they probably think car tires, bike tires, basketballs, air pumps, scuba diving tanks, air compressors, and pneumatic tools. All these are examples of modern marvels that utilize or produce compressed air. In actuality the human race has been utilizing compressors and compressed air longer than many think. Humans have utilized compressors air since Adam, the first human being in the Holy Bible, took his first breath. When humans breathe they expand their lungs first. Expansion of the lungs creates a pressure in them lower than the surrounding atmosphere. Air moves from the atmosphere into the lungs due to this pressure differential. While air

resides in human lungs a biological process occurs. Red blood cells release carbon dioxide while capturing oxygen from the air. Carbon dioxide fills the lungs and the fresh oxygen is sent to other body parts such as the heart and brain. The carbon dioxide must leave the lungs because it is a poisonous to the human body. Carbon dioxide is removed by contracting the lungs or decreasing the volume of space the lungs occupy. This compresses the carbon dioxide inside the lungs. Once the carbon dioxide's pressure, due to compression, is greater than the surrounding atmosphere it exits the lung due to a pressure differential. Understanding this explains why it is harder to breath at higher altitudes. For example atmospheric pressure is 14.696 pounds per square inch atmospheric (psia) at sea level while a top Mount Everest, a little under 30,000 feet, the atmospheric pressure is roughly 4.371 psia (3). There is a smaller pressure differential on top Mount Everest than sea level, because of a smaller pressure differential air does not enter the lungs easily on top Everest.

Many years ago humans discovered fire. To start fires humans use compressed air generated by their lungs. As bigger and hotter fires were needed, for things such as metal forging and smelting, more compressed air was needed. The earliest device to mechanically compress air for metal smelting was used around 3,000 BC, historians estimate (4). These devices were hand operated bellows similar to the ones found by a fireplace today. Metal smelting led to the first reciprocating (piston) compressor design also. The first piston operated compressors were found in eastern countries and utilized a bamboo cylinder, piston, and bird feathers (5). Bird feathers were used as a piston ring. They allowed air to pass on the piston's suction

stroke and sealed the cylinder on the compression stroke. More advanced piston devices were used by the Greeks around 150 BC (6). These piston cylinder devices were made of metal. As metallurgy advanced more compressed air was needed therefore larger more efficient bellows were needed. Eventually people discovered the energy potential of water (hydro power) and used it to operate larger bellows. Examples of large water powered bellows can be seen at historical blacksmith shops around the country (4).

Compressed air became such a reliable muscle for work that engineers started playing with the idea of using it as a viable energy source in the early 1800's (4). At this time steam was widely used to power industrial devices, but steam was a messy and dangerous energy source that was hard to transmit over long distances. Therefore compressed air seemed very attractive. In the mid 1800's, engineers were debating whether electricity or compressed air was a better energy source. At one time an 18,000 kW compressed air plant existed in Paris, France to power devices, simple as clocks, useful as elevators, and advanced as industrial processes. Paris was the first city to offer compressed air as a public utility similar to electricity today (4). Eventually electricity won the long debate, whether electricity or compressed air would be a good energy source for mass distribution. Ironically compressed air and electricity are still used today in most manufacturing plants; thus, no utility truly won the debate. Some other advances of compressed air during the 1800's were for underwater diving and mining (4). Compressed air allowed people to dive deeper for longer and allowed miners to mine faster and easier.

The mining industries drastically changed compressed air. Lightweight pneumatic drills and hammers allowed miners to move through rock faster than ever, which truly showed the reliability of compressed air to perform work. Mining not only advanced compressed air tools but also paved the path for the screw compressor used today. In 1854 the Roots blower was invented by the Roots brothers (6). The Roots blower was initially designed to remove water from mines by pumping it. Unfortunately the blower was not very successful at this task and was adjusted for air and gas compression. A patent of the Roots blower was made in 1860 (5) (6). The patented Roots blower boosted gas for furnaces and ventilated mines for miners. In 1878, Krigar expanded on the Roots blower design and designed an advanced blower that utilized helical lobes (6). Figure 1 below shows the original lobe design for a Roots blower and Figure 2 below shows the helical blower lobe design.

Figure 1: Originally Three Lobe Blower Design (5)

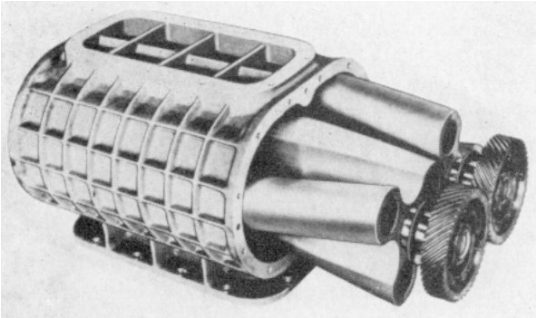

Figure 2: Helical Blower Lobes (7)

Originally compressed air was not easily available to everyone because it required a substantial power source usually in the form of steam which only large companies could afford. In the 1900's packaged compressed air units that utilized electricity were being produced because advancements in electrical power distributions and electrical motors were made (4). Packaged compressors were great for most businesses that desired a reliable robust energy source because they had access to electrical power now. As packaged compressed air units were produced more pneumatic tools were produced driving down the cost of pneumatic tools. Pneumatic tools were great because they are reliable and they helped manufactures increase production rates. During World War II anything that could increase production was embraced, thus, compressed air was embraced and proved its reliability once again (4).

One of the biggest advancement of compressed air in the 1900's was the twin screw compressor. A Swedish engineer, Alf Lysholm, invented the twin screw compressor for a

gas turbine in the 1930's. Lysholm needed a compressor that used a rotating action at high speeds that could not surge. Therefore Lysholm developed the twin screw compressor using the 1878 helical blower design by Krigar (6). Lysholm's twin screw compressor could not supply the volume of air his turbine needed at a practical size. Even though the twin screw compressor did not do what Lyshom wanted a Swedish research group Svenska Rotor Maskiner developed the idea of using the twin screw compressor for industrial applications (6). In 1946 companies started using the twin screw compressor. Industrial twin screw compressors utilized an oil free design until 1957 when the first oil injected machine hit the market (6). Oil free means zero particles of oil in the air stream that is compressed and oil flooded means oil particles are injected into the compressor's inlet, along with air, as air is compressed. 40 to 500 horsepower compressor are most common sizes today (4). A set of screw compressor screws is shown in Figure 3 below.

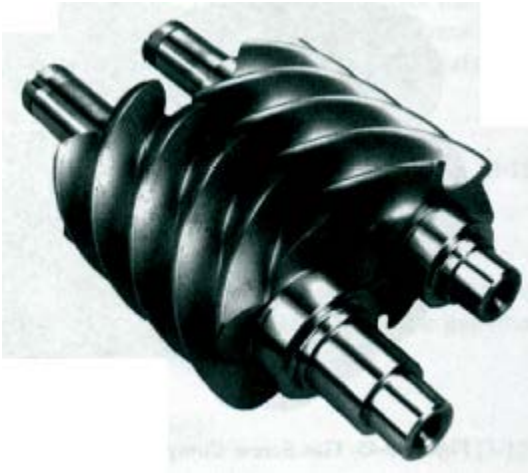

Figure 3: Rotary Screw Compressor Screws (5)

1.3 Why Theory is Important

Predicting how much energy is consumed by an air compressor is a useful tool to anybody that uses compressed air. Unfortunately predicting energy consumption can be difficult because both practical and theoretical knowledge is needed. Theoretical knowledge demonstrates why compressed air is energy intensive. Plus modifying a compressor in theory, on a piece of paper, is easier than modifying one in the real world. For example, take a plant engineer who spends a large sum of money to modify his air compressor to accept outside air in the winter months. With these modifications there is an expectation of a quick payback to the investment. If the fast payback period is not achieved, then reputations are damaged and other energy conservation measures are questioned. A situation such as this could be avoided by first testing the modification on paper with a thorough understanding of the physics. If desired results are found on paper then they should be implemented.

Practical knowledge is needed because sometimes theoretical ideas maybe impossible to implement. On paper using compressor waste heat for space heating appears great and has the potential for substantial savings. If the space is already hot in the winter time, due to other process equipment, then there are not savings from dumping compressor waste heat into this space. In actuality employees may not like this because the space could get too hot. There are endless ideas that sound good on paper but once it’s time to implement them, they fall through.

1.4 Pressure

Pressure is defined as *a normal force exerted by a fluid per unit area* (8). This is a simple concept yet there are so many different ways to represent units of pressure it makes pressure confusing. For example when calculating compressor performance absolute pressure must be used but in industry pressure may be referred to in one of the following ways, pound force per square inch (psi), pound force per square inch gauge (psig), pound force per square inch absolute (psia), or pound force per square inch differential (psid). Most commonly pressure is referred to in units of psi. Pressure is referred to in psi most commonly because most gauges report in psi, see Figure 4.

Figure 4: Air Pressure Gauge (9)

This unit is useless when predicting or estimating compressor performance because absolute pressure is needed. Absolute pressure is the actual pressure at a given position (8). It is

measured relative to a vacuum with absolute zero pressure. Most pressure gauges are calibrated to measure zero at atmospheric conditions. Therefore when a pressure gauge is read on a car tire or compressed air pipe the atmospheric pressure needs to be added to the gauge pressure to know absolute pressure inside the tire or pipe. This is analytically written as follows:

$$P_{absolute} = P_{guage} + P_{atmosphere} \qquad \textit{(Eqn. 1.4.1)}$$

Most of the time it is safe to assume atmospheric pressure is 14.7 psia, but atmospheric pressure does vary with altitudes. Atmospheric pressure is greatest towards the center of earth because the atmosphere gets denser as you approach the center. It is like swimming in a pool, as you approach the bottom of the pool you feel a greater force exerted on your body due to an increasing pressure. This is because the weight of water molecules stacked on top of each other increases with depth. This example can be expressed with the following equation for pressure:

$$P_{absolute} = \rho g h \qquad \textit{(Eqn. 1.4.2)}$$

Where,

P	=	*Pressure*
ρ	=	*Density of fluid*
g	=	*Gravitational constant at location*
h	=	*Height from desired location of pressure*

The weight of air, around 0.075 lbf/ft^3 depending on conditons, might not seem like much but it adds up. As an example the normal force exerted on a football field due to atmospheric air

is 121,894,000 pounds force. The following table shows variations of atmospheric pressure with altitude.

Table 1: Atmospheric Pressure at Different Altitudes (10)

Geo potential Altitude above Sea Level - *h* - (*ft*)	Temperature - *t* - (^oF)	Acceleration of Gravity - *g* - (ft/s^2)	Absolute Pressure - *p* - (lb/in^2)	Density - *ρ* - (10^{-4} $slugs/ft^3$)	Dynamic Viscosity - *μ* - (10^{-7} $lb.s/ft^2$)
-5000	76.84	32.189	17.554	27.45	3.836
0	59	32.174	14.696	23.77	3.737
5000	41.17	32.159	12.228	20.48	3.637
10000	23.36	32.143	10.108	17.56	3.534
15000	5.55	32.128	8.297	14.96	3.430
20000	-12.26	32.112	6.759	12.67	3.324
25000	-30.05	32.097	5.461	10.66	3.217
30000	-47.83	32.082	4.373	8.91	3.107
35000	-65.61	32.066	3.468	7.38	2.995
40000	-69.70	32.051	2.730	5.87	2.969
45000	-69.70	32.036	2.149	4.62	2.969
50000	-69.70	32.020	1.692	3.64	2.969
60000	-69.70	31.990	1.049	2.26	2.969
70000	-67.42	31.959	0.651	1.39	2.984
80000	-61.98	31.929	0.406	0.86	3.018
90000	-56.54	31.897	0.255	0.56	3.052
100000	-51.10	31.868	0.162	0.33	3.087
150000	19.40	31.717	0.020	0.037	3.511
200000	-19.78	31.566	0.003	0.0053	3.279
250000	-88.77	31.415	0.000	0.00065	2.846

1.5 Ideal Gas or Not

Probably the most common and most misused equation relating pressure, volume, and temperature is the ideal gas equation of state. This famous equation is commonly written as follows:

$$Pv = RT \qquad \textit{(Eqn 1.5.1)}$$

Where,

P = *Absolute pressure (psia)*

v = *Specific volume* (ft^3/lbm)

R = *Gas constant for a particular gas (BTU/lbm-R)*

T = *Absolute temperature (R)*

Note R can be found for any gas as follows:

$$R = R_u / M \qquad \textit{(Eqn. 1.5.2)}$$

Where,

R_u = *Universal gas constant (1.98588 BTU/lbmol-R)*

M = *Molecular weight of particular gas (lbm/lbmol)*

This equation is commonly misused because, in reality, a truly ideal gas does not exist. However, under the right conditions it is safe to assume a gas is ideal with little error. A compressibility factor was created to show how ideal a gas is. The compressibility factor is represented by the letter Z and found with the following equation:

$$Z = {}^{v_{actual}}/_{v_{ideal}} \qquad \textit{(Eqn. 1.5.3)}$$

Where,

v_{acutal} = Actual specific volume of gas (ft^3/lbm)

v_{ideal} = ideal specific volume of gas (ft^3/lbm) = RT/P

If the compressibility factor is equal to one the gas is ideal and there is zero error in using the ideal gas equation. As this factor deviates from one, the error increases. As an example take steam at one psia and 101.69 ºF. The actual specific volume of steam at this pressure is 333.49 ft^3/lbm. Using the ideal gas law, the ideal specific gas volume is calculated to be 334.54 ft^3/lbm. The compressibility factor for steam at this temperature and pressure is 0.997. Percent error for this example is a mere 0.32%

Now evaluate steam at another temperature and pressure, 200 psia and 381.8ºF. At this state steam tables give an actual specific volume of 2.2882 ft^3/lbm. Assuming steam is an ideal gas at this state gives an ideal specific gas volume of 2.507 ft^3/lbm. Steam does not deviate from the ideal gas assumption much. The compressibility factor for this example is 1.15 with a percent error of 13.0%.

Evaluate steam as ideal gas at a pressure of 3,200.1 psia and 705ºF. Note this is nearly the critical pressure (3,200 psia) and temperature (704.8ºF) of water. The actual specific volume of steam at this state is 0.04975 ft^3/lbm. Using the ideal gas assumption the ideal specific volume of steam is 0.217 ft^3/lbm. The compressibility factor for this example is 0.229 with a percent error of 336%. It is fairly obvious using the ideal gas assumption for steam around this temperature and pressure will result in significant error.

There is a trend associated with these examples above. As a gas approaches its critical point its compressibility factor deviates from one, in other words, as a gas approaches its critical point considering it to be ideal is inaccurate. When equation 1.5.1 was originally discovered, by J. Charles and J. Gay-Lussac with experimental observations in 1802, it was found to work at low pressures in the vapor phase of a substance only (8). Further testing and experimentation supported evidence that gases were not ideal as they approached their critical point.

To realize when a gas is approaching its critical point two unitless normalizations were created. The reduced pressure (P_r) and reduced temperature (T_r). These normalizations are found as follows:

$$P_r = P_{actual} / P_{critical} \qquad \textit{(Eqn. 1.5.4)}$$

$$T_r = T_{actual} / T_{critical} \qquad \textit{(Eqn. 1.5.5)}$$

To demonstrate the benefits from normalizing pressure and temperature with their critical points consider the example of steam given earlier. Below in Table 2, reduced temperature and pressure are reported. Two states of steam are added to the example given earlier also.

Table 2: Steam as an Ideal Gas and Related Error

State	Actual Temp. (ºF)	T_r	Actual Pressure (psia)	P_r	v_{actual} (ft^3/lbm)	v_{ideal} (ft^3/lbm)	Z	Error
1	102	0.482	1	0.00	333	335	0.997	0.32%
2	382	0.723	200	0.06	2.88	2.51	1.15	13%
3	518	0.840	800	0.25	0.569	0.73	0.78	28%
4	705	1.00	3,200	1.00	0.050	0.22	0.23	336%
5	2,000	2.11	3,000	0.94	0.485	0.488	0.99	1%

As discovered by experiments it is safe to assume a gas is ideal when reduced pressure is less than one, pressure is low. Looking at state one and two in Table 2 supports this. At both these states the reduced pressure is low and the error from assuming steam is an ideal gas is fairly low. Assuming steam is an ideal gas at very high temperatures works as well. State five in Table 2 above has a reduced temperature of 2.11, meaning very high temperature, and

the error associated with assuming steam is ideal, 1%. Table 2 shows as the reduced temperature and pressure approach one, the critical point, the error assuming ideal gas increases drastically. State three has an error of 28% and state four has an error of 336%.

Researchers have created plots to support this evidence with experimentally determined Z values plotted against reduced pressure and reduced temperature. Figure 5 is a plot of reduced temperature, reduced pressure, and experimentally determined compressibility factors for several different gases.

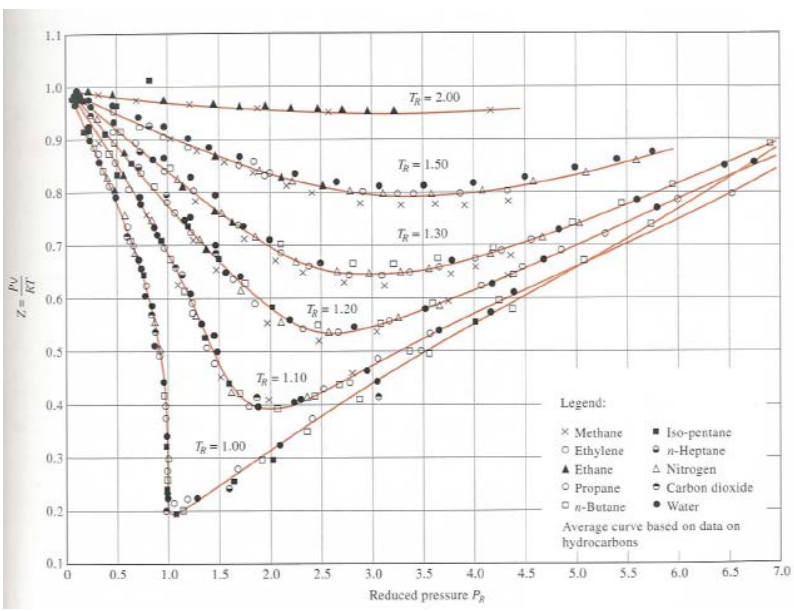

Figure 5: Compressibility Factor vs. Reduced Pressure (8)

In conclusion it is safe to assume a gas is ideal when its pressure is very low (P_r <<1), when its pressure is low with a corresponding high temperature, or when its temperature is

extremely high (T_r >>1). It is important to understand that a gas deviates from the ideal gas law when its actual temperature and pressure approach its critical temperature and pressure.

Chapter 2 – Today's Air Compressor

2.1 The Modern Compressed Air System

In the past compressed air was used to feed fires and today its common knowledge compressors are energy hogs, thus, why is compressed air still used today? Compressed air is very valuable to industrial facilities today because of its power and reliability. Compressed air is commonly used to operate tools such as impact wrenches, drills, and grinders. More efficient electrical motors could be used to power these tools but they are heavy and cause people, such as plant employees, to fatigue over time. Fatigue is not good because it can decreases production. To show the difference between an electric tool and a pneumatic tool the power to weight ratio for a grinder can be compared. The power to weight ratio of random electric grinder is 0.09 horsepower per pound while a pneumatic grinder is 0.31 horsepower per pound (4). The compressed air tool is nearly three and half times more powerful per pound!

Compressed air is commonly used for process controls today also. For particular manufacturing facilities that require explosion proof devices, such as chemical plants, can save much money on initial builds utilizing compressed air controls instead of electrical control systems. Compressed air controls are cheaper because they do not pose the risk of fire like their electrical counterparts (4). Until electrical devices become more robust, cheaper, and safer compressed air has a long future ahead of it.

To make sure pneumatic tools and or controls operate reliably and efficiently they must have a reliable source of compressed air. The most important device to supply compressed air is an air compressor. An air compressor compresses air from atmospheric pressure to an elevated pressure. When air expands from an elevated pressure to atmospheric pressure it can be used to do work or operate several pneumatic tools like the ones mentioned above. There are several different types of air compressor for different applications. It is very important to choose the correct air compressor depending on the task needed.

Storing high pressure air is essentially storing energy. Commonly a receiver, usually in the shape of a tank, is installed in a compressed air system to store compressed air. A receiver is an essential part to a compressed air system because it keeps system pressure consistent and it acts as a buffer. Consistency is important because many pneumatic tools, controls, and process require a particular pressure. When a receiver acts as a buffer it allows a compressor to unload more often. An unloaded compressor will consume less energy than a loaded compressor.

Pipe is another important component to a compressed air system. Pipe is a must because it transmits compressed air around a facility. Having the correct size pipe is essential to maximizing the efficiency of the compressed air system because it keeps pressure drop to a minimum. When pressure is lost (i.e. pressure drop across a pipe) and it is not used to do work then energy used to generate that high pressure air has been wasted.

Air commonly has moisture in it that needs to be removed before it is used in tools and or process. An air dryer is a device used to remove moisture in a stream of compressed air. Two types of air dryers are commonly used a refrigerant type and desiccant type. The refrigerant air dryer utilizes a vapor compression cycle to condense and remove moisture. The desiccant dryer utilizes compressed air and a desiccant to remove moisture. The desiccant dryer consumes more energy but removes more moisture. Selecting a dryer depends on what the compressed air is used for.

There are other devices that help control the pressure and what enters a compressed air device such as filters and pressure regulators. But, the most essential components to a compressed air system are the compressors, receivers (storage tanks), piping, and air dryers. An example of modern compressed air system, excluding end use tools, is shown on the following page in Figure 6.

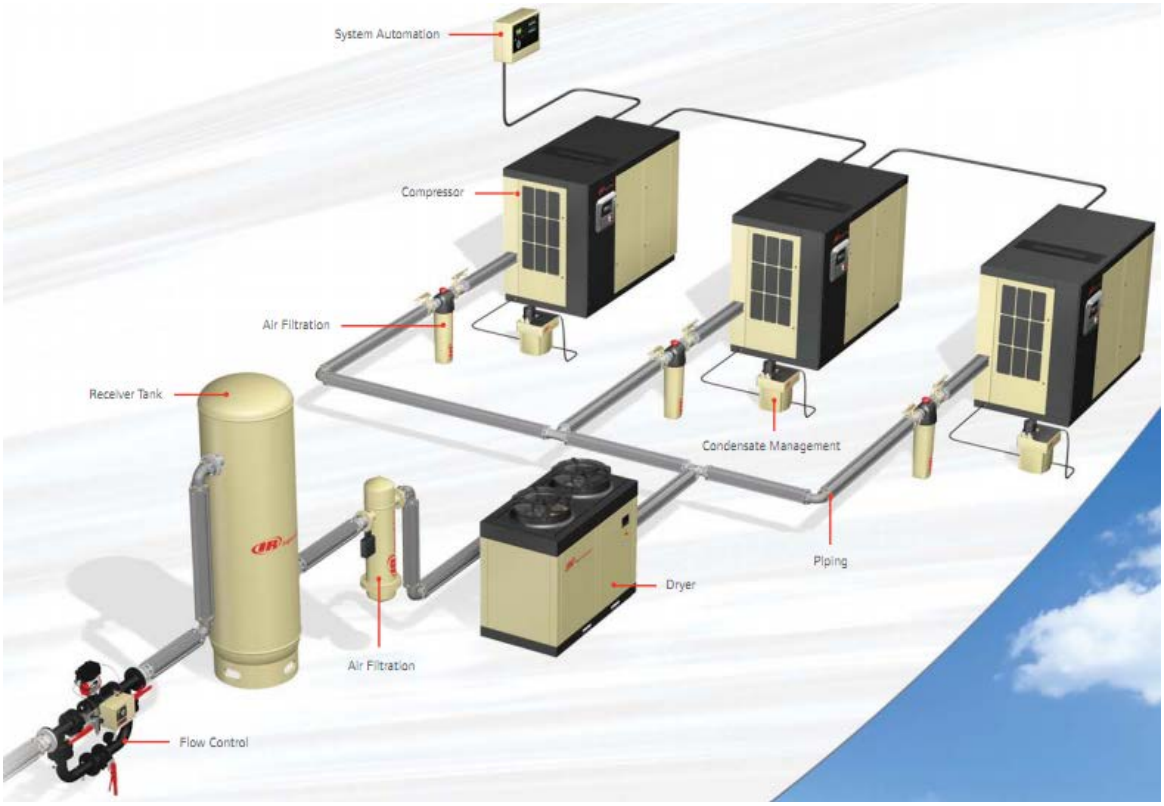

Figure 6: Modern Compressed Air System (11)

2.2 Physical Compressor Operation: Reciprocating and Twin Screw

The simplest mechanical air compressor is a reciprocating one. A reciprocating compressor is a piston cylinder device with an inlet and exit valve. The compression cycle begins with the piston at top dead center, where the volume inside the cylinder is essentially zero. In actuality there is a small volume between the piston and cylinder head at top dead center. This volume is referred to as clearance volume.

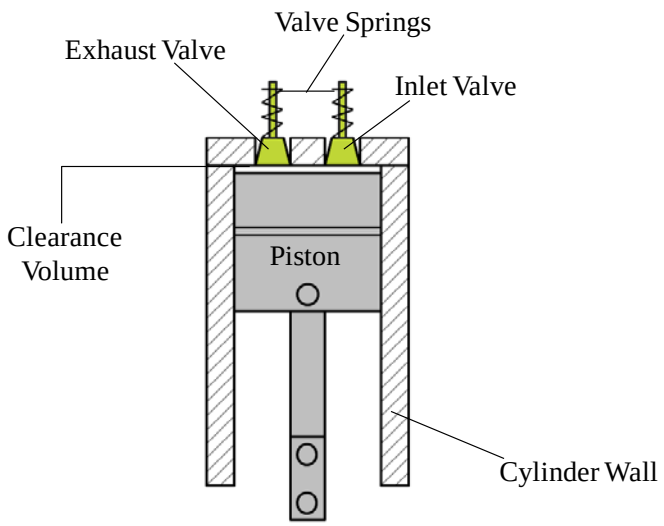

Figure 7: Piston at Top Dead Center

A force is applied to the piston to pull it down. As the piston is pulled down the cylinder's volume increases creating a vacuum inside the cylinder. Atmospheric pressure on top the valve is greater than gauge pressure inside the cylinder. This pressure differential causes the intake valve to open and causes air to fill the cylinder (i.e. blue dots represent air in Figure 8).

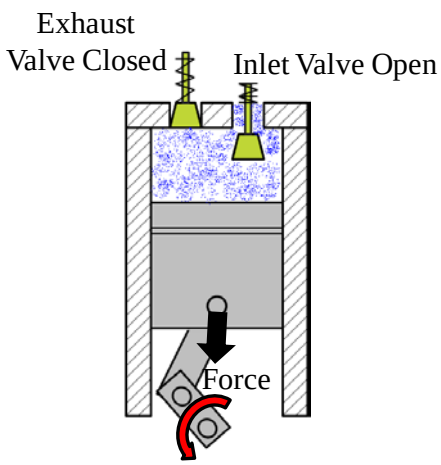

Figure 8: Cylinder Volume Increasing

Once the piston is at the bottom of its stroke, the largest volume of the cylinder, it is pushed back up by an external force.

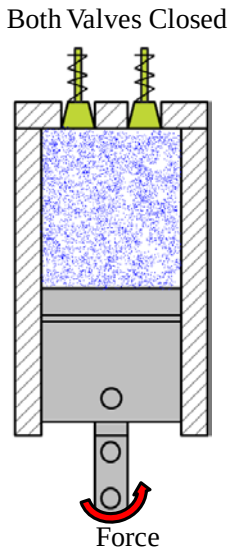

Figure 9: Piston at the Bottom of Stroke

As the piston moves towards top dead center the cylinder's volume decreases and pressure increases. According to the ideal gas law, equation 1.5.1 above, as volume decreases the pressure increases. Once the pressure inside the cylinder is greater than atmospheric pressure the intake valve closes. The exhaust valve opens once a desired pressure inside the cylinder is met. Usually the predetermined pressure is controlled by a spring forcing the valve shut. Using different spring rates or adjusting how much the spring is compressed allows the compressors operator to adjust the compressors outlet pressure. As the exhaust valve opens compressed air is released from the piston cylinder device.

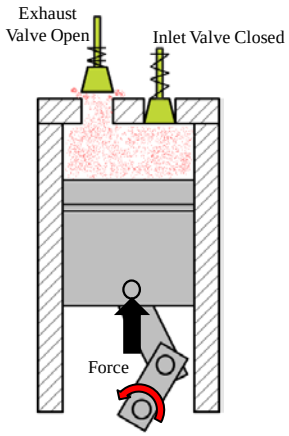

Figure 10: Piston at Top Dead Center Again

Note compressed air is represented with red dots in Figure 10 because compressing air generates heat. Once the piston reaches top dead center again most the compressed air has been pushed out the cylinder and both the intake and exhaust valve are closed and the cycle is ready to repeat. Today, the force to drive the piston is most commonly done with an

electric motor. In the past, steam was used to generate this force. Below, Figure 11, shows nearly a complete cycle.

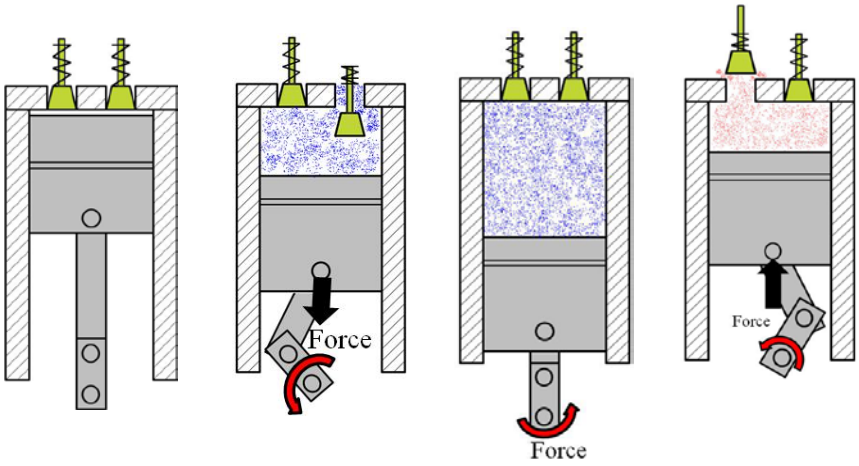

Figure 11: Reciprocating Compressor Full Cycle

A disadvantage to the reciprocating compressor is it's up and down motion causes vibration. Therefore a reciprocating compressor must be mounted on a secure platform or it will vibrate so much it fails over time.

The force generated by the electric motor is very important. This force translates to energy consumed by the compressor and the rate at which this force operates is power. As an example if the compressor's piston requires 550 lbf applied throughout a distance of one foot every cycle, where every cycle takes one second, then the compressor needs a one horse power electric motor. One horsepower is defined as 550 foot pound force per second (550 ft-lbf/sec). How much air is pulled into this compressor in every cycle is very important also.

This air at a higher pressure (compressed state) is the compressor's objective because compressed air can be used to drive robust tools such as drills.

One of the most common compressor found in the manufacturing world today is the twin screw compressor, commonly referred to as the screw compressor. The screw compressor does not compress air as shown above. However, the screw compressor in principle compresses air the same way, via mechanically shrinking a volume of air. The most basic screw compressor has two screws, a male and female, and a case. Below, Figure 12, shows the male screw (left) and female screw (right) meshed together.

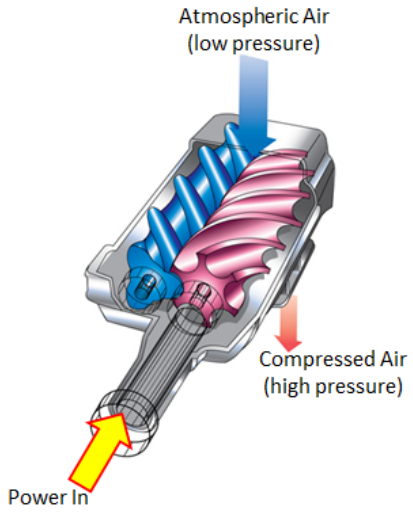

Figure 12: Compressor Screws (12)

As the input shaft rotates the screws begin meshing together. As screws mesh together they draw in air from the compressors intake. This air is trapped between the meshing screws and

as the screws rotate they compress the air against the screw's housing. The air compresses because of the screws conical taper (12). Once the air is at the end of the screws it is discharged. An advantage of the screw compressor is its ability to compress air with a coolant, such as oil. The coolant removes heat during the compression process, lubricates screws, and ensures meshing of screws. Another advantage of the screw compressor is its rotational operation. This allows the compressor to be mounted anywhere. Unlike the reciprocating compressor, vibration is less of an issue with the screw compressor. This also means less maintenance.

Both the compression process above are completed with positive displacement compressors. Another way to compress gases is using a dynamic compressor. Dynamic compressor work using a fluid's kinetic energy to compress itself against the compressor housing. Dynamic compressors are not seen as often as positive displacement compressors therefore will not be discussed further.

2.5 Volumetric Efficiency

Theoretically it is possible to design a positive displacement compressor where all the air drawn into the compressor is compressed and discharged. In the real world, this is impossible due to design tolerances. These tolerances strongly dictate a compressor's volumetric efficiency. Volumetric efficiency is defined as the ratio of volume actually delivered by the compressor compared to the displaced volume or theoretical volume of the

compressor (6). This means if the volume of air drawn into a reciprocating or screw compressor is measured and the volume of air discharged out the compressor is measured and converted back to inlet conditions the values would differ. An easier way to understand volumetric efficiency maybe stating: the mass of air exiting the discharge port of a compressor is smaller than the mass of entering the suction port.

The greatest effect on volumetric efficiency, for reciprocating compressors, is clearance volume. During a compressor's compression stroke some of the compressed air does not leave the compressor but instead occupies the compressor's clearance volume. Note air left in the clearance volume is around the compressor's discharge pressure. When the compressor begins its suction stroke the air occupying the clearance volume expands and keeps the inlet valve from opening until the cylinders pressure drops below that of atmosphere. This limits the amount of air the compressor can draw in during its suction stroke, thus, the volume of air discharged is smaller than the cylinder's volume. The greater a compressors discharge pressure ratio the lower its volumetric efficiency. The reason being higher discharge pressure means higher pressure air occupies the clearance volume. Therefore it takes longer for the cylinders internal pressure to drop below that of atmosphere. The other factors that lead to a reciprocating compressors volumetric efficiency are maintenance and age (6). As the compressor ages or maintenance is not done its piston rings wear down. Eventually the rings will wear down so much air can passes between the ring

and cylinder wall without being compressed. A plot of volumetric efficiency versus pressure ratio is shown below in Figure 13

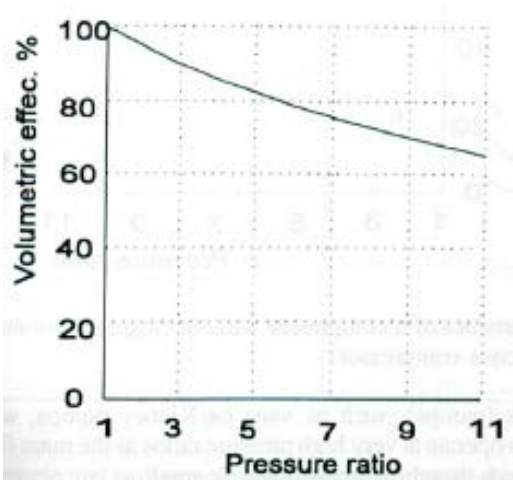

Figure 13: Volumetric Efficiency vs. Pressure Ratio for Reciprocating Compressor (6)

Volumetric efficiency losses in oil flooded screw compressors are much less than a reciprocating compressor due to the nature of their design. A clearance volume does not exist in screw compressors, but other tolerance issues do exist. Even though all the parts inside a screw compressors seem to touch; small tolerances or gaps exists between rotors, bearings, shafts, and seals. These tolerances pose an issue because air can escape through these gaps due to a pressure difference. Note the oil in an oil flooded screw compressor helps seal these tolerances tremendously. Therefore the biggest contributor to volumetric efficiency for a screw compressor is pressure ratio. The higher the discharge pressure the greater the pressure difference across tolerances or gaps therefore the greater the driving

force for air to escape. Plots of volumetric efficiency versus pressure ratio (Figure 16 and Figure 17) for screw compressors are found in Section 2.9.

2.6 Efficient Compression: Isentropic, Polytropic, or Isothermal

When a system goes from one equilibrium state to another it is considered a process. The act of compressing air is a process because air goes from a one state to another, usually a state of low pressure at some temperature to a state of high pressure at an elevated temperature. There are three different process paths air can follow as it is compressed. In particular air can be compressed using an isentropic process or an isothermal process. In actuality air is compressed in a polytropic process. Understanding all three processes is important because they clarify how compressors work in the physical world and help predict how much energy is consumed by a compressor.

As an example the amount of horsepower required to compress air with a volumetric flow rate of 690 CFM of air at 14.5 psia and 68ºF to 139.5 psia will be modeled using all three process and compared to a real world 150 horsepower compressor designed for this. Air at 14.5 psia and 68ºF is utilized because compressors are tested under these conditions, note this will be discussed further in Section 2.7. For all three processes air is assumed to be an ideal gas. This is a safe assumption because the critical temperature of air is (-221.5ºF) and critical pressure of air (547 psia). The lowest reduced temperature of air, for these examples, is 2.21, well above one. Looking at Figure 5, gases with a reduced temperature of two obtain a

compressibility factor of nearly one no matter the reduced pressure. The maximum reduced pressure will be 0.26 for every case. The following diagram will be used when calculating compressor energy requirements.

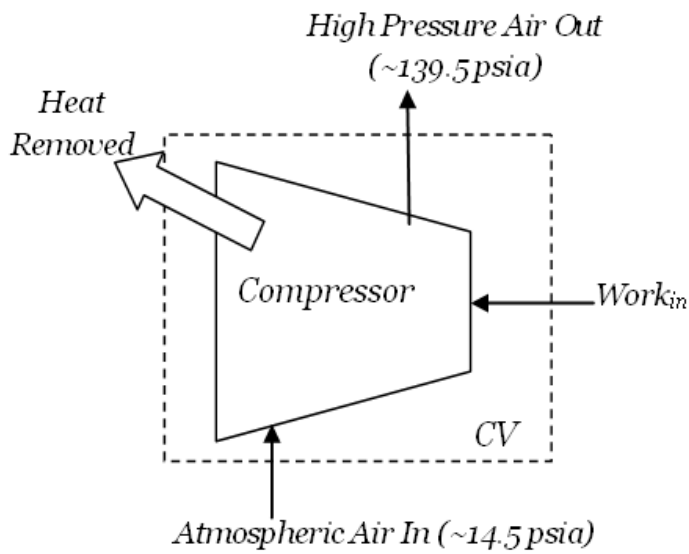

Figure 14: Thermodynamic Air Compressor Diagram

The following symbols will be used during derivations:

ΔE_{CV}	=	*change in rate of energy of control volume (BTU/hr)*
$E_{in\ toCV}$	=	*rate of energy into control volume (BTU/hr)*
$E_{out\ of CV}$	=	*rate of energy out of control volume (BTU/hr)*
Q_{in}	=	*rate of heat into control volume (BTU/hr)*
Q_{out}	=	*rate of heat out of control volume (BTU/hr)*
m_1	=	*mass flow rate of air entering control volume (lbm/hr)*
m_2	=	*mass flow rate of air exiting control volume (lbm/hr)*

h_1	=	*enthalpy of air entering control volume (BTU/lbm)*
h_2	=	*enthalpy of air exiting control volume (BTU/lbm)*

2.6.1 Isentropic Case

An isentropic compressor is a steady flow, reversible and adiabatic device (i.e. heat is not removed from the air being compressed) where kinetic and potential energy changes can be neglected meaning no entropy will be generated. Air will also be assumed to have constant specific heats and to be an ideal gas. Using these assumptions and the control volume in Figure 14 the work required to compress air from 14.5 psia at 68ºF to 139.5 psia is found as follows:

$$\Delta E_{CV} = 0$$

$$E_{in\ toCV} = E_{out\ ofCV}$$

$$W_{in} + m_1\,(h_1) = m_2(h_1)$$

For a steady flow device the mass entering must equal the mass leaving, therefore:

$$W_{in} = [m(h_2 - h_1)] \cdot C \qquad \textit{(Eqn. 2.6.1)}$$

For an ideal gas enthalpy is a function of temperature only. Therefore equation 2.6.1 can be rewritten as:

$$W_{in} = m \cdot c_p \cdot (T_2 - T_1) \cdot C \qquad \textit{(Eqn. 2.6.2)}$$

Where,

W_{in}	=	*Rate of energy or power required to compress air (BTU/hr)*
T_1	=	*Temperature of air entering compressor (ºF)*

T_2 = *Temperature of air leaving compressor (ºF)*

c_p = *Specific heat of air at constant pressure (BTU/lbm-ºF)*

C = *Conversion constant, (1 hp-hr/2,545 BTU)*

Equation 2.6.2 will predict the amount of horsepower required to compress 690 CFM of air from 14.5 psia to 139.5 psia. One problem exists with equation 2.6.2, the outlet temperature of the compressor is not known. This outlet temperature could be measured but this can be difficult to do. Assuming air is compressed isentropically the outlet temperature can be found theoretically from one of the three isentropic relationships. For an ideal gas with constant specific heats the following equations can be derived:

$$s_2 - s_1 = c_v ln\left(\frac{T_2}{T_1}\right) + Rln\left(\frac{v_2}{v_1}\right) \qquad \textit{(Eqn. 2.6.3)}$$

$$s_2 - s_1 = c_p ln\left(\frac{T_2}{T_1}\right) - Rln\left(\frac{P_2}{P_1}\right) \qquad \textit{(Eqn. 2.6.4)}$$

Where,

s_2 = *Entropy leaving the compressor (BTU/lbm-R)*

s_1 = *Entropy entering the compressor (BTU/lbm-R)*

c_v = *Specific heat of air at constant volume (BTU/lbm-ºF)*

R = *Gas constant for air (BTU/lbm-R)*

P_2 = *Absolute pressure of air exiting the compressor (psia)*

P_1 = *Absolute pressure of air entering the compressor (psia)*

v_2 = *Specific volume of air exiting the compressor (ft^3/lbm)*

v_1 = *Specific volume of air entering the compressor (ft^3/lbm)*

Isentropic means no entropy is generated, therefore (s_2-s_1 = 0). Using the isentropic assumption and algebra each equation can be rearranged as follows:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{R/c_v} \qquad \textit{(Eqn. 2.6.5)}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{R/c_p} \qquad \textit{(Eqn. 2.6.6)}$$

Equations 2.6.5 and 2.6.6 are fine to use, but in many thermodynamic books these equations are rearrange to look as follows:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{(k-1)} \qquad \textit{(Eqn. 2.6.7)}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \qquad \textit{(Eqn. 2.6.8)}$$

Equations 2.6.7 and 2.6.8 are found knowing:

$$R = c_p - c_v \qquad \textit{(Eqn. 2.6.9)}$$

and,

$$k = c_p / c_v \qquad \textit{(Eqn. 2.6.10)}$$

What most thermodynamic books do not show is how to get from equations 2.6.5 and 2.6.6 to equations 2.6.7 and 2.6.8. To get from equation 2.6.5 to 2.6.7 simply plug equation 2.6.9 into equation 2.6.5 as follows:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{(cp\,-\,cv)/c_v}$$

Using simple algebra and inserting equation 2.6.9 the exponent is rearranged:

$$\frac{c_p}{c_v} - \frac{c_v}{c_v} = (k-1)$$

and the final result is equation 2.6.7. There is a special trick to finding equation 2.6.8 from 2.6.6. The trick is to multiply the exponent in equation 2.6.6 by a ratio equivalent to one as follows:

$$\frac{R}{c_p} \cdot 1 = \frac{R}{c_p} \cdot \left(\frac{1/c_v}{1/c_v} \right) = \frac{R/c_v}{c_p/c_v} = \frac{(cp\ -\ cv)/c_v}{k} = \frac{(k-1)}{k}$$

Finally equation 2.6.8 is found. Using equations 2.6.7 and 2.6.8 is easier than 2.6.5 and 2.6.6 because fewer variables have to be found. Looking at equations 2.6.7 and 2.6.8 there is a third relationship that can be found.

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{v_1}{v_2}\right)^k \qquad \textit{(Eqn. 2.6.11)}$$

Equations 2.6.7, 2.6.8, and 2.6.11 are known as the isentropic relationships. It must be remembered when using these equations the system is assumed to be isentropic therefore no entropy is generated. In reality this will never occur because it is never a totally reversible process.

Back to the original problem, equation 2.6.2 could not be solved because the outlet temperature of air leaving the compressor was not known. Rearranging equation 2.6.8 to find T_2, inserting it into equation 2.6.2, and including the conversion from Fahrenheit to Rankin the following is found:

$$\mathrm{W}_{in} = m \cdot c_p \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C \qquad \textit{(Eqn. 2.6.12)}$$

To remove the c_p in equation 2.6.12 another trick is needed. Both sides of this equation need to be multiplied by R/R, which equal 1. Using algebra the following is found:

$$\left[\mathrm{W}_{in} = m \cdot c_p \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C \right] \frac{R}{R}$$

$$\mathrm{W}_{in} = m \cdot \frac{c_p}{R} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C$$

Where,

$$\frac{c_p}{R} = \frac{k}{k-1}$$

Therefore equation 2.6.12 can be re written as:

$$\mathrm{W}_{in} = m \cdot \frac{k}{k-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C \qquad \textit{(Eqn. 2.6.13)}$$

Another variable is still unknown, (*m*) the mass flow rate. The mass flow rate of air is fairly simple to determine with the following equation:

$$m = CFM \cdot \rho \cdot 60 \qquad \textit{(Eqn. 2.6.14)}$$

Where,

CFM = *volumetric flow into the compressor (ft^3/min)*

ρ = *density of air at inlet conditions (lbm/ ft^3)*

60 = *conversion constant (60 min/hr)*

Finally the equation to find the isentropic work required to compress air adiabatically at a constant flow rate assuming air is an ideal gas with constant specific heats is:

$$\mathrm{W}_{in} = CFM \cdot \rho \cdot 60 \cdot \frac{k}{k-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C \quad \textit{(Eqn. 2.6.15)}$$

In reality a 150 horsepower compressor is need to compress 690 CFM of air from 14.5 psia to 139.5 psia. Equation 2.6.15 predicts the following:

$$\mathrm{W}_{in} = 690 \cdot 0.074 \cdot 60 \cdot \frac{1.4}{1.4-1} \cdot 0.06855 \cdot (68 + 460) \left(\left(\frac{139.5}{14.5}\right)^{(1.4-1)/1.4} - 1 \right) \cdot \frac{1}{2{,}545}$$

$$\mathrm{W}_{in} = 139 \; horsepower$$

The isentropic power required to compress air is lower than the actual power required. An isentropic process is ideal; it assumes no entropy generation therefore there are no losses (i.e. friction). An isentropic efficiency needs to be included to calculate the actual power required to compress air at the given conditions. The isentropic efficiency of a compressor is defined as:

$$\eta_{isentropic} = \frac{\mathrm{W}_{isentropic}}{\mathrm{W}_{actual}} \quad \textit{(Eqn. 2.6.16)}$$

2.6.2 Isothermal

In the isentropic compression process above the compressor was assumed to be adiabatic (i.e. no heat transfer), thus heat was not removed during the compression process. So, the air's temperature leaving the compressor was higher than the temperature entering it, equation 2.6.8 above explains this analytically. An interesting question, to ask is what will happen if

all the heat is removed from air during the compression process? Logically thinking the compressor will require less power to compress air. As temperature rises, air expands (ideal gas law) equation 1.5.1. Therefore in the adiabatic process above the compressor is fighting expansion of air as its temperature rises. If the temperature is held constant (isothermal) the compressor will not have to fight the expanding air.

Assume the compressor is a steady flow non-adiabatic device where all the heat is removed during the compression process, where kinetic and potential energy can be neglected. Also, assume air has constant specific heats and an ideal gas. Using these assumptions and the control volume for a compressor, shown in Figure 14, the work required to compress air from 14.5 psia at 68ºF to 139.5 psia is found as follows:

$$\Delta \mathrm{E}_{CV} = 0$$

$$\mathrm{E}_{in\ toCV} = \mathrm{E}_{out\ ofCV}$$

$$\mathrm{W}_{in} + \mathrm{m}_1\,(h_1) = Q_{out} + \mathrm{m}_2(h_2)$$

For a steady flow device the mass entering must equal the mass leaving, therefore:

$$\mathrm{W}_{in} = [Q_{out} + m(h_2 - h_1)] \cdot C \qquad \textit{(Eqn. 2.6.17)}$$

Finally another assumption should be made. Assume the work done in this process is quasi-equilibrium. Since the work done is quasi-equilibrium it is also reversible work in theory. Net heat transfer from an internally reversible process is defined as:

$$\mathrm{Q}_{int.rev} = \int_1^2 T\mathrm{d}S \qquad \textit{(Eqn. 2.6.18)}$$

From Gibbs equation the following is known:

$$T \cdot dS = dH - VdP \qquad \textit{(Eqn. 2.6.19)}$$

Equation 2.6.17 can be rewritten as follows:

$$W_{in} = [-Q_{int.rev} + m(h_2 - h_1)] \cdot C \qquad \textit{(Eqn. 2.6.20)}$$

$$W_{in} = \left[-m\int_1^2 Tds + m(h_2 - h_1)\right] \cdot C \qquad \textit{(Eqn. 2.6.21)}$$

$$W_{in} = \left[-m\int_1^2 dh + m\int_1^2 vdP + m(h_2 - h_1)\right] \cdot C \qquad \textit{(Eqn. 2.6.22)}$$

$$W_{in} = \left[m\int_1^2 vdP\right] \cdot C \qquad \textit{(Eqn. 2.6.23)}$$

To find the power needed to compress air, specific volume (*v*) as a function of pressure needs to be known. Since air is assumed to be an ideal gas rearranging the ideal gas equation (equation 1.5.1) and plugging it into equation 2.6.23 can be done as follows:

$$W_{in} = \left[m\int_1^2 \frac{R(T_1+460)}{P} dP\right] \cdot C \qquad \textit{(Eqn. 2.6.24)}$$

Integrating equation 2.6.24 gives the following:

$$W_{in} = mR(T_1 + 460)ln\left(\frac{P_2}{P_1}\right) \cdot C \qquad \textit{(Eqn. 2.6.25)}$$

Inserting equation 2.6.14 for mass flow rate the following is found:

$$W_{in} = CFM \cdot \rho \cdot 60 \cdot R \cdot (T_1 + 460) \cdot ln\left(\frac{P_2}{P_1}\right) \cdot C \qquad \textit{(Eqn. 2.6.26)}$$

Equation 2.6.26 represents, analytically, how much power is required to compress air from a low pressure to an elevated pressure assuming the process is isothermal, the gas is ideal, and specific heats are constant. The amount of power required to compress 690 CFM into the compressor at 68ºF from 14.5 psia to 139.5 psia is found as follows:

$$W_{in} = 690 \cdot 0.074 \cdot 60 \cdot 0.06855 \cdot (68 + 460) \cdot ln\left(\frac{139.5}{14.5}\right) \cdot \frac{1}{2{,}545}$$

$$W_{in} = 99\ horsepower$$

This is a 29% reduction in power from the isentropic process. The idea that the compressor has to fight the gas expanding is correct. In the actual world creating a compressor to compress a gas at constant temperature is impossible.

2.6.3 Polytropic

The isentropic and isothermal processes, to compress air, have been discussed. Some information that was not mentioned is how pressure and volume are related for a closed system compression process. Pressure and volume for a closed system isentropic process are related by the following equation:

$$Pv^k = Constant \qquad \textit{(Eqn. 2.6.27)}$$

and pressure and volume for an isothermal process are related by the following:

$$Pv = Constant \qquad \textit{(Eqn. 2.6.28)}$$

In the real world a compressor compresses a gas somewhere in between the isentropic and isothermal process where the following is true:

$$Pv^n = Constant \qquad \textit{(Eqn. 2.6.29)}$$

This process in between is called the polytropic process. Figure 15 is a plot of pressure versus specific volume for all three processes:

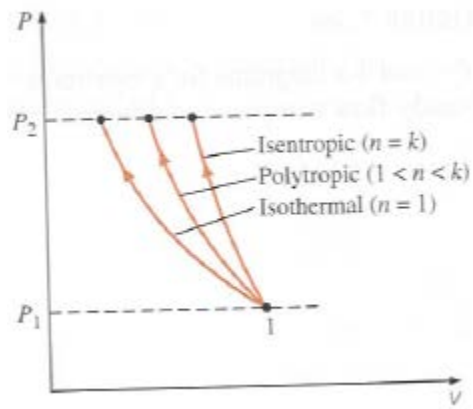

Figure 15: Compression Processes Pressure vs. Specific Volume (8)

The area under each curve is the amount of work required to compressor a mass of air. It can be seen that the isentropic process requires the most work and the isothermal the least amount of work. These processes, the isentropic and isothermal, are both impossible to do in the real world. It is impossible to insulate a compressor where zero heat is loss, or to compress a gas in which all heat is rejected and the inlet temperature stays constant throughout. Therefore in reality the polytropic process occurs.

Looking at Figure 15, equation 2.6.27, equation 2.6.28, and equation 2.6.29 it can be seen that as *n*, for the polytropic process, approaches *k*, the specific heat ratio of a gas, the process becomes more isentropic. It is also seen that as *n* approaches one the process becomes more isothermal (8). Knowing this, the rate of work required to compressor a mass of air in a polytropic process assuming the process is internally reversible, with constant specific heats, and the gas is ideal is found is found by modifying the isentropic power equation as follows:

$$\mathrm{W}_{in} = m \cdot \frac{n}{n-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1 \right) \cdot C \qquad \textit{(Eqn. 2.6.30)}$$

Using the same procedure used in the isentropic process equation 2.6.30 can be written in terms of CFM as follows:

$$\mathrm{W}_{in} = CFM \cdot \rho \cdot 60 \cdot \frac{n}{n-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1 \right) \cdot C \quad \textit{(Eqn. 2.6.31)}$$

The power to compress 690 CFM of air polytropically from 14.5 psia at 68ºF to 139.7 psia assuming *n = 1.3* is:

$$\mathrm{W}_{in} = 690 \cdot 0.074 \cdot 60 \cdot \frac{1.3}{1.3-1} \cdot 0.06855 \cdot (68 + 460) \left(\left(\frac{139.5}{14.5}\right)^{(1.4-1)/1.4} - 1 \right) \cdot \frac{1}{2{,}545}$$

$$\mathrm{W}_{in} = 130\ horsepower$$

As discussed the worked required to compress 690 CFM of air at standard conditions polytopically lies in between the isentropic process, 139 horsepower, and the isothermal process 99 horsepower.

2.7 Air Properties

Power required to compress air is believed to be a function of three air properties. Two of these properties are easily seen in equations 2.6.13, 2.6.25, and 2.6.30 above:

$$\mathrm{W}_{in} = m \cdot \frac{k}{k-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \right) \cdot C \qquad \textit{(Eqn. 2.6.13)}$$

$$W_{in} = mR \cdot (T_1 + 460) ln \left(\frac{P_2}{P_1}\right) \cdot C \qquad \textit{(Eqn. 2.6.25)}$$

$$W_{in} = m \cdot \frac{n}{n-1} \cdot R \cdot (T_1 + 460) \cdot \left(\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right) \cdot C \qquad \textit{(Eqn. 2.6.30)}$$

These two properties are inlet temperature, T_1, and inlet pressure, P_1. The lower the inlet temperature the less power needed to compress a mass of air. As the inlet pressure gets closer to the outlet pressure, P_2, the power required to compress a mass of air decreases. This concept may be easier to explain by stating: as the pressure ratio, P_2 / P_1, approaches one less power is needed to compress a mass of air. The property of air that is not easily seen in the above equations is humidity, which affects the mass of air entering a compressor.

2.7.1 Humidity Effect

Humidity is a word commonly heard on the weather forecast during summer months. Humidity refers to the amount of water vapor in the air. To ways to quantity humidity are specific humidity, ω, and relative humidity, RH. Specific humidity is easier to understand conceptually. Specific humidity, for a sample of air, is a ratio of water vapor mass to dry air mass (3). Specific humidity is commonly expressed analytically as follows:

$$\omega = \frac{m_{wv}}{m_{da}} \qquad \textit{(Eqn. 2.7.1)}$$

Where,

m_{wv} = *Mass of water vapor (lbm)*

m_{da} = *Mass of dry air (lbm)*

Specific humidity is very useful because it gives the mass of water vapor in a volume of air but humidity is not commonly referred to in terms of specific humidity. Instead, humidity is referred to in terms of relative humidity.

Relative humidity is the percent value listed on weather reports (i.e. 70% humidity at 75ºF). This percentage is important because it gives an idea of how close moist air is to saturation, meaning the point where air cannot hold anymore water vapor. Air that is completely dry has a RH of zero percent and saturated air has a RH of 100%. It is important to note as air rises in temperature the amount of water vapor air can hold increases. This can be seen on the psychrometric chart, see *Appendix C*. On the psychrometric chart taking a fixed specific humidity it can be seen that relative humidity decreases as temperature rises. Therefore relative humidity is a ratio of the amount of water vapor air holds relative to the maximum amount of moisture or water vapor the air can hold at the same temperature (8). Relative humidity is commonly found with the following equation:

$$RH = \frac{P_{wv}}{P_{sat\,@\,T}} \qquad \textit{(Eqn. 2.7.2)}$$

Where,

P_{wv}	=	*Partial pressure of water vapor (psia)*
$P_{sat\,@\,T}$	=	*Saturation pressure of water vapor at a particular temperature (psia)*

Why does humidity affect a compressor performance? It should be obvious as a mass of air is compressed water vapor in the air is compressed also. The desired output of an air

compressor is compressed air, hence the name, and not high pressure water vapor. Water vapor is usually removed from a compressed air stream by the air dryer because water can be harmful to pneumatic tools and or processes. When a cubic foot of air is drawn into a compressor and compressed, the moisture in the air takes up space. Therefore if humid air is drawn into a compressor the compressor will compress more cubic feet of humid air to reach a desired output of dry air. Since more air is being compressed the air compressor will consume more energy. The energy increase for humid air is not significant. For example, the specific humidity for air at 90ºF and zero percent RH is zero and the specific humidity for air at 90ºF and 100% RH is just over 0.03. This means the humid air should only see a three percent reduction of dry air produced when compressed. Therefore compressor energy consumption will only increase about three percent. For this reason it's also believed the biggest problem with compressing moist air is the chance or water vapor entering tools and or processes.

2.8 The Standard Cubic Foot

Pneumatic process and tools are usually rated in standard cubic feet per minute (SCFM) and compressor capacity is rated in actual cubic feet per minute (ACFM) at inlet conditions. To make terminology more confusing compressors sometimes have an inlet cubic feet per minute (ICFM) rating. A standard cubic foot of air is utilized for processes and tools because it specifics the mass of air needed. The mass of air, not the volume, and pressure determine how much work compressed air can do as it expands. A standard cubic foot of air always has

a mass of 0.0741 pounds. Therefore when a pneumatic grinder requires 41 SCFM at 90 psig it means the tool requires 3.04 pound mass of air per minute at 90 psig. According to the Compressed Air and Gas Institute (GAGI) a standard cubic foot of air has temperature of 68ºF, a pressure of 14.5 psia, and zero percent RH. CAGI's website supplies the following formula for converting SCFM to ACFM:

$$ACFM = SCFM \times \frac{P_{standard}}{[P_{atmospheric} - (ppm \times RH)]} \times \frac{(T_{atmospheric} + 460)}{(T_{standard} + 460)} \qquad \textit{(Eqn. 2.8.1)}$$

Where,

$P_{standard}$	=	*Standard pressure (14.5 psia)*
$P_{atmospheric}$	=	*Atmospheric pressure (usually 14.7 psia)*
ppm	=	*Partial pressure of moisture at atmospheric temperature*
RH	=	*Relative humidity*
$T_{atmospheric}$	=	*Atmospheric temperature (ºF)*
$T_{standard}$	=	*Standard temperature (68ºF)*

The CAGI website supplies the following definitions also: ACFM is flow rate of air at a certain point at a certain condition. ICFM is CFM flowing through the compressor inlet filter or inlet valve under rated conditions. SCFM is free air measured and converted to a standard set of reference conditions (14.5 psia, 68ºF, and zero percent RH).

2.9 Theory Conclusion

All the equations above are difficult to derive and require several assumptions. These equations only give an estimate of the power required to compress air. A better way to think about these equations is they estimated the amount of power a fluid, air in this case, absorbs when its compressed (6). In actuality this power needs to be applied to a fluid. This power can be applied to the fluid many ways. Two examples of doing this are using a piston compressor or twin screw compressor. Since a machine needs to apply power to a fluid other losses need to be accounted for also. Some examples are electrical motor losses, inertia of moving parts, bearing friction, vibrations, and gearbox losses. This can get very difficult fast.

A compressor has been built were the fluids inlet temperature was higher than its outlet temperature. This was done by injecting a specific amount of coolant at a particular temperature while a fluid was compressed, the same theory as an oil flooded screw compressor (6). This appears to be an isothermal compression process, or even better, but when the compressor's efficiency was tested it was worse than the projected isothermal efficiency. The compressors efficiency was even lower than its projected isentropic efficiency. Therefore it is common to categorize compressors by their isentropic efficiency no matter if they are oil flooded or not (6). The figures below give isentropic or adiabatic and volumetric efficiency for oil free and oil flooded screw compressors.

