TECHNOLOGICAL POTENTIAL OF SMALL SCALE ICE THERMAL STORAGE BASED AIR CONDITIONING SYSTEM IN THE GENERATION PHASE FOR HOTEL & ENTERTAINMENT INDUSTRY OF SRI LANKA

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Degree of Master of Science

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DECLARATION

The research work submitted in this dissertation is my own investigation except where otherwise stated. This dissertation has not been accepted for any degree and not concurrently submitted for any degree in a university or any other institution.

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I endorse the declaration by the candidate.

Prof. R. Attalage

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ABSTRACT

Demand for electricity during a day, varies with the time due to various factors. Electricity demand of Sri Lanka for a typical day could easily be divided into three main categories. One segment characterizes a very high sudden demand increase during later in the evening, and sharp decrease of the demand during the mid night until the next day early morning time and average daily demand during the day time. As a remedial action in facing this change of demand, electricity service providers generally encourage users to reduce the demand through different measures and also shift their consumption during the high demand period to the low demand period. This is achieved by introducing different electricity tariffs based on the time of the day. As Air Conditioning systems demand considerable percentage of building electricity consumption, Cold Thermal Storage technologies is an ideal candidate for electrical load shifting applications of buildings. This study explores the technological feasibilities and also reviews the engineering economics of building small scale Ice Thermal Storage based air conditioning system in the generation phase which has average capacity of 32 Ton Hours. The development of small scale thermal storage based air conditioning system is progressed through a detailed research work and final design was reviewed of its economic feasibility to be used for hotel rooms & 100 capacity movie theaters under Hotel & General electricity tariff structures respectively.

This study further investigates in particular the ice building process in a water filled, limited length horizontal rectangular enclosure with the constant temperature glycol circulation system at the top & bottom surfaces. The rates of ice building on top and bottom surfaces were mathematically modeled based on the equations published by Et al. P. Bhargavi & Et al. Liang Yong. The dimensionless equations were then converted to dimensional and set of equations were derived to find the ice thickness Vs time, temperature profile along ice thickness at a given time and several other associated parameters necessary to calculate the heat transfer during water freezing. The goal was to find the maximum achievable ice thickness during 6.5 hours period and total energy extracted by the glycol circuit. Three glycol temperatures of -12C, -6C and -3C were considered and 3 data sets were built.

By considering a given glycol temperature, the built ice thickness was calculated and tabulated for 6.5 hours period at 20 minutes intervals. Thereafter the temperature profile along the ice thickness was tabulated during the end of each 60 minutes (1 hour) up to 6 hours and final data set was tabulated at the end of 6.5 hours. Here the temperature profile was estimated at every 2.5 mm distance along the built ice thickness. The width & length of water filled rectangle enclosure was selected as 10 cm & 110 cm respectively and this unit is called Primary Ice Making Chamber. The height was selected based on the final built ice thickness which was decided based on temperature of glycol circuit. Finally relevant total energy extracted and final ice volumes were calculated for 3 different glycol circuit temperatures.

In order to achieve the uniform ice thickness inside the Primary Ice Making Chamber, counter flow arrangement was introduced to glycol circuits placed at top & bottom surfaces of it. Still there is a drop of final ice volume. The volume reduction was calculated and relevant total energy removed by the glycol circuit was calculated. This was repeated for 3 different glycol temperatures. The glycol circuits were designed and relevant flow rates were calculated to maintain the heat transfers for 3 designs scenarios. Finally 3 ice thermal storage designs were evaluated. The cost of manufacturing was calculated for all three designs. The operational cost was calculated for all three cases under hotel tariff for using at hotels and under general tariff for using at Movie Theaters. It was revealed that the price of chiller contributes to more than 50% of the cost of manufacturing for all 3 designs.

The payback period for the use case of hotel rooms under hotel tariff was found to be 4.3 years. The use case of Movie Theater has a 3.4 years payback period. This clearly indicates the further possibility of reducing the payback period under both cases used by cutting down the capital cost of chillers. When these units are manufactured on an industrial scale, it would further reduce the cost of chillers by volume discounts. This study makes a clear indication that small scale ice thermal storage systems are economically feasible.

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LIST OF ABBREVIATIONS

CTS	Cold Thermal Storage
CEB	Ceylon Electricity Board
GP-2	CEB General purpose tariff for each individual point of supply delivered and metered at 400/230 Volts nominal and where the contract demand exceeds 42 KVA.
ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
TR	Ton of Refrigeration
Ton Hr	Ton Hours
TES	Thermal Energy Storage
PIMC	Primary Ice Making Chamber
ATES	Aquifer Thermal Energy Storage
CIMC	Composite Ice Making Chamber
M	Mass
P	Density
V	Volume
M	$p * V$
C_p	Specific heat

Nomenclature used for mathematical model

b	Wall of the capsule
c	Specific heat
h	Thickness of the capsule
Hs	Solidification latent heat of ice
k	Thermal conductivity
l	Liquid phase change material
m	Freezing point of water
S	Solidification thickness
Ste	Stephan number
S	Solid phase change material (ice state)
t	Time
T	Temperature
x	Longitudinal coordinate
X	Dimensionless longitudinal coordinate
y	Solidification thickness of ice
α	Thermal diffusivity
θ	Dimensionless temperature
ρ	Density
τ	Dimensionless time

LIST OF APPENDICES

1. INTRODUCTION

1.1 Background

The demand for electricity is varied during the day and it usually experience sudden increase as well as sharp drop during a typical day in Sri Lanka. Electricity service providers around the world prefer to maintain an average flat demand curve and employ different strategies to encourage consumers to shift their electricity demand from high demand peak time to low demand off peak periods. This is usually done through introducing low tariffs for off peak times. Ice Thermal Storage is a very established technology used around the world for shifting electricity loads from peak to off peak times. This technology has very limited use in Sri Lanka due to the various reasons which are beyond the scope of this thesis. Even though there are large number of thesis and technical documents were published for using of cold thermal storage technology for large scale air conditioning system, it is rare to find a thesis or technical documents giving due attention to the use of Cold Thermal Storage technology for small scale domestic air conditioning systems.

According to the data published by Ceylon Electricity Board, more than 40% of its total electricity generation is sold to domestic & religious purpose category. This is a passive indicator that there is a growing demand for domestic air conditioning systems in Sri Lanka. This has opened a huge market potential for room air conditioning systems. Almost all the apartment units so far built in the city of Colombo has equipped with air conditioning systems. This growing trend contributes to large electricity consumption by domestic air conditioning systems. Currently there are no attractive electricity tariff structures during off peak time for domestic users but hotel industry is currently enjoying an attractive low tariff during the off peak period which has opened possibilities of using small scale ice thermal storage systems for room air conditioning. Especially in resort type hotels & small hotels, individual split type & package type air conditioning systems are being preferred to central chiller units.

Current typical daily load curve of Sri Lanka characterized by sudden demand increase during the period from 6.30 PM to 10.00 PM and sharp drop in demand thereafter until the following day 6.30 AM. During the day there exists a average demand until 6.30 PM. The change in typical daily electrical load curve of Sri Lanka over the years is presented in the **Fig 1.1**.

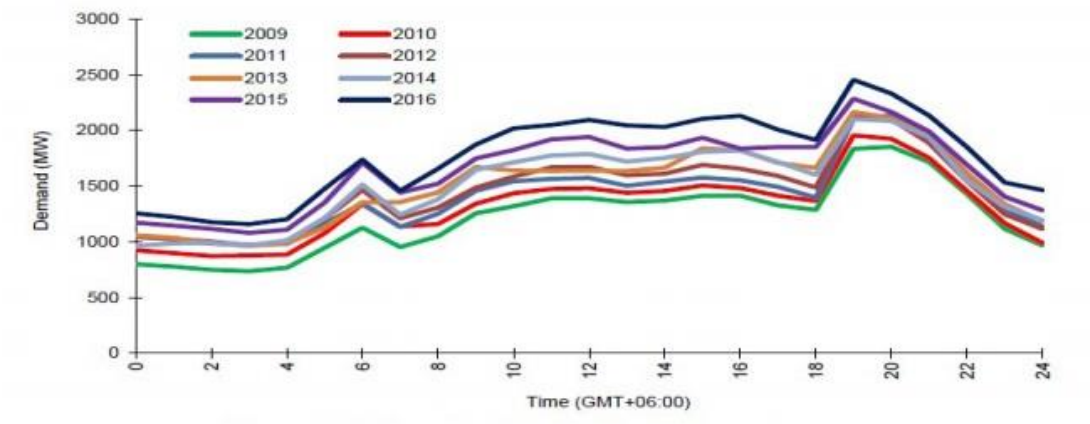

Fig. 1.1 : Change of typical daily electricity load curve over the years. [4]

1.2 Research Question

a.) What are the technologies used in Cold thermal storage based air conditioning systems and their evolution.

b.) How to make a mathematical analysis of water freezing and ice forming inside a rectangular closed enclosure with heat absorbing system in place at the top & bottom wall.

c.) Is it economically feasible & technologically viable to build and use small scale ice thermal storage based air conditioning system for hotel & entertainment industry of Sri Lanka.

1.3 Research Objective

a) Investigate & develop a research approach for designing of small scale ice thermal storage based air conditioning systems.

  a. Investigate the most economical glycol temperatures to be used for generating ice.

  b. Investigate the most appropriate dimensions of Primary Ice Making Chambers for the design of 32 Ton Hours capacity ice thermal storage based air conditioning unit.

2. LITRERATURE REVIEW

2.1 Back ground and concept of cold thermal storage systems

Cold Thermal Storage is one of the best demand management technologies in MVAC industry. This would provide air conditioning more economically by using cheaper electricity at night/low tariff period to freeze water into ice and the ice storage is used to produce chilled water by avoiding or reducing electricity consumption during the peak tariff period.

Similar concept was used during 1930's to cool the fresh milk produced in dairy farms. Rather than operating high capacity chillers during the day time for short period, they began to produce ice with using reduced capacity chillers running for little longer hours during off time. This enabled dairy farms to reduce their investment on high capacity chiller equipment. The batches of fresh milk was cooled quickly by melting ice. The ice storage will be recharged during the time gap between consecutive milking cycles. Thermal ice storage still plays a vital role in milk cooling in dairy farms. [15]

Ice still plays a major role in thermal comfort applications in building services engineering. Even the definition of a Ton of refrigeration is derived based on ice. The latent heat of fusion (amount of thermal energy release/absorb during the transition of water to ice or ice to water) is 144 British thermal units (Btu) per pound of water. One ton of ice is equivalent to 2,000 pounds and energy required to convert 2,000 pounds of water to ice will be 144 x 2000 = 288,000 British thermal units or Btu's. According to the definition this is achieved during 24 hours period. The rate energy release/absorb during each hour is amounting to 12,000 Btu. This rate of exchange of energy is defined as a Ton of Refrigeration. During late 70's and early 80's refrigeration engineers started to explore the possibilities of designing ice thermal storage based air conditioning systems. No sooner electricity companies realized this technology is an ideal way to shift electrical load from peak to off peak time slots of the day. This helped them to reduce their investment on unnecessary plant expansions. They later began to offer various incentive schemes and other benefits to consumers to encourage the use of ice thermal storage based air conditioning systems.

Today the thermal energy storage (TES) is a matured technology with proven track record which is widely popular in most of the countries in the world. This technology now available in many forms under different application scenarios includes solar collectors, Seasonal Thermal Storage, