CAES: Combined Compressed Air Energy Storage

Final Design

4/12/2010

Team #7
Bryan Campbell
Timothy Davis
Ray Melendez

Table of Contents

Executive Summary

The Compressed Air Energy Storage (CAES) project is based on the concept of storing renewable energy in an environmentally safe and friendly manner. The project is sponsored by the Center for Advanced Power Systems (CAPS) and Keuka Wind. The overall concept is to store wind energy when the demand for electrical power is low. The goal of the project is to determine the feasibility for small scale applications on the range of 20 kW to 200 kW.

For this project, wind turbines provide the mechanical power input to run an air 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 electrical power. We have developed a MATLAB program that contains experimental wind data, residential load curve, compressor and air motor specifications, and governing equations. This code can easily be manipulated to find the optimal system for any scale, simply by modifying the input parameters. The model used for analysis in this project was for a 20 kW rated turbine to mechanically drive the corresponding compressor.

Once the theoretical model was programmed, it was quickly discovered that the given pressure vessel is extremely undersized. The main obstacle for the design is the sheer volume of compressed air that any sufficient and sustainable power generation requires. The other major obstacle encountered is the finite time period that surplus power is generated which can be utilized to compress air.

It was determined that our analysis program assumes an efficiency of 100% for the system, which in reality is impossible. Therefore it should be noted that the model generated is an attempt to maximize power generation during peak load and minimize power input from the turbine. The only way to achieve this is to either increase the volume of air stored and the maximum fill pressure, or decrease the required load.

The final conclusions are that while the integrated CAES system is feasible on a small scale, it is highly inefficient. Even on small scale, the system requires a very large volume of air storage and power input compared to a small power output. However, in a completely off grid application, the CAES system designed to the specifications of our recommendation would be capable of generating a sustainable power supply. While the initial startup costs for a unique system are high, in large production these costs could be minimized and the pay off period for the integrated system could be optimized.

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 Compressed Air Energy Storage (CAES) systems. We will design and analyze a system driven by wind turbines and a power generation unit to convert wind energy to electric power. Analyses will be done on the system performance, efficiency and energy balance. This will be done while keeping the cost at a minimum and the scalability open. The scalability will be kept open so that if the customer desires to change the scale of the system they will be able to easily analyze the changes.

The CAES system will be comprised of three subsystems: a compressor, a storage device, 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 providing a sustainable renewable energy source for off grid applications.

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 hours, 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.

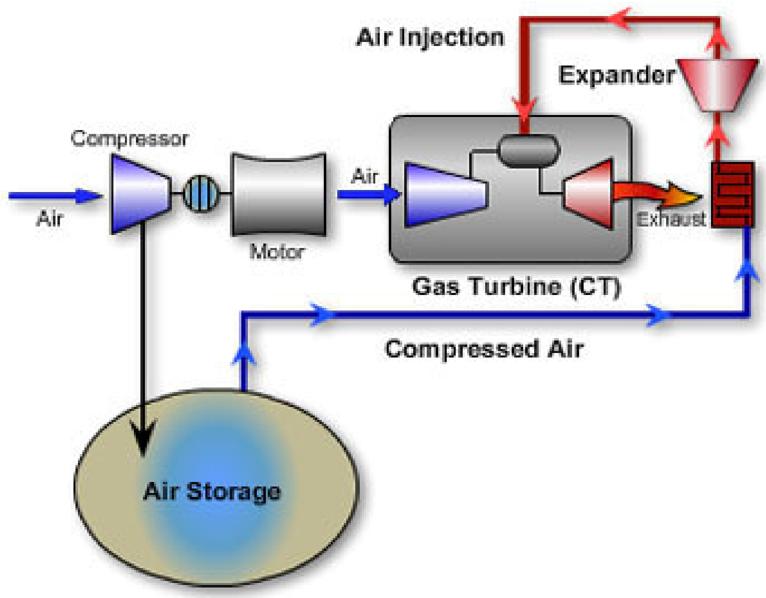


Figure 1 -Diagram of German CAES Power Plant

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, Keuka Wind. As mentioned before we will have four different wind turbines with power ratings of 20 kW, 50 kW, 100 kW and 250 kW. Because of the range of this project, we will focus on the 20kW wind turbine to make general analysis easier and consistent. All analysis will be done in MATLAB or Microsoft Excel and have made the analysis easy for adjustment. This is so that in the future, if the sponsor decides to use our analysis program for a different power rating, they can easily change the power inputs and outputs for different size systems. Furthermore, we have designed that the air compressor will be mechanically driven by the wind turbine, for reasons that will be discussed later. The pressure vessel is provided by our sponsors and we will go into detail about it later. The air motor will have the ability to be throttled according to the output power required.

Wind Turbines

Wind turbines essentially operate in one of two ways; either they utilize lift forces or drag forces to create rotational mechanical energy. Lift based turbines mimic the profile of an airplane wing to create rotational motion whereas drag based turbines 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. Larger lift based wind turbines require approximately 12 mph of wind to begin generating electricity and the generator is located at the center or hub. Rim based turbines only require approximately 2 mph of wind to begin generating electricity and the generator is at the base of the turbine. This design takes advantage of the fact that the velocity at the tip of the blades, or the rim of the turbine, is significantly higher than at the center.

Figure 2-Kueka Wind Rim based wind turbine

The blades of the wind turbines supplied by Keuka wind are connected by a metal rim. A rope or belt is tied around the rim of the turbine; this rope is fed through a system of pulleys that turn one or two electric generators to produce the equivalent electric power rating of the wind turbine. This is shown in Figure 3 below.

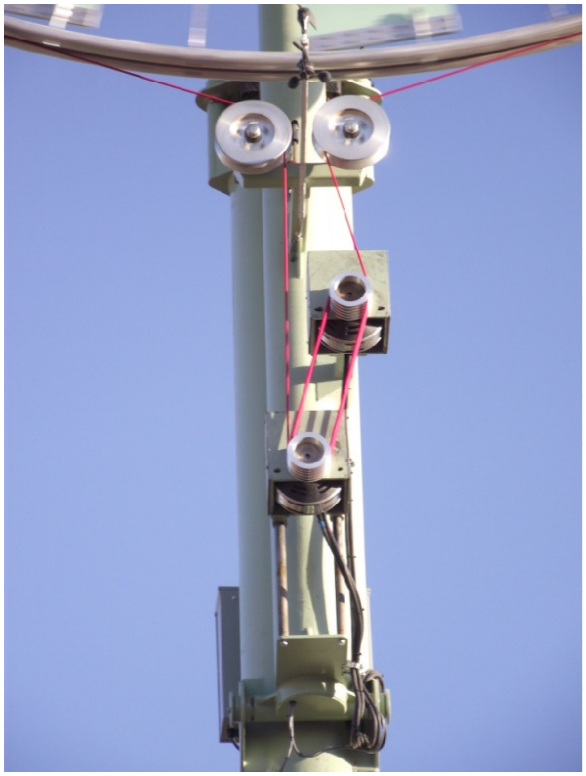


Figure 3-Rope/Pulley system on Keuka Wind Turbine

The wind turbine used for analysis in this project, shown in figure 2, has a diameter of 25 feet and a maximum power generation of 20 kW. The power curve obtained from experimental testing is shown in Figure 4.

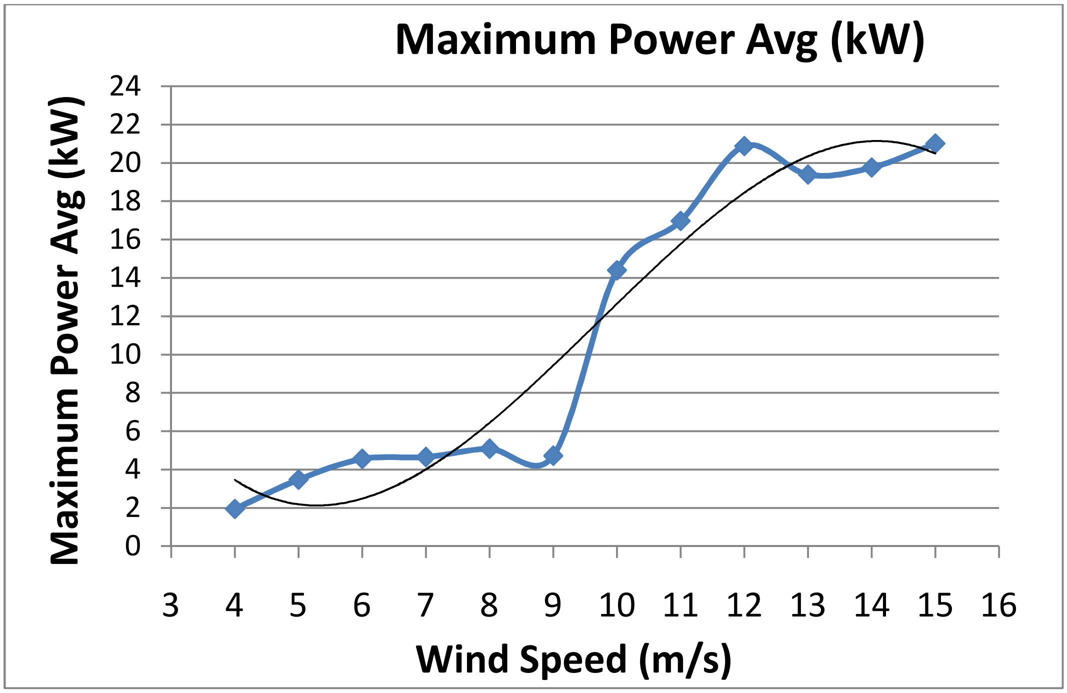


Figure 4-Power Curve for 20 kW wind turbine

The graph shows the experimental data points obtained during testing, and the third order polynomial fit power curve used for analysis. As shown in the power curve, the wind turbine is able to begin generating power at approximately 4 m/s. The power generation increases significantly from 7 to 10 m/s and reaches maximum power generation at 14 to 15 m/s.

