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CO2 HEAT PUMPS FOR COMMERCIAL BUILDING APPLICATIONS WITH SIMULTANEOUS HEATING AND COOLING DEMAND

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CO_2 HEAT PUMPS FOR COMMERCIAL BUILDING APPLICATIONS WITH SIMULTANEOUS HEATING AND COOLING DEMAND

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of

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by

Supriya Dharkar

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For my family

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NOMENCLATURE

SYMBOL	DESCRIPTION
Q	Load (kW)
W	Work (kW)
m	mass flow rate (kg/sec)
T	Temperature (°C)
h	Specific enthalpy (kJ/kg)
η	Efficiency (%)
M	Mass of medium in the tank (kg)
C_p	Specific Heat Capacity (kJ/kg-K)
U	Overall heat loss coefficient (kW/m^2-K)
A	Area (m^2)
COP	Coefficient of Performance (-)

TEWI	Total Equivalent Warming Impact (kg CO_2)
GWP	Global Warming Potential (-)
L	Leakage rate (kg/year)
n	Life of the system (years)
m	Refrigerant charge (kg)
α	Recycling factor of the refrigerant (%)
β	CO_2 emission factor (kg CO_2/kWh)
E	Annual energy consumption (kWh/year)

SUBSCRIPTS

coolingload	Cooling load of the data center
supply, cold	Cold supply water
return,cold	Cold return water
supply, hot	Hot supply water
return, hot	Hot return water

evap	Evaporator
gascooler	Gas cooler
comp	Compressor
exp	Work recovery expansion device
w	Water
tank	Tank parameters
amb	Ambient
CO2heating	Heating provided by CO_2 heat pump system
r	refrigerant
gen	Generator
turb	Turbine
is	Isentropic

ACRONYMS

ODP	Ozone Depletion Potential
MATH	Mathematics Department Building

CDU	Cooling Distribution Units
ICT	Information and Communication Technologies
ITE	Information Technology Equipment
TES	Thermal Energy Storage
PCM	Phase Change Material

ABSTRACT

Dharkar, Supriya P. M.S.M.E, Purdue University, August, 2015. CO_2 Heat Pumps for Commercial Building Applications with Simultaneous Heating and Cooling Demand. Major Professor: Eckhard A. Groll, School of Mechanical Engineering

Many commercial buildings, including data centers, hotels and hospitals, have a simultaneous heating and cooling demand depending on the season, occupation and auxiliary equipment. A data center on the Purdue University, West Lafayette campus is used as a case study. The electrical equipment in data centers produce heat, which must be removed to prevent the equipment temperature from rising to a certain level. With proper integration, this heat has the potential to be used as a cost-effective energy source for heating the building in which the data center resides or the near-by buildings. The proposed heat pump system utilizes carbon dioxide with global warming potential of 1, as the refrigerant. System simulations are carried out to determine the feasibility of the system for a 12-month period. In addition, energy, environmental and economic analyses are carried out to show the benefits of this alternative technology when compared to the conventional system currently installed in the facility. Primary energy savings of ~28% to ~61%, a payback period of 3 to 4.5 years and a decrease in the environmental impact value by ~36% makes this system an attractive option. The results are then extended to other commercial buildings.

CHAPTER 1. INTRODUCTION

1.1. Objectives

The environmental concerns related to the currently used refrigerants have led to research on climate friendly refrigerants. The objective of this research is to analyze the application of heat pumps, which use natural refrigerants such as CO_2 for commercial buildings with a simultaneous heating and cooling load. The heat rejected by the electrical equipment in commercial buildings, such as data centers, can be used as a source to provide district heating to the near-by buildings. The cooling load of a data center on the Purdue University campus and the heating load of the building in which the data center resides is collected. Based on the simultaneous heating and cooling load, a model is created to determine the total primary energy savings compared to the current conventional system. Various other commercial buildings such as hotels and hospitals are evaluated using the model to determine the scope of primary energy savings, leading to other environmental and economic benefits.

1.2. Current Refrigerants

The first practical refrigeration system can be dated back to 1834, which used ethyl ether as a refrigerant. This invention was followed by several compression refrigeration systems, which used natural refrigerants such as ammonia, water and CO_2. However, use of these refrigerants posed technical and safety concerns. Hence, the focus shifted to discovering inexpensive, non-flammable and non-toxic refrigerants which had lesser engineering challenges. In April 1930, the development of the fluorocarbon refrigerants was announced. Commercial chlorofluorocarbon (CFC) production began with R-12 and continued in 1930's with R-11, R-114 and R-113. Later, hydrochlorofluorocarbon (HCFC's) such as R-22 started being used in several applications. In 1974, Molina and Rowland's (United Nations Ozone Secretariat, 2007) ozone layer depletion theory aroused great interest which led to CFC's ban in aerosols in USA and several other countries. This was followed by the Vienna Convention for Protection of the Ozone Layer. The Montreal Protocol in 1987, was instrumental in phasing out of CFC's and discovery of low Ozone Depletion Potential (ODP) refrigerants namely the hydrofluorocarbons (HFC). However, these refrigerants have a high Global Warming Potential (GWP) of up to 4500, an effect that has been attributed to the tremendous increase in greenhouse gases such as CO_2, CH_4 and F-gases emissions. U.S. Environmental Protection Agency, 2015, states that fluorinated gas emissions in the United States have increased by about 83% between 1990 and 2012. The increase in emissions can be traced to a 310% increase in hydrofluorocarbons (HFCs) emissions since 1990. HFC emissions are projected to grow by nearly 140% between 2005 and 2020 as demands for refrigeration continue to grow and as more ozone-depleting substances are replaced. Refrigerants contribute directly as well as indirectly to the high GWP. This growth poses a serious threat and many countries are regulating or phasing out HFC's. Refrigerants contribute to global warming in two ways. Direct contribution is caused by the refrigerant leak and indirect contribution is due to the CO_2 emitted during the generation of the power required by the systems. In lieu of this revelation about refrigerants contribution to the global warming, researchers are looking towards various other alternatives like hydrofluoroolefin (HFO). The recent advancement of technologies has also propelled the research back to natural refrigerants, such as CO_2 and NH_3. The refrigerant industry has almost come full circle as represented in Figure 1-1. However, no single refrigerant fits all the application and hence the refrigerants should be carefully chosen based on application.

This research examines the application of heat pumps utilizing a natural refrigerant, carbon dioxide (R-744) in commercial buildings applications, which have simultaneous heating and cooling loads. Although CO_2 was first used as a refrigerant about 150 years ago, it has been intensively researched recently. There is still significant scope for further product and system development. With recent urgency in tackling the issue of global warming, the use of natural refrigerants, such as CO_2, have become extremely important.

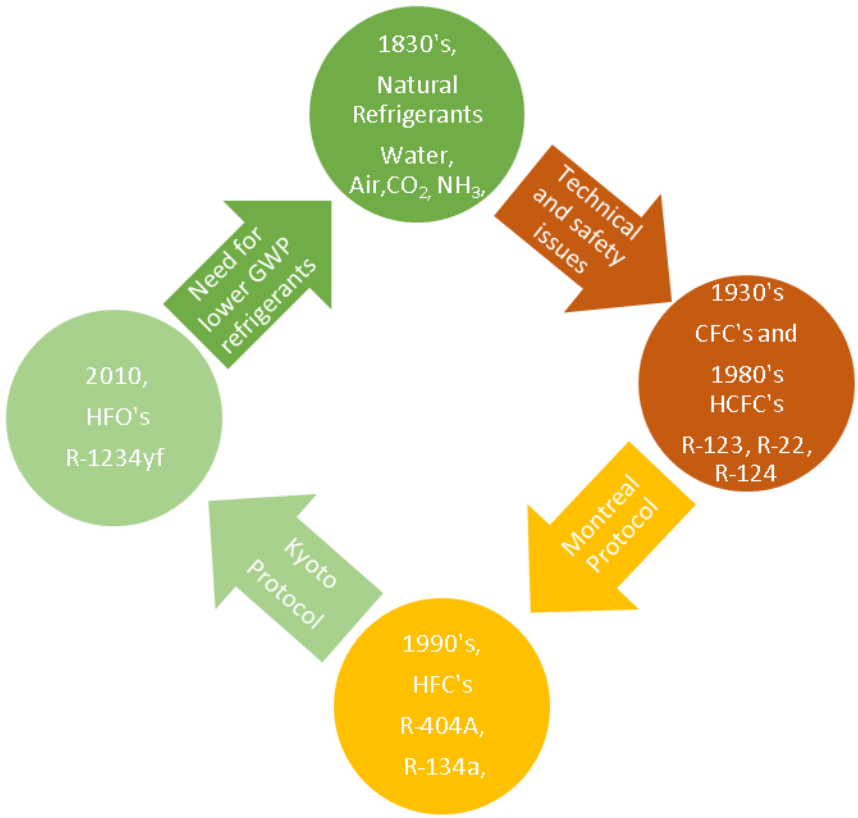

Figure 1-1. Transition of refrigerants along the years.

1.3. Natural Refrigerants – CO_2 as a Refrigerant

Carbon dioxide has unique properties which if integrated appropriately can be used to advantage. Carbon dioxide is colorless, odorless, and heavier than air. With a Global Warming Potential of 1, CO_2 is the reference value for comparing a refrigerant's direct impact on global warming. It is non-flammable and non-toxic making it a refrigerant with A1 safety classification (the same as most fluorocarbon refrigerants). CO_2 as a refrigerant is sourced from a number of production methods as a by-product. Though it is nontoxic, oxygen will start getting displaced if enough carbon dioxide builds up in an enclosed space, leading to asphyxiation to anyone present over a certain period within the space. In terms of environmental impact, CO_2 does not lead to any harmful decay products.

CO_2 as a refrigerant has several unique properties. The most significant being higher operating pressure of CO_2 systems. Adding to this are the extremely low viscosity and the low critical temperature of Carbon dioxide. It is now possible to source compressors for use in refrigeration system up to 16MPa (Britzer, 2015). The high pressure of CO_2 delivers many of the inherent benefits of the system. The compressors are relatively small and so is the pipework. The gas is dense and hence results in good transport and heat transfer properties. If a system is designed to capitalize on the advantage of high pressure, CO_2 systems can offer significantly more efficient systems.

The viscosity of liquid CO_2 is one-tenth of water at 5°C. Unlike brine and glycol solutions, the viscosity of CO_2 at lower temperatures of -50°C remains uniform. This makes CO_2 an attractive heat transfer fluid at low temperature.

The critical temperature of CO_2 is 31°C which is considerably low compared to the other refrigerants. This offers unique opportunities to design a system in a suitable way so as to generate significant savings when compared to conventional technology. Heat recovery at these higher temperatures is very advantageous. Various water heaters, domestic, commercial and light industrial heat pump system installation in the market show that if this property is tapped in appropriately, the system can have significant advantages. The benefits of this property are more noteworthy when both, the cooling and heating systems are integrated. An example of application in commercial buildings with simultaneous heating and cooling demand is discussed in this thesis.

One of the first transcritical CO_2 systems was a prototype automotive air conditioning system built and tested by Lorentzen and Pettersen (1992). This was succeeded by first prototype experiment on bus air conditioning (Sonnekalb & Kohler, 1996) and studies on feasibility of train air conditioning application (Quack & Kraus, 1994). The CO_2 transcritical cycle system proves to be reliable system for air conditioning system of vehicles because of its low weight, safety, reliability, tightness and overall system performance. However, research shows that the system performs better than R-134a system only at lower ambient temperature. Apart from automobiles, CO_2 systems have also been used as commercial refrigeration systems. Transcritical CO_2 refrigeration cycles have been used in vending machines. In December 2009, the Coca Cola Company announced converting all their vending machines to CO_2 systems. Coca Cola Company have installed one million coolers till 2014 and claim to have increased energy efficiency of the cooling equipment by 40% (The Coca Cola Company, 2015). CO_2 systems have also been utilized in cascade type industrial refrigeration system and as a secondary fluid in food display applications in super markets. An interesting application of CO_2 heat pump system discussed by Li & Groll, 2004 is carbon dioxide based Environmental Control Units (ECU) used by the U.S. Army to provide air conditioning for field installation and personnel. The research shows a higher cooling COP and a higher cooling capacity for carbon dioxide based ECU when compared to the traditional R-22 based ECU. Apart from the various applications discussed, tap water heating is another promising application of the transcritical CO_2 system. The temperature glide of CO_2 during supercritical heat rejection results in a very good temperature adaptation when heating tap water, which undergoes a large temperature glide itself. This, together with the efficient compression and good heat transfer characteristics of CO_2, makes it possible to design very efficient heat pumping systems (Groll &

Kim, 2007). Stene (2005) proposed CO_2 heat pumps for residential applications. The author tested a 6.5 kW prototype CO_2 heat pump under three different modes: space heating only, Domestic Hot Water (DHW) heating only, and simultaneous heating and DHW heating and documented the results. In laboratory settings, Wang et al. (2014) concluded that the best COP of application of transcritical CO_2 heat pump system with simultaneous heating and cooling using a thermal battery is found to be approximately 7.0 at a specific operating condition.

Table 1-1 lists the existing CO_2 systems in different countries over the globe (Sheeco, 2013). Some of these systems are prototypes in specific locations where as others are commercialized systems in the corresponding country. This research is concentrated on the application of CO_2 heat pumps in commercial buildings in particular such as a data center

1.4. Scope of Application of CO_2 Heat Pumps in Data Centers

With the growing use of digital content, big data and e-commerce, data centers have become one of the fastest growing industries using electricity and one of the key drivers in the construction of new power plants. Based on moderate estimates, Mills (2013) predicts the world's Information and Communication Technologies (ICT) ecosystem uses about 1500 TWh of electricity annually. The report also forecasts that ICT is now approaching 10% of worlds electricity generation. Greenpeace International's (2012) findings show the large IT companies reliance on fossil fuels to power the clouds. The dependance on coal is represented in Figure 1-2. This reliance on coal highlights the urgent need to move towards more energy efficient equipments and utilization of renewable sources of energy.

Table 1-1. Existing CO_2 systems in different countries.

Country	Prototype or Commercial Applications
USA	• Vending Machines: Coca- Cola Company • Beverage Coolers: Red Bull • Wine producer Somerston Wine Co, Napa Valley, California • Supermarket installations: Sam's Club, Whole Foods, Fresh and easy, SUPERVALU, Wegman's Woodmore store
Canada	• CO_2 ice rink, in the Marcel Dutil Arena of the town of Les Coteaux • Quebec Laboratory refrigeration plant in Québec • Dairy/Cheese plant- Fromagerie • Polyethnique / Fromagerie Fritz Kaiser dairy/cheese plant • Super market application- Sobey's, Maple leaf Gardens store, Loblaws
Europe	• 1,300 supermarkets confirmed for CO2-only transcritical systems • Norwegian ice cream manufacturer Diplom-Is. • Variety of cascade and secondary system solutions, together with ammonia and hydrocarbons, or synthetic refrigerants • Mobile air conditioning
UK and Ireland	• Cúil Dídin Nursing Care Facility and Beechdale manor care home • O'Donovan's Hotel in Ireland
Japan	• CO_2 water heaters – residential and commercial • Government provides subsidy for residential applications • Use in hotel in Tottori
China	• Potential production capacity of CO_2 heat pumps has reached 100,000 units.

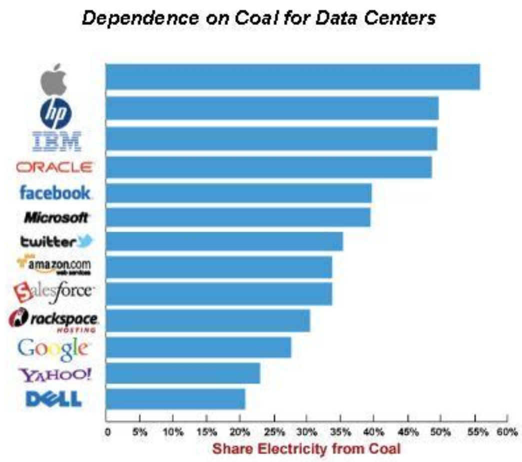

Figure 1-2. Electricity share from Coal (Mills, 2013).

