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Study to Reduce Peak Electrical Demand Charges for the John C. Hodges Library

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Office of Sustainability — University of Tennessee

Electrical Load Shifting to Reduce Peak Demand Charges for Comfort Conditioning of the John C. Hodges Library

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May 2016:

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ACRONYMS

Acronym	Definition
TVA	Tennessee Valley Authority
KUB	Knoxville Utility Board
UT (UT)	University of Tennessee
USACE	United States Army Corp of Engineers
CERL	Construction Engineering Research Laboratory
BEopt	Building Energy Optimization
DOE	Department of Energy
TES	Thermal Energy Storage
RFB	Redox Flow Battery
VRB	Vanadium Redox Battery
ZNFB	Zinc-Nickle Flow Battery
PV	Photovoltaic
CHP	Combined Heat and Power
MT	Micro-turbine
RE	Reciprocating Engine
DD	Desiccant Dehumidifier
OFT	Oberflachentechnik

EXECUTIVE SUMMARY

The University of Tennessee (UT) campus at Knoxville spent $14.5 million in 2012 for electrical service, of which $3.7M (25%) was cost attributed to demand charges for comfort conditioning UT's buildings from June through September. It is not uncommon to find demand charges of the same magnitude as energy charges. Eliminating the entire demand charge is difficult, but the demand can be reduced by simply shifting the electrical load from on-peak to off-peak periods. UT current peak demand charges are $14.57 per kW from 2PM till 8PM in the summer and $14.57 per kW from 5AM till 11AM in the winter. In addition, an on peak demand fee of $14.92 per kW will also be charged UT if any demand exceeds the predetermined contractual power limits set by the local power company, KUB.

Undergraduate students in the MABE department conducted a Capstone study to design electrical and thermal storage systems for the John C. Hodges Library that would shift the electrical load. The Hodges Library in 2012 had demand charges of about $750k. Therefore, the Office of Sustainability provided funding to the MABE department through the Green Fee initiative to support faculty and students with the development of 3 technology scenarios to help support the UT sustainability policy for renovation and new construction based on LEED for Commercial Interiors (LEED-CI) and on LEED for New Construction and Major Renovations (LEED-NC). The 3 technologies were investigated:

1. **Redox flow batteries (RDF)** have been under development since the 1980s. A breakthrough occurred with the development of a Teflon based fluoropolymer membrane that improved battery life. Currently, there are three RFB types showing excellent potential for the commercial market: Vanadium Redox Batteries (VRBs), Zinc-Nickel (ZNFB), and Hydrogen-Bromine (HBr). The ZNFB and HBr batteries are still in bench testing; however, the VRB is commercially available. The VRB has a demonstrated lifespan of 30 years with proven reliability exceeding 100,000 cycles. Pump and the recharging of electrolyte fluid are the only maintenance items. RDF has excellent application to retrofit and new construction.

2. **Thermal storage** has the potential to provide added cooling capacity at less investment as compared to additional chiller infrastructure. Instead of expanding one of the 5 UT chiller plants, a chilled water storage (TES) or phase change material (PCM) storage system can be implemented to meet UT's growing demand. The thermal storage would serve the cooling load of a building by day and be regenerated the following night with an ice-maker heat pump harvesting ice or with the existing central chillers cooling PCM using off peak power. TES showed the best potential because of water's high latent heat and the reduced volume for ice storage as compared to PCM. Either has good retrofit application and excellent opportunity for new construction.

3. **Combined Heat and Power (CHP) with supplemental dehumidification and comfort cooling** merges energy systems into a power generation subsystem to produce electrical energy and a thermally activated subsystem that captures the hot generator exhaust to produce useful thermal energy for industrial processes and/or comfort conditioning of buildings. The technology has two useful energy outputs for the fuel energy input required by the power generator, i.e., an additional useful energy stream, in the form of thermal energy, is produced from the exhaust energy and would be used to drive adsorption chillers for comfort cooling and desiccant dehumidifiers for supporting the latent load of the Hodges library. CHP has excellent application to new construction for making the building its own autonomous power plant.

Space limitations in retrofit applications would favor selection of ice storage system or flow battery technologies. The thermal energy storage system using ice-maker heat pumps yields a payback of 5 years with an investment of $258.5k to shift the peak electrical load. Under current costs the RDF flow battery has a 15 year payback but as product cost drops to near-term levels, the payback improves to 9.7 years. The CHP with reciprocating engine as the prime mover and desiccant/adsorption HVAC shows excellent potential for new construction application and realizes a simple payback within 7.9 years.

Therefore for retrofit applications the TES and RDF are better selections if space is a limiting factor. An ice storage system yields superior payback but requires the inclusion of outdoor footprint for an air-cooled condenser. The CHP scenario targets new construction, and could make the library's electrical needs almost totally independent of the electrical grid. Payback as compared to UT's existing chiller plant in the Hodges Library is 7.9 years! UT's energy conservation policy also requires that campus buildings must operate with as low as feasible environmental impact. All 3 scenarios have minimal impact on the environment. The CHP system reduces the annual electrical consumption, produces low NO_x and CO_x emissions and therefore supports the reduction in carbon footprint for the UT campus.

Table 1.0 Cost, annual savings and simple payback for the 3 technology scenarios. TES and RDF are sized for shifting 100% of the HVAC peak summer load for the John C. Hodges Library. CHP includes HVAC equipment and best payback occurs for continuous CHP operation.

ITEM	**TES**	**Flow Battery (RDF)**		**CHP with Comfort Conditioning****	
	Ice Storage	Current Cost	Near Term Cost	Micro-turbine	Recip. Engine
Total Cost	$258.5k	$934.6k	$559.3k	$18,498k	$12,498.0k
Annual Savings	$52.9k	$70k	$70k	$913.7k	$1,581.8k
Payback (years)	4.9	15	9.7	20.2	7.9
** CHP savings compared to the library's existing chilled water plant. Assumes 7 adsorption chillers, each of 90 kW capacity.					

1. INTRODUCTION

The Tennessee Valley Authority (TVA) and one of its many local power companies, the Knoxville Utility Board, provide all electrical service to the University of Tennessee (UT) Knoxville campus. UT generates steam at its boiler plant for comfort conditioning of its buildings in the winter. UT uses a chilled water system for comfort cooling and has 5 chiller plants at various locations on campus. Each plant is equipped with about 5,000 tons of refrigerant capacity for chilling brine that is pumped to throughout the campus for supplying comfort cooling. (Note: A refrigerant ton (RT) is equivalent to 12,000 Btu per hour of cooling. Thermodynamically, 288,000 Btu are required to make one ton of ice; divide 288,000 Btu by 24 hours to get 12,000 Btu per hour to make one ton of ice in one day)

The University has an ambitious long-term plan for new building construction and for the renovation of its existing buildings, all of which will require expansion of the chilled water and steam distribution systems. Therefore UT's plans must include the addition of supplemental capacity. The Capstone therefore investigated 3 options: (1) Redox flow batteries (RFBs); (2) thermal storage using phase change materials with the existing chiller plant or ice-maker heat pumps that produce ice at night to serve the cooling load of a building by day; and, (3) Combined Heat and Power (CHP) with inclusion of desiccant dehumidification and adsorption cooling. Option 3 is better suited to new building construction.

The Capstone report addresses battery technology, thermal storage and CHP as viable options for UT Facility Services and the Office of Sustainability to consider in future power plant retrofits and/or new building designs.

1.1 THERMAL LOAD COMPUTATIONS

The thermal load for the Hodges Library was computed using the heat balance procedure in the ASHRAE Fundamentals (2013). The 99% heating design dry-bulb temperature of -6.2°C (20.8°F) was assumed for peak heating load. For peak cooling load the 1% dry-bulb/coincident wet-bulb condition was used; it being 32.5°C DB /22.8°C WB [90.5°F/73.1°F]. The peak heating and cooling loads were used to estimate the magnitude of the electrical load that could be shifted to off-peak hours. A load profile is a graph that shows electrical load versus time. For buildings, a load profile typically rises during the morning, peaks during the day, falls in the evening, and remains consistently low throughout the night. However, as weather conditions change on a day-to-day basis, the specifics of the load profile change. In order to properly approximate the load profile of the Hodges Library, an average load profile was gleaned from the Knoxville Utilities Board (KUB) for three time periods (Summer, Winter, and Transitional). The Hodges Library is currently equipped with two electric meters, but there is no measure for the power draw of the chillers or other HVAC equipment in the library. Because of this, the load profile of the library had to be approximated, which was done using the ASHRAE Fundamentals [2013] as well as computer modeling programs.

UT Facility Services provided the building's blueprints, which documents the square footage of the walls, fenestration, floors, and ceilings as well as the construction materials of each envelope system. The ASHRAE Fundamentals [2013] provided properties for the materials used in the building's construction (i.e. conductivity, specific heat, density and thermal resistance). Parallel resistance paths were used to compute the effective thermal resistance for the walls, floor and ceilings where framing factors made an applicable difference in heat flow. TYM3 weather data [NREL, 2016] for Knoxville, TN was used to find the most reasonable temperatures in both summer and winter that would cause the largest temperature gradients between the temperature the library is held at and the outside temperature during on-peak hours. Finally, ASHRAE Fundamentals [2013] and the library's design documentation were used again to find the expected power cost from infiltration, ventilation, equipment (such as lights and computers), and an assumption of how crowded each floor would be during peak hours (100 occupants per floor). Once all this was calculated/looked up, the power needed to maintain the library's temperature assuming the outside temperature was creating as much of a temperature gradient as Knoxville's climate can reasonably produce was calculated, and the results are shown below in Table 1. The results show that the summer load is nearly twice as much as the winter load, which is probably due to the internal miscellaneous loads, which in the summer are a detriment to cooling but in winter are a supplement to heating the building.

Table 1.1. Hodges Library Building Peak Loads from ASHRAE Fundamentals [2013]

Floor	Summer Load (BTU/hr)	Winter Load (BTU/hr)	Latent Load (BTU/hr)
Ground	278,899.70	-107,882.81	79,969.97
1	286,454.66	-104,233.91	79,969.97
2	338,944.19	-144,605.73	79,969.97
3	185,710.84	-82,650.42	54,022.50
4	179,594.69	-81,456.37	54,022.50
5	174,330.69	-77,074.93	54,022.50
6	153,518.43	-229,773.00	26,862.58
Total	1,597,453.21	-827,677.18	428,840.00
Summer	2,026,293 Btu/h [593.8 kW]		
Winter	827,677 Btu/h [242.6 kW]		

1.2 MODELS USED FOR THERMAL LOAD

To better approximate and benchmark the load profile of Hodges Library, various software tools were used to model the library. The first software used was BEopt (Building Energy Optimization), a free software developed by the National Renewable Energy Laboratory. BEopt allows the user to model his building and enter in the materials used in the building's construction. The code uses the Department of Energy (DOE) software Energy Plus as its engine to compute and analyze the energy consumption of the building given the construction, geometry, internal loading, make-up air and weather conditions. However, BEopt is more suited to residential applications and its model showed BEopt was unable to accurately simulate the makeup air requirements for a building the size of Hodges. It did, however, help benchmark our spreadsheet tool and showed that the summer load was approximately twice as much as the winter load. The next two programs looked at were DesignBuilder and OpenStudio, both of which could take building files and run them with imported weather data to generate load profiles. Eventually, OpenStudio was selected due to its cost (OS is free, developed by the DOE) and its ability to better model commercial buildings. A key feature of OpenStudio is its ability to accept a Revit file, software used in 4th year Architecture and Design curriculum. Revit is a very robust tool usable for modeling and rendering buildings and is built for Building Information Modeling (BIM) to assist professionals in the design of more energy-efficient buildings. Two 4th year architecture students (Paul Bamson and Austin Winter) were hired to model the library in Revit, which was uploaded into OpenStudio, Figure 1.1. Unfortunately, due to time constraints, a load profile from OpenStudio is still in process. However, an approach was demonstrated for developing a cross-disciplinary curriculum between Engineering and Architecture for a future BIM model of the entire campus. As field data becomes available for the power consumed by the

Hodges HVAC system, a benchmark will be completed using OpenStudio and the results used to better predict energy savings and payback for implementation of advanced technologies discussed herein that reduce peak electrical demand changes.

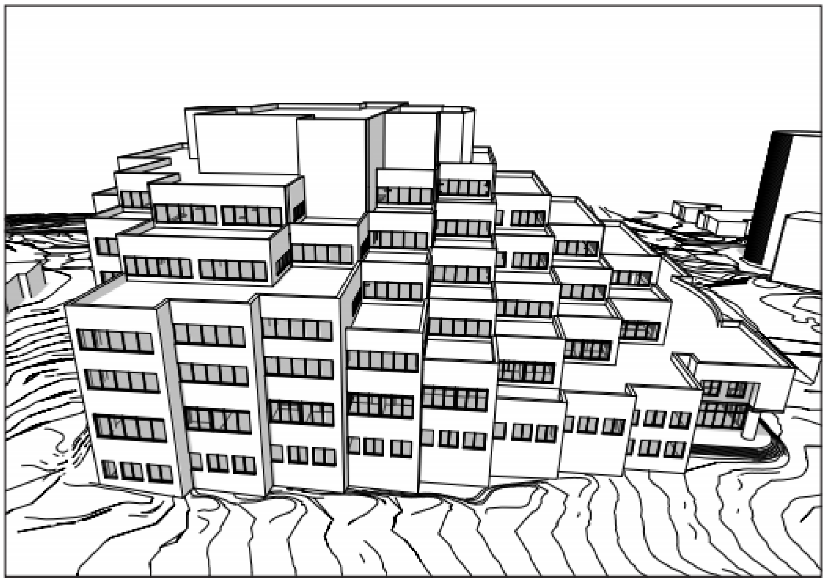

Figure 1.1 The Revit Model for the John C. Hodges Library was developed by architect students Paul Bamson and Austin Winter.

1.3 POWER INSTRUMENT INSTALLATION IN LIBRARY

A portion of the UT Green Fee stipend was used to purchase instrumentation for measuring and recording the energy usage of all HVAC equipment in the library. WattNode Modbus power meters were purchased from Continental Control Systems LLC. A micro-logger was also purchased from Campbell Scientific to record the power data for the two centrifugal chillers, assorted pumps and fans associated with cooling towers on the roof of the Library. All equipment has been delivered and is awaiting install by UT Plant at a scheduled power outage. Once installed, data will be recorded continuously and output at 15 min intervals to coincide with demand charge monitoring done by the KUB.

2. THERMAL ENERGY STORAGE FOR PEAK LOAD SHIFTING

Thermal energy storage (TES) is a technology that stocks energy thermally in a material for use at a later time. There are two main types of TES; sensible TES and latent TES. Sensible TES takes advantage of the energy required to heat or cool a material (specific heat value) without the material undergoing any change of phase. Latent TES utilizes the energy required to change the phase of a material (latent heat value) from solid to a liquid, or liquid to a gaseous state. Latent TES is generally favorable due to the amount of energy required to change the phase of a material. For example, the energy required to heat 1 gram of water from 0°C (32°F) to 80°C (208°F) is the same energy needed to melt 1 gram of ice. Because of this fact, the latent TES system can be sized smaller which is favorable for retrofitting a TES system to the existing Hodges Library. The following presented latent TES systems are retrofit system designs using equipment from the manufacturing and HVAC industry.

2.1 ICE STORAGE SYSTEM

Water is the most common thermal storage material due to its high specific heat (4.18 J/(g·ºC)), high latent heat of fusion (333.55 J/g), availability, and stability. For the application of peak load shifting in the Hodges library, water stored in a tank can be frozen (charged) at night by various methods during off-peak hours and melted (discharged) during on-peak hours to provide cooling energy to the library HVAC with minimal need of electricity. This could lower or completely eliminate the load received by the plant chillers during on-peak hours; thus drastically reducing the demand charges accrued by UT. The following latent TES system utilizes water as the storage medium, Fig. 2.1.

At the 2016 AHR EXPO we were able to meet manufacturers as well as discuss and view state-of-the art HVAC products. An ice storage system design was chosen based on the reliability and compatibility of component parts with the current plant setup. The system design consists of 8 CALMAC™ 1190C IceBank™ storage tanks and a Trane™ RTAC185 air-cooled chiller. The storage tanks have a total cooling capacity of 1296 RT-h, which would allow for discharge of cooling energy during the entire on-peak period. During the discharge cycle the existing plant brine (chemically treated water) returning from the air handlers would be chilled by a heat exchanger that transfer heat to a glycol solution, which flows in a closed loop between the storage tanks and the air-cooled chiller. The IceBank™ storage tanks enclose counter-flow heat exchanger tubes that would be surrounded by ice. As the glycol solution passed within the tubes and exchanged heat with the ice, it would be cooled so that more heat could be absorbed from the plant brine. During the charging cycle the glycol solution supplied to the storage tanks by the air-cooled chiller must be below the freezing temperature of the stored water to create ice as it circulates. Using the existing Trane™ centrifugal chillers for ice making would de-rate the chillers; causing issues with the refrigerant pressure leading to cavitation of the centrifuge impellers. Thus additional ice making equipment is required and the Trane™ RTAC185 air-cooled chiller could easily be placed on the roof of Hodges Library for the purpose of creating ice during off-peak hours. Fig. 2.1 below displays a CALMAC™ 1190C IceBank™ tank and the Trane™ RTAC185.

a)

b)

Figure 2.1 a) CALMAC™ 1190C IceBank™ Storage Tank b) Trane™ RTAC185 air-cooled chiller.

Each CALMAC™ tank has a diameter of 7.5 ft. and a height of 8.5 ft. Due to the storage tanks modular design, the internalized main headers can be bolted to each other which in turn will reduce the total footprint, piping, and insulation required. The total footprint required for 8 CALMAC™ IceBank™ storage tanks is 15 ft by 30 ft standing at a height of 8.5 ft which fits the designated area in the Hodges Library. The Trane™ RTAC185 has dimensions of 15.5 ft by 8 ft by 7 ft which would easily fit on the rooftop of Hodges Library. Fig. 2.2 below displays the schematic for a retrofit application of this system to the Hodges Library plant. The heat exchanger between the plant brine and glycol solution is not shown in Fig. 2.2.

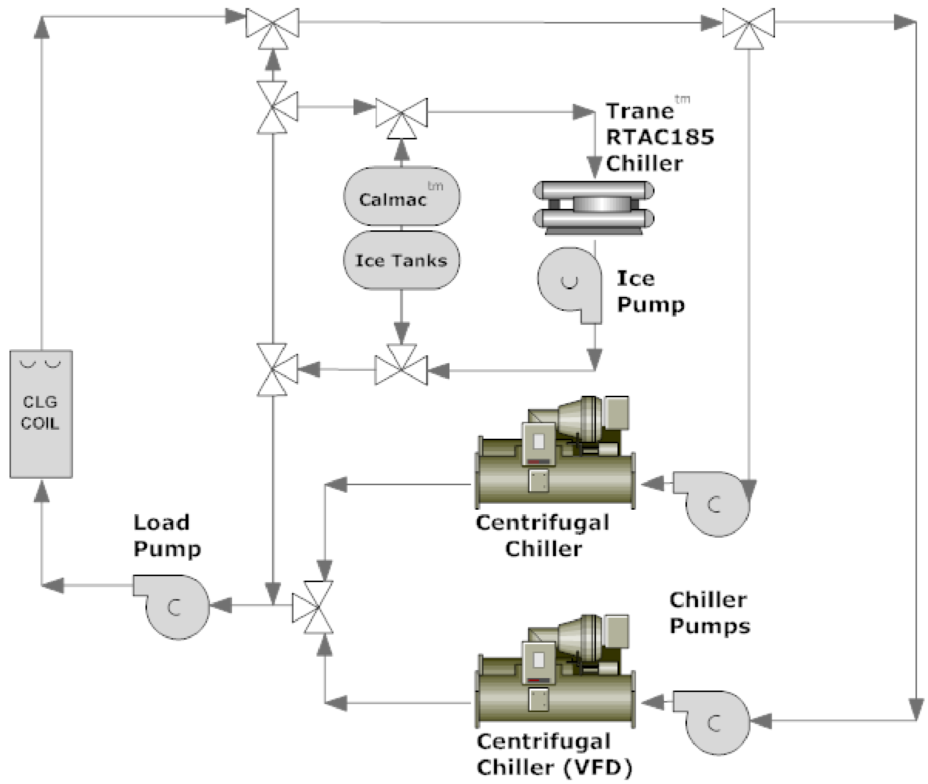

Figure 2.2 Schematic of retrofit latent TES system using ice as the storage medium.

The storage tanks ideally would be placed in the existing plant bypass loop. During the discharge cycle this would allow the centrifugal chillers to be bypassed, which would reduce the electricity consumption during the on-peak period. The chillers could also be operational during the on-peak period if the load was too large to be supported by the ice storage system. Additionally, this retrofit setup also allows for the ice storage tanks to be bypassed during the charging cycle and maintenance periods. Table 2.1 shows the total budget equipment cost of the presented ice storage system including installation.

Table 2.1 Total Cost of Retrofit Ice TES System

System Equipment	Equipment Cost	Installation (15%)	Total
RTAC185*	$166,500	N/A	$258,500
8 IceBank Tanks	$80,000	$12,000	
*Installation Costs Included			

The payback period presented in Table 2.2 was computed based on the percentage of the on-peak load that could be reduced by the TES system. For instance, a 30% reduction in on-peak demand would alleviate 30% of the energy consumed by the centrifugal chiller compressor during the on-peak period. Using the current rate structure for energy usage and monthly demand charges to calculate the savings for summer, winter, and transition months; we determined the total yearly savings by summing the monthly savings and subtracting the estimated yearly maintenance cost of $4,000. Comparing the total yearly savings to the total cost of the ice TES system allowed us to estimate the payback period. As it is shown in Table 2.2, if the entire on-peak load could be covered by the ice storage system the payback period would be less than 5 years.

Table 2.2 Estimated Payback Period of Retrofit Ice TES System

Percentage of Load Demand	Summer Month Savings		Transition Month Savings		Winter Month Savings		Total Yearly Savings	Payback Period of Ice Storage
	usage	demand	usage	demand	usage	demand		
15%	$2,411	$1,221	$1,843	$1,139	$2,128	$1,139	$5,881	44.0
30%	$4,822	$2,443	$3,685	$2,278	$4,257	$2,278	$15,762	16.4
50%	$8,037	$4,071	$6,142	$3,796	$7,094	$3,796	$28,936	8.9
100%	$16,073	$8,142	$10,441	$6,454	$10,287	$5,504	$52,901	4.9

Trane™ and Calmac™ often work together on TES systems such as this and have proven to be successful in implementation. For example, the University of Arizona retrofitted CALMAC ice storage tanks into the existing campus cooling system. The ice storage system demonstrated monthly savings of $38,000 [Tarcola, 2009]. The majority of the yearly maintenance cost comes from the upkeep of the air-cooled condenser which entails compressor oil analysis, leak tests, and inspection of safety controls. The IceBank™ tanks and RTAC185 chiller are backed by warranty and since the tanks have no moving parts the only maintenance required is checking the quality of the glycol fluid, checking the water level in the tanks, and adding biocide every other year to eliminate algae growth. Additionally, Trane™ has developed software called IcePick™ that is designed for controlling ice storage systems in tandem with Trane™ chillers making this design attractive due to its short payback period and ease of setup and control.

2.2 PHASE CHANGE MATERIAL STORAGE SYSTEM

Phase change materials (PCMs) are gaining recognition as a viable alternative to water as the storage medium. The most popular PCMs are fatty acids and salt hydrates. Although PCMs have a lower latent heat of fusion than water (thus less storage capacity) they can be designed to change phase at various temperatures allowing for flexibility in thermal storage system designs. PCM™ is a worldwide company that designs various phase change materials for temperature control management. They also have designed storage tanks that prevent separation of the PCM and corrosion of the tank. The system design proposed would use the S10 PCM (salt hydrate produced by PCM™) with a PlusICE™ storage tank that would be retrofitted to the current chiller system in Hodges library. The S10 salt hydrate has a specific heat of 1.9 J/(g·ºC) and a latent heat of fusion value of 155 J/g. Because the S10 PCM is designed to change phase at 10°C (50°F), this would allow the newly installed centrifugal chiller to charge the storage tank during off-peak hours while the other centrifugal chiller provides for the cooling load. This is possible because the PCM melting temperature lies between the chiller supply brine temperature (7.2°C,45°F) and return brine temperature (12.8°C,55°F); thus 7.2°C (45°F) brine can be provided to solidify the PCM during the charging cycle. During discharge the thermal storage tank would lower the temperature of the brine returning to the chiller to 10°C (50°F), reducing the temperature difference the chiller must provide by one half. Because the new chiller being installed has a variable frequency drive (VFD), the compressor will not consume as much energy to supply brine at 7.2°C (45°F) which will greatly reduce chiller electricity usage during on-peak hours. The schematic for the proposed PCM TES system can be seen below in Fig. 2.3.

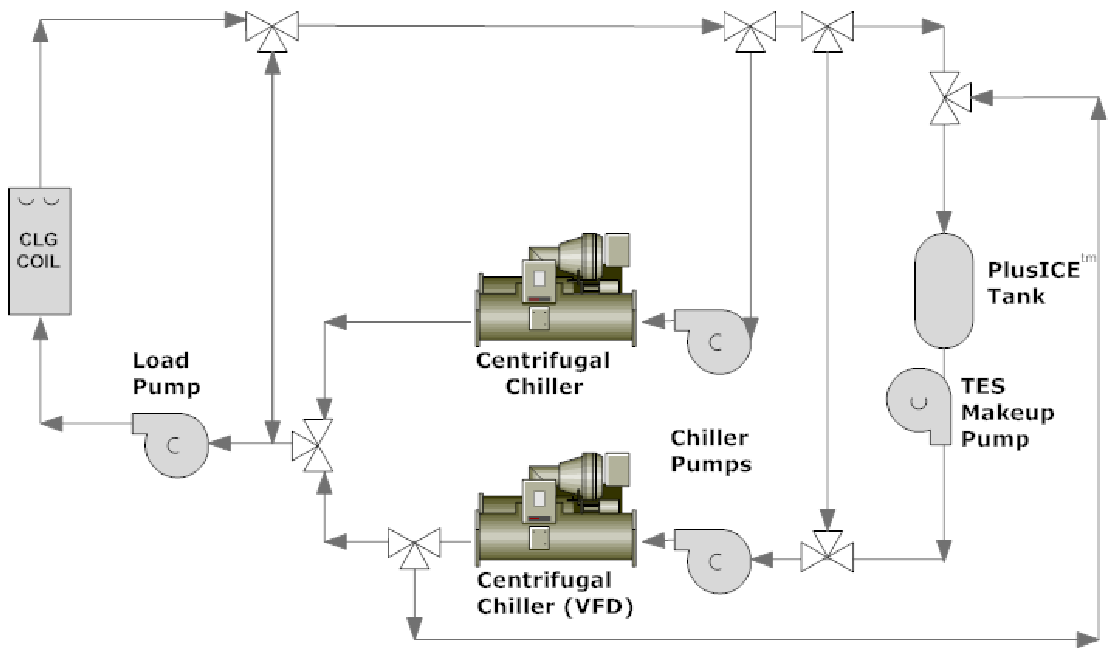

Figure 2.3 Schematic of the retrofit thermal storage system using a phase change material as the storage medium.

Fig. 2.4 below displays the PlusICE™ storage tank design. The S10 salt hydrate would remain enclosed within the FlatICE™ containers shown. Each of these containers has internal support beams so that multiple containers can be stacked on top of one another.

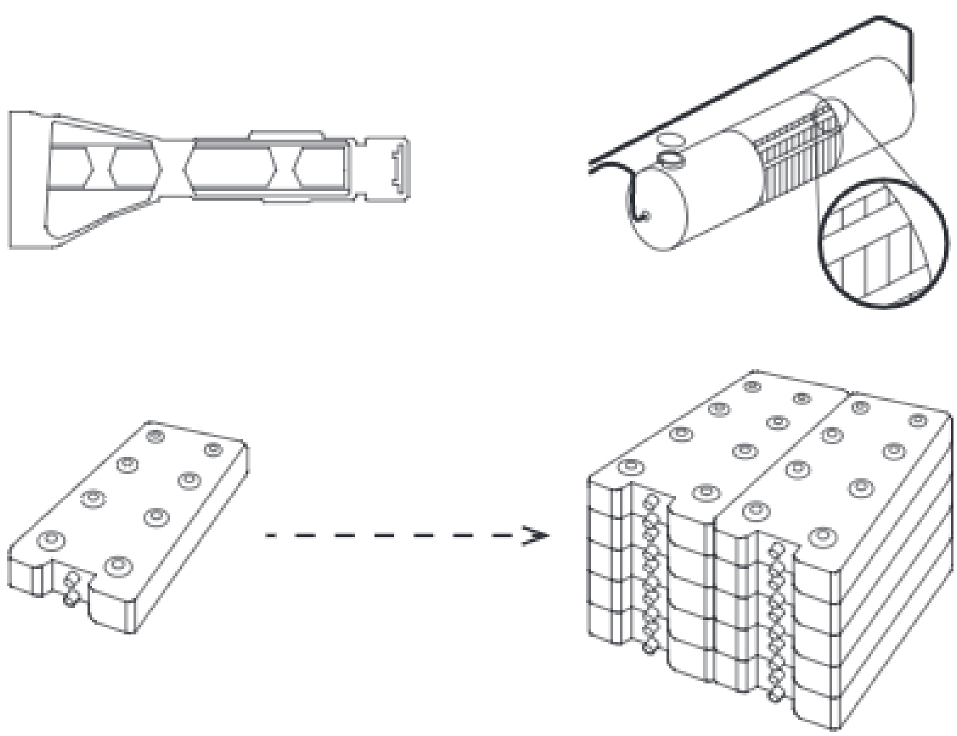

Figure 2.4 PlusICE™ storage tank design for the S10 salt hydrate.

The containers are designed to allow the brine circulated by the chillers to flow evenly between each container with the only drawback being a loss of fluid pressure as the brine flows through the storage tank. This loss however is counteracted by the tank being pressurized to 150 psi and any additional pressure head loss can easily be made up with a pump. A circular storage tank is illustrated in Fig. 2.4; however, the proposed storage tank would be rectangular so that the majority of the circulating brine must pass between the stacked FlatICE™ containers; not around them. The storage tank would have a footprint of 9 ft by 39.5 ft which would stand at a height of 8 ft and would include 16,900 FlatICE™ containers. The PCM storage system would have a cooling capacity 1060 RT-h and would be able to discharge for the entire on-peak period. The total budget equipment cost including installation is shown below in Table 2.3.

Table 2.3 Total Cost of Retrofit PCM TES System

System Equipment	Equipment Cost	Total
S10 PCM Material*	$200,000	$300,000
Pressurized Storage Tank*	$100,000	
*Installation Costs Included		

The simple payback is presented in Table 2.4 and was determined based on the percentage of the on-peak load that could be reduced by the TES system. For instance, a 30% reduction in on-peak load demand would alleviate 30% of the energy consumed by the centrifugal chiller compressor during the on-peak period. Savings for summer, winter, and transition months are based on current rate structure for energy usage and monthly demand charges. The total yearly savings were deduced by summing the monthly savings and subtracting the estimated yearly maintenance cost of $1,000. Comparing the total yearly savings to the total cost of the PCM TES system allowed us to estimate the payback period. Unlike the ice TES system; 100% reduction of on-peak load is not shown because the PCM TES system could at most alleviate 50% of the cooling load due to the nature of its placement in the plant. As it is shown in Table 2.4, if 50% of the on-peak load was covered by the PCM storage system the payback period would be just over 9 years.

Table 2.4 Estimated Payback Period of Retrofit PCM TES System

Percentage of Load Demand	Summer Month Savings		Transition Month Savings		Winter Month Savings		Total Yearly Savings	Payback Period of PCM Storage
	usage	demand	usage	demand	usage	demand		
15%	$2,411	$1,221	$1,843	$1,139	$2,128	$1,139	$8,881	33.8
30%	$4,822	$2,443	$3,685	$2,278	$4,257	$2,278	$18,762	16.0
50%	$8,037	$4,071	$6,142	$3,796	$7,094	$3,796	$31,936	9.4

Since the PlusICE™ storage tank has no moving parts the only maintenance required would be the periodic check of tank pressure and inspection of the FlatICE™ containers for leaks. Because PCMs are relatively new in TES application, the cost is relatively high. However as PCMs see more use, it is projected for the price to drop making their implementation even more appealing.

3 REDOX FLOW BATTERIES FOR LARGE-SCALE ENERGY STORAGE

Since the advent of electricity, there has been an engineering challenge to develop ways to efficiently store and release generated electrical energy. Batteries were the answer to the question of electrical storage, but primitive batteries could not discharge the electricity efficiently nor hold the charge for extended periods. In the past few decades, there has been a push to create highly efficient batteries. This change can be seen from the departure from traditional battery types such as lead-acid and Nickel-Cadmium (NiCad) to Lithium-Ion. Lithium-ion is more efficient at charging, discharging, and storing energy than lead-acid or NiCad batteries. Lithium-ion batteries typically are capable of twice the energy density of lead-acid batteries and have approximately a 37% longer life cycle (charge + discharge = 1 cycle), [Albright, 2012]. Due to the increase in renewable energy generation from solar and wind farms in recent years, a less conventional battery technology has begun to emerge into industry. Redox flow batteries (RFBs) have been under development since the 1980s; however, issues regarding electrolyte fluid corrosiveness and cell membrane materials inhibited their introduction into the industry. A basic schematic of a redox flow battery follows (Figure 3.1).

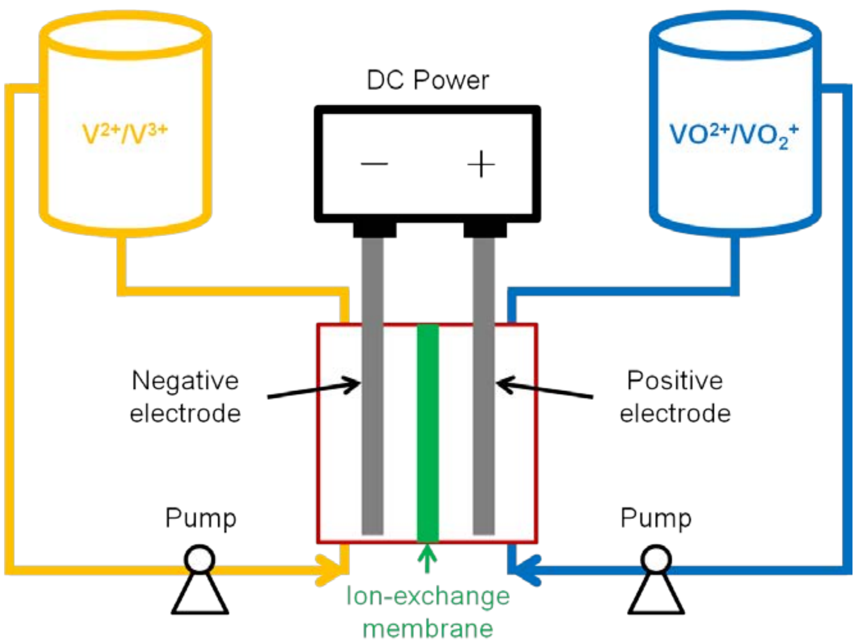

Figure 3.1 Redox flow battery schematic. [Photo Credit: Xie]

The redox flow battery consists of two tanks containing an electrolyte fluid, two pumps to pump the fluid back and forth across the membranes, cells containing the membrane, and stacks comprised of multiple cells. Charging and discharging is initiated by simply pumping the electrolyte fluid from one tank to the other. The membranes within the cells let the positive ions of the fluid pass through, but not the electrons. Hence charge is created and dissipated as the fluid is forced through the membrane. It is important to note that the electrolyte fluid is not consumed during the ion exchange process, it is merely a change of metal valence ions [Alotto, 2013] implying the flow battery can have exceptionally long lifespans. One of the main factors drawing industry to flow batteries is their uniqueness in the way which they deliver power and energy independently. The amount of energy that can be stored (kWh) is determined by how much volume of electrolyte fluid is in the tanks. The power which can be delivered (kW) is reliant on the size of the cell stacks. Also, in tandem with the scalability, the stacks can be modified to deliver different amounts of voltage or current depending on stack arrangement in series or parallel. For instance, if a load of 25 kW needs to be supplied via the flow battery and the equipment needs to run for 3-hours, a system would have a cell stack which can supply 25 kW of power and tanks which can supply 75 kWh of energy. This example assumes 100% efficiency of the battery and no parasitic loads, such as the electrolyte fluid pumps. Degradation due to efficiency losses and parasitic loads will be discussed further in a later section.

3.1 REDOX FLOW BATTERY STATE OF THE ART

Several flow battery technologies were tested over the years comprised of varied electrolyte solutions paired with different types of membranes. Early on, one of the main hindrances to the technology was the lack of a membrane material which could resist the corrosive nature of electrolyte solutions. Iron-Chromium electrolyte solutions were utilized in some of the first flow battery experiments; however, no compatible cell membrane was available, so the solution chemically reacted with the membrane which decreased the lifespans of the cells. No membrane type was specified by the researcher concerning Fe-Cr flow batteries. [Alotto, 2012] The issue of cell membrane degradation was overcome with the use of Nafion, a Teflon based fluoropolymer produced by DuPont. Currently, there are three RFB types leading in research as well as the commercial market: Vanadium Redox Batteries (VRBs), Zinc-Nickel (ZNFB), and Hydrogen-Bromine (HBr).

3.1.1 VANADIUM REDOX BATTERIES

All-Vanadium redox flow batteries are currently the most widely adopted RFB technology on the market. The system is comprised of a solution of vanadium dissolved in aqueous sulfuric acid (~5M) and a Nafion membrane for the cells. It was found that the vanadium solution was extremely stable compared to Fe-Cr and paired well with the Nafion membrane, leading to extremely long lifespans in terms of batteries. Projections and research state that VRBs today have an expected lifespan of 30 years with proven reliability over 100,000 cycles. [Shigematsu, 2011] The stability of both the electrolyte and membrane also means that yearly maintenance is limited to pump maintenance and recharging the electrolyte fluid to ensure its stability. Vanadium electrolyte solution also has a high energy density of ~50 kJ/L of solution. [CellCube, 2015] The higher the energy density, the less the amount of storage space needed for a right-sized VRB system. It is also important to note that VRBs have a 75% round-trip efficiency, meaning that from power generation to end use, 25% of the energy will be lost due to system inefficiencies Currently, cost is the limiting factor to VRB application. VRBs typically cost $400 to $450 per kWh. In comparison a lithium-ion production plant with a manufacturing capacity of 2 GW/year can produce batteries for $237 per kWh due to bulk production. [Ha, 2015] As can be seen, VRBs are currently $200 per kWh more expensive than Li-ion, which in a 2 MWh storage system can mean a cost increase of almost $450,000. However, like any technology, as VRBs become more widely adopted and manufacturing increases, the cost per kWh will decrease. If a 2 GW/year plant was created for VRB production it is predicted that costs could reduce to $158 per kWh, making VRBs an extremely competitive option to Li-ion for energy storage. [Ha, 2015]

3.1.2 ZINC-NICKEL

The Zinc-Nickel flow battery (ZNFB) is a different style of flow battery than the VRB. The ZNFB has no membrane. It is considered a semi-solid flow cell (SSFC) meaning that energy is transferred via slurry-like electrodes interaction with solid particles and nano-scale conductors [Liu, 2015]. In theory, since the slurry ZnNi solution has a higher concentration of energy components, it should have a higher energy density than that of vanadium solution. Bench test results showed energy densities of 460 kJ/L at a coulombic efficiency of 60 to 70%. [Liu, 2015] Even though ZNFBs have much higher energy densities than VRBs, they are still in the experimental phase and not commercially available.