Wind Data

The compressed air energy storage (CAES) system is dependent on the power output provided by the Keuka Wind turbine. The power generation of the wind turbine is dependent on the wind speed that is available; therefore wind data for the area must be analyzed in order to determine the available power output as well as sustainability of the system. Our sponsors have a 20kW wind turbine located just outside of Texas Tech University where they are doing testing and collecting wind data. This wind data was given to us for analysis for this project. The data was normalized to take out any fluctuations and plotted versus the time of day, from 0 to 24 hours. This is shown in Figure 5 below.

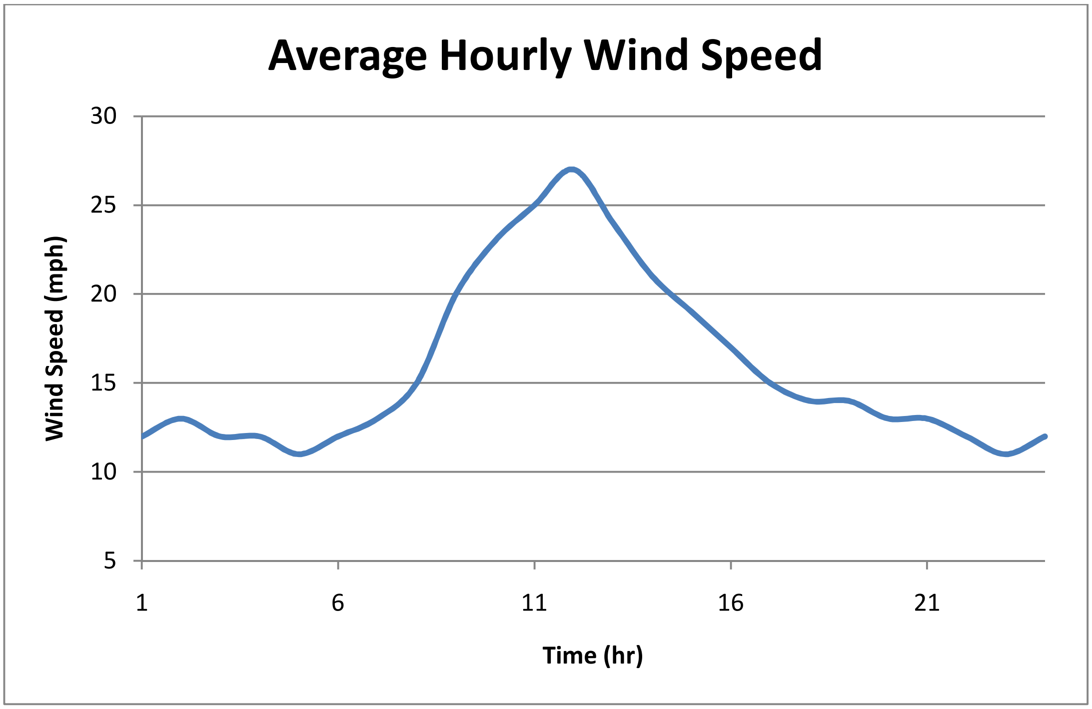


Figure 5- Wind Data from 20kW turbine outside TTU

In the above graph (Figure 5) it is clear that the peak wind speed occurs in the middle of the day from about 10 A.M until around 3 P.M. The peak generation will also occur during this period. The CAES system must be designed to maximize storage of energy during these peak generation hours in order to supply an adequate power supply during times of reduced generation.

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 this system being designed for a low speed wind turbine the compressor must be capable of operating at a low power input. Additionally the compressor must be capable of synchronizing with the fluctuating power input provided by the wind turbine. For the purpose of this design project we will design the compressor to be capable of operating at an input power of 20kW. At this operating power a low flow rate into the compressor can be expected; therefore as can be noted in Figure 6 below, a rotary type compressor will be best suited for our system.

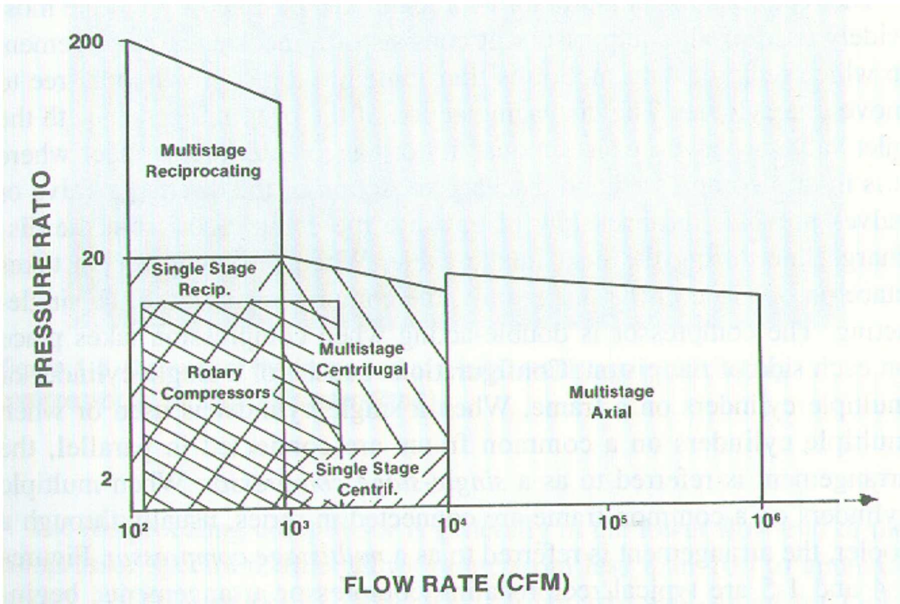


Figure 6-Typical application ranges of compressor types

For this application a helical-lobe type of compressor air end is best suited because of the compression type's high efficiency. This is due to the compression devices utilization of two 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 7. The screw compressors primary function that makes it the suitable choice however is its ability to perform at variable angular velocities without significant damage to the rotors. This ability is only available as a mechanically driven air end however. Therefore, the typical packaged units that compressor manufacturer's sell with a dedicated motor to power the air end at a single RPM speed will not work. A manufacturer and model of screw compressor air end must then be selected that has the best characteristics for our application.

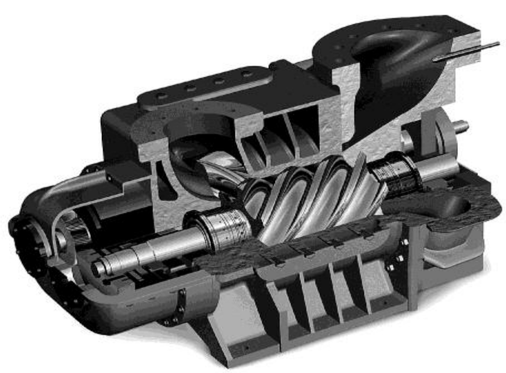


Figure 7- Helical-Lobe Compressor

Compressor Selection

As was discussed previously a rotary twin screw compressor air end will provide the characteristics that are necessary for this system to operate efficiently and will define the success of our design. Therefore 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.

The Quincy QGV 40 compressor was selected for the system that utilizes the 20 kW wind turbine. The QGV 40 has 127.5 mm rotors and a maximum operating speed of 3600 RPM. The compressors variation in flow rate in CFM due to change in RPM is shown below in Table 1 and has been plotted in Figure 8. This data was consolidated from the QGV 40 technical specifications sheet shown in Appendix 1.

Table 1-Compressor air end statistics for each input load

QGV 40			
	100psi	**125 psi**	**150 psi**
RPM	**CFM**	**CFM**	**CFM**
1000	42.8	40.3	40.7
1500	71	67.1	69.3
2000	98.5	97.5	97.3
2500	126.1	123.7	121.6
3050	156.6	150.9	152.9
3120	160.2	158.2	
3600	185.3		

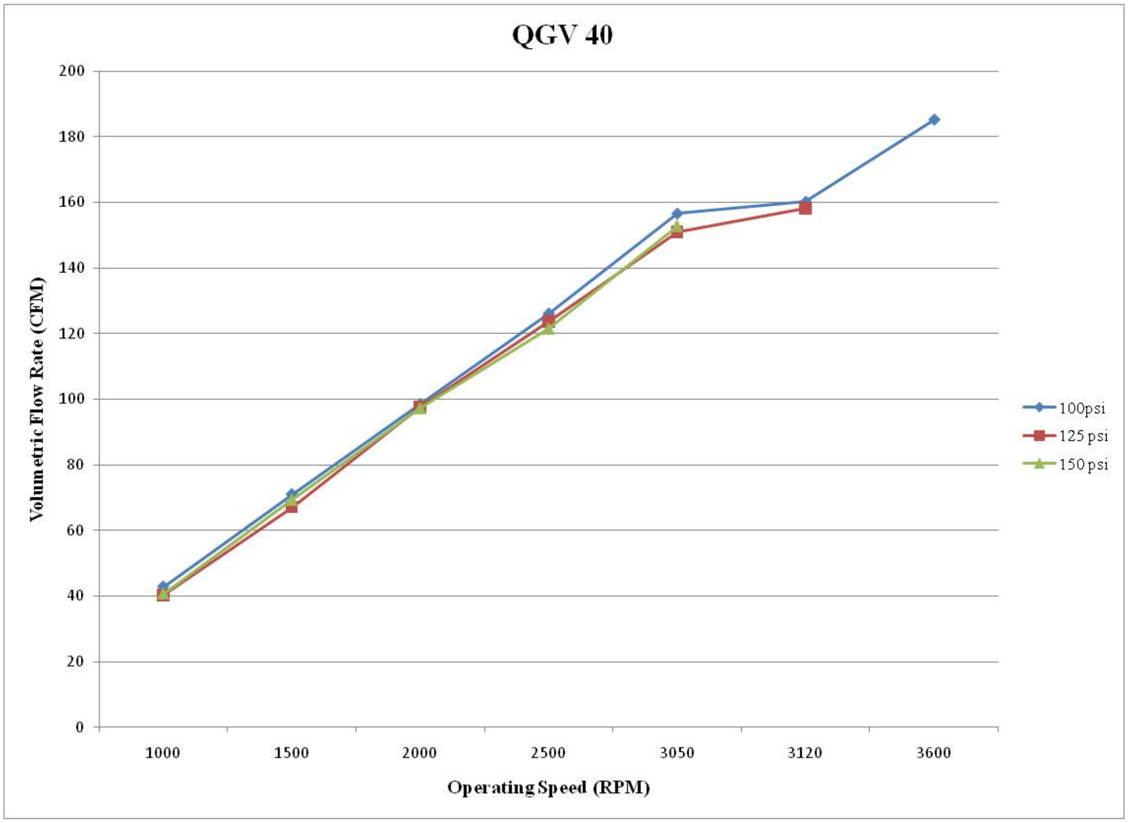


Figure 8 –QGV 40 Volumetric Flow Rate vs. Operating Speed

From the compressor statistics shown above it should be noted that the volumetric flow rate increase with each individual increase in input speed, this should cause our efficiency to increase as the wind speed increases. 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.

It has been determined that the size of the pressure vessel is not large enough to store the available power. From our analysis, we have shown that the vessel will need to be 2 or 3 times as large if the wind data we were given is considered an average wind day for the area.

Generation

For the smaller scale 20 kW system a gas turbine or micro turbine would not be feasible to operate due to high pressure and volume requirements. For this power range, as well as the other ranges given to use by our sponsors, an air motor will be integrated into the system to return the compressed air to mechanical energy that can then power an electric generator.

Air motors come in several different varieties such as vane, piston and turbine. Due to our customer's needs we have determined that piston air motors will be the best choice for this design.

Piston Air Motors

Piston air motors operate just like their internal combustion counterparts except they replace the fuel-air mixture with 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 use of speed controls costs can rise due to the complexities involved with controlling flow, power, maintenance and so forth. Also, these motors require excellent lubrication and higher maintenance due to the size and number of parts involved. Piston air motors are capable of generating up to 23 kW if the supply pressure and volumetric capacity is sufficient. Shown in Figure 9 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.

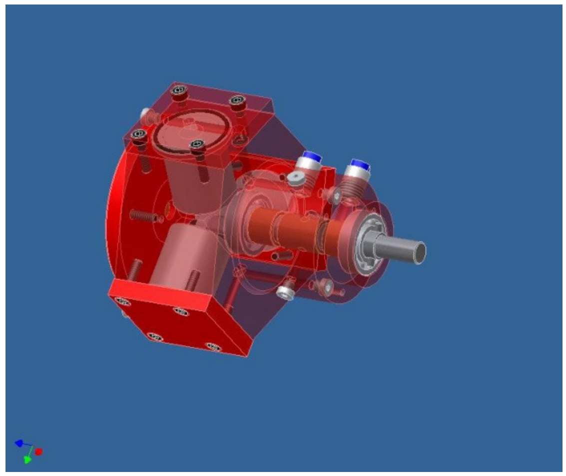


Figure 9-Huco Dynatork Radial Piston Air Motor

Performance of all types of air motors is highly dependent on the inlet pressure and flow rate available. 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 air motors supplied power curve. This optimum pressure and volume flow rate is only limited by the supplied pressure vessel.

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.

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 radial 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,

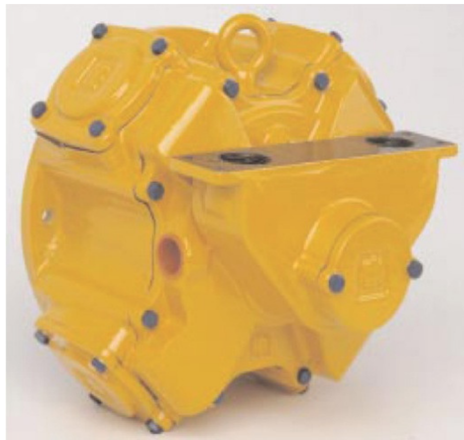


Figure 10- Ingersoll Rand MMP150

Due to the limitations of our pressure vessel and the analysis of the power generation and load curves we were led to the Ingersoll Rand MMP150 Air Motor. This air motor gives an output of 16 hp which is approximately 12 kW.

The Ingersoll Rand MMP150 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. Additionally they 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. These pressure limits therefore create a need for a flow control system which will be comprised of pressure regulators and solenoid valves. The regulators will maintain 90 psi to the motor and once pressure falls to 70 psi the solenoid valve will activate and close the piping to the air motor.

Run time analysis was done the same way as the fill time for the pressure vessel was. As will be shown later, it was found that the power this air motor produces exceeds the needed power requirement. Therefore, we have decided to either throttle this air motor down to produce only the necessary power of approximately 7 kW. If the required power output is half of what the motor will produce, then the required flow rate is half as well. This allows for almost twice the run time of the motor, which means that the system will produce a more constant power for a longer period of time. This is shown in the Results section as well as Appendix III.

Theoretical Analysis

The governing equations for the CAES system were derived using the Continuity Equation (Equation 1).

$$0 = \left.\frac{\partial m}{\partial t}\right|_{CV} + \iint_{CS} \rho V_n dA \tag{1}$$

The continuity equation is a form of the conservation of mass property, stating that the time rate of change of mass through the control volume, in this case the pressure vessel, plus the mass flow across the control surface is, by definition, zero. This equation can be solved for the pressure differential by assuming the ideal gas law, equation 2, which was verified with a compressibility factor of 0.99. In equation 2, m represents the mass, p the pressure, V the volume, R the gas constant for air, and T the temperature.

$$m = \frac{p\cancel{V}}{RT} \Rightarrow \left.\frac{\partial m}{\partial t}\right|_{CV} = \frac{\cancel{V}}{RT}\frac{dp}{dt} \tag{2}$$

In the case of surplus power from the wind turbine, the compressor will be operating to fill the pressure vessel with compressed air. Thus the air motor will not be operating and the mass flux leaving the control surface will be zero. Therefore equation 1 reduces to equation 3.

$$\frac{\cancel{V}}{RT}\frac{dp}{dt} = \iint_{in} \frac{p}{RT} V_n dA \tag{3}$$

By assuming that the gas constant R and the temperature T are relatively constant throughout the process, equation 3 can be reduced and solved for the pressure differential with respect to time, shown in equation 4, where p is pressure, p_{in} is the output pressure from the compressor to the pressure vessel, t is time, V is the volume of the pressure vessel and $\dot{V}$ is the volumetric flow rate of the compressor.

$$\frac{dp}{dt} = \frac{1}{V} p_{in}(t)\dot{V}(t) \tag{4}$$

In the second case in which the required load is greater than the power output from the wind turbine, compressed air stored in the pressure vessel is used to power an air motor. In this case, there will be no incoming mass flux across the control surface, as the compressor will be shut off. However, as the air, motor will be in operation, there will be a mass flux exiting the control surface. In this case, equation 1 and 4 become equation 5.

$$\frac{dp}{dt} = \frac{1}{V}\left[-p(t)_{out}\dot{V}(t)_{out}\right] \tag{5}$$

Note that equation 5 is very similar to equation 4; however, the pressure time derivative is negative as the vessel is being depleted, p_{out} refers to the operating pressure of the air motor, and $\dot{V}_{out}$ is the operating flow rate of the air motor.

For component selection and initial analysis, the input power was assumed to be a constant 20 kW and equation 4 was solved for the fill and run times of the compressor and air motor respectively. In this case, the compressor outlet pressure and flow rate are held constant and the separable equation can be integrated with respect to time to solve for the pressure in the tank. Likewise, equation 5 was solved for the maximum power generation of the air motor, yielding a constant operating pressure and volumetric flow rate.

For actual analysis of the integrated CAES system, equations 4 and 5 were solved numerically using MATLABs ordinary differential equation solver and the provided data for wind speed and residential load requirements.

