CAES: Combined Compressed Air Energy Storage

Final Design

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Executive Summary

The compressed air energy storage (CAES) project is based on the concept of storing renewable energy in environmentally safe and friendly manners. The project is sponsored by the Center for Advanced Power Systems (CAPS) and the Keuka Wind farm in Interlachen, FL. The overall concept is to harness and store wind energy during off-peak hours when the demand for electrical power is lower. The main goal of the project is to research current CAES systems that operate on the larger MW scale and determine if similar technology can be implemented for small scale applications.

For this project, wind turbines provide the power input to run a compressor. The compressed air is stored underground in a large pressure vessel, which is then used to run an air motor and generator to produce and electrical power output. For this small scale application, wind turbines rated at 20, 50, 100, and 250 kW are used for the input power. A steel pressure vessel with a volume of 11, 310 cubic feet and a working pressure of up to 200 psi is provided at the Keuka facility and was used for analysis. For each power input, an appropriate compressor air end was selected for the design. For the fixed power inputs, the corresponding initial fill times of the pressure vessel is 130, 44, 21 and 6 minutes respectively. The corresponding fill time decrease significantly for secondary refills after the air motor has ran the pressure in the vessel to 70 psi.

Also for each power input, a corresponding air motor was selected. Air motors of 12 and 24 kW were selected, which can be combined for larger power outputs. These allow for run times of approximately 24 and 11 minutes respectively. For the 20 and 50 kW systems, this yields an efficiency of approximately 12% and 13%. However, this rises to 22% and 25% for refills with 70 psig in the pressure vessel. While in terms of power input, the efficiencies rise with the larger systems of 100 and 250 kW, the run times of the air motors decrease so significantly due to the increase in required air flow rate that at this time it does not seem reasonable to operate air motors. It is our recommendation that for the provided pressure vessel, the 20 and 50 kW power systems are most applicable.

While the calculated efficiencies seem low, it is important to note that compressing air is among the least efficient processes in engineering. However, the entire system is essentially run off of free and renewable wind energy. The next step in this project is to calculate the data for a true variable power input and attempt to integrate the components for testing.

Project Introduction

Renewable and sustainable energy sources have become a major topic of interest with the depletion of oil and natural gas supplies. In addition, the need for cleaner and more efficient energy processes are becoming increasingly apparent. Wind energy is an obvious choice when searching for sustainable and environmentally friendly energy sources. However, there currently lacks an efficient means of storing renewable wind energy for later use. Our project is to design a more efficient means of harnessing surplus wind energy by compressing air, storing it, and defining its later use.

The focus of this project is to identify the need for coupling wind turbines with Combined Air Energy Storage (CAES) systems. We will construct and design a system driven by wind turbines and a power generation unit to convert energy to electric power. Analysis will be done on the system performance, efficiency and energy balance. This will be done while keeping the cost and scalability of the system at a minimum while keeping efficiency high.

The CAES system will be comprised of three subsystems: a compressor, a storage device such as a pressure vessel, and an energy generator that will allow the stored compressed air to be converted to electrical energy. The primary focus of this project is efficiency of the system, while keeping the system scalable for use within large and small systems used for power generation. The system will have a variety of power inputs that include 20kW, 50kW, 100kW and 250kW.

CAES Background

Currently there are only two power plants in the world that use CAES, one in Germany and one in Alabama. However, both plants do not use a renewable energy source to power the compressor; they use excess grid electricity as a power source. There is currently a project in Iowa that will use wind turbines such as the type our project is focused on. The systems in use currently generate 290 MW and 190 MW respectively. Our system will not generate as much power because our system is focused on more local small scale use rather than power distribution.

The current power plants utilize abandoned mines or empty caverns as their pressure storage area. These vessels are able to store massive volumes of air. For example, the plant in

Germany is able to store approximately 300,000 m^3 of air at a pressure between 700 psi to 1000 psi but operating pressure is around 600 psi. When the extra power is needed, air is released from the cavern and injected into a gas turbine which is connected to a motor-generator. The motor generator functions as a two in one machine; as a motor to drive the compressor during off peak hour then as a generator when the extra power is needed. The air is compressed for around 8 hours and then is able to be used for 2 hours. The total power efficiency of the plant is approximately 40-50%. A simple diagram of the plant is shown below.

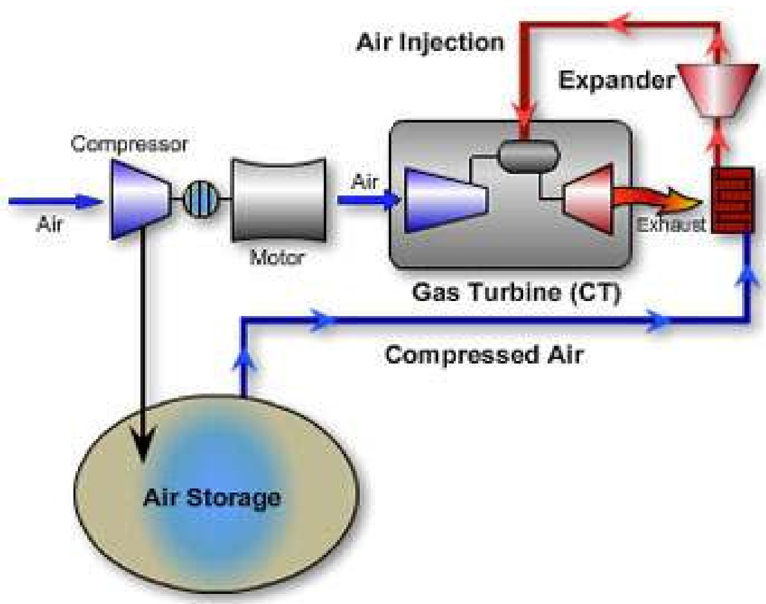

Figure 1 -Diagram of German CAES Power Plant

Wind Turbines

Wind turbines come in all shapes and sizes, from lift based turbines that mimic the profile of an airplane wing to create rotational motion to drag based turbines that rely on the wind to push the blade. The turbines our system will utilize are known as rim based turbines as shown in Figure 2 below. Whereas the larger lift based wind turbine require approximately 12 mph of wind to begin generating electricity and has the generator at the center or hub. The rim based turbines only require approximately 2 mph wind to begin generating electricity and the generator is at the base of the turbine.

Figure 2-Kueka Wind Rim based wind turbine

Wind Data

The compressed air energy storage system is dependent on the power output provided by the Keuka Wind turbine based out of Interlachen Florida. The power generation of the wind turbine at Keuka is dependent on the wind speed that is available; therefore wind data for the area must be analyzed in order to determine the equivalent power outputs. Wind data that has been averaged over the past 25 years is shown below in Figure 3 for Gainesville, Florida which is in close proximity to Interlachen. From the data it can be noted that an average yearly wind speed of 6.3 miles per hour can be expected. So the wind turbine should be capable of producing power in the range of 20 kW to 250 kW as was expected due to the design of the wind turbine. The drag wind turbine is capable of operating at wind speeds as low as two miles per hour, meaning that at our average speed a sizeable amount of power will be created for consumption, and for use within the CAES system during the off peak hours.

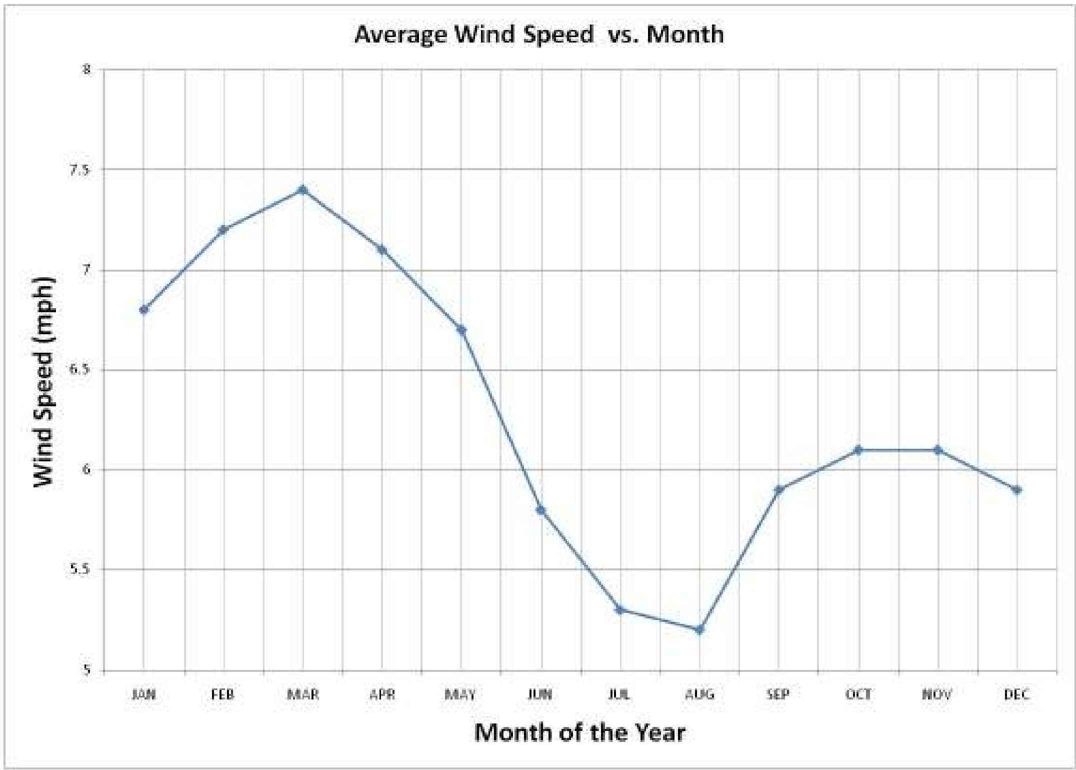

Figure 3- Wind Data for Gainesville Florida

Compressors

For the application of compressed air energy storage the efficiency of the compressor within the unit will define the success of the system. Due to the location of the wind turbine at Interlachen, Florida an average wind speed of 6.3 miles per hour can be expected. Therefore a small amount of power will be usable as energy to run the compressor during the off peak hours. For the purpose of this design project we will design the compressor to be capable of operating at an input power range of 20kW to 250kW. At these power ranges we can expect a low flow rate into the compressor; therefore as can be noted in Figure 4 below, a rotary type compressor will be best suited for our system. Within the rotary type of compressors there are two different variations of compressor that will be considered, helical-lobe and sliding vane compressors. However another type of compressor exists that has been created since Figure 4 was created. The guided rotor compressor is a relatively new positive displacement device that has become the industry standard for many compressions applications since its inception in the early 1990’s. Each of these three variations will be described below in detail.

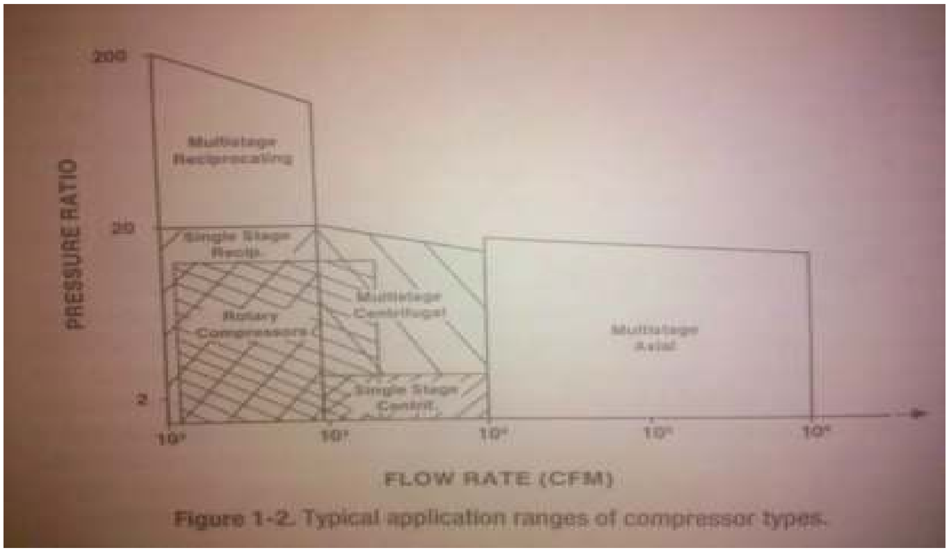

Figure 4-Typical application ranges of compressor types

Helical-Lobe Compressors

Helical-lobe compressors are a positive displacement type of compression device they utilize rotating helical screws in mess to compress the gas. Typically these compressors are referred to as a screw compressor due to the design which can be seen below in Figure 5. Helical-lobe compressors come in two forms dry and flooded. In the dry form a timing gear set is required to reduce wear on the rotors; the flooded form utilizes a liquid media to keep the rotors from contacting one another. The dry configuration has a capacity range of approximately 500 to 35,000 cubic feet per minute (CFM). Discharge pressure is limited to 45 psi in single stage configuration with atmospheric suction. However, supercharged or multistage applications have an obtainable discharge pressure of 250 psi, with a maximum obtainable pressure ratio of 21 to 1.

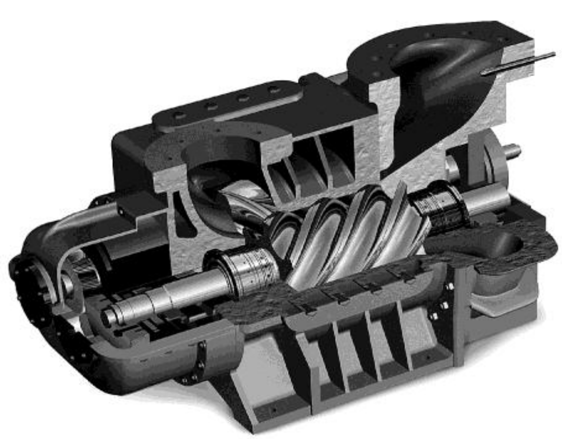

Figure 5- Helical-Lobe Compressor

Sliding Vane Compressors

Sliding vane compressors are another form of a positive displacement compressor, they use a single rotating element to compress gas. The rotor of the sliding vane compressor is mounted eccentric to the center of the cylinder portion of the casing and is slotted and fitted with vanes which can be seen below in Figure 6. The vanes are free to move in and out within the slots as the rotor revolves. Gas is trapped between a pair of vanes as the vanes cross the inlet port; the gas is then moved and compressed circumferentially as the vane pair moves toward the discharge port. The port locations control the pressure ratio. The design requires an external source of lubrication to ensure efficiency.

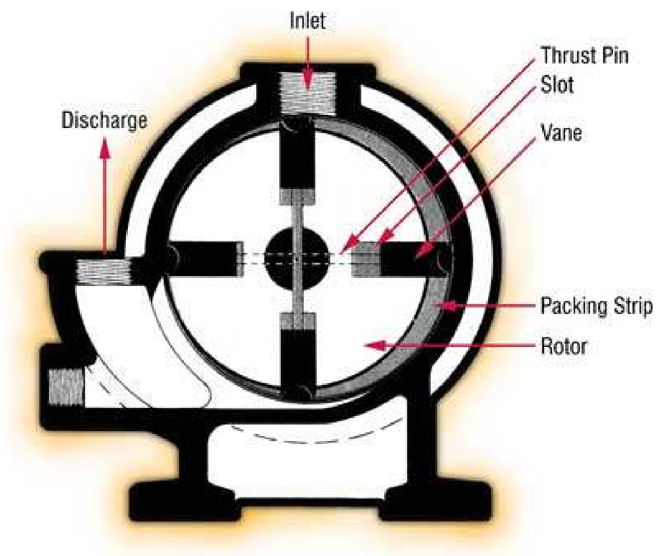

Figure 6- Sliding Vane Compressor

Sliding vane compressors are commonly used as vacuum pumps as well as compressors; with volume flow rates from 50 to 6,000 cfm. A single-stage compressor is capable of generating discharge pressures up to 50 psi, while in booster service units can produce up to 400 psi.

Guided Rotor Compressors

The guided rotor compressor is a rotary positive displacement device that utilizes a trochoid curve to define its basic compression volume, trochoidal design can be seen below in Figure 7. A single rotor compressor assembly is made up of a trochoidal housing, a rotor, roller seals, suction side plate, discharge side plate, crankshaft, rotor bearing, main bearings, end covers, and a ceramic face seal. The guided rotor compressor does not require a timing gear and does not require speed increasers to achieve cost effective delivery. Guided rotor compressors are capable of high adiabatic efficiencies ranging from 75% to 88%, at pressure ratios ranging from 2.8 to 4.6. Also, in comparison to helical-lobe and sliding vane compressors the guided rotor compressor is much smaller in size as well as being a quieter running device due to its balanced design.

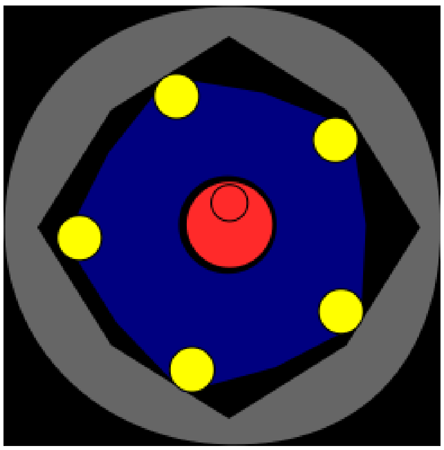

Figure 7- Trochoidal Design

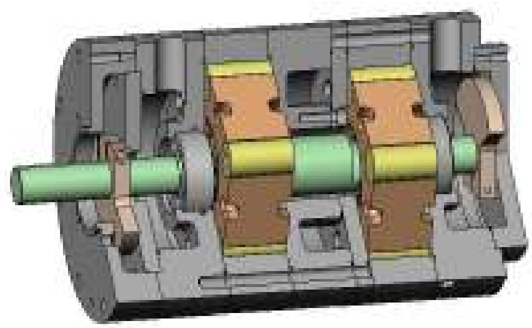

Figure 8- Guided Rotor Compressor

Compressor Decision

In order to fulfill the requirements of the system the compressor chosen must have a high adiabatic efficiency, a good pressure ratio, and be within our budget. Since a compressor is available for use at the Keuka wind facility it may be used if it fulfills these requirements, however due to the lack of information on the compressor this research on compressors has been completed. From the results of our research it has been determined that the helical-lobe compressor would be the best fit for our application due to its variable input power range and excellent pressure ratio. The guided rotor compressor does boast equivalent statistics to the screw compressor however due to its relatively recent conception the cost of such a device is well out of our price range. Therefore ideally a Helical-Lobe or screw compressor should be integrated into our system to compress air for storage.

Generation

For the smaller scale applications of 20 kW and 50 kW we believe that due to the constraints of the provided pressure vessel, a gas turbine or micro turbines would not be feasible to operate due to high pressure and volume requirements. For these power ranges we believe the proper type of generation would be to use an air motor. Furthermore, for the 100 kW and 250 kW power ratings we may be able to connect a few air motors together and combine a gear set to turn the electric generator. However, this idea is also limited by our pressure vessel as will be shown later.

Air Motors

Air motors come in several different varieties such as vane, piston and turbine. Each has their own advantages and disadvantages according to what to customers need is. Vane motors operate similar to a rotary internal combustion engine. They are also the same design as some of the compressor discussed earlier; they just operate in reverse when compressing air. A slotted rotor rotates eccentrically in the chamber formed by the cylinder and cylindrical end plates. Since the rotor is off-center and its diameter smaller than that of the cylinder, a crescent-shaped chamber is created. This is shown in Figure 9-Vane Air MotorFigure 9 below where 'a' is an intake port and 'b' and 'c' are exhaust ports, rotation is clockwise. The Vanes are the brown slots and are allowed to extend to provide a seal between the cylinders inside surface. These motors are best suited for low to medium power outputs ranging from a few quarters of a horsepower to several horsepower. Also they have an operating range of 100 to 25,000 rpm but they provide more power per weight than a comparable piston air motor.

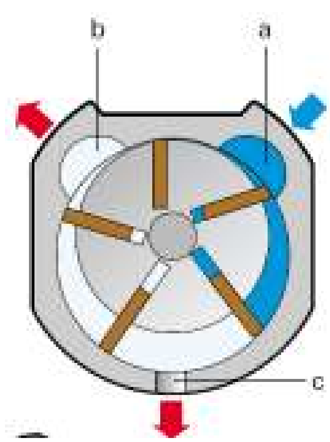

Figure 9-Vane Air Motor

Piston Air Motors

Piston air motors operate just like their internal combustion counterparts except they replace the fuel for compressed air. Piston air motors are best suited for applications requiring high power, high starting torque, and accurate speed control at low speeds. With the speed control however, cost can rise due to the complexities involved. Also, these motors require excellent lubrication and higher maintenance due to size and number of parts involved. But they can output as much as 23 kW if the supply pressure is sufficient. Shown in Figure 10 below is a current model of radial piston air motor from Huco Dynatork that has been shown to use up to 80% less air than a comparable vane air motor.

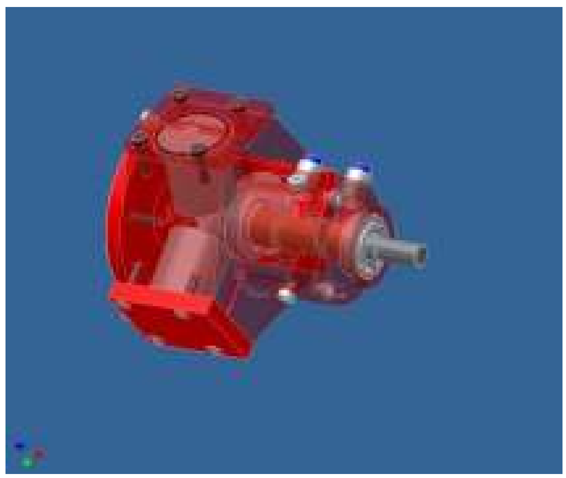

Figure 10-Huco Dynatork Radial Piston Air Motor

Performance of all types of air motors is highly dependent on the inlet pressure. Maintaining a fairly constant inlet pressure will assure the highest efficiency possible for the air motors as well as the optimum power output. This will be done by selecting the proper operating pressure according to the ability of the supplied pressure vessel.

Concept

Our project will not be to the scale of the current CAES power plants. Those power plants produce hundreds of Mega Watts whereas our system is geared to produce up to 200 kW. The system will start with a wind turbine. This wind turbine is supplied to us from our sponsor, Kueka Wind. As mentioned before we will have four different wind turbines with power ratings of 20 kW, 50 kW, 100 kW and 250 kW. For this portion of the project we will be assuming that the compressor will have a power input equal to what the wind turbine is rated at. Next semester we will research how the power truly fluctuates with data provided from the wind turbines and make adjustments for the compressors to operate accordingly.

Compressor Selection

As was discussed previously rotary twin screw air compressors will provide the characteristics that are necessary for this system to operate efficiently and will define the success of our design. Therefore a copious amount of research was conducted in order to decide what compressor air end manufacturer produces the best product for this design. Through the process of our research it was found that the manufacturer Quincy Compressors produces very reliable and durable air ends that are capable of fulfilling our requirements.

Quincy's products were chosen because their compressors have a life expectancy of 100,000 hours when routinely maintained every 4,000 operational hours. Since our system will only operate on the off peak power hours we can expect that the compressor will need maintenance at most twice a year which is very acceptable. Quincy's product line covers a vast range of readily available air ends with several compressor lines that will suit our power input requirements.

In order to size the appropriate air end for each individual power input the powers had to be converted from kilowatts to horsepower due to all of Quincy's products being rated in horsepower. The new power inputs are 26.8, 67, 134, and 335 HP. As can be expected to acquire a motor capable of operating at these odd power ratings it would have to be custom built, and as our project's budget is low this seems inappropriate so the power inputs have been normalized to 25, 60, 125, and 300 HP these values were brought down in order to ensure that sufficient power is supplied.

Once these normalized power inputs had been found two statistics sheets, shown in Appendix I, were used to make a decision of which particular air end would be best suited for each power, the results are shown below in Table 1.

Table 1-Compressor air end statistics for each input load

Compressor Model	QSB/T 25	QSI-250	QSI-540	QSI-1400
Power Input (hp)	25	60	125	300
Full Load Pressure (psi)	150	125	125	125
Maximum Pressure (psi)	165	150	150	150
Full Load Capacity (acfm)	87	256	540	1400
Manufacturer	Quincy	Quincy	Quincy	Quincy

From the compressor statistics shown above it should be noted that the volumetric flow rate shown in actual cubic feet per minute (acfm) increase with each individual increase in power input, this should cause our efficiency to increase as the systems become more powerful. Also the maximum operating pressure that we can expect is about 150 psi which is slightly lower than we had hoped to achieve, however this sacrifice was made in order to allow for higher flow rates out of the compressors and into the storage device thereby decreasing the filling time. These filling times along with the power inputs will allow us to calculate our energy usage to compress air and with each reduction in filling time our efficiency should increase.

Pressure Vessel

The pressure vessel provided at the Keuka wind farm facility is a steel pipe with welded caps buried underground. The given dimensions for this vessel are a length of 100 ft, diameter of 12 ft and a thickness of ¾”. Using this data, we were able to calculate the allowable pressures the vessel would be able to withstand. The yield stress of steel is approximately 30 ksi which translates to a yield pressure of 312 psi (Appendix III). The maximum allowable stress for steel used in designing pressure vessels is 16.9 ksi. The recommended operating pressure was calculated using the ASME code for boilers and pressure vessels. This includes the appropriate safety factors as decided by the ASME. This yields a maximum recommended operating pressure of 176 psi (Appendix III). However, due to the fact that the pressure vessel is buried underground, the surrounding pressure will be greater than the atmospheric pressure. This allows the vessel to safely withstand a larger pressure. According to the Keuka facility, the pressure vessel will operate safely up to 200 psi.

In calculating the energy output and efficiencies of the integrated CAES system, it was important to calculate the fill and unload times of the pressure vessel based upon the flow rates of the compressor and air motor. In these calculations, the ideal gas law and a constant temperature in the pressure vessel was assumed using the Continuity Equation (Appendix III). The fill unload times of the pressure vessel through the corresponding compressor and air motors are shown in Appendix III.

Power Generation

As mentioned before, we have decided that air motors will be the best way to extract power from our system. This is because of the limitations of the pressure vessel and the volume of air that is available. It was recommended to us by our sponsors to use different size air motors or at least different configurations for the different power ratings given to us. For example, for the 20 kW wind turbine we should use an air motor of approximately 20 kW. As mentioned before, vane air motors are best suited for low power high speed applications and piston air motors are suited for high power medium speed applications.

Within the selection of piston air motors there are two sections: Axial and Radial. Axial piston air motors are ideal for limited space mounting, they have a much more complex design and a greater cost than air motors but, their maximum power output is about 4 HP. Radial piston air motors are more robust and have higher starting torques and smoother power output due to the radial design. Power outputs range from a few horsepower to a maximum of about 35 HP (26 kW). Therefore, Radial piston air motors are ideal for our application due to their power output,

Figure 11- Ingersoll Rand MMP150

Due to pricing of air motors and availability we were led to the Ingersoll Rand MMP150 Air Motor. This air motor gives an output of 16 hp which is approximately 12 kW. We were told that the electric generator at Keuka Wind needs a minimum of 7 kW of power to start generating electricity. Therefore, we believe that this MMP150 will meet our needs for this power rating. For the 50 kW power rating we decided to use a Tonson M18 Air Motor which outputs approximately 31 hp which equates to about 23 kW. This air motor would also suffice for the 20 kW power rating if needed but with the increased power out comes a dramatic increase in price which would be something that we will address later in the cost analysis section of this report.

Figure 12-Tonson M18 Air Motor

The Ingersoll Rand air motor requires 425 CFM of air flow with a maximum operating pressure of 90 psi. After talking to the Ingersoll Rand distributer they recommend operating the air motor at 90 psi in order to attain the rated power out, also the recommend not running the motor below 70 psi so that we do not induce any damage to the motor and still have a sufficient power output. With these pressure limits and shutoff point arises the need to control the flow somehow; this will be done by means of pressure regulators and solenoid valves. The regulators will maintain 90 psi to the motor and once pressure falls to 70psi the solenoid valve will activate and close the piping to the air motor.

Run time analysis was done by the same was the fill time for the pressure vessel was. The difference is that the vessel would go from a low pressure to a high pressure and the air motor calculation will go from a high pressure to a low pressure. This is shown in Appendix III. To summarize the run time for the Ingersoll Rand MMP150 12 kW motor came to be approximately 23 minutes and the Tonson M18 23 kW motor can run for approximately 11 minutes.

Our team ran into problems when trying to find air motors above 30 hp. They are very rare and extremely difficult to find, not to mention extremely expensive. When calculating the power out for the higher power ratings using multiple air motors it was found that the motors would only be able to run for less than 5 minutes. This small time would add a dramatic cycling fatigue to the air motors which will reduce their lifetime. These calculations can be found in Appendix III.

Cost Analysis

The provided budget for the CAES project is $2500. After researching the components needed to integrate the CAES system, we found that the cost of the individual components dramatically exceeds our given budget. After consulting with distributors and vendors, even with discounts the MMP150 air motor is priced at $9463. The air compressor for the 20 kW power rating was quoted at $5400. In addition, the pressure regulating devices and solenoid valves totals approximately $900. These prices are summarized below in Table 2. Given that the total for the components far exceeds our budget, we are making recommendations to the Keuka facility for the individual components required to complete the CAES system.

Table 2-Cost Analysis

Item	Description	Quantity	Price
QSB/T 25	25 HP (20kW) Airend	1	$5,486
IR MMP150	16 HP (12 kW) Air Motor	1	$9,463
Air Centers of FL	Pressure Regulator	2	$600
Air Centers of FL	Solenoid Valve	2	$300
Travel to Kueka	Fuel Cost	2	$100
		Total	$15,949
		Budget	$2,500

Conclusion

Wind is one of the easiest renewable energies to use on earth. However, the major concern is how to store this energy when the wind is blowing yet its energy is not immediately required. This is where Compressed Air Energy Storage (CAES) enters. Small systems such as the one described here can be readily installed and used for small power ranges such as 20, 50, 100, and 200 kW.

When the wind turbines are creating electricity but it is not needed, that power will drive the Quincy Air compressor and store the energy in the form of compressed air. As discussed, for a Kueka Wind Turbine rated at 20 kW, it will take approximately 2 hours to fill the pressure vessel to the proper pressure. Afterwards the pressurized air is released to power an air motor which will in turn power an electric generator for approximately 24 minutes. This process can be duplicated for power levels of 50 kW, 100 kW, and 200 kW. At different power ratings, different air compressors and air motors were selected accordingly. The difference between the power levels will be noticed in the fill times, run times and efficiencies of the system. We have noticed that as the power rating increases the run time drops dramatically due to the pressure vessel limitations. This will need to be analyzed further to find any possible solutions if we must keep the same pressure vessel.

While the calculated efficiencies of 22% and 25% at the power ratings of 20 kW and 50 kW respectively seems discouraging, we must remember that this was done using renewable energy. Essentially, the power it took to create that 22% efficiency was free. It did not come from the grid or fossil fuels and we did not pay for it.

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Appendix I- Air Compressor Specifications

Compressor Model		QSI-245	QSI-300	QSI-370	QSI-500	QSI-600	QSI-750	QSI-1000	QSI-1250	QSI-1500
Motor Horsepower	hp	50	60	75	100	125	150	200	250	300
Motor RPM	rpm	1800								
Full Load Capacity	acfm	245	288	370	500	623	757	1010	1264	1515
Min Operating Pressure	psig	75								
Sound Data, Unenclosed (@ 1 meter) air/water cooled	dB(A)	82/80	83/83	83/83	87/86	89/86	89/87	89/86	95/92	96/92
Sound Data, Std Enclosure (@ 1 meter) air/water cooled	dB(A)	76/73	79/79	79/79	83/80	87/80	87/80	85/79	91/84	93/84
Sound Data[1], Low-Sound Enclosure (@ 1 meter)	dB(A)	71	76	76	80	80	80	82	85	85
Dimensions[1] (approx) (L x W x H)	inch	78x48x58	78x48x66	84x52x58	92x56x60	102x56x60	116x68x76	120x76x73	132x80x89	
Weight[1] (approx)	lbs	3100	3200	3600	4400	4500	7500	9000	10300	10500

125 psig Full Load Pressure - High Capacity / 140 psig Maximum pressure

Compressor Model		QSI-245	QSI-300	QSI-370	QSI-500	QSI-600	QSI-750	QSI-1000	QSI-1250	QSI-1500
Motor Horsepower	hp	60	75	100	125	150	200	250	300	350
Motor RPM	rpm	1800								
Full Load Capacity	acfm	243	286	364	495	615	751	1003	1255	1504

125 psig Full Load Pressure - Low Horsepower[2] / 140 psig Maximum pressure

Compressor Model		QSI-220	QSI-250	QSI-335	QSI-440	QSI-540	QSI-675	QSI-925	QSI-1175	QSI-1400
Motor Horsepower	hp	50	60	75	100	125	150	200	250	300
Motor RPM	rpm	1800								
Full Load Capacity	acfm	220	256	335	440	540	675	925	1175	1400
Dimensions[1] (approx) (L x W x H)	inch	78x48x58	78x48x66	84x52x58	92x56x60	102x56x60	116x68x76	120x76x73	132x80x89	
Weight[1] (approx)	lbs	3100	3200	3600	4400	4500	7500	9000	10300	10500

High Pressure[2]

Compressor Model		QSI-245	QSI-370	QSI-500	QSI-750	QSI-1000	QSI-1250
Motor Horsepower	hp	75	100	150	200	300	350
Full Load Capacity @175 psig, (190 max psig)	acfm	227	351	468	715	951	1216
Motor Horsepower	hp	75	125	150	250	300	
Full Load Capacity @210 psig, (225 max psig)	acfm	224	346	461	702	933	

Model: QSB/T 25

TECHNICAL DATA

Compressor Drive Motor	Nameplate HP	25	25	25
Inlet Capacity	ACFM	115	105	87
Full Load Operating Pressure	PSIG	100	125	150
Max. Operating Pressure	PSIG	115	140	165
Min. Operating Pressure	PSIG	75	75	75
Max. Ambient Operating Temp.	Degrees F	110	110	110
Min. Ambient Operating Temp.	Degrees F	35	35	35
Rotor Diameter	mm	127.5	127.5	127.5
Male Rotor Speed	RPM	2157	2031	1750
Female Rotor Speed	RPM	1448	1364	1177
Rotor Tip Speed	M/sec	14.50	13.66	11.78

COOLING DATA

Heat Rejection - Oil Cooler	BTU/MIN	938	992	974
Aftercooler	BTU/MIN	197	181	148

Appendix II- Air Motor Specifications

MMP150 Air Motors

PERFORMANCE

Model	Max. Power		Speed at Max. Power	Free Speed	Starting Torque		Stall Torque		Air Consumption at Max. Power	
	hp	kw	rpm	rpm	lb.-ft.	Nm	lb.-ft.	Nm	scfm	m³/m
MMP 150	16.0	11.9	1800	3800	61.0	82.7	78.0	105.8	425	12.0

Performance figures are at 90 psig (6.2 Bar) air pressure.

Figure 1: MMP150 Air Motor Performance
425 cfm Air Supply at 90 psi

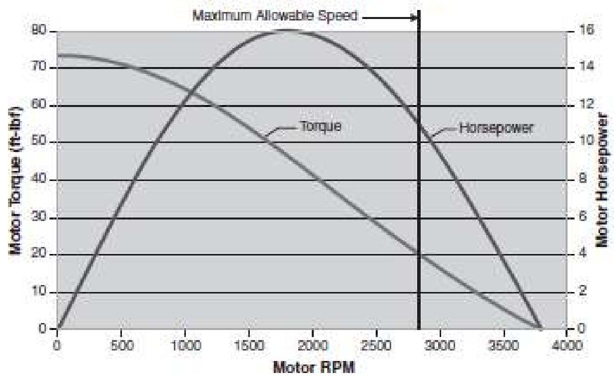
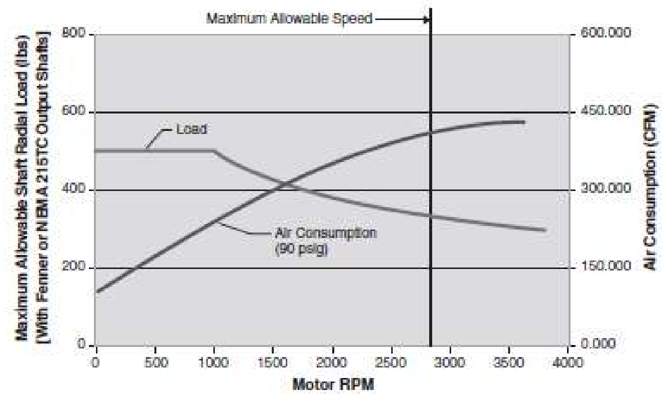


Figure 2: MMP150 Air Motor Performance



Appendix III- Calculations

Compressor Data - for 20, 50, 75, 100, 250 kW

20 kW	$Power_1 := 25hp$	$P_1 := 150psi$	$Q_1 := 87cfm$	$20kW = 26.82\cdot hp$
50 kW	$Power_2 := 60hp$	$P_2 := 150psi$	$Q_2 := 256cfm$	$50kW = 67.051\cdot hp$
100 kW	$Power_3 := 125hp$	$P_3 := 150psi$	$Q_3 := 540cfm$	$75kW = 100.577\cdot hp$
100 kW	$Power_4 := 200hp$	$P_4 := 150psi$	$Q_4 := 925cfm$	$100kW = 134.102\cdot hp$
250 kW	$Power_5 := 300hp$	$P_5 := 200psi$	$Q_5 := 1400cfm$	$250kW = 335.256\cdot hp$

Pressure Vessel - Steel pipe with welded ends (underground)

$d := 12ft$ $\quad L := 100ft$ $\quad t := \frac{3}{4}in$ $\quad \sigma_{yield} := 30ksi$ $\quad \sigma_{allow} := 16.9ksi$

$r := \frac{d}{2}$ $\quad V_{vessel} := \pi \cdot r^2 \cdot L = 11309.734ft^3$

Yield Stress

$P_{yield} := \frac{\sigma_{yield} \cdot t}{r}$ hoop stress $\quad P_{yield} = 312.5psi$

$P_{yield\_axial} := 2 \cdot P_{yield} = 625 \cdot psi$

Allowable Stress

$P_{allow} := \frac{\sigma_{allow} \cdot t}{r}$ $\quad P_{allow} = 176.042psi$

$P_{allow\_axial} := 2 \cdot P_{allow} = 352.083psi$

Note: The allowbale stress is calculated using the allowable stress for steel pressure vessels based on the ASME codes for Boilers and Pressure Vessels. Due to the fact that the pressure vessel is underground, the maximum operating pressure will be 200 psi.

Fill Time

Assumptions:
ideal gas law
negligible temperature change
initial fill with P = 0 psig

$P_0 := 0psi$ $\quad P_{max} := 150psi$ $\quad P_{min} := 70psi$

$$t_{initial1} := \frac{(P_{max} - P_0) \cdot V_{vessel}}{P_1 \cdot Q_1} \qquad t_{initial1} = 129.997 \cdot min \qquad t_{initial1} = 2.167 \cdot hr$$

initial

$$t_1 := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_1 \cdot Q_1} \qquad t_1 = 69.332 \cdot min \qquad t_1 = 1.156 \cdot hr$$

secondary

$$t_{initial2} := \frac{(P_{max} - P_0) \cdot V_{vessel}}{P_2 \cdot Q_2} \qquad t_{initial2} = 44.179 \cdot min \qquad t_{initial2} = 0.736 \cdot hr$$

initial

$$t_2 := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_2 \cdot Q_2} \qquad t_2 = 23.562 \cdot min \qquad t_2 = 0.393 \cdot hr$$

secondary

$$t_{initial3} := \frac{(P_{max} - P_0) \cdot V_{vessel}}{P_3 \cdot Q_3} \qquad t_{initial3} = 20.944 \cdot min$$

initial

$$t_3 := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_3 \cdot Q_3} \qquad t_3 = 11.17 \cdot min$$

secondary

$$t_{initial4} := \frac{(P_{max} - P_0) \cdot V_{vessel}}{P_4 \cdot Q_4} \qquad t_{initial4} = 12.227 \cdot min$$

initial

$$t_4 := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_4 \cdot Q_4} \qquad t_4 = 6.521 \cdot min$$

secondary

$$t_{initial5} := \frac{(P_{max} - P_0) \cdot V_{vessel}}{P_5 \cdot Q_5} \qquad t_{initial5} = 6.059 \cdot min$$

initial

$$t_5 := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_5 \cdot Q_5} \qquad t_5 = 3.231 \cdot min$$

secondary

Run Time - Air Motor

$P_{operate} := 90psi$

$P_{min} := 70psi$ $\quad$ $P_{max} = 150 \cdot psi$

$Q_{motor1} := 425cfm$ $\quad$ $Power_{motor1} := 11.9kW$ $\quad$ $rpm_1 := 1800rpm$

$Q_{motor2} := 900cfm$ $\quad$ $Power_{motor2} := 23.3kW$ $\quad$ $rpm_2 := 1500rpm$

12 kW

$$t_{run1} := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_{operate} \cdot Q_{motor1}}$$

$t_{run1} = 23.654\,min$

$t_{run1} = 0.394\,hr$

$$\eta_{fill1} := \frac{Power_{motor1} \cdot t_{run1}}{Power_1 \cdot t_{initial1}} = 11.615\%$$

initial fill

$$\eta_{secondary1} := \frac{Power_{motor1} \cdot t_{run1}}{Power_1 \cdot t_1} = 21.778\%$$

secondary fill

24 kW

$$t_{run2} := \frac{(P_{max} - P_{min}) \cdot V_{vessel}}{P_{operate} \cdot Q_{motor2}}$$

$t_{run2} = 11.17 \cdot min$

$t_{run2} = 0.186\,hr$

$$\eta_{fill2} := \frac{Power_{motor2} \cdot t_{run2}}{Power_2 \cdot t_{initial2}} = 13.167 \cdot \%$$

initial fill

$$\eta_{secondary2} := \frac{Power_{motor2} \cdot t_{run2}}{Power_2 \cdot t_2} = 24.688\%$$

secondary fill

Compressibility factor

$P_{cr} := 573psi$ $\qquad T_{cr} := 132.41K$ $\qquad T := 75°F$

Reduced Pressure $\qquad P_R := \dfrac{P_{max}}{P_{cr}}$ $\qquad P_R = 0.262$

Reduced Temperature $\qquad T_R := \dfrac{T}{T_{cr}}$ $\qquad T_R = 2.243$

thus $\qquad Z := 0.99$ $\qquad P_{cr} = 39.507\,bar$

Thus ideal gas law assumption is applicable with a very small error