3.1.3 HYDROGEN-BROMINE

Hydrogen-Bromine (HBr) flow batteries offer a more cost effective alternative to VRBs. Previously, HBr flow batteries were plagued with low lifecycles due to the degradation of platinum catalysts by the HBr solution. Through the application of platinum-iridium catalysts, researchers have been able to stabilize HBr batteries and increase their lifecycle to something similar to a VRB. Also, the HBr battery utilizes a composite cell membrane comprised of Nafion and polyvinylidene fluoride (PVDF) which is a cheaper alternative to that of traditional Nafion membranes. Since hydrogen and bromine are more readily available than vanadium, the HBr electrolyte solution is also less expensive than VBR electrolyte solution. With the decrease in cost of both the cell membrane and electrolyte, HBrs can return costs of approximately $210 per kWh, which is around $200 per kWh less than VRBs. [Lin, 2015] Although initial testing has proven that HBr flow batteries are a lower cost alternative to that of VRBs, they are not yet commercially available and require more development.

3.2 FIELD-VALIDATED VANADIUM REDOX FLOW BATTERY INSTALLATIONS

A multiplicity of flow batteries types are under study. Field demonstrations are needed to benchmark performance in industrial and commercial applications. However, the VRB has been tested time and time again in industrial and commercial applications ranging from peak load shifting at food processing plants to large-scale energy storage at wind or solar farms. For this reason, VRBs were chosen as the applicable flow battery technology for analysis of this project. VRBs have been implemented for a wide range of reasons and their usage is increasing as more manufacturers bring them to market. Currently, the major manufacturers of commercial VRBs are Prudent Energy, Imergy, CellCube, American Vanadium, and redT. VRB installations of key interest in regards to the UT Green Fee project were Gills Onions food processing plant and Fort Leonard Wood in Missouri.

Gills Onions in Oxnard, CA demonstrated the use of a VRB to lower costs by shifting electricity generation from on-peak to off-peak periods. The utility rate structure which Gills Onions prescribes to is similar to UTs in that there are on-peak and off-peak rates, with higher utility rates being charged during on-peak hours which is typically a six hour time period in the afternoon. Gills Onions has an on-site energy generation system which uses plant waste to create energy; their goal was to store the energy generated at night, when plant energy need was low, and use the stored energy during the six hour on-peak time period. Prudent Energy installed a VRB-ESS system comprised of three 200 kW modules with vanadium electrolyte storage to provide six hours of energy. Another benefit of the VRB was its ability to avoid exorbitant demand charges. The Gills Onions VRB-ESS was commissioned in 2012 and has been estimated to save the company over $100,000 per year in electricity bills. [ESA, 2015]

Another installation bearing semblance to the UT Green Fee project is the air conditioning system powered via a VRB at Fort Leonard Wood, Missouri. A Prudent Energy 5 kW/20 kWh system was installed to supply power for HVAC equipment. Field data was analyzed to document the response time of a VRB as well as how it behaves in a microgrid system. The VRB was charged via a 6 kW photovoltaic (PV) array while the system load was generated by two pumps, two condensers, resistive heating elements, and the HVAC system. Testing was performed during May 2013 and the largest peak demand observed was 2 kW. During the day, the HVAC system was powered by the PV array and excess power was sent to charge the VRB. The VRB was then discharged during times of low sunlight or at nighttime to power the HVAC system. It was determined that the VRB could supply power to the HVAC equipment adequately and even respond to the change in loads created by the equipment in milliseconds. [Qui, 2014] The VRB application at Fort Leonard Wood applies directly to the UT Green Fee project. The Hodges Library is keenly interested in using energy storage methods to supply power for cooling loads and a VRB has the ability to reliably supply power to the HVAC equipment. These field demonstrations validate VRB systems ability to shift electricity loads for cooling without compromising the resilience of the grid.

3.3 VANADIUM REDOX FLOW BATTERY FOR PEAK SHIFTING OF THE HODGES CHILLER PLANT

Vanadium redox flow batteries are a proven technology with a lifespan exceeding 30 years. VRBs are also proven to work with HVAC systems and as large scale energy storage for peak shifting. The John C. Hodges chiller project differs slightly from the reviewed installations, but is a mixture of both large scale storage as well as HVAC power supply. Through the implementation of a VRB system, UT would be able to shift the cost of running the chiller plant to off-peak utility rates, avoiding high on-peak energy rates as well as on-peak demand charges. Currently, UT prescribes to a utility rate structure provided by the Knoxville Utility Board (KUB) which specifies off-peak rates as well as on-peak rates which is a six hour time period typically from 2-8 pm. Along with increased energy charges during on-peak hours, there is an increased on-peak demand charge, also, if power usage exceeds contract limits, the demand charge is even greater. A brief overview of UTs utility rates can be seen in the table below.

Table 3.1 UT Utility Rate Structure

Rate Season	Peak Hours	On Peak Demand Rate ($/kW)	On Peak Energy Rate ($/kWh)	Off Peak Demand Rate ($/kW)	Off Peak Energy Rate ($/kW)
Transition	14:00-20:00	$14.57	$0.07	$5.32	$0.02
Summer	14:00-20:00	$14.57	$0.09	$5.32	$0.02
Winter	05:00-11:00	$14.57	$0.08	$5.32	$0.02

In addition to the rate structure above, an on peak demand fee of $14.92 per kW will be charged for any demand which exceeds contractual power limits. By implementing energy storage methods, the university can avoid costly on peak charges and ensure that peak demand limits are not exceeded year round.

3.3.1 HODGES VRB STORAGE

There are several hurdles to overcome prior to installing a large-scale VRB at the John C. Hodges Library. First, equipment necessary to the operation of the chiller plant must be metered to determine how much power they draw as well as how long the equipment will be operating during the six hour on peak period. Knowing how much power is drawn from the grid during plant operation is crucial for properly sizing the cell stacks in the VRB. For instance, the current chiller is rated at 360 kW. To run the chiller at full capacity, minus any ancillary pumps or fans, the VRB must supply 360 kW of power. Once the power need is known, that kW value can be multiplied by six, the number of on peak hours, to determine how much energy must be stored to completely take the chiller plant off of the grid during on peak times. Using the previous example, to run the chiller for six hours, the VRB would need to supply 360 kW of power and the electrolyte storage tanks would need to store approximately 2,160 kWh of energy.

Once stack size and storage capacity of the VRB is determined, considerations must be taken regarding the climate in which the VRB will be operating. Typically VRBs are deployed in standalone packages which contain necessary controls, storage tanks, cell stacks, electrolyte pumps, and space conditioning equipment to maintain the VRB in an environment where the temperature ranges between 10-30°C [Qui, 2014]. Since the VRB in the library would be placed in a conditioned space, ancillary HVAC equipment for the battery would not be necessary, as the equipment room of Hodges Library will not see temperatures below 10°C or above 30°C. The current space available in the library is 27-ft by 40-ft by15-ft high which is 16,200 ft^3 (458.7 m^3). One of the largest prepackaged VRB available today is the Imergy ESP250, rated at 250 kW and 1 MWh of storage; it is comprised of one 40-ft shipping container and one 45-ft shipping container equating to 183 m^3 of space. Assuming that the VRB can be installed in a non-package form, there is enough space in the Hodges Library to install a VRB system rated at 500 kW and 2 MWh of storage with room to spare.

3.3.2 RIGHT-SIZING THE VRB SYSTEM

To correctly size any energy storage system for a chiller plant, the load profile of the plant must be known. The load profile allows the determination of the peak load demand and what the average load demand is. Given these data, steps can be taken to determine whether partial or full storage is more applicable in a given scenario. Currently, the Hodges Library does not have the necessary metering equipment to create a load profile; however, power transducers and data loggers are being installed to perform this task. Based on consultations with chiller operators, a load between 80-90% of the chiller capacity is seen much of the year. Therefore a mock load profile was created for the purpose of this study. During summer months (May, June, July, August, September) the chiller load is assumed to be operating at 100% capacity, in the transition months (March, April, October, November) the chiller is operating at 80% capacity, and during winter months (December, January, February) the chiller is running at 70% capacity. The Trane chiller installed in Hodges library has a full load (primary) power draw of 312.4 kW for 565 tons of cooling which will be used as 100% capacity. In order for the VRB to power the chiller plant during summer on-peak operating hours, it must be able to supply 312 kW of power and 1,872 kWh of energy. To account for flow battery inefficiency and depth of charge (around 75%), the flow battery was sized 25% larger than the capacity of the chiller plant; the VRB being 312.4 kW and 2,344 kWh. One benefit of flow battery energy storage over thermal storage is that the full capacity of the battery can be used year-round regardless of chiller load. The battery is sized to provide 100% of the load, but sometimes the load will not be 100%, reducing the amount of energy shifted to off peak hours and therefore reducing savings. However, if the battery were tied into the library’s main power distribution as well, it could be used to shift the load of other equipment in the library during times when the climate conditioning load is not 100%. A flow battery can deliver its maximum peak shifting capabilities 364 days a year, leaving one day for preventative maintenance, regardless of chiller load. Based on a full load of 312 kW/1,874.4 kWh, three flow battery sizes were analyzed: 15% load coverage, 30% load coverage, and 100% load coverage. Load coverage is based on the 565 ton maximum load during the summer. Note that the percent load coverage in transition and winter months, where the maximum load is smaller, will be larger than 15% and 30%. A detailed illustration of load coverage can be seen in the table 3.2.

To provide 100% load coverage for summer loading, the VRB will require 155 m^3 of vanadium solution, 30% load coverage would require 46.7 m^3 of solution, and 15% load coverage would require 23.4 m^3 of solution. The volumes stated here are the tank sizing to store the vanadium electrolyte, more space will be required for installation of cell stacks and ancillary equipment.

Table 3.2 Hodges Library Chiller Plant Load Coverage

	Summer Load 100% - 565 RT	Transition Load 80% - 456 RT	Winter Load 70% - 399 RT
Load Coverage	15% - 85 RT – 30.6 kW	19% - 85 RT – 30.6 kW	21% - 85 RT – 30.6 kW
	30% - 170 RT – 61.2 kW	37% - 170 RT – 61.2 kW	43% - 170 RT – 61.2 kW
	100% - 565 RT – 312.4 kW	100% - 456 RT – 164.2 kW	100% - 399 RT – 143.6 kW

3.3.3 PAYBACK PERIOD DEPENDING ON LOAD COVERAGE

Analysis of the operation and payback of the VRB to shift chiller power consumption differs from that of the analysis of thermal storage. In thermal storage, a separate chiller, generally air-cooled, is used to freeze ice in tanks at night, taking advantage of the lower off-peak utility rates. During the day, the chiller is shut off or idled and the brine is pumped through the ice tanks and chilled instead of passing through the chiller. The VRB takes advantage of low utility rates during off-peak hours by charging at those times, such as overnight. During on-peak hours, the chiller power supply is then switched from the Hodges Library main to the flow battery, effectively running the chiller at off-peak utility rates during on-peak hours. However, savings cannot be calculated simply as the cost difference between on-peak and off-peak rates. Since a flow battery is generally 75% efficient, 25% more energy (kWh) than what is required must be stored in order to power the chiller for a full six hours, meaning operating the chiller via flow battery actually consumes more energy than operating it straight from the grid. Even with a 25% increase in energy consumption the cost difference between on-peak and off-peak rates allows for the generation of substantial savings. The payback period for the VRB also depends on the initial cost and lifecycle cost of the system. Flow battery cost per kWh decreases as the size of the battery increases, so the greatest payback potential occurs with a larger load which can be shifted. The lifecycle of a VRB has been demonstrated to be over 100,000 cycles and general maintenance is limited to routine pump maintenance as well as the restoration of the electrolyte fluid. Restoration of the fluid is done by cycling the battery without any load being pulled from it. Maintenance costs are assumed to be $1,000 per yr. Since the peak summer load must be covered, the VRB was sized to accommodate peak loading criteria. A detailed cost structure for the respective load coverage is provided Table 3.3 below.

Table 3.3. Flow Battery Cost

Summer Load Criteria			Battery Size		Battery Cost ($/kWh)		Total System Cost	
Load	**Load Coverage**	**Tons**	**kW**	**kWh**	**Current**	**Near Term**	**Current**	**Near Term**
100%	15%	85	47.0	324.3	$793	$472	**$279,513**	**$166,521**
	30%	170	94.0	705.1	$518	$321	**$365,250**	**$226,576**
	100%	565	312.4	2343	$399	$239	**$934,617**	**$559,331**

The cost of each battery system was determined using Pacific Northwest National Lab's (PNNL) flow battery cost calculator [PNNL, 2015]. Many parameters can be altered in the calculator, but only two parameters were changed: depth of charge (DOC) and Nafion cost. The DOC was set to 75%, meaning the battery is 75% efficient, and the cost per sq. meter of Nafion was changed to $750/m^2. The program returned two battery costs for each sizing, current cost and near term cost. Near term cost is how much the flow battery can be expected to cost 5 years in the future when the technology is more mature and widely adopted. Note that the battery sizing listed in Table 3.3 is 25% greater than the amount of energy the chiller uses to compensate for battery inefficiency. When 15% of the maximum load is covered, a 47 kW/353 kWh VRB is required at a current cost of $793/kWh. In comparison, a battery which covers 100% of the maximum load is 312.4 kW/2,344 kWh at a cost of $399/kWh. At current costs, to cover 100% of the maximum load, a vanadium redox flow battery system would cost $934,617 and would cost $559,331 in the near term. Due to the newness of commercially available flow batteries, the initial capital cost is exceptionally large, but cost is expected to decrease as evidenced by a near term system cost over $350,000 cheaper.