Results

In approximating residential load requirements for the integrated CAES system, data was obtained from the NREL Habitat for Humanity Zero energy house. This data, scaled up for the CAES system, should serve as an appropriate residential load, as the goal of the CAES system is the provide power for an off grid residential community. The available data from NREL is shown in Figure 11 below.

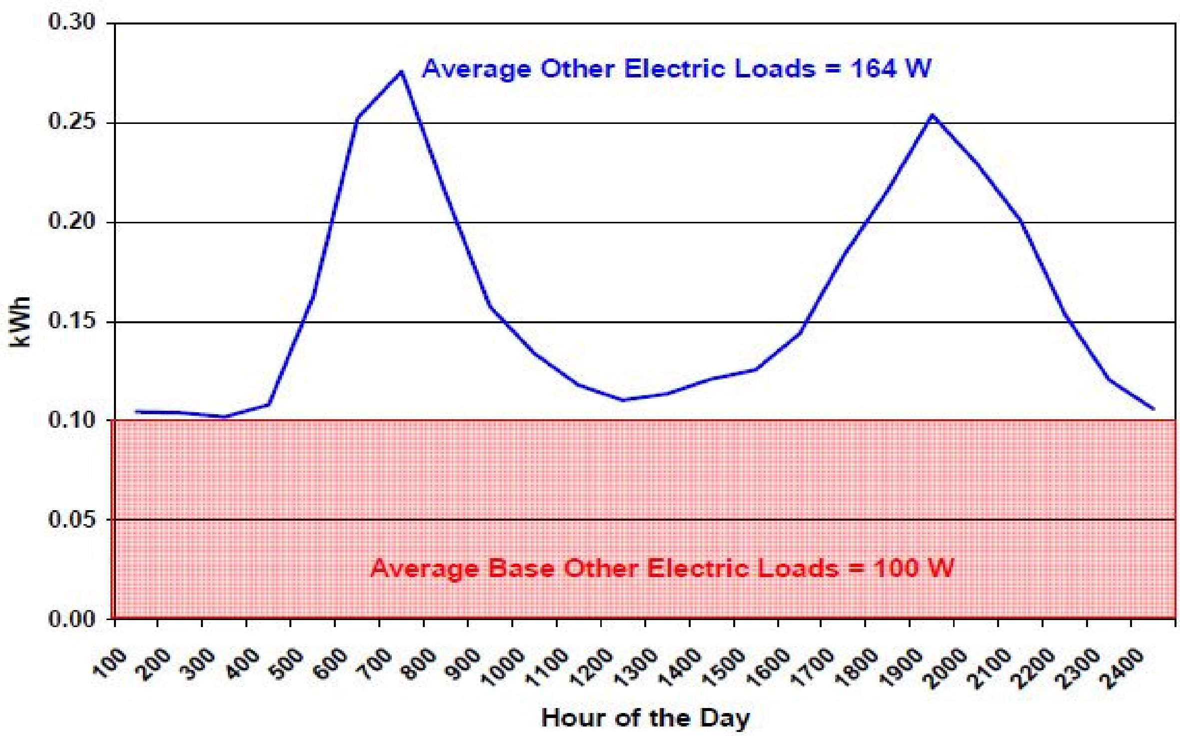


Figure 11-NREL Habitat for Humanity Zero Energy House

This graph shows the power consumption for a single zero energy house. As shown on the graph, the average base load requirement is 100W and the varying load averages an additional 164W. The varying load peaks both in the morning hours and evening hours, at approximately 8:00 AM and 8:00 PM respectively.

Using the power curve experimentally obtained for the 20 kW wind turbine, shown in Figure 4, we transformed the experimental wind data, Figure 5, to plot the daily power generation of the wind turbine. The NREL residential load was then scaled to an appropriate power consumption to match the power for the wind turbine, shown in Figure 12 below.

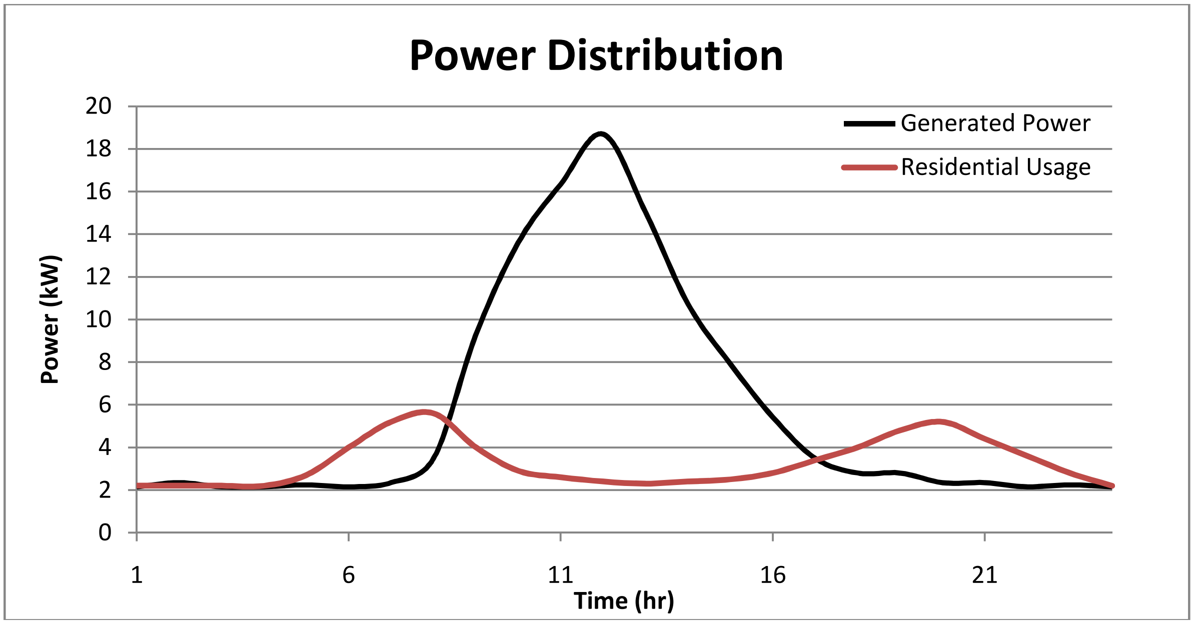


Figure 12-Experimental Power Distribution Curve

For theoretical analysis, a polynomial fit was generated for the turbine power production, and a sinusoidal curve was fit for the residential power usage. These curves are shown in Figure 13 below.

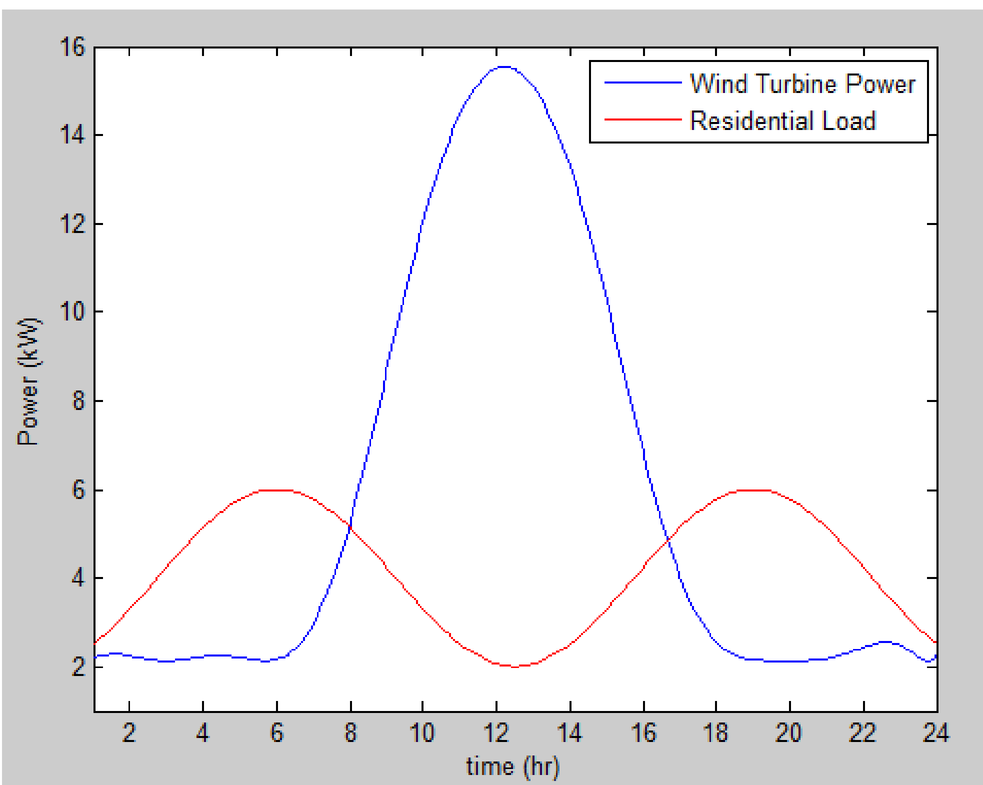


Figure 13-Theoretical Power Distribution

The theoretical power distribution show in above in Figure 13 closely approximates the experimental data shown in Figure 12. However, the sinusoidal residential load curve slightly over estimates the power consumption in the early morning hours.

In analyzing the power distribution curve, it is apparent that from approximately 10:00 AM to 4:00 PM, the power generated by the turbine exceeds the residential load. During these hours, the surplus power can be diverted to the compressor to store air in the vessel. Using the compressor power curve as a function of flow rate, as shown in Figure 8, the volumetric flow rate was calculated using the surplus power. With the given compressor, the pressure variation is shown in Figure 14 below.

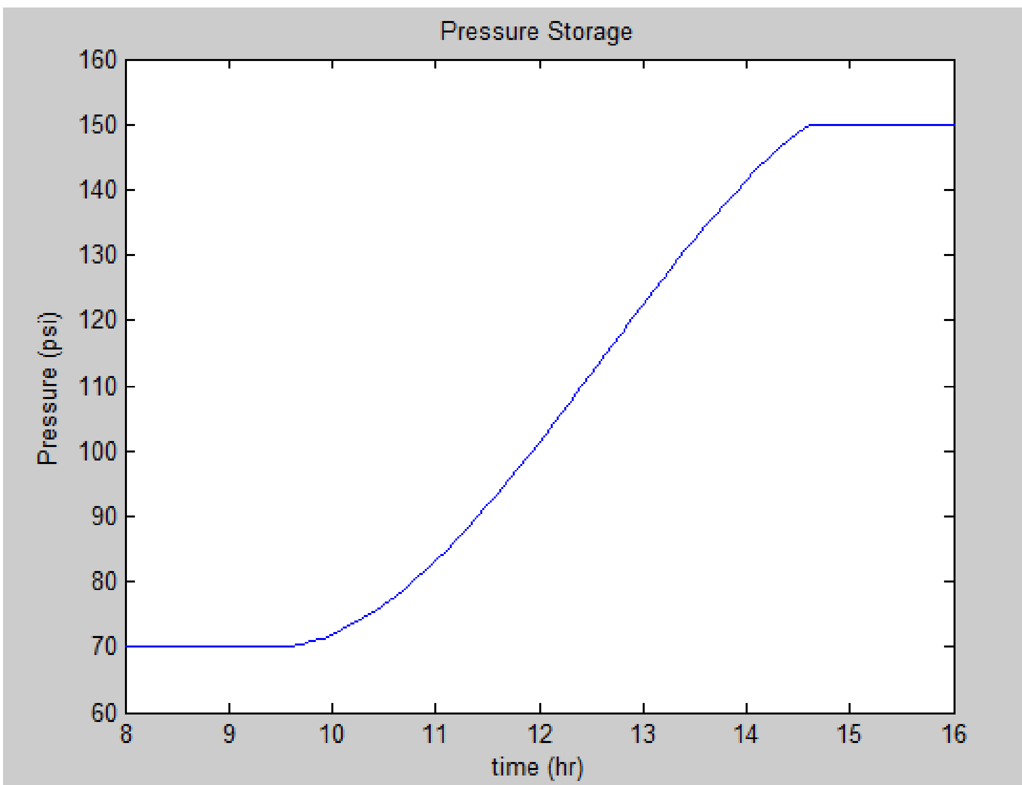


Figure 14-Pressure Storage using surplus power

This graph assumes that the pressure vessel has an initial pressure of 70 psi, the minimum operating pressure of the air motor and the compressor is shut off when the pressure reaches 150 psi in the pressure vessel. In addition, the pressure vessel used for this graph is 2.5 times the size of the provided vessel, to achieve maximum use of the surplus power and run time of the air motor.

By combining the fill profile with the discharge profile during power generation for peak load requirements, a profile for the variation in pressure in the vessel can be plotted. This is shown in Figure 15 below.

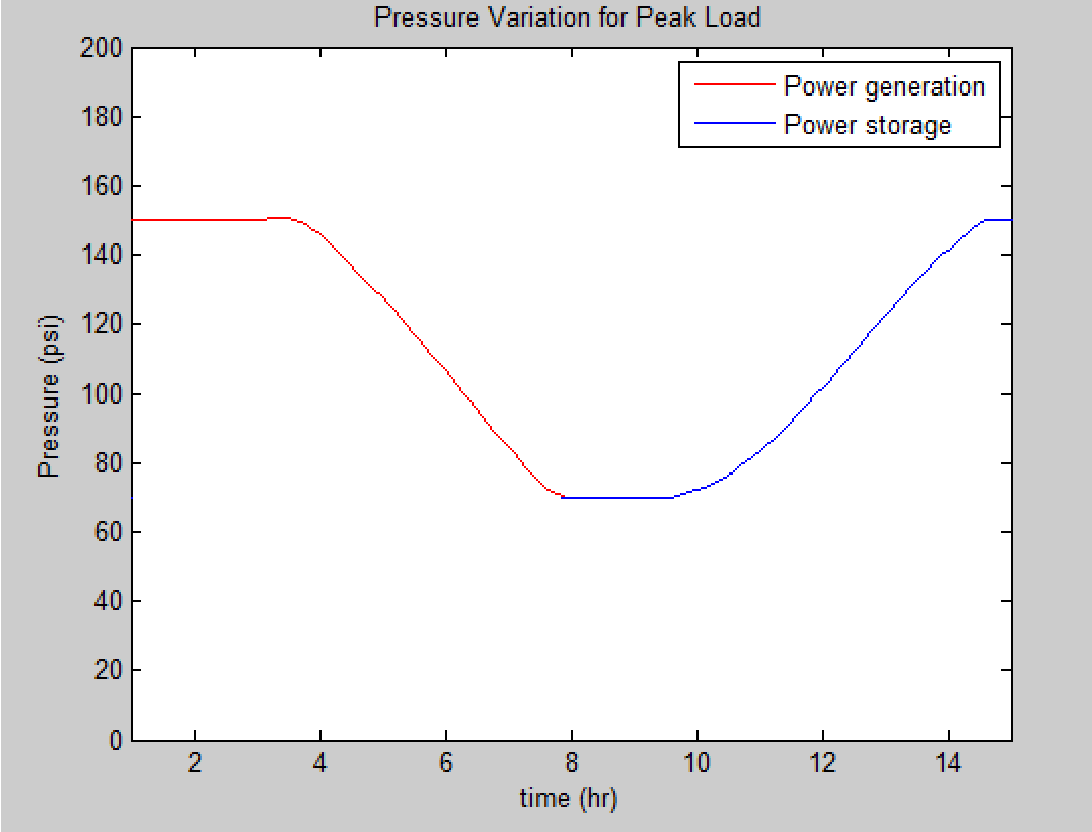


Figure 15-Pressure variation for peak load and surplus storage

Figure 15 shows that for the morning hours, when the residential load is highest and the wind turbine power generation is lowest, the air motor is used for power generation. Thus, during this time, the pressure is depleted from the tank. Then, as the wind speed increases and surplus power is generated, the compressor begins to replenish the compressed air storage to 150 psi.

It is important to note that while our initial selection was for a 12 kW air motor, it was found during analysis that this was not needed as the required power during peak load hours does not exceed 5 kW for this model. Also, the required volumetric flow rate was too large for the required period of operation using the assumed pressure vessel volume. Therefore, a 7 kW air motor was used in this model, with a required flow rate of nearly half that of the 12 kW air motor; which increased our run time enough to satisfy the load requirement.

Cost Analysis

The provided budget for the CAES project was $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 was quoted at $5400. In addition, the pressure regulating devices and solenoid valves totals approximately $900. These prices are summarized below in Table 22. Given that the total for the components far exceeds our budget, and our experience with operating and maintaining the equipment, we are making recommendations to our sponsors at the Center for Advanced Power Systems at Florida State University 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
		Total	$15,949
		Budget	$2,500

Reliability

With any renewable energy system being designed, the reliability, or equivalently the sustainability, of the system must be analyzed. Due to the system being fully reliant on wind energy the biggest sustainability risk is that off the location of the system. The location must be carefully chosen to ensure that there will be sufficient wind speed and frequency of wind in order to truly sustain the system. If there is no wind, no power will be generated, and if there is not any power being generated then there will surely not be enough power to compress air for later use.

Another reliability issue that arose from our analysis is that our current analysis assumes that the system will be able to convert 100% of the wind energy into compressed air energy. Essentially, we have assumed an efficiency of 100% and therefore have neglected inefficiencies of the system as a whole in our analysis. This was due to the fact that we were not able to test or operate any of the equipment due to the scale and location of the system. The wind turbine for which the data was provided is in Texas, but the pressure vessel and air compressor is in Florida. And assuming preset efficiencies for equipment would skew the data and results even more.

Our analysis has also shown that there is a sufficient need for controls in this system. The controls for the system need to be able to tell the system when to turn on and off the air compressor. As was discussed earlier, the pressure vessel will need control or isolation valves which will need to be controlled according to whether the vessel is being filled, emptied, or resting. Controls will also lend a great deal of flexibility to the air motor by being able to throttle the power of the air motor according to the power requirement of the system.

These issues are inherent in any project that is relying on wind energy as an energy source. Other issues can easily be resolved with enough testing or data. Since we were not able to resolve some of these issues we must focus on the analysis program created. Our program is easy to use for anyone who is somewhat familiar with MATLAB and we have made it such that the parameters of the system can be easily changed. This assures that our sponsors will be able to use our analysis program for any size system they have in mind and they can use the results of the program for a good starting base for further analysis.

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. As partial fulfillment in the project description, we have developed a MATLAB program that contains the wind data, residential load curve, compressor and air motor specifications, and governing equations. This code can easily be manipulated to find the optimal system for any scale, simply by modifying the input parameters.