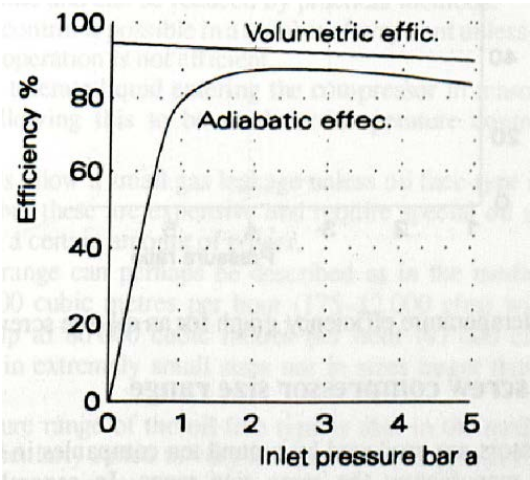

Figure 16: Oil Free Screw Compressor Volumetric and Adiabatic eff. vs. Inlet Pressure (6)

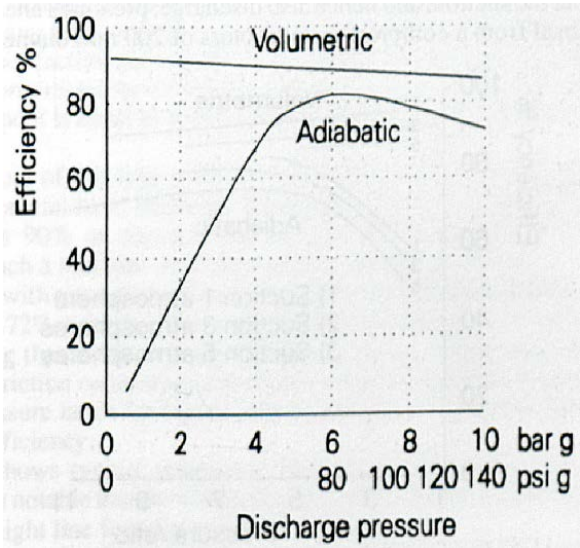

Figure 17: Oil Flooded Screw Compressor Volumetric and Adiabatic eff. vs. Discharge Pressure (6)

It is important to note all examples above have been analyzing the control volume of a compressor only. If an entire compressed air system where to be evaluated many more

variables would come in to play. Unfortunately evaluating an entire compressed air system is outside the scope of this thesis, but the following control diagram was created to represent some of the inputs and outputs of an entire compressed air system.

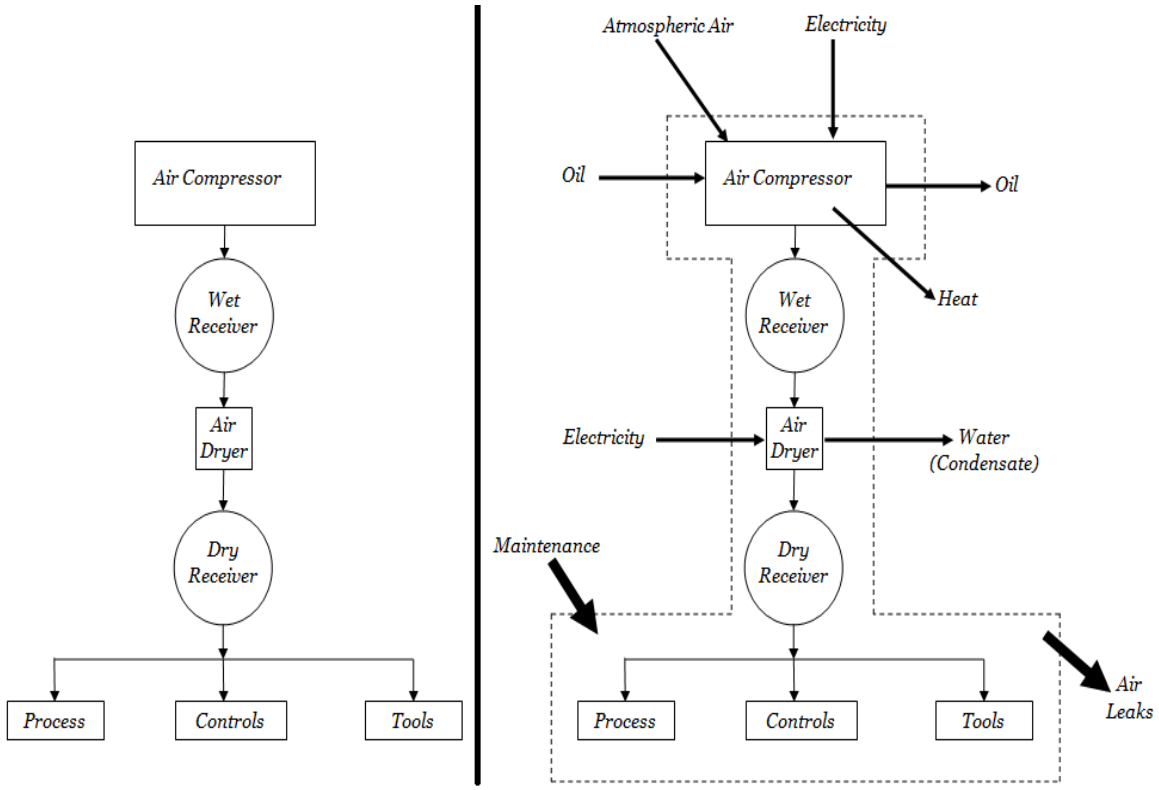

Figure 18: Compressed Air System Control Volume

Finally it is important to note efficiency is a ratio desired output over input. Therefore air compressor efficiency is usually rated in SCFM (desired output) per horsepower (electrical input).

Chapter 3 – Understanding Electricity

Compressing air requires a substantial amount of power and energy, as seen in Chapter 2. This energy can be obtained numerous ways such as steam, fossil fuels, or electricity. Three phase electricity is the most common form of energy used, to power an air compressor's motor, in today's industrial environment. Therefore if the mechanical side of an air compressor is improved (i.e. the rotors or pistons compressing air require less energy) the electrical motor should draw less electrical power and energy. Electric motors do not operate the same as internal combustion engines many people are familiar with. If the mechanical side of an air compressor powered by an internal combustion engine is improved, in terms of SCFM produced per pound mass of fuel burned, the motors throttle can be simply backed down to get the same compressed air output consuming less fuel. The same should happen for an electrical motor in theory (i.e. SCFM per electrical horsepower should increase) but alternating current three phase electrical motors cannot be throttled back simply, thus, what happens? The discussion of this matter will begin with three phase alternating current electricity, followed by some electrical motor basics.

3.1 Electricity and three phase power

Three phase electricity is simply three single phase sinusoidal supplies or electricity offset by 120º. A visual example of three phase power is shown below in, Figure 19.

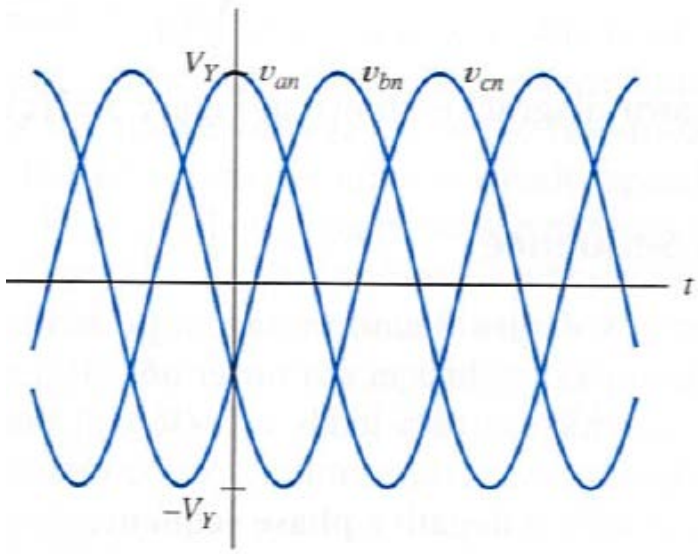

Figure 19: Three Phase Power (13)

One reason for utilizing three phase electricity, in industrial settings, is it supplies a fairly constant voltage. An example of single phase electricity versus three phases is: phase A (v_{an}), of Figure 19, is connected to an electrical motor. In this situation the electrical motors power output will fluctuate in a sinusoidal manner ever cycle, not very consistent. If phase A, B (v_{bn}), and C (v_{cn}) are connected to an electrical motor its power output will be fairly constant because one of the three phases will be nearing its peak voltage at any given point in time (14).

In the United States three phase power is generated at high voltage, distributed over large power lines, and finally supplied to a customer. When the power lines meet a customer electricity is supplied to them at a desired voltage utilizing a transformer. A transformer is

composed of two sides, a primary and secondary. The primary side is where the power plants high voltage electricity flows through it and the secondary side is where the customer's power flows out at a reduced voltage. The reduced voltage depends on what the customer needs electricity to power. Common voltages on the secondary side of a transformer are 120 volt, 208 volt, 277 volt, 480 volt, 2,300 volt, and 4,160 volt (15). Big power consumers use a higher voltage so smaller wires can be used to power their electrical equipment. This makes power distribution more economical because less copper (conductive material) is needed. The reason this works is electrical power is the product Volts and Amps therefore the higher the Voltage the less Amps needed. It is common to use the analogy Volts is equivalent to pressure in a pipe and Amps is equivalent to the amount of water flowing through a pipe.

Continental Control Systems LLC has a website that shows how different electrical services are supplied. Their diagrams show the secondary windings of several transformers and each has a brief description. Here are two of the common three phase services utilized:

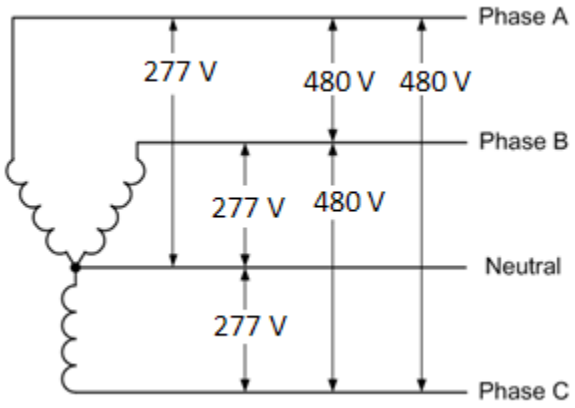

Figure 20: Three Phase Four Wire Wye

According to Continental Control Systems LLC, the most common electric service used in North America is the 4 wire wye setup. This service is utilized to power equipment such as lighting and large HVAC loads. Each phase is supplied voltage which is 277 Volts above ground and 480 Volts different from the other 2 phases.

Below is a three phase wire delta. According to Continental Control Systems LLC, three phase wire delta is primarily used to power three phase electric motors.

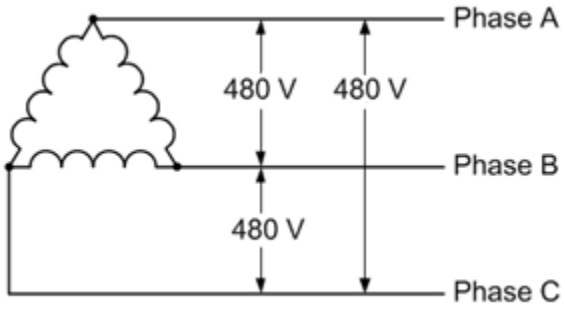

Figure 21: Three Phase Wire Delta

Another delta configuration is shown in, Figure 22. This delta is also known as a corner grounded delta. The reason for using this configuration is to save money on wiring cost because a service cable with only two insulated conductors is used.

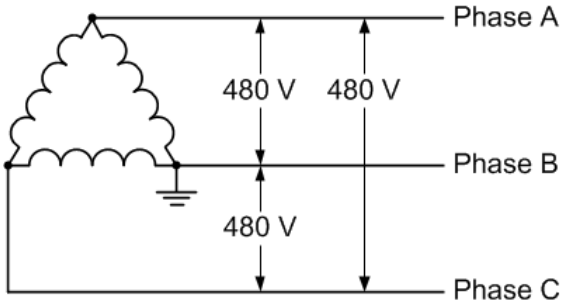

Figure 22: Three Phase Two Wire Corner-Grounded Delta

Assuming all of these electrical services supply balanced three phase electricity, meaning amps and voltage are the same on each phase, and assuming current and voltage are in phase the power delivered for each service is found as follows:

$$P_{electrical} = \sqrt{3} \cdot V_{line} \cdot I_{line} \qquad \textit{(Eqn 3.1.1)}$$

Where,

$P_{electrical}$	=	*Real electrical power delivered (kW)*
V_{line}	=	*Line to line voltage (volts)*
I_{line}	=	*Line to line current (amps)*

3.2 AC Electric Motors

Three phase AC motors are commonly used in industrial environments because they require practically no maintenance and power is commonly supplied in three alternating phases. Most commonly the squirrel cage electric motor is used because of its simplicity and ruggedness (16). Squirrel cage electrical motors operate at nearly constant speed in terms of revolutions per minute (rpm). Therefore if a variable industrial process exists, which requires speed regulation, these motors may not be desired. Unfortunately all AC motors are synchronous devices in theory. Meaning, they operate in a direct proportion to the frequency of electricity supplied to them, where the synchronous speed of an electrical motor is found as follows:

$$RPM = \frac{Frequency \times 120}{P} \qquad \textit{(Eqn. 3.2.1)}$$

Where,

Frequency	=	*Supplied electric frequency (60 Hz)*
P	=	*Number of poles (commonly 2, 4, 6, or 8)*

Therefore an electrical motor made with 4 poles has a synchronous speed of 1,800 rpms. In reality an electric motor does not have a synchronous relationship with frequency due to a load being applied to the motors output shaft. The difference between actual motor speed and synchronous speed is slip (16). An electrical motors percent slip is found as follows:

$$Slip\ (\%) = \frac{Synchronous\ Speed - Operating\ Speed}{Synchronous\ Speed} \times 100 \qquad \textit{(Eqn. 3.2.2)}$$

Since slip is dependent of motor loading the greater the load the greater the slip. In reality a fully loaded electric motor, in terms of design load, slips between 1 and 4% (16). How an electric motor's current and torque are affected compared to synchronous speed are shown below.

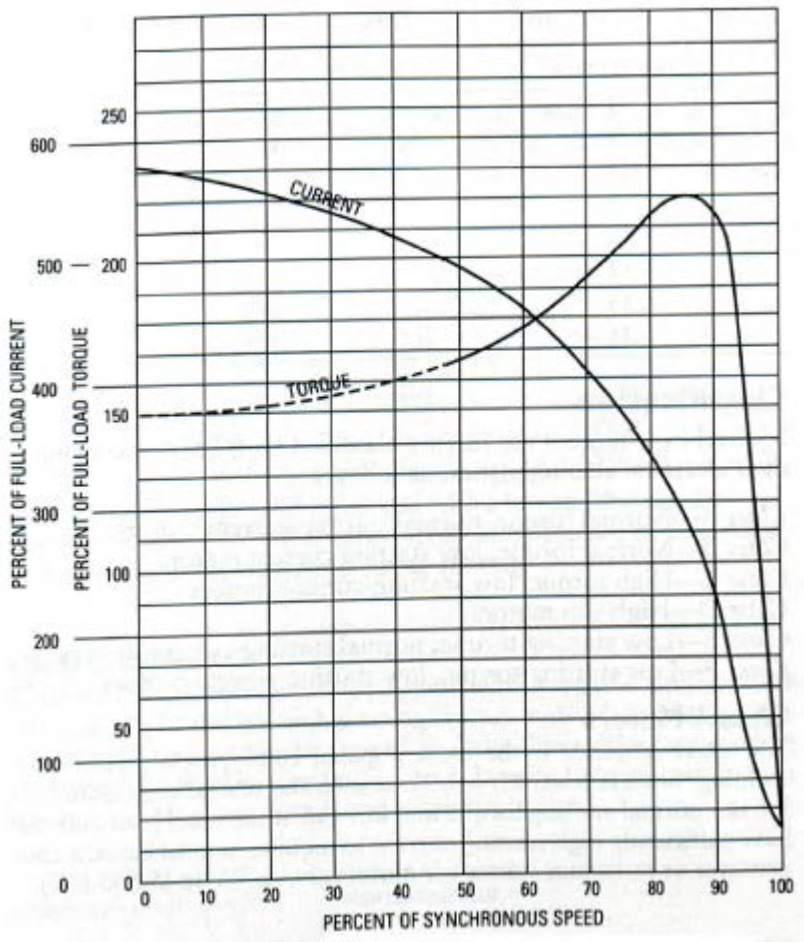

Figure 23: Characteristic Curve for an Electric Motor (16)

Motor loading also affects the electric motors efficiency and power factor. Electric motors are commonly designed to operate from 50% to 100% of their rated output (17). When electric motor operates in this range, 50 to 100%, their efficiency is fairly constant. If an electric motors load drops below 50% of its rated output its efficiency begins to drop. Motors loaded more than 100% of their rated output tend to overheat, which leads to

inefficiencies, and eventually failure. Failure occurs due to deteriorating insulation. A plot of percent full load current versus time, in minutes, shows how quickly a motor can overheat.

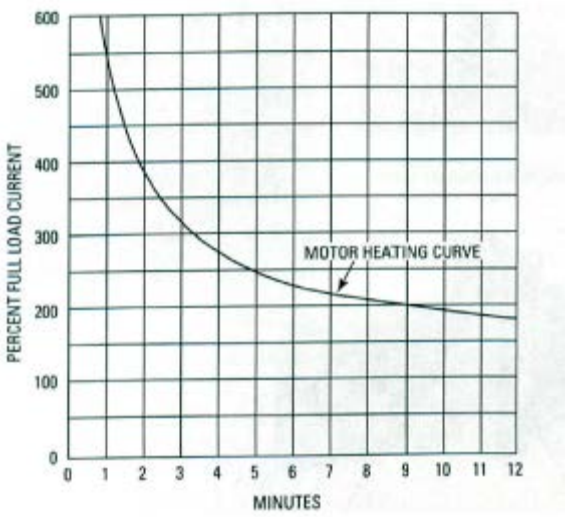

Figure 24: Motor Heating Curve (16)

Electric motors are said to operate in their service factor when they deliver more than their rated power output. Usually electric motors are designed to operate in their service factor for short periods of time (17). Typical service factors are around 115% to 120% of an electric motors output. This means a 150 horsepower compressor can output 173 to 180 horsepower for short periods of time without being damaged. The following figure visually shows how electric motors efficiency changes with a varying load.

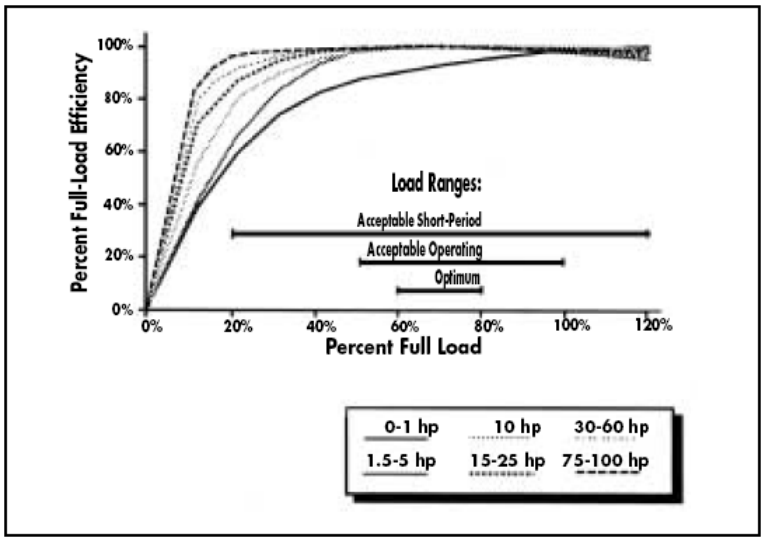

Figure 25: Motor Efficiency vs. Percent Full Load (17)

Power factor occurs when voltage and current are not supplied in phase. This is a concern because some utilities charge a penalty for poor power factor. Figure 26 shows how current can lead voltage or how current can lag voltage. Both these plots are visual examples of voltage and current drawn out of phase.

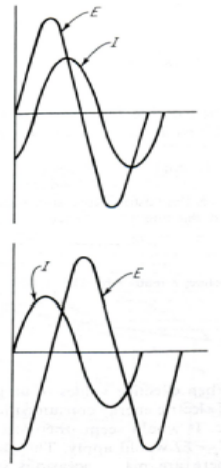

Figure 26: Electricity Leading and Lagging (18)

A power factor of 100% means current and voltage are in phase perfectly and a power factor of 0% means voltage and current are out of phase by 180 degrees. Current and voltage drawn out of phase reduce the amount of power supplied because their peak values do not coincide (18). Therefore equation 3.1.1 must be adjusted to include power factor. The real power supplied to an electrical device is found as follows:

$$P_{electrial} = \sqrt{3} \cdot V_{line} \cdot I_{line} \cdot PF \qquad \textit{(Eqn 3.2.3)}$$

Where, *PF* is power factor expressed as a percentage. It is important to understand that the power factor of electricity supplied to an electric motors decreases as the motors loads decreases. A plot of power factor versus load for different sized electrical motors is shown below.

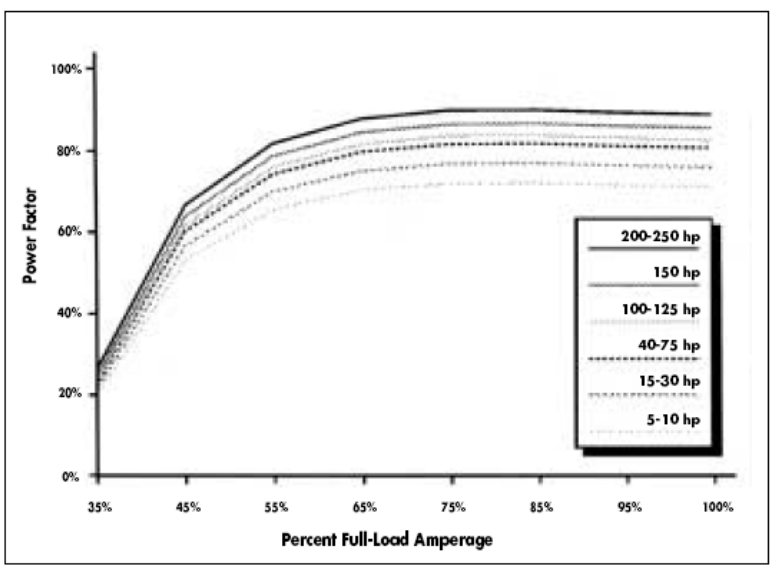

Figure 27: Motor Power Factor vs. Percent Full Load Current Draw (17)

The objective of this section is to show how correctly sizing an electric motor is important and how making the mechanical side of a compressor more efficient will result in electrical power savings. Viewing Figure 25 shows if a compressor's loading is reduced by 10% the compressor will draw nearly 10% less electricity. Unfortunately one of the easiest ways to reduce the power required to compress air is to match air capacity and air demand exactly which requires speed regulation. The only ways to adjust the speed of an electrical motor is adjusting slip, number of poles, or frequency.

Chapter 4 – Best Practices

4.1 Capacity Control for Oil Injected Screw Compressors

Ideally a compressor's power consumption will vary with desired capacity in a one to one proportion or better. Where capacity is "the amount of air flow delivered under specific conditions, usually expressed in cubic feet per minute (CFM)" (19). An example of ideal capacity control is a facility with an industrial process that needs 300 SCFM of air. Assume at full load, meaning the compressor consume 100% of its rated input power, the facility's compressor delivers 600 SCFM of air. If ideal capacity control is used only 50% of the compressors rated input power is consumed because only 50% of the compressors capacity is needed for process. In theory ideal capacity control can be completed with screw compressors. The volume of air traveling through a screw compressor is directly proportional to the frequency or speed the screws are rotating at. This is good news but there lies a problem, most industrial screw compressors are driven by electrical motors. If compressors were driven by another power source, such as a combustion engine (like your car), a throttle could be adjusted to meet air demand, kind of like a car on cruise control. The best way to adjust the speed of an alternating current electric motor is to use a variable frequency drive (VFD). These drives are becoming more common but they are expensive.

Therefore other capacity control methods are used today. Several capacity control methods are used for oil flooded screw compressors, some are: modulating, modulating with blowdown, load / unload, variable displacement, and speed control. A 150 horsepower compressor with a 690 SCFM capacity is used to describe five capacity control methods.

Modulating capacity control is sometimes referred to as suction throttling, because there is a valve located at the compressor's inlet that opens and closes depending on air demand. When the inlet valve is fully open the compressor is in a full-load state. Meaning, the compressor is delivering maximum air capacity (690 SCFM) while operating at full speed consuming maximum power (150 horsepower). As the inlet valve shuts, the compressor's capacity and power consumption decrease because less air mass is drawn through the compressor. The power consumption does not drop in a one to one proportion though, reason being, a pressure drop develops across the compressor inlet valve as it shuts. For the compressor to keep the same discharge pressure the pressure ratio (P_2 / P_1) must increase causing the compressor to consume more power. When the inlet valve is completely shut the compressor goes into an unloaded state. Theoretically, when unloaded the compressor delivers no air (i.e. 0 SCFM of air) while consuming 70% of its rated power (105 horsepower). The compressor still consumes 70% of its rated power because high pressure air on the discharge side of the compressor screws restricts the screws from turning effortlessly. If the compressors' electrical drive motor is small enough it turns off when the unloaded state is met. In most cases compressors have electrical motors that are too large to

simply turn off. When an electric motor is switched on there is a large and sudden Amperage spike to overcome the motor's inertia. This large ampere spike prevents large electric motors from rapidly switching on and off because it causes heating in the windings that leads to premature failure. However, when the drive motor is run for an extended period of time unloaded, maybe 20 minutes, it turns off. Below, Figure 28, is a plot of percent input power versus percent capacity output.

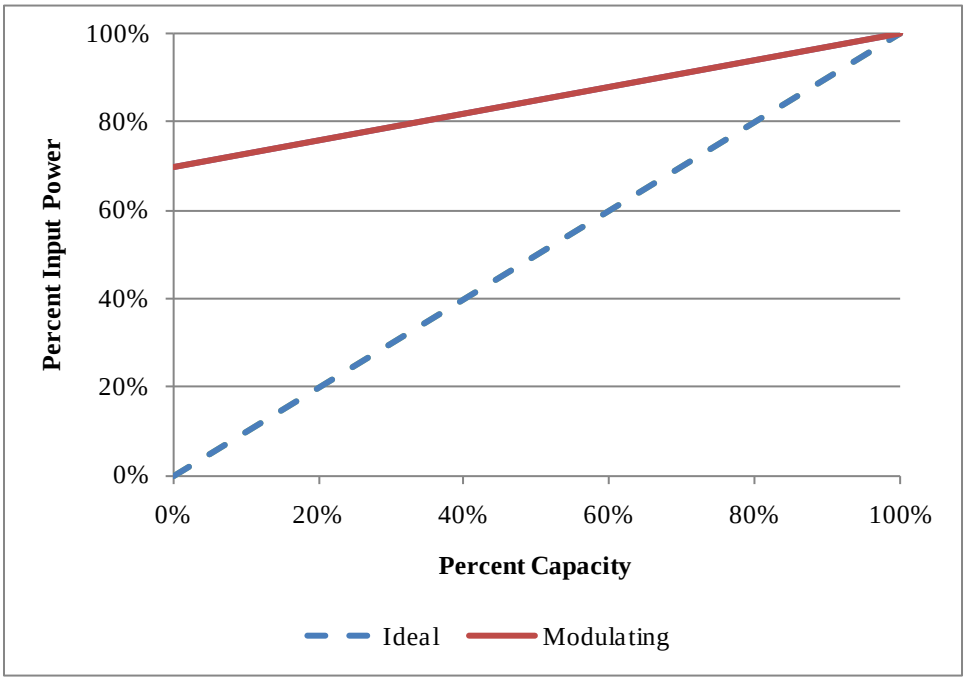

Figure 28: Oil Injected Screw Compressor Modulating Power vs. Capacity

The plot above gives the ability to derive a compressor load factor. CAGI defines load factor as "ratio of average compressor load to the maximum rated compressor load over a given

period of time". Using the plot above and basic algebra a load factor, excluding time dependency, for a modulating screw compressor is found as flows:

$$LF = 0.3\left(C_{process}/C_{rated}\right) + 0.7 \qquad \textit{(Eqn.4.1.1)}$$

Where,

LF	=	*Load Factor*
$C_{process}$	=	*Process capacity (SCFM)*
C_{rated}	=	*CAGI datasheet rated capacity (converted to SCFM)*

Knowing a process capacity, in terms of SCFM, allows a load factor to be calculated. The load factor is then used to determine the actual amount of electrical horsepower needed to compress a quantity of air for a process as follows:

$$P_{input} = LF \cdot P_{rated} \qquad \textit{(Eqn. 4.1.2)}$$

Where,

P_{input}	=	*Power required to compress a quantity of air (horsepower)*
P_{rated}	=	*CAGI total package input power (horsepower)*

To make suction throttling or modulating capacity controls more efficient, oil separator blowdown should be included. After the air and oil mixture travel through the compressor they travel to an oil separator, as shown in Figure 29.

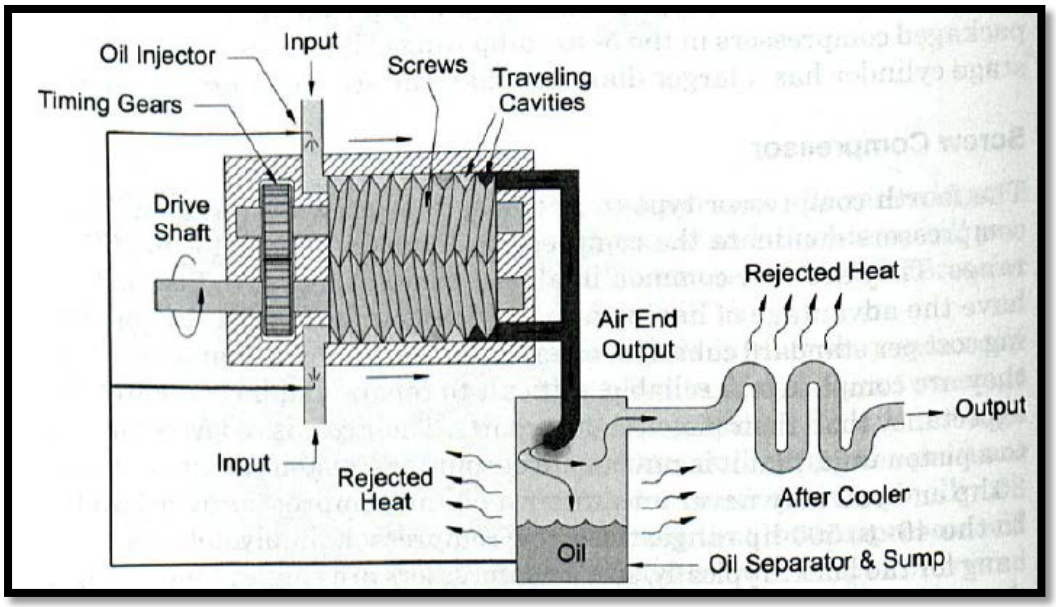

Figure 29: Screw Compressor, Oil Separator, and After Coolers (4)

The oil separator does as named, it separates oil and air. Separated oil is sent through a heat exchanger to cool, before returning to the compressor inlet. The air leaving the oil separator is sent to the facility's compressed air system.

Blowdown begins when the air compressor reaches an unloaded state. When the unloaded state is reached a valve in the oil separator is opened. The blowdown valve relieves pressure in the oil separator until a pressure ratio, across the compressor, is small enough to circulate a sufficient amount of oil through the compressor is met. This reduced pressure ratio, across the compressor, lowers the amount of power needed to turn the compressors screws and circulate oil. A non return valve (check valve) is usually installed in the compressed air

system after the oil separator to prevent compressed air from flowing back into the oil separator (6). Note compressor blowdown does not occur instantly. If the oil separator is blown down too fast oil will foam causing problems. "The time for blow down may range from 30 to 120 seconds and, unless there is sufficient air receiver/system volume, the compressor may reload before blow down has been completed and the reduce power requirement is realized" (20). When a compressor is unloaded and completely blown down power consumption is around 15-35% of full load power (5). That is 22.5 to 52.5 horsepower for the 150 horsepower example compressor given above. Below, Figure 30, shows visually the operation of a modulating compressor with blow down capabilities against a modulating compressor without blowdown capabilities.

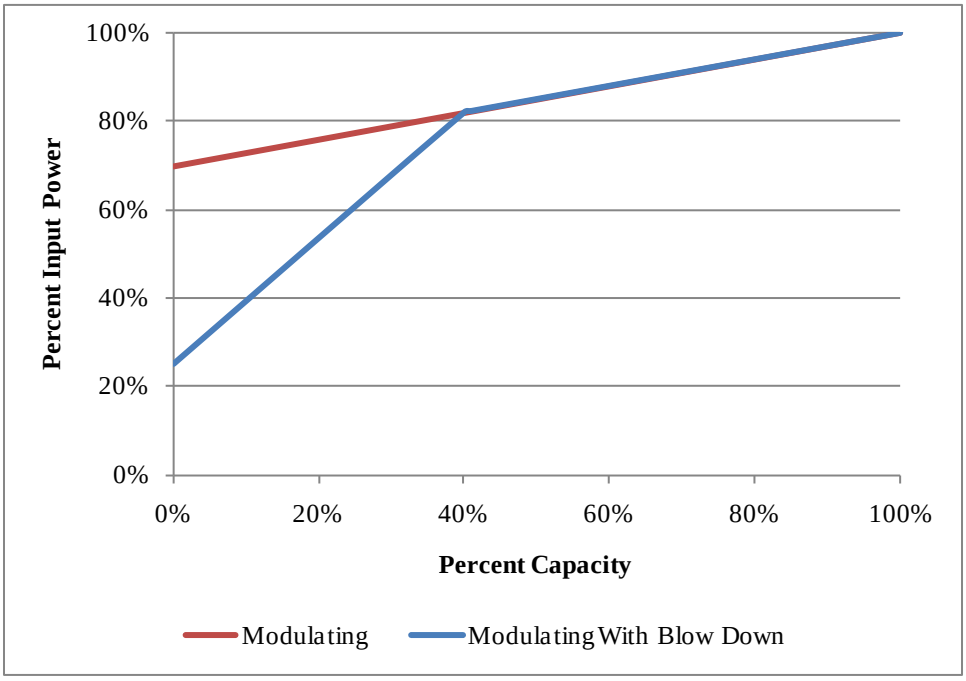

Figure 30: Oil Injected Screw Compressor with Blowdown Power vs. Capacity

The loaded factor for a modulating compressor with blowdown capabilities to supply a specific capacity of air is found with the following equation:

$$LF = 0.647\left({C_{process}}/{C_{rated}}\right) + 0.416 \qquad \textit{(Eqn. 4.1.3)}$$

Note power is calculated using the load factor above, equations 4.1.3, and equation 4.1.2 above.

A third method of capacity control is load/unload. Load/unload capacity control uses an inlet valve also, but the valve is either open or closed. A compressor utilizing this control scheme can be in one of the following states: loaded, unloaded, blown down, or off. Load/unload operation depends greatly on receiver size as seen in, Figure 31, below.

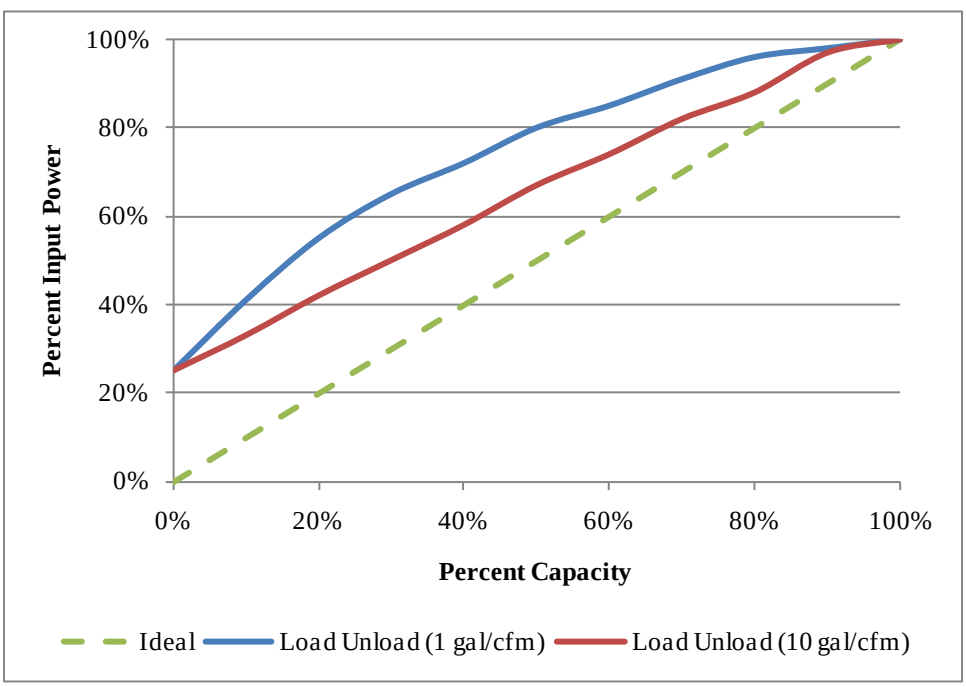

Figure 31: Load Unload Capacity Control

Gallons per CFM means gallons of storage per SCFM of are delivered. For example, if the compressor has a capacity of 690 SCFM then 6,900 gallons or storage is needed to achieve the results shown in Figure 31 for the 10 gal/cfm line. The load factors for a load unload compressor to supply a specific capacity at one gallon per CFM and 10 gallons per CFM is found with the following equations:

$$LF_{1\ gal/cfm} = 0.719\left({C_{process}}/{C_{rated}}\right) + 0.375 \qquad \textit{(Eqn. 4.1.4)}$$

$$LF_{10\ gal/cfm} = 0.772\left({C_{process}}/{C_{rated}}\right) + 0.265 \qquad \textit{(Eqn. 4.1.5)}$$

Another capacity control method, variable capacity control, is commonly done using two methods: sliding valves or poppet valves. "A valve is built into the rotary screw air compressor casing to control output to match demand by varying the displacement of the rotors. Rising discharge pressure causes the valve to be repositioned, progressively reducing the effective length of the rotors and delaying the start of compression. The inlet pressure and compression ratio remain constant so part-load power requirements are less than for inlet valve modulation. The normal capacity range is from 100 percent to 50 percent, followed by inlet valve modulation to zero capacity, or to 40 percent capacity, below which the compressor is unloaded" (20).

"Step control valves or poppet valves may be used on lubricant injected rotary air compressors to have an effect similar to slide, spiral, or turn valves, but with discreet steps of

percent capacity rather than infinitely variable positioning" (20). A plot of percent power versus percent capacity for a variable displacement compressor is shown below.

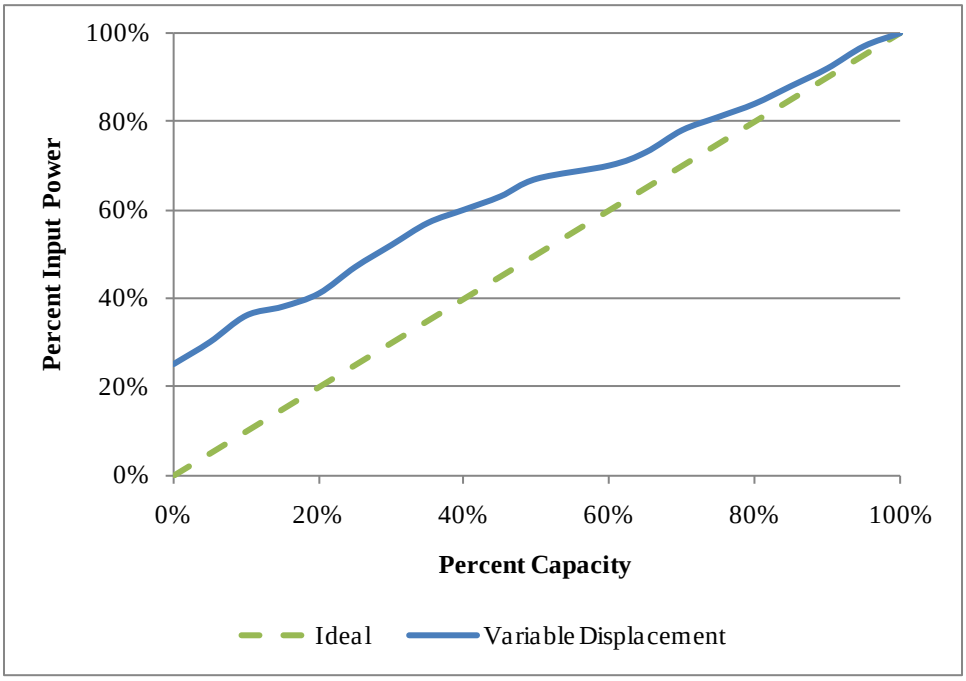

Figure 32: Variable Displacement Percent Power vs. Capacity

Load factor for a variable displacement compressor to supply a specific capacity of air is found with the following equation:

$$LF = 0.716\left(C_{process}/C_{rated}\right) + 0.283 \qquad \textit{(Eqn. 4.1.6)}$$

Speed control is the ideal because capacity is directly proportional to speed for a screw compressor. As mentioned earlier most air compressors are driven using a large alternating current (AC) electrical motor. A VFD is needed to adjust the speed of these motors.

Sometimes a variable frequency drive is also called a variable speed drive (VSD). If a facility's compressed air demand is unknown, purchasing a VFD is the most sensible solution for sizing a compressor. Below, Figure 33, it can be see that a VFD compressor is almost ideal.

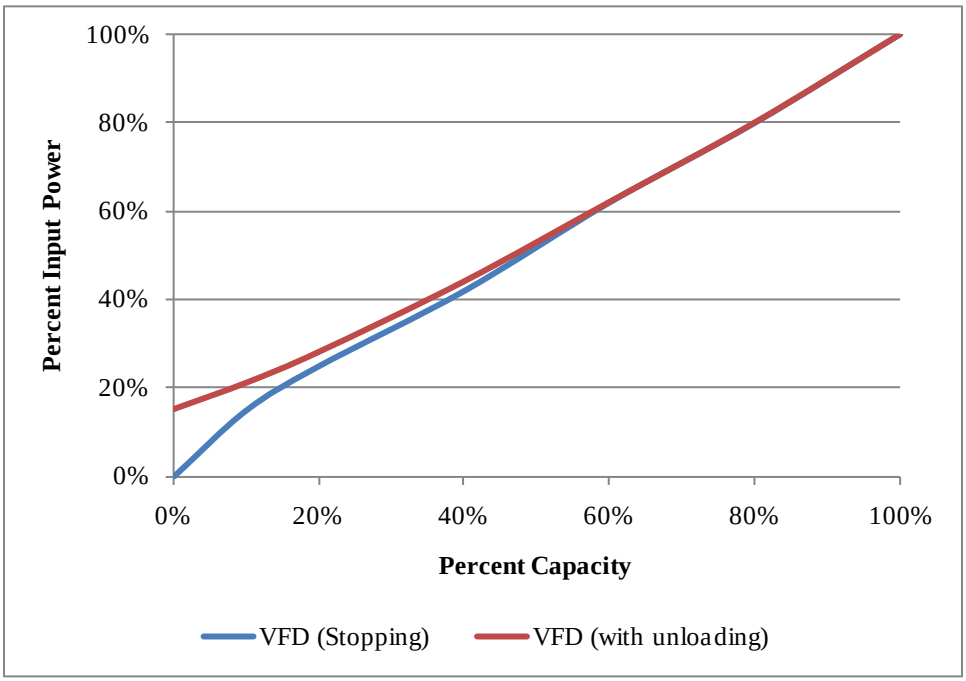

Figure 33: Power vs. Capacity Using a VFD Screw Compressor

The load factor for VFD compressors with unloading controls to supply a specific capacity of air is found with the following equation:

$$LF_{unloading} = 0.854\left(C_{process}/C_{rated}\right) + 0.122 \qquad \textit{(Eqn. 4.1.7)}$$

And the load factor for a VFD compressor with stopping controls is:

$$LF_{stopping} = 0.9677\left(C_{process}/C_{rated}\right) + 0.0343 \qquad \textit{(Eqn. 4.1.8)}$$

A VFD is not always the most efficient air compressor is terms of capacity per horsepower. There are some losses operating the VFD drive. Therefore a fully loaded modulating of load unload compressor is slightly more efficient than a VFD, because there are no losses with electricity flowing through the VFD control box.

Ideally a VFD compressor will be operated as a swing compressor. A swing compressor is used to supply air capacity to a varying air demand and a base compressor supplies capacity to a constant air demand. The plot below shows where the base loaded compressors will operate and where swing compressors will operate.

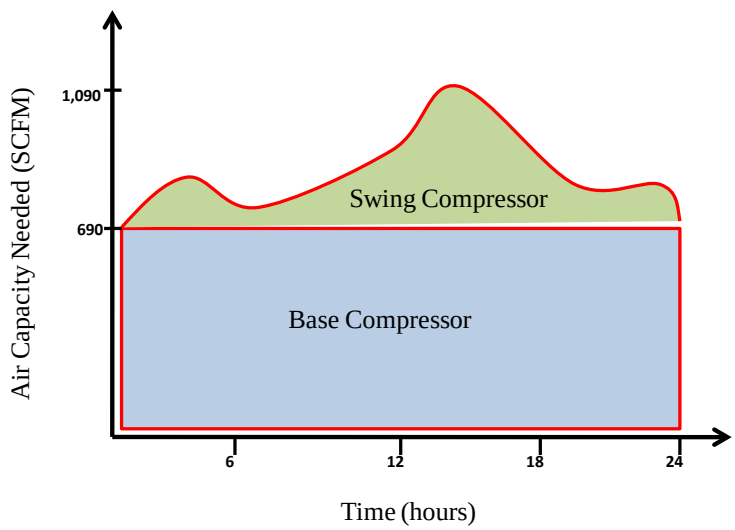

Figure 34: Swing Compressor Operation

When deciding which capacity control method to use it is very important to understand what compressed air is used for (i.e. hand tools or pneumatic controls for process). In any case compressor blowdown should be utilized, for load / unload operation correct storage or receiver sizing must be ensured, and VFD compressors should always be used as swing compressors. Finally, facilities that use multiple compressors should always base load their biggest compressor and utilize their smaller ones as swing compressors. This will make sure big compressors operate fully loaded, thus efficiently.

The plots above, Figure 28 through Figure 33, were created using data from Best Practice for Compressed Air book published by the U.S. DOE (20). Equations 4.1.1 through 4.1.8 were found using Excel's plot trend line with equation tool. All trend lines were assumed to be linear. The worst correlation was Figure 30 with a R^2 equivalent to 0.814.

4.2 Lower Compressor Output Pressure

Conceptually, the easiest way to lower the input power to a compressor is lower its pressure ratio. Anyone who has tried to blow up a child's air balloon can understand this concept. As discussed earlier our lungs are a simple air compressor. When blowing up a child's air balloon from atmospheric pressure to some elevated pressure, say three psig, it is fairly simple. Now imagine blowing up a child's water balloon instead. Water balloons are usually made with a thicker material than air balloons because they need to hold the weight of water inside them. Note the weight of water usually applies a higher pressure to the walls

inside a water balloon therefore they are built sturdier. That is why blowing up a water balloon is harder with your lungs; it requires higher pressure, say 10 psig, to expand the balloon's volume. This is why a person usually turns red in the face and struggles to blow up a water balloon using his or her lungs.

Air compressors usually have a fixed inlet pressure, atmospheric pressure. Thus lowering an air compressors discharge pressure effectively lowers its pressure ratio. There is plenty of literature showing reduction of a compressors pressure ratio lowers the amount of power required to compress a fluid or gas. In particular the DOE has released several compressed air tip sheets that suggest lowering a compressed air systems operation pressure. Lowering a compressed air systems operation pressure with, hopes of saving energy, is done by lowering the compressors discharge pressure. DOE tip sheets one, seven, eight, nine, ten, and eleven suggest lowering compressed air system operation pressure (21).

The DOE tip sheets do not provide a technical explanation to show lowering a compressor's discharge pressure saves input power and energy. A common component of all the theoretical equations derived in Chapter 2 is a ratio of discharge pressure (P_2) to inlet pressure (P_1) also known as pressure ratio. Since compressors are usually rated with an isentropic efficiency the isentropic power equation, equation 2.6.12 above, can be used to derive a percent power reduction due to lowering a compressor's discharge pressure. Percent power reduction (PR_p) due to lowering discharge pressure is derived analyzing a

compressor's initial performance (W_{inital}) versus proposed performance ($W_{proposed}$) at a lower discharge pressure as follows:

$$W_{inital} = m \cdot c_p \cdot (T_1 + 460) \cdot \left(\left(\frac{P_{2\_inital}}{P_1} \right)^{(k-1)/k} - 1 \right) \cdot C$$

$$W_{proposed} = m \cdot c_p \cdot (T_1 + 460) \cdot \left(\left(\frac{P_{2\_proposed}}{P_1} \right)^{(k-1)/k} - 1 \right) \cdot C$$

The percent power difference using the two equations above is:

$$Precnt\ Difference = 1 - \frac{W_{proposed}}{W_{inital}} \qquad \textit{(Eqn. 4.2.1)}$$

Using algebra equation 4.2.1 can be rearranged as follows:

$$PR_p = 1 - \frac{\left(\frac{P_{2\_proposed}}{P_1} \right)^{(k-1)/k} - 1}{\left(\frac{P_{2\_inital}}{P_1} \right)^{(k-1)/k} - 1} \qquad \textit{(Eqn. 4.2.2)}$$

Knowing the inlet pressure, initial discharge pressure, proposed discharge pressure, capacity control method, and hours of operation for an air compressor the annual energy savings can be predicted with the following equation:

$$ES = LF \cdot P_{rated} \cdot PR_p \cdot 0.746 \ \frac{kW}{hp} \cdot Hrs \qquad \textit{(Eqn. 4.2.3)}$$

Where,

ES	=	*Energy Savings (kWh)*
P_{rated}	=	*CAGI total package input power (horsepower)*
LF	=	*Compressor's load factor see Section 4.1*

PR_p	=	*Power reduction due to discharge pressure reduction (%)*
Hrs	=	*Compressors annual operation time (hours)*

4.3 Fix Air Leaks

Fixing air leaks in a compressed air system is a simple way to save energy, but in practice fixing air leaks is not commonly done well. Air leaks are easy to neglect because they do not generally affect production or critical maintenance; i.e. air leaks do not leave a mess like hydraulic lines that leak nor do they pose the threat of an explosion like a gas leak (4). Therefore air leaks are not commonly fixed and over time they become a big expensive problem. It is important to note that even a well-managed air leak program will have some air leaks, but if leaks are not controlled an estimated 20 to 30 percent of a compressor's output can be consumed by these leaks (20).

One way that air leaks can be a significant (but silent) issue is for the plant to have to purchase a larger compressor due to air leaks reducing its useful capacity. Take a new manufacturing facility that has sized their compressor correctly, i.e. their compressors output matches demand almost perfectly, as an example. If this manufacturer does not fix air leaks, after some time, the system's pressure will drop. When the system's pressure drops tools and processes will not operate correctly, thus production will be affected. The quickest fix to this problem may be purchasing a new compressor instead of taking time to fix all the air leaks. In the long run this is a terrible decision because capital is wasted purchasing a new

compressor and energy is wasted because the new compressor is oversized. If the manufacturer would have set up a simple leak detection program leaks would have been fixed as they occurred saving the facility money and energy.

A leak detection program can be as simple as having a couple of employees walking around a manufacturing facility once a month to detect leaks and fix them. Larger leaks can be heard by ear and smaller leaks can be found using soapy water. Soapy water placed on compressed air system components should bubble if a leak is present. The ear and soap method can be very time consuming therefore purchasing an ultrasonic acoustic leak detector is suggested. These detectors are simple to use and have an estimated training time of 15 minutes (20). According to the DOE's third compressed air tip sheet the most common problem areas, for are leaks, are:

1) Couplings
2) Hoses
3) Tubes
4) Fittings
5) Pipe joints
6) Quick disconnects
7) Filter regulator and lubricator
8) Condensate traps
9) Valves
10) Flanges
11) Packings
12) Thread sealants
13) Point of use devices

Conceptually fixing air leaks in a compressed air system to save energy is easy to understand, but estimating energy and power savings due to fixing an air leak is a difficult task. If a leak

exists then a volume of air that was compressed will escape out of the leak before the air is utilized for useful work. Deriving a theoretical equation to represent the energy and or power loss due to a leak in a compressed air line is very difficult to do. According to the *THERMODYNAMICS An Engineering Approach, Sixth Edition* the mass flow rate of air exiting a compressed air leak, in pound mass per second, is found with the following equation:

$$m = C_d \left(\frac{2}{k+1}\right)^{1/k-1} \frac{P_2}{RT_2} A \sqrt{kR\left(\frac{2}{k+1}\right) T_2} \qquad \textit{(Eqn. 4.3.1)}$$

This equation assumes compressed air exits a leak from line pressure to atmospheric pressure adiabatically, and is based on choked flow theory. Equation 4.3.1 above is difficult to use because it requires conversions from BTU's to foot pound force and conversions from pound force to pound mass. The conversions utilized are:

$$1\ BTU = 778\ lbf \cdot ft$$

$$1\ lbf = \frac{1 lbm \cdot 32.2 ft}{sec^2}$$

Equation 4.3.1 can be rewritten to include these conversions as follows:

$$m = Area \cdot C_d \left(\frac{2}{k+1}\right)^{1/k-1} \frac{P_2}{R_p(T_2+460)} \sqrt{25{,}052 \cdot kR_T \left(\frac{2}{k+1}\right)(T_2 + 460)} \qquad \textit{(Eqn. 4.3.2)}$$

Where,

Area	=	area of leak (ft^2)
C_d	=	discharge coefficient is 0.65 if actual data is unknown
R_p	=	gas constant for air in units of (0.3704 psia-ft^3 / lbm-R)

25,052 = conversion constant in units of (lbm-ft^2 / BTU-sec^2)

k = specific heat ratio of air (1.4)

P_2 = compressor discharge pressure (psia)

T_2 = temperature of air at leak (ºF)

R_T = gas constant for air in units of (0.06855 BTU / lbm-R)

Equation 4.3.2 is quite cumbersome. Assuming the leak is in the shape of a perfect circle, and substituting in for known properties, the equation can be reduced to find the mass flow rate of air in pounds per minute as follows:

$$m = 25.07 \cdot D^2 \cdot C_d \cdot \frac{P_2}{\sqrt{T_2+460}} \qquad \textit{(Eqn. 4.3.3)}$$

In equation 4.3.3 the variable *D* represents the diameter of a leak in inches. After the mass of air leaving a leak is calculated some practical knowledge about compressors is needed before a power reduction, due to fixing air leaks, can be calculated. Practical knowledge is need because two simple mistakes can be made. First, once the mass of air exiting a leak is known it seems logical to calculate how much power is required to compress this mass. This can be done simply using one of the equations shown in Chapter 2. The problem with this method is fixing air leaks causes a compressor to run unloaded for longer periods of time. As discussed above, in Section 4.1, an unloaded compressor still consumes power. Therefore using the mass of air exiting a leak and one of the equations shown in Chapter 2 will over predict energy savings unless the compressor is a VFD.

In reality a reduction in power consumption due to fixing air leaks is the difference in the compressor power draw due to an air leak load minus the compressor unloaded power draw. To calculate these savings the volumetric flow rate of air, at standard conditions, escaping a leak needs to be calculated. Knowing the mass flow rate of air escaping through a leak and utilizing the ideal gas law the volumetric flow rate of air at standard conditions can be calculated as follows:

$$\dot{V}_{air\ leak} = \frac{m\ (460+T_s)R_p}{P_s} \qquad \textit{(Eqn. 4.3.4)}$$

Where,

$\dot{V}$ = *Volumetric flow rate (SCFM)*

T_s = *CAGI standard air temperature (68ºF)*

R_p = *gas constant for air in units of (0.3704 psia-ft^3 / lbm-R)*

P_s = *CAGI standard pressure (psia)*

A generic table of air leaks at different system pressures and leak sizes is shown below. Note this table assumes line temperature is 75ºF.

Table 3: Generic Leak Rates

Pressure (psig)	1/64" Leak Diameter Flow Rate (SCFM)	1/32" Leak Diameter Flow Rate (SCFM)	1/16" Leak Diameter Flow Rate (SCFM)	1/8" Leak Diameter Flow Rate (SCFM)	1/4" Leak Diameter Flow Rate (SCFM)	3/8" Leak Diameter Flow Rate (SCFM)
70	0.20	0.79	3.14	12.6	50.3	113
80	0.22	0.88	3.52	14.1	56.2	127
90	0.24	0.97	3.89	15.5	62.2	140
100	0.27	1.06	4.26	17.0	68.1	153
125	0.32	1.30	5.19	20.7	83.0	187

Once the SCFM of air exiting a leak is known a compressor's loading due to this leak can be found by replacing $C_{process}$ with $\dot{V}_{air\ leak}$ in one of the load factor equations shown above: equations 4.1.1, 4.1.3, 4.1.4, 4.1.5, 4.1.6, 4.1.7, and 4.1.8. As an example the load factor equation due to air leak in a modulating compressor with blow down capabilities is expressed below:

$$LF_{air\ leak} = 0.647\left(\dot{V}_{air\ leak}/C_{rated}\right) + 0.416$$

A compressors unloaded load factor needs to be found to calculate power savings due to fixing air leaks also. The unloaded load factor is found by replacing $C_{process}$ with zero SCFM, because a compressor produces zero SCFM when unloaded. Again the unloaded load factor for a modulation compressor with blow down is used as an example as follows:

$$LF_{unloaded} = 0.647\left(0/C_{rated}\right) + 0.416$$

$$LF_{unloaded} = 0 + 0.416$$

$$LF_{unloaded} = 0.416$$

With the load factor due to an air leak and a unloaded load factor the power reduction due to fixing air leaks is found:

$$PR_{air\ leak} = (LF_{air\ leak} - LF_{unloaded}) \cdot P_{rated} \qquad \textit{(Eqn. 4.3.5)}$$

Annual energy savings due to fixing air leaks can be found also:

$$ES_{air\ leak} = PR_{air\ leak} \cdot 0.746\ \frac{kW}{hp} \cdot Hrs \qquad \textit{(Eqn. 4.3.6)}$$

The second easy to make mistake involves the volumetric flow rate of air escaping through a leak and compressor data. It is common to take compressor name plate data or CAGI data and find a compressors efficiency rating in terms of power per SCFM, i.e. kW/SCFM or hp/SCFM. Once this efficiency is known it seems reasonable to multiple it by the volumetric flow of air leaving a leak. To make matters worse the DOE suggest using this method in one of their compressed air tip sheets. The DOE's compressed air tip sheet for air leaks, number three, suggest using the following equation:

Energy savings = # of leaks x leakage rate (cfm) x kW/cfm x # of hours *(Eqn. 4.3.7)*

Note for a VFD compressor this method is assumed to work well, but VFD units are not common.

Using the air leak equations, shown above, the annual energy savings for fixing a single 1/8 inch air leak in a 125 psig compressed air system utilizing a 150 horsepower compressor are found for different capacity control methods.

Table 4: Estimated Air Leak Energy Savings

Capacity Control Method	$LF_{unloaded}$	$LF_{air\ leak}$	Power Savings (horsepower)	Annual Energy Savings (kWh)
Modulating Without Blowdown	0.700	0.709	1.64	10,726
Modulating With Blowdown	0.416	0.435	3.54	23,132
Load Unload at (1 gal/SCFM)	0.375	0.397	3.93	25,706
Load Unload at (10 gal/SCFM)	0.265	0.288	4.22	27,601
Variable Displacement	0.283	0.305	3.92	25,599
VFD with Motor Stopping	0.034	0.063	5.29	34,598
VFD with Motor Unloading	0.122	0.148	4.67	30,533

The estimated savings shown in Table 4 are made assuming the 150 horsepower compressor has an output capacity of 690 SCFM, total input power of 182 horsepower, operates 8,760 hrs per year, and a line temperature of 75ºF.

A simple way to determine the amount of air leaks present is a compressed air system is to conduct an experiment. To conduct this experiment the total volume of a compressed air system including all parts, i.e. hoses or lines and receiver tanks, is needed. A pressure gauge downstream of the compressor's receiver and a stopwatch are also needed. To conduct the experiment the compressed air system should be brought to full operating pressure. Once the system is up to operating pressure the compressor should be shut down and the amount of time it takes the compressed air system to drop to half of its operating pressure should be recorded. Once this is completed the following equation can be used to find the volumetric flow rate of all air leaks:

$$\dot{V}_{All\ air\ leaks} = \frac{V \cdot \left(P_2 - \frac{P_2}{2}\right)}{Tx14.7} \times 1.25 \qquad \textit{(Eqn. 4.3.8)}$$

Where,

$\dot{V}_{All\ air\ leaks}$	=	*flow rate of all air leaks in CFM of free air*
V	=	*volume of entire compressed air system (ft^3)*
P_2	=	*compressed air system operation pressure (psig)*
T	=	*time (minutes)*

Note CFM of free air is assumed to be equivalent to SCFM and the 1.25 multiplier corrects for a reduced leakage rate as system pressure drops. The suggested experiment and equation shown above, equation 4.16, where found using the *BEST PRACTICES for COMPRESSED AIR SYSTEMS* book (20).

4.4 Utilize Compressor Waste Heat

When air is compressed heat is generated. In fact it is estimated that 80 to 93 percent of the electrical energy consumed by an air compressor turns into heat (20). Referring back to Chapter 2 this can be seen theoretically. Chapter 2 shows the more heat that is rejected, while compressing air, the more efficient the compression process. In fact if air is compressed isothermally then 100% of the energy put into the compressor is rejected in the form of heat. Unfortunately this cannot be done in the real world, but compressors do reject as much heat as possible to be efficient as possible. Since most the electrical power consumed by an air compressor turns into heat, which is rejected to atmosphere, air

compressors are commonly considered inefficient but if this heat is captured and utilized air compressors can obtain an overall efficiency or nearly 100%.

Oil flooded screw compressors inject an oil to absorb heat, during the compression process, and then reject this heat using an air to oil heat exchanger or oil to water heat exchanger. Hot oil travels into the heat exchanger and ambient air is forced across the heat exchanger to cool the oil. Potentially, this energy could be utilized for space heating in the winter months. This could be done by sending the hot air off a heat exchanger into a facility (via. duct work and maybe an additional fan) where space heating is needed. Savings will be seen by a reduction in energy (probably natural gas) used for space heating. A damper should be installed in the duct work so heat is rejected to atmosphere when space heating is not needed in the facility's space. The damper could be manually operated or have a simple control system that works off a thermostat inside the facility. Space heating is not needed annually, maybe 5 months of the year maximum, depending on geographical location. Therefore if hot air needed for a continuous process, such as inlet air to an ovens burner, energy savings will be increased because waste heat will be utilized 12 months per year instead of 5 months.

Some air compressors use water to cool their lubricant. Heat absorbed by the water is eventually dumped to atmosphere using a cooling tower. A cooling tower requires energy to operate. If hot water off the air compressor is used for process (saving energy) a reduction in cooling tower operation is seen also saving additional energy. Some possible opportunities

are central heating or boiler systems, industrial cleaning processes, plating operations, heat pumps, laundries, any application where hot water is needed (20).

Before installing a waste heat recovery system (i.e. ducting for space heating or piping and a heat exchanger to preheat boiler makeup water) the amount of energy being wasted needs to be tabulated. To be conservative it is suggest to estimate the potential amount of waste heat available for space heating or process is 80% of the energy consumed by the compressor. Therefore the annual amount of waste heat available from an oil flooded screw compressor for space heating is estimated as follows:

$$ES = LF \cdot P_{rated} \cdot Hrs \cdot 80\% \cdot \frac{Mon}{12} \cdot C \qquad \textit{(Eqn. 4.4.1)}$$

Where,

ES	=	*Energy Savings (BTU/yr.)*
LF	=	*Load Factor (see Section 4.1 for values)*
P_{rated}	=	*CAGI total package input power (horsepower)*
Hrs	=	*Compressors annual operation time (hours / yr.)*
Mon	=	*# of months space heating is required (5 months depending location)*
C	=	*Conversion constant (2,545 BTU / hp-hr)*

Equation 4.4.1 can be used to estimate about the amount of energy available to heat water also. If water is used annually then the number of months used should be 12.

4.5 Utilize Outside Air

Most energy engineers have heard using outside air in an air compressor should save energy because the average industrial floor temperature is higher than the average outside air temperature. This is especially true for manufacturers with air compressors located close to heat generating equipment such as boilers. A lower inlet temperature should lower compressors power consumption because as air cools it becomes denser. This means the fluid's specific volume decreases as it cools. The goal of a compressor is to compress or shrink a volume of air. Therefore if a mass of air entering a compressor is cold the machinery used to compress the air will not work as hard as this same mass of air at an elevated temperature. Note a compressor's power consumption is a function of mass entering the compressor not volume therefore if cooler intake air is used the compressor could consume more power if the volumetric flow rate of air is not reduced. In the case were volume is held constant and not mass the air compressor will produce compressed air more efficiently in terms or capacity per horsepower, SCFM / hp. In effect, if the compressor consumes more power with a lower air temperature, the compressor will not have to run as long because the compressor will charge or fill the compressed air system faster. Thus the compressor will run for shorter periods of time and may reduce the amount of energy the compressor consumes. If the mass flow rate of air entering a compressor is held constant, by reducing the volumetric flow rate of air entering the compressor, then an instantaneous power reduction should be found when lower temperature air is used compared to high temperature air.

There are reliable articles that suggest using outside air in industrial air compressors. The Department of Engineer (DOE) has created compressed air tip sheets to help manufactures keep their air compressors operating efficiently. Tip sheet number fourteen is titled *Effect of Intake on Compressor Performance* (11). This tip sheet suggest: locating the compressor intake outdoors to keep ambient intake temperature to a minimum to lower energy consumption and maintenance work. The tip sheet also suggests locating the intake away from humid environments. Note the DOE tip sheet has a disclaimer stating savings are less pronounced for lubricant-inject rotary screw compressors. This claim is made because the incoming air is mixed with hot oil. Another source had a similar statement: "Variation of ambient temperature will have a related effect on discharge temperature in any air compressor. The extent will depend on whether cooling during compression is taking place due to oil or water injection. Without this the change will be significant, with it will be minor." (6).

The 1997 ASHRAE Transaction article titled "*Energy-Saving Opportunities for Positive-Displacement Air Compressors*" evaluates whether using outside air in an air compressor saves energy or not. The article discusses how it is common knowledge to energy auditors that using outside air in an air compressor should save energy. The article concludes that air compressors do not save instantaneous power by utilizing outside air but compressor run operation times are reduced, thus, lowering overall energy consumption. Unfortunately the ASHRAE article does not mention if any of the four air compressors tested were oil flooded

rotary screws. Two other sources had similar knowledge (i.e. they mention air temperature affects compressor performance but no mention of oil injection or not). These sources are *Compressor Handbook: Principles and Practice* (5) and *A Practical Guide To Compressor Technology* (22).

Assuming a compressor is a isentropic process the power reduction compressing are at an average outdoor temperature of 70ºF versus an average plant indoor temperature of 90ºF is found simply using equation 2.6.13 above. Assuming mass flow is fixed at 3,074 pound mass of air per hour, about 690 SCFM, inlet pressure is assumed to be 14.7 psia, and outlet pressure is assume to be 139.7 psia the power to compress air at 90ºF is 144 horsepower. Using the same assumptions the power to compress air at 70ºF is 139 horsepower. That is a power reduction of 3.47%. This power reduction may appear to be negligible, but if the compressor operates for 8,760 hours per year fully loaded the savings are significant, around 32,675 kWh per year.

A simple equation to find the percent power reduction due to using outside air in an oil free compressor can be found as follows:

$$PR_{OA} = 1 - \frac{\mathrm{W}_{outdoor}}{\mathrm{W}_{indoor}}$$

$$PR_{OA} = 1 - \frac{(T_{outdoor} + 460)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}{(T_{indoor} + 460)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}$$

$$PR_{OA} = 1 - \frac{(T_{outdoor} + 460)}{(T_{indoor} + 460)} \qquad \textit{(Eqn. 4.5.1)}$$

Where,

PR_{OA}	=	*power reduction due to outside air (%)*
$T_{outdoor}$	=	*average outdoor air temperature (ºF)*
T_{indoor}	=	*average indoor air temperature (ºF)*
P_1	=	*compressor inlet pressure (psia)*
P_1	=	*compressor outlet pressure (psia)*
k	=	*ratio of air's specific heats, 1.4 (unitless)*

Note the equation above assumes mass flow rate of air entering a compressor is held constant, using a control system. Knowing the percent power reduction the annual energy savings for a fully loaded compressor can be found using the equation below.

$$ES = P_{rated} \cdot LF \cdot PR_{OA} \cdot 0.746 \, \frac{kW}{hp} \cdot Hrs$$

Where,

ES	=	*Energy Savings (kWh)*
P_{rated}	=	*CAGI total package input power (horsepower)*
LF	=	*Compressor's load factor see Section 4.1*
PR_{OA}	=	*Power reduction due to utilizing outside air (%)*
Hrs	=	*Compressors annual operation time (hours)*

Chapter 5 – Utilizing Outside Air for Compressor Inlet Temperature Analysis

5.1 Hypothesis and Experimental Overview

The objective of this thesis is to experimentally determine if utilizing outside air in an oil flooded screw compressors, used for industrial purposes, affects compressor energy consumption due to changing inlet air temperature. It is common knowledge that utilizing outside air instead of warm plant air in air compressor should save energy. This common knowledge does not define what type of compressor though. According to the isentropic compression process using outside air in an oil free compressor should save energy, sec Section 4.5 for details. It is hypothesized that changing inlet air temperatures in oil flooded screw compressors will have a slight affect on compressor performance (i.e. energy consumption) at best. This hypothesis is made because the compressor's coolant, injected oil, is maintained at specific temperatures. The oil represents most of the mass because of its higher density, therefore the compressor's oil temperature is presumed to be the determining factor in compressor performance.

This hypothesis can be determined theoretically. Compressor efficiency is usually related to an isentropic compression process, so it seems equation 2.6.13 will sufficiently estimate an

oil flooded screw compressor's power consumption. However it does not, the biggest reason is equation 2.6.13 uses an isentropic relation (equation 2.6.8) to determine the temperature of air leaving the compressor. For an oil flooded screw compressor this is not the case. The air and oil exiting the compressor is controlled to a specific temperature, usually around 212ºF (6). To get a more accurate view of how an oil flooded screw compressor operates the following diagram can be utilized.

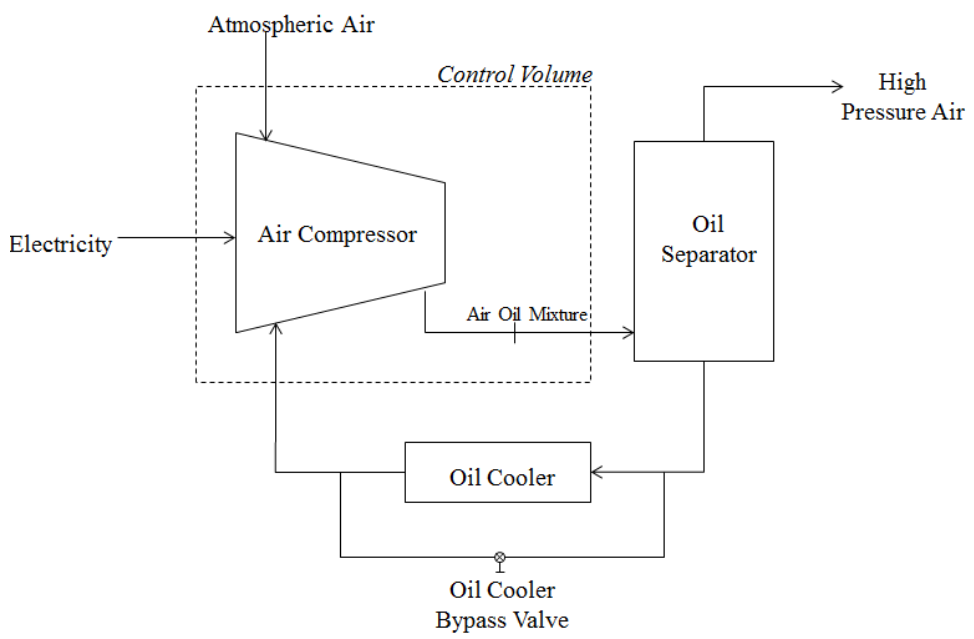

Figure 35: Oil Flooded Screw Compressor Diagram

As Figure 35 shows electricity or work, air, and oil enter the compressor. A mixture of oil and air exit the compressor. Using thermodynamics an energy balance across the oil flooded screw compressor control volume is derived as follows:

$$\Delta E_{CV} = 0$$

$$E_{in\ toCV} = E_{out\ of CV}$$

$$W_{in} + m_{air}\,(h_{air\ in}) + m_{oil}\,(h_{oil\ in}) = m_{air}\,(h_{air\ out}) + m_{oil}\,(h_{oil\ out})$$

Assuming a steady flow device the mass entering must equal the mass leaving, therefore:

$$W_{in} = m_{air}\,(h_{air\ out} - h_{air\ in}) + m_{oil}(h_{oil\ out} - h_{oil\ in})$$

Since the air and oil do not change phases specific heats can be used instead of enthalpies.

$$W_{in} = m_{air}\,C_{p\ air}(T_{air\ out} - T_{air\ in}) + m_{oil}C_{p\ Oil}(T_{oil\ out} - T_{oil\ in}) \qquad \textit{(Eqn. 5.1.1)}$$

Where,

W_{in}	=	*rate of electrical energy or power required to compress air (BTU/hr)*
m_{air}	=	*mass flow rate of air (lbm/hr)*
$T_{air\ in}$	=	*temperature of air entering compressor (ºF)*
$T_{air\ out}$	=	*temperature of air leaving compressor (ºF)*
$c_{p\ air}$	=	*specific heat of air (0.24 BTU/lbm-ºF)*
m_{oil}	=	*mass flow rate of oil (lbm/hr)*
$T_{oil\ in}$	=	*temperature of air entering compressor (ºF)*
$T_{oil\ out}$	=	*temperature of air leaving compressor (ºF)*
$c_{p\ oil}$	=	*specific heat of mineral oil (0.4 BTU/lbm-ºF)*

Compressor specifications were found for the oil flooded screw compressor studied in this work but they are not public information. Therefore the following temperatures are rules of thumb instead of actual manufacture data. Oil and air exit temperature are assumed to be 194ºF (22). Air inlet temperature is assumed to be 68ºF in accordance with CAGI. The oil

injection temperature is assumed to be 140ºF (20). Mineral oil properties are used for the calculation. Assuming the compressor draws in 690 SCFM of air, the mass flow rate into the compressor is around 3,105 pound mass of air per hour. Oil flow rates are usually between 10 to 20 gallons per minute per 100 horsepower (5). Assuming the oil's volumetric flow rate is around 30 gallons per minute its mass flow rate is 13,716 pound mass per hour. The estimated power to operate a compressor like this is:

$$\mathrm{W}_{in} = 3{,}105\frac{lbm}{hr}\cdot\frac{0.24BTU}{lbm\,°F}\cdot(194-68)°F + 13{,}716\frac{lbm}{hr}\cdot\frac{0.4\,BTU}{lbm\,°F}\cdot(194-140)°F$$

$$\mathrm{W}_{in} = 93{,}895\,\frac{BTU}{hr} + 296{,}266\frac{BTU}{hr}$$

Multiplying each side by the conversion constant, 2,545 BTU = 1 hp-hr, the following is found:

$$\mathrm{W}_{in} = 37\,hp + 116\,hp$$

$$\mathrm{W}_{in} = 153\,hp$$

This means 76% of the electrical horsepower put into the air compressors is absorbed by the oil and rejected as heat.

To determine if this hypothesis is correct an industrial oil flooded screw compressor that uses outside air had to be found. Once a compressor was found particular parameters needed to be data logged optimally for a year. One year was desired so the greatest changes in air temperature could be evaluated. Fortunately, a very gracious manufacturing facility, located in western North Carolina, allowed their air compressor to be data logged for this

experiment. Unfortunately, there were unknown problems with the compressor which delayed the experiment's anticipated start date, early January 2011. Therefore data logging did not begin until March 23, 2011. To make matters worse the compressor began to operate erratically in May and failed. Therefore data from May through June is scarce.

On June 30, 2011 the H22 energy logger inside the compressor's enclosure melted. At this point it was decided to stop recording data and begin analyzing it. This decision was made because the facility began utilizing their backup compressor, which utilized inside plant air, for process and it would be an extended period of time before the main compressor started operating again. Later it was determined the compressor failed due to a dirty oil/coolant, cooler. According to plant personnel the compressor is operating well today.

5.2 The Compressor and Operating Parameters

A manufacturing facility in western North Carolina agreed to our request for their air compressors to be data logged for this experiment. Their air compressor is an Ingersoll Rand, model number R110IU-A-125, 150 horsepower oil flooded, contact cooled, rotary screw compressor. This is a packaged air compressor meaning, screws, housing, oil separator, oil cooler, after cooler, moisture separator and some other parts are packaged in one enclosure as seen in Figure 36.

Figure 36: Packaged Air Compressor (23)

Most the parts inside the compressor enclosure can be found in *Appendix B.* CAGI has verified the performance of this compressor model. The following parameters can be found using the CAGI data sheet and simple kW to horsepower (hp) conversions, note the actual CAGI data sheet is located in *Appendix A.*

Table 5: CAGI Highlights in Standard Units

Total Package Input Power (hp)	182
Nameplate Power (hp)	150
Drive Motor Efficiency	95%
Capacity at full Load Operating Conditions (acfm)	690
Fan Motor Nameplate Rating (hp)	7.5
Fan Motor Efficiency	90.5%
Unloaded Package Power (hp)	59.5
Rated Capacity and Full Pressure at Full Input Power (acfm/hp)	3.79

The CAGI data sheet defines acfm, actual cubic feet per minute, as: “ACFM is actual cubic feet per minute at inlet conditions”. Note CAGI performs their test at standard conditions meaning their test are done with a compressor compressing air at 68ºF, zero percent relative humidity, and 14.5 psia. Since the CAGI data sheet states ACFM is at inlet conditions its assumed 690 ACFM is about 690 SCFM.

The compressor data logged for this experiment is utilized at a manufacturing facility that coats textile products. Compressed air is used for pneumatic controls and some hand tools. The compressor has the ability to control capacity by modulating or utilizing load/unload control. Note both control schemes have oil separator blowdown capabilities. Plant personal said they believe the compressor is controlled by two pressure set points. Once the air pressure reaches 125 psig the compressor begins unloading until pressure drops to 115 psig. Once 115 psig is met the compressor starts loading until a pressure of 125 psig is met. This cycle repeats twenty four hours per day seven days per week. If pressure drops to 100 psig then an additional compressor comes online to help with air demand. This control scheme resembles load/unload control but the data suggests the compressor operates in a modulating fashion. The compressor is wired in a dead leg configuration; this configuration is discussed in Chapter 3. Power distributed to the compressor is three phase 480 volts. The physical compressor packaged is located outside in the parking lot.

5.3 Measurement Procedure

Originally energy, in units of kWh's, consumed by the compressor was going to be measured with a WattNode meter counting pulses from a kWh meter. Unfortunately the compressors connection box is rather small and fear of creating a short inside the box arose when attempting to wire the WattNode Pulse Output kWh transducer inside the box. However, the compressors apparent current was measured instantaneously every minute using current transformers, CT's. To be specific Magnelab, part number SCT-1250-200, 200 amp CT's were utilized. CT's were placed around wires inside the compressors control panel. Only two of the three legs had CT's because of the compressor's dead leg wiring. CT's operate by reading a magnetic field induced by current flowing through a wire. This magnetic field then induces a smaller current and voltage in the CT. The smaller current is recorded and then a multiplier is used to compute the actual current traveling through the compressors wire. The CT's utilized are shown in Figure 37 below.

Figure 37: Current Transformer (24)

Compressor inlet air temperature was measured every minute by placing a sensor inside the compressors air intake filter. The sensor used is Onset's 12-Bit Temp Smart Sensor with a six meter cable: model S-TMB-M006. This sensor used is shown in Figure 38 below.

Figure 38: Inlet Temperature Probe (24)

Compressed air flow was measured using CDImeter, model number CDI 5400, compressed air flow meter. This meter has a flow range from 7 to 700 SCFM. The flow meter was installed on a three inch steel pipe 20 pipe diameters downstream of the air dryer and 10 pipe diameters up stream of any obstruction. CDImeter's flow meter measures flow by keeping one probe hotter than the other. Then the probe calculates the amount of heat required to keep it hot. When the heat rate is known mass velocity of the air can be found and the flow of air traveling through a pipe can be determined. The sensor does all the calculations internally. The flow meter is shown in Figure 39 below.

Figure 39: Flow Meter (24)

The flow meters output is in a pulse fashion. Meaning, the meter produces a pulse every time 0.2 standard cubic foot air travels through it. The logger recorded the number of pulses accumulated every minute. Therefore if 500 pulses were recorded in a minute then 100 standard cubic feet of air traveled through the pipe in that minute on average. Note flow meter had to be installed after the air dryer to prevent contamination, per manufacturer's requirement.

The CTs, flow meter, and temperature meter were connected to HOBO® Energy Loggers, model H22-001. These loggers are very convenient because all measurements or data can be downloaded from one source. After data was downloaded from the H22 loggers it was exported to Excel for data analysis. A picture of a H22 logger is shown below in Figure 40.

Figure 40: H22 Energy Logger (24)

Relative humidity of air inside the compressor's enclosure was recorded. Onset's HOBO U12 Temperature/Relative Humidity/Light/External Data Logger – U12-012 was used to record relative humidity. Note all loggers, inlet temperature, current, flow, and relative humidity were synchronized to record data at the exact same moment. The data logger used to record relative humidity is shown below in Figure 41

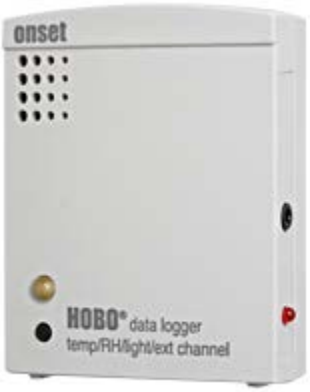

Figure 41: RH Meter (24)

The following diagram shows the compressor lay out and locations of measurement devices.

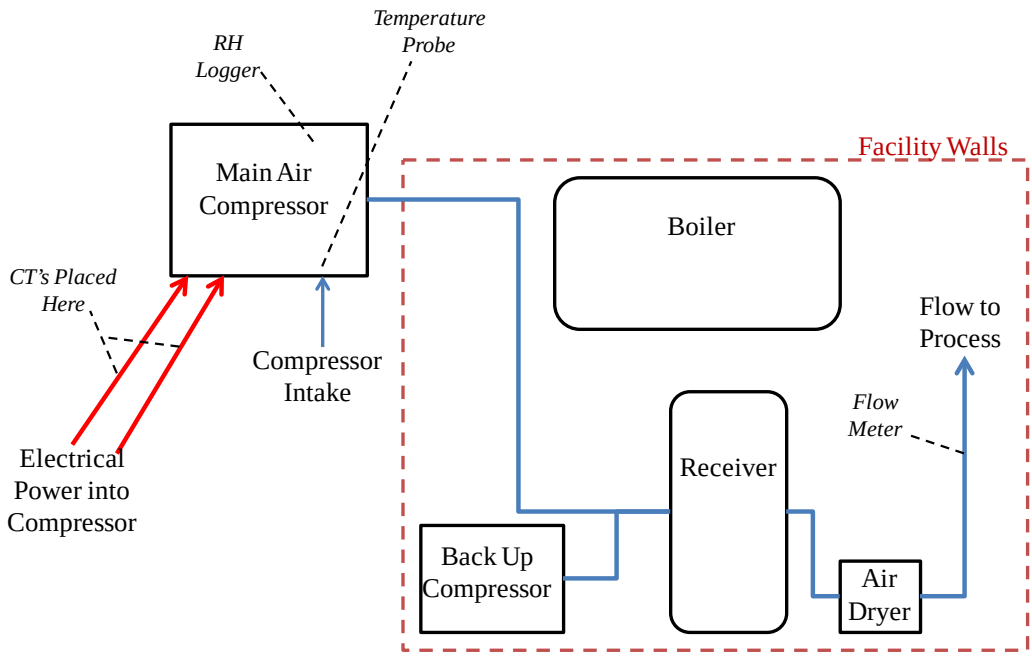

Figure 42: Data Measurement Diagram

5.4 Data Analysis

As discussed, the following compressor parameters were measured: current or Amperes, capacity or average cubic feet or air per minute, inlet temperature, and relative humidity of surrounding air. Compressor current, an assumed power factor of 90%, and equation 3.2.3 were utilized to calculate instantaneous input horsepower. Capacity, in terms of cubic feet per minute, was found by taking the pulse count, raw data, and dividing it by five. Temperature data was recorded in ºF therefore no adjustments were needed. Relative

humidity was recorded directly too. At the end of the experiment 634,040 data points were recorded.

5.4.1 Weekly Data Analysis

Around 141,120 rows of data were recorded for this experiment. Excel is limited to plotting 32,000 rows of data. Therefore it was decided to plot data by weeks consisting of 10,080 rows of data. Data was recorded every minute therefore 10,080 rows of data is equivalent to 168 hours or 7 days of data collection. Plotting data week by week also organizes data and allows potential trends to be seen.

Data, excluding relative humidity, recorded for the first week of this experiment, 3/23/11 through 3/30/11, are shown visually in Figure 43.

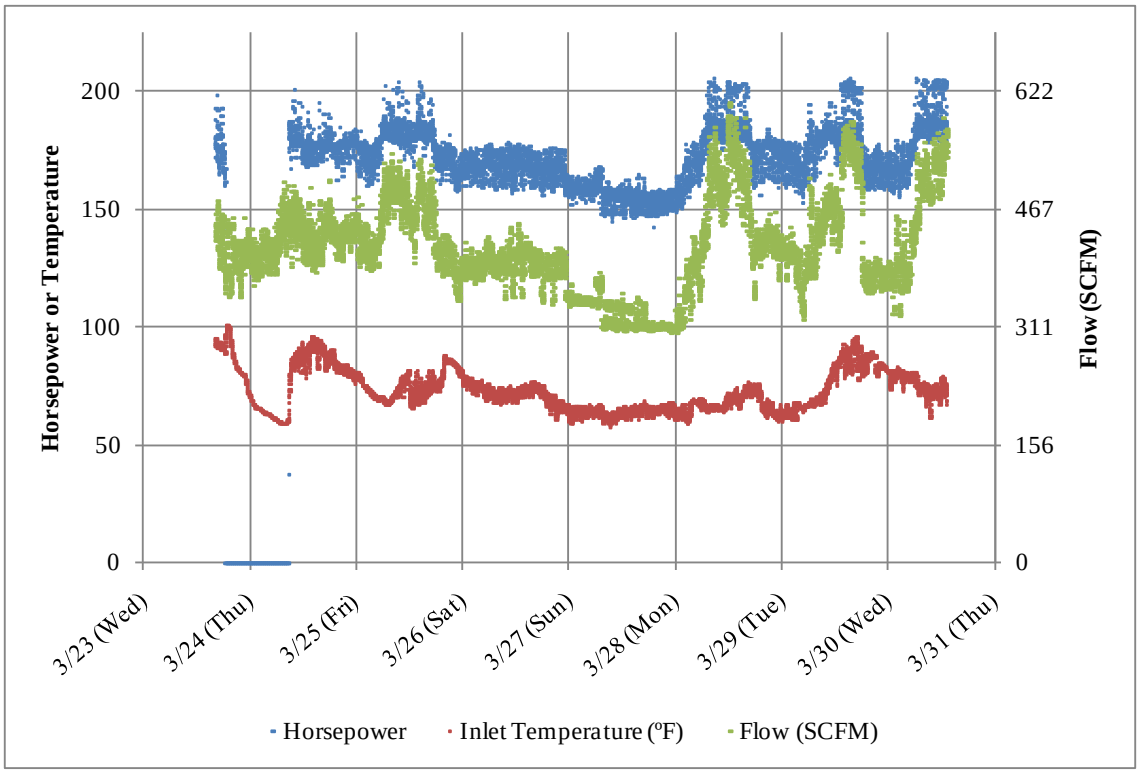

Figure 43: First Week Compressor Data

Horsepower is represented with blue dots, delivered flow (SCFM) by green dots, and inlet temperature by red dots. Looking at Saturday, Sunday, and Monday of Figure 43 it appears flow rate correlates with horsepower. As horsepower drops from Saturday to Sunday so does flow, then horsepower increases substantially Monday as does flow rate. Therefore it appears average flow rate with instantaneous input power data is reasonable to make conclusions with. The only period of time on the graph where horsepower and flow did not correlate was part of Thursday and Friday. At these times the compressor shut down (i.e. consumed no power) and compressed air flow was still recorded. This was a concern

initially, but the meter was installed downstream of the air dryer which receives air from the facility's back up compressor also. Thus the periods when flow existed and the main compressor was not operating is presumed to be flow due to the backup compressor. Unfortunately it was impossible to tell if the backup compressor ever turned on concurrently with the main compressor since pressure was not measured.

Looking at Figure 43 it is difficult to determine whether temperature is related to power because the horsepower plot and temperature plot do not follow the same trend for the entire week. Compressor inlet temperature appears to be a function of outdoor temperature if the compressor is operating. This is believed to happen because the compressor's inlet, where temperature was measured, is located inside the compressor's enclosure. The air temperature inside the enclosure will increase as the compressor consumes more power because the 150 horsepower electric motor will increase in temperature as it operates. This causes the electric motor to heat air entering the compressor's enclosure which eventually travels into the compressors inlet. Evidence of this is shown Wednesday, 3/23/11, when the compressor shuts down and the temperature drops consistently. Then the compressor switches on middle of the day Thursday, 3/24/11 and the temperature increases rapidly. Note even though the air entering the compressor is not outside air, technically, it varies quite often therefore compressor performance can still be determined at different temperatures.

Summary results for the compressor's first week of data collection were computed for time the compressor was in operation; these results are found in Table 6. Operation was defined as anytime the compressor was consuming a quarter of its rated horsepower (37.5 horsepower) or more. Highest, lowest, and average quantities were tabulated for horsepower, temperature, and flow rate. The amount of time the compressor operated during week 1 was tabulated also. Note a full week has 168 hours in it.

Table 6: First Week Summary Results

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414

All 14 weeks of data collection were analyzed as week 1 was. The plots and weekly highlights for weeks 2 through 14 are be found in *Appendix D.* The summary results for all 14 weeks of data collection are shown in Table 7.

Table 7: All Weeks Summary Results

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350
11	3.00	204	41	188	119	82	108	494	403	446
12	12.5	211	92	193	119	76	107	622	383	475
13	23	208	52	180	120	80	102	607	318	380
14	15.0	203	86	182	125	74	97	541	308	381

Looking at Table 7 it is hard to tell whether a correlation between inlet air temperature and compressor power exist. Therefore a plot of average horsepower versus average temperature for each week was created and a trend line was fitted to this data using Excel. The results of this plot and trend line are shown in Figure 44.

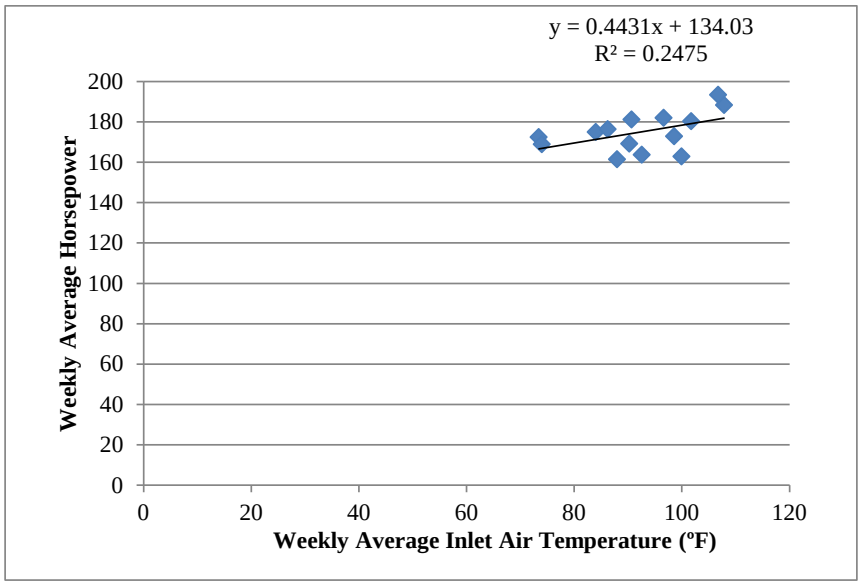

Figure 44: Average Horsepower versus Average Temperature

The plot's trend line does have an increasing slope, suggesting power increases with temperature, but there is a weak correlation (R^2 = 0.248) between average inlet temperature and average horsepower at best. One reason for a weak correlation is operation data for weeks 11 through 14 is scarce. The compressors operated for 142 hours on average for weeks 1 through 10 and 13.4 hours on average for weeks 11 through 14. As mentioned earlier the compressor started operating erratically in week 10 of data collection. Also, the plot above does not include variables such as compressor demand or capacity which compressor power is a function of.

Facility personnel stated the compressor's oil cooler become dirty causing the compressor to shut down. After talking to Ingersoll Rand personnel, it was discovered their air compressor

has a failsafe that shuts the compressor off when oil discharge temperature rises above a specific set point. If the oil cooler were dirty it could not reject heat causing the oil temperature to rise and shut the compressor down.

5.4.2 SCFM per Horsepower vs. Inlet Air Temperature

The weekly data analysis showed a weak correlation at best. Therefore it was decided to plot compressor efficiency versus inlet air temperature. Compressor efficiency is defined as SCFM per horsepower (SCFM /hp). As mentioned in Section 5.4.1 data for weeks 11 through 14 is scarce and compressor operation was nonexistent or erratic at the end of week 10. Therefore it was decided to only plot SCFM /hp versus temperature for weeks 1 through 10 only.

To generate a plot of SCFM /hp versus temperature all data points from Excel spreads for weeks 1 through 10 had to be extracted and placed in one Excel spread sheet. This resulted in 84,908 rows of data when the compressor was operating. Note 25 outliers that had efficiencies above 5 SCFM per horsepower where removed too. Excel can only plot 32,000 data points, thus, the 84,908 temperature and SCFM /hp data points had to be reduced. These data points were reduced by averaging the data by a factor of 3 minutes. Averaging data by a factor of 3 minutes means every 3 data points were average into a single data point (i.e. 30 minutes of data points will be reduced to 10 data points). After averaging data the following graph was generated:

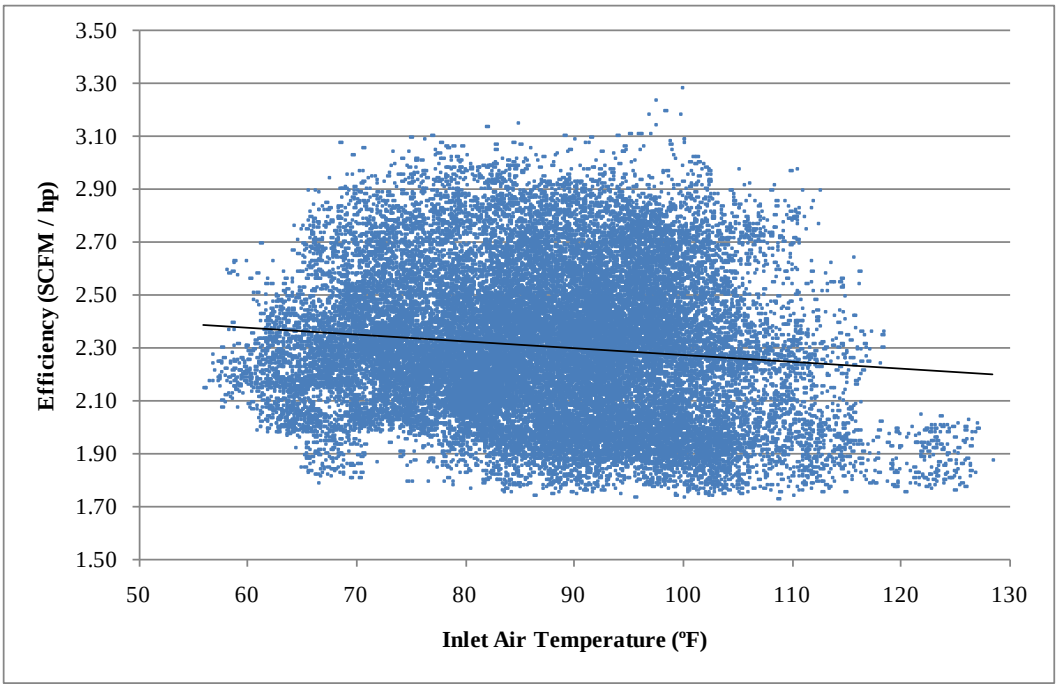

Figure 45: SCFM /hp vs. Inlet Air Temperature Averaged Over 3 Minutes

Figure 45 appears to be an uncorrelated group data points. The black line seen in the plot is a linear trend line generated by Excel. The trend line has a slope of negative 0.0026 and a R^2 value of 0.01313. This suggests an extremely weak correlation exist between compressor efficiency and inlet air temperature. The weak correlation may be due to instantaneous power data versus average flow rate data. Increasing the amount of data points averaged from 3 to 30 may tighten the data spread by leveling instantaneous power data. The

following plot is of data averaged using a 30 minute factor:

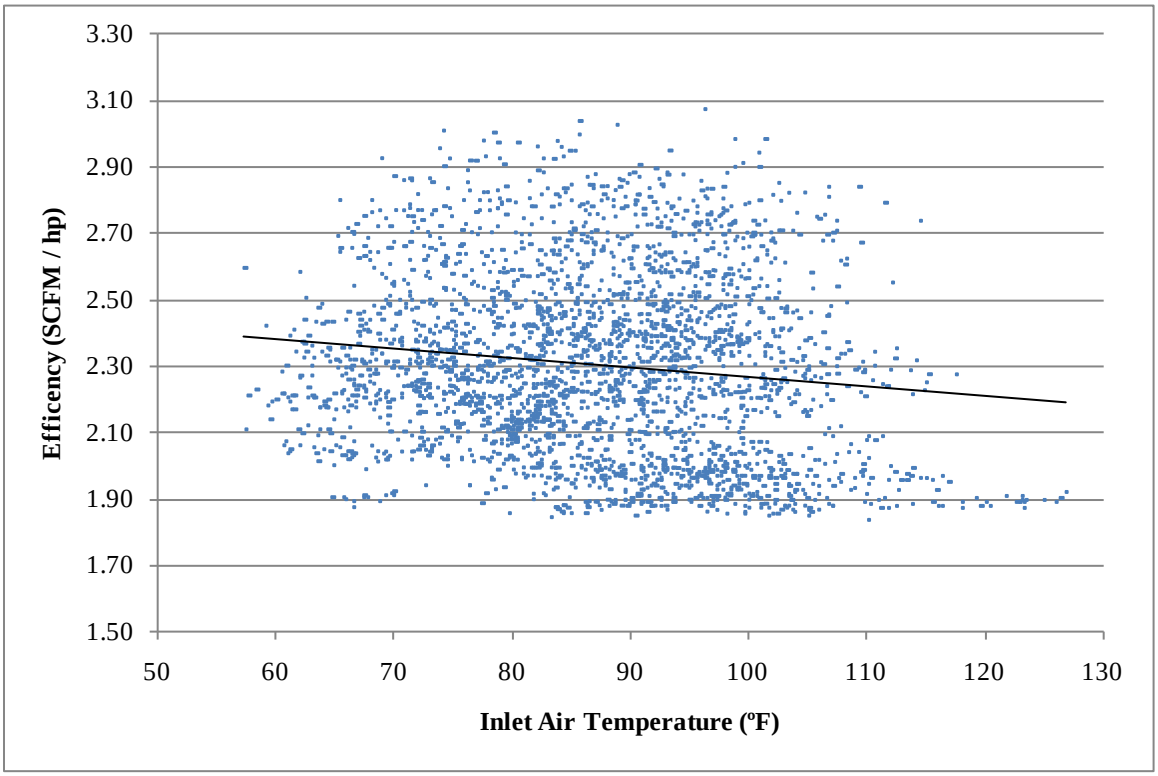

Figure 46: SCFM /hp vs. Inlet Air Temperature Averaged Over 30 Minutes

Averaging data by a 30 minute factor slightly improved correlation. The R^2 value increased from 0.0131 to 0.0165. The trend line slope became steeper increasing from negative 0.0026 to negative 0.0028.

Plotting SCFM /hp versus temperature was believed to be an excellent way to evaluate compressor performance due to temperature changes. Unfortunately it is difficult to draw reliable conclusion from this data due to weak correlations.

5.4.3 Input Power vs. Delivered Capacity at Constant Inlet Temperature

Plotting horsepower, flow, and inlet temperature versus time does not appear to show any correlation. Plotting average horsepower versus average inlet temperature has a weak correlation at best as does plotting compressor efficiency versus inlet air temperature. Therefore it was decided to plot electrical input power (horsepower) versus delivered flow (SCFM) at nearly constant inlet air temperatures. Generating a plot of horsepower versus SCFM is a good way to see compressor performance because it can be compared to compressor capacity control methods shown in Section 4.1 for creditability.

If inlet air temperatures affect an oil flooded screw compressor then a plot of power versus flow is hypothesized to change in a similar fashion as shown below.

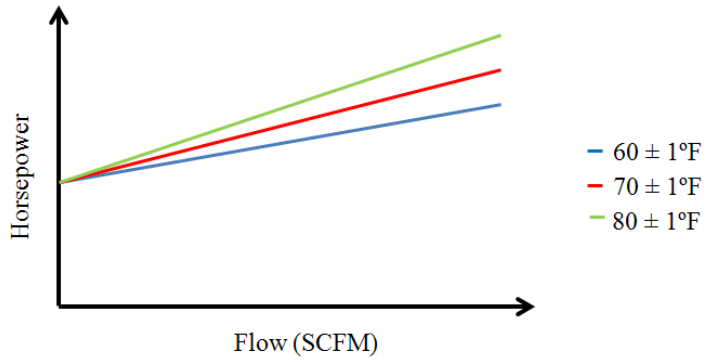

Figure 47: Hypothesized Horsepower vs. SCFM Plot if Temperature Affects Power

Plots are hypothesized to looks like this because compressors have a fixed unloaded input power and input power linearly increases until their peak power draw is meet. Therefore the y-axis (horsepower) of Figure 47 is fixed. If lower inlet air temperatures do improve performance than the compressors peak power will be reduced lowering the compressor curve's slope. Therefore the compressor curve is hypothesized to have its steepest slope at 80 ± 1ºF and flattest slope at 60 ± 1ºF.

As mentioned in Section 5.4.1 and 5.4.2 data points for weeks 11 through 14 are scarce and compressor operation was nonexistent or erratic at the end of week 10. It was decided to investigate compressor operation for weeks 11 through 14 further before more conclusions were made. A plot of horsepower versus flow, grouping data from weeks 1 through 10 together, and grouping data from weeks 11 through 14 together, was generated to do this. The results are seen in Figure 48.

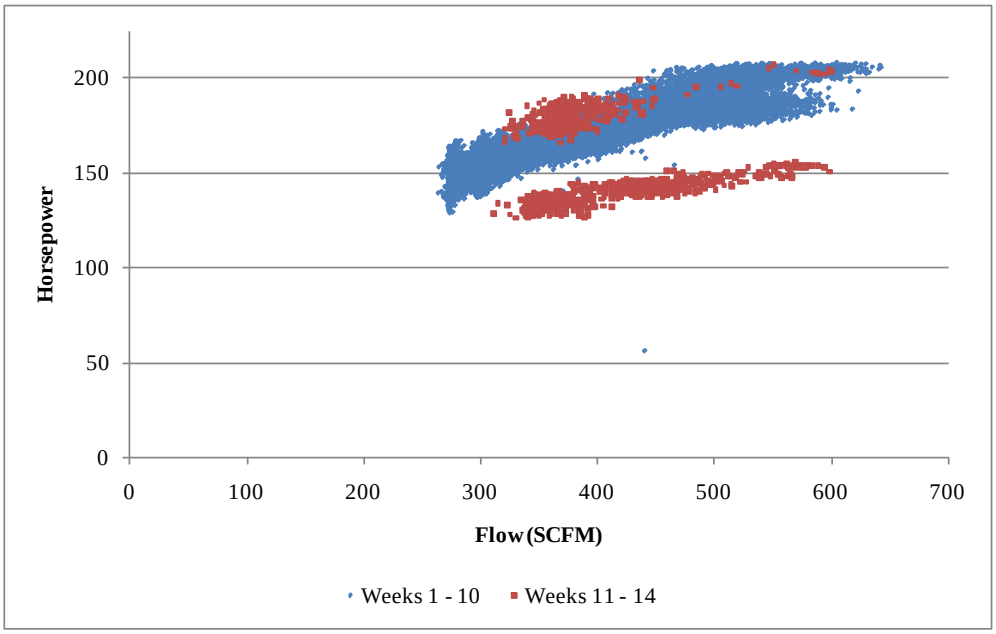

Figure 48: Horsepower vs. Flow All Weeks

Figure 48 shows two distinct data trends. Inspecting these trends closer showed 35,099 instantaneous power data points ranged from 130 horsepower to 210 horsepower and 625 data points ranged from 130 horsepower to 160 horsepower. The larger data trend was determined to consist of data from week 1 through 10 and 13. The average inlet air temperature for this data set was 89ºF. The smaller data trend had an average inlet air temperature of 104ºF and consisted of weeks 11, 12, and 14. Knowing the average air temperatures of these data trends and looking at Figure 48 appears the compressor performs better at higher inlet air temperatures. This does not match the theory shown in Section 4.5 nor does it correlate with the common knowledge that using cooler intake air saves energy. Note the compressor's oil cooler caused the compressor to shutdown as data collection

progressed. As data collection took place it was assumed the oil cooler would have gotten dirtier and dirtier reducing the amount of heat that could be rejected. As learned from Chapter 2, the power required to compressor a fluid decreases when heat is rejected during the compression process. Therefore the data for weeks 11 through 14 were hypothesized to consume more power. In actually the operation data for weeks 11 through 14 make the compressor appear to perform better when heat is not rejected and inlet air temperature rises.

Therefore it was decided to take another look at compressor theory. As the oil cooler rejects less heat the compressor becomes more adiabatic. Using equation 5.1.1 above a quick evaluation of compressor operation as the oil cooler becomes infinitely dirty (i.e. reject zero heat) can be done. Since the oil cooler could not reject heat sufficiently the oil injection temperature would continue to rise until it reached the oil and air ejection temperature. At this point the temperature rise across the oil entering and exiting the compressor would be small. Since this analysis is theoretical assume the oil temperature rise across the air compressor is very small, essentially zero. This assumption would cause the second half of equation 5.1.1 to reduce to zero. This in effect would lower the overall input power to the compressor if the air temperature didn’t rise. If the oil temperature entering and exiting the compressor were the same then the oil would not absorb any heat generated during the compression process, thus, the air temperature would increase because the air would be compressed adiabatically. To clarify what was written equation 5.1.1 is used as follows:

$$W_{in} = 3{,}105\frac{lbm}{hr}\cdot\frac{0.24BTU}{lbm\,°F}\cdot(545-68)°F + 13{,}716\frac{lbm}{hr}\cdot\frac{0.4\,BTU}{lbm\,°F}\cdot(194-194)°F$$

$$W_{in} = 335{,}460\,\frac{BTU}{hr} + 0\frac{BTU}{hr}$$

Note the air's exit temperature was estimated using the isentropic relation shown in equation 2.6.8 with an outlet pressure of 139.7 psia, inlet pressure of 14.7 psia, and inlet temperature of 68ºF. Multiplying each side by the conversion constant, 2,545 BTU = 1 hp-hr, the following is found:

$$W_{in} = 140\,hp + 0\,hp$$

Finally since the compressor is assumed to be adiabatic when the oil reaches a steady temperature an adiabatic efficiency needs to be included as follows:

$$W_{in} = {}^{140\,hp}/_{80\%} = 175\,hp$$

The adiabatic efficiency was estimated using Figure 17. Assuming the calculations are correct it is hypothesized an air compressor with a dirty oil cooler will consume more power. Since the hypothesis and data do not show the same results another theory for the smaller data set was made.

The second theory is flow rates in weeks 11, 12, and 14 are related to the facility's backup air compressor instead of the 150 horsepower compressor that was data logged. Looking at *Appendix D*, data for weeks 11 through 14 can be viewed. It appears the compressor operated sporadically weeks 11 and 12. The sporadic operation in theses weeks may have

skewed data. As the compressor sporadically turned on and off instantaneous power would have been recorded from the CT meters and average capacity from the flow meter. The instantaneous power recorded may reflect an unloaded 150 horsepower compressor because the systems storage was already charged due to the backup compressor, thus, the 150 horsepower compressor would not have fully loaded. Also, the flow meter may have recorded a average loaded capacity (SCFM) due to the backup compressor. The compressor operated for nearly 23 hours continuously week 13 of data collection. This means the backup compressor was less likely to turn on skewing data. This may be why red data points are shown to trend normally in Figure 48. Week 11 the compressor only operated 3 hours sporadically and week 12 the compressor operated 12.5 hours sporadically. Week 14 the compressor operated nearly 15 hours continuously but the data is skewed. Note week 14 is the week the data logger melted which may have some affect on data.

Using theory and logic data is presumed to be skewed due to the backup compressor operating. Since operation data collection for weeks 11 through 14 were not as consistent as weeks 1 through 10 it was decided to remove weeks 11 through 14 for further data analysis.

To generate an actual plot similar to the hypothesized one shown in Figure 47 inlet air temperature, horsepower, and delivered flow data points were extracted from Excel spread sheets for weeks 1 though 10 and placed in a single Excel spread sheet. Once all the data points were extracted and placed in one spread sheet they were filtered into 12 different temperature groups. The temperature groups range from ~60ºF to ~105ºF by increments of

5ºF. The (~) means ± 1ºF; plus or minus 1ºF was used ensure a substantial number of data points were found. The following scatter plot was generated utilizing Excel:

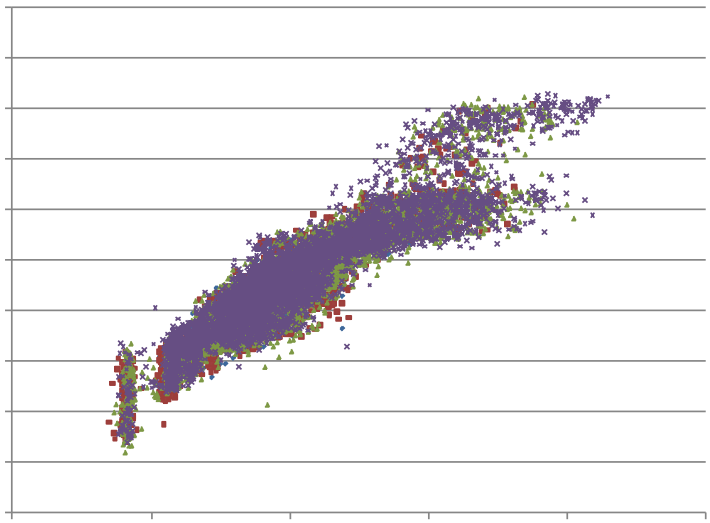

Figure 49: Horsepower vs. SCFM at Various Temperatures

Looking at the scatter plot in Figure 49 above, some concerns are found. First the lowest horsepower recorded, about 130 horsepower, seems relatively high. According to the CAGI data sheet the unloaded compressor power draw should be 44.4 kW or 59.5 horsepower. The CAGI data sheets also states the compressor's fully loaded power draw should be 135.8 kW or 182 horsepower. Referring back to Section 4.1 modulating compressor performance without blowdown ranges from 70% input power to 100% input power. Ingersoll Rand claims their compressors can modulate down to 60% input power. Assuming 210 horsepower is fully loaded input power, according to data, then unloaded horsepower should

be around 126 horsepower (210 hp x 60% = 126 hp). The scatter plot also shows power modulation is mostly linear as seen in compressor capacity control methods. With this knowledge the compressor's power modulation makes since assuming the compressor does not successfully blowdown. The high horsepower numbers are still a concern and the only justification maybe a 90% power factor used to calculate power was a bad assumption.

Unfortunately it is difficult to tell if inlet air temperature affects compressor performance in Figure 49. The reason being data points lie on top each other. To resolve this issue trend lines were found for each temperature group assuming a fixed y-intercept of 105 horsepower. Equations for these trend lines, R^2 values, and the number of data points in each temperature group are shown in Table 8.

Table 8: Trend Line Power Equations vs. SCFM at Various Temperatures

Temperature	Predicted Horsepower Equation	R^2	Data Points
60 ± 1 ºF	Power = (0.1602 x SCFM) + 105	0.6926	497
65 ± 1 ºF	Power = (0.161 x SCFM) + 105	0.8739	2,266
70 ± 1 ºF	Power = (0.1628 x SCFM) + 105	0.8605	3,787
75 ± 1 ºF	Power = (0.1643 x SCFM) + 105	0.8566	4,850
80 ± 1 ºF	Power = (0.1642 x SCFM) + 105	0.8529	6,937
85 ± 1 ºF	Power = (0.1664 x SCFM) + 105	0.859	7,436
90 ± 1 ºF	Power = (0.1671 x SCFM) +105	0.8447	7,762
95 ± 1 ºF	Power = (0.167 x SCFM) + 105	0.8259	7,282
100 ± 1 ºF	Power = (0.1661 x SCFM) + 105	0.7674	5,275
105 ± 1 ºF	Power = (0.1657 x SCFM) + 105	0.6929	2,620
110 ± 1 ºF	Power = (0.1657 x SCFM) + 105	0.6929	1,278
115 ± 1 ºF	Power = (0.1626 x SCFM) + 105	0.4699	556

In Table 8 the column labeled “Predicted Horsepower Equations” are trend line equations found using Excel. These equations can be used to predict compressor power consumption at a specific temperature. As an example, the predicted horsepower to compress 600 SCFM of air at ~60ºF is estimated as follows:

$$Horsepower = (0.1602 \times 600\, SCFM) + 105$$

$$Horsepower = (96) + 105$$

$$Horsepower = 201$$

These equations are assumed to be reliable because R^2 values show a strong correlation for all temperature data sets except ~115ºF. An important observation to make form Table 8 is predicted power equations have a increasing slope as temperature rises from ~65ºF to ~95ºF. After ~95ºF predicted power slopes start decreasing. To better visualize what is happening here it was decided to plot predicted horsepower versus delivered flow from 0 to 690 SCFM. The results are shown in Figure 50.

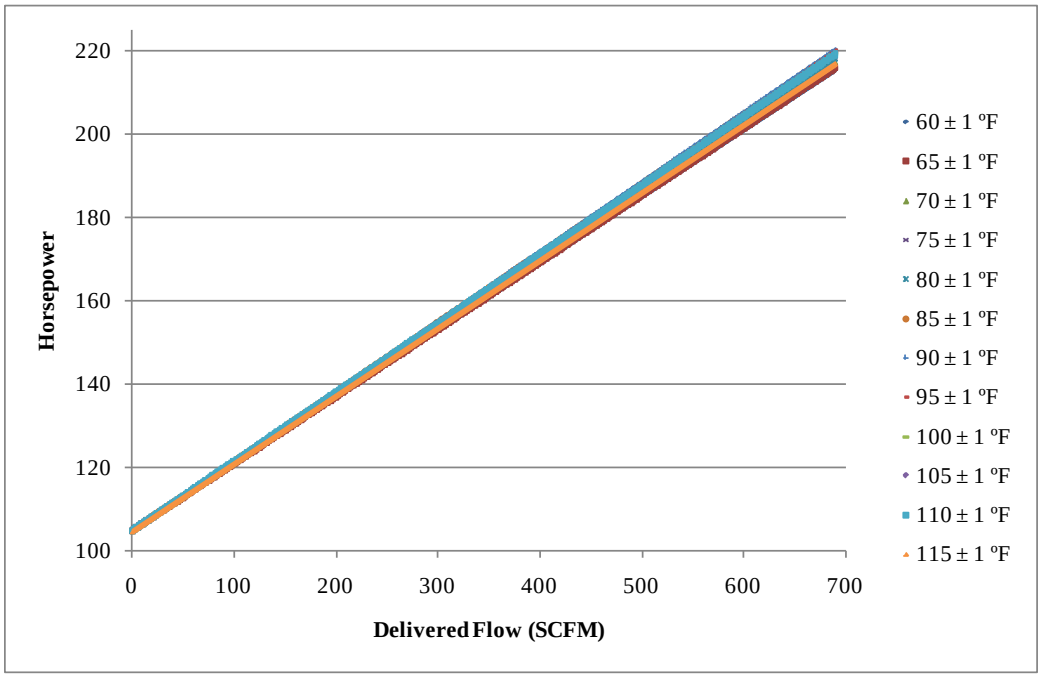

Figure 50: Predicted Horsepower vs. Delivered Flow at Various Temperatures

Looking at Figure 50 it can be seen that predicted horsepower, for all temperature groups, starts at 105 horsepower and peak power varies slightly. The highest predicted power draw at maximum capacity (690 SCFM) was 220 horsepower at ~85ºF, ~90ºF, ~95ºF, and ~100ºF. The lowest power draw at 690 SCFM was 216 horsepower at ~60ºF. The percent difference between these values is 1.82%. This data suggest inlet air temperature slightly affects compressor performance. To better visualize changes in predicted horsepower the y-axis and x-axis of Figure 50 can be adjusted as follows:

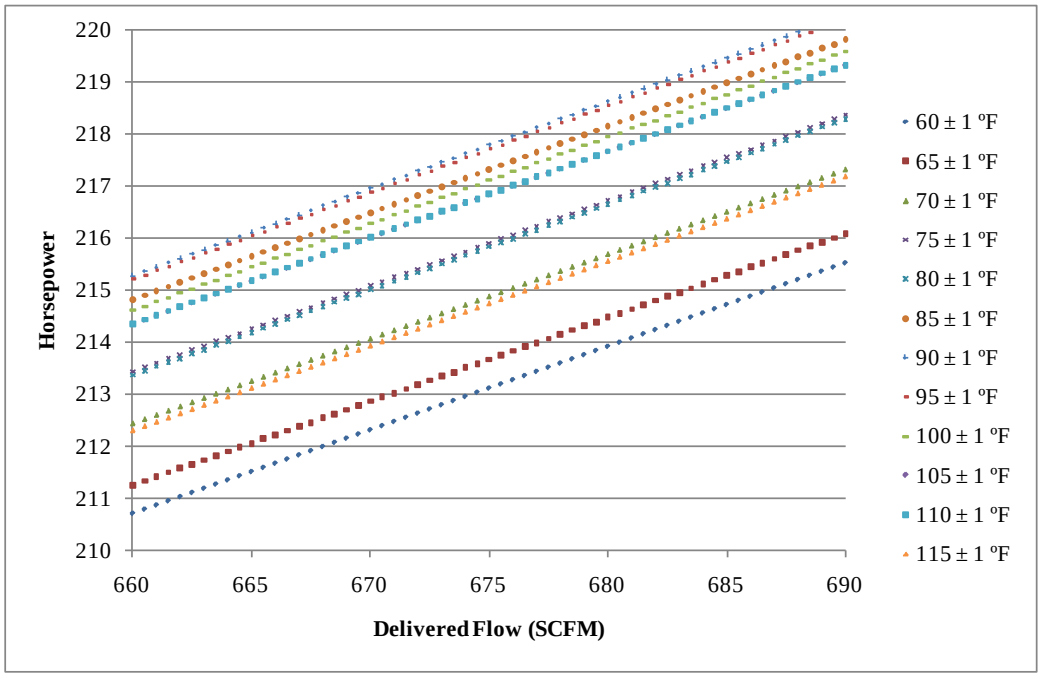

Figure 51: Predicted Horsepower vs. Delivered Flow at Various Temperatures Zoomed In

The average delivered flow for the data show in Figure 49 was 396 SCFM. It was decided to make a table of predicted horsepower at this flow rate to compare theoretical power differences to actual power differences. Theoretical power increases and reductions were predicted using equation 4.5.1 and 70ºF as a base. 70ºF was chosen because CAGI test their compressors at 68ºF. The results of predicted horsepower and predicted power differences at a flow of 396 SCFM compared theoretical power differences are shown in Table 9.

Table 9: Predicted Power Differences at Various Temperatures and 396 SCFM

Temperature	Predicted Horsepower	Efficiency (SCFM / Horsepower)	Predicted Horsepower Percent Difference	Theoretical Horsepower Difference
60 ± 1 ºF	168	2.35	-0.608%	-1.89%
65 ± 1 ºF	169	2.35	-0.421%	-0.94%
70 ± 1 ºF	169	2.34	0.000%	0.00%
75 ± 1 ºF	170	2.33	0.351%	0.94%
80 ± 1 ºF	170	2.33	0.327%	1.89%
85 ± 1 ºF	171	2.32	0.841%	2.83%
90 ± 1 ºF	171	2.31	1.005%	3.77%
95 ± 1 ºF	171	2.31	0.981%	4.72%
100 ± 1 ºF	171	2.32	0.771%	5.66%
105 ± 1 ºF	171	2.32	0.678%	6.60%
110 ± 1 ºF	171	2.32	0.678%	7.55%
115 ± 1 ºF	169	2.34	-0.047%	8.49%

Table 9 has the same results as Table 8 and Figure 50; compressor performance is best at ~60ºF and performances get worse until temperature rises to ~95ºF. At 95ºF compressor performance begins to increase. Comparing the percent difference columns in Table 9 shows the isentropic theoretical power reduction equation does not work for a oil flooded screw compressor. To visually see this difference the following plot was generated:

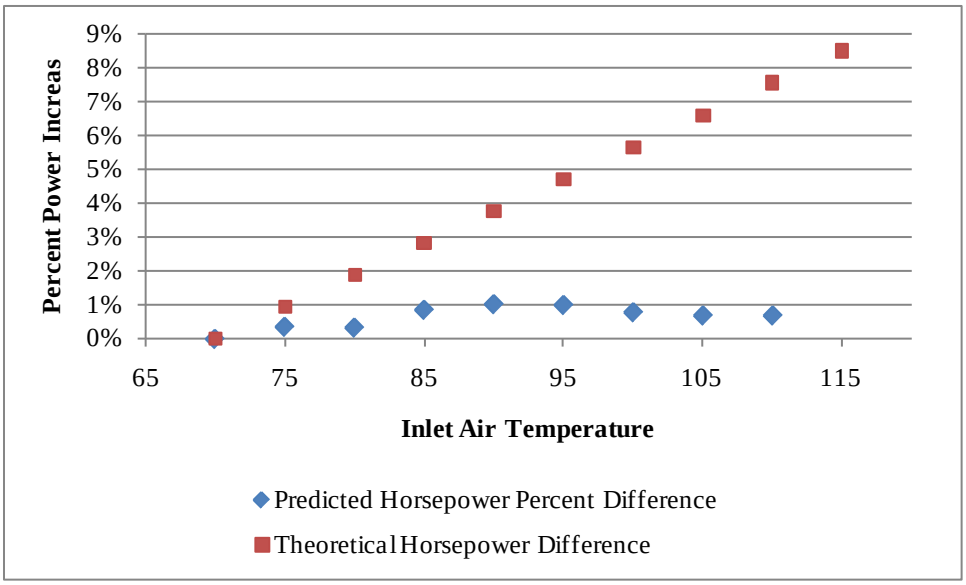

Figure 52: Percent Power Increase vs. Inlet Air Temperature

Figure 52 shows using the isentropic compression process to predict power savings, due to cooler inlet air temperatures, in an oil flooded screw compressor is incorrect. Table 8, Figure 50, and Table 9 show evidence of compressor performance being affected by inlet air temperatures. The maximum affect of inlet air temperature is predicted to be 1.82% when the compressor is operating at maximum capacity. A power increase of 1.82% is small. As mentioned in Section 2.7.1 relative humidity is believed to affect compressor performance by a maximum of 3%, thus, the affects seen is this data analysis may be due to relative humidity instead of inlet air temperature.

5.4.4 Input Power vs. Inlet Mass at Various Temperatures

As discussed in Section 2.7.1 air has moisture in it, thus, the mass entering a compressor is a combination of water vapor and dry air. Data analyses done in Sections 5.4.1, 5.4.2, and 5.4.3 were for input power versus compressor output, where compressor output was mass of dry air only. Since temperature affects the amount of moisture in a volume of air and properties of dry air it was decided to conduct an analysis of input power versus input mass.

The flow meter measured data in SCFM which is essential a measurement of dry air mass. Note a standard cubic foot of air has a density of 0.0741 pound mass of dry air per foot cubed (lbm/ft^3). It is assumed no dry air exits the compressor after it is drawn in. Therefore, the amount of dry entering the compressor is the SCFM measurement taken by the flow meter. The amount of moisture entering the compressor can be found knowing the relative humidity and dry bulb temperature of air entering the compressor. Knowing the temperature of air entering the compressor allowed the saturation pressure of water vapor, at that temperature, to be found for each data point. Instead of using steam tables to find the saturation pressure of every data point an Excel program named XSteam was utilized. Using the saturation pressure of water vapor at a specific temperature ($P_{sat@T}$) and the relative humidity recorded from the experiment the partial pressure of water vapor (P_{wv}) entering the compressor could be found with the following equation shown in Chapter 2:

$$RH = \frac{P_{wv}}{P_{sat\,@\,T}} \qquad \textit{(Eqn. 2.7.2)}.$$

Knowing the partial pressure of water vapor entering the compressor and the atmospheric pressure (14.7 psia) allowed specific humidity of water vapor to be found with the following relation:

$$\omega = 0.62198 \frac{P_{wv}}{14.7 - P_{wv}} \qquad \textit{(Eqn. 5.4.1)}$$

Finally the mass of water vapor entering the compressor could be found using specific humidity and the mass of dry air entering the compressor.

Unfortunately relative humidity data is scarcer than inlet air temperature data. The reason being relative humidity data logging did not begin until the second week of data collection and trips were taken to western North Carolina every six weeks. Six weeks was too long a time for the relative humidity logger to record data at one minute intervals. Fortunately there is a large amount of relative humidity data. Relative humidity was successfully recorded for weeks 2, 3, 4, 8, 9, 10, and 14.

Data for weeks 2, 3, 4, 8, 9, and 10 were compiled into one Excel spread sheet. Once data was compiled in one spread sheet equation 4.5.1, XSteam, and density of air standard conditions were used to calculate the mass of air moist and dry entering the air compressor. Next data was filtered in to temperature groups as done in Section 5.4.3. The results of this work are visually represented in the following scatter plot:

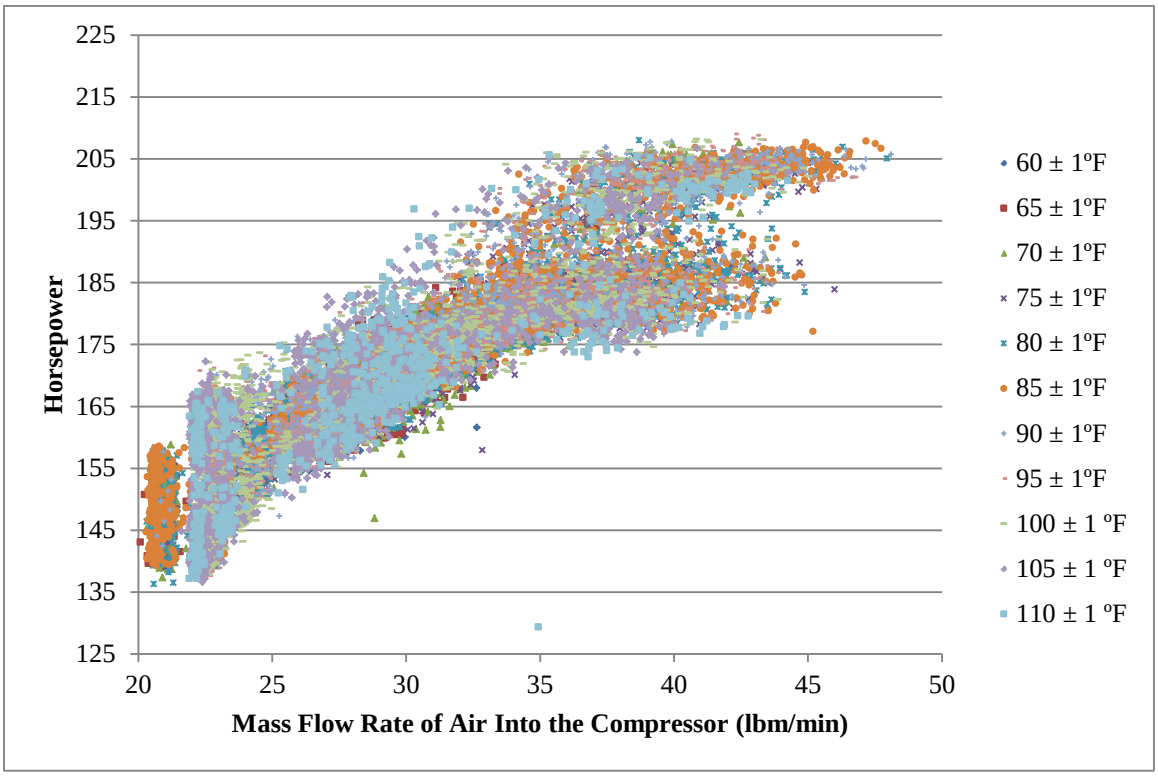

Figure 53: Horsepower vs. Inlet Mass Flow Rate

It is important to note the mass flow rate along the x-axis of this plot is the sum of dry air and moist air. This scatter plot resulted in data points being plotted on top each other, thus, it is difficult to determine if air temperature is affecting compressor performance. Therefore it was decided to find the trend line of each temperature group. Results are seen in Table 10.

Table 10: Predicted Horsepower Equations vs. Mass Flow Rate

Temperature	Predicted Horsepower Equation	R^2	Data Points
60 ± 1 ºF	Power = (2.2609 x lbm) + 102	0.6258	297
65 ± 1 ºF	Power = (2.3013 x lbm) + 102	0.8416	805
70 ± 1 ºF	Power = (2.2852 x lbm) + 102	0.8513	2,019
75 ± 1 ºF	Power = (2.2854 x lbm) + 102	0.8622	2,228
80 ± 1 ºF	Power = (2.2933 x lbm) + 102	0.8562	3,177
85 ± 1 ºF	Power = (2.307 x lbm) + 102	0.8669	3,141
90 ± 1 ºF	Power = (2.3181 x lbm) + 102	0.8229	3,029
95 ± 1 ºF	Power = (2.3179 x lbm) + 102	0.8141	2,534
100 ± 1 ºF	Power = (2.2919 x lbm) + 102	0.693	2,445
105 ± 1 ºF	Power = (2.2913 x lbm) + 102	0.6118	1,556
110 ± 1 ºF	Power = (2.2974 x lbm) + 102	0.7159	813

Table 10 shows data correlated strongly for most temperature groups. Correlations for data at ~60ºF and ~105ºF are neither strong nor weak. Predicted horsepower equations in Table 10 show trend line slopes increasing as temperature rises from ~60ºF to ~90ºF excluding ~65ºF. At ~95ºF slopes begins decreasing until ~110ºF. At ~110ºF slope is steeper than ~70ºF but not as steep as slope at ~90ºF. A plot of these trend lines was generated from a mass flow rate of zero pound mass per minute to 51.13 pound mass per minute; note 51.13 lbm/min is equivalent to 690 SCFM.

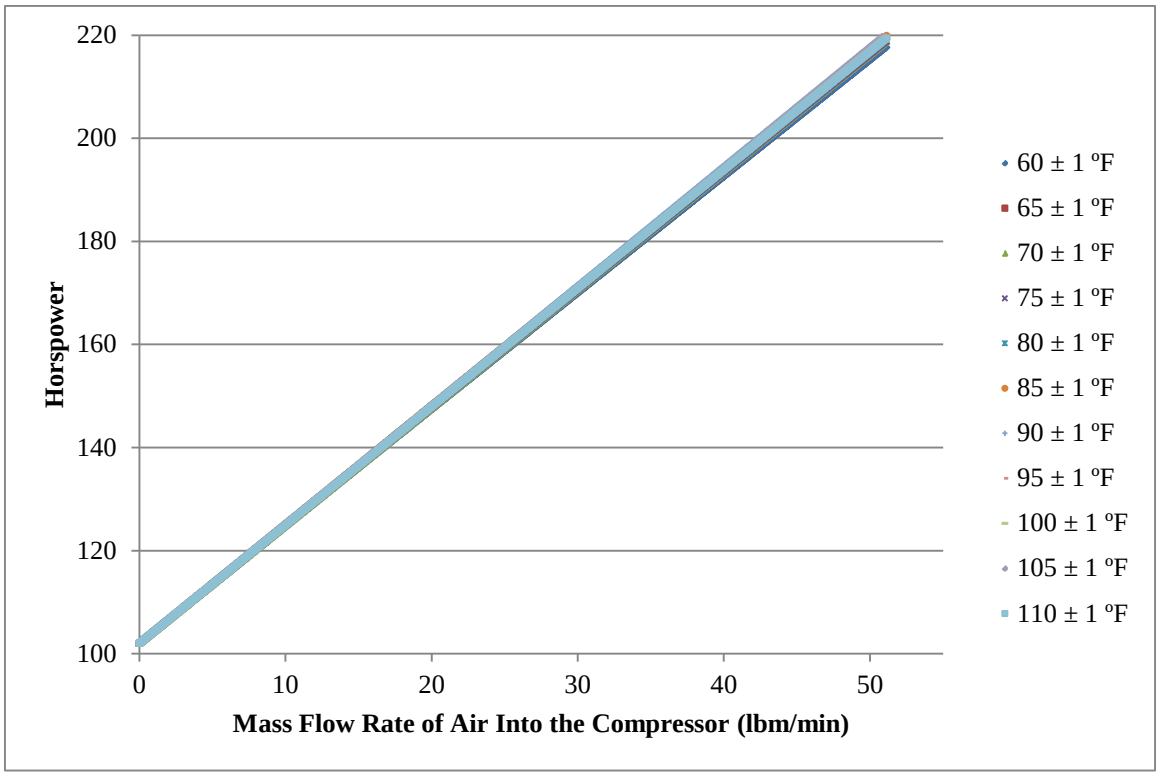

Figure 54: Horsepower vs. Mass Flow Rate

Unfortunately Figure 54 appears to show one trend line only because the predicted horsepower trend lines are nearly identical. Adjusting the y and x-axis of Figure 54 gives a more detailed view of what predicted horsepower is doing at the different temperature ranges when the compressor is operating at maximum capacity.

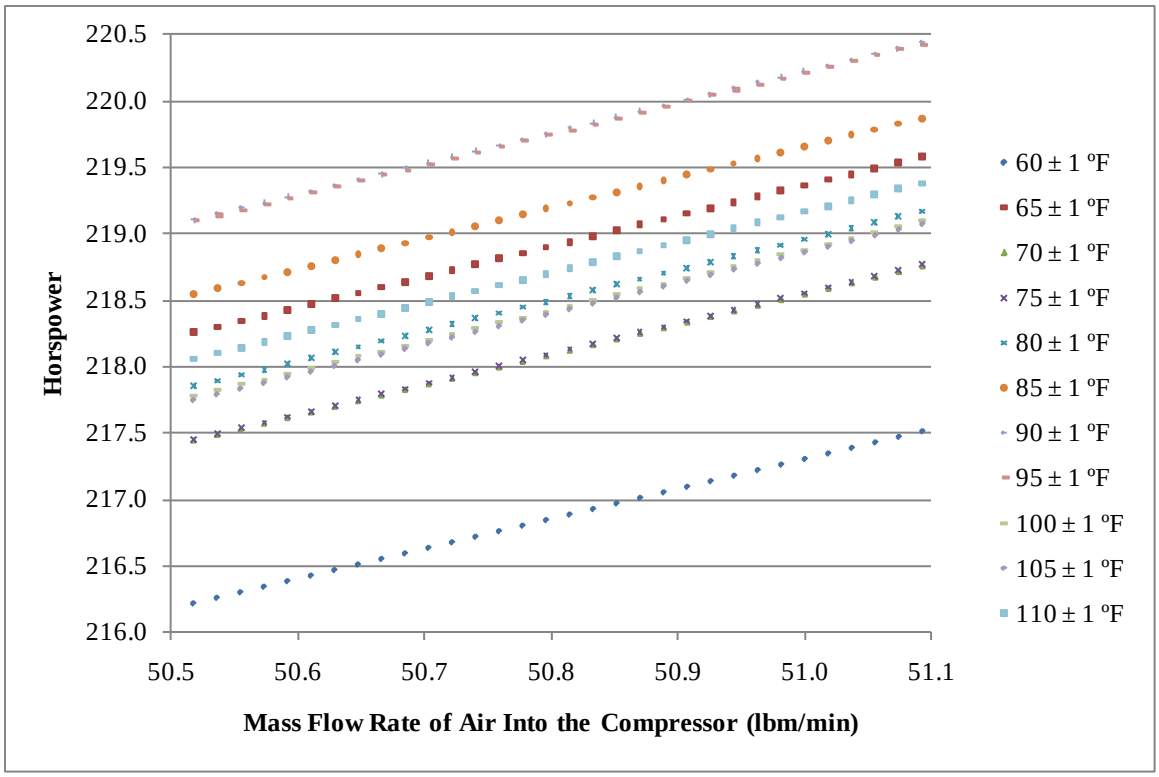

Figure 55: Horsepower vs. Mass Flow Rate Zoomed In

The plot shown in Figure 54 was scaled down tremendously to generate the plot seen in Figure 55. The x-axis only varies from 50.5 to 51.1 pound mass per minute and the y-axis only varies from 216 to 220.5 horsepower. This suggests predicted horsepower lines are very similar. The greatest percent difference in power was due to lines at ~95º and ~60ºF. The percent difference between these two lines at 51.1 lbm/min is 1.38%. Looking at Figure 55 predicted horsepower from greatest power draw to lowest is listed as follows:

~90ºF, ~95ºF, ~85ºF, ~65ºF, ~110ºF, ~80ºF, ~100ºF, ~105ºF, ~75ºF, ~70ºF, and ~60ºF.

This list does not appear to have a pattern.

It was decided to create a table to calculate the predicted horsepower for each temperature group at the average mass flow rate of the entire data set to compare theoretical horsepower percent differences to predicted horsepower percent differences with a base line of 70ºF. The average mass flow rate of the entire data set was 29.9 lbm/min.

Table 11: Predicted Horsepower Equations for Horsepower vs. Inlet Mass Flow Rate

Temperature	Predicted Horsepower	Efficiency (SCFM / Horsepower)	Predicted Horsepower Percent Difference	Theoretical Horsepower Difference
60 ± 1 ºF	169.6	2.38	-0.427%	-1.89%
65 ± 1 ºF	170.8	2.36	0.283%	-0.94%
70 ± 1 ºF	170.3	2.37	0.000%	0.00%
75 ± 1 ºF	170.4	2.37	0.004%	0.94%
80 ± 1 ºF	170.6	2.37	0.142%	1.89%
85 ± 1 ºF	171.0	2.36	0.383%	2.83%
90 ± 1 ºF	171.3	2.36	0.578%	3.77%
95 ± 1 ºF	171.3	2.36	0.574%	4.72%
100 ± 1 ºF	170.5	2.37	0.118%	5.66%
105 ± 1 ºF	170.5	2.37	0.107%	6.60%
110 ± 1 ºF	170.7	2.36	0.214%	7.55%

Table 11 shows utilizing the isentropic compression process to predict power savings for an oil flooded screw compressor does not work. The predicted horsepower percent difference never matches the theoretical power difference. The differences between these two columns can be seen in the following plot:

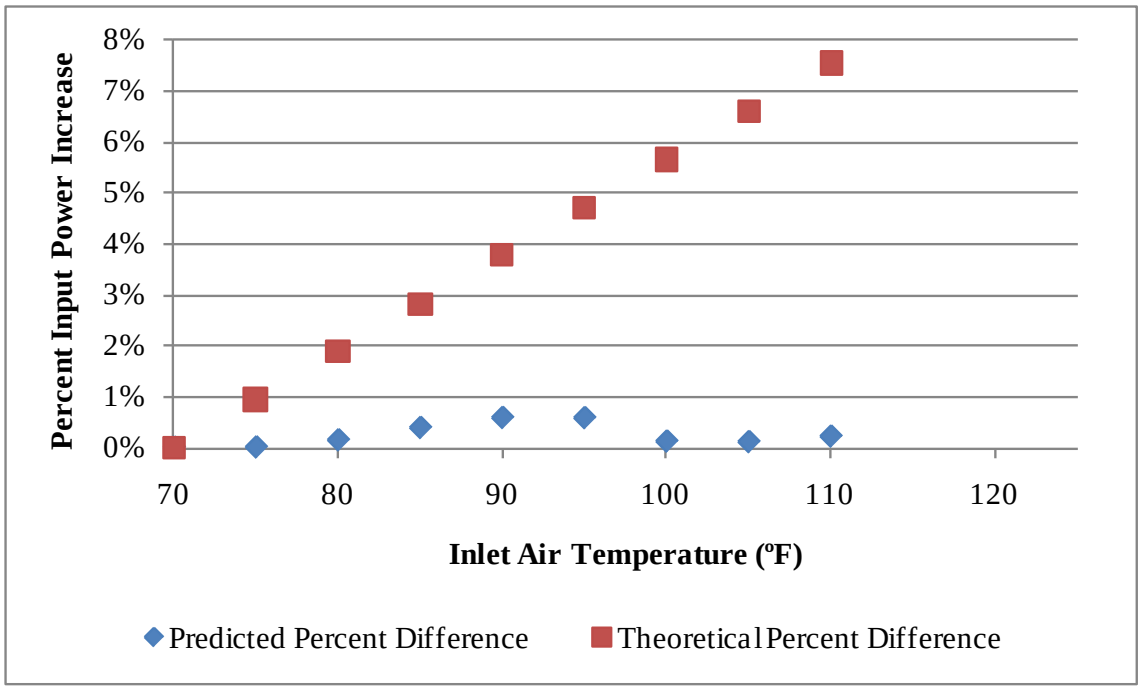

Figure 56: Percent Input Power Increase vs. Air Temperature Due to Inlet Mass

Chapter 6 – Conclusions

6.1 Results and Conclusion

For a compressor used in industry, it is commonly believed compressing outside air instead of warm plant air saves energy. The objective of this thesis was to experimentally determine if this is true for an oil flooded screw compressor. It was hypothesized that using outside air in an oil free screw compressor will save energy but using outside air in oil flooded screw compressor will not. To determine if this hypothesis was true a compressor, compressing outside air, utilized for industrial processes was data logged. 14 weeks of data were collected before the data logger and compressor prematurely failed.

The data analyses begin by plotting power, inlet air temperature, and flow versus time for a duration of a week. Fourteen of these plots were made. Studying these plots did not prove or disprove the hypothesis stated above. However, these plots validated data loggers operated correctly and the compressor's inlet air temperature was affected by climate. A table summarizing each week's data was created. The summary included lowest, highest, and average measured power, inlet air temperature, and capacity (SCFM). A plot of average horsepower versus average inlet air temperature was created. The plot did show average input power increased with average inlet air temperature. However, the slope of this increase was merely 0.443 and the correlation was weak obtaining a coefficient of determination

equivalent to 0.248. Since correlation for this data analysis was weak further analysis had to be done before a conclusion could be made.

Commonly air compressor performance in rated in terms of SCFM per horsepower, thus, plotting SCFM per horsepower versus inlet air temperature should be a good way to evaluate compressor performance. Horsepower, capacity (SCFM), and inlet air temperature data for weeks 1 through 10 were compiled to generate a plot of SCFM/hp versus inlet air temperature. Weeks 11 through 14 were not used because suspicions of skewed measurements arose. The first plot of SCFM/hp versus inlet air temperature could not be generated because Excel can plot 32,000 rows of data only. Therefore data was averaged by a factor of 3. The plot of SCFM/hp versus temperature average by a factor of 3 appeared to show zero correlation. A best fit line or trend line was found for the data. The trend line suggested compressor performance decreased as inlet air temperature increased but the slope was small (-0.0026) and correlation was extremely weak ($R^2 = 0.013$). The data was averaged again by a factor of 30 and plotted. Averaging data by 30 slightly helped correlations raising it from a R^2 of 0.013 to 0.0165. Slope was affected also increasing from -0.0026 to -0.0028. The second data analysis had results very similar to the first data analysis. Both analyses showed compressor performance slightly decreasing as inlet air temperature raised, however, both analyses had weak correlations. Therefore further data analysis was needed.

Due to very weak correlations in data collection, so far, it was believed compressor performance was affected by another variable. Viewing weekly data analysis and capacity control methods for oil flooded screw compressors (Section 4.1), again, it was decided the data logged compressor for this experiment utilized a modulating capacity control scheme. A modulating compressor's power consumption is a function of the compressor's inlet valve because as the valve shuts the pressure ratio across the compressor increases. This causes the compressor to operate less efficiently yet consume less power because the mass of air being compressed is decreased. Therefore another hypothesis was made; as inlet air temperature raises the compressors inlet valve has to open more, due to air's lower density, to deliver a specific mass of compressed air. The more open inlet valve causes the compressor to operate more efficiently and hides the decrease in efficiency due to higher air temperatures. To solve this issue the compressor's inlet valve needed to be isolated.

To fix the compressor inlet valve operation it was decided to plot input power versus capacity (SCFM) at various temperatures. Initially a scatter plot was generated and no conclusions could be made because data points laid on top each other at various temperatures. Best fit lines or trend lines were found for each temperature group. Note trend lines were found by fixing their y-intercept because literature for compressor performance did this. Trend line slopes increased as temperature rose then slopes began decreasing around 95ºF. Note correlations for temperature groups were strong with R^2 values ranging from nearly 0.7 to 0.85. Trend lines were used to predict compressor power. A predicted

horsepower datum was set at 70ºF. Percent differences were tabulated according to predicted horsepower numbers below and above 70ºF. Percent differences due to predicted horsepower's were compared to theoretical percent power differences using the isentropic compression process. A maximum predicted horsepower percent differences of 1% was found and the maximum theoretical percent power difference was 8.5%. This data analysis suggested compressor performance was not affected by inlet air temperature and using the isentropic compression process to predict power savings is incorrect.

When outside air is compressed water vapor is compressed also. Data analyses before compared the amount of power required to compressor air to its desired output of dry air. Therefore it was decided to compare the compressor input power to input mass, that being mass of water vapor plus mass of dry air. Temperature affects the properties of dry air and the amount of moisture air can hold, thus, plotting compressor power versus input mass should give a more accurate view of how inlet air temperature affects compressor performance. It was also believed moisture in air would affect the compressor's inlet valve as high temperature air did; as humidity increases the valve would open, due to moisture taking up space, more causing the affects of humidity to be hidden due to a lower pressure ration. Unfortunately, humidity data was scarcer than other data therefore a plot of input power versus input mass could be made for weeks 2, 3, 4, 8, 9, and 10 only. Once data was compiled together a plot of input power versus input mass as various temperature groups was made. Trend lines were fit to this data like the power versus SCFM data analysis. The

results were very similar to the input power versus SCFM data analysis. A maximum predicted horsepower percent difference of 0.578% was found were the maximum theoretical difference was 7.55%. Trend line slope for this data increased from 60ºF, excluding 65ºF, until 95ºF then started decreasing again. This data analysis also suggested compressor performance is not affected by inlet air temperature and using the isentropic compression process to predict power reductions or increases is incorrect.

The hypothesis that the compressor's inlet valve opens more due to outside air temperature and humidity appears to be correct since data correlation increased dramatically once input power versus capacity was compared. In conclusion predicting or estimating power reductions and or energy savings using the isentropic compression process for an oil flooded screw compressor is incorrect. Comparing predicted horsepower percent differences to theoretical power percent differences show no similarities. This was shown when comparing input power to SCFM output and input power to input mass. Oddly these data analyses, input power versus SCFM and input power versus input mass, show compressor performance decreasing as temperature rises to 90ºF then performance starts increasing from 95ºF onward. Also, both data sets had a power increase at 65ºF for unknown reasons. The other two data analyses, average horsepower versus average inlet air temperature and SCFM/hp versus inlet air temperature, suggest compressor performance degrades slightly as air temperatures rise. Unfortunately these analyses had very weak correlations. From all four data analysis it can

be concluded that inlet air temperature does not have significant affect on the oil flooded screw compressor tested in this experiment.

The amount of heat rejected by a compressor's oil cooler is dictated by ambient air temperatures. The compressor in this experiment has its oil cooler located outside, thus, changes in weather would affect the oil coolers ability to reject heat. Unfortunately the effects of weather on the compressor's oil cooler were not measured in this experiment. In theory, decreases in outside air temperatures would allow the oil cooler to reject more heat causing the compressor to operate more efficiently. As outside air temperatures increase the oil cooler's ability to reject heat would decrease causing the compressor to operate less efficient. In actuality, oil injected screw compressors control the temperature of oil they inject, therefore, changes in ambient air temperatures surrounding the oil cooler are not believed to have skewed data in this experiment.

6.4 Future Work and Suggestions

If this experiment is redone in the future it is suggested to measure energy instead of instantaneous power and measure more variables. Measuring energy in a pulse fashion versus capacity, flow rate, in a pulse fashion would result in a less error prone analysis. This is believed because average energy would be compared to average capacity. Compressor outlet and inlet pressure should be measured as well. This information would give more details on how the compressor operates (i.e. is the inlet valve a function of temperature and

humidity). Pressure would also show when the backup compressor switched on thus eliminating skewed data. Oil flow rate, inlet, and exit temperature should be measured. They will give an idea of what the air's exit temperature is and they will allow the amount of power rejected through the oil cooler to be calculated. Knowing the amount of power being rejected from the oil cooler will show how much power the air absorbs and show how compressor performance is affected by heat rejection. Data logging an oil flooded screw compressor using a load unload screw compressor would be interesting also. Using a load/unload compressor would eliminate complications due to the compressors inlet valve. Measuring a non oil flooded screw compressor's performance due to outside air would show how much power consumption actually varies compared to theoretical isentropic predictions also.

Finally, if a facility is trying to decide where to install a oil flooded screw compressor locating it outside is not a bad idea. The compressor is estimated to see very slight performance increases in the winter months. However, if a compressor is located inside a facility it is not suggested to install ducting to the compressors inlet. The reason being duct work may cause a pressure drop into the compressors intake. It is believed pressure drop will affect compressors performance in a negative matter more than the positive affect of using low temperature outside air. To ensure a compressor operates efficiently its suggested to fix air leaks, operate at the lowest outlet pressure possible, utilize waste heat, and choose the correct capacity control scheme depending on facility needs.

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APPENDICES

Appendix A: CAGI datasheet

Ingersoll Rand
Industrial Technologies

COMPRESSOR DATA SHEET
Rotary Screw Compressor

MODEL DATA - FOR COMPRESSED AIR			
1	Manufacturer: **Ingersoll Rand**	Date: **October 2009**	
2	Model Number: **R110IU-A-125** [X] Air-cooled [ ] Water-cooled [X] Oil-injected [ ] Oil-free	# of Stages: **1**	
		VALUE	UNIT
3	Rated Capacity at Full Load Operating Pressure [a, f]	**690**	acfm[a,f]
4	Full Load Operating Pressure [b]	**125**	psig[b]
5	Maximum Full Flow Operating Pressure [c]	**128**	psig[c]
6	Drive Motor Nameplate Rating	**150**	hp
7	Drive Motor Nameplate Nominal Efficiency	**95.0**	percent
8	Fan Motor Nameplate Rating (if applicable)	**7.5**	hp
9	Fan Motor Nameplate Nominal Efficiency	**90.5**	percent
10	Total Package Input Power at Zero Flow[e]	**44.4**	kW[e]
11	Total Package Input Power at Rated Capacity and Full Load Operating Pressure[d]	**135.8**	kW[d]
12	Specific Package Input Power at Rated Capacity and Full Load Operating Pressure[g]	**19.7**	kW/100 cfm[g]

NOTES:
a. Measured at the discharge terminal point of the compressor package in accordance with the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217). ACFM is actual cubic feet per minute at inlet conditions.
b. The operating pressure at which the Capacity (Item 3) and Electrical Consumption (Item 11) were measured for this data sheet.
c. Maximum pressure attainable at full flow, usually the unload pressure setting for load/no load control or the maximum pressure attainable before capacity control begins. May require additional power.
d. Total package input power at other than reported operating points will vary with control strategy.
e. Tolerance is specified in the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217).
f, g. Tolerance is specified in the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217) as follows:

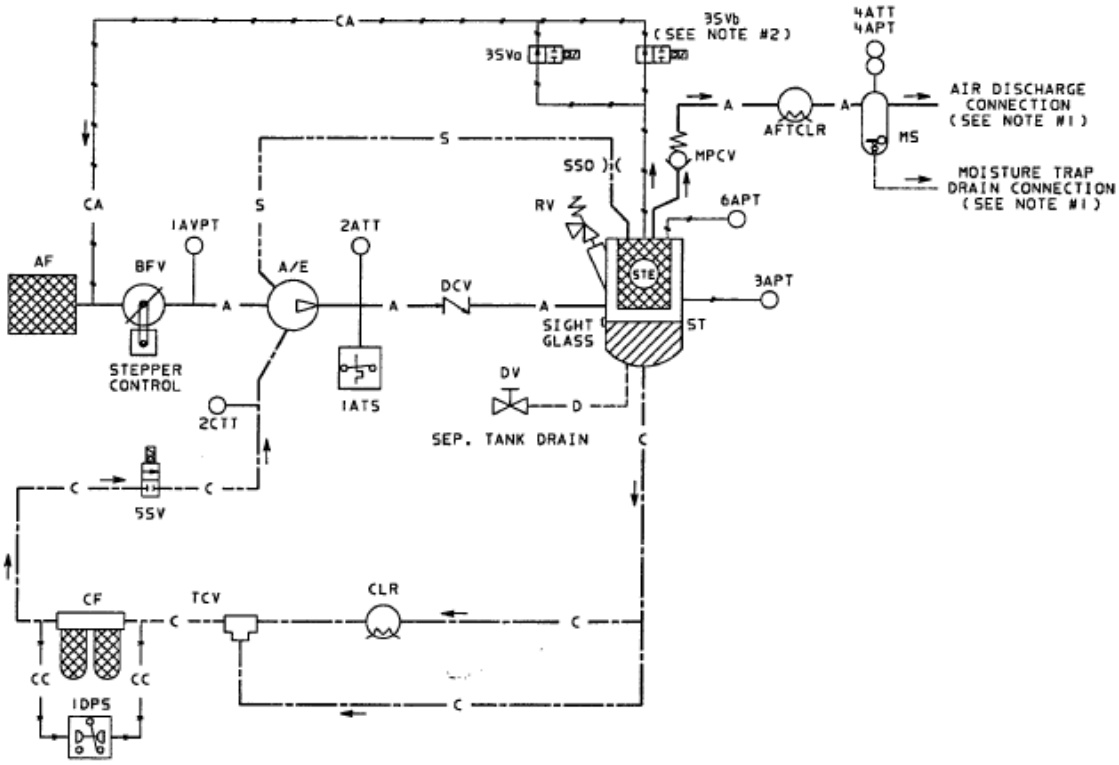
Member
COMPRESSED AIR CAGI AND GAS INSTITUTE ®

Volume Flow Rate at specified conditions		Volume Flow Rate[f]	Specific Energy Consumption[g]
m^3 / min	ft^3 / min	%	%
Below 0.5	Below 15	+/- 7	+/- 8
0.5 to 1.5	15 to 50	+/- 6	+/- 7
1.5 to 15	50 to 500	+/- 5	+/- 6
Above 15	Above 500	+/- 4	+/- 5

This form was developed by the Compressed Air and Gas Institute for the use of its members.
CAGI has not independently verified the reported data.

Form Number ROT 030 Rev 4 1-05

Appendix B: Packaged Compressor Parts



LEGEND

ABBR	DESCRIPTION
A/E	AIREND
ST	SEPARATOR TANK
STE	SEPARATOR TANK ELEMENT
AF	INLET AIR FILTER
BFV	AIR INLET CONTROL VALVE
DCV	DISCHARGE CHECK VALVE
MPCV	MINIMUM PRESSURE CHECK VALVE
RV	PRESSURE RELIEF VALVE
CF	COOLANT FILTER
TCV	THERMOSTATIC CONTROL VALVE
DV	COOLANT DRAIN VALVE
35Vo,b	BLOWDOWN SOLENOID VALVE
5SV	COOLANT STOP SOLENOID VALVE
CLR	COOLANT COOLER
AFTCLR	AFTERCOOLER
MS	MOISTURE SEPARATOR
SSO	SEPARATOR SCAVENGE ORIFICE
1AVPT	INLET VACUUM PRESSURE TRANSDUCER
3APT	SEPARATOR TANK WET SIDE AIR PRESSURE TRANSDUCER
4APT	DISCHARGE AIR PRESSURE TRANSDUCER
6APT	SEPARATOR TANK DRY SIDE AIR PRESSURE TRANSDUCER
2CTT	COOLANT TEMPERATURE SENSOR
2ATT	AIREND DISCHARGE AIR TEMPERATURE SENSOR
4ATT	PACKAGE DISCHARGE AIR TEMPERATURE SENSOR
1ATS	HIGH AIR TEMPERATURE SWITCH
1DPS	COOLANT FILTER DIFFERENTIAL PRESSURE SWITCH

NOTES:

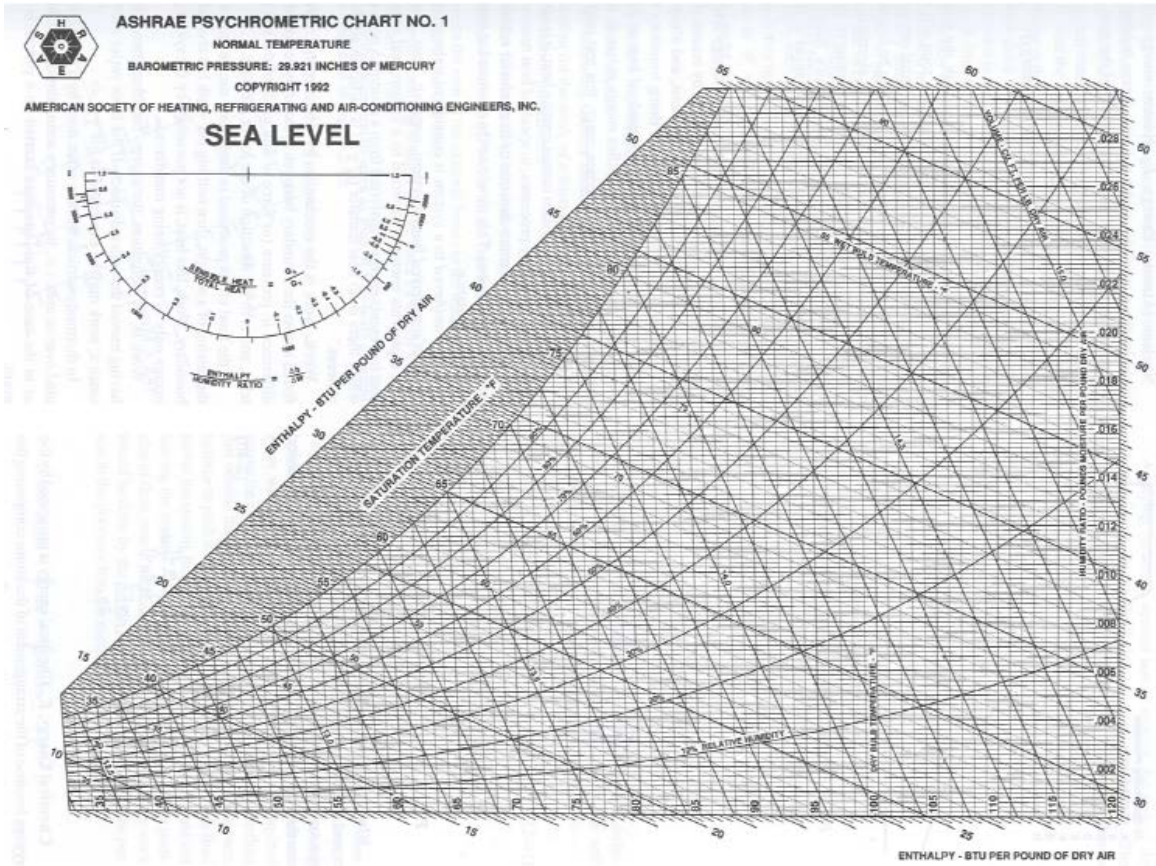
1. FOR CUSTOMER CONNECTIONS SEE FOUNDATION PLAN OF UNIT.
2. 35Vb REQUIRED ON 350-450HP ONLY.

PIPING LEGEND

A	AIR PIPING
C	COOLANT PIPING
CA	CONTROL AIR PIPING
CC	CONTROL COOLANT PIPING
S	SCAVENGE PIPING
D	DRAIN PIPING

This schematic is from Ingersoll Rand's Industrial Technologies website (25)

Appendix C: Psychrometric Chart



Appendix D: Data Analysis Weeks Two – Fourteen

Week two data

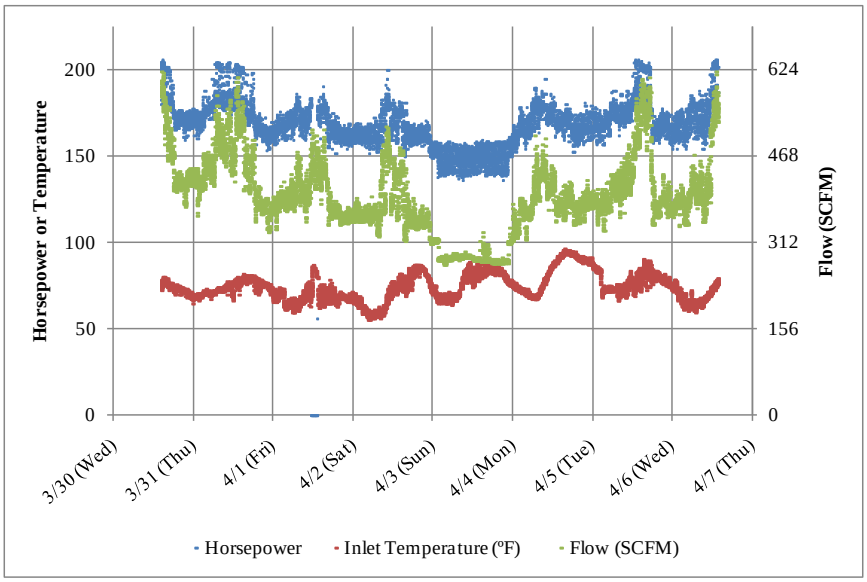


Figure: Second Week Compressor Data

Table: Second Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391

Week three data

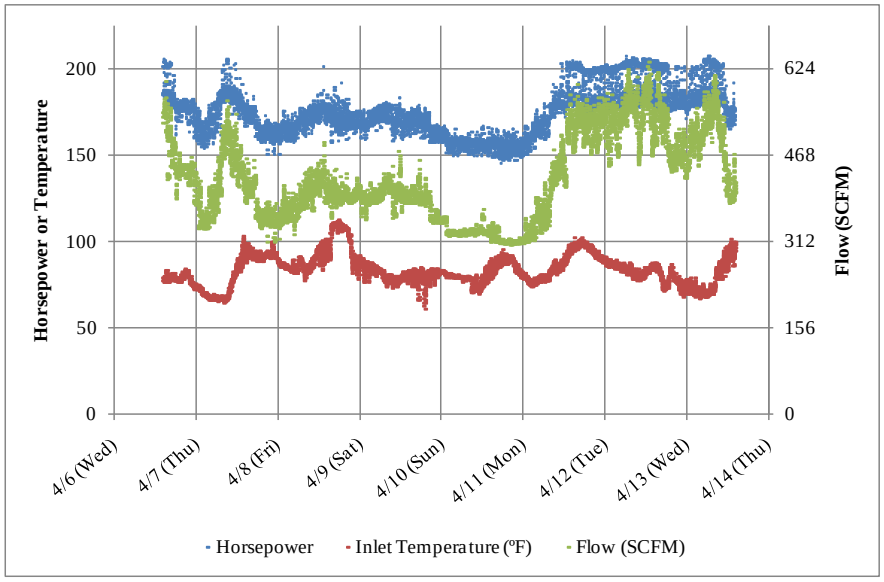


Figure: Third Week Compressor Data

Table: Third Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422

Week Four Data

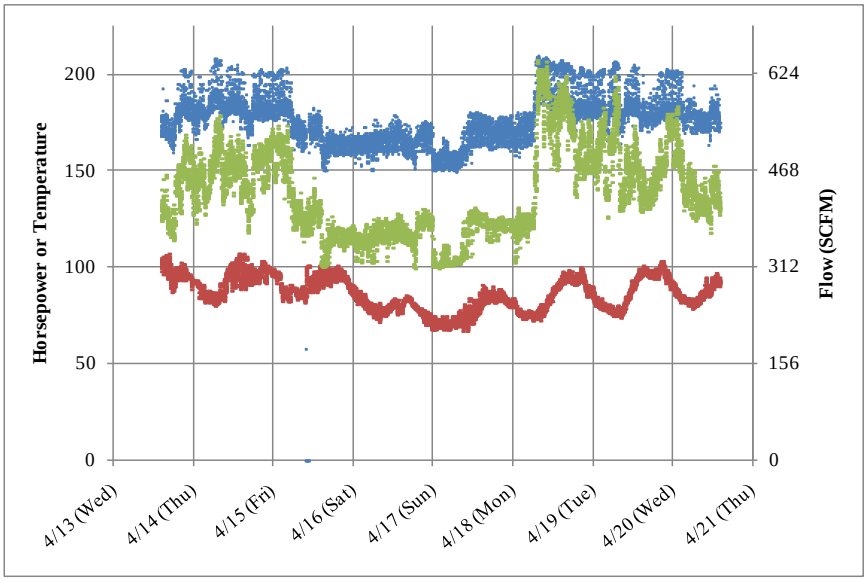


Figure: Fourth Week Compressor Data

Table: Fourth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427

Week Five Data

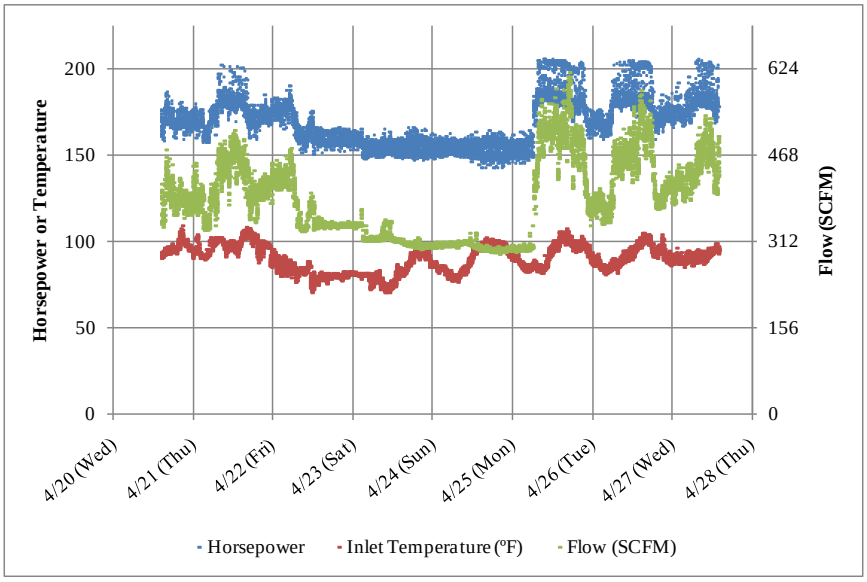


Figure: Fifth Week Compressor Data

Table: Fifth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384

Week Six Data

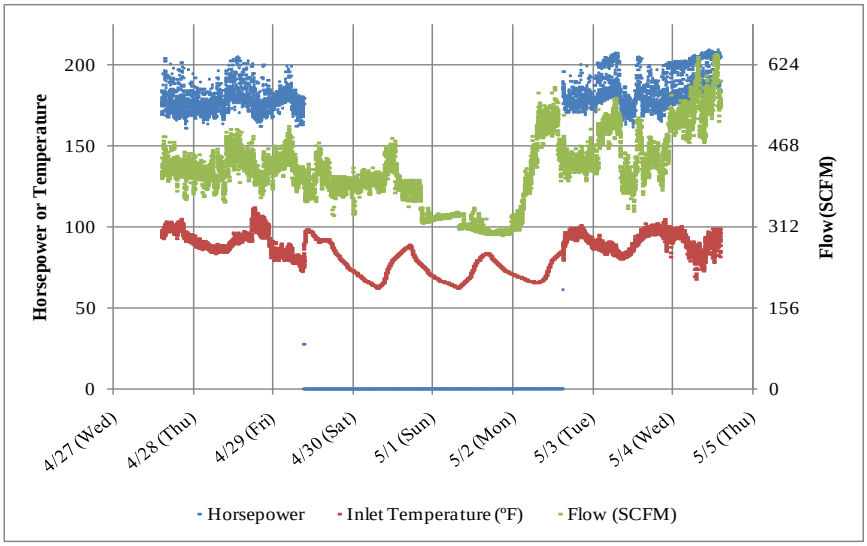


Figure: Sixth Week Compressor Data

Table: Sixth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451

Week Seven Data

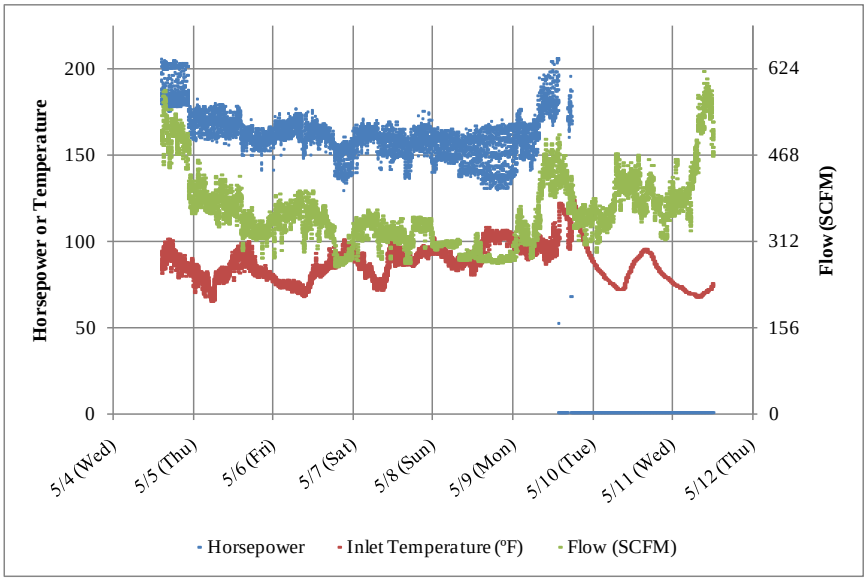


Figure: Seventh Week Compressor Data

Table: Seventh Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346

Week Eight Data

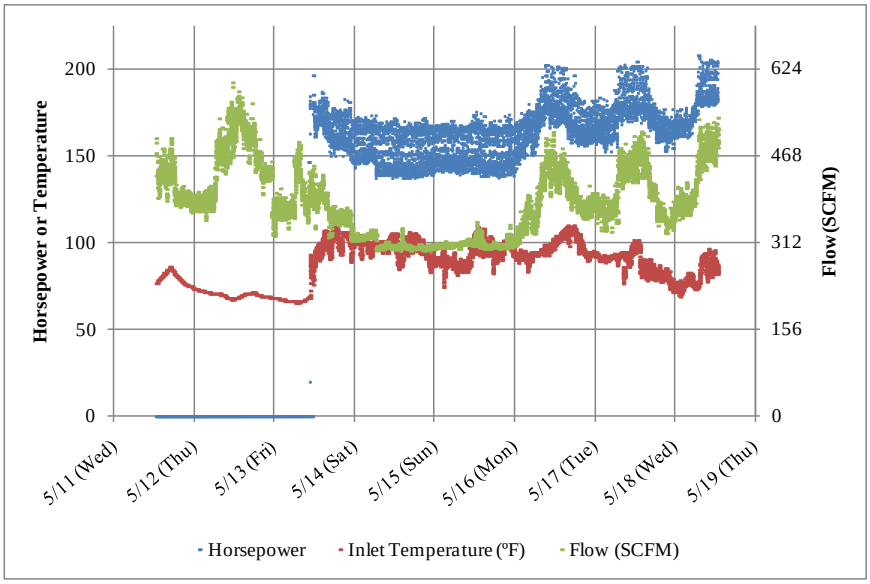


Figure: Eighth Week Compressor Data

Table: Eighth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358

Week Nine Data

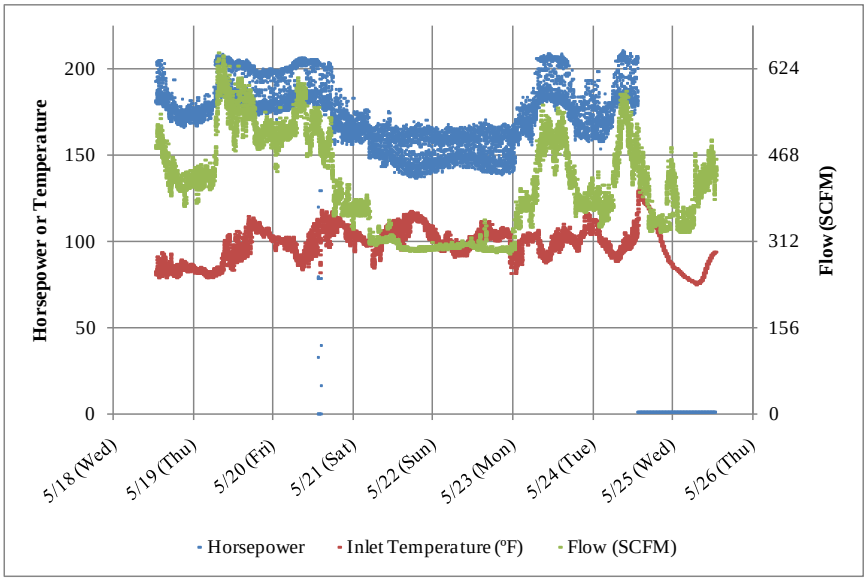


Figure: Ninth Week Compressor Data

Table: Ninth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410

Week Ten Data

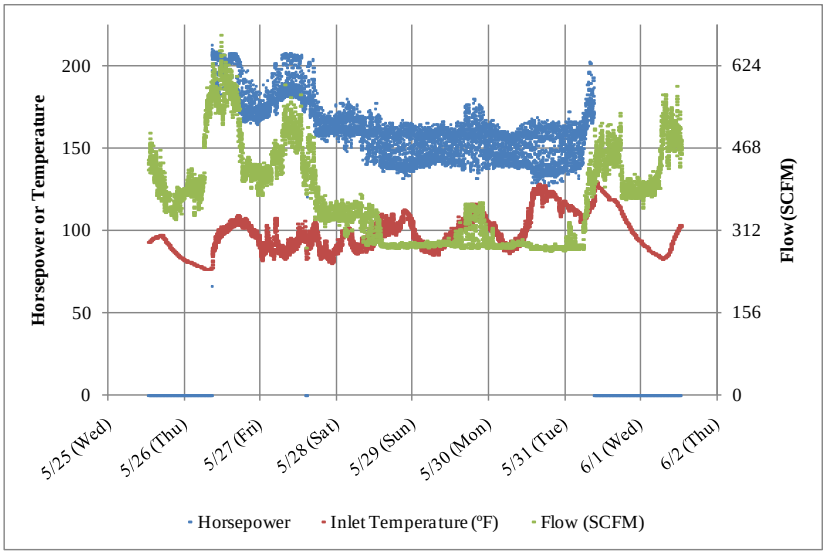


Figure: Tenth Week Compressor Data

Table: Tenth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350

Week Eleven Data

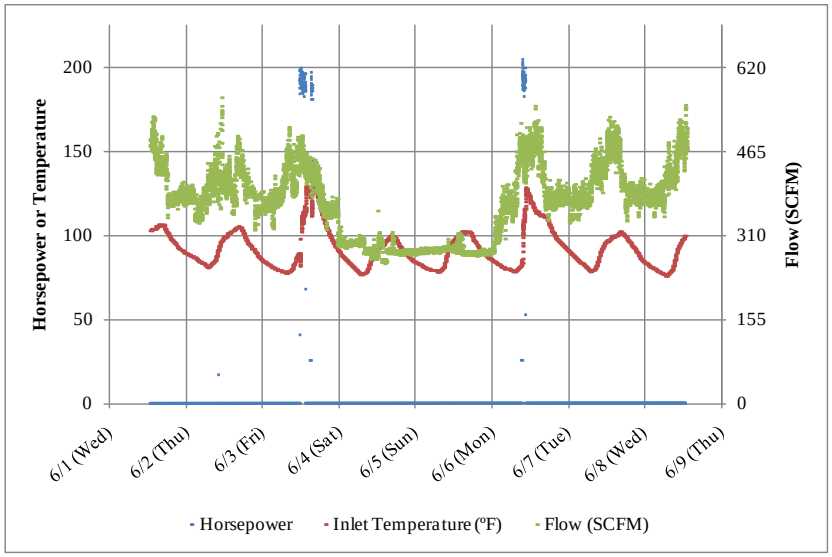


Figure: Eleventh Week Compressor Data

Table: Eleventh Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350
11	3.00	204	41	188	119	82	108	494	403	446

Week Twelve Data

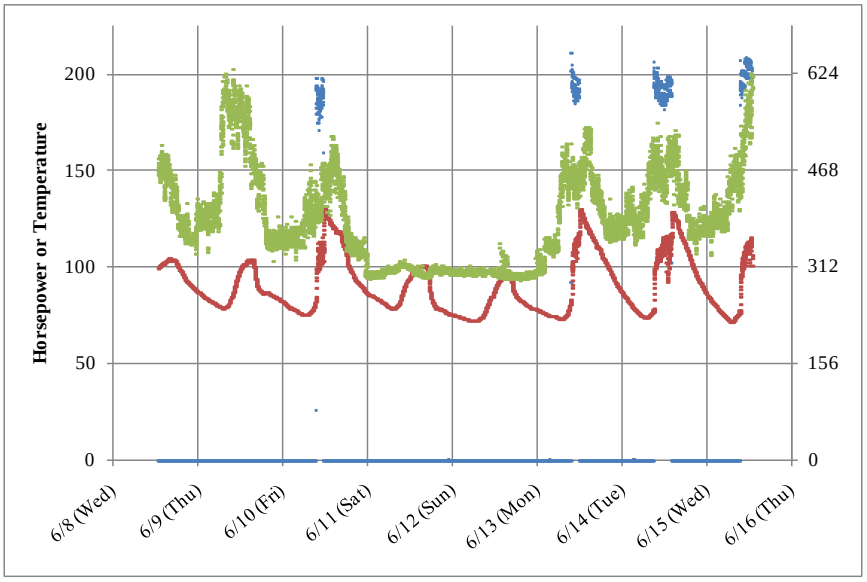


Figure: Twelfth Week Compressor Data

Table: Twelfth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350
11	3.00	204	41	188	119	82	108	494	403	446
12	12.5	211	92	193	119	76	107	622	383	475

Week Thirteen Data

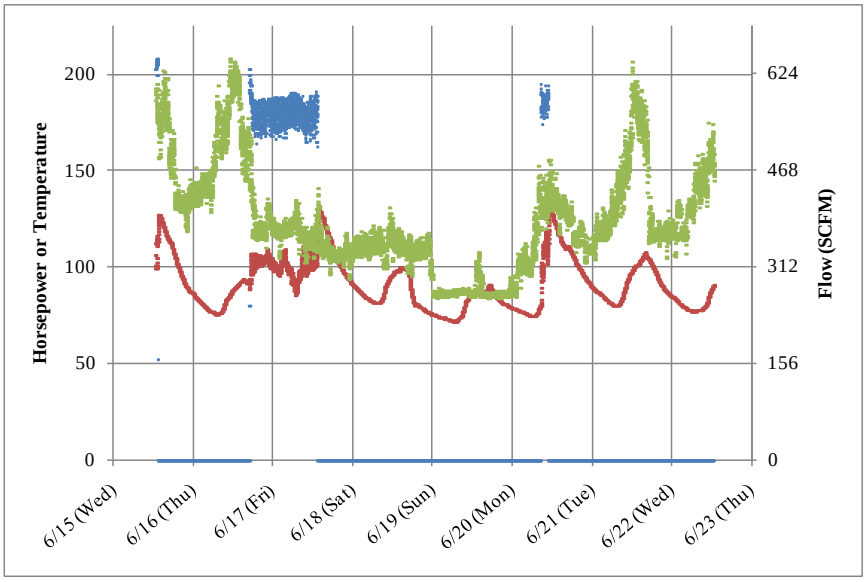


Figure: Thirteenth Week Compressor Data

Table: Thirteenth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350
11	3.00	204	41	188	119	82	108	494	403	446
12	12.5	211	92	193	119	76	107	622	383	475
13	23	208	52	180	120	80	102	607	318	380

Week Fourteen Data

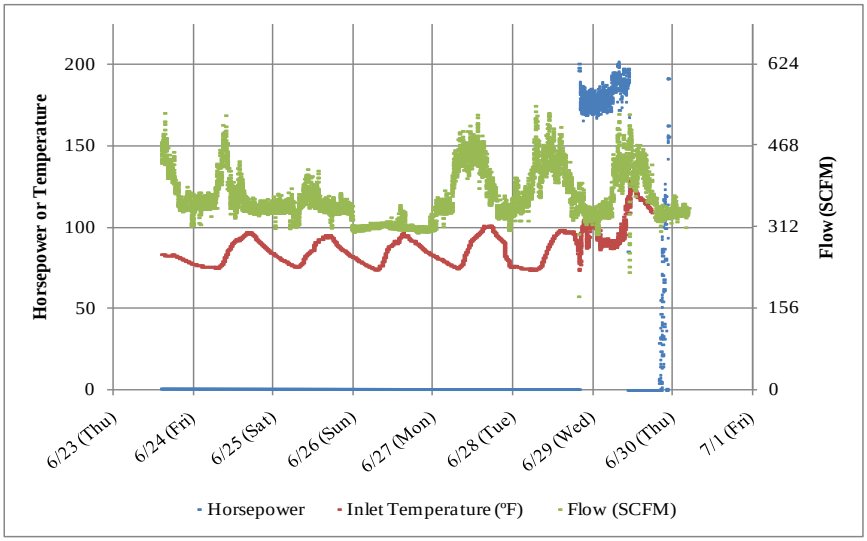


Figure: Fourteenth Week Compressor Data

Table: Fourteenth Week Stats

Wk	Operation Time (hrs.)	Horsepower			Temperature (ºF)			Flow (SCFM)		
		High	Low	Avg.	High	Low	Avg.	High	Low	Avg.
1	151	206	141	172	96	58	73	608	303	414
2	166	206	56	169	96	55	74	618	270	391
3	168	208	146	175	112	61	84	634	305	422
4	167	209	58	176	107	67	86	644	309	427
5	168	206	143	169	109	70	90	617	289	384
6	90.2	209	61	181	112	67	91	637	338	451
7	120	206	52	162	115	66	88	585	264	346
8	121	207	77	164	110	68	93	532	294	358
9	144	209	39	173	121	79	99	650	287	410
10	120	213	66	163	129	77	100	681	272	350
11	3.00	204	41	188	119	82	108	494	403	446
12	12.5	211	92	193	119	76	107	622	383	475
13	23	208	52	180	120	80	102	607	318	380
14	15.0	203	86	182	125	74	97	541	308	381