With a capital cost approaching one million dollars, the VRB system must be used to shift on-peak electrical loads equivalent to its full capacity as many days during the year as possible. This fully-loaded operating criteria as well as traditional operating schedule similar to thermal storage were compared. The load profile was split into three sections where 565 Tons was the maximum load: Summer – 100%; Transition – 80%; and Winter – 70%. When using thermal storage, if the system is sized to cover 100% of the load and the library is operating at 70% of the maximum load, 30% of the thermal storage capacity is not active. Only the current chiller load can be compensated for and no other equipment, operating the VRB in this way was called Traditional Operation. The first analysis of VRB payback was completed following Traditional Operation, where any leftover capacity after powering the chiller was not utilized. The second VRB analysis was coined Optimum Operation and uses any leftover capacity of the battery system to power other equipment in the library, utilizing the VRBs maximum potential year-round. Based on the previously described load profile, load coverages of 15%, 30% and 100% were analyzed through traditional operation to calculate yearly savings generated from shifting the electrical load of the chillers. Savings were split into two sections, energy charge savings and demand charge savings, these two sections sum together to yield total savings generated by the system. The bar graph (Fig. 3.2) illustrates and compares potential yearly savings generated through traditional operation at each load coverage.

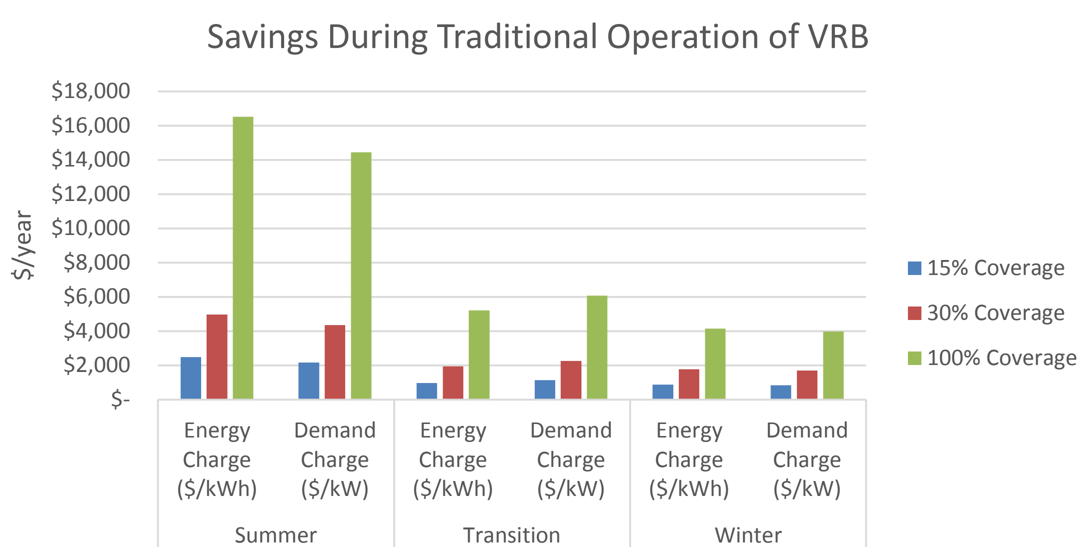

Figure 3.2. Yearly Savings Generated by Traditional Operation of VRB.

The total amount of money saved through traditional operation of the VRB at 15% load coverage was $8,497 per year, $16,993 per year for 30% load coverage, and $50,394 per year for 100% load coverage. While each load coverage amount has the potential to save the university thousands of dollars per year, the large initial cost of the flow battery system means that even with large sums saved each year, the payback period is exceptionally long. Table 3.4 below summarizes the payback period, including lifecycle cost, for each load coverage percentage in traditional operation.

Table 3.4. Payback Period for Traditional Operation of VRB

		Battery Cost					Yrs to cover Battery		Payback Period	
Load Coverage	**Savings per Year**	**Current**	**Near Term**	**Maintenance Cost/yr**	**Install (15% of avg. system cost)**	**Yrs to cover Install**	**Current**	**Near Term**	**Current**	**Near Term**
15%	$ 8,497	$ 279,513	$ 166,521	$ 1,000	$ 33,453	4.5	37.3	22.2	**41.7**	26.7
30%	$ 16,993	$ 365,250	$ 226,576	$ 1,000	$ 44,386	2.8	22.8	14.1	**25.6**	16.9
100%	$ 50,394	$ 934,617	$ 559,331	$ 1,000	$ 112,046	2.3	18.9	11.3	**21.2**	13.6

According to a 2012 study by the United States Building Energy Efficiency Retrofits committee **[Fulton, 2012]**, an acceptable payback for the installation of a high-efficiency VFD chiller is 8-12 years. Similarly, many thermal storage applications have a payback period of 8-10 years. As illustrated by the above table, no VRB in traditional operation mode can achieve a payback of 8 to 10 years; the quickest being 21.2 yrs. Even when near term costs are used, the shortest payback period is 13.6 years.

Therefore the VRB was analyzed for yearly savings during optimum full-load operation which would maximize the savings potential of the battery system. Through optimum operation, any energy stored in the VRB which is not being used by the chiller is used to power other equipment in the library. For instance, in the winter when the loading is 70%, 30% of the battery's capacity is being not used and the potential for savings is decreased. This leftover energy could be used to provide additional power to the library if it was wired into the library's main power supply, effectively maximizing the VRB's savings potential all year long, Fig. 3.3.

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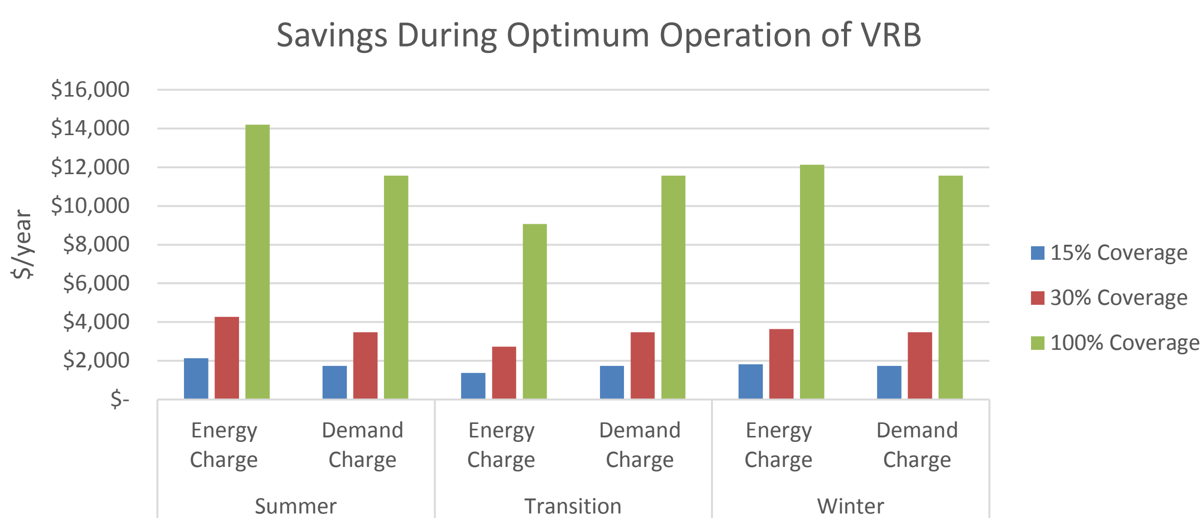

Figure 3.3. Yearly Savings Generated by Optimum Operation of VRB.

Through optimum operation of the VRB, the maximum amount of load which the VRB can shift is utilized year-round during on-peak hours, increasing the amount of savings generated over traditional operation. Optimum operation techniques have the potential to save $10,542/year with 15% load coverage, $21,085 per year with 30% load coverage, and $70,076 per year with 100% load coverage. Savings generated through optimum operation of the system at 100% load coverage are approximately $20,000 per year more than traditional operation. Due to the larger amount of yearly savings, the payback period under this type of operation decreases as illustrated below.

Table 3.5 Payback Period for Optimum Operation of VRB

		Battery Cost					**Yrs to cover Battery**		**Payback Period**	
Load Coverage	**Savings per Year**	**Current**	**Near Term**	**Maintenance Cost/yr**	**Install (15% of avg. system cost)**	**Yrs to cover Install**	**Current**	**Near Term**	**Current**	**Near Term**
15%	$ 10,542	$ 279,513	$ 166,521	$ 1,000	$ 33,453	3.5	29.3	17.5	**32.8**	21.0
30%	$ 21,085	$ 365,250	$ 226,576	$ 1,000	$ 44,386	2.2	18.2	11.3	**20.4**	13.5
100%	$ 70,076	$ 934,617	$ 559,331	$ 1,000	$ 112,046	1.6	13.5	8.1	**15.2**	9.7

The payback period through optimum operation is more acceptable than the payback generated through traditional operation. The fastest payback period is at 100% load coverage with a payback in 15.2 years for current battery costs and 9.7 years for near term battery costs. By utilizing the VRB's potential year round for more than just shifting chiller loads, it becomes a much more viable method of peak shifting. While a current payback period of 15.2 years is not optimal, it demonstrates that VRBs can effectively save money for the university when used for peak load shifting. With a near term payback of 9.7 years, VRBs will become a competitive option to thermal storage. The payback of a VRB system could be improved if supplemented with renewable energy resources. VRBs are proven to work well in storing energy produced by solar and wind farms and due to their long cycle life, the repeated cycling of the battery will not decrease its life. A future study of how VRBs combined with small scale renewables can reduce on-peak electricity usage should be explored, as it would reduce the payback period even further.

3.4 CONCLUSION OF VRB ANALYSIS

Vanadium redox flow batteries are a rapidly increasing storage technology on the cusp of mass commerciality. Multiple smaller companies manufacture VRBs for large and small scale industrial use with proven reliability and payback. Each VRB installation is analyzed individually and there are many factors which go into the analysis of whether a VRB will work for the given application. Many times, renewable resources, excess power generation, and very high utility rates (such as those in California) make a profitable case for VRB storage. In the case of VRB implementation for peak shifting of the Hodges Library chiller plant, renewable energy is not an option and the utility rates seen in Knoxville, TN are lower than much of country. These factors combine to decrease the profitability of a VRB installation for peak shifting at the library. However, calculations performed on approximated chiller load profiles have shown payback on the order of 15 years under the most optimum operating condition. Therefore charging a VRB during off-peak utility rates to be discharged during on-peak utility rate hours can save the university over $70,000 per year. With proven life over 100,000 cycles, the VRB will last well beyond its current payback period, and with a near term payback of 9.7 years it will be a competitive option to thermal storage in the future. Based on research, there has not been a case of a VRB being used in the methods described, but calculations and research show that a VRB used for simple peak load shifting is viable and can be a competitive energy storage solution.

4 COMBINED HEAT AND POWER LOAD SHIFTING

As members of the Green Fee Project team we were tasked with finding ways to save energy and/or utility cost for the University of Tennessee, specifically at the Hodges Library. As mentioned above, this is a multifaceted approach that includes exploring the possibility of battery storage (vanadium redox technology) and thermal storage (ice or phase change material). Another area in which significant potential for energy or financial savings exists is in the use of combined heat and power (CHP) systems with comfort conditioning with or without thermal storage. Therefore, the following section will explore the feasibility and potential payback of a stand-alone CHP system located in the basement of the Hodges Library. This section will discuss fundamentals of CHP design and feasibility studies, including descriptions of prime movers, heat recovery, and thermally activated technology. Also, two possible CHP systems will be proposed, as well as suggestions for future work to more accurately assess the potential return on investment for proposed CHP systems.

4.1 LITERATURE REVIEW OF ABSORPTION TECHNOLOGY

An extensive literature search was conducted and included absorption technology. Absorption technology typically refers to absorption chillers. For an absorption chiller, the compressor (as seen in traditional centrifugal chillers) is replaced by an absorber (that maintains a low vapor pressure in the evaporator) and a generator in which heat is added to boil the refrigerant. A number of issues limiting the durability and feasibility of absorption chillers have been well documented, including crystallization, high toxicity and corrosion, risk of Legionella bacteria, and tendency towards catastrophic failure. A significant portion of the literature review was devoted to the search for alternative working pairs (ammonia-water, ionic liquids, etc.), but none of these has proven to be a viable solution to the issues mentioned above. Therefore, absorption technology does not appear to be a successful technology, and there are no longer any manufacturers of absorption chillers in the United States, as we discovered at the AHR Expo.

4.1.1 AHR EXPO

In order to get more acquainted with the specific technologies and meet company representatives of potential equipment supply, we attended the Air Conditioning, Heating, and Refrigeration (AHR) Expo in Orlando, FL on January 25-26, 2016. The goal in regards to CHP systems was to talk to representatives and developers of absorption technology, desiccant dehumidification, and micro-turbine technology. A list of all companies and sales representatives talked to at the Expo can be seen in Appendix A.