The model used for analysis in this project was for a 20 kW rated turbine to mechanically drive the corresponding compressor. Once the theoretical model was programmed, it was quickly discovered that the given pressure vessel with a volume of approximately 11,300 cubic feet was tremendously undersized. The main obstacle for the design is the sheer volume of compressed air that any sufficient power generation requires. Thus the model was modified for a pressure vessel with 2.5 times the volume of the given tank. The other major obstacle encountered is the finite time of surplus power generation that could be applied to compress air. As the pressure vessel size increases, the amount of time required to fill the vessel increases accordingly. With the original compressor selected and the actual wind data, the volume 2.5 times that of the current vessel is the maximum amount the system can charge with surplus power from the turbine.

Initially, our task was to set the residential load curve and power generation curve of the wind turbine to match. This implies that the integral of the load curve equals that of the turbine generation curve. However, this assumes that the system has a100% efficiency and is clearly not the case with compressed air, or any power system. Therefore the model was generated in an attempt to maximize power generation during peak load and minimize power input from the turbine. This is the model shown in the Results section and uses a 20 kW rated wind turbine for power generation and a community of 20 NREL zero energy homes.

It has been determined that the current model is also impossible to achieve for both morning and night peak loads, as it stores only enough compressed air for the morning peak.

The central issue for residential loads is that the spikes in required power occur twice daily, while the spike in surplus power occurs only once at midday. Therefore, the task becomes to store enough surplus power during the midday to supply power to peak loads in both the morning and evening. The only way to achieve this is to either increase the volume of air stored and the maximum fill pressure, or decrease the required load. The first option is only possible up to a point, with the limitations being the maximum pressure of the vessel, the maximum output pressure of the compressor, and the time duration of surplus power.

In an effort to match the load and power curves to maximize efficiency, it has been shown that the efficiency of the system is far too low to provide enough power on the scale selected. Thus, using our current component selection, the residential load must be decreased for our model to operate. It is impossible to store enough compressed air during the five hours of surplus power from the wind turbine to generate the residential load requirement for both morning and evening. For a more realistic model, the residential load must be decreased. This would effectively increase the input power and, while efficiency will decrease, the system will be operable. The new results for this model are given in the appendix, which decreases the load to 13 NREL zero energy homes. This system allows for the surplus power during midday to store enough compressed air to generate power for both morning and evening peak loads.

The final conclusions is that while the integrated CAES system in feasible on a small scale, it is highly inefficient. Even on small scale, the system requires a very large volume of air storage and power input compared to a small power output. However, in a completely off grid application, the CAES system could be capable of supplying residential load requirement for a properly designed system. While the initial startup costs for a unique system are high, in large production these costs could be minimized and the pay off period for the integrated system could be optimized.

Bibliography

Allen, Carolyn. Air compressors for Manufacturing Energy Efficiency. 2006. <http://www.californiagreensolutions.com/cgi-bin/gt/tpl.h,content=1297>.

Dart Creations. Air Motors. 2005. <http://www.dart-creations.com/business-tree/auto/air_motors.html>.

Davey Compressor. Different Types of Compressors. 2006. <http://www.daveycompressor.com/differenttype.html>.

GE. Positive Displacment Compressors. 2008. <http://www.gepower.com/prod_serv/products/compressors/en/downloads/ac_pos_displ_compr.pdf>.

Hydraulics and Pneumatics. Air Motors. 2010. <http://www.hydraulicspneumatics.com/200/TechZone/FluidPowerAcces/Article/True/6422/TechZone-FluidPowerAcces>.

Brown, Royce. Compressors Selection and Sizing. Houston, Texas: Gulf Publishing Company, 1997.

Florida State University Meteorology . Average Wind Speed Florida. 2010. <http://coaps.fsu.edu/climate_center/data/normals_avgwindspeed.shtml>.

Appendix I- Air Compressor Specifications

TECHNICAL DATA QGV-40 QUINCY COMPRESSOR DIV.

GENERAL OPERATING DATA

FULL LOAD TARGET PRESSURE				
MAXIMUM	75	PSI (g)	5.2	BAR (g)
MINIMUM	150	PSI (g)	10.5	BAR (g)
MAXIMUM OPERATING PRESSURE	165	PSI (g)	11.5	BAR (g)
MINIMUM OPERATING PRESSURE	75	PSI (g)	5.2	BAR (g)
MAXIMUM AMBIENT TEMPERATURE				
WITHOUT ENCLOSURE	N/A	°F	N/A	°C
WITH ENCLOSURE	104	°F	40	°C
MINIMUM AMBIENT TEMPERATURE	34	°F	1	°C
SOUND LEVEL @ 1 METER, AIR COOLED, UNENCLOSED			N/A	dBa
SOUND LEVEL @ 1 METER, WATER COOLED, UNENCLOSED			N/A	dBa
SOUND LEVEL @ 1 METER, AIR COOLED, ENCLOSED			69	dBa
SOUND LEVEL @ 1 METER, WATER COOLED, ENCLOSED			N/A	dBa

AIREND DATA

FIRST STAGE		
ROTOR DIAMETER	127.5	MM
DRIVE TYPE	DIRECT	
DRIVEN ROTOR SPEED, MAXIMUM note 6	3600	RPM

FREE AIR DELIVERY (CFM / M^3/HR.)

At Specified Operating Pressure

Compressor RPM	100 PSI (g)	6.8 BAR (g)	125 PSI (g)	8.5 BAR (g)	150 PSI (g)	10.3 BAR (g)
1000	42.8	71.4	40.3	67.3	40.7	67.9
1500	71.0	118.5	67.1	112.0	69.3	115.7
2000	98.5	164.4	97.5	162.7	97.3	162.4
2500	126.1	210.5	123.7	206.5	121.6	203.0
3050	156.6	261.3	150.9	251.9	152.9	255.2
3120	160.2	267.3	158.2	264.0		
3600	185.3	309.2				

PACKAGE SPECIFIC POWER (kW/100 CFM)

At Specified Operating Pressure

	Air Cooled			Water Cooled		
Compressor RPM	100 PSIG	125 PSIG	150 PSIG	100 PSIG	125 PSIG	150 PSIG
1000	27.8	33.7	41.8	N/A	N/A	N/A
1500	22.7	27.3	31.7	N/A	N/A	N/A
2000	21.1	24.0	27.7	N/A	N/A	N/A
2500	20.1	23.3	27.6	N/A	N/A	N/A
3050	19.9	23.0	26.4	N/A	N/A	N/A
3120	19.9	22.6		N/A	N/A	N/A
3600	19.7			N/A	N/A	N/A

FULL PACKAGE SPECIFIC POWER, PER CAGI-PNEUROP PN2CPTC2 TEST CODE

AIR COOLED COOLER PERFORMANCE

HEAT REJECTION FLUID COOLER	1412	BTU/MIN	24.8	kW
HEAT REJECTION AFTER COOLER	314	BTU/MIN	5.5	kW
APPROACH TEMPERATURE A/C note 2 & 3	5	°F	2.8	°C
MAXIMUM ALLOWABLE STATIC BACKPRESSURE	0.25	in H_20	0.0006135	ATM
AIR COOLED COOLERS FAN	1.5	HP	1.1	kW
NOMINAL MOTOR SPEED	1200		RPM	
COOLERS COOLING FAN CAPACITY@1.0" WG SP	3218	CFM	5467	m^3/hr

WATER COOLED COOLER PERFORMANCE

HEAT REJECTION FLUID COOLER	N/A	BTU/MIN	N/A	kW
HEAT REJECTION INTERCOOLER	N/A	BTU/MIN	N/A	kW

ENGQGV1

03.23.10 REV. 001

HEAT REJECTION AFTER COOLER	N/A	BTU/MIN	N/A	kW
TEMPERATURE APPROACH TO COOLING WATER @ 70°F note 2&3	N/A	°F	N/A	°C
WATER FLOW W/AFTERCOOLER @ 70°F	N/A	GPM	N/A	L / s
@ 90°F	N/A	GPM	N/A	L / s
WATER FLOW L/AFTERCOOLER @ 70°F	N/A	GPM	N/A	L / s
@ 90°F	N/A	GPM	N/A	L / s
WATER PRESSURE MIN., MAX.	N/A	PSIG	N/A	BAR (g)
WATER PRESSURE DROP W/AC @ note 8 note 7	N/A	PSIG	N/A	BAR (g)
WATER PRESSURE DROP L/AC @ note 8 note 7	N/A	PSIG	N/A	BAR (g)
WATER CONNECTIONS SIZE IN/OUT	N/A		NPT	
PACKAGE RADIATE HEAT note 9	N/A	BTU/MIN	N/A	kW
VENT FAN	N/A	HP	N/A	kW
NOMINAL MOTOR SPEED	N/A		RPM	
VOLUMETRIC FLOW RATE	N/A	CFM	N/A	m^3/hr

COMPRESSOR FLUID

SYSTEM CAPACITY	7.5	GAL.	28.4	LITER
FLUID RESERVOIR CAPACITY	6.0	GAL.	22.7	LITER
COMPRESSOR FLUID FLOW RATE, MAX	21.0	GPM	1.3	L / s
TYPICAL FLUID CARRYOVER	2 - 4		PPM	
NORMAL DISCHARGE TEMPERATURE	190	°F	88	°C

