ABSTRACT

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The purpose of this paper is to analyze and determine the feasibility of a cold thermal storage system in manufacturing Industries. Cooling loads and actual plant data for some manufacturing facilities will be examined and the benefits of implementing thermal storage will be weighed against its absence. In this paper, two cases of thermal storage use in industry will be analyzed. The first case will deal with thermal storage use for process cooling in a facility. The second case will deal with thermal storage use for air conditioning.

In case one, the use of a thermal storage system for the process cooling of gases before they undergo compression proved to be beneficial since the work done by the compressors on the gas is significantly reduced. The payback for the installation of a thermal storage system is 4 years.

Case two dealt with the most common use of thermal storage in industry, air conditioning. The payback and cost savings from operating the thermal storage system at different operational schedules were analyzed. Of the three analyzed schedules, Load Shifting a type of partial storage operating schedule had the shortest payback of 16 years.

Thermal Energy Storage: Analysis and Application

by
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DEDICATION

This thesis is dedicated to my parents for the opportunities they have provided for me.

BIOGRAPHY

Dolanimi Ogunkoya was born in Lagos, Nigeria on August 4, 1985. After earning his Junior School Certificate in 1998 from Nigerian International School, Benin Republic, he returned to Nigeria where he graduated from CTC International School in 2001. In 2002, he moved with his parents to Harare, Zimbabwe where he spent a year in grade eleven at the Harare International School. Dolanimi graduated with a Bachelor of Science in Mechanical Engineering from New Jersey Institute of Technology in 2007. In the fall of the same year, he was accepted as a graduate student in the Mechanical Engineering department at the North Carolina State University.

ACKNOWLEDGMENTS

For his constant guidance and support throughout the research, I would like to thank my advisor Dr. Stephen Terry whose guidance saw me through the completion of this paper. I would like to sincerely thank Dr. Herbert Eckerlin who began it all when he introduced me to the IAC and Dr. Tiegang Fang for taking time off their busy schedules and serving on my advisory committee.

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Chapter 1 Background

1.1 INTRODUCTION

Energy storage is a means of retaining a particular form of energy for more efficient utilization and discharge. Reasons for energy storage are limitless and expected to continue to grow. Some of the major reasons for energy storage are [1]

- To store energy from renewable energy sources whenever possible as the energy output is usually inconsistent.
- Energy storage is used to reduce cost incurred by energy generating plants which still run at full capacity even when energy demand is low.
- Portability of the stored energy.
- Reduction in the degradation of the environment.
- More efficient management of energy when stored.
- The option of storing energy from inconsistent energy sources is another reason for energy storage.

Thermal Energy Storage (TES) is simply the storage of high or low – temperature energy for later use [2]. Thermal energy storage can be classified into two; sensible heat storage which is a storage system based on changing the temperature of a material and latent heat

storage which is a storage system that utilizes a materials ability to change phase. Thermal storage applications are widespread; they are used in power generation, HVAC applications, electricity demand side management amongst others. Reasons for thermal energy storage range from simply reducing demand charges and peak power usage to environmental concerns.

1.2 Thermal Storage Classifications

The classifications of thermal energy storage are based on the functions imposed on the materials responsible for thermal storage. Thermal energy storage deals with the storing of energy by cooling, heating, melting, solidifying or vaporization of a material which when reversed provides thermal energy [2] for later use. The two classifications of thermal energy storage are sensible thermal energy storage and latent thermal energy storage.

Sensible Thermal Energy Storage (STES): This method of storing thermal energy works by changing the temperature of the storage medium such as water, soil or oil. However, no phase change of the thermal storage medium occurs regardless of the changes in temperature of the medium. The amount of heat stored by the medium can be obtained by using equation 1.0.

$$Q = mc_p\Delta T \quad (1.0)$$

Where,

Q = heat stored

m = mass of media

ΔT = change in temperature

c_p= specific heat

The above equation shows that the amount of energy into a STES is directly proportional to the mass of the medium used for storing the energy, the specific heat of the medium and the difference in the storage medium's initial and final temperature.

The choice of storage medium depends on the temperature range needed for thermal storage. A material's ability to store sensible heat, its cost and the rate of heat release from the material are all prerequisites the material must meet in order to serve as an acceptable medium. A typical sensible thermal energy storage system consists of a storage tank (holds the storage medium and consequently the thermal energy), the medium for 'holding' thermal energy such as water, input and output devices such as piping to allow for the transfer of energy in and out of the storage tank.

Latent Thermal Energy Storage (LTES):

This is the storage of heat as the latent heat of fusion in suitable substances that undergo melting and freezing at a desired temperature level. Latent heat storage has the ability to provide a high energy storage density while storing heat at a constant temperature. The constant temperature corresponds to the temperature at which the heat storage medium changes phase. Examples of such substances include ice, paraffins or salts. Storage of heat in substances as the heat of crystallization, for materials undergoing phase change from one solid phase to another is also classified as latent heat storage [1]. Heat stored in materials going from a liquid to vapor state is not classified as latent heat storage, as problems arise with storing of gaseous phase and the requirement of a pressurized vessel.

There are three basic requirements for a latent heat thermal energy storage system [1]. These requirements are:

- A heat storage substance that undergoes solid to liquid phase transition and vice versa.
- A storage vessel for the heat storage media.
- A heat exchanger surface for transferring heat from the heat source to the heat storage media and from the heat storage media to the sink.

Thermodynamic, kinetic, chemical and economic criteria need to be considered when determining materials to be used for latent heat energy storage.

1. Thermodynamic criteria:
    - Material must have a melting point in the desired temperature range.
    - High thermal conductivity to allow for quick charging and discharging of energy stored in the material.
    - High latent heat of fusion per unit mass, to allow a small amount of material to store a large amount of energy.
    - High density, this allows for the use of a smaller storage tank.
2. Kinetic Criteria
    - There should be no super cooling during freezing. The storage media should crystallize at its thermodynamic freezing point.
3. Chemical Criteria
    - The material should be chemically stable. It should be non toxic and non-explosive.
    - Material should be non corrosive to materials which may include piping and storage tank.
    - The phase change material should show no chemical decomposition to ensure a longer lasting system.

4. Economic criteria
    - The material to be used as a medium must be cheap and readily available.

The heat exchanger as required by this system must meet the three requirements below.

1. The heat exchanger surface must provide a high effective heat transport rate in order to facilitate rapid charging and discharging of the stored heat.
2. It should allow only small temperature changes for charging and discharging of the stored energy.
3. The heat exchanger must be able to sustain a high thermal diffusivity.

Chapter 2 Cold Thermal Energy Storage

2.0 Definition of Cold Thermal Energy Storage

Cold Thermal Energy Storage is a means of storing energy at night during off peak periods when electricity demand is lower for use during on-peak periods. Energy is stored in cold thermal energy storage (CTES) in the form of cooling capacity. Cooling capacity can be stored by chilling or freezing various fluids and media such as water, glycol and eutectic salts. The choice of media will depend on the purpose of the storage device and how effective the medium of storage is in storing latent energy or sensible energy. Water is the most widely used media of storage for various reasons, which include its availability, its compatibility with a wide variation of cooling systems and its being non toxic.

In industry, air-conditioning constitutes a high fraction of a facility's electricity demand. Air-conditioning runs typically during the day when the facility is running at full capacity and the outside weather is less than desirable (warm/hot). The cooling load on the air-conditioning system will therefore more than likely be at its peak during the day which falls in the on- peak demand period. In order to offset the otherwise high cost of electricity a facility will incur during the day from air-conditioning, cold thermal storage is now been employed by many facilities as a means of transferring electric demand to night or off peak periods when electricity it is less expensive. At night during off peak periods, a cold thermal storage system employs refrigeration to cool the storage media from which cooling capacity will be obtained during the day.

A typical cold thermal storage system is made up of a storage tank, the medium (typically water), a refrigeration system (chiller), piping and pumps, and a heat exchanger depending on whether it is a latent energy system (ice storage) or a sensible energy system (chilled water). Figure 2.0 shows a simple schematic of a cold thermal storage system connected to a system load which can for instance be an air-conditioning system for a facility.

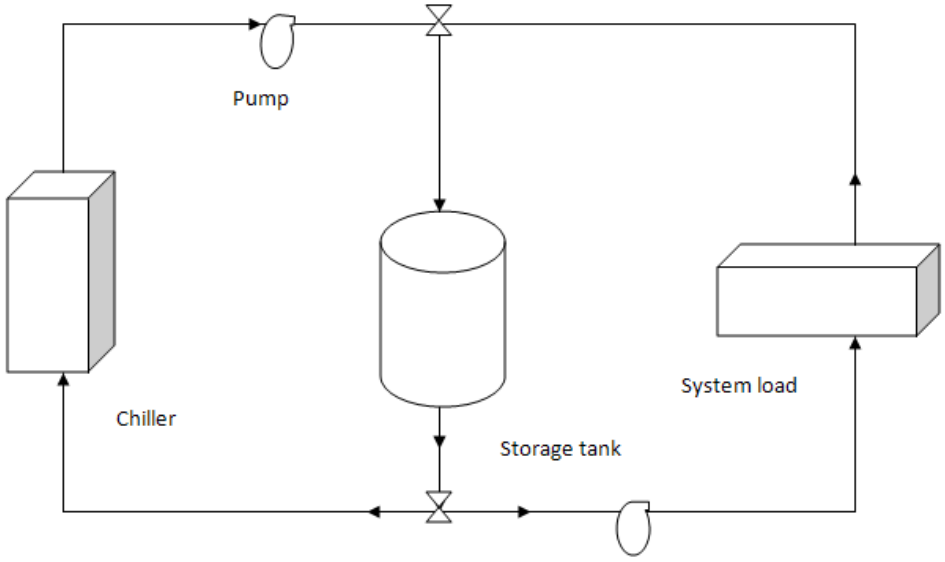

Figure 2.0: Schematic of thermal storage system.

Cold water or glycol depending on specific requirements of the system, which may include the systems desired temperature, is pumped into the storage tank from the chiller. The chiller is used to remove energy from glycol.

The cold glycol solution enters the tank from the top and as it pumped through the tank, it removes energy from the ice balls thus freezing the distilled water in the balls. The now "warmer" glycol leaves from the bottom of the storage tank and is pumped back through the

chiller as the process is repeated. The process of creating ice in the storage tank is referred to as the “charging cycle”.

In the day - time during on-peak periods and when cooling capacity is desired, the “discharge cycle” starts. In this cycle, glycol solution is pumped into the storage tank from the top. Ice balls in the storage tank absorb energy from the glycol solution thus cooling the solution as it flows through and out of the tank. The cold glycol solution leaving the storage tank is directed to the system load which may be air-conditioning or a process cooling load at the facility. Depending on the air-conditioning system or process cooling in place, a heat exchanger may be required to transfer energy from the cold glycol solution to the chilled water. For instance, a facility that cannot afford to run glycol solution to cooling coils either due to cost or process restrictions will need a heat exchanger to remove energy from water to cold glycol. The cold water leaving the heat exchanger is then pumped through the cooling coil in the air handling unit to provide cooling.

There are three main types of storage media used in cold thermal storage systems. They are ice, water and eutectic salts. A chilled water system has the lowest storage density and is the least complex of the three systems. This system requires the largest storage tanks but can also interface easily with existing chiller systems at a facility [3]. It uses the sensible heat capacity of water to store cooling capacity; it operates at an ideal operating temperature of (38^{o}F to 41^{o}F) and is most economical for systems greater than 2000 ton-hours.

Ice systems use the latent heat of fusion of water to store cooling capacity. They require a more complex chiller system and may either use special ice making equipments or modified chillers for low temperature service.

A thermally insulated storage tank depending on the type of cold storage system may be filled with ice balls. An ice ball is a spherical plastic container with expandable sides or indentations an example is shown below.

Figure 2.1: Ice Ball

The steps to determine storage tank capacity, quantity of ice balls and consequently cost are now considered. The first step will be to determine storage capacity which can be obtained from the cooling load calculations to be discussed in Chapter 3.

Theoretical ice quantity = $Cooling\ load\ (ton - hrs)$ x $\frac{12{,}000\ Btu}{1\ ton{-}hr}$ x $\frac{1\ lb\ ice}{144\ Btu}$

From the ice quantity, the volume of ice can then be obtained.

Theoretical ice volume = $ice\ Quantity\ (lb\ ice)$ x $\frac{1\ gal\ ice}{7.5\ lbs}$

Recalling that ice balls have indentations that bulge out once ice is formed in the ball, there will be a 20% void space between the packed ice balls in the storage tank. The volume of the tank must then be calculated by factoring in this void space.

A facility's installation of a cold thermal storage system is typically affected by a variety of factors the biggest of which is cost. Considerations made by a manufacturing industry before installing a cold thermal storage system are [2]:

- Cost of installation and payback.
- Expansion of an existing cooling system
- Expensive on-peak demand prices
- A facility's maximum cooling load is much greater than the average load.

Incentives provided by the utility company may also be a good reason for a facility to install cold thermal storage. Other reasons may include the desire for a cooling media as part of the facilities process and the existence of a storage tank on site.

2.1 Cold thermal storage system load

Full and partial storage are the two plans of action by which a thermal storage system charges and discharges its cooling capacity. With a full storage plan, all the cooling loads are moved to off-peak hours. A partial storage cold thermal storage system will have the chiller operate to meet a facility's peak period load with the cold storage system providing the rest of the cooling load.

Full cold storage systems are set to operate when cooling load is expected to be at its peak. This system is more beneficial for industries that have a very high demand charge during on peak hours or short on peak periods. With these conditions met, the system is used to offset the cooling load needed during on peak periods with cooling capacity generated at night during an off- peak period when electricity is cheaper. This system may be referred to as a load shifting operational schedule as the on-peak loads are redistributed to off-peak periods. Figure 2.2 shows how chiller loads and process loads vary throughout a typical day. Note that for this operational schedule, the chiller is turned off during on-peak hours and the thermal storage system provides all of the required cooling.

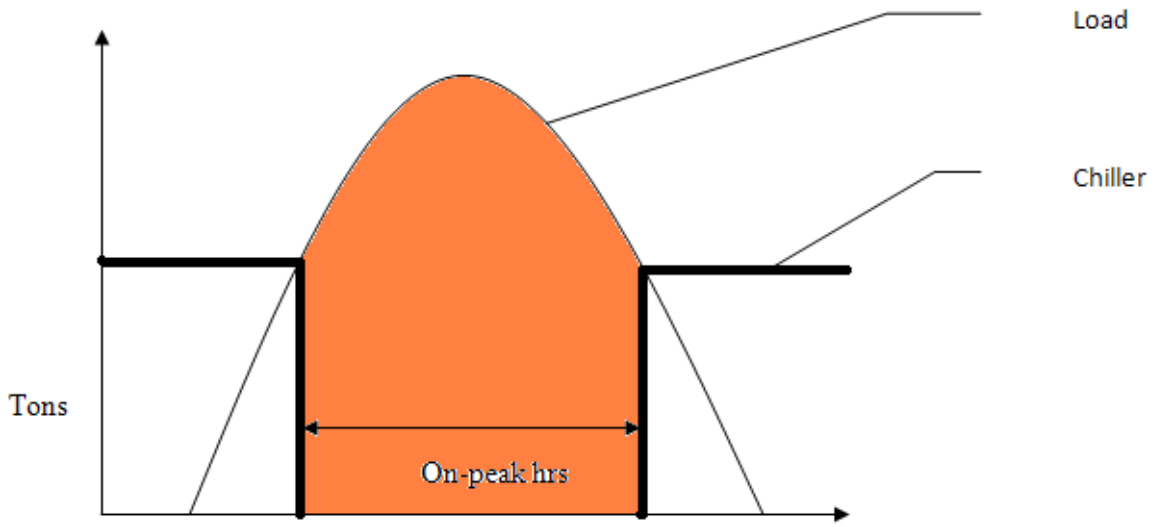

Figure 2.2: Full storage systems[2]

A partial storage cold thermal storage system offsets the cooling load during on peak periods with both a chiller and the pre-generated cooling capacity. This system may operate as either a *load leveling or a demand limiting system*. As a load leveling system, the chiller runs all day at full capacity, when the cooling load is lower than the chiller output; the surplus is stored in the storage tank. When the cooling load requirement exceeds the chillers output, the thermal storage system discharges cooling capacity to meet the additional load. A plot of a load leveling system is shown in Figure 2.3.

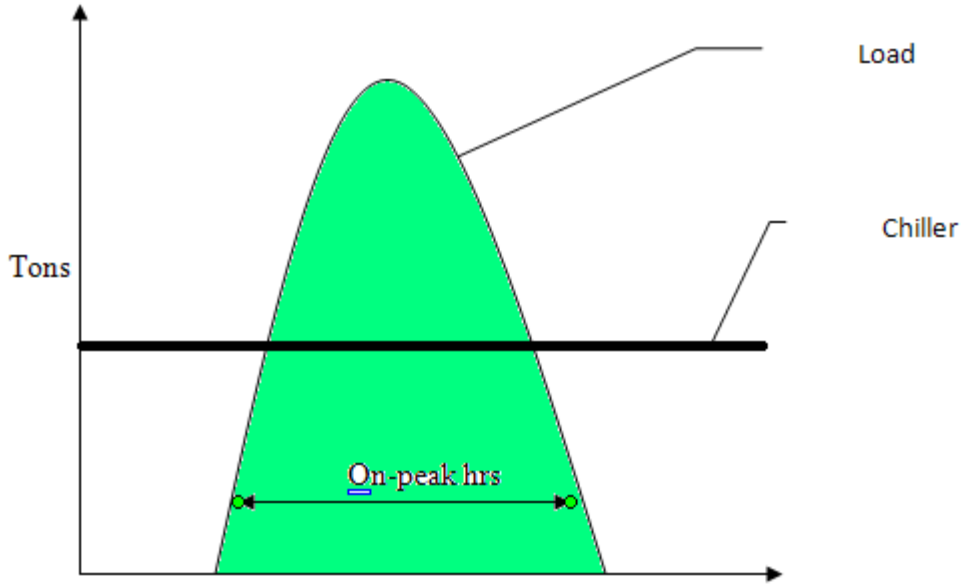

Figure 2.3: Partial storage load leveling [2]

In a *demand limiting system*, the chiller runs at reduced capacity during on-peak periods in order to reduce a facilities peak demand charge. The cost of equipment and demand savings for this systems operation are higher than for a load leveling system but lower than a full storage system. The system operation is shown in Figure 2.4

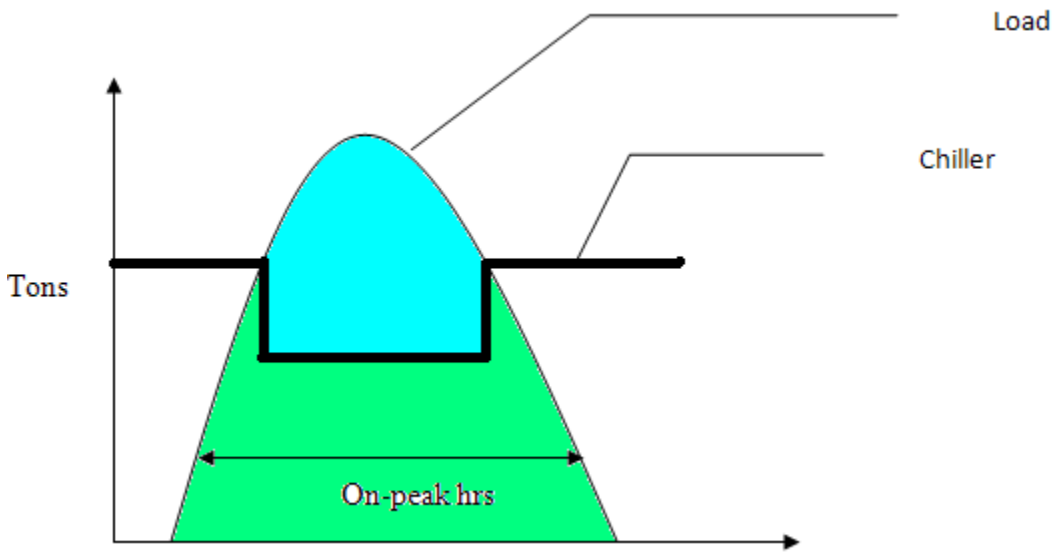

Figure 2.4: Partial –storage demand limiting system [2]

2.2 Types of Cold Thermal Storage Systems Based on storage media

There are three main storage media used for cold thermal storage. The choice of storage media has been discussed above. Ice, chilled water and eutectic salts are the three main storage media used for a cold thermal storage system.

2.2.1 Chilled Water CTES:

This system requires the largest storage tank. It is not a complex storage media to use when compared to the other media choices. The use of chilled water is most economical for systems greater than 2,000 ton-hrs capacity. This system uses the sensible heat capacity of water (1 BTU/lbm-^{o}F) to store energy, it does not involve water changing phase. The typical operating temperature range is 5-10^{o}F. It is estimated that a chilled water system will require 6-7 times the storage capacity of a similar ice system.

A chilled water system works in harmony with typical industrial chillers and regular heat exchangers without the need for modifications. To minimize the size of the storage tank, it is essential that the difference between the cooling water supply and return temperatures are maximized as this maximizes the energy storage capacity of water. Chilled water may be stored in a variety of tank arrangements for this system. It may be stored in tank arranged in series, parallel or stratified tanks.

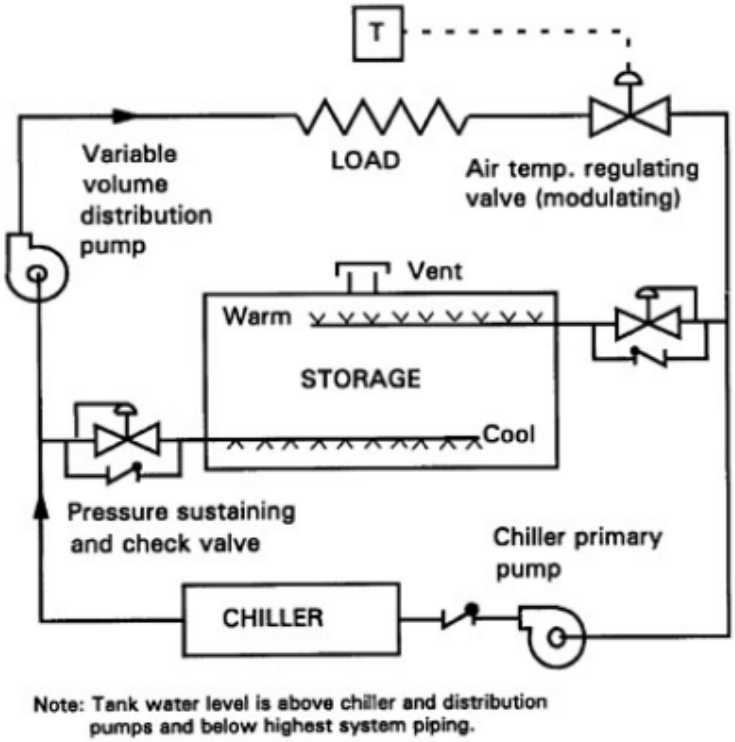

Figure 2.5: Schematic of a Chilled water thermal Storage system [4]

The series arrangement is the least complex. It requires that the chilled water storage tanks be placed in series. When the system chiller provides more chilled water than is needed, the chilled water is diverted to the tanks in series where it displaces the warmer water from

cooling load. Conversely, when more chilled water is needed than has been provided by the chiller, warm water returning from cooling load is used to displace chilled water in the tank.

Parallel storage tanks require that the storage tanks be connected in parallel with each tank having individual drain and fill valves. During the charging cycle, some tanks are left empty while others are filled with chilled water. When discharging begins, the fill valves on the empty tanks are opened to allow for return warm water from the cooling, drain valves on the initially filled tanks are also opened as the discharge progresses.

A stratified storage tank uses a single tank to store both chilled water and the warm water from cooling load. Since water stratifies because of its change in density with respect to temperature, the cold chilled water will sit at the bottom of the tank while warm return water sits at the top of the tank. By using a “special” piping network water in and out of the storage tank, mixing does not occur allowing the formation of a boundary layer between the warm and cold water as shown in Figure 2.5.

2.2.2 Eutectic salts for CTES:

These are phase change materials suitable for cold thermal storage. They undergo phase changes at temperatures of about 47^{o}F and typically change phase from liquid to solid. They absorb and release large amounts of energy during phase change. Eutectic salts are mixtures of inorganic salts and additives. The salts are sealed in plastic wrappings then placed in the storage tank with water or other working fluid flowing around them in the same tank.

Eutectic salts in a thermal storage tank perform the same functions as ice balls. As they release heat to the working fluids around them, they change to solids. The chiller only needs to reduce the temperature of the working fluid to about 47^{o}F and as such the chiller is able to operate more efficiently. This system is considered when space available for storage tank is limited and low temperatures are not required.

2.2.3 Ice CTES:

Ice CTES systems are made up of a heat exchanger, ice making system and a storage tank. They may be classified into static or dynamic systems depending on how ice is delivered to the storage tank.

Static systems:

Ice is produced inside the storage tank by an ice-making coil inside the storage tank. Ice may either be produced inside this coil (Ice-in-tube), outside the coil (ice-on-coil).

Ice-on pipe CTES produce ice by pumping very cold refrigerant through an array of pipes immersed in a tank of water. The ice produced from this system forms on the exterior surface of the pipe or coil. Bubbles are allowed to flow around the pipes in order to promote dense and even ice formation during the charging cycle and uniform melting during the discharge cycle. This system requires that the chiller use a complex liquid-overfeed system.

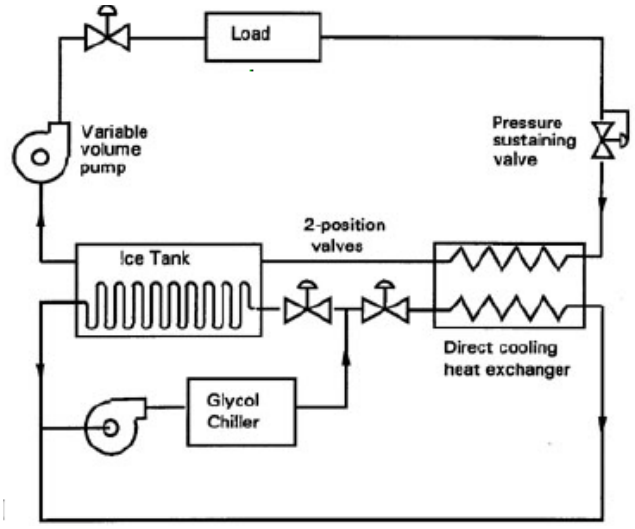

Figure 2.6: Schematic of a Ice-on pipe thermal Storage system [4]

Dynamic systems:

Ice is produced outside the storage tank and removed from the ice making surface periodically. The removed ice is transferred to the storage tank where it may undergo melting usually by feeding the storage tank with warm water from the load.

Ice harvester system:

Ice is produced by a single component called the ice harvester. The harvester is installed above an open tank where it creates ice, which falls into the storage tank capable of storing cold water and flakes of ice. Water is pumped from the bottom of the storage tank and is used to make the ice. The storage tank is discharged by pumping water from the tank to meet the cooling load. Warm return flows into the ice-water mixture in the storage tank to speed up heat transfer between water and ice.

Ice harvesting systems are expensive and are ideal for applications that require relatively low refrigeration capacity.

Ice – Slurry CTES systems:

Ice-slurry is a crystallized water based ice solution, it can be pumped and offers secondary cooling medium for CTES. It is pumped like a typical chilled water system but has a higher energy storage capacity. Ice slurry systems do not suffer from efficiency losses in the defrosting cycle of ice harvesters. Slush is a mixture of small ice particles in a solution of glycol and water. To discharge the slush that has been pumped into the storage tank from an ice slurry generator, a heat exchanger would be required in order to isolate the cooling load from the ice slurry system. Warm solution returning from the cooling load is sprayed on the surface of the cold slurry solution in the storage tank. It is the preferred system when high storage capacity is needed.

2.3 Exergy Analysis of a CTES system

The Exergy of an energy form or a substance is a measure of its usefulness or quality to cause change and provide the basis for an effective measure of the potential of a substance and or energy form to impact the environment [3]. Exergy is based on the second law of thermodynamics. The Kelvin – Planck statement states that it is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work [5].

The law basically states that the entropy of an isolated system not in equilibrium will tend to increase over time till it reaches a maximum value at equilibrium.

An exergy analysis of a cold thermal storage system allows for the accounting of how the energy in the system is used. In the case of an energy analysis, the first law of thermodynamics is put to play and as such accounts only for the transfer of energy. The specific exergy (∈) for a closed system is defined as [6]

$$\epsilon = KE + PE + (u - u_r) + Pr(v - v_r) - T_r(s - s_r) + \epsilon_{ch} \quad (2.1)$$

For an open system the specific exergy can be written as

$$\epsilon = KE + PE + (H - H_r) - T_r(s - s_r) + \epsilon_{ch} \quad (2.2)$$

Where,

Є = specific exergy	r = properties of the reference environment.
PE = Potential energy	H = enthalpy.
KE = specific kinetic energy	ϵ_{ch} = specific chemical exergy
u = Internal energy	P = Pressure
s = specific entropy	T = temperature

The exergy of a system is conserved only for reversible or ideal processes. Thermal storage systems apply natural laws in the transfer of energy and as such undergo an irreversible process till a thermodynamic equilibrium is attained.

Since thermal energy storage systems deal primarily on energy conservation principles, a very efficient TES system must therefore be able to transfer as much energy that has been stored up to do work with little losses to the environment. That is, a thermal storage system must be as close to an ideal reversible system as possible. While it is true that energy losses can be accounted for, the degrading of energy quality can only be accessed through an exergy analysis.

2.3.1 EXERGY ANALYSIS

To perform an exergy analysis, mass and energy balances must first be evaluated. In the proceding analysis, data values will not be used for any calculations; instead the steps taken to analyze a thermal storage system will be shown. The analysis below is for a cold thermal energy storage system. Methods of determining exergy efficiencies will also be accounted for.

Mass Balance

Mass input - Mass output = Mass accumulation (2.3)

For a non- steady state flow process, the mass balance can be written as:

$$\sum_i m_i - \sum_e m_e = m_2 - m_1 \quad (2.4)$$

Energy Balance

Considering a Cold thermal storage unit,

$$Q_t \quad [Q_{rec} \mid Q_l] - \quad u_c \qquad (2.5)$$

From Equation 2.5, Q_t is the heat removed from the material to be frozen in the storage tank. It is this heat removal by the heat transfer fluid (water/glycol solution) that facilitates freezing during the charging cycle. Q_{rec} is the heat removed from the heat transfer fluid by the material (water or salts) in the storage tank. Q_l is the total heat gained by the storage media from the environment possibly due to poor insulation. u_c is the reduction in internal energy of the storage media for the complete charging and discharging cycles.

The energy balance can also be written as:

$$\dot{Q}_s = \dot{Q}_{load} + \dot{Q}_{loss} - \dot{Q}_{chil} \qquad (2.6)$$

From equation 4 above, $\dot{Q}_s$ is the rate at which energy is stored in the storage tank during the charge cycle. $\dot{Q}_{load}$ is the cooling load rate as required by air-conditioning or other industrial cooling process $\dot{Q}_{loss}$ is the rate at which the storage tank looses heat to the environment. $\dot{Q}_{chil}$ is the heat removed by the chiller.

The energy transferred by the heat transfer fluid during charging

$$Q_i = mc_p\Delta T \tag{2.7}$$

and is related to $\dot{Q}_s$ since,

$$\dot{Q}_s = \frac{Q_i}{t} \tag{2.7.1}$$

Where, t = time.

$$\dot{Q}_{loss} = UA(T_{outside} - T_{tank}) \tag{2.8}$$

Where, UA is the heat transfer coefficient for the tank insulation, T is the temperature.

The rate of heat loss from the chiller is

$$\dot{Q}_{chill} = \dot{m}_r(h_d - h_c) \tag{2.9}$$

Where $\dot{m}_r$ is the refrigerant mass flow rate, h_d and h_c are the heat transfer fluid enthalpy at the end of the discharge cycle and at the beginning of the charging cycle respectively.

Exergy Balance

The exergy balance equation is similar to the energy balance equation given in equation 3 above and is given as [6]:

$$\epsilon_i - [\epsilon_{rec} + X_l] - I = \Delta\Xi \tag{2.10}$$

In terms of work, for the CTES system which is a closed system,

$$\sum_o (X_o)_{1,2} - (W_x)_{1,2} - I_{1,2} = 0 \qquad (2.11)$$

$(W_x)_{1,2}$ = work done by the system from charging up the storage tank to discharging.

$(X_o)_{1,2}$ = flow of exergy from start of the charge cycle to end of the discharge cycle.

X_l = exergy loss and is related to the heat loss by the storage tank.

ϵ_i = net exergy input and is based on the exergy of the heat transfer fluid from the charging cycle through storage to the discharge cycle.

ϵ_{rec} = net exergy recovered which when added to the exergy loss accounts for the net exergy output.

$\Delta \Xi$ = exergy accumulation.

The exergy consumption can be expressed as [7]:

$$I - T_r \Pi \qquad (2.12)$$

Equation 2.12 above characterizes the thermodynamic irreversibility in the system and shows that exergy consumption is proportional to entropy creation. Equation 2.13 is given in rates.

$$I = T_r \dot{S}_g \qquad (2.13)$$

Equation 2.13 is called the Gouy-Stodola relation [6]. Where $\dot{S}_g$ is the entropy generation rate.

Exergy into the storage tank during the charging cycle is given as:

$$\epsilon_i = (H_1 - H_2) - T_{ref}(S_1 - S_2) \quad (2.14)$$

Exergy out of the storage cycle during the discharge cycle is given as:

$$\epsilon_{rec} = (H_4 - H_3) - T_{ref}(S_4 - S_3) \quad (2.15)$$

From the above set of equations the subscripts 1,2 represent the inlet and outlet of the charging cycle respectively. The subscripts 3,4 represent the inlet and outlet of the discharge cycle respectively.

Recall that exergy is a means of measuring the degradation of energy during the energy transfer process, especially during the storing process. The exergy loss from the storage process can be determined from:

$$X_{l,f} = \left(1 - \frac{T_r}{T_j}\right) Q_{l,f} \quad (2.16)$$

T_r = temperature of the reference environment

T_j = mean storage tank temperature.

$Q_{l,f}$ = heat loss to the environment and can be obtained from equation 6.

Cycle Efficiency

For a cold thermal energy storage system, the overall energy efficiency is defined as [3]:

$$\eta = 1 - \frac{Energy\ loss}{Energy\ input} \tag{2.17}$$

The overall exergy efficiency for a CTES system is given as:

$$\eta = 1 - \frac{Total\ exergy\ destruction\ rate\ of\ overall\ system}{Total\ exergy\ input\ rate\ into\ CTES\ system} \tag{2.18}$$

And can also be written as:

$$\psi = \frac{Exergy\ recovered\ from\ CTES\ duringdischarge}{Exergy\ input\ to\ CTES\ during\ charging} \tag{2.19}$$

The charging cycle energy efficiency is given as:

$$\eta_c = \frac{Energy\ accumumlation\ in\ CTES\ during\ charging}{Energy\ input\ to\ CTES\ during\ charging} \tag{2.20}$$

The charging cycle exergy efficiency is given as:

$$\psi_c = \frac{Exergy\ accumumlation\ in\ CTES\ during\ charging}{Exergy\ input\ to\ CTES\ from\ charging} \tag{2.21}$$

The discharge cycle energy efficiency is:

$$\eta_d = \frac{\textit{Energy recovered from CTES during discharge}}{\textit{Energy input to CTES during charging}} \tag{2.22}$$

The discharge cycle exergy efficiency is:

$$\psi_d = \frac{\textit{Exergy recovered from storage tank during discharge}}{\textit{Energy accumulation in CTES during charging and storing}} \tag{2.23}$$

The exergy efficiency during storing is given as:

$$\psi_c = \frac{\textit{Exergy accumumlation in CTES during charging and storing}}{\textit{Exergy accumulation in CTES during charging}} \tag{2.24}$$

Chapter 3 Air Conditioning

Air conditioning as the name implies is the modification of air to meet specified requirements. It refers to the control of the temperature, moisture content, cleanliness and circulation of air in a particular location. Depending on the type of air-conditioning system, building type, cost and medium of energy transfer (usually air or water), air may be conditioned on site, pre-conditioned and then transferred to the space in need of the conditioned air or both.

Space to be conditioned may require heating, cooling, humidification or dehumidification, and cleaning and as such “air conditioners” are able to perform all of this functions as dictated by the requirements of the space to be conditioned.

Heating: The heating of a space is performed for two main reasons - either to increase the temperature of the air in the conditioned space or to replace the energy lost by the space to its surroundings so as to maintain the temperature of the conditioned space. Hot gases from a furnace may be used to heat air directly before been sent to the conditioned space. Hot water or steam from boilers may be piped through heating coils located in ductwork or air handlers. Heated air is transferred back to the conditioned space [8].

The rate of sensible heat transfer from the heating coils to the air stream is given by Equation 3.1.

$$\dot{q}_s = \dot{m} c_p (t_e - t_i) = \frac{\dot{Q} c_p}{v} (t_e - t_i) \qquad (3.1)$$

Where,

$\dot{q}_s$ = rate of sensible heat transfer

$\dot{m}$ = mass rate of air flow

c_p = constant –pressure specific heat of air

t_e = Temperature of airstream out

t_i = Temperature of airstream in

v = specific volume of air

$\dot{Q}$ = volume flow rate of air

$\dot{Q}$ = volume flow rate of air

Cooling: Cooling is the removal of energy from a space to be conditioned in order to make up for the energy gained by that space. Air to be cooled is circulated through a heat exchanger where chilled water or other working fluids already circulating through the same heat exchanger absorb energy from the air and transfers it to the refrigeration system. The cooling of a conditioned space is done mostly for the same reason as heating. It is done to reduce or maintain the temperature of a space.

During the cooling of a space, cool air is introduced while an equal amount of warm air is removed from the space. Energy from the warm air leaving the conditioned space is rejected to the atmosphere through the use of a refrigerant system like a chiller.

Cooling can be done by a direct expansion system for smaller cooling loads or by means of a chilled water system for higher cooling capacity of 100 or more tons.

Dehumidification: This is the removal of water vapor from an airstream entering a conditioned space in order to maintain the humidity levels of the space. Dehumidification occurs in heat exchanger coils during the cooling process by condensation. The liquid water from condensation is then drained out of the heat exchanger. In a similar manner, humidification can be done by injecting water vapor into the airstream through the use of humidifiers or wet plates or mats. Humidification is usually done in winter when there's less water vapor in airstream. Energy required for moisture content removal is called latent cooling.

The latent energy transferred in a humidifying or dehumidifying process is:

$$\dot{q}_l = \dot{m}_w h_{fg} \quad (3.2)$$

Where,

$\dot{q}_l$ = latent heat rate (it is positive for humidification and negative for dehumidification)

$\dot{m}_w$ = rate of water vaporization or condensation

h_{fg} = enthalpy of vaporization for water vapor

Cleaning: The cleaning of the airstream is done to improve air quality and reduce the amount of contaminants in the conditioned space. It is done with the use of filters placed in air handling units which stop solid particulates from entering the conditioned space.

3.1 Methods of cooling

Most manufacturing plants typically install cold thermal storage system for the purpose of reducing on-peak demand loads. There are generally two methods by which cooling may be accomplished, cooling by direct expansion of a refrigerant and chilled water systems. The method of cooling is mostly dictated by the cooling load or the cooling capacity required by the manufacturing plant and of course the cost of implementing each system. Direct expansion systems are small with a small cooling capacity of about 50 tons or less per unit. In large facilities, many units must be installed to meet the desired cooling loads. They are cheaper to install per unit but have short life spans of about 15 years. The chilled water system is more expensive to implement but has a greater cooling capacity per unit. They are less expensive to maintain, have a greater life span and only one system need be in place to meet a plants' cooling load.

3.2 Direct Expansion Systems

A direct expansion system is made up of four main parts, the compressor, the condenser, the evaporator and the expansion valve.

Compressor: A compressor is a mechanical device used to increase the pressure of a gas while reducing its volume. In a direct expansion system, refrigerant which is typically the working fluid enters the compressor as a low temperature gas. Upon compression, it leaves the compressor as a high pressure, high temperature gas as it heads towards the condenser. The compressor is typically located outside of the conditioned space.

Condenser: A condenser is a device used to condense a vapor to its liquid state. The refrigerant enters the condenser as a high temperature and high pressured gas and leaves as a low temperature high pressure liquid. The condenser for an air cooled system is a heat exchanger to remove heat from the refrigerant through the use of heat transfer fins with fans blowing cool outside air through the fins. The condenser is usually located outside of the conditioned space.

Evaporator: Liquid refrigerant leaves the condenser as a high pressure low temperature fluid as it heads for the evaporator. An evaporator is a heat exchanger that facilitates the refrigerants phase change from liquid to vapor.

Expansion Valve: High pressure liquid refrigerant enters the expansion valve where it expands and returns to a vapor. Upon the expansion of the fluid, it absorbs heat from air in the room around it while losing some of the heat to the atmosphere at the same time since the evaporator acts as a heat sink. Refrigerant leaves the evaporator as a cool gas and heads for the compressor where the cycle is repeated.

A schematic showing a direct expansion air conditioning system is shown in Figure 3.1.

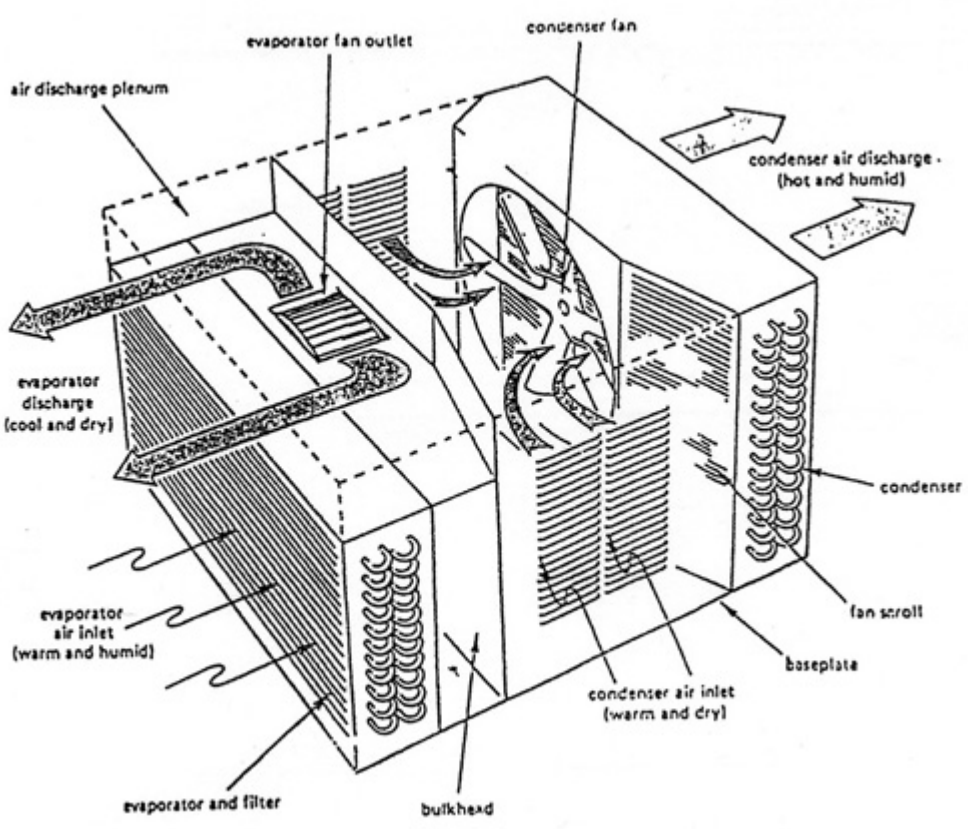

Figure 3.1: Direct Expansion Air conditioning system [9]

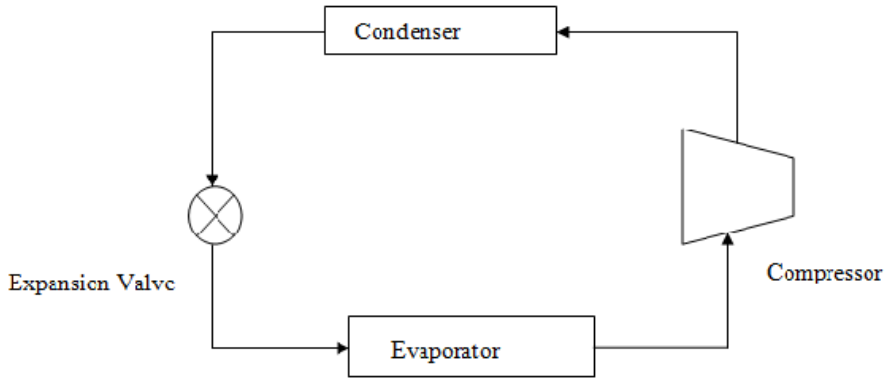

Figure 3.2: Direct Expansion Cycle

3.3 Chilled water system

A chilled water system is a better alternative to a direct expansion system when cooling capacity needed is very high, especially in manufacturing plants. Although more expensive to implement than the smaller direct expansion system, it generally tends to last longer if well maintained, is a lot more efficient and less expensive to maintain. The chilled water system can replace direct expansion systems with some modifications. By direct replacement, the refrigerant gas cooling coil and a few controls need only be changed in the air handling unit when installing a chilled water system. The refrigeration cycle used for the chilled water system needs to be in a central location relative to the air handling units. The direct expansion cooling coil is replaced with chilled water cooling coil while the rest of the heating coils, filters and humidifiers and dehumidifiers in the air handler remain untouched.

A chilled water system unlike the direct expansion system is made up of the chiller, the water tower, centrifugal pumps and a chilled water cooling coil.

Chiller: A chiller is a device used to remove energy in the form of heat from a working fluid (usually water or glycol) through a vapor – compression cycle or absorption-refrigeration cycle. In a chilled water system, water is used to remove energy from a conditioned space. The energy transferred to water from the conditioned space is removed by the chiller. The refrigeration cycle by which the chiller works is much the same as the direct expansion system previously mentioned, the only difference is that the chiller houses all components of the system and heat is removed from water to be chilled rather than from air around the evaporator. Water cooled in this manner leaves the chiller as chilled water and is pumped to the chilled water cooling coil in the air handler.

Chillers may be air cooled or water cooled. Air cooled chillers force air from the outside surroundings to cool the condenser in the chiller while water cooled chillers use water to cool the condenser. Water used in cooling the condenser is circulated to a cooling tower where it is able to release the heat absorbed from the refrigerant to the outside by evaporation. Water cooled chillers are more efficient than air cooled chillers, they require a smaller condenser size and ultimately a smaller chiller; since fans for cooling the condenser are eliminated, the noise level of the unit is also significantly reduced. A typical chiller used in industry is shown below.

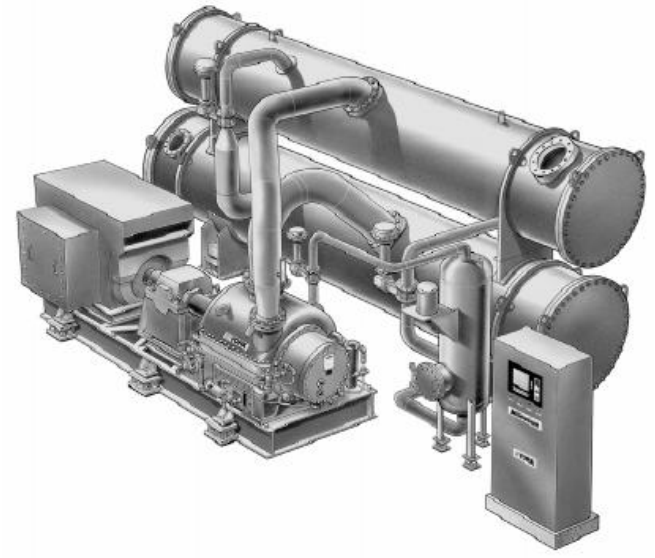

Figure 3.3 York Chiller[10]

Cooling tower: A cooling tower is one of the most important parts of a water cooled chiller system. It is the unit responsible for the ultimate removal of heat from a plant's process to the atmosphere. The process in the case of an HVAC system is the heat removed from the conditioned space. Heat is removed from the condenser through which the refrigerant is flowing by either a dry means or by a wet means depending tower type.

In a Wet type cooling tower, water is sprayed through fins onto the condenser bundle though which water is flowing, as water on the condenser evaporates, it cools the water running through the condenser. A dry type cooling tower functions as an air cooled heat exchanger. Wet type cooling towers are typically more efficient and as such are the more used cooling towers in manufacturing industries. The systems described above are closed systems. Open systems are however more common in industry.

A chilled water system uses water as the working fluid. Air handling units typically have cooling coils installed in them, by pumping chilled water through this cooling coil and then using a fan to move the airstream through the cooling coils, the airstream can be cooled. Warm air removed from the room to be conditioned is moved through cooling coils in the air handling unit, the airstream loses its energy to the water in the coils and has a significant temperature drop before it is re-circulated back into the conditioned room. The energy saturated water leaves the cooling coil at a higher temperature than it initially had, it is then pumped through a chiller which is used to remove the energy in the water. The removed energy from the working fluid is then rejected to the atmosphere through the use of a cooling tower system or through the use of air cooled condensers. Water after losing its energy leaves the chiller at a lower temperature before its pumped back to the cooling coil where the process is repeated. A typical chilled water system uses centrifugal pumps to move water through it.

3.4 Air - Conditioning systems

There are three basic types of air conditioning systems based on how energy and air is distributed. These systems are selected based on building design and purpose of the system. The three systems are the All- air systems, Air and water system and all water system.

All-Air Systems: Energy and ventilating air are carried by duct work between the air-handling unit and the conditioned space. As the system can also be used for heating, energy and air may also be moved through ductwork from a furnace to the conditioned space. It is a system used in buildings with different areas in need of conditioning for example schools and offices. The different areas in need of conditioning can adjust the air conditions to meet their respective needs. This is a system that is also used for situations requiring close control of temperature and humidity as such it is the most commonly found air conditioning system in manufacturing industries. A simple schematic showing air and energy distribution in all-air systems is shown below.

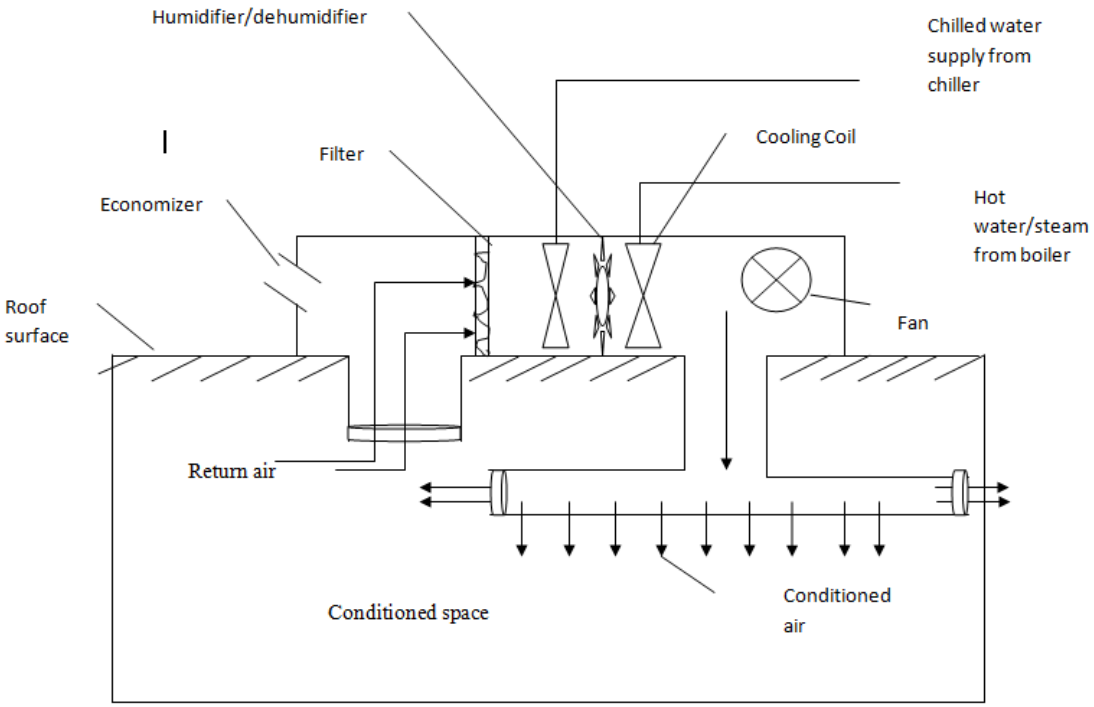

Figure 3.4: A schematic of a air-handling unit for an All-Air air conditioning system

There are various types of all- air system mostly based on how air is moved to and from the conditioned space. Some of the types of all air –air systems are Single Zone, Reheat, Variable air volume, Dual duct and Multizone systems.

Air and Water systems: This air conditioning system requires that both air and water be used for the conditioning of a space. Water in this case is used to remove sensible heat from the conditioned space while air is used to ventilate the space and removing moisture. Air may also be used to transfer energy from the conditioned space in addition to water. This system is particularly useful when the least amount of duct work is desired. Water which has a greater specific heat and density than air is able to transfer energy more effectively and the piping space required by water for the same cooling capacity is much less than duct work needed by air. The horsepower of the pump needed to move water for this system is much less than the fan horsepower that would otherwise have been needed. The air and water system is as such a very economical system that is applicable in large offices and schools as the system has a lower operating cost, initial cost and savings on space.

All –Water systems: In this system, unconditioned ventilation air is supplied by an opening in the wall or through infiltration. Cooling and dehumidification are typically done by circulating chilled water through coils in the room terminal. Heating may also be accomplished by circulating hot water or steam from a boiler through the same coil. Humidification of the conditioned space cannot be accomplished using this system. It is a system that provides great flexibility, is cost and space effective but may not meet indoor air quality standards.

3.5 Cooling Load Calculations

In order to calculate the cooling load requirements for a space in need of conditioning, some terminologies need to be understood.

Cooling load: This is the rate at which energy must be removed from a conditioned space in order to maintain the desired temperature and humidity of that space. Energy transfer into the conditioned space in the form of conduction and radiation are absorbed by object in that space, as such, only energy transferred in the form of convection adds energy to the room air and is a part of the cooling load. Similar but different heat transfer rates may also be considered.

Heat gain: This is the rate at which energy is transferred to or generated in a space. It has two components, sensible heat and latent heat that must be accounted for separately. Heat gain occurs in various forms such as solar radiation through windows, heat conduction through exterior walls and ceilings boundaries, ventilation and infiltration. Sensible and latent heat may also be generated in the space through activity.

Heat extraction rate: This is the rate at which energy is removed from a space by cooling and dehumidifying equipments. It is equal to the cooling load when the equipments are on and space conditions are constant.

Cooling Coil Load: Cooling coil load is the rate at which energy is removed at the cooling coil (present in the air handling unit) serving a conditioned space. It is equal to the summation of the instantaneous cooling load and the external load imposed on the system away from the conditioned space.

3.5.1 Methods of calculating space cooling load

There are several means of estimating space cooling loads. The three most common methods are Total Equivalent Temperature Differential method, Transfer Function method (TFM) and Cooling Load Temperature Differential method (CLTD).

Total Equivalent Temperature Differential Method: This method employs a time averaging technique to convert instantaneous total space heat gain to an instantaneous space cooling load.

Transfer Function Method: This method uses transfer functions as weighing factors in calculating space cooling load. It requires a computer analysis to predict the hourly cooling loads due to different types of walls, roofs and fenestration.

Cooling Load Temperature Differential Method: This method makes use of temperature differences for walls and roofs and cooling load factors (CLF) for solar gain and internal heat sources.

The CLTD and CLF are derived from the Transfer function method. They both vary with time and are functions of the environmental conditions and building parameters. CLTD also accounts thermal response due to radiation from interior surfaces of the wall and in the heat transfer through a wall or roof. The CLTD method will typically require the use of a computer to estimate the hourly cooling loads for the day. With this method, the cooling load may be estimated as follows [11]:

$$\text{Cooling Load} = \Sigma(U \times \text{Area} \times CLTD_{cor}) + \text{Internal Load} + \text{Infiltration}$$

Where,

U	=	Overall thermal conductivity of a surface, BTU/hr-ft^2-°F
Area	=	Area of a surface, ft^2
$CLTD_{cor}$	=	Corrected CLTD value for a surface calculated below
Internal Load	=	Cooling load due to operation of equipment, people, etc, BTU/hr
Infiltration	=	Cooling load due to air infiltrating into the space, BTU/hr

The corrected CLTD value is found using:

$$CLTD_{cor} = CLTD + A + B + C$$

Where,

$CLTD_{cor}$	=	CLTD corrected for outdoor temperature, indoor temperature and solar angle
CLTD	=	Cooling load temperature difference for an outdoor temperature of $85^{o}F$, an indoor temperature of $78^{o}F$, for a particular construction at a particular hour of the day, ^{o}F
A	=	Outdoor temperature correction, (T_{out} - 85)
B	=	Indoor temperature correction (78 - T_{in})
C	=	Latitude and month correction, °F from tabulated values

The values for CLTD and C are found in [12]. The CLTD value is then corrected by applying correction factors, A, B, and C. The corrected CLTD value accounts for conduction between the outdoors and the inside, solar loads, and for the thermal mass of the building construction itself.

CHAPTER 4 Electric Billing and Rate Schedules

A thermal storage system is one installed for the purpose of cost and demand savings. Without careful considerations of the billing rates and the rate schedules to which a manufacturing facility is tied to, the thermal storage system may have an opposite effect and cost the facility more in its installation and operation. As will be observed from the rate schedules discussed below, a manufacturing facility that runs more than two shifts and is under the Time of Use rate schedule will get the most benefit when it comes to installing a thermal storage system. The time of use rate schedule clearly benefits a facility that can transfer as much of its operation to off-peak times when energy charges are significantly lower.

From the discussion on air-conditioning, we know that in the summer, a manufacturing facility will generally have its peak demand during the day when its air conditioners are in full operation. By producing the required cooling at night and running a cold thermal storage system, the facility can clearly save.

Under the General rate schedule, there would not be any cost savings in installing a thermal storage system seeing as the rates are relatively flat. In fact for both of these rate schedules installing thermal storage system will increase cost because the energy charge would be

rather high during a charging cycle which may off-set any additional savings from when the air-conditioning system is not running.

The choice of power supplier is typically dictated by the location of the manufacturing plant. Plants located to the east in towns like Rocky Mount, New Bern and Tarboro are served by North Carolina Eastern Municipal Power Agency (NCEMPA) while towns facilities located to the west are served by North Carolina Municipal Power Agency Number 1 (NCMPA1). NCEMPA have part ownership of 5 generating plants (2 coal and 3 nuclear) and buys only 32% of its power needs from Progress energy an investor owned utility. NCMPA1 has part ownership of Catawba nuclear plant operated by Duke Energy and has about 34 diesel generators tied to the grid to meet high demands and offset high prices [18].

North Carolina's Electric Cooperatives are made up of 26 private, independent, not for profit electric cooperatives. They are owned by the consumers they serve. NC co-ops are made up of distribution cooperatives and generation and transmission cooperatives. Distribution cooperatives are supplied by generating and transmission cooperatives. They may also buy power from investor owned utilities before distributing it to their members. Unlike investor owned utilities that provide power for profit maximization, NC co-ops operate to provide at-cost electric service to their consumer members. A co-op's net margin goes to the individual consumer member of the co-op and not the utility.

As manufacturing industries may require a thousand kilowatts or more for their operation, power companies are required to have generating capacities capable of meeting a facility's maximum load even if the maximum load occurs for only a few hours a day. Peak loads may occur rarely, but when they do occur, the power generating companies must meet the excess load requirement by turning on expensive power generation equipments such as natural gas turbines. Natural gas turbines are an expensive power generation system when compared to nuclear and coal and when turned on to meet unexpected peak loads, the cost of power generation is increased.

For the purpose of this paper, rate schedules according to Duke Power will be used for all calculations. For facilities serviced by Duke Power, there are two different rate schedules that service manufacturing facilities in North Carolina. They are the General Industrial Schedule and The Optional Power Service, Time of Use (OPT) schedule. To better understand this rate schedules, a few terminologies must first be understood [13].

Power Factor: This is a measurement of the phase relationship between voltages and current in an electric system. This makes power companies supply more power than is registered on meters. Low power factors can also reduce the efficiency of an entire electrical system thus making utilities charge extra for facilities with low power factors. It is typically caused by electric motors which draw current for inductive loads and may be rectified by the use of capacitors.

Peak Demand: This is the largest of 50% of the contract demand, the maximum integrated 30 minutes for demand during the summer or winter on-peak period for which the month lies and 15 kilowatts. It is an essential part of the monthly bill.

Load factor: This is the ratio of the kilowatt-hours (kWhs) to the maximum demand (kW) times the number of hours in the billing month.

Demand Charge: This is the electric utility charge, usually based on the highest energy consumption per demand interval each month.

4.1Rate Schedules

4.1.1 General Industrial Rate schedule

This is an industrial service schedule [14] available to businesses classified as manufacturing industries by the Standard Industrial classification Manual. The total bill paid by a facility for a month is made up of the demand charge, the energy charge, basic facilities charge, taxes and other charges such as fuel riders.

According to this rate schedule, the billing Demand is the largest of –

- The Maximum integrated 30- minute demand measured during the month for which the bill was rendered.

- 50% of the maximum integrated 30- minute demand during the billing month of June-September, within the previous 12 months and including the month for which the bill is rendered.
- 50% of the contract demand.
- 30kW.

For the General rate schedule, the energy charge is the total energy used in the specified month multiplied by an energy charge rate. The energy charge is an integral part of the monthly electric bill.

4.1.2 Optional Power Service, Time of Use (OPT) schedule

The Time of Use rate schedule is an industrial service schedule available to businesses classified as manufacturing industries by the Standard Industrial Classification Manual. It is generally more expensive to make power during the day when demand is high than at night when demand is lower. A time of use bill consist of the billing demand, energy charge and facilities charge.

The demand charges are made up of the on peak demand charge and the economy demand charge for both the summer months (June 1 – September 30) and the winter months (October 1- May 31). The peak demand charge is less in the winter months than in the summer months, but the economy demand charge is the same for both winter and summer months.

For the summer months, the first 2000 kW of billing demand is charged at $10.9490 per kW, the next 3000 kW is charged at $10.0296 per kW and for 5000 kW and over billing demand is charged at $9.1017 per kW. In a similar manner, the demand charge for winter months for first 2000 kW, 3000 kW and 5000 kW and over are charged at $6.4446 per kW, $5.5167 per kW and $4.5804 per kW respectively.

This rate schedule further comprises of the energy charge. The energy charges are based on both all - on peak energy per month and all off - peak energy per month. On-peak periods are in the summer between 1:00 pm and 9:00 pm Monday to Friday and 6 am to 1:00 pm Monday to Friday in the winter. Off-peak are all other weekday hours, Saturday, Sunday and holidays. Summer and winter months are defined in the same way as for demand. The energy charge for a TOU rate schedule is made up of the on-peak energy charge which is the total amount of energy used during the on-peak periods of the month multiplied by an on-peak energy charge rate. In a similar manner, the off-peak energy charge is the total off-peak energy used multiplied by the off -peak energy charge rate. The total energy charge is the summation of the on-peak and off-peak energy charge. The on-peak energy charge is $0.046646 per kWh and off-peak energy charge is $0.028629 per kWh.

Contract Demand: The power company requires that the manufacturing facility specify the maximum demand that they require of the power company. This may then be used to determine the billing demand.

For the Time of Use Rate schedule, the total bill paid for by a facility for a month is made up of the demand charge, energy charge, the basic facilities charge and taxes.

There are two sections to the billing demand as previously mentioned. The on-peak billing demand and the economy demand. The on-peak billing demand each month is the largest of –

- The Maximum integrated 30- minute demand during the applicable summer or winter on-peak period for the month in which the bill was accessed.
- 50% of the contract demand or 50% of the on-peak contract demand.
- 15kW.

Economy demand: To determine the economy demand, the maximum integrated 30 minute demand during the entire month (day and night) is compared to on-peak billing demand determined from above. If the maximum demand exceeds the on-peak billing demand, the difference between them is the economy demand. The economy demand charge in the summer and winter months is $0.8688 per kW.

Benefits of the time of use rate schedule are:

1. Total cost of operation may be reduced by load shifting. That is energy demanding operations are scheduled during off-peak periods when demand charge is a cheaper.

2. It supports and rewards facilities that may use load shifting systems such as facilities operating thermal storage devices which generate cooling capacity at night for use during on-peak periods.
3. Time of Use rate schedules benefits facilities that run three or more shifts. Facilities that can run 7 days a week and manufacture at night get the most reward from this rate schedule.

Chapter 5 Feasibility Studies

The two cases discussed below represent actual case studies performed by the Industrial Assessment Center at NCSU. The results obtained from these case studies are based on data obtained on site at the facilities visited. The results obtained are by no means “perfectly accurate” and only represent estimations as to savings that could be made should the recommended actions be considered.

5.1 Case 1: Thermal storage for processing CO2

This case pertains to a facility that compresses CO_2 gas to make liquid CO_2 and dry ice. The facility currently buys waste CO_2 gas from a nearby plant. The CO_2 gas is piped in at a temperature of 95^oF and is reduced to 49^oF in the inlet cooler before going to the first stage CO_2 compressor. The inlet cooler currently uses ammonia to cool CO_2. Figure 5.1 shows a part of the facilities process flow.

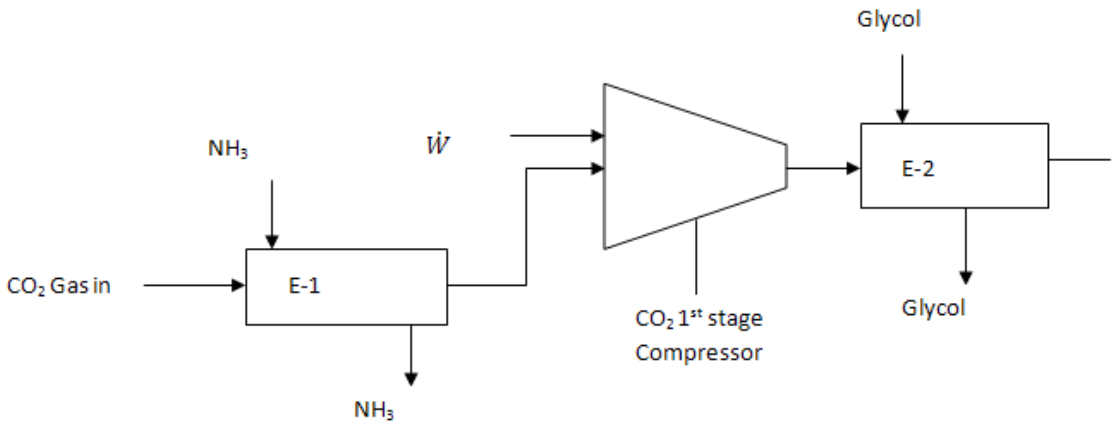

Figure 5.1: Partial Process Flow Chart (Current system)

The installation of a thermal storage system will replace ammonia cooling of CO_2, cooling the gas to around 34^{o}F. CO_2 entering the CO_2 compressor at a lower temperature will result in reduction in power demand by the compressor. A cold thermal storage system in place, on the process line will reduce the load on the ammonia refrigeration system during on peak periods. This is made possible by running cold glycol from the cold storage tank to cool CO_2 in place of ammonia that was previously been used. Figure 5.2 shows the layout of the proposed process at the facility.

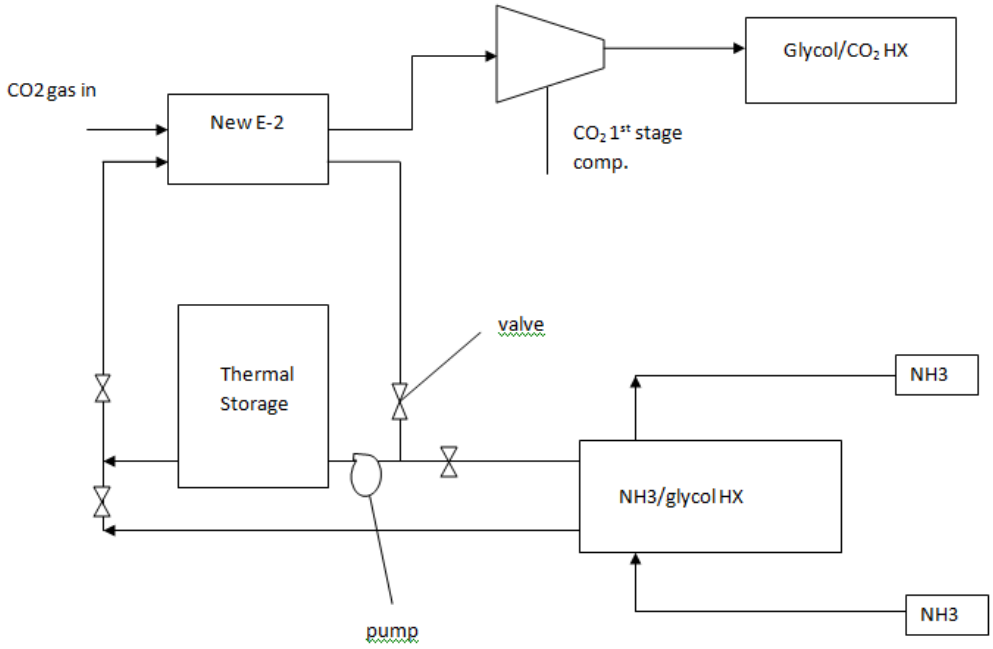

Figure 5.2: Partial Process Flow Chart (Proposed system)

Anticipated Savings

Table 5.1 Demand and Energy rates at the facility

Demand	$13.98kW
Energy	$0.04222/kWh

Unlike the standard thermal storage system, an additional chiller will not be necessary for this system since an ammonia refrigeration cycle is in use on site. The thermal storage system can readily use this refrigeration cycle to create ice at night during off peak periods.

The first heat exchanger is a glycol to ammonia heat exchanger and the second heat exchanger for glycol to CO_2. Ammonia will absorb heat from glycol and cool the solution down to 25^{o}F. The cooled glycol solution will freeze the ice balls as it is pumped over them. This all takes place at night during off peak periods to reduce cost.

During on peak periods, the glycol solution is pumped to a glycol to CO_2 heat exchanger and absorbs heat from the CO_2 entering the facility. The cool CO_2 leaves the heat exchanger at about 34^{o}F as it heads for the CO_2 compressor. The glycol solution is circulated back through the tank. Rather than operating the ammonia compressor to keep the glycol cool, the ice in the thermal storage media absorbs the heat throughout the day. The following evening, the ammonia refrigeration cycle goes back to freezing the ice and the cycle begins again. The savings that may be obtained by following this plan of action are calculated below.

The mass flow rate of CO_2 gas entering the facility is 80,000 lbm/hr at a temperature of 95°F. At full load, the current heat transfer, for design purposes, will be:

$$\dot{Q} = 80{,}000 \text{ lbm/hr} \times 0.203 \text{ BTU/lbm-°F} \times (95 - 49\text{°F})$$

$$= 747{,}040 \text{ BTU/hr} \times (1 \text{ ton}/ 12{,}000 \text{ BTU/hr})$$

$$= 62.3 \text{ tons}$$

With the installation of a thermal storage system, the heat transfer rate is:

$\dot{Q}$ = 80,000 lbm/hr x 0.203 BTU/lbm-°F x (95 – 34°F)

= 990,640 BTU/hr x (1 ton/ 12,000 BTU/hr)

= 82.5 tons

The ammonia system has an estimated efficiency of about 0.6 kW/ton [17].

The current on-peak demand by the ammonia compressor is:

Current Demand = 62.3 tons x 0.6 kW/ton

= 37.4 kW

With the installation of a thermal storage system, we will assume the ammonia compressor can be shut off during on-peak periods and as such will not provide 62.3 tons of cooling to CO_2. The cooling of CO_2 will be provided for by glycol from the cold storage.

This is, however, not the actual demand savings. When screw compressors unload, they do so inefficiently. On a typical screw compressor, reducing the load by 3% will only result in a 1% energy savings. These savings are only expected to be about $5/kW. We will neglect these savings, to be conservative.

As mentioned above, reducing the temperature of the CO_2 gas entering the compressor will reduce the compressor power. The following relation will quantify this.

<u>CO_2 compressor:</u>

Isentropic work into the compressor per unit mass is given as [5]:

$$\text{W comp, in} = \frac{kRT_1}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{(k-1)/k} - 1\right]$$

From the above equation,

k = specific heat ratio

R = ideal gas constant

T_1 = Temperature of CO_2 into the compressor

T_2 = Temperature of CO_2 out of the compressor

P_1 = Pressure of CO_2 into the compressor

P_2 = Pressure of CO_2 out of the compressor

The ratio of the work into the compressor at T_1= 34^{o}F with thermal storage present, and at T_1 = 49^{o}F in the absence of thermal storage is used to determine the energy savings. With all other values remaining constant in the equation for work into the compressor, the percent saving can be calculated as:

Percent savings = [1 – ratio of the absolute temperatures] x100

$$= \left[1 - \left(\frac{34^{o}\text{F} + 459.67\text{R}}{49^{o}\text{F} + 459.67\text{R}}\right)\right] \text{x } 100$$

= 2.95 %

CO_2 is compressed by two 850 hp compressors. The demand reduction on the compressor is:

Demand Reduction = 2 x 850 hp x 0.0295 x 0.746 kW

Current Operating Condition = 37.412 kW

The CO_2 compressor operated for 8,600 hr/yr.

Energy Savings = 37.412 kW x 8,600 hr/yr

= 321,743.2 kWh/yr

CO_2 Comp Cost Savings = 321,743.2 kWh/yr x \$0.04222 /kWh

+ 37.412 kW x 12 months x \$13.98 /kW

= \$19,860.24/yr

There are three other factors considered to approximately balance out one another: Thermal losses from the storage tank, increased compressor work to drive ammonia to lower temperatures for freezing thermal storage, yet lower condenser temperatures (at night) causing for more efficient refrigeration cycle.

Thermal losses from the storage tank will be calculated by assuming a steady state conduction system. The storage tank is assumed to be made of steel with half inch thick cellular glass insulation. By using 3E Plus Insulation Thickness computer program, the heat loss from the storage tank is 204,400 BTU/ft^2/yr. For a 10,000 gallon tank of diameter 10 ft, the surface area of the tank is 691.81 ft^2. The heat loss from the tank is therefore:

Heat Loss	=	204,400 BTU/ft2/yr x 691.81 ft^2
	=	141,405,964 BTU/yr
	=	141,405,964 BTU/yr x (1 ton-hr/12,000 BTU)
	=	11,783.83 ton-hrs/yr x 0.6 kW/ton
	=	7,070.3 kWh/yr

The work done by the ammonia compressor can be calculated by using a power consumption factor of 0.207 kW/ton for ammonia [17].

Current power by NH3 comp.	=	62.3 tons x 0.282 hp/ton x 0.746kW/hp
	=	13.106 kW

The work done by the ammonia compressor to cool glycol to 25^oF for freezing the ice balls in the storage tank is calculated below.

Proposed power by NH3 comp.	=	82.5 tons x 0.282 hp/ton x 0.746 kW/hp
	=	17.356 kW

For a year, the increase in work done by the ammonia compressor will be assumed to shift the ammonia compressor from an unloaded state to a slightly less unloaded state, a factor of about 20%.

Increase in work done = (17.356 - 13.106) kW x 8760 hrs/yr x 0.2

= 7,446 kWh/yr

From the previous equations, the thermal losses from the tank is very close to the increase in work done by the ammonia compressor as such the previous statement suggesting that the losses and work done cancel out is correct.

The condenser in the ammonia refrigeration cycle will run more efficiently at night because of the lower outside temperature. The effect of which will be noticed on the evaporator thus the overall effect of a lower outside temperature will cancel out.

5.1.1 Implementation

To implement the recommended action, a thermal storage system will need to be designed and built. The system will be made up of two heat exchangers, a storage tank filled with balls containing distilled water. The system will need to be sized for the capacity of the plant, 80,000 lb/hr. The cooling capacity needed to cool CO_2 entering the facility from 95^{o}F to 34^{o}F is calculated as follows:

Heat	=	80,000 lb / hr x 0.203 BTU / lb-°F x (95 – 34°F)
	=	990,640 BTU / hr x 1 ton / 12,000 BTU / hr
	=	82.6 tons

In the summer, on peak periods are on weekdays between 1 pm and 9 pm and in the winter between 6 am to 1 pm. If we assume it will take 16 hrs for ammonia to cool glycol for ice formation, the storage capacity may be calculated as follows.

At 80,000 lb/hr production rate, the storage capacity is:

Storage needed	=	82.6 tons x 8 hr
	=	661 ton-hr

Theoretical ice quantity	=	661 ton-hrs x 12,000 BTU/ton-hr x 1 lb ice /144 BTU
	=	55,083 lb ice

Theoretical ice volume	=	55,083 lb ice x 1 gal ice / 7.5 lbs
	=	7,344 gal

Ice balls have an estimated 20% void space between them when placed together in a storage tank.

Tank Size	=	7,344 gal x 120%
	=	8,813 gal

To account for losses a 10,000 gallon tank will be needed to hold all the ice balls that will be required to meet the cooling load. The ice balls will typically cost $40 /ton-hr for a total of $26,440. A 10,000 gallon storage tank with insulation is estimated at $16,250 [15] with $9,250 [16] for additional controls, wiring, piping and labor.

The system will also need two heat exchangers. One will transfer heat between the CO_2 and the glycol. The other will transfer heat between the ammonia and glycol. Each is expected to cost about $3,200[16].

Implementation Cost	=	$26,440 + $16,250 + $9,250 + $6,400
	=	$58,340

The simple payback period if thermal storage is installed and operates at full load is:

Simple Payback	=	Implementation cost / Total cost savings
	=	$58,340/$19,860.24
	=	3 years

5.2 CASE 2: Thermal Storage system in use with a Chilled water system

5.2.1 Replace Rooftop Air conditioners with Chilled water system

Background

The facility currently uses air cooled rooftop units for plant conditioning. These systems are less efficient than water cooled systems. It is recommended that the facility change from the current air cooled system to a centralized, chilled water system. A chilled water system will require that the cooling coil in the air handling units be replaced with a chilled water cooling coil. Controls will also have to be either replaced or altered, but the heating components in the package units will remain unchanged. A water cooled chiller system has an efficiency of about 0.65 kW per ton while the current rooftop units use around 1.4 kW per ton. The calculations below are based on Duke Power's rate schedules discussed in Chapter 4.

Table 5.2 Demand and Energy rates at the facility

		on-peak	**Off- peak**
Demand		$7.48/kW	Negligible
Energy	$0.05560/kWh	$0.06055/kWh	$0.04852/kWh
Peak Hours	1:00pm - 9:00 pm		

Anticipated Savings

The facility currently operates about 30 rooftop air conditioning units for a total of 470 tons. Smaller units of 4 or less tons will not be considered for this recommendation. An estimate of the cooling loads for the facility throughout the year will be calculated by using the cooling load temperature differential method previously discussed in Chapter 3. The calculations will be done with the use of an analysis package developed at the NCSU IAC. For this calculation, it was assumed that the process cooling load is negligible. The internal heat generation for each shift at the facility was estimated from the energy into the facility during these shifts.

The maximum hourly load for the entire facility was calculated to be 447 tons. The values for cooling in the table below correspond to average daily maximum loads over the entire month to account for differences between the maximum cooling load and the maximum electrical demand. This is a conservative assumption because the peak electrical demand will probably occur when the cooling load is a maximum. We will assume that the existing air conditioning system operates at an efficiency of 1.4 kW/ ton. Assume also that the load in the winter is negligible; Table 5.3 shows the calculated cooling loads for March to October.

Table 5.3: Calculated Cooling Load - Production and Office Spaces

Month	Total Load ton-hrs	Max Load tons
March	54,601	346
April	79,549	351
May	112,636	411
June	140,761	447
July	154,899	447
August	139,491	426
September	105,624	403
October	66,248	330
Total	**853,809**	

From the above table, the demand, energy and cost can be calculated. For the month of June as an example,

Demand = Max Load x 1.4 kW/ton

= 447 tons x 1.4 kW/ton

= 625.8 kW

Energy = Total Load x 1.4 kW/ton

= 140,761 ton-hrs x 1.4 kW/ton

= 197,065 kWh

The current demand and energy cost can now be obtained from the above energy and demand.

Demand and energy cost = 625.8kW x $7.48/kW +

197,065 kWh x $0.05560/kWh

= $15,638

In a similar manner, the estimated electrical demand and energy consumption for the existing air conditioning system is given in the Table 5.4.

Table 5.4: Current Air Conditioning Electrical Demand and Energy Usage

Month	Demand (kW)	Energy (kWh)	Demand & Energy Cost
March	484	76,441	$7,873
April	491	111,369	$9,868
May	575	157,690	$13,072
June	626	197,065	$15,638
July	626	216,859	$16,738
August	596	195,287	$15,319
September	564	147,874	$12,442
October	462	92,747	$8,613
Total	**4,424**	**1,195,332**	**$99,563**

The air conditioning units are old and they are currently being replaced at a rate of two per year. It is estimated that each of the units replaced every year is at least 17 tons. The cost of a 17 ton air conditioning unit is about $20,100. With two air conditioning units replaced every year, the total replacement cost is $40,200.

Assume a total maintenance cost of $50,000 /yr including labor based on contractor estimate. This includes the cost of two air conditioning units, their installation and the cost of maintaining other units in place.

Accounting for electricity and maintenance, the cost to air condition the plant is $149,563/yr
A new chilled water system will operate more efficiently and will cost less to maintain. From vendor data, a typical water chiller operates with an efficiency of 0.65 kW/ ton or less. The proposed cooling load for the facility is shown in Table 5.5. The demand, energy and cost are calculated in the same manner as was done for the estimated existing system.

Using the month of June as an example,

Demand	=	Max Load x 0.65 kW/ton
	=	447 tons x 0.65 kW/ton
	=	291kW
Energy	=	Total Load x 0.65 kW/ton
	=	140,761 ton-hrs x 0.65 kW/ton
	=	91,495 kWh

The current demand and energy cost can now be obtained from the above energy and demand.

Demand and energy cost = 291 kW x $7.48/kW + 91,495 kWh x $0.05560/kWh

= $7,260

Table 5.5: Proposed Air Conditioning Electrical Demand and Energy Usage

Month	Demand (kW)	Energy (kWh)	Demand & Energy Cost
March	225	35,491	$3,656
April	228	51,707	$4,581
May	267	73,213	$6,069
June	291	91,495	$7,260
July	291	100,684	$7,771
August	277	90,669	$7,112
September	262	68,656	$5,777
October	215	43,061	$3,999
Total	**2,056**	**554,976**	**$46,225**

The maintenance cost for the central chiller is estimated to be $25,000/yr. The total cost savings, accounting for electricity and maintenance are:

Demand/ Energy cost difference = $99,563/yr - $46,225/yr

= $53,338/yr

Maintenance cost difference	=	$50,000/yr - $25,000/yr
	=	$25,000/yr
Cost Savings	=	$53,338/yr + $25,000/yr.
	=	$78,338/yr.
Energy Savings	=	1,195,332 kWh /yr – 554,976 kWh/yr
	=	640,356kWh /yr

Implementation

To implement this recommendation, a 450 ton chiller will be installed. Table 5.6 is an estimate of the implementation cost which includes labor cost [16].

Table 5.6: Implementation Cost

	Cost($)
450 ton chiller	228,500
Cooling tower	84,100
Piping and valves	6,200
Total	$318,800

Estimated Payback	=	Implementation cost / Total cost savings
	=	$318,800 / $78,338
	=	4 years

5.3 INSTALL A COLD THERMAL STORAGE SYSTEM

Economic feasibility of a cold thermal storage system depends on whether the total first cost is sufficiently low to justify the operational savings over the life of the system [1].
We will assume that this plant will go with the above chilled water system recommendation. As previously discussed, a thermal storage system can save facilities under the Time of Use rate schedule a lot of money. The set up for this system will be the same as the chilled water thermal storage system discussed in Chapter 2.

For this feasibility study, we will assume that the chilled water system will only be in operation from April 1 to October 31, as this are the months that require the most cooling load. The thermal storage system can be set to operate either as a full storage system or as a partial storage system. The operational loading that will best suit the facilities need will be determined from the cost savings and payback.

In order to determine the cost savings and payback for each of the possible operational loading schedule, the cooling load for every hour between April 1 and October 1 will need to be analyzed. The data used in these analyses were obtained from the CLTD calculations used previously for the chilled water system recommendation. In the absence of a thermal storage system, the average energy, demand and consequently cost of meeting the cooling load requirements can be calculated as follows. For the following calculations, June will be used as an example.

The average on-peak cooling loads from each day in June are determined by taking the average daily values of the cooling load during peak periods (1:00 PM to 9:00 PM) based on cooling load data obtained from the CLTD calculations. The average on-peak cooling loads for June is determined by taking the average of the daily on-peak cooling load averages. For the month of June, the average on-peak cooling load is 305 tons. This is repeated for April to October. In a similar manner, the average off-peak energy is determined from averaging the average daily cooling loads during off-peak periods (12:00 am to 11:59 pm, 9:01pm to11:59 pm). The average off - peak cooling load is 135 tons for June. The chilled water system will be estimated to have an efficiency of 0.65 kW/ton. It will also be assumed that off-peak demand is negligible.

On-peak Demand	=	On-peak cooling load x efficiency of system
	=	305 tons x 0.65kW/ton
	=	198.60 kW

On-peak Energy	=	On-peak Hours x On-peak cooling load x Efficiency
	=	8 hrs/day x 22 day/month x 305 tons x 0.65kW/ton
	=	34,954 kWh/month

Off-peak Energy = Off-peak Hours x On-peak cooling load x Efficiency

= ((16 hrs/day x 22 days/month) +

(24 hrs/day x 8 days/month)) x 135 tons x 0.65 kW/ton

= 47,699kWh/month

Cost = (On-peak demand x On-peak Demand rate) +

(On-peak Energy x On-Peak Energy rate) +

(Off-peak Energy x Off-peak rate)

Cost = (198.60 kW x $7.48 /kW) +

(34,954 kWh /month x $0.06055/kWh) +

(47,699 kWh/month x $0.04852/kW)

= $6,605.99/month

Table 5.7: Energy, Demand and Cost without Thermal Storage.

Month	on-peak Demand (kW)	on-peak energy (kWh)	off-peak Energy (kWh)	Cost
April	139	24,427	36,064	4,936.42
May	173	30,370	37,123	5,643.10
June	199	34,954	47,699	6,605.99
July	204	35,877	50,861	6,816.40
August	193	33,907	43,225	6,222.55
September	163	28,734	37,756	5,534.28
October	112	19,750	34,160	4,460.58

Full storage:

The full storage operational loading was discussed previously in Section 2.1. This is a load shifting system in which cold storage is used to off- set cooling load needs during peak periods. For this analysis, the peak period will be 1:00 PM to 9:00 PM. With this operational schedule, demand and energy during the peak periods for each month is zero. The cooling load during the peak period will be completely transferred to the off-peak period. Cooling during on peak periods will be provided for by the thermal storage system. The energy previously used during on-peak periods will be transferred to off-peak. Table 5.8 shows load shifting demand, energy and cost from April to October. For load shifting, assume a 5% loss of energy in transferring on peak energy to off-peak.

Using the month of June as an example,

On-peak Demand	=	0 kW/month
On-peak Energy	=	0 kWh/month
Off-peak Energy	=	140,761 ton - hrs x 0.65kW/ton / 0.95
	=	96,310 kWh/month
Cost	=	96,310 kWh/month x $0.04852 /kWh
	=	$4,672.97/month
Cost savings	=	Cost without thermal Storage – Load shifting Cost
	=	$6,605.99/month - $4,672.97/month
	=	$1,933/month

Table 5.8: Load shifting demand, Energy and cost

Month	On-Peak Demand kW/month	On-Peak Energy kWh/month	Off-Peak Energy kWh/month	Total Monthly Cost	Monthly Cost Savings
April	0	0	54,428	$2,640.86	$2,295.56
May	0	0	77,067	$3,739.28	$1,903.82
June	0	0	96,310	$4,672.97	$1,933.02
July	0	0	105,984	$5,142.32	$1,674.08
August	0	0	95,441	$4,630.81	$1,591.74
September	0	0	72,269	$3,506.49	$2,027.79
October	0	0	45,328	$2,199.29	$2,261.28
Total	0	0	546,827	$ 26,532.02	$13,687.29

Partial –Storage: This was previously discussed in Chapter 2. For partial storage systems, the chiller runs either fully or partially loaded simultaneously with the thermal storage system. There are two types of operational schedules to be considered.

Load Leveling Operation:

The chiller is set to run at full capacity all day long. It eliminates the peak loads observed at on-peak times as the cooling load is uniformly distributed during the day. In order to calculate the energy and demand, the peak cooling loads will be redistributed over all hours of the day for an average load of 201 tons.

Considering the month of June,

On-Peak Demand	=	Average cooling load per month x 0.65kW/ton
	=	201ton x 0.65 kW/ton
	=	131 kW
On-Peak Energy	=	201 tons x 0.65kW/ton x 8 hrs/day x 22days/month
	=	22,986 kWh/month
Off-Peak Energy	=	(201 tons x 0.65kW/ton x 16 hrs x 30days/month) + (24 hrs x 8 days/month x 201 tons x 0.65 kW/ton)
	=	87,764 kWh/month
Cost	=	(131 kW x $7.48 /kW) + (22,986 kWh/month x $0.06055 /kWh) + (87,764 kWh/month x $0.04852 /kWh)
	=	$6,627.02/month
Cost savings	=	Cost without thermal Storage – Load shifting Cost
	=	$6,605.99 - $6,627.02/month
	=	-$21.04 /month

Table 5.9: Load leveling demand, Energy and cost

	On-Peak Demand (kW/month)	On-Peak Energy (kWh/month)	Off-Peak Energy (kWh/month)	Total Monthly Cost	Monthly Cost Savings
April	102.8	18,094	69,088	$5,216.79	-$280.37
May	110.3	19,404	74,090	$5,594.46	$48.64
June	130.6	22,986	87,764	$6,627.02	-$21.04
July	135.3	23,818	90,941	$6,866.89	-$50.50
August	123.2	21,674	82,756	$6,248.87	-$26.32
September	110.8	19,492	74,423	$5,619.65	-$85.36
October	94.1	16,563	63,239	$4,775.15	-$314.57
Total	807.0	142,031	542,301	$40,948.83	-$729.52

Table 5.9 above shows the load leveling demand, energy and cost from April to October.

Demand – Limiting systems:

This operating schedule has also been previously discussed in Chapter 2. For this schedule, the chiller runs at a reduced capacity during on – peak periods. For this analysis, the chiller will be set to run at 70% of the average chiller capacity during on-peak periods. Reducing the capacity of the chiller drastically could result in a loss of efficiency on the chiller which may result in additional losses. Off-peak cooling load will not be affected.

Considering the month of June,

The average on-peak energy was estimated from CLTD data to be 306 tons and the average off-peak energy was estimated to be 135 tons.

On-Peak Demand = 306 tons x 0.65kW/ton x 0.7

= 139 kW

On-Peak Energy = 8 hrs/day x 22 day/month x 306 tons x 0.65 kW/ton

= 34,954 kWh/month

Off-Peak Energy = (135 tons x 0.65kW/ton x 16 hrs x 30 days/month) +

(24 hrs x 8 days/month x 135 tons x 0.65kW/ton)

= 58,922 kWh/month

Cost = (139 kW x $7.48 /kW) +

(34,954 kWh/month x $0.06055 /kWh) +

(58,922 kWh/month x $0.04852 /kWh)

= $6,015.23/month

Cost savings = Cost without thermal Storage – Demand Limiting cost

= $6,605.99- $6,015.23/month

= $590.75 /month

Table 5.10: Demand-Limiting, demand, Energy and cost

	On-peak Demand (kW/month)	**On-Peak Energy (kWh/month)**	**Off-Peak Energy (kWh/month)**	**Total Monthly Cost**	**Monthly Cost Savings**
April	97.2	24,427	44,550	$4,367.34	$569.07
May	120.8	30,370	45,857	$4,967.40	$675.69
June	139.0	34,954	58,922	$6,015.23	$590.75
July	142.7	35,877	62,829	$6,288.13	$528.27
August	134.9	33,907	53,395	$5,652.56	$569.99
September	114.3	28,734	46,640	$4,857.60	$676.68
October	78.6	19,750	42,197	$3,830.87	$629.71
Total	827.4	208,019	354,390	$35,979.13	$4,240.16

5.3.1 Implementation

The cost of installing the thermal storage system and setting it up to operate as either full storage or partial storage will need to be determined. From the implementation cost obtained, the payback of each of the operating schedules will be obtained.

Load Shifting:

The storage capacity of a thermal storage system to be used for load shifting operations will require that the storage system be sized based on the maximum cooling loads that expected during peak periods. The month of July has the maximum cooling load of 447.6064 tons and as such will be used in determining the storage requirements of the system for the year.

Using the steps outlined in Chapter 2, the system can be sized.

Storage capacity = 447.6064 tons x 8 hr

= 3,580.85 ton-hr

The ice required to store this cooling capacity is

Ice Quantity = 3,580.85 ton-hrs x 12,000 BTU / ton-hr

x 1 lb ice /144 BTU

= 298,404 lb ice

Ice Volume = 298,404 lb ice x 1 gal ice / 7.5 lbs

= 39,787 gal

Ice balls have an estimated 20% void space between them when placed together in a storage tank.

Tank Size = 39,787 gal x 120%

= 47,745 gal

An estimated 50,000 gal tank will be needed to hold all the ice balls that will be required to meet the cooling load. The ice balls will typically cost $40 /ton-hr for a total of $143,234. A 50,000 gallon storage tank with insulation is estimated at $69,395 [15]. Table 5.11 accounts for material and labor cost for this implementation [16].

Table 5.11 Implementation cost for the load shifting operation schedule

Material	Cost($)
50,000 gallon tank	69,395
Ice Balls	143,234
Piping and valves	12,400
Total	$225,029

The total cost savings from April to October is $13,687.29 from Table 5.8.

Simple Payback	=	Implementation cost / Total cost savings
	=	$225,029 / $13,687.29/yr
	=	16.4 years

Load Leveling:

To determine the thermal storage capacity needed to operate this schedule, the day with the maximum total load is obtained from CLTD. The largest day total load is then divided by 24 hours to obtain the desired tons. July has the largest daily total with a ton value of 238.74 tons and as such will be used to size the tank required.

Storage capacity = 238.74 tons x 8 hr

= 1,909 ton-hr

The ice required to store this cooling capacity is

Ice Quantity = 1,909 ton-hrs x 12,000 BTU / ton-hr x 1 lb ice /144 BTU

= 159,083 lb ice

Ice Volume = 159,083 lb ice x 1 gal ice / 7.5 lbs

= 21,211 gal

Ice balls have an estimated 20% void space between them when placed together in a storage tank.

Tank Size = 21,211 gal x 120%

= 25,453 gal

An estimated 30,000 gal tank will be needed to hold all the ice balls that will be required to meet the cooling load. The ice balls will typically cost $40 /ton-hr for a total of $76,397. Table 5.12 shows the breakdown of the implementation cost labor cost factored in.

Table 5.12 Implementation cost for the load leveling operation schedule

Material	Cost($)
30,000 gallon tank	40,105
Ice Balls	76,397
Piping and valves	12,400
Total	$128,902

The total cost savings from April to October is -$729.52 from Table 5.9 which is a deficit. This operational schedule should not be implemented.

Demand Limiting:

To determine the thermal storage capacity needed to operate this schedule, the maximum on-peak load for the year will be reduced by 30%. The maximum on-peak load is in the month of July with a maximum load of 447.61 tons

Storage capacity = 447.61 tons x 8 hr x 0.7

= 2506.62 ton-hr

The ice required to store this cooling capacity is

Ice Quantity	=	2506.62 ton-hrs x 12,000 BTU / ton-hr
		x 1 lb ice /144 BTU
	=	208,885 lb ice

Ice Volume	=	208,885 lb ice x 1 gal ice / 7.5 lbs
	=	27,851 gal

Ice balls have an estimated 20% void space between them when placed together in a storage tank.

Tank Size	=	27,851 gal x 120%
	=	33,421 gal

An estimated 35,000 gal tank will be needed to hold all the ice balls that will be required to meet the cooling load. The ice balls will typically cost $40 /ton-hr for a total of $100,264.

Table 5.13 shows the breakdown of the implementation cost with the cost of labor factored in.

Table 5.13 Implementation cost for the Demand limiting operation schedule

Material	Cost($)
35,000 gallon tank	45,847
Ice Balls	100,264
Piping and valves	12,400
Total	$158,510

The total cost savings from April to October is $4,240.16 from Table 5.10.

Simple Payback = Implementation cost / Total cost savings

= $158,510 / $4,240.16

= 37 years

Figure 5.3 below shows the total cost savings obtained for each operational schedule. From the figure, we note that load shifting has the greatest cost saving. This result is expected since a load shifting operation moves peak loads to off peak and completely eliminates peak demand charge. From the figure we also note that load leveling has a deficit cost savings, this deficit suggests that the facility will have to spend more money to operate this schedule.

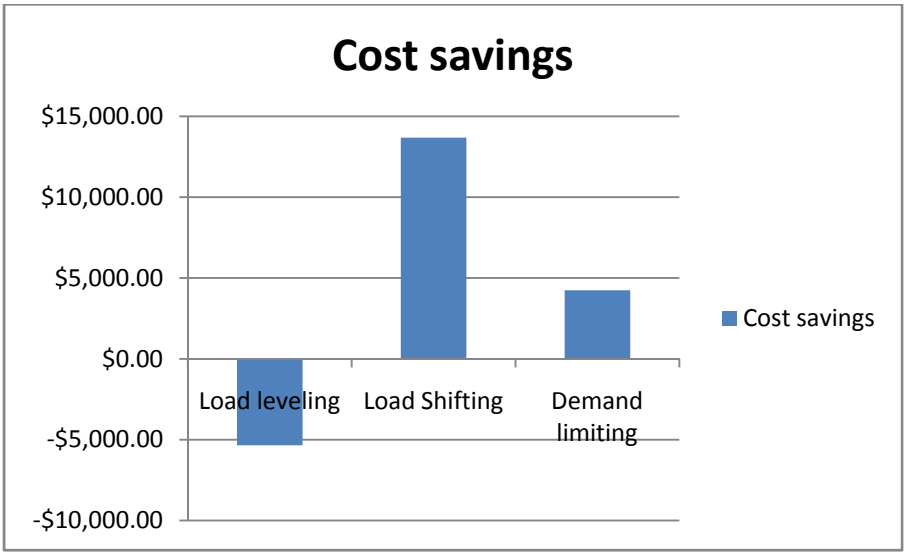

Figure 5.3 Cost Savings for each operational Schedule

Figure 5.4 further shows how long it would take the facility to recover its investment in the thermal storage system. From the figure, a load shifting operational schedule will have the shortest payback. If the facility decides to install a cold thermal storage system, they should operate on a load shifting operational schedule as it provides the shortest payback.

It is also worth noting that the storage tank requirement for a load shifting operational schedule is about 15,000 gallons greater than the tank capacity for the other two operational schedules. The total cost savings for this schedule more than makes up for the extra cost of the tank.

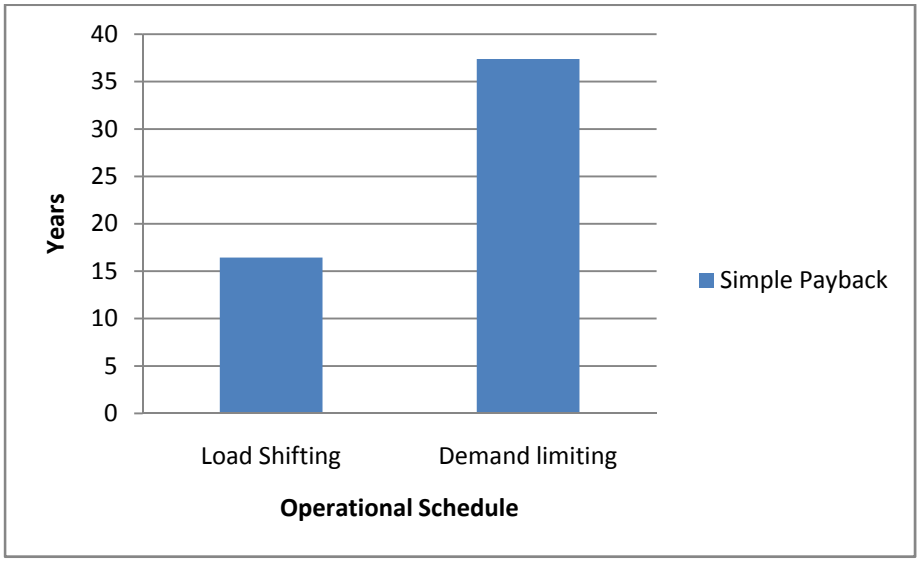

Figure 5.4 Simple payback for each operational Schedule

CONCLUSION

Cold thermal storage systems are installed at a facility to help manage it cooling load requirements during on-peak periods. The choice of installing a thermal storage system is affected by cost, efficiency, cooling needs, expansion of HVAC systems amongst others. The two cases studied depict two possible uses of a thermal storage system in industry.

In case 1 where thermal storage was used to cool CO_2 gas before compression, with a significant energy savings obtained from the CO_2 compressor. The savings will also pay for the installation of the cold storage system in about 4 years. While this storage system does not actually shift cooling loads from on-peak periods to off peak periods, the facility saves because its CO_2 compressors does not require as much power to compress the cool CO_2 gas. The savings on the ammonia compressors at the facility were neglected as they are assumed to cancel out the heat loss from the thermal storage tank. The payback of only 4 years suggests that the facility should go ahead with the installation of the thermal storage system.

In case 2, thermal storage was considered for installation at a facility that will be replacing its rooftop air-conditioning units with a chilled water system. Three different operational scenarios where looked into for installing a thermal storage system. For the load shifting scenario, a significant savings was made each month when compared to running the chilled water system without thermal storage.

From the three operational schedules, load shifting provided the best cost savings and the shortest payback of 16 years. Should the facility decide to install a thermal storage system, it should operate the system based on this schedule. We note from Figure 5.4 that the simple payback for a load leveling operational schedule has been omitted. This is because the facility will incur a deficit if this operational schedule is implemented.

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