Very few companies were willing to discuss absorption chillers, and therefore absorption was dismissed as a viable option for our application. Capstone Turbine Corporation manufactures modular micro-turbines that could fit nicely in our system. The most promising companies representing desiccant technology were Concepts and Designs, Inc., and Munters Corporation. Perhaps the most interesting and beneficial meeting at the AHR Expo occurred with Invensor. Carl Isohito, a representative from Invensor and AAA Save Energy at the AHR Expo introduced us to a different thermally activated technology, adsorption. The Invensor adsorption chillers could potentially perform the same function as absorption chillers, but without the debilitating issues discussed above. The adsorption process and other specifics will be described later.

4.2 CHP FUNDAMENTALS

Potential fuels used in CHP systems in the United States include oil, waste oil, wood, biomass, biofuels, liquid fuels (propane, etc.), coal, and natural gas, which is the preferred choice for many CHP systems. This is likely due to the relatively constant cost of natural gas (as opposed to rising electricity costs), as well as its low NO_x and CO_2 emissions. A simple measure of the feasibility of a proposed CHP system is the spark spread, or the "relative difference between the price of fuel and the price of electric power" [Foley, et al, 9]. Spark spread depends primarily on the efficiency of energy conversion of the system. According to the U.S. Energy Information Administration, a simple way to calculate spark spread is shown below in Equations 1-2, based on the annual costs of energy**.**

$$Spark\ Spread = \left(\frac{\$}{MWh}\right)_{Elec} - \left(\frac{\$}{MMBtu}\right)_{NG} * Heat\ Rate\ \left(\frac{MMBtu}{MWh}\right) \quad (1)$$

$$Spark\ Spread\ \geq \frac{\$12}{MMBtu} \quad (2)$$

As seen in Equation 2, CHP systems offer opportunity for significant payback if the spark spread is greater than $\$12/MMBtu$.

4.2.1 CHP DESIGN

The primary goal of a CHP system is to provide the system operator with a reasonable return on investment (ROI), although other benefits may be realized. Any CHP system can be divided into three main units: a unit in which the fuel is combusted, an electric generator, and a heat recovery unit. The device used to convert fuel to electricity is called the prime mover, and the selection of the correct prime mover is a critical issue for any CHP system. [Foley, et al, 14]

4.2.2 CHP DESIGN GUIDELINES

In order to overcome the often-large capital costs of a CHP system, the system should operate almost continuously at the rated load. Typically, CHP systems are most effective in situations where there are high coincident electric and thermal loads. Therefore, it becomes essential to fully understand the nature of the site's thermal and electric loads. Most CHP failures occur because the system is sized incorrectly to meet the site's thermal loads. The selection of a prime mover is primarily a function of site requirements, including the comparison of fuel cost and electricity cost (spark spread), the equipment capital costs, installation and permitting. [Foley, et al, 15]

4.3 PROPOSED CHP SYSTEMS

Two possible CHP systems are proposed, only differing in the choice of prime mover. The systems consist of either a micro-turbine (Capstone) or reciprocating engine (Tecogen) providing electricity and waste heat. The waste heat is used to drive a desiccant dehumidifier (Concepts and Designs), a heat recovery steam generator, and an adsorption chiller (Invensor). (Pertinent data for all equipment can be seen in Appendix A.)

The Hodges Library is potentially a good candidate for CHP because it has large electric loads (lighting and computers, etc.) that are coincident with the large cooling/heating loads (particularly in summer and winter). Also, the library has long operating hours, and potentially little variation in load (especially electric) for day of the week and time of day. For example, the library likely does not see as wide a variation between the electric loads at 10 am compared to the loads at 10 pm as would be expected for a typical office or commercial building. Nor should the library see drastic differences between the loads on Saturday or Sunday versus Tuesday or Wednesday. Also, using the CHP system in tandem with a thermal storage system provides more opportunity for benefit. Charging the thermal storage system at night allows for the CHP system to run at higher load during a time when the load would typically be lower.

Each of the proposed systems focuses on one main purpose and potentially three useful byproducts. The main purpose of the CHP system is to provide electric power to UT's grid and the library itself, thereby reducing the amount of electric power purchased from KUB, especially during on-peak periods. The waste heat from the micro-turbine (or reciprocating engine) can be used for three purposes. First, the heat can drive a desiccant dehumidifier, providing dry air to the library (necessary to preserve books and other documents) and reducing the latent load seen by the HVAC chiller. Second, the heat will be used to provide hot water as the driving input for adsorption chillers, resulting in a reduction of electricity needed to provide the cooling load for the library. Finally, the heat can be used in a heat recovery steam generator to provide steam if necessary to other nearby buildings on campus or for the condensate return to the main boiler plant. A schematic of the proposed system can be seen below in Figure 4.1.

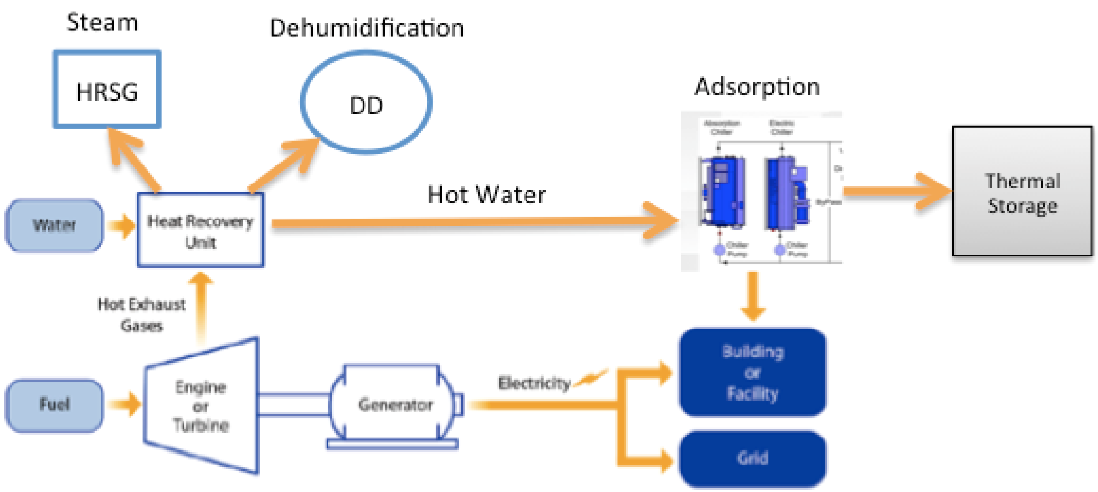

Figure 4.1. Proposed CHP Design

4.3.1 PRIME MOVER— MICRO-TURBINE

BASICS: There are two major producers of micro-turbines (MT) in the United States, Capstone and FlexEnergy. MTs typically can produce up to 1 MW when modular packages are used. The thermal output of MTs consists of exhaust heat at temperatures around 260-315°C (500-600°F), providing the opportunity for steam production. Typical fuels used include natural gas, sour gas, and liquid fuels such as gasoline, kerosene, diesel fuel and heating oil. Lifespans of MTs range from 40,000 to 80,000 hours, including major overhauls. MTs produce low emissions (NO_x), especially when fired by natural gas. [ICF International: Micro-turbines, 2]

Process and Components: MTs operate on the Brayton thermodynamic cycle, in which air is compressed, heated, and combusted with fuel to drive an expansion turbine. A schematic of a typical MT can be seen below in Figure 4.2.

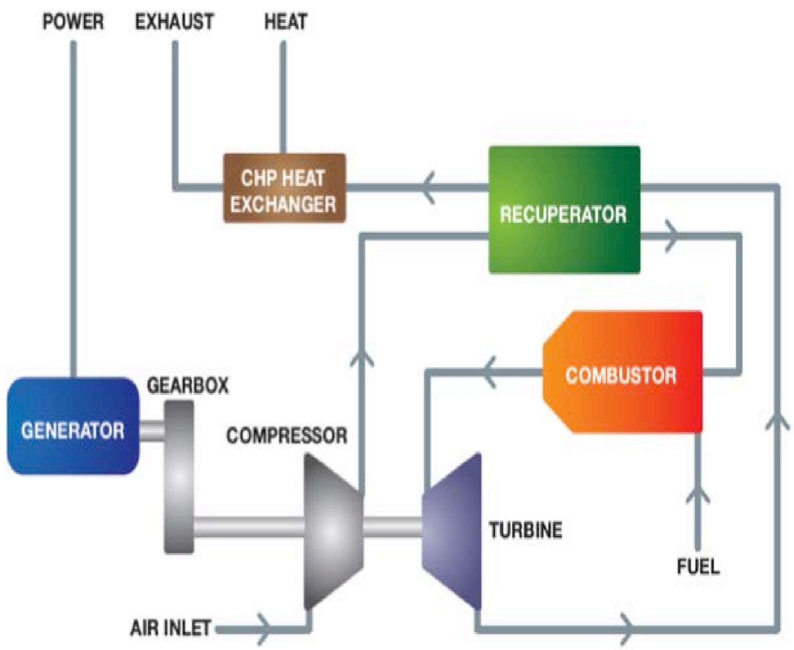

Figure 4.2. Micro-turbine

The expansion turbine in turn drives the inlet compressor and the drive shaft for power generation. The air entering the compressor is preheated using the hot gases of combustion in the recuperator, lowering the amount of fuel required to heat the compressed air. The use of a recuperator can increase the fuel efficiency of the MT tremendously, but can cause a 10-15% drop in power output. A heat exchanger extracts the remaining energy in the turbine exhaust exiting the recuperator. [ICF International: Micro-turbines 5-7]

Performance Characteristics: Most MTs are designed to operate at full load for as long as possible. Significant efficiency degradation occurs at loads less than 70% full capacity. For modular packages, load following can be achieved by successively turning on and off more units. The effect of ambient conditions on performance is also significant. As the ambient inlet air temperature increases, the power output and efficiency of the MT decrease. Power decreases because the higher temperature air is less dense, resulting in a lower mass flow rate. Efficiency decreases because the power requirement to the compressor increases as a result of the lower density air. The effect of ambient conditions is significant for temperatures above 20-25℃ (70-80°F). [ICF International: Micro-turbines, 8-10]

Maintenance and Availability: The ASHRAE CHP Design Guide suggests the following maintenance schedule.

- 8,000 hrs: replace air and fuel filters
- 16,000-20,000 hrs: replace fuel injectors, igniters, and thermocouples
- 20,000 hrs: battery replacement (standalone units)

- 40,000 hrs: major overhaul, core turbine replacement

The fuel compressor also requires maintenance and inspection. Modular packages and achieve availability as high as 98%. [Foley, et al, 129]

Emissions: When natural gas is used as the fuel, MTs have very low NO_x emissions, but carbon monoxide (CO) and volatile organic compounds (VOC) emissions can be high. However, CO/VOC emissions can be reduced significantly if necessary with the use of an oxidation catalyst. [ICF International: Micro-turbines, 18]

4.3.2 PRIME MOVER— RECIPROCATING ENGINE

Basics: Reciprocating engines (RE) typically range in capacity from 10 kW to 18 MW. The thermal output consists of hot water and low-pressure steam. REs are characterized by fast startup times, making them useful for peaking and emergency power applications. REs typically have high part-load efficiencies (compared to MTs), which is beneficial for electric load following. High electric efficiencies result in lower fuel-related operating costs. Most REs used today are natural gas-fired spark ignition engines. Good CHP site candidates for RE as a prime mover are those in which there exists relatively high coincident electric and hot water demand. [ICF International: Reciprocating Engines, 2-3]

Process: The RE operates on either the 4-stroke Otto (spark ignition) or 4-stroke Diesel (compression ignition) thermodynamic cycle. In both cases, the hot combustion gases expand, pushing the piston downward (power stroke) and causing the crankshaft to rotate, generating power, Figure 4.3. Diesel engine REs were more commonly used due to higher efficiencies (as a result of higher compression ratio and combustion temperature). However, stricter emission regulation standards have made Diesel-based

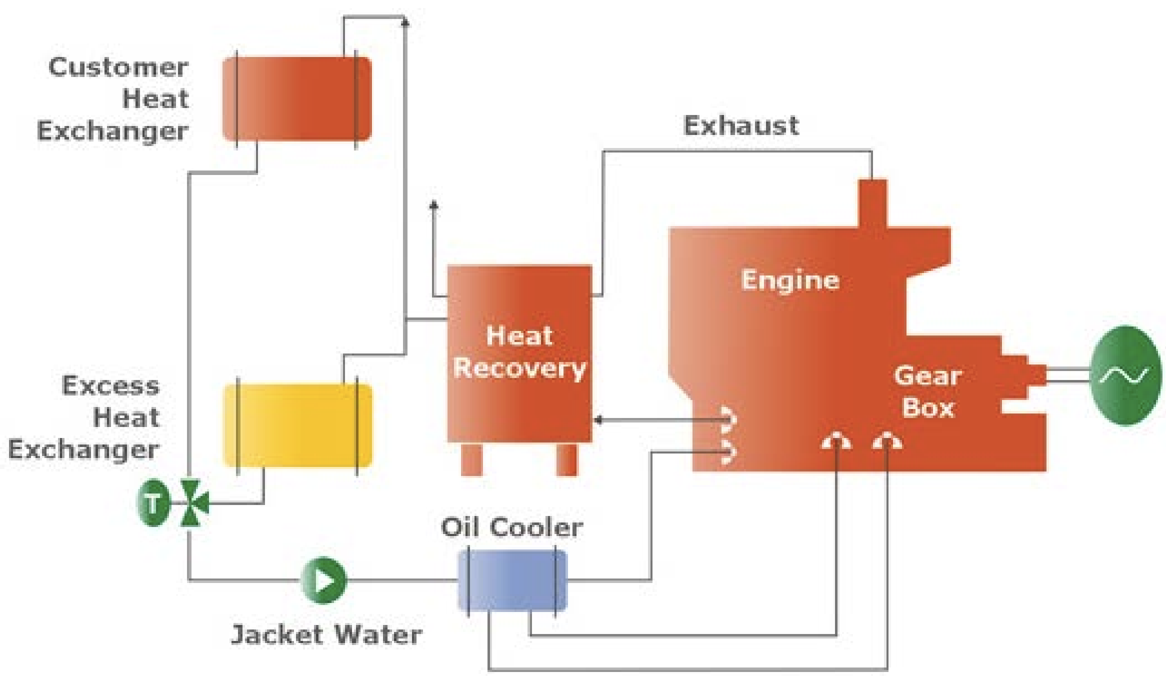

Figure 4.3 Reciprocating Engine.