The U.S. Environmental Protection Agency (2007) estimates that the ITC sector consumed about 61 billion kilowatt-hours (kWh) in 2006 (1.5 percent of total U.S. electricity consumption) for a total electricity cost of about $4.5 billion. This number is expected to grow tremendously according to the growth trends documented in the above report.

Figure 1-3 shows the current and future data center market size. The data from 2007-2010 is the predicted data documented in the EPA's report. Based on the extrapolation, predictions have been made for the growth in market by historic, current and improved efficiency.

On examining the data center's energy consumption, it is realized that about 50% of consumed total electrical power is spent on cooling equipment in data centers. This is represented in the form of a pie chart in Figure 1-4. Cooling is very critical to data centers. Electronic equipment performs better in cool environment and though data centers can now work at higher temperatures, cooling is still required and consumes significant amount of resources. The heat rejected by the electronic equipments is typically wasted. This heat has the potential to be utilized in other applications such as domestic water heating or district heating if integrated properly. This will be discussed in the next chapters.

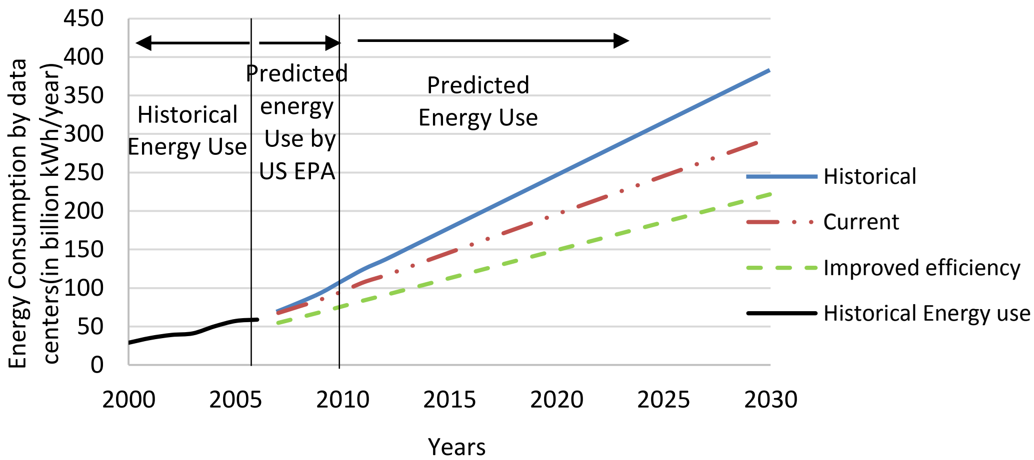

Figure 1-3. Increase in data center market size.

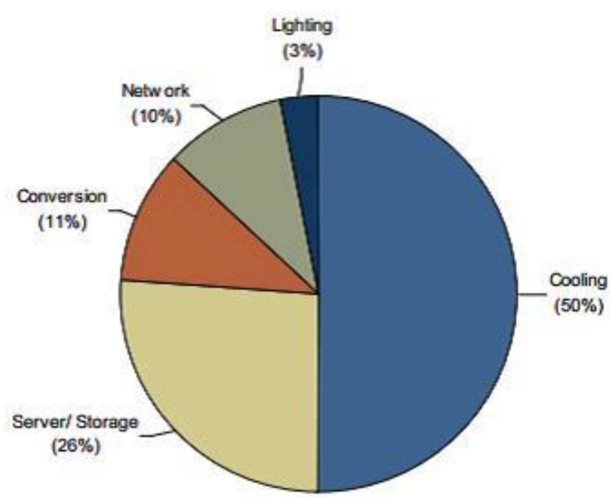

Figure 1-4. Typical breakdown of energy consumption in a data center (Info-tech Research, 2007).

1.5. Energy Consumed by Heating Systems in Buildings

The waste heat rejected by the data centers can be utilized by buildings. U.S Energy Information Administration (EIA), 2015 states that 41% of the total U.S energy consumption was consumed by residential and commercial buildings in 2014. The building energy sector energy consumption is expected to grow faster than that of other sectors such as industry and transportation. D&R International, Ltd. under contract to Pacific Northwest National Laboratories, 2011 published a 2010 Buildings Energy Data book which predicts the energy consumption of commercial buildings to be 18.6% in 2015. It also enumerates the 2010 commercial energy end-use splits by energy consumed on site. The maximum energy consumed is by space heating followed by lighting. If the waste heat from data centers or other such commercial buildings is integrated appropriately, the energy consumed by space heating can be decreased significantly.

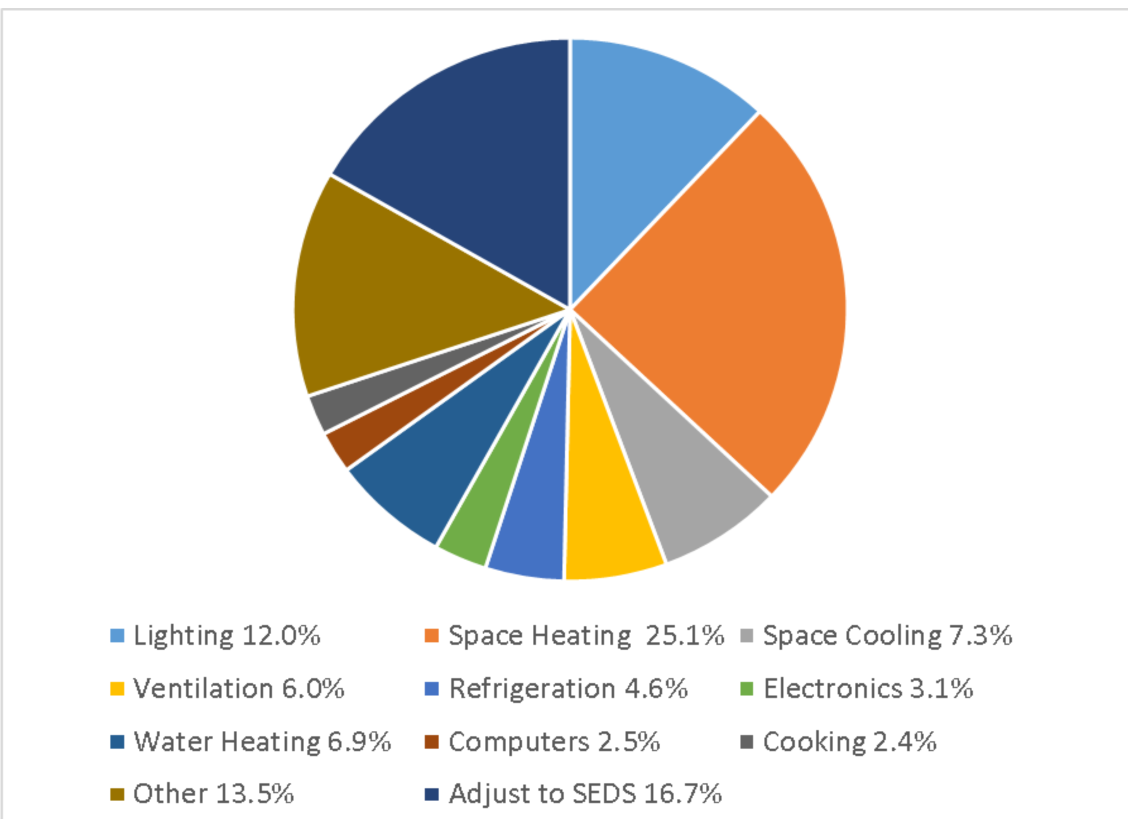

Figure 1-5. 2010 Commercial Energy End-Use Splits.

The CO_2 heat pump technology can be utilized in any commercial buildings which have simultaneous heating and cooling requirements. Figure 1-6 shows the amount of heat utilized by heating and cooling equipment in different types of commercial buildings. Buildings such as food services, health care, inpatient and strip malls have considerable cooling needs. CO_2 heat pumps can be used in buildings such as these to meet the cooling needs and the waste heat can be utilized to fulfill at least part of the heating demand. Some results regarding the application in such buildings is presented in Chapter 7.

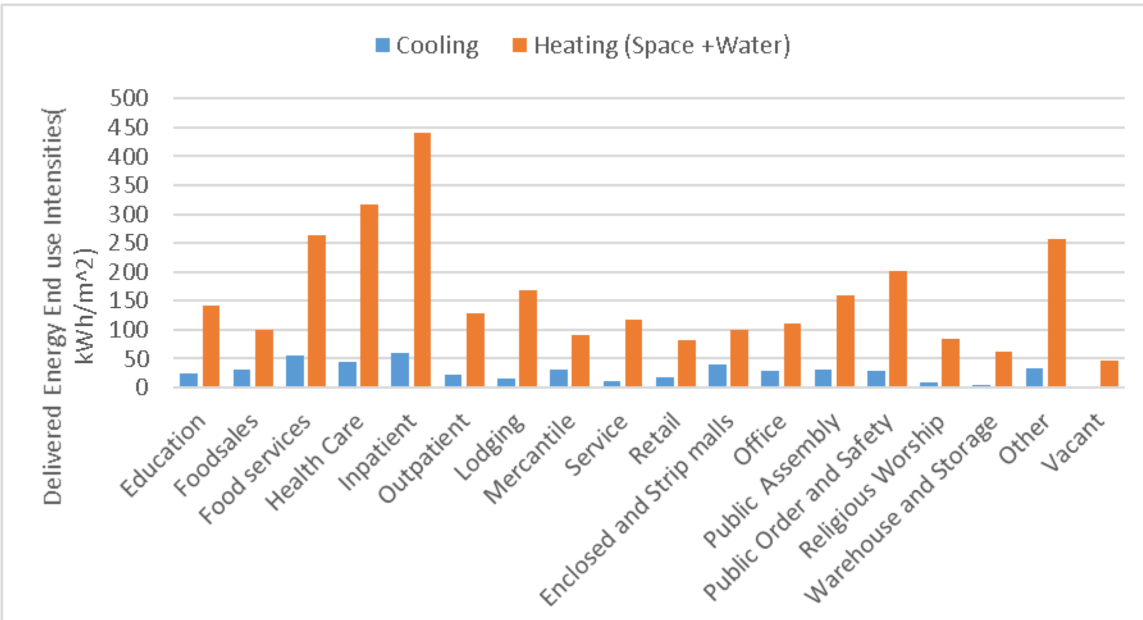

Figure 1-6. 2003 Commercial buildings delivered end-use intensities by building activity (D&R International Ltd., 2011).

CHAPTER 2. CASE STUDY: DATA CENTER

2.1. Introduction

As discussed in Subchapter 1.4, data center industry is one of the fastest growing industry and a reason for constructing more power plants in the United States. This makes data center a very important case study. A data center on Purdue University, West Lafayette campus is taken as a case study to carry out simulations. Based on the previous monitored growth, Wagstaff et al. (2009) projected the total data center space on Purdue University, West Lafayette campus to double since 2009. The demand for additional research space has been propelled by growth of research funding, as well as discipline- based shifts to utilizing more cyber infrastructure. Purdue houses Conte, the nation's fastest university-owned supercomputer, developed in collaboration with HP, Intel, and Mellanox.

Figure 2-1. Conte, Purdue’s most powerful research computer.

As Purdue continues to make significant strides in research funding, the power and cooling infrastructures will be strained continuously. Several projects are being carried out to make the data centers on campus more energy efficient. Currently, there are 40 data centers on campus. Amongst them, the data center located in the basement of the Mathematics department (MATH) is the most energy intensive data center. This critical data center is considered as a case study, and simulations are carried out with proper integration to decrease the total energy utilization of the entire building.

2.2. Mathematics Department – Current System

The data center in the Mathematics Department is located in the basement of the MATH building. With super computers like Conte, located in this data center, considerable amount of energy is required for cooling the data center. The instantaneous energy consumption of the data center is 1.1-1.5 MW. The energy consumed by the equipment used for cooling is approximately 700 kW, which is 50% of the total energy consumed by the data center. To meet this high cooling load, liquid cooling in the form of close coupled passive Rear Door Heat Exchanger is utilized. Figure 2-2 shows the liquid cooling system in place. The chilled water supplied to the data center is distributed to various rear door heat exchangers by the Cooling Distribution Unit (CDU).

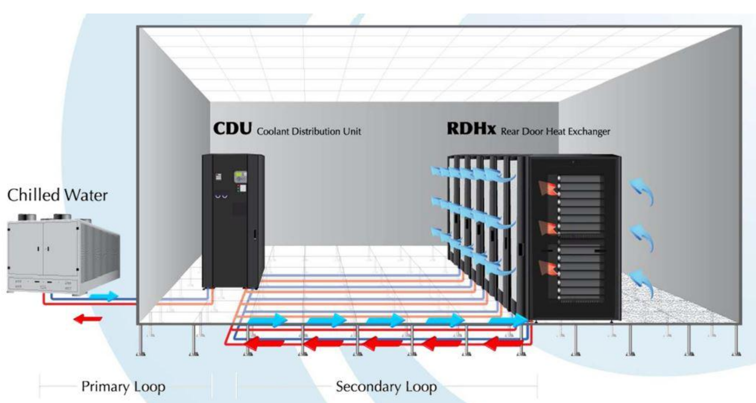

Figure 2-2. Current system overview (Gagnon, 2011).

There are 14 external Cooling Distribution Units (CDU) as the one shown in Figure 2-4(a). These CDU units are connected to 101 rear door cooling units (Figure 2-3 b) through underfloor piping as shown in Figure 2-4(b). The maximum chilled water capacity is 65 kg/s but the actual hourly usage is approximately 30 kg/s.

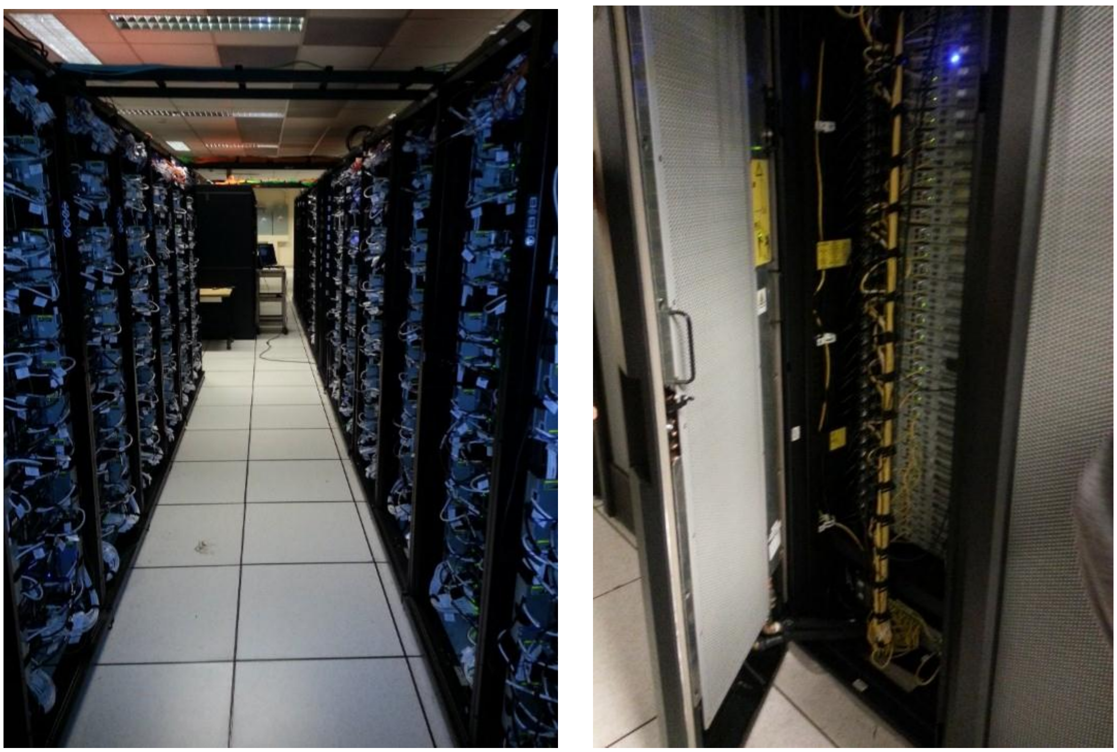

Figure 2-3. (a) Data server racks (b) Rear door cooling units.

The Purdue University West Lafayette campus benefits from a combined heat and power system that utilizes steam not only to provide heat to facilities but also to generate electricity and the chilled water that is used to cool facilities such as data centers. Figure 2-5 shows the energy flow diagram of the current system. Chilled water at 7°C is provided to the MATH data center by 10 electric driven chillers and 3 steam driven chillers. A turbine and a generator are used to produce electricity from the high temperature and pressure steam at 435°C and 4.48 MPa, which is produced by 1 coal fired and 3 natural gas fired boilers. The lower grade steam at 288°C and 0.86 MPa is used as a source for the steam driven chillers. The MATH data center can be classified as a W1/W2 class data center according to the ASHRAE cooling guidelines (Steinbrecher & Schmidt, 2011). The MATH building is one of the biggest buildings on the Purdue University, West Lafayette campus.

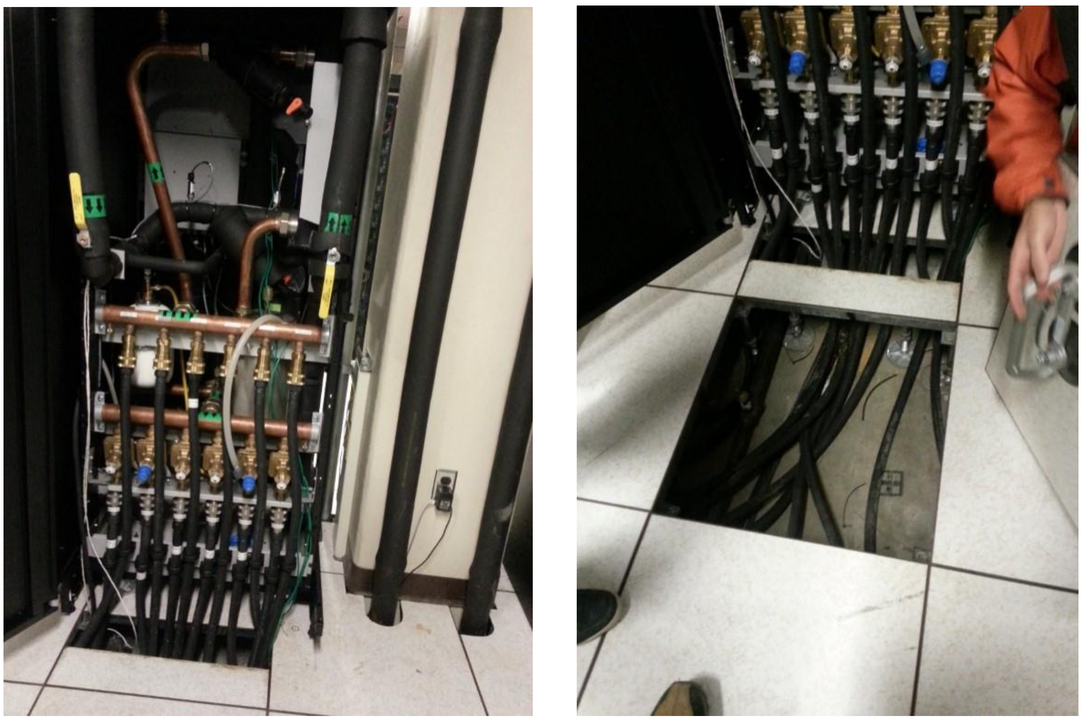

Figure 2-4. (a) Cooling Distribution Units (CDU) (b)Underfloor piping distributing chilled water from CDU to the rear door cooling units.

To meet the high heating load, low grade steam at ~180°C is supplied to the building in which this data center resides. The water condensate returns to the utility plant at ~52°C.

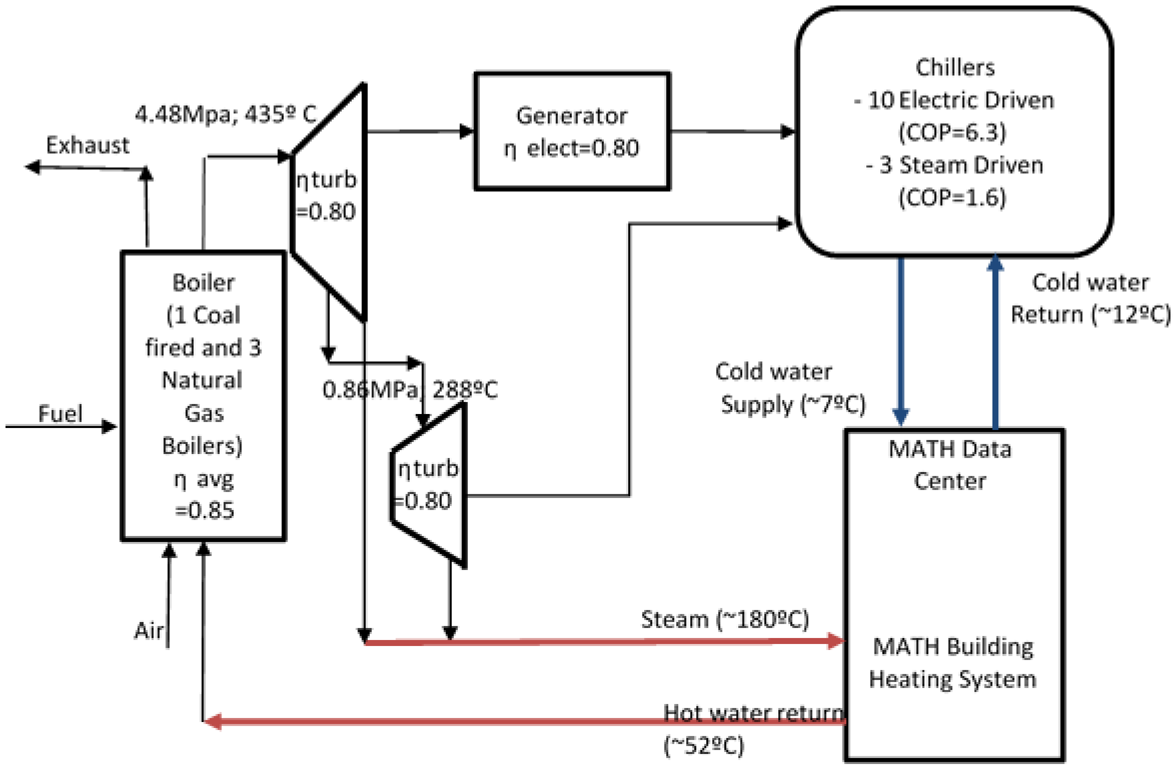

Figure 2-5. Energy flow diagram of the current conventional system.

A bar graph in Figure 2-6 shows the heating load of the MATH (Mathematics) building in red bar and the cooling load of the MATH data center in blue bar. The collected data is over a 12-month period ranging from August'13 - July'14. While the cooling load of the data center is relatively constant over the entire year, the heating load varies considerably depending on the season.

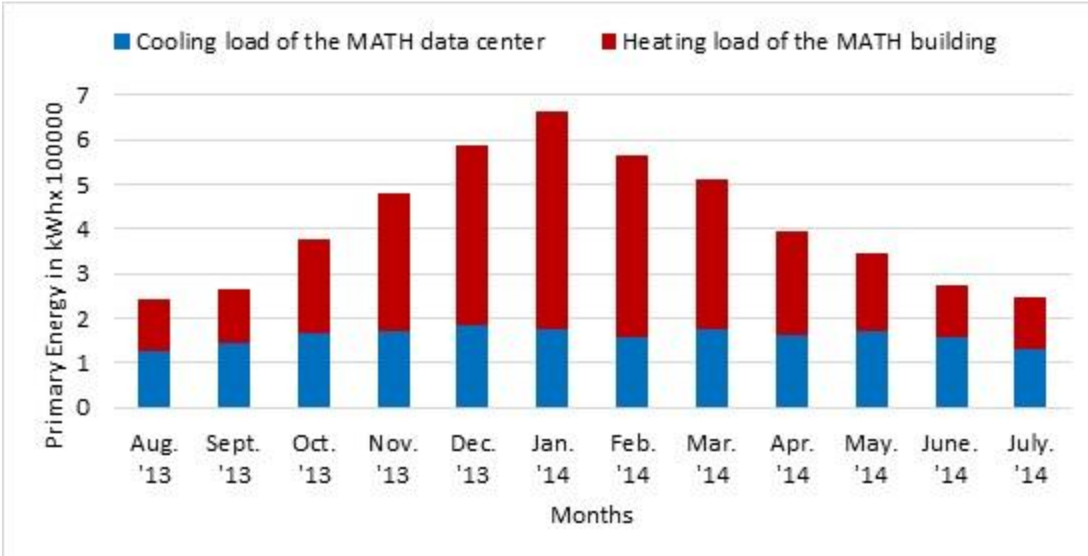

Figure 2-6. Heating load of the MATH building and cooling load of the MATH data center.

2.3. Mathematic Department - Proposed System

The proposed CO_2 heat pump system would replace the 13 chillers shown in Figure 2-5. The chilled water required to cool the data center is instead provided by a CO_2 heat pump system. The heat rejected by the electrical equipment will be pumped to a higher temperature of approximately 62° C which can be utilized to heat the rest of the building. The energy flow diagram of the proposed system is shown in Figure 2-7. The CO_2 heat pump system is designed to meet the cooling demand of the data center. The waste heat is utilized to heat the MATH building in which the data center resides. Figure 2-8 is the schematic diagram of the proposed the system. The cold water from the evaporator is used to cool the data center. The system has two compressors with intercooler. It takes advantage of single stage compression in cold winter months and two-stage compression in warm summer months. A gas cooler is used to cool the refrigerant in heat exchange with water. The heated water is then used to provide heating to the building. After passing through the gas cooler, the refrigerant passes through the air cooler.

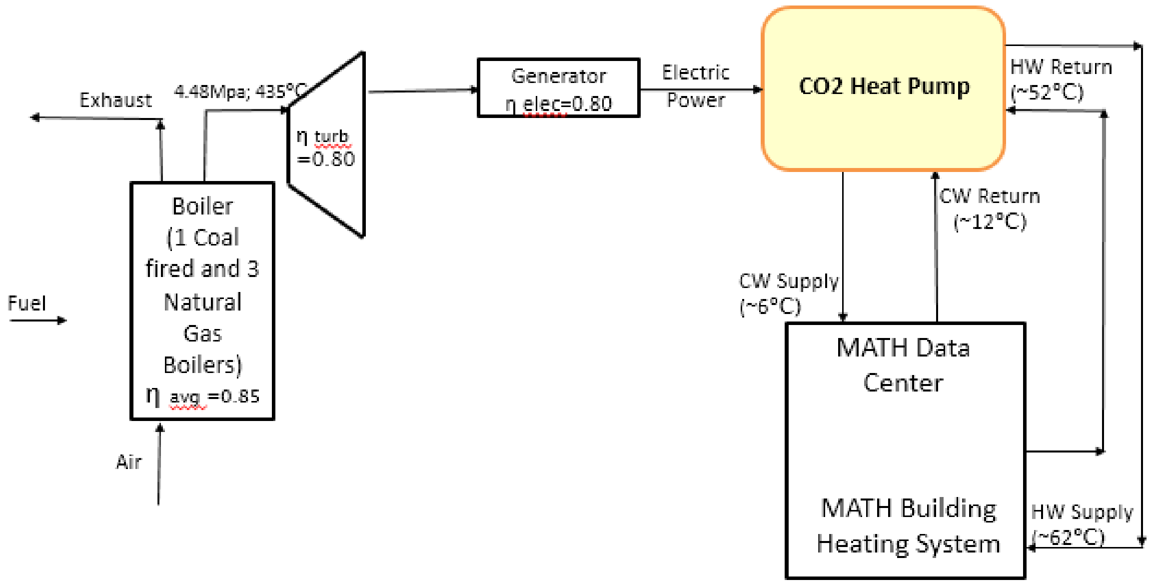

Figure 2-7. Energy flow diagram of the proposed system.

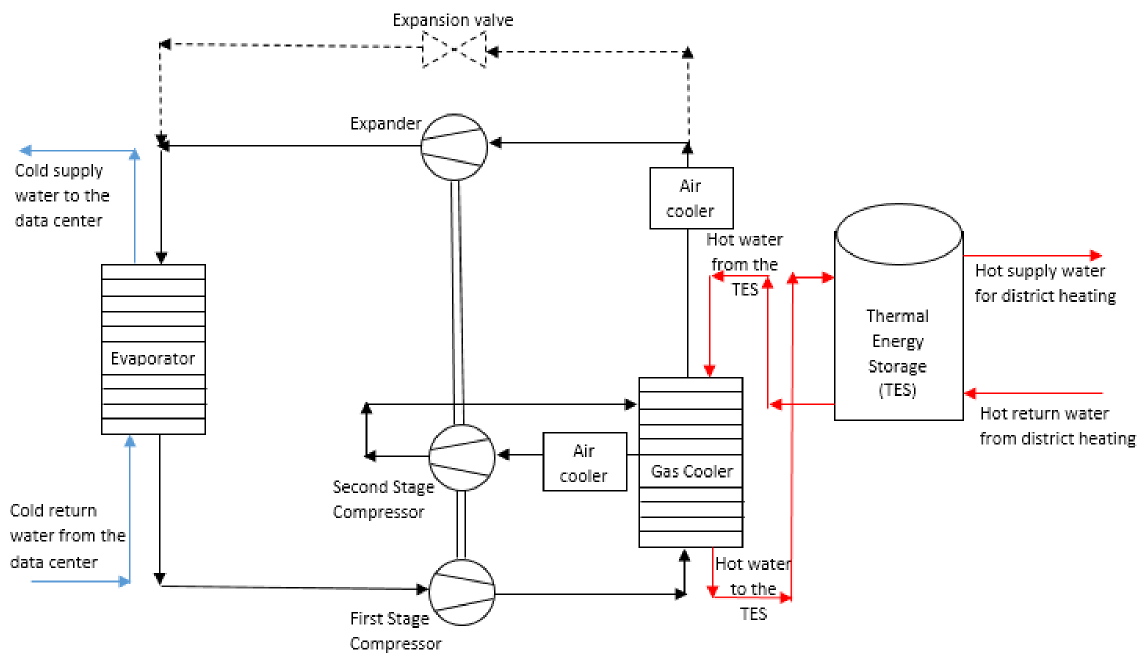

Figure 2-8. Schematic diagram of the system.

The air cooler increases the efficiency of the system by making use of the cold ambient air temperatures in the winter. The system is simulated with two expansion devices: An expansion valve (represented by dotted line in Figure 2-8) and an expansion work output device, which recovers work to assist in driving compressor. The integrated, high performance expander/compressor unit (ECU) helps to recover normally lost expansion work, significantly boosting the performance of the transcritical CO_2 cycle. A hot thermal energy storage (TES) is also integrated with the heat pump system. Daily storage is found to be most suitable for the system. A couple of storage materials are discussed in the following chapters.

2.4. Modeling

Figure 2-9 is a flow diagram which describes the modeling approach undertaken. Engineering Equation Solver (EES) and MATLAB were utilized to carry out the simulations. Hourly pressure, temperature, mass flow rate and ambient temperature data are collected from the sources at the Wade Utility Plant on the campus and input into the simulation model. The data is attached in the extra files. These parameters are used to calculate the cooling load using Equation (2.1).

$$Q_{coolingload} = m_{coolwater} * C_{pw} * (\mathrm{T}_{\mathrm{supply,\ cold}} - \mathrm{T}_{\mathrm{return,\ cold}}) \tag{2.1}$$

$$Q_{heatingload} = m_{hot} * (\mathrm{h}_{\mathrm{supply,\ hot}} - h_{\mathrm{return,\ hot}}) \tag{2.2}$$

Using the pressure and temperature, the enthalpy of the steam can be calculated which is utilized in Equation (2.2) to determine the heating load.

The model uses some constant parameters, which are listed in the Table 2-1. A discharge pressure is selected. Using energy balance equations, the specific gas cooler ($Q_{gascooler}$), evaporator (Q_{evap}), work done by the compressor (W_{comp}) and work extracted from the expander (W_{exp}) is calculated. The COP of the heat pump system is given by Equation (2.3), Equation (2.4) and Equation (2.5).

$$COP_{cooling} = \frac{Q_{evap}}{W_{comp} - W_{\exp}} \tag{2.3}$$

$$COP_{heating} = \frac{Q_{gascooler}}{W_{comp} - W_{\exp}} \tag{2.4}$$

$$COP_{total} = COP_{cooling} + COP_{heating} \tag{2.5}$$

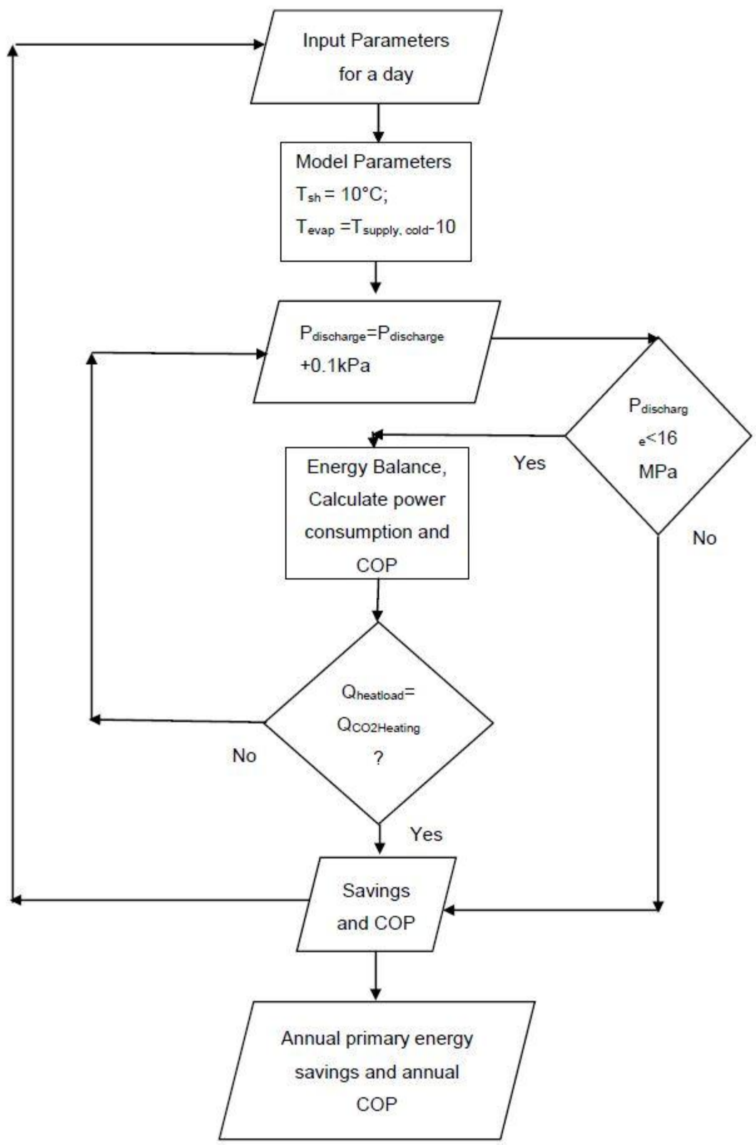

Figure 2-9. Model Flow-diagram.

Table 2-1. Model parameters.

*Parameters to calculate the energy savings of the proposed system compared
Source: Wade Utility Plant, Brenton Dunham

Model Parameters	**Value**
Superheat	10°C
Gas Cooler pinch temperature	5°C
Evaporator pinch temperature	10°C
Air cooler pinch temperature	10°C
Isentropic Efficiency(η_{is})	70%
$T_{supply,dh}$	62°C
$T_{return,dh}$	52°C
$T_{sinitial}$	62°C
Conventional System Parameters*	
Generator efficiency(η_{gen})	80%
Turbine efficiency(η_{turb})	80%
$COP_{electric}$	6.38
COP_{steam}	1.60

It should be noted that in the summer months, the $Q_{gascooler}$ includes the gas cooler heat from both the gas cooler and intercooler. Similarly, during the summer months, power is consumed by two compressor and both of them should be taken into consideration while calculating the COP.

If the heat load of the building cannot be met by the heating generated by the CO_2 heat pump system, calculations are done again with a higher discharge pressure. An optimum discharge pressure is such that the heating load is completely met. The cooling COP is high at lower high-side pressures. However, the recovered heat in the gas cooler is not sufficient for heating purpose of the MATH building at such low high-side pressures. Instead, if the cycle is made to operate at a higher pressure, more heat can be obtained at the cost of cooling COP. The loss in cooling COP is made up by the gain in heating COP and thus, results in approximately the same total COP. This is represented in the Figure

2-10. The heating percentage is the amount of heating attainment level, a ratio of heating produced by the CO_2 heat pump system in a day with respect to the actual heating load in a day. The cooling COP decreases by a unit of one as the heating percentage increases. However, the gain in heating COP makes the overall system attractive.

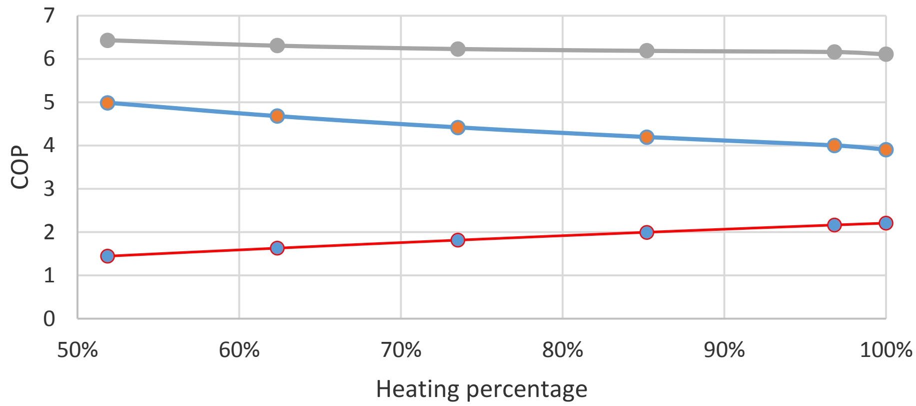

Figure 2-10. COP vs heating percentages.

16 MPa is the highest discharge pressure that a commercially available CO_2 compressor can withstand (Britzer, 2015). Hence a condition is included in the model that stops the increase in discharge pressure if such a state arises. There are some days (Winter days) in the year when there is excessive heating requirement which can lead to such a condition. Supplement heating is required to fulfil the heating load of such days. This supplement heating can be provided by power plants or by designing appropriate storage to meet the extra heating load. Figure 2-11 and Figure 2-12 show the P-h diagram for the CO_2 cycle on a typical cold day in November and a hot day in July respectively.

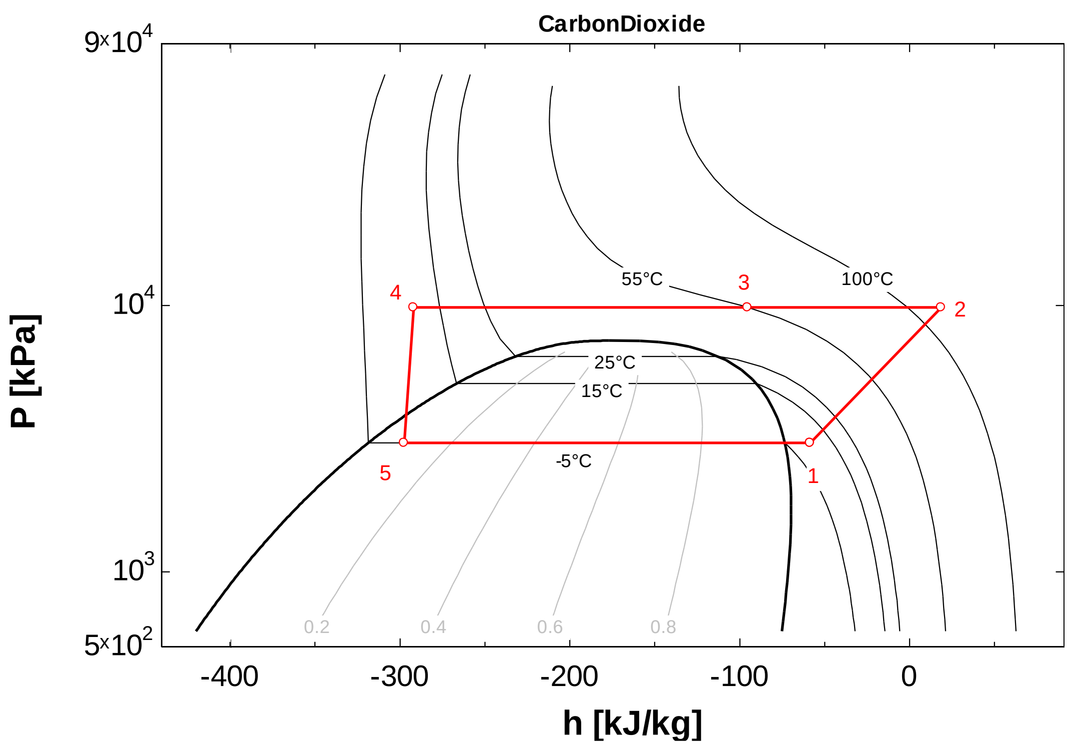

Figure 2-11. P-h diagram of the CO_2 cycle on a typical cold day (November).

1-2: Compression

2-3: Gas cooler, which can provides hot water ≧ 62℃ for district heating of the building

3-4: Air Cooler

4-5: Work recovery expansion device

5-1: Evaporator, which provides cooling water of ~6℃ to the data center.

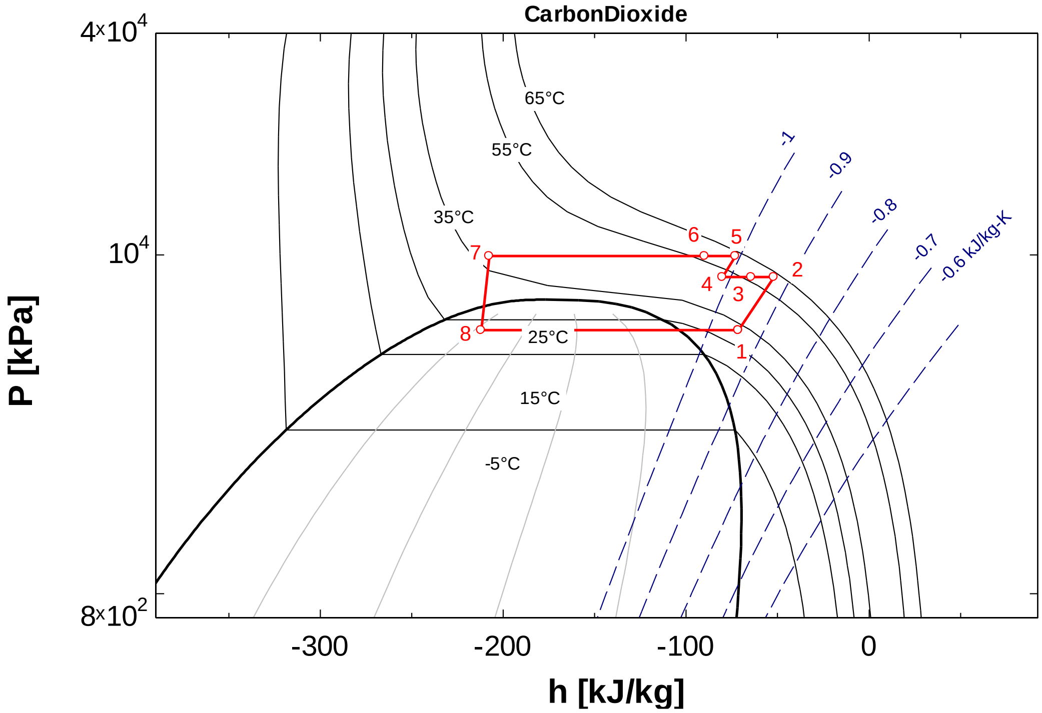

Figure 2-12. P-h diagram of the CO_2 cycle on a typical hot day (July).

1-2: Compression
2-3: Intercooler
3-4: 2nd Stage Compression
4-5: Gas cooler, which can provides hot water ≧ 62℃ for district heating
5-6: Air Cooler
6-7: Work recovery expansion device
7-1: Evaporator, which provides cooling water of ~6℃ to the data center

Total primary energy savings is calculated using Equation (2.6)

$$Savings = Q_{primheating} + Q_{primarycooling} - Q_{primco2\,power} \tag{2.6}$$

The savings for each day are added to calculate the annual primary energy savings.

For the ease of modeling, a fully-mixed model is utilized to model the Thermal Energy Storage. The entire liquid (water) in the storage tank is assumed to have a uniform temperature which changes with time as a result of net energy addition or withdrawal during the charge or discharge process or due to the interaction with the surrounding. The model equation (Duffie & Beckman, 2013) representing this thermal balance is written as Equation (2.7).

$$M * C_{pw} * \frac{dT_{s,\tan k}}{dt} = Q_{CO2heating} - Q_{heatingload} - U_{\tan k} * A_{\tan k} * (T_{S,\tan k} - T_{amb}) \quad (2.7)$$

It is assumed that the tank is heated to 62°C initially. The value of tank mass and area is determined by optimization of the tank size. The value of the overall heat loss coefficient (U_{tank}) depends on the thickness of the insulation which is discussed in Chapter 3.

CHAPTER 3. RESULTS AND DISCUSSIONS

The annual data was input into the model discussed in Chapter 2 to derive various results discussed below. Figure 3-1 shows that annual potential energy savings of 15% can be achieved when a simple single stage compression transcritical CO_2 cycle with a throttling valve is simulated.

	Aug. '13	Sept. '13	Oct. '13	Nov. '13	Dec. '13	Jan. '14	Feb. '14	Mar. '14	Apr. '14	May. '14	June. '14	July. '14	Annual
Savings	-30%	-34%	20%	29%	33%	42%	38%	33%	23%	-6%	-49%	-41%	15%

Figure 3-1. Potential primary energy savings.

Various system alterations which were simulated to increase the overall efficiency of the system are discussed in the following subchapters.

3.1. Two-Stage Compression in Summer Months

The system performance greatly depends on the ambient temperature. From Figure 3-1, it is evident that the system does not perform well in the summer months. The negative energy saving symbolizes greater use of primary energy when compared to the current conventional system in place. Previous studies (Almeida, 2011) have shown that the use of 2-stage compression with intercooling results in greater energy efficiency. 2-stage compression especially increases the potential savings in the summer months. Figure 3-2 shows the potential improvement in the savings when two-stage compression is used in summer months. If the system uses single stage compression in these 4 warmer months, the proposed CO_2 heat pump system would utilize 195,958 kWh (-17.3%) units of energy more than the present system. The use of 2-stage compression in the summer months when the ambient temperature is high reduces the additional energy unit consumption to 80,685 kWh (-7%).

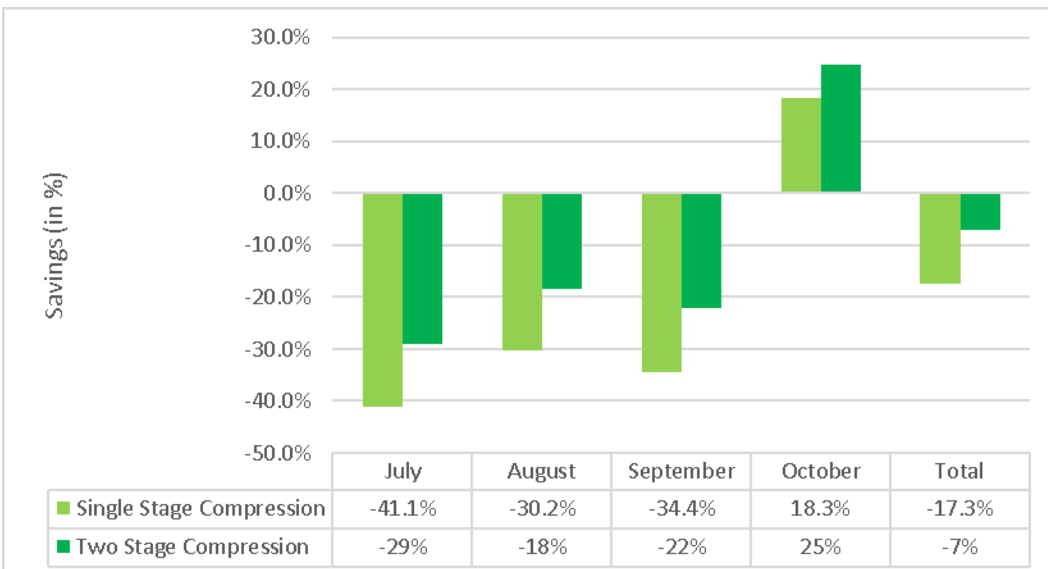

Figure 3-2. Potential primary energy savings when single stage compression is compared to 2-stage compression.

The results show that the most efficient system will be the one which works as a single-stage compression cycle in the winter months and a two-stage compression cycle with intercooling in the summer months.

3.2. Energy Savings with Addition of Work Recovery Device

A couple of studies have been conducted by researchers to compare the use of an expansion work recovery device in comparison to an expansion valve. Robinson & Groll (1998) in their study, derived that the expansion valve is the component with the largest percentage of total irreversibility. The study concludes that replacing the valve with an expansion work recovery turbine with an isentropic efficiency of 60% reduces the process's contribution to total cycle irreversibility by 35%. There is a notable increase in the Coefficient of Performance (COP) of the system as well.

The model discussed in Chapter 2 is used to compare the primary energy savings of the system with an expansion valve and a work recovery turbine with an isentropic efficiency of 70% . The results are shown in the Figure 3-3. The light green bar in Figure 3-3 shows the primary energy savings in percentage of the system with an expansion valve over an entire year and the dark green bar shows the primary energy savings of the system with an expander. The annual primary energy savings of the system increases to ~28% from ~19% in the case of an expansion valve. This is a significant improvement of ~40%. The results show that a work recovery expansion device will lead to more significant primary energy savings.

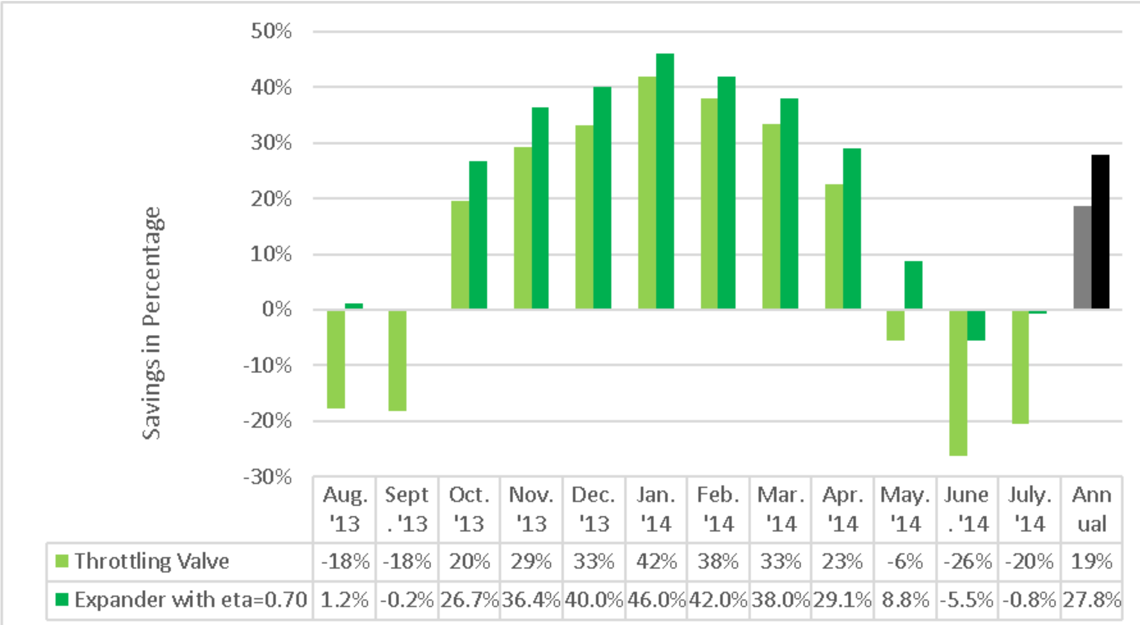

Figure 3-3. Primary energy savings - Comparison of system with throttling valve to a system with a work recovery expander.

3.3. Energy Savings with Increase in Data Center Supply Temperature

Conventional data centers such as the Purdue MATH data center uses chilled water of ~6° C. However, according to the new ASHRAE guidelines data centers can be operated at higher temperatures. Thermal guidelines have been evolving to address the growing concerns of energy efficiency. Steinbrecher & Schmidt (2011) summarize the evolving thermal guidelines to include wider temperatures and humidity ranges. The article discusses the changes in both the air cooling and liquid cooling guidelines. The new liquid cooling guidelines are listed in Table 3-1. Class W1 and W2 consist of data centers, which are typically cooled using chillers and a cooling tower, but depending on data center location uses an optional water-side economizer to improve energy efficiency.

Table 3-1. ASHRAE Liquid Cooled Guidelines (Steinbrecher & Schmidt, 2011).

Liquid Cooling Classes	Typical Infrastructure design		Facility supply water temperature
	Main Heat Rejection Equipment	Supplemental Cooling Equipment	
W1	Chiller/Cooling Tower	Water-side Economizer (With Dry cooler or Cooling Tower)	2°C to 17°C
W2			2°C to 27°C
W3	Cooling Tower	Chiller	2°C to 32°C
W4	Water-side Economizer (With Dry cooler or Cooling Tower)	N/A	2°C to 45°C
W5	Building Heating System	Cooling Tower	> 45°C

The data centers which use chillers only as supplemental cooling are classified under class W3. Class W4 and W5 data centers do not use chillers. Class W5 data centers are special data centers in which the ITE can withstand high temperatures of more than 45°C. In this sub-section, the change in energy savings with increase in supply temperature is shown. The results are limited to the temperature range of the class W1, W2 and W3 data centers since the technology being discussed uses chillers to cool the data centers. The primary energy savings over a year for the current data center requirements of chilled supply temperature of 6°C is shown in Figure 3-3. Figure 3-4, Figure 3-5 and Figure 3-6 represent the primary energy savings over a year if supply temperature are increased to 17°C (Class W1), 27°C (Class W2) and 32°C (Class W3). The total annual primary energy savings increase from ~28% to ~61% and the total annual COP increases from 4.5 to 8.1 when the supply temperature increases from 6°C to 32°C. This is clearly represented in Figure 3-7. This increase in performance is because of the increase in the evaporating temperature from -4°C to 22°C.

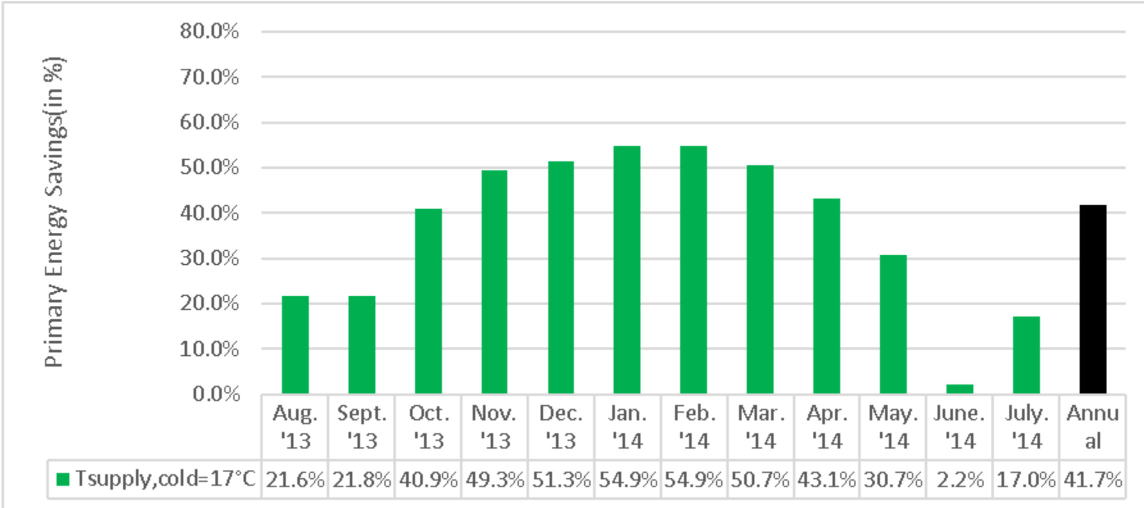

Figure 3-4. Annual primary energy savings for $T_{supply,\ cold}$ = 17° C.

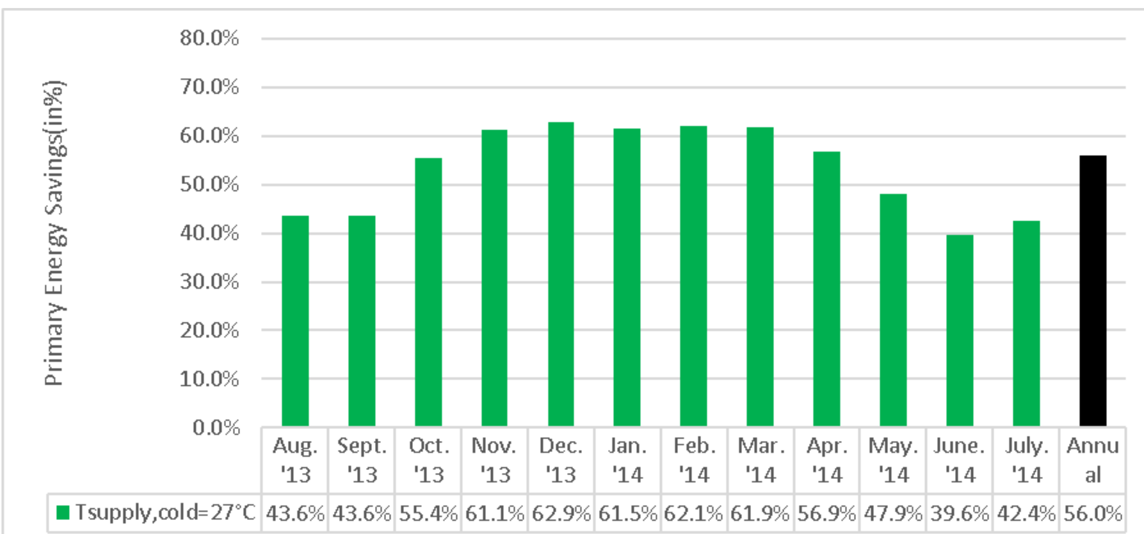

Figure 3-5. Annual primary energy savings for $T_{supply,\ cold}$ = 27° C.

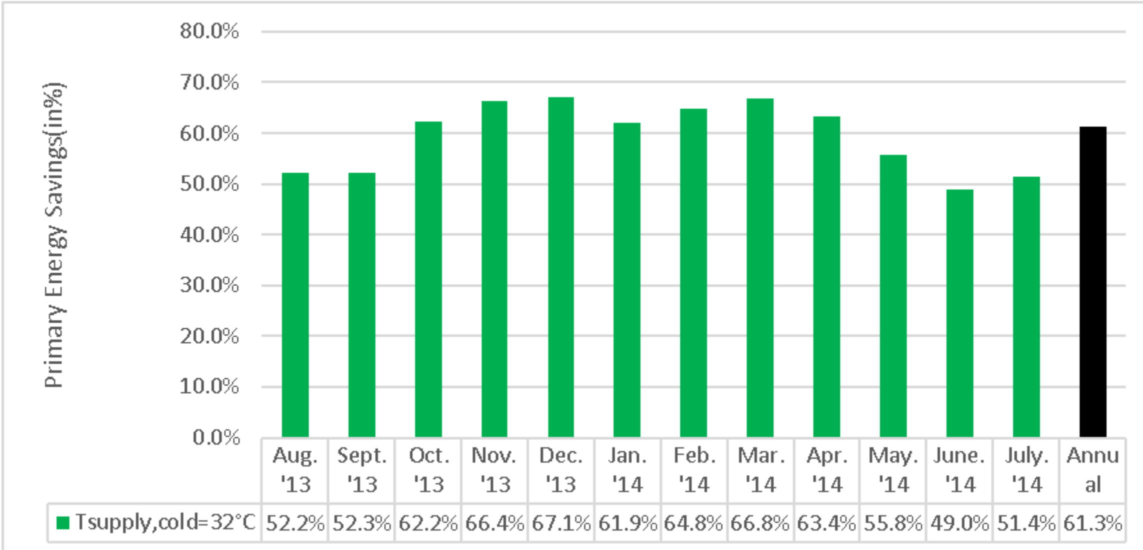

Figure 3-6. Annual primary energy savings for $T_{supply,\ cold}$ = 32° C.

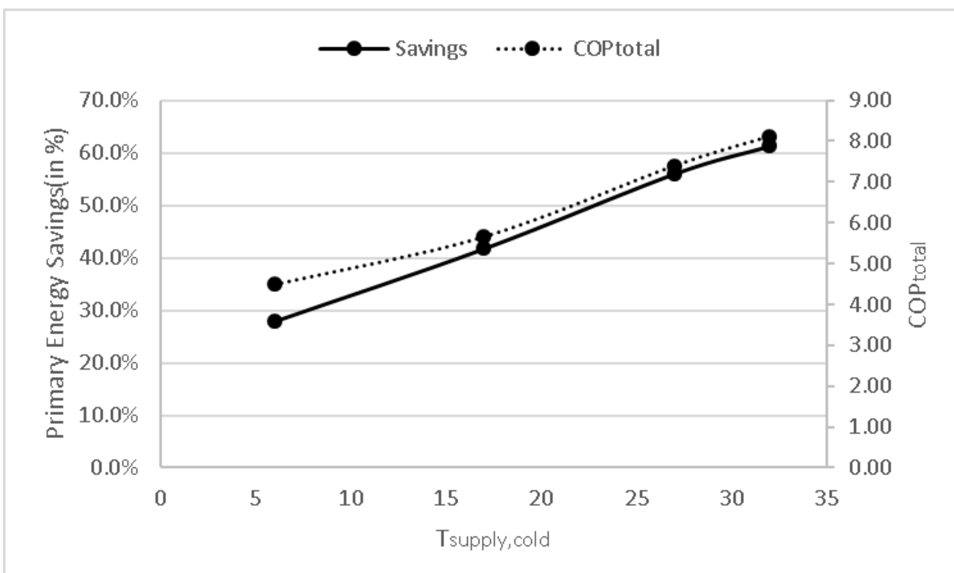

Figure 3-7. Variation of annual primary energy savings and total annual COP.

Increase in data center temperature might seem to be an attractive way of saving energy and reducing carbon emissions, however it shows some serious concerns such as impact on long term system reliability, effect on server performance, effect on server energy consumption and higher fan speeds. There is vast discrepancy in the results documented by various research on increased temperature on the system reliability. Nosayba et al.(2012) conducted a study on more than a dozen data centers at different organizations and discussed their observations. According to the research, the effect of high data center temperature on reliability might not be as prominent. Various factors determine failure of equipment and the previous temeperature correlation such as the Arrhenius model does not take the other factors into consideration. The research further discusses that a bigger issue with increasing data center temperature is increase in power consumption of individual servers. This increase in power consumption is caused by power leakage in the processor and increased server internal fan speeds. Further research needs to be concentrated on designing algorithms for fan speeds accurately as they make up for a significant fraction of total energy consumption.

3.4. Energy Savings with Daily Thermal Storage

On close examination of the heating load profile, it is realized that the heating load is low till 8:00 am. The heating load depends on the occupancy and peaks from 8:00 am until 10:00 pm. After which the load reduces again. To meet the peak load during the day, the system can be operated during the night at higher pressures than required until 8:00 am. The extra heat energy can be stored and then used to meet the peak load during the day. This is represented in Figure 3-8. The orange line indicates the heating provided by the CO_2 system. Whereas, the red line indicates the actual heating load of the building on a cold winter day. The shaded area represents the scope of recovery of heat stored during the night which can be liberated during the peak hours. Storage of thermal energy from the waste heat can reduce the power consumption and decrease the CO_2 emissions and lower the need for costly peak power and heat production capacity. This also gives the scope of inclusion of intermittent power sources such as solar PV-T technology. Finding optimum thermal storage capacity for reasonable response of time-dependent demands at minimum electricity will makes the system most attractive.

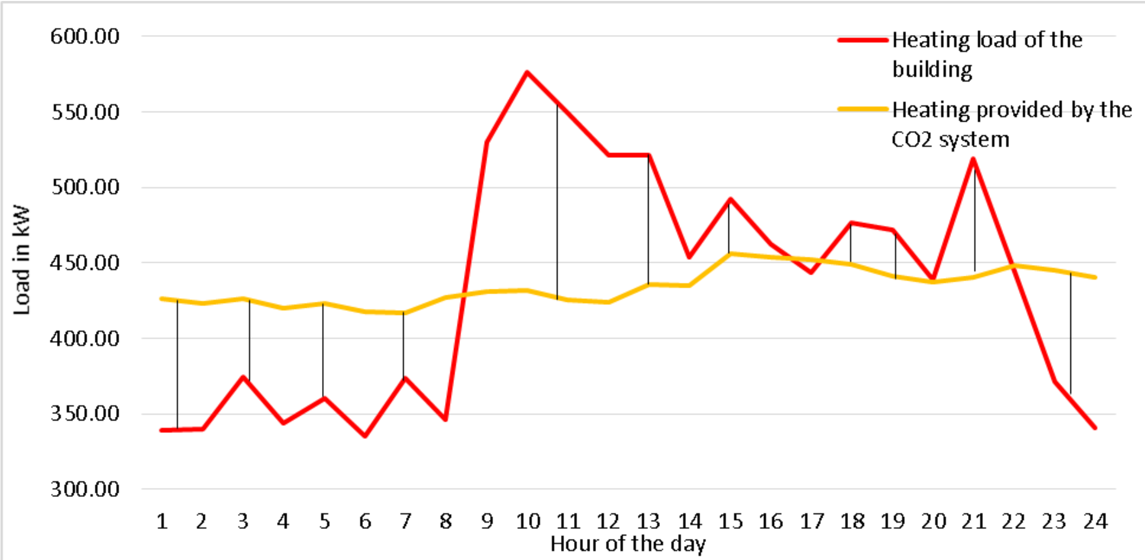

Figure 3-8. Potential use of storage.

Heier et al. (2014) reviewed the various storage techniques which are categorized in Figure 3-9. The CO_2 heat pump system can make use of storage that falls in the category: ‘Storage in HVAC’. Sensible heat storage makes use of the specific heat of the storage medium. Thermal Energy Storage (TES) can be stored at wide range of temperature as sensible heat, latent heat and chemical energy using chemical reactions.The most common sensible heat storage, which will also be discussed in this thesis, is the water storage. However, the storage capacity is limited by the specific heat. Phase change materials (PCMs) make use of the latent heat of storage and can offer higher storage capcity. A suitable PCM is dicussed in the following sub chapter. International Renewable Energy Agency (2013) briefs the various thermal storage technologies available and their commercialization status. It states that storage systems can be defined in terms of the following properties: capacity, power, efficiency, storage period, charge and discharge time and cost. The typical parameters of the various thermal storage systems are listed in Table 3.2.

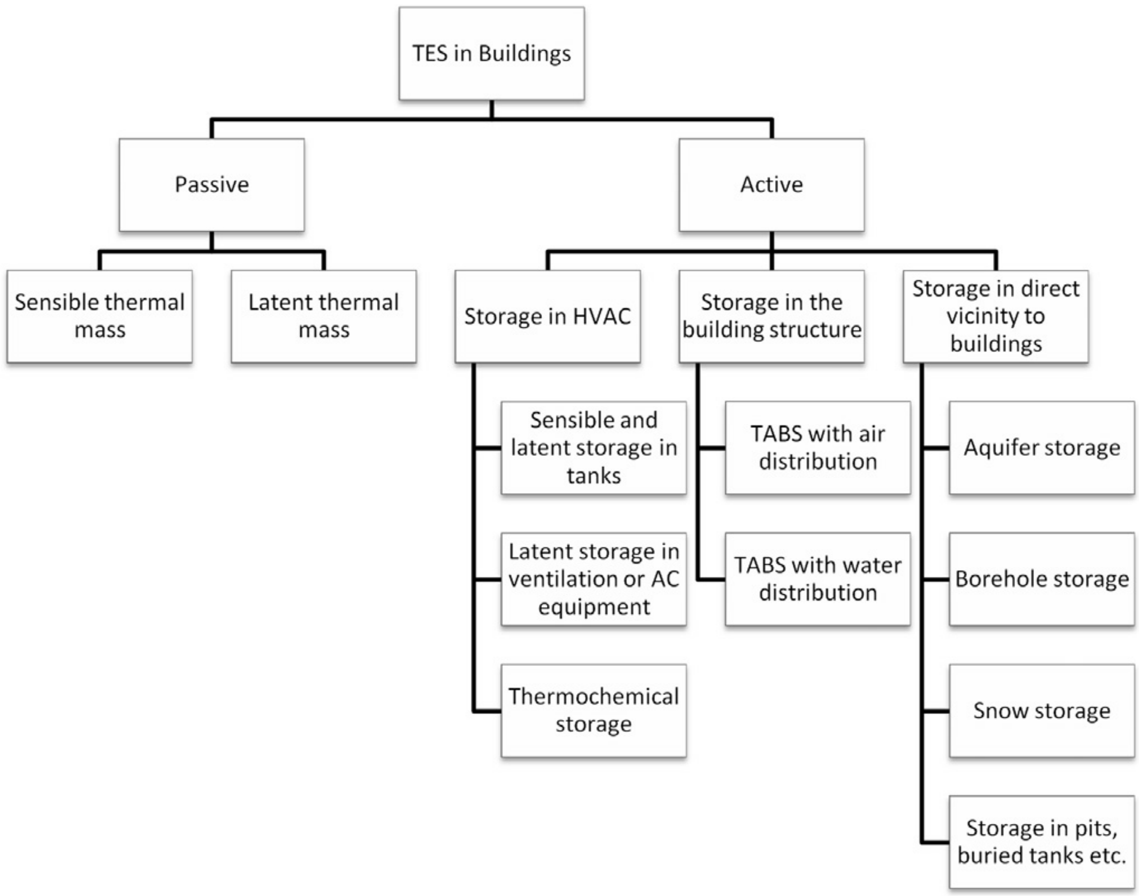

Figure 3-9. Categorization of various storage techniques (Heier et al., 2014).

Table 3-2. Typical Parameters of Thermal Energy Storage (International Renewable Energy Agency, 2013).

TES System	**Capacity (kWh/t)**	**Power(MW)**	**Efficiency(%)**	**Storage Period(h,d,m)**	**Cost($/kWh)**
Sensible(hot water)	10-50	0.001-10	50-90	d/m	0.11-11.35
PCM	50-150	0.001-1	75-90	h/m	11.35-56.73
Chemical Reaction	120-250	0.01-1	75-100	h/d	9.08-113.46

In the following subchapters, the size of sensible and latent thermal storage will be discussed.

3.4.1. Sensible Storage and Latent Storage System

Storage can be designed for meeting the daily load or seasonal load. However, for the present system, a daily storage will be more beneficial, taking into consideration, the large storage area and the cost. Figure 3-10 shows the savings in percentage with a daily storage unit. The annual savings in primary energy increase to 35%.

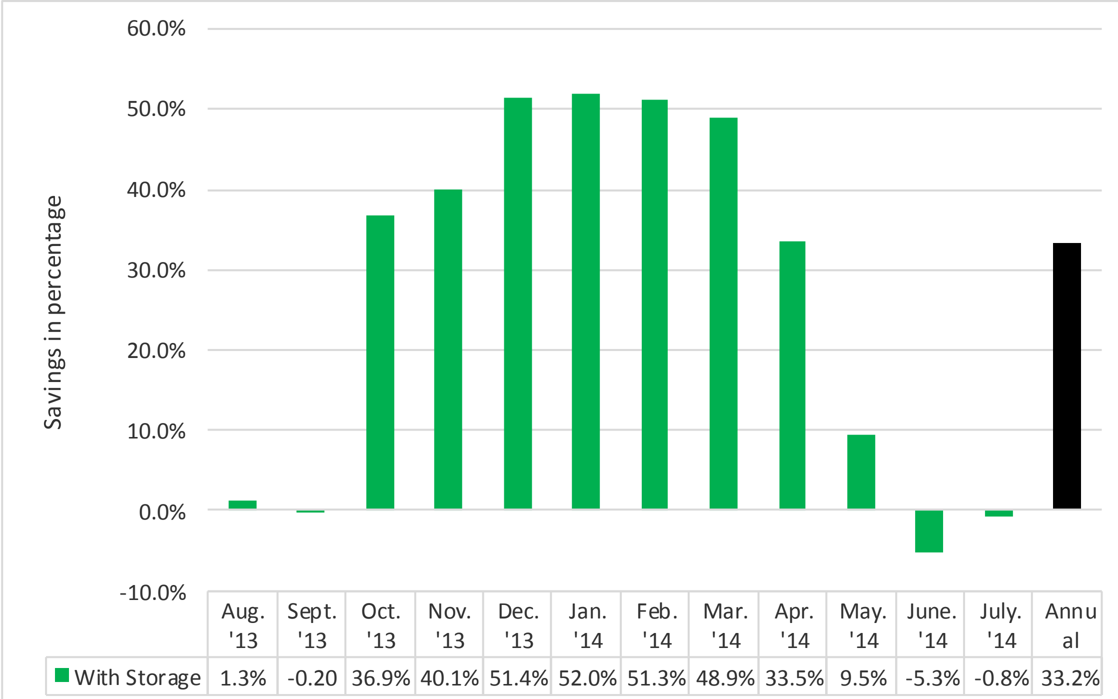

Figure 3-10. Savings in primary energy with daily storage.

On analyzing the data, the month of February seems to have the greatest demand for storage. To effectively design a storage system, the system has to be designed for the day which requires the maximum storage. The maximum storage required by the system is for 411 kWh units of energy at site. Water as a sensible Thermal Energy Storage (TES) medium is the most commonly used and commercially available storage. Large standardized as well as custom made water tanks with appropriate insulation are available in market. However, the size restriction might make water a less feasible option.

Phase Change Materials (PCMs) are being researched as an alternative to water as TES medium. PCMs is a developing technology, however some PCMs are commercially available in the market. Zalba et al. (2003) reviews all the potential PCMs, which can be used as energy storage materials. The author further tabulates the PCM's which are commercially available. From the commercially available PCMs, paraffin wax (RT 60) seems to be the best fit, taking into consideration its melting point temperature range. The properties of RT-60 provided by the manufacturer are listed in Appendix B. The melting temperature range of the TES is 55°C-61°C. This suites the application perfectly, since water is required for district heating at 62°C and the CO_2 heat pump system can provide water at ~65°C. This is shown in the schematic diagram Figure 3-11.

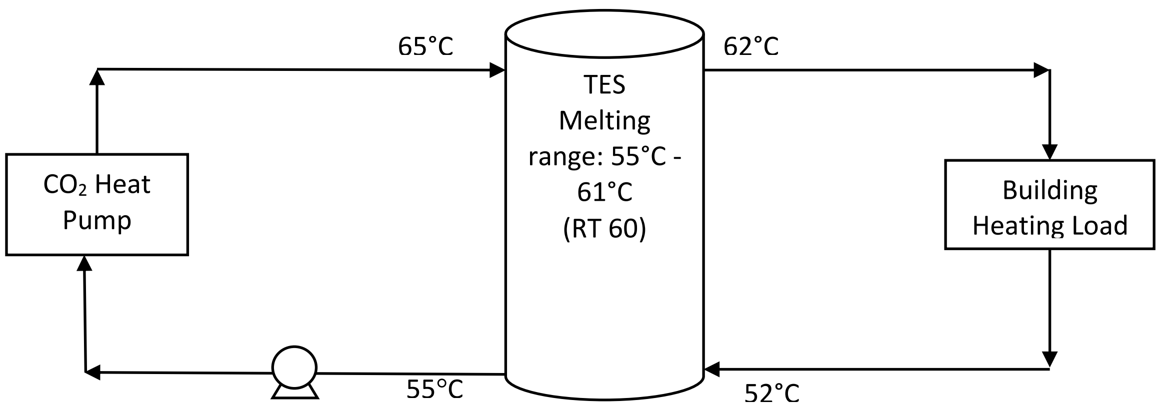

Figure 3-11. Schematic diagram of the system with PCM (RT-60) as the storage.

The major advantages of using Paraffin waxes are:

- Four to five times higher heat capacity by volume or mass, than water, thus making them more compact.
- Non water endangering substances (if melting above 27 °C)
- 100 % recyclable.
- Neither toxic nor dangerous to health

Table 3-3 gives a comparison between the size of water and PCM as storage for this application. PCM reduces the size of the storage by one-third. This is a significant decrease in the size and potentially an important factor since storage has to be placed inside a building in this application. However, Paraffin waxes are up to four times more expensive than water storage (He & Setterwall, 2002).

Table 3-3. Comparison between the size of water and PCM storage.

Storage: 411kWh = 1479600kJ		
	Water	**PCM (RT-60)**
Heat Capacity	4.186 kJ/kg-K	144 kJ/kg
Mass	35346 kg	10275 kg
Density	983.2kg/m³	770 kg/m³
Volume	**35.95 m³**	**13.34m³**

Paraffin waxes can be designed to be utilized in various geometries. One effective storage module is discussed in by Agyenim & Hewitt (2012). It consists of a horizontally mounted cylindrical storage shell containing Paraffin wax. A copper tube with longitudnal fins welded on the surface is placed in the center through which water to be heated passes. The fins are required to improve the heat transfer characteristics since paraffin wax has a low thermal conductivity (0.2W/m-K). A similar geometry will be beneficial in the sytem being discussed in this thesis.

From, the above size comparision PCMs prove to be a better option. However, economic analysis is required to make an appropriate decision which is discussed in Chapter 5.

CHAPTER 4. ENVIRONMENTAL ANALYSIS

After the successful phase out schedules of the ozone depleting refrigerants, such as CFC's and HCFC's, the concentration has shifted to lowering the harmful effects of substances that have high global warming potential. The phenomenon of Global Warming is a very critical issue and has been rightly receiving much social, economic and scientific interest. The fluorinated gases contribution to global warming is currently only 3% of the total greenhouse gas emissions (United States Environmental Protection Agency, 2015). Velders et al., 2009 projects that the contribution of HFC's to global warming will amount to 45% by 2050.

However, decisions should not be based only by comparing the global warming potential of the refrigerants. Refrigerants contribute directly as well as indirectly to the global warming phenomenon. Direct impact is a result of the global warming potentials developed by the Inter Panel on Climate Change (IPCC) that uses carbon dioxide as a reference gas. Indirect emissions are a consequence of the CO_2 emissions emitted while producing the power required by the equipment. Hence, energy efficiency of the equipment using different refrigerants should not be ignored. This research will utilize the concept of Total-Equivalent Warming Potential (TEWI) to compare the proposed technology with the conventional system. TEWI can be defined as in Equation (4.1).

$$TEWI_r = GWP_r * L_r * n + GWP_r * mass_r * (1-\alpha_r) + n * E_r * \beta \tag{4.1}$$

Parameters mentioned in Table 4-1 are used to calculate the TEWI for the proposed CO_2 heat pump system and the R410A chillers used to supply chilled water to the data centers. However, the CO_2 heat pump system provides heating to the building as well. The global warming equivalent from heating can be calculated using Eq. 4.3. The value of annual energy consumption for heating is 3026297kWh/year from the gathered data and simulation. The addition of TEWI due to heating and cooling gives the total TEWI value for the conventional system (Equation (4.4)).

Table 4-1. Parameters of the refrigerants input into TEWI.

1. Refrigerant Charge: Mycom Industrial CO_2 heat pump uses 10.8kg for a 74 kW system (Sheeco, 2013; Shecco Publishing, 2015)
2. Refrigerant charge of 410A (Emerson Climate Technologies, 2010)
3.Leakage rate of CO_2 system (Navigant Consulting, Inc, 2015)
4. Leakage rate of 410A system (The Australian Institute of Refrigeration, Air Conditioning and Heating, 2012)
5. GWP value (Rajendran, 2011)
6. Recycling factor (Maykot et al., 2004)
7. Annual Energy consumption from the data and simulation
8. CO_2 emissionfactors (Maykot et al., 2004)

Refrigerants	CO_2	410 A
Refrigerant Charge(mass) in kg/kWh	0.146[1]	0.5[2]
Annual Leakage rate(L)	10%[3]	7%[4]
GWP	1	2088[5]
Recycling Factor(α)	75%[6]	75%[6]
Annual Energy Consumption(E) in kWh/year	3572045[7]	19125307[7]
CO_2 emission factor of coal (β_{coal})	1.075kg CO_2 /kWh[8]	
CO_2 emission factor of natural gas($\beta_{naturalgas}$)	0.847kg CO_2/kWh[8]	
System Lifetime(n)	10 years	

For the calculations, the value of CO_2 emission factor (β) is calculated by using the coal and natural gas emission factor as listed in Table 4-1. There is 1 coal-fired and 3 natural gas boilers. Hence the total value is calculated as 0.904 kg CO_2/kWh using Eq. 4.2

$$\beta = 0.25 * \beta_{coal} + 0.75 * \beta_{naturalgas} \quad (4.2)$$

$$TEWI_{heating} = E_{heating} * \beta * n \quad (4.3)$$

$$TEWI_{conventional} = TEWI_{410A} + TEWI_{heating} \quad (4.4)$$

The above equations are used to tabulate (Table 4-2) the TEWI values of the current separate heating and cooling systems and the proposed combined CO_2 heat pump system.

Table 4-2. TEWI of the conventional and the proposed system.

<table>
<tr><th colspan="2">Systems</th><th>Direct Emissions (kg CO₂)</th><th>Indirect Emissions (kg CO₂)</th><th colspan="2">TEWI (kg CO₂)</th></tr>
<tr><td colspan="2">Combined CO₂ heat pump system</td><td>136.8</td><td>3.23E+07</td><td colspan="2">3.23E+07</td></tr>
<tr><td rowspan="2">Conventional System (Separate Heating and Cooling)</td><td>410A Chillers</td><td>5.68E+06</td><td>1.74E+07</td><td>2.31E+07</td><td rowspan="2">5.04E+07</td></tr>
<tr><td>Heating</td><td></td><td>2.74E+07</td><td>2.74E+07</td></tr>
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Figure 4-1 shows a significant reduction of 35.97% in the Total Equivalent Warming Impact (TEWI) value.

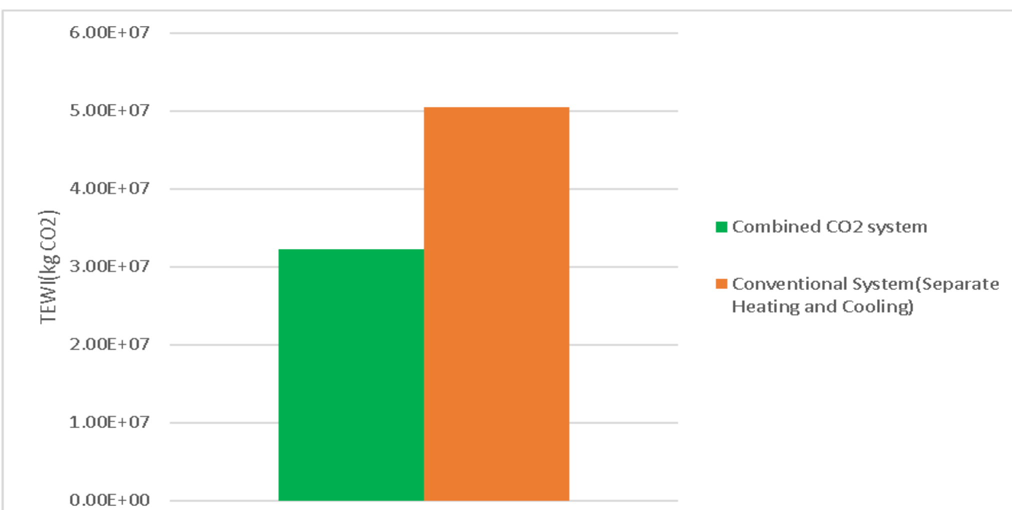

Figure 4-1. Comparison between the TEWI values.

CHAPTER 5. ECONOMIC ANALYSIS

While designing a new system, it is important to analyze the economic benefits of the technology as well. Various system improvements are discussed in Chapter 3. The significant energy savings lead to noteworthy operating cost savings. First, the operating cost savings are discussed and then a payback analysis is carried out to calculate the feasibility of the technology. Purdue University benefits from a power plant responsible for providing the required heating and cooling to campus buildings. It also provides a portion of electricity. The deficit in electricity need is bought from Duke Energy. The aggregate power cost averaged over a year is 0.0486$/kWh (Source: Wade Utility plant- Anthony Covarrubias).

The annual operating cost savings with change in the supply temperature is shown in Figure 5-1. The system can save $137,219 - $203,858/year.

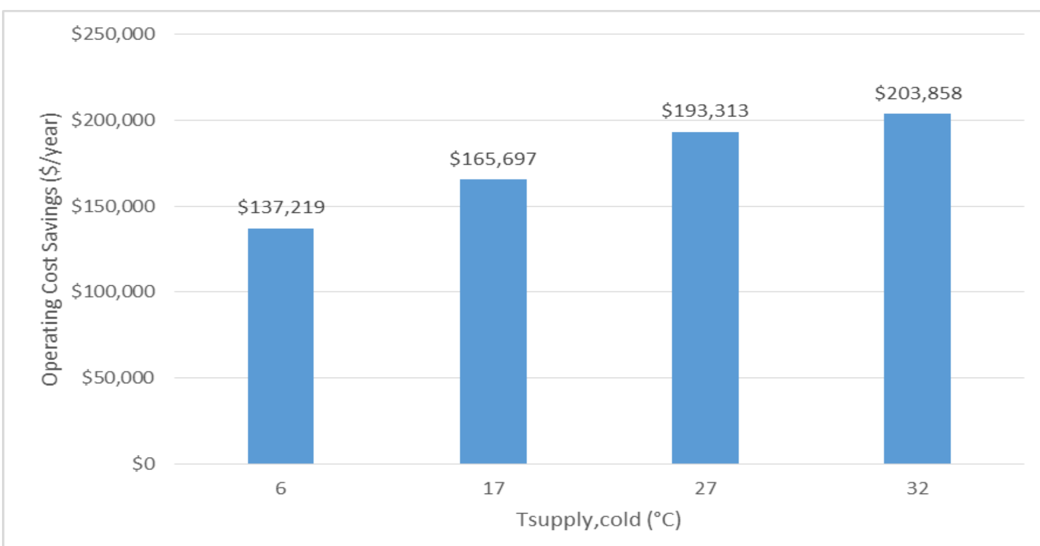

Figure 5-1. Operating cost savings.

This is a retrofit application since the cooling and heating systems are already in place. Currently installed CO_2 heat pump system prototypes have not been designed to provide such high load. Hence, the initial cost of this system is difficult to predict. Figure 5-2 shows the achievable pay back period with change in the initial cost of the system. Commercial production of CO_2 heat pump system is expected to lower the initial cost and thus, decrease the payback period. With commercialization of the system will make the system more feasible.

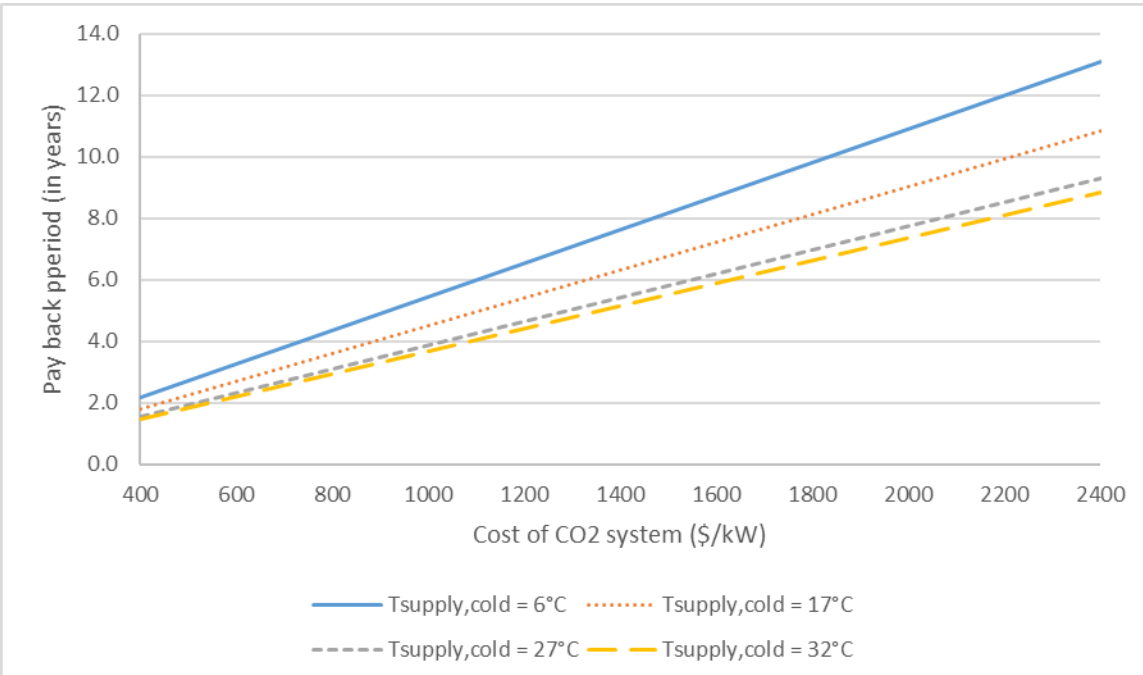

Figure 5-2. Variation of payback period with initial cost of the system.

5.1. Payback Analysis with Storage

Subchapter 3.4 discusses the advantages of including Thermal Energy Storage (TES). Integration of daily TES increases the energy savings to 33.2% from 27.8%. Cost of a standard water storage tank of ~37 m^3 is estimated to be $13,500 (State of Michigan, 2015). This cost is the average cost of the factory coated, bolted steel surface reservoirs erected on sand or gravel with a steel ring curb, including typical accessories such as roof, ladders, manways, vents, fittings on tank, and liquid level indicators, etc. Another important aspect of a water storage tank is the insulation. Cooperative Extension Service publication,2015 tabulates the effectiveness and cost of various thickness of insulation for a tank size of ~45m^3, which is represented in Figure 5-3.

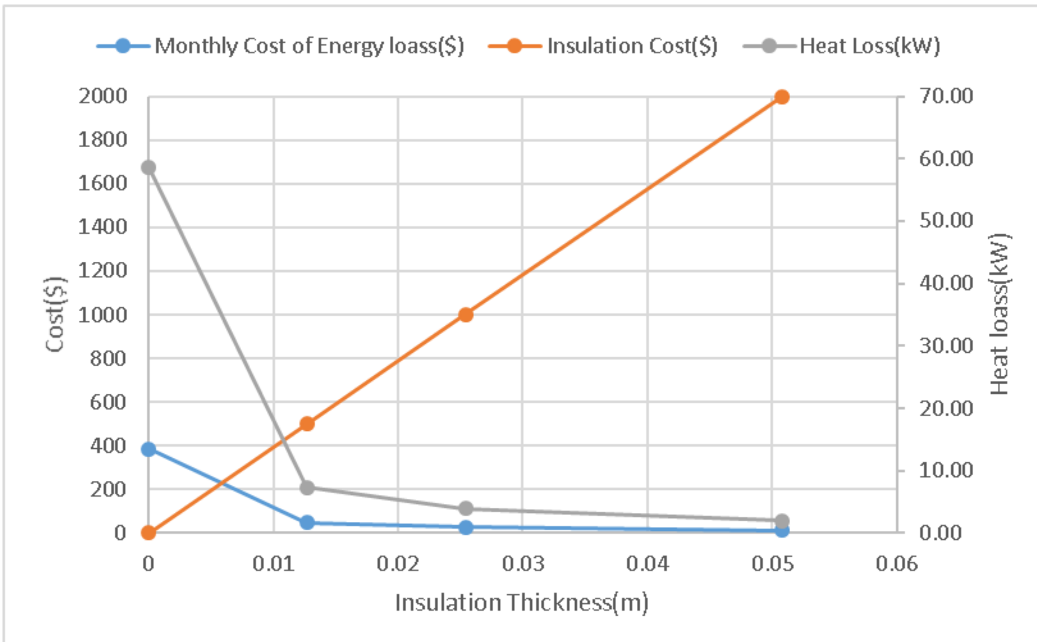

Figure 5-3. Effectiveness of insulation for a water tank.

The optimum insulation thickness from the graph can be assumed to be ~0.015m.

The total cost of water storage can be calculated using the following Equation (5.2)

$$Cost_{storage} = Cost_{\tan k} + Cost_{insulation} \quad (5.1)$$

Cost of insulation can be calculated using the area of tank which is ~67 m^2 (State of Michigan, 2015). The total cost (Fixed capital + Working Capital) of Paraffin

Wax is assumed to be 132.66-141.02 \$/kWh in He & Setterwall(2002). However the cost can increase significantly when integrated with a system such as a CO_2 heat pump system. These costs are difficult to assume because of the complexity and novelty of the system. Hence payback analysis with a range of cost values for storage is tabulated in Table 5-1.

Table 5-1. Payback analysis of the system with storage.

		Payback period (in years) for system for various cost of CO_2 heat pump (\$/kW)			
		400	**1000**	**1600**	**2200**
Cost of Paraffin Wax (in \$/kWh)	100	2.33	5.37	8.42	11.46
	150	2.48	5.53	8.57	11.61
	200	2.64	5.68	8.72	11.76
	250	2.79	5.83	8.87	11.92
	300	2.94	5.98	9.03	12.07
	350	3.09	6.14	9.18	12.22
	400	3.24	6.29	9.33	12.37
Cost of water tank storage(\$)	47000	2.53	5.81	9.09	12.37

The cost of storage is a small percentage of the whole system. When the cost of CO_2 heat pump system is considered low, the cost of the storage system becomes prominent and hence the difference between storage systems become significant. An appropriate comparison between the storage systems can be made with better knowledge of the cost. However, with expected reduction in cost of paraffin wax storage in future, paraffin wax seems to be a more preferable option as a storage system.

CHAPTER 6. SAFETY CONCERNS

Safety is a major concern in refrigeration application and is one of the main reasons why synthetic refrigerants were introduced and are being exploited. The threat from the phenomenon of global warming has propelled researchers to concentrate again on natural refrigerants. Apart from its other unique properties, CO_2 is the only non-flammable and non-toxic refrigerant amongst the other natural refrigerants. ASHRAE's safety group classification illustrated in Figure 6-1 is classified by two categories. The letters determine the increasing toxicity and the numbers denote the increasing flammability.

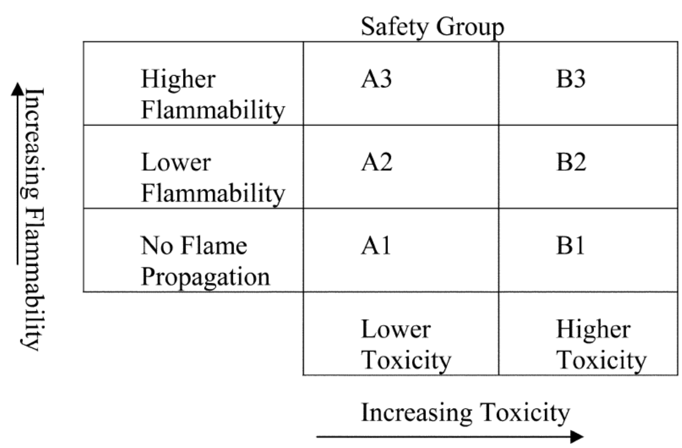

Figure 6-1. Refrigerant safety group classification (Environmental Protection Agency, 2011).

CO_2 is classified under the A1 category as most other hydrocarbons. Carbon dioxide is non-toxic, but it can be a cause of concern at higher concentrations. Carbon dioxide replaces air and causes lack of oxygen. There have been a couple of studies on the safety of carbon dioxide based air conditioning. Amin et al. (1999) studied the risk of high CO_2 concentration in a vehicle cabin. The authors discussed the safety concerns and presented some suggestion for CO_2 as a refrigerant in mobile air conditioning systems. A study relevant to the present application was conducted by Sawalha (2008). The author discussed the safety concerns in specific application of supermarket refrigeration. Safety is a very important factor especially in the case of data centers at Purdue University because of its proximity to the students. The system will be placed in a department building and hence careful consideration is required if a prototype of the transcritical CO_2 system is installed.