PACKAGE ENERGY DATA (kW)

At Specified Operating Pressure

	Air Cooled			Water Cooled		
Compressor RPM	**100 PSIG**	**125 PSIG**	**150 PSIG**	**100 PSIG**	**125 PSIG**	**150 PSIG**
1000	11.9	13.6	17.0	N/A	N/A	N/A
1500	16.1	18.3	22.0	N/A	N/A	N/A
2000	20.8	23.4	27.0	N/A	N/A	N/A
2500	25.4	28.8	33.6	N/A	N/A	N/A
3050	31.2	34.7	40.3	N/A	N/A	N/A
3120	31.9	35.7		N/A	N/A	N/A
3600	36.6			N/A	N/A	N/A

WEIGHTS & DIMENSIONS

LENGTH	97.1	INCHES	2466	MM
WIDTH	39.5	INCHES	1003	MM
HEIGHT	64.7	INCHES	1643	MM
WEIGHT - WITH ACOUSTICAL ENCLOSURE	2335	LBS	1059	KG.
WEIGHT - WITHOUT ACOUSTICAL ENCLOSURE	N/A	LBS	N/A	KG.
SERVICE CONNECTION	1.25		NPT	

1. When facing control panel 'X' from left end , 'Y' from panel side
2. Approach temperature is dependent on ambient temperature,relative humidity, & barometric pressure & may be different than the values given .
3. After cooler approach temp based on 100°F ambient heat load based on amb. conditions 68°F,36%R.H.29.92"Hg
4. Maximum ambient based on 220°F discharge temperature
5. Consult factory when using ethylene or propylene glycol as a coolant
6. Driven rotor speed based on nominal motor speed
7. Does not include water temperature regulating valve
8. At flow specified @70°F
9. Based on 90°F ambient temperature

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

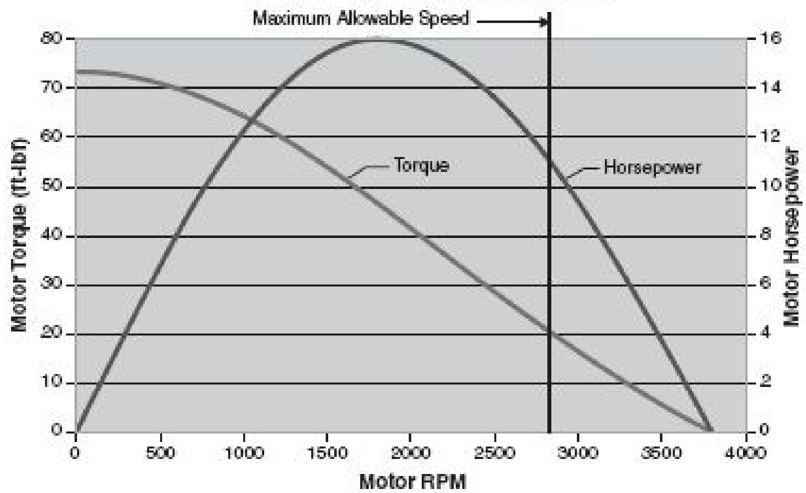
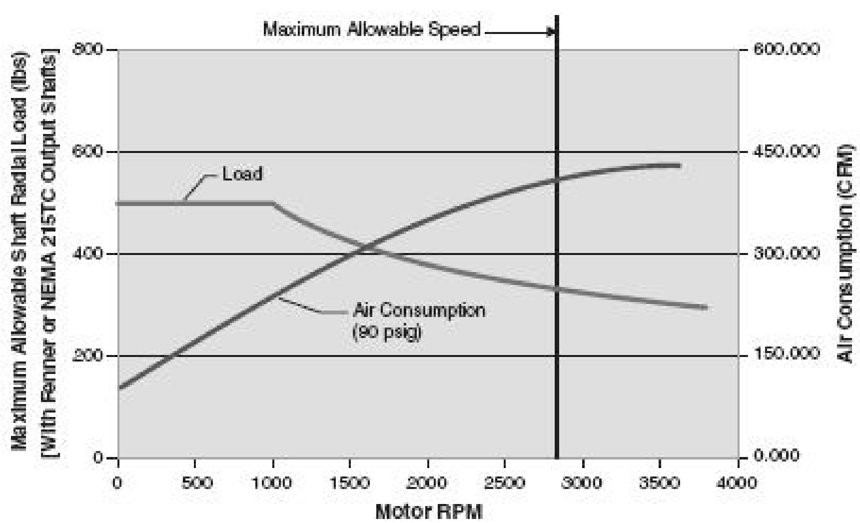


Figure 2: MMP150 Air Motor Performance



Appendix III- Calculations (constant power input)

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

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

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 $$P_{yield} = 312.5\,psi$$

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

Allowable Stress

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Run Time - Air Motor

$P_{operate} := 90\,psi$

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

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

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

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

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

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

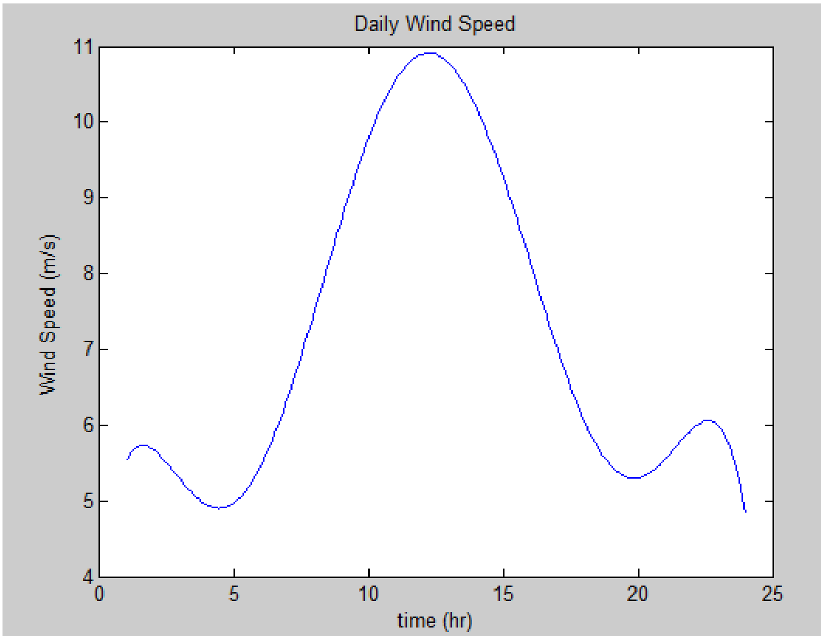
Reduced Temperature $T_R := \frac{T}{T_{cr}}$ $T_R = 2.243$

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

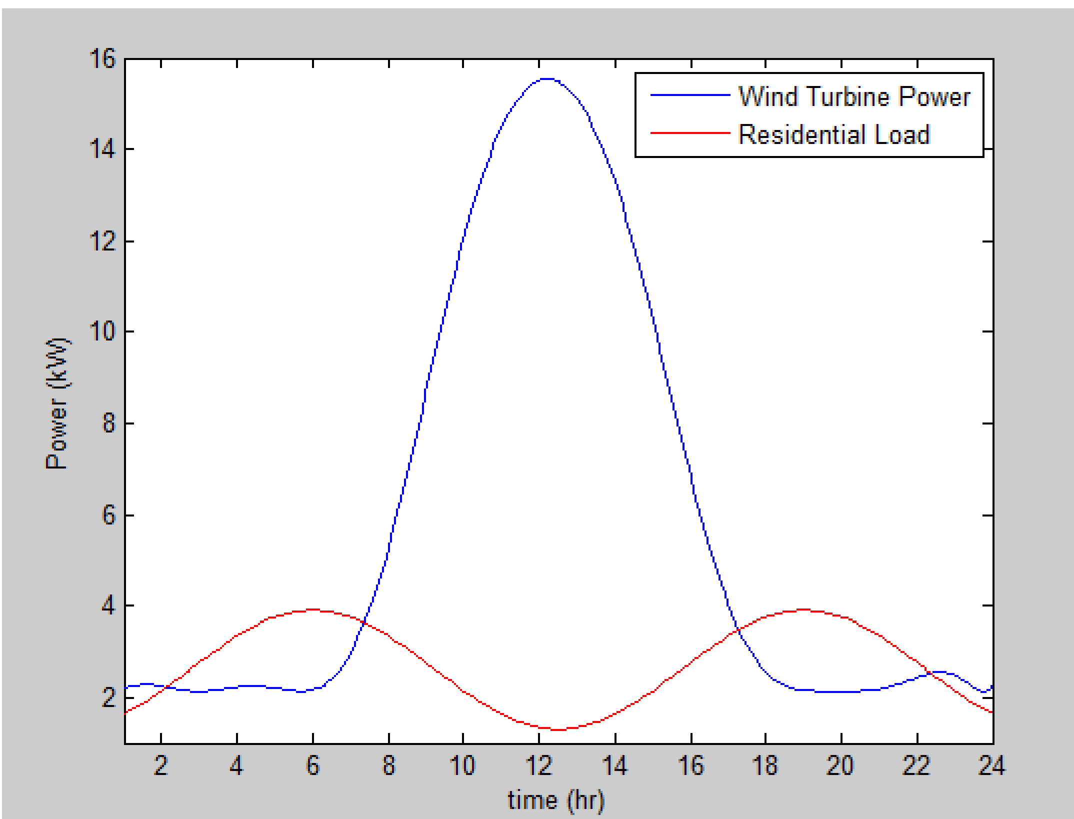
Thus ideal gas law assumption is applicable with a very small error

Appendix IV – Variable power input

Theoretical Wind Profile



Actual Power Distribution Curve – Community of 13 NREL homes and 20 kW turbine



Appendix V – MATLAB code

```
%System_Simulation
psi = 60;
tspan = 1:0.1:24;
P_0 = 70/psi;

[t,P] = ode45('fill_ode', tspan, P_0);
P = P*psi;

for i = 1:length(t)
    if P(i) > 150
    P(i) = 150;
    else if P(i) < 70
            P(i) = 70;
        else P(i) = P(i);
        end
    end
end


figure(4)
plot(t,P),axis([8 16 60 160]), title('Pressure Storage'), xlabel('time
(hr)'), ylabel('Pressure (psi)')

Wind = (-0.00001206)*t.^6 + 0.0008783*t.^5 - 0.023577*t.^4 + 0.2818397*t.^3 -
1.439047*t.^2 + 2.793162*t + 3.9378176;
Req_Power = 13*(0.1*sin((2*pi/13)*t-1.33) + 0.2);
Gen_Power = -0.0557*Wind.^3 + 1.6192*Wind.^2 - 12.443*Wind + 30.89;

for i = 1:length(t)
    if Gen_Power(i) > Req_Power(i) + 10/3
        Comp_Power(i) = Gen_Power(i) - (Req_Power(i) + 10/3);
    else Comp_Power(i) = 0;
    end
end


RPM_Comp = 150*Comp_Power;
CFM = 0.0543*RPM_Comp - 12.697;

figure(1), plot(t,Wind),xlabel('time (hr)'), ylabel('Wind Speed (m/s)'),
title('Daily Wind Speed')
figure(2), plot(t,Gen_Power,t,Req_Power,'-r'),xlabel('time (hr)'),
ylabel('Power (kW)'), legend('Wind Turbine Power','Residential Load'),
axis([1 24 1 16])
%figure(3), plot(t,CFM),
figure(5), plot(t,Comp_Power)

tspan = 1:0.1:24;
P_init = 150/psi;
[t,pr] = ode45('discharge_ode',tspan,P_init);
pr = pr*60;
```

```
for j = 1:length(t)
    if Req_Power(j) < Gen_Power(j)
        pr(j) = 150;
    else pr(j) = pr(j);

    end

end


figure(6), plot(t,pr,'-r',t,P), title('Pressure Variation for Peak Load'),
xlabel('time (hr)'), ylabel('Pressure (psi)'),legend('Power
generation','Power storage')


%-----------------------------------------------------------------------------

function dpdt = discharge_ode(t,pr)

Wind = (-0.00001206)*t.^6 + 0.0008783*t.^5 - 0.023577*t.^4 + 0.2818397*t.^3 -
1.439047*t.^2 + 2.793162*t + 3.9378176;

Req_Power = 13*(0.1*sin((2*pi/13)*t-1.33) + 0.2);

%Power Curve = -0.0557*wspeed^3 + 1.6192*wspeed^2 - 12.443*wspeed + 30.89
%Provided by Sponsors

Gen_Power = -0.0557*Wind.^3 + 1.6192*Wind.^2 - 12.443*Wind + 30.89;


if Req_Power > (Gen_Power + 1.5)
    Load_Power = Req_Power - Gen_Power;
else Load_Power = 0;
end

if Load_Power == 0
    RPM_air = 0;
    CFM_air = 0;
else RPM_air = 21.711*Load_Power.^2 + 89.182*Load_Power + 28.268;
    CFM_air = (0.1684*RPM_air + 133.35)*1/1.77083;
end


L = 100; D = 12; Vol = 3*pi/4*D^2*L;
P_in = 80;

dpdt = -1/Vol*(P_in*CFM_air);

%-----------------------------------------------------------------------------

function dpdt = fill_ode(t,p)
```

```
Wind = (-0.00001206)*t.^6 + 0.0008783*t.^5 - 0.023577*t.^4 + 0.2818397*t.^3 -
1.439047*t.^2 + 2.793162*t + 3.9378176;

Req_Power = 13*(0.1*sin((2*pi/13)*t-1.33) + 0.2);

%Power Curve = -0.0557*wspeed^3 + 1.6192*wspeed^2 - 12.443*wspeed + 30.89
%Provided by Sponsors

Gen_Power = -0.0557*Wind.^3 + 1.6192*Wind.^2 - 12.443*Wind + 30.89;

if Gen_Power > (Req_Power + 10/3)
    Comp_Power = Gen_Power - Req_Power - 10/3;
else Comp_Power = 0;
end

RPM_Comp = 150*Comp_Power;
CFM = 0.0543*RPM_Comp - 12.697;

if CFM > 10
    CFM = CFM;
else CFM = 0;
end


 D = 12; L = 100; %dimensions in feet
 Vol = 3*pi/4*D^2*L;
 P_out = 150;

dpdt = 1/Vol*(P_out*CFM);

%------------------------------------------------------------------------------------------------------------------------------------

function Pressure
%Solution of ODE for pressure variation in vessel

%Time span
day2min = 24*60;
psi = day2min;
day2hour = 24;
t0 = 0.05;
tf = 0.75;
tspan1 = [t0 tf]; %days -> 12 midnight = 0
P_0 = 0; %psi

for i = 1:10

options = odeset('Events', @eventP150);
[t,p,te,pe,ie] = ode45(@fill_ode, tspan1, P_0, options);

MaxPressure = 150*ones(size(t));

P = p*psi;
T = t*day2hour;
```

```
TE = te*day2hour;
PE = pe*psi;

Fill_Time = TE - T(1)

plot(T,P,T,MaxPressure), xlabel('Time (hours)'), ylabel('Pressure (psi)')
title('Pressure Variation in Vessel')
grid on
hold on

tspan2 = [te tf];
P_max = pe;

options = odeset('Events', @eventP90);
[t,p,te,pe,ie] = ode45(@discharge_ode, tspan2, P_max, options);

P = p*psi;
T = t*day2hour;
TE = te*day2hour;
PE = pe*psi;
MaxPressure = 150*ones(size(t));

Run_time = TE - T(1) %hours
max_power = 11.9; %kW - using 12 kW air motor
Power_Generated = max_power*Run_time %kWhr
Time_of_day = TE

tspan1 = [te tf];
P_0 = pe;

plot(T,P,T,MaxPressure), xlabel('Time (hours)'), ylabel('Pressure (psi)')
title('Pressure Variation in Vessel')
grid on
hold on

end

%----------------------------------------------------------------------------

%ODE for solving pressure variation within pressure vessel - fill only

function dpdt = fill_ode(t,p)

%Define wind speed (wind turbine rpm, loaded) equation
RPMturbine = -8874.3*t.^6+27287*t.^5-30860*t.^4+15845*t.^3-
3669.3*t.^2+347.25*t+1.5469;

%Define gearing ratio to determine compressor RPM
N = 100;
RPMcompressor = N*RPMturbine;

%Define relationship for compressor rpm and flow rate (cfm)
%Vdot = 0.0543*rpm - 12.697 (for P = 150, QGV40 compressor)
Vdot = 0.0543*RPMcompressor - 12.697;
```

```
%(pressure vessel calculations)
L = 100;
D = 12;
Vol = pi/4*L*D^2;

%ODE derived from Continuity Equation
%ODE1 - No Power generation ->: dP/dt = 1/Vol*(P_out*Vdot)
P_out = 150; %psi

dpdt = 1/Vol*(P_out*Vdot);

%---------------------------------------------------------------------------

% ODE for solving pressure variation in tank with power generation

function dpdt = discharge_ode(t,p)

%Define wind speed (wind turbine rpm, loaded) equation
RPMturbine = -8874.3*t.^6+27287*t.^5-30860*t.^4+15845*t.^3-
3669.3*t.^2+347.25*t+1.5469;

%Define gearing ratio to determine compressor RPM
N = 100;
RPMcompressor = N*RPMturbine;

%Define relationship for compressor rpm and flow rate (cfm)
%Vdot = 0.0543*rpm - 12.697 (for P = 150, QGV40 compressor)
Vdot = 0.0543*RPMcompressor - 12.697;

%(pressure vessel calculations - units in ft)
L = 100;
D = 12;
Vol = pi/4*L*D^2;


P_out = 150; %psi

%Air motor data for constant power generation (12 kW air motor)
P_airmotor = 90; %operating pressure (psi)
Vdot_airmotor = 425; %flow rate (cfm)

%ODE derived from Continuity Equation

dpdt = 1/Vol*((P_out*Vdot) - (P_airmotor*Vdot_airmotor));

%---------------------------------------------------------------------------

%Event Location for P = 150 psi

function [value, isterminal, direction] = eventP150(t,p)

psi = 24*60;
```

```
value = p(1)*psi - 150;
isterminal = 1;
direction = 1;

%------------------------------------------------------------------------------

%Event Location for P = 70 psi (Power Generation)

function [value, isterminal, direction] = eventP90(t,p)

psi = 24*60;

value = p(1)*psi - 90;
isterminal = 1;
direction = -1;
```