REs more rare, and natural gas-fired spark ignition REs have become the norm. Heat recovery in REs consists of waste heat from the engine exhaust, jacket coolant, and lubrication oil cooler. Exhaust temperatures are often too low for steam production, but hot water ranging from 88-110℃ (190-230°F) can be produced. [ICF International: Reciprocating Engines, 6-9] The adsorption chillers (described below) require an inlet hot water temperature of 70℃ (158°F), so REs are more than capable of providing this input. A basic schematic of a RE can be seen in Figure 4.3.

Performance Characteristics: Typically, as is the case for just about any prime mover, the efficiency decreases with decreasing percentage of maximum load. In other words, REs run more efficiently at higher loads. However, the part-load performance for REs is significantly better than for gas turbines or micro-turbines. Increasing ambient temperatures also negatively affect the efficiency of REs, but not as severely as micro-turbines. [ICF International: Reciprocating Engines, 13-15]

Maintenance and Availability: Maintenance of REs consists of the following:

- 500-2,000 hrs: routine inspection and periodic replacement of engine oil filters, coolant, and spark plugs
- 8,000-30,000 hrs: tapered overhaul including cylinder head and turbocharger rebuild
- 30,000-72,000 hrs: piston replacement, crankshaft inspection, inspection of bearings and seals

RE availability is as high as 95% have been achieved. [ICF International: Reciprocating Engines, 20]

Emissions: Emissions are typically more of a concern for REs than for micro-turbines, but lean burn natural gas-fired REs produce the lowest NO_x emissions. Many of these issues can be alleviated using emissions control options, including combustion process control, post-combustion control, oxidation catalysts, Diesel particulate filters, three-way catalyst, and selective catalytic reduction. [ICF International: Reciprocating Engines, 26-7]

4.3.3 HEAT RECOVERY—DESICCANT DEHUMIDIFIER

Purpose: A desiccant dehumidifier (DD) is a type of thermally activated technology that provides dehumidification (dry air) using waste heat from the prime mover. Specifically, at the Hodges Library, the desiccant wheel can supply the library with dry air necessary for books and moisture-sensitive documents and for human comfort. The desiccant technology has the ability to reduce the latent cooling load seen by the chiller, potentially saving money on electricity costs.

Process: As mentioned above, a DD is driven by waste heat from the CHP system (i.e. micro-turbine or reciprocating engine). A diagram detailing the process can be seen in Figure 4.4.

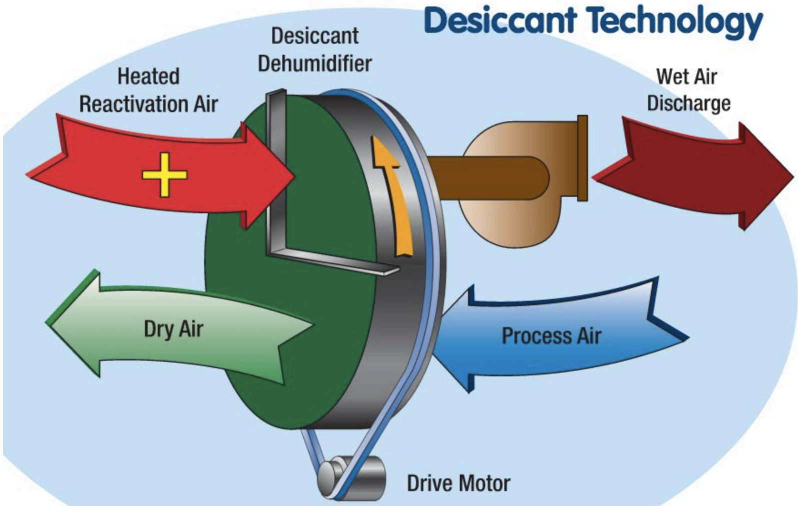

Figure 4.4 Desiccant Dehumidifier.

The DD consists of a wheel containing a desiccant material, usually a type of silica gel (for solid desiccants). As the DD wheel slowly rotates, moist outdoor air is allowed to pass over the silica gel bed. An area of low pressure on the surface of the desiccant material attracts the water vapor due to the air's higher vapor pressure. The moisture in the air adsorbs to the silica gel in a fashion similar to a sponge absorbing liquid water. The air that passes through the DD is now conditioned and can be supplied to the building. As the wheel continues to rotate, hot reactivation air (around 78℃ or 170°F) heated by the CHP exhaust passes over the adsorbed moist half of the desiccant wheel. The reactivation air picks up the moisture from the desiccant wheel and is exhausted to the outdoors. The now dry half of the wheel is available to adsorb moisture again, and the cycle repeats. [Foley, et al,184]

4.3.4 HEAT RECOVERY—ADSORPTION CHILLER

Purpose: An adsorption chiller is a type of thermally activated technology that utilizes the heat from a prime mover to provide cooling for a building or facility. In the case of the Invensor Adsorption Chillers, hot water at 70℃ drives the process. Specifically, at the Hodges Library, using adsorption technology allows for further reduction of the existing centrifugal chiller's cooling load, and more potential for electricity cost savings.

Process: Adsorption is a solid-vapor sorption process in which the sorbent is a solid (typically silica gel) and the sorbate is a gas (water vapor). (See Figure 4.5 below for a schematic of the process.)

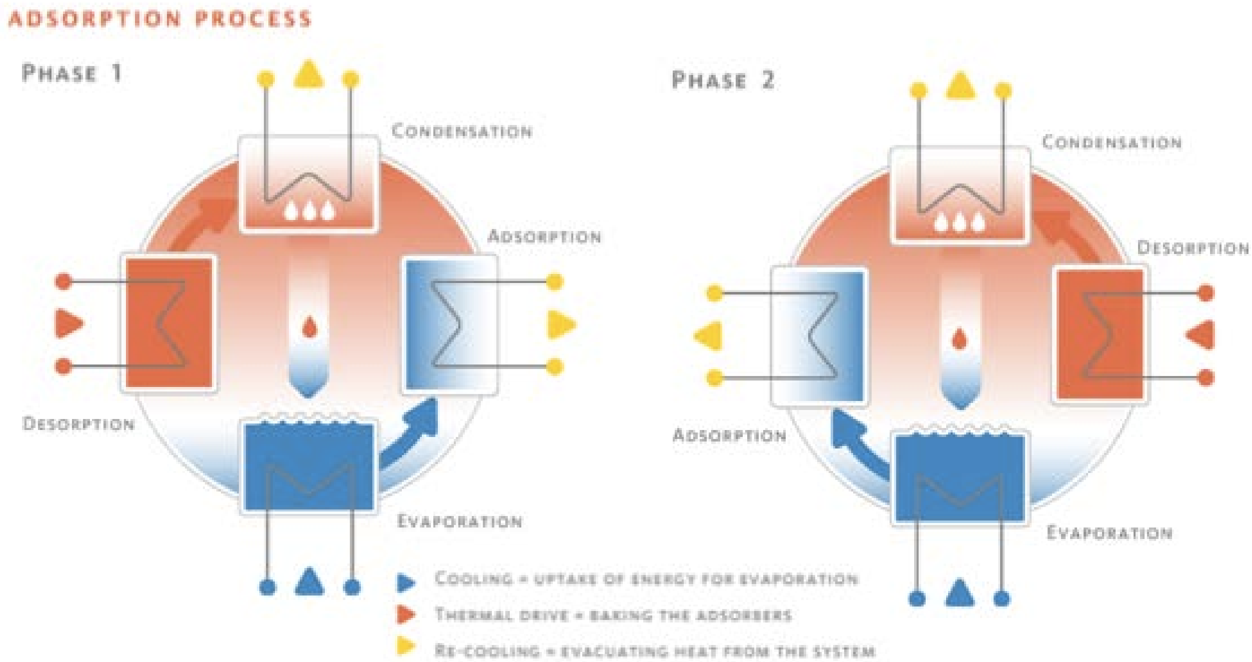

Figure 4.5 Adsorption Chiller.

The refrigerant adsorbent is water. The cycle differs from the typical vapor-compression cycle in that the compressor is replaced with dual adsorbing and desorbing beds that alternate between adsorption and desorption. Heat is added to a desorber, driving the water vapor from the silica gel. The water vapor enters the condenser, and the heat is exhausted to a cooling tower. Silica gel in the adsorber adsorbs cool water vapor from the evaporator. Once the adsorbate (silica gel) becomes saturated, the role of the adsorber switches to a desorber, and the silica gel is regenerated for further adsorption. The two beds alternate to supply a continuous flow of chilled water to the process. The low-pressure refrigerant (water vapor) chills brine in the evaporator, which, in turn, provides comfort conditioning to the library. [Foley, et al, 181-2]

4.3.4.1 Invensor Adsorption Chillers

A meeting with representatives from Invensor (including Carl Isohito, Sophie Schieler, and Hayo Angerer) on April 5, 2016 generated the following information specific to Invensor technology.

Capacity: At 30 kW capacity per chiller, 10 chillers connected in parallel can provide a maximum capacity of 300 kW, or approximately half of the estimated peak load. Invensor also has a 90 kW machine, and 7 chillers would handle the entire peak load for the library.

Specifications

- Capacity: 30 or 90 kW
- Driving Hot Water Temp: 70°C (can handle 60-100°C)
- Vacuum pumps at 10-30 mbar (subatmospheric)
- Part load: can run down to 40% peak load (switch off below 40%)

Maintenance, Durability and Availability: The Invensor chillers use an Activac vacuum pump system, resulting in very little maintenance within the system. The only regular maintenance consists of replacing exterior valves and pumps annually. There are no long heat-up periods as typically expected with absorption chillers.

Noise, Vibrations and Environmental Impact: The Invensor chillers have very low noise levels. There is no compressor, so the only noise comes from the turning and switching of exterior valves. The refrigerant is water, so there are very little corrosion issues. The chillers typically use a dry cooling tower, eliminating Legionella and other hygienic concerns associated with wet cooling towers.

Invensor Chillers Currently in Operation: As Invensor is a company based in Berlin, there are a number of examples of Invensor adsorption chillers in operation in Germany. The chillers were used at Oberflachentechnik (OFT) Dobeln, an electroplating company near Dresden. OFT Dobeln installed a CHP system at their facility, adding adsorption chillers in 2014. The chillers provide necessary heating and cooling for the electroplating process. (OFT Dobeln)

Another example of Invensor technology in use is at the TRANSpofix plant[1] located between the German cities of Nuremberg and Regensburgnea. The plant specializes in plastic injection molding. The company produces its own electricity with the help of three micro cogeneration units (CHP) and uses the resulting waste heat for heating and cooling. Three LTC 10 plus adsorption chillers provide cooling for the injection molding and cooling for the entire 900 square foot office building during the summer. The company also installed hot-water and cold-water buffer tanks to store heat in order to meet its needs in the event of high demand for cold or hot water. Representatives state that cost savings for the year-round use of the three CHP units and the InvenSor adsorption cooling units has paid for the entire system cost in only five years.

4.3.4.2 Adsorption at UT and Prime Mover Selection Concerns

Although these particular applications are much smaller in capacity than the project at the library, by connecting multiple chillers in parallel, it is possible to achieve cooling capacities that could provide significant benefit to UT. Talking to the representatives from Invensor, many of the installations in Germany have worked well in CHP systems in which the prime mover is a reciprocating engine. The high electric efficiency of reciprocating engines coupled with the potentially high electric loads of the library make a reciprocating engine an intriguing choice for prime mover in our CHP system. The choice of prime mover may ultimately be decided based upon the need for or potential benefit from CHP steam production. If there is a benefit to UT of additional steam from the library's CHP system, then a micro-turbine is ideal because of the high exhaust temperatures. If steam generation is not a perceived benefit, then a reciprocating engine will likely be a better option. Of course, sizing, emissions, and cost are separate issues that will affect prime mover selection as well.

4.4 ECONOMIC ANALYSIS

Ultimately, the acceptance of a CHP system will be based upon economics. In other words, the system is successful if a return on investment can be achieved that is acceptable to the UT building owners. (Please see "CHP Calcs" Excel Document for all following calculations).

[1] http://chiefengineer.org/?p=12856&page=2

4.4.1 SPARK SPREAD

The first step in determining the potential feasibility of a CHP system is to calculate the spark spread. As mentioned above, the spark spread is the relative difference between the cost of electricity and the cost of natural gas. Typical spark spreads for various regions in the U.S. can be seen below in Figure 4.6. As seen in the graph, the spark spreads in the Southeast are generally lower than for larger cities in the Northeast and Midwest, thus sizing the CHP system specifically to the library's load will be an essential issue.

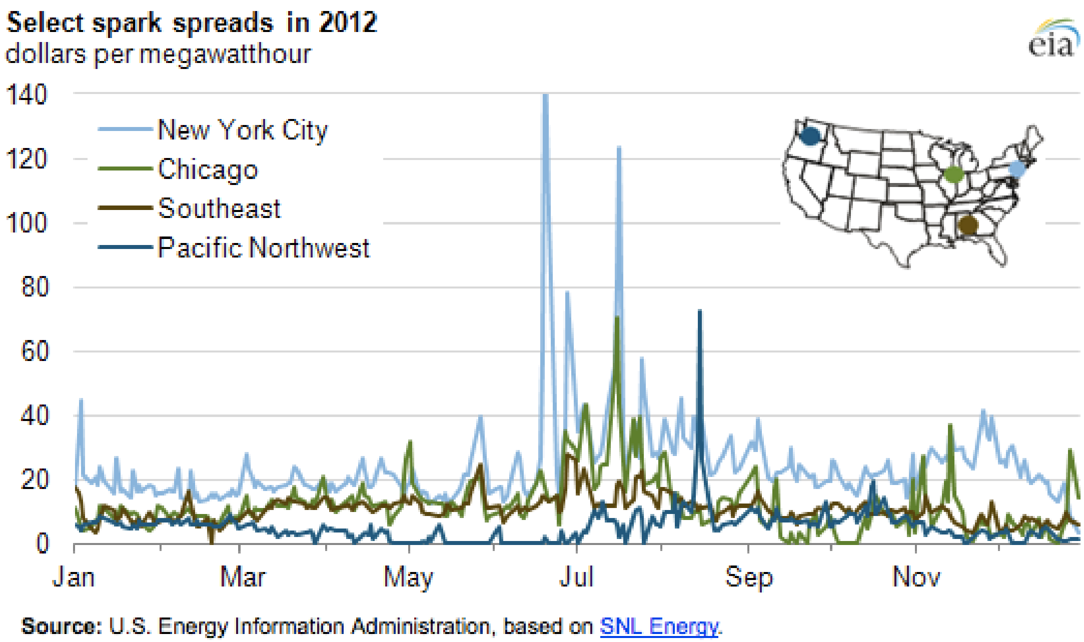

Figure 4.6 Spark spread by region of U.S. in 2012

Using total electricity and natural gas cost and usage data from 2014, as well as heat rate data for the micro-turbine and reciprocating engine, the spark spread was calculated using Equation 1 listed above. The results are seen below in Table 4.1

Table 4.1 Spark Spread

Prime Mover	Spark Spread ($/MW-h)	Spark Spreak ($/MMBtu)
Micro-turbine	82.94	24.30
Reciprocating Engine	126.99	37.21

Note that the spark spread for the reciprocating engine is significantly larger than for the micro-turbine. This exemplifies the higher electric efficiency exhibited by the reciprocating engine. However, both prime movers show that potential for payback exists given the current electricity rate structures and natural gas prices.

4.4.2 PEAK LOAD COVERAGE

The CHP system was analyzed at various percentages of peak load in order to estimate the amount of savings due solely to peak load shifting. Specifically, the ability of the CHP system to cover the daily 6-hour peak load of the existing chiller at 100%, 50%, 30%, and 15% was analyzed economically. For this analysis, the chiller was assumed to run at 100% from May through September, 80% from March, April, October and November, and 70% from December through February. The results of this analysis are tabulated in Table 4.2

Table 4.2 Peak Load Coverage (Yearly Savings)

Peak Load Percentage	Micro-turbine ($)	Reciprocating Engine ($)
100%	23,720	32,080
50%	10,960	14,780
30%	6,580	8,870
15%	3,290	4,440

Once again, the reciprocating engine shows significantly better potential for savings due to its higher electric efficiency.

4.4.3 PAYBACK

While peak load coverage is likely a good measure for effectiveness of thermal storage, a better measure for CHP systems is the overall payback, because CHP generates its own electricity as opposed to simply shifting the electricity to off-peak periods.

Equipment Costs: Table 4.3 shows total equipment costs for the micro-turbine, reciprocating engine, desiccant dehumidifier, and adsorption chiller for various sized systems.

Table 4.3 Total Equipment Costs (in thousands of U.S. Dollars). Assumes 10 adsorption chillers, each of 30 kW capacity.

Capacity (MW)	MT ($k)	RE ($k)	DD ($k)	AC ($k)	Total (MT) ($k)	Total (RE) ($k)
2	6,000	4,000	200	425.6	6,625.6	4,625.6
3	9,000	6,000	200	425.6	9,625.6	6,625.6
4	12,000	8,000	200	425.6	12,625.6	8,625.6
5	15,000	10,000	200	425.6	15,625.6	10,625.6
6	18,000	12,000	200	425.6	18,625.6	12,625.6

Yearly Electric and Natural Gas Cost: Taking the most recent electricity rate structures and 2014 electric usage data, the current electricity cost for an entire year was estimated as a base for comparison. A CHP electricity cost was estimated by subtracting the CHP-generated electricity (kW-h and kW) from the 2014 usage data. The total natural gas use to generate the electricity was determined by dividing the total CHP-generated electricity by the electric efficiency of the prime mover. Taking an average natural gas cost of $0.62/therm from UT's natural gas data, the total natural gas cost can be estimated. Adding the annual electricity and natural gas costs with CHP gives the total annual CHP cost.

Payback Period: By subtracting the total yearly cost with CHP from the total yearly cost without CHP, the total yearly savings can be determined. A sellback price of $.15/kW-h was assumed for selling excess electricity back to KUB. An estimated annual maintenance cost of $100,000 was subtracted from the yearly savings to get a more accurate payback period. Dividing the upfront equipment costs by the yearly savings gives the payback period. The results of this calculation can be seen in Table 4.4

Table 4.4 Yearly Savings and Payback Period. Assumes 10 adsorption chillers, each of 30 kW capacity.

Capacity (MW)	Micro-turbine		Reciprocating Engine	
	Savings ($)	Payback (yrs)	Savings ($)	Payback (yrs)
2	147,000	45	370,000	12.5
3	395,000	24,3	730,000	9.1
4	624,000	20.2	1,000,000	8.1
5	790,000	19.8	1,300,000	7.9
6	904,000	20.6	1,570,000	8.0

4.5 CONCLUSIONS AND FUTURE ANALYSIS FOR CHP

In general, due to its higher electric efficiency, the reciprocating engine appears to be the better option for prime mover. However, the micro-turbine may still be the better option when taking into account the thermal side of CHP. Due to lack of available thermal data, this analysis takes into account only the electricity savings and neglects any economic benefits of steam production and latent chiller load reduction.

Without the thermal data and analysis, large capacity systems (2 MW or greater) are necessary to show reasonable payback periods. It may be that with economic analysis of the thermal benefits, smaller capacity systems (less than 1 MW) can offer similar payback periods with smaller upfront costs. For the current analysis, the 5-MW reciprocating engine offers the shortest payback, but the 3 and 4 MW systems offer similar paybacks with not as much upfront cost.

Finally, to correctly choose the prime mover for this application, a full thermal analysis must be performed, and a load profile developed. Once this data is available, the possible benefit of steam production can be analyzed, and the prime mover can be properly chosen. If it is determined that steam production is necessary or desired, then a micro-turbine could be the best choice due to its higher exhaust temperatures.

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APPENDIX A. CHP INFORMATION

AHR Expo Companies/Contacts

Desiccant Dehumidifiers

Concepts and Designs, Inc.
Owatonna, MN
Chris Bogart

Munters Corporation
Kista, Sweden

Alfa Laval Kathabar
Tonawanda, NY
Pat Leach

SEMCO
Columbia, MO
Don Frye

Micro-turbine

Capstone Turbine Corporation
Wayne, PA
Tom McGeehan

Adsorption Chiller

InvenSor/AAA Save Energy, LLC.
Berlin, Germany
Des Plaines, IL
Carl Isohito

Equipment Data

InvenSor Adsorption Chiller

Dimensions of the machine

Length	1.560 mm
Height	1.845 mm
Width	800 mm
Weight LTC 30 e plus	1.200 kg
Weight LTC 30 e plus-FC	1.205 kg

Position of the connectors

from the ground	1.905 mm

Nominal widths

Drive (2x)	G 1 1/2"
Cooling (2x)	G 1 1/2"
Recooling (2x)	G 1 1/2"

General technical specifications				
Output range – cooling	kW / RT	10-35 / 2,8-9,9		
COP maximum		0,72		
Max. overpressure	bar	4		
Electrical connection	V~ | Hz | A	230 | 50/60 | max. 9,5		
Approx. electrical power consumption	W	25		
Approx. electrical power consumption (incl. pumps)	W	895 | EER = 33		
Nominal data		**Cooling circuit**	**Recooling circuit**	**Drive circuit**
Temperatures – possible application	°C	10–25	20–37	60–99
Volume flows	l/h	6.600	11.400	6.300
Available ext. pressure head	mbar	400	400	300

Capstone Micro-turbine

Rating: 200 kW

Electrical Efficiency: 33%

Combined Heat Power Efficiency: Up to 90%

Voltage: 400-480 VAC

Frequency: 50/60 Hz, Grid Connect 10-60 Hz, Stand Alone

Electrical Service: 3-Phase, 4-Wire

Width: 1.7 m (67 in)

Depth: 3.8 m (150 in)

Height: 2.5 m (98 in)

Weight: 2,776 kg (6,120 lb), Grid Connect 3,413 kg (7,525 lb), Dual Mode

Exhaust Temperature: 280 °C (535°F)

Exhaust Gas Flow: 1.3 kg/s (2.9 lbm/s)

Net Heat Rate LHV: 10.9 MJ/kWh (10,300 BTU/kWh)

Desiccant Dehumidifier (Concepts and Designs, Inc.)

DH Series Capacity Data

		Basic Unit Model #	DH-122	DH-130	DH-138	DH-142	DH-148	DH-160	DH-168	DH-176	DH-185	DH-196	DH-1108	DH-1114
Process Air Data														
Desiccant Rotor Diameter		[in]	22	30	38	42	48	60	68	76	85	96	108	114
Max Process Air Volume		[scfm]	1,500	3,000	5,000	5,800	7,500	12,000	15,000	18,000	24,000	29,000	38,000	47,000
Max External Static Pressure		[in W.C.]	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00
Max Outside Air Volume		[scfm]	300	600	1,000	1,160	1,500	2,400	3,000	3,600	4,800	5,800	7,600	9,400
Return Air Volume	Min	[scfm]	1,200	2,400	4,000	4,640	6,000	9,600	12,000	14,400	19,200	23,200	30,400	37,600
	Max	[scfm]	1,500	3,000	5,000	5,800	7,500	12,000	15,000	18,000	24,000	29,000	38,000	47,000
Process Air Moisture Removal Rate	Inlet Condition 1*	[lb H20/hr]	34	67	112	128	166	263	332	404	531	640	838	1,000
	Inlet Condition 2**	[lb H20/hr]	50	98	164	187	242	383	489	593	776	935	1,223	1,443
	*1 based on inlet conditions of 45° F db and 44.2 gr/lb **2 based on inlet conditions of 75° F db and 100 gr/lb													
Approximate Unit Dimensions	Width (W)	[in]	53.5	55.0	63.0	63.0	63.0	92.0	92.0	110.0	120.0	140.0	140.0	140.0
	Base Height (BH)	[in]	5.0	5.0	5.0	5.0	5.0	7.0	7.0	7.0	9.0	9.0	9.0	9.0
	Height (H)	[in]	41.0	42.0	54.0	54.0	60.0	72.0	84.0	99.0	99.0	123.0	132.0	138.0
	Length (L)	[in]	64.0	86.0	90.0	94.0	98.0	112.0	154.0	160.0	192.0	192.0	192.0	192.0
Approximate Unit Weight		[lb]	1,500	2,100	2,700	2,800	4,500	5,300	6,000	9,300	12,400	15,300	16,900	19,200

Reciprocating Engine (Tecogen)

Model	CM-60 Low Emissions	CM-60 Ultra Low Emissions	CM-75 Low Emissions	CM-75 Ultra Low Emissions
Electrical Output (kW)	60 kW		75 kW	
Thermal Output (Btu/hr)	458,000	439,000	511,000	489,000
Engine Jacket/Exhaust Manifolds **Remote Exhaust Gas Heat Exchanger**	301,000 157,000	301,000 138,000	336,000 175,000	336,000 153,000
Gas Input	782 scfh		927 scfh	
Overall Efficiency **@ LHV of 905 Btu/scf** **@ HHV of 1020 Btu/scf**	93.6% 83.1%	90.9% 80.7%	91.4% 81.1%	88.8% 78.7%
Required Gas Pressure	10-28" wc			
Design Hot Water Flow	22 gpm (24 gpm max)			
Maximum Leaving Water Temperature	230° F			
Maximum Entering Water Temperature	180° F			
Electrical Service	208V / 230V / 460V, 3 PH, 3-wire			
Acoustic Level	70 dBa @ 20'			
Dimensions	7' 2"L x 3' 8"W x 3' 10"H			
Weight	3000 lbs			