The most critical issue that is faced in CO_2 systems is the high pressure at standstill. In case of component failure, power cut or if the system has to be stopped for mantainence, the plant might start gaining heat from the surrounding as a result of which the pressure inside the plant will increase. The system might not be able to withstand very high pressure. The most common and easiest way is to release the charge. Since CO_2 is inexpensive, this method turns out to be the most feasible option. However, the CO_2 concentrations should be monitored by appropriate placement of the sensors. The health consequences at different concentrations are tabulated in Table 6-1. The study by Sawalha (2008) in supermarket refrigeration concludes that the use of CO_2 in supermarket refrigeration systems does not result in exceptional health risks for the customers or workers. However, CO_2 detectors are advised because of the non-self-alarming nature of carbon dioxide. Proper ventilation and alarm system should be mandatory in the machine room in case of an emergency.

Table 6-1. Different concentrations of CO_2 and their expected health consequences (Sawalha, 2008).

1. Time-Weighted Average (TWA) concentration that must not exceed during any 8 hour per day 40 hour per week; Threshold Limit Value (TLV): TWA concentration to which one may be repeatedly exposed for 8 hours per day 40 hours per week without adverse effect.

PPM	Effects on Health
350	Normal value in the atmosphere
1000	Recommended not to be exceeded for human comfort
5000	TLV-TWA[1]
20000	Can affect the respiration function and cause excitation followed by depression of the central nervous system. 50% increase in the breathing rate
50000	100% increase in breathing rate after short time exposure
100000	Lowest Lethal concentration
200000	Death accidents that have been reported
300000	Quickly results in unconsciousness and convulsions

CHAPTER 7. ANALYSIS OF MARKET BENEFITS

As mentioned earlier in this paper, the data center industry is expanding greatly. If the use of data centers keeps increasing at the present rate, data centers are expected to consume 222 billion kWh/year of total electricity by 2030. Out of this, 50% of the electricity is consumed by the cooling equipment. Waste heat can be recovered from the data centers to heat any buildings nearby. Assuming half of the existing data centers are located at places with climatic conditions similar to West Lafayette, Indiana, 95 TWh of energy can be saved annually by 2030. The model discussed in the previous section can be extended to calculate the potential savings in 2030 in other commercial buildings as shown in Table 7-1. The energy consumption data for these commercial buildings (D&R International, Ltd., 2010) is input into the model to determine the potential savings. Significant energy savings per year can be seen which can lead to important cost and environmental benefits.

Table 7-1. Potential savings in 2030 if the heat pump system is applied to other commercial buildings.

	Food Services	**Health Care**	**Inpatient**	**Lodging**	**Total**
Cooling required(TWh/year)	10.6	31.3	16.9	10.4	69.2
Heating provided by heat pump (TWh/year)	16.7	41.9	26.7	16.4	101.7
Percentage of total heating provided by heat pump	33%	23%	21%	14%	21%
Savings (TWh/year)	**7.58**	**22.5**	**12.2**	**7.47**	**49.75**

CHAPTER 8. CONCLUSIONS

CO_2 heat pump systems are an attractive alternative technology for commercial building applications which have simultaneous heating and cooling demands. Such a system leads to significant energy, cost and environmental benefits. A case study of a data center on Purdue University campus proves that the system is especially advantageous in a data center application. Some of the major conclusions from the analysis are:

- 2-stage compression with intercooler during the summer months increases the efficiency of the system.
- Adding an expansion work recovery device improves the primary energy savings by 40%.
- Increasing the data center supply temperature according to the new ASHRAE guidelines increases the primary energy savings to ~61%.
- The total annual Coefficient of Performance of the system varies from 4.5 to 8.1 depending on the data center supply temperature
- Addition of an optimally sized storage system further increases the primary energy savings from ~28% to 33.2%
- Size and economic analysis shows PCM (Paraffin Wax-RT60) to be a better alternative when compared to water as a TES medium. This is because size is an important factor in commercial buildings. Addition of PCM does not increase the initial cost and payback period considerably.
- The payback period is approximately between 3 to 4.5 years
- Environmental analysis shows the decrease in Total Equivalent Warming Impact (TEWI) value by ~36%

- Previous studies show that CO_2 leakage does not pose any exceptional health risks, however appropriate placement of CO_2 detector, proper ventilation and alarm system are strongly advised.
- Significant energy savings can be achieved if system is integrated appropriately in commercial buildings with simultaneous heating and cooling load. This system can potentially aid Department of Energy (DOE's) goal to enable 50% primary energy savings in the USA building sector by 2030.

CHAPTER 9. SCOPE OF FUTURE WORK

The major scope of future work is installation of a prototype system in a data center. CO_2 systems capable of handling smaller loads are discussed in Subchapter 1.3. However, a system designed for high loads as discussed in this thesis is not available commercially. Hence, additional research for components might be required.

Before creating a prototype of the system, the model discussed above can be made more robust by addition of averaged data over a longer period of time. Appropriate forecasting of data is important for design of a system. Due to unavailability of data for longer periods, the current model was created using data for a year.

Safety is a great concern due to the placement of the system in a university building. Simulations using computational fluid dynamics will be beneficial to make a decision of placement of sensors and ventilation.

The data center discussed in this study is a critical facility with high load. With appropriate funding, a prototype system can be installed in a smaller data center on campus and real time data can be collected to explore the feasibility of the system. The results from this prototype can then be extended to other commercial buildings.

Two TES storage mediums are discussed in this research. Some other storage medium and types can also be researched. One such opportunity is using geothermal storage. Such a storage system might be beneficial. However, various aspects such as location and cost need to be considered before making appropriate decision. Simulations with other storage types such as borehole thermal storage should be carried out.

There have been some prior studies on integration of renewable energy sources with data center. The proposed CO_2 heat pump system with storage gives a good opportunity for integration of intermittent power sources. Further system simulations will aid in increasing the understanding of such hybrid system.

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APPENDICES

Appendix A.

```
%Heating load of the building and cooling load of the data center
clear all
clc
%global P_2 h_2 T_2 s_2 T_3 P_3 h_3 s_3 T_4 P_4 h_4 s_4 P_5 h_5 T_5 s_5
filename= 'data.xlsx';
[num, txt,raw]=xlsread(filename);
[Rows,Columns]=size(num);

%%Constants
fluid='co2.fld';
rho_water=1000;                          %kg/m^3
P_cold=101.325;                          %kPa

T_sh=10;
eta=0.70;
T_hot_supply_dh=62+273;
T_hot_return_dh=51+273;
eta_boiler=0.85
eta_generator=0.80
eta_turbine=0.80
COP_electric=6.381
COP_steam=1.6
COP_avg=(10*COP_electric+3*COP_steam)/13

begday=1
endday=24
c=1;
k=2;
Savings(1)=0;
count(1:1)=c;
T_s(1:1)=62;
Delta_t=1*3600
for i=1:Rows
T_amb=((num(i,10)-32)/1.8)+273;
T_amb_tank=mean(T_amb);
end
C_p_water=4.186 %kJ/kg-K
while endday<Rows+1
    P_2=8000
    Q_CO2_day_heating(c)=300
    Q_hot_day(c)=100

while abs(Q_CO2_day_heating(c)-(Q_hot_day(c)))>1
    if Q_CO2_day_heating(c)>(Q_hot_day(c))
    P_2=P_2-0.5;
else
    P_2=P_2+0.5;
    end
if P_2>16000
    break;
end
for i=begday:endday
```

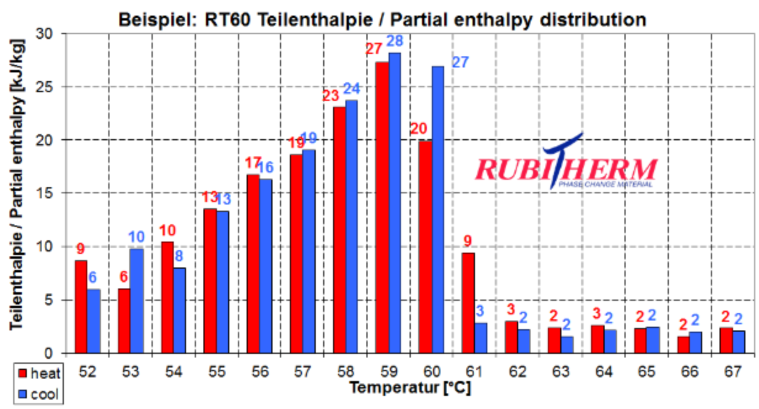
```
m_hot=num(i,5)*0.00000105*rho_water;    %Conversion rates to convert to
kg/s
T_hot_supply=((num(i,7)-32)/1.8)+273;          %F to C conversion
P_hot_supply=num(i,6)*6.895;            %psia to kPa
T_hot_return=((num(i,9)-32)/1.8)+273;          %F to C
P_hot_return=num(i,8)*6.895;            %psia to kPa
h_hot_supply=refpropm('H','T',T_hot_supply,'P',P_hot_supply,'water.fld'
);
h_hot_return=refpropm('H','T',T_hot_return,'P',P_hot_return,'water.fld'
);
Q_hot_load(i)=abs(m_hot*(h_hot_return-h_hot_supply)/1000);

m_cold=num(i,2)*0.0000631*rho_water;              %Conversion to kg/s
T_cold_supply=((num(i,4)-32)/1.8)+273;                   %F to C
conversion
T_cold_return=((num(i,3)-32)/1.8)+273;                    %F to C
conversion

h_cold_supply=refpropm('H','T',T_cold_supply,'P',P_cold,'water.fld');
h_cold_return=refpropm('H','T',T_cold_return,'P',P_cold,'water.fld');
Q_cold_load(i)=m_cold*(h_cold_return-h_cold_supply)/1000;

%%Compressor Inlet and Evaporator Outlet
T_evap=T_cold_supply-10;
T_1=T_evap+T_sh;
Q_1=1;
P_1=refpropm('P','T',T_evap,'Q',Q_1,fluid);
h_1=refpropm('H','T',T_1,'P',P_1,fluid);
s_1=refpropm('S','T',T_1,'P',P_1,fluid);

%%Compressor Outlet and Gas Cooler Inlet
h_2s=refpropm('H','P',P_2,'S',s_1,fluid);
h_2=h_1+(h_2s-h_1)/eta;
T_2=refpropm('T','P',P_2,'H',h_2,fluid);
s_2=refpropm('S','P',P_2,'H',h_2,fluid);

%%Gas cooler outlet
T_3=T_hot_return_dh+3;
P_3=P_2;
h_3=refpropm('H','T',T_3,'P',P_3,fluid);
s_3=refpropm('S','T',T_3,'P',P_3,fluid);
Q_gc(i)=h_3-h_2;

%%Air Cooler outlet
T_4=max(273,T_amb+10);                                  P_4=P_3;
h_4=refpropm('H','T',T_4,'P',P_4,fluid);
s_4=refpropm('S','T',T_4,'P',P_4,fluid);
Q_gc_air=h_4-h_3;

%%Expander Outlet
P_5=P_1;
h_5s=refpropm('H','P',P_5,'S',s_4,fluid);
h_5=h_4-eta*(h_4-h_5s);
```

```
T_5=refpropm('T','P',P_5,'H',h_5,fluid);
s_5=refpropm('S','P',P_5,'H',h_5,fluid);

Q_evap(i)=h_5-h_1;
m_dot(i)=Q_cold_load(i)/abs(Q_evap(i));
Power_exp(i)=abs(h_4-h_5);
Power(i)=((h_2-h_1)-Power_exp(i));
Power_consumption(i)=m_dot(i)*Power(i);
COP_h(i)=abs(Q_gc(i)/Power(i));
COP_c(i)=abs(Q_evap(i)/Power(i));

time(i)=i;
COP_total(i)=COP_h(i)+COP_c(i);

Q_CO2_cooling(i)=m_dot(i)*Power(i)*COP_c(i);
Q_CO2_heating(i)=m_dot(i)*Power(i)*COP_h(i);
Storage_hot(i)=Q_CO2_heating(i)-Q_hot_load(i);
end
Q_CO2_day_heating(c)=0;
Q_CO2_day_cooling(c)=0;
Q_cold_day(c)=0;
Q_hot_day(c)=0;
Power_consumption_day(c)=0;

for j=begday:endday
Q_CO2_day_heating(c)=Q_CO2_heating(j)+Q_CO2_day_heating(c)
Q_CO2_day_cooling(c)=Q_CO2_cooling(j)+Q_CO2_day_cooling(c);
Q_cold_day(c)=Q_cold_load(j)+Q_cold_day(c);
Q_hot_day(c)=Q_hot_load(j)+Q_hot_day(c);
Power_consumption_day(c)=Power_consumption(j)+Power_consumption_day(c);
end
Q_primaryheating_day(c)=Q_hot_day(c)/eta_boiler;
Q_primarycooling_day(c)=Q_cold_day(c)/(COP_avg*eta_turbine*eta_boiler);
Q_primarypower_day(c)=sum(Power_consumption_day(c))/(eta_generator*eta_
turbine*eta_boiler);
Difference_day(c)=Q_CO2_day_heating(c)-Q_hot_day(c);
Difference_primary_day(c)=Difference_day(c)/eta_boiler;
Savings(k)=Q_primaryheating_day(c)+Q_primarycooling_day(c)-
Q_primarypower_day(c)+Difference_primary_day(c);

if Savings(k)<Savings(k-1)
    Savings_day(c)=Savings(k);
    break;
end
pressure(k)=P_2;
Q_h_load(k)=Q_CO2_day_heating(c);
Savings_day(c)=Savings(k)
k=k+1;
end
k=2;
P_high(c)=P_2
Savings(1)=0;
begday=endday+1
endday=begday+23
```

```
c=c+1;
count(c)=c;
end
%%Sizing of the storage
Storage_max=max(abs(Storage_hot))*3600;
Delta_T_tank=(T_hot_supply_dh-T_hot_return_dh);
m_storage=Storage_max/(C_p_water*Delta_T_tank);
V_storage_m3=m_storage/1000;
V_storage_gallons=V_storage_m3*264.172052
l_tank_iter=V_storage_m3/(pi*(d_tank^2/4))


%%Annual Energy Savings

COP_heat=sum(COP_h)/Rows
COP_cold=sum(COP_c)/Rows
COP_total=COP_heat+COP_cold
Q_total_heatload=sum(Q_hot_load)
Q_Co2_total_heat=sum(Q_CO2_heating)

Q_total_primaryheating=sum(Q_hot_load)/eta_boiler
Q_total_primarycooling=sum(Q_cold_load)/(COP_avg*eta_turbine*eta_boiler
)
Q_total_primarypower=sum(Power_consumption)/(eta_generator*eta_turbine*
eta_boiler)

Difference_total=Q_Co2_total_heat-Q_total_heatload
Difference_total_primary=Difference_total/eta_boiler

Savings_total=Q_total_primaryheating+Q_total_primarycooling-
Q_total_primarypower+Difference_total_primary
Reduction_total=(Q_total_primaryheating+Q_total_primarycooling-
Q_total_primarypower+Difference_total_primary)/(Q_total_primaryheating+
Q_total_primarycooling)
```

Appendix B.

The most important data:	Typical Values	
Melting area	55-61 main peak: 60	[°C]
Congealing area	61-55 main peak: 61	[°C]
Heat storage capacity ± 7,5%	144	[kJ/kg]*
Combination of latent and sensible heat in a temperatur range of 53°C to 68°C.	40	[Wh/kg]*
Specific heat capacity	2	[kJ/kg·K]
Density solid at 15 °C	0,88	[kg/l]
Density liquid at 80 °C	0,77	[kg/l]
Heat conductivity (both phases)	0,2	[W/(m·K)]
Volume expansion	12,5	[%]
Flash point (PCM)	>200	[°C]
Max. operation temperature	80	[°C]



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The product information given is a non-binding planning aid, subject to technical changes without notice. Version: 26.09.2014

Measured with 3-layer-calorimeter.

Figure B-1. Paraffin Wax (RT-60) properties from Rubitherm GmbH. (http://rubitherm.com/english/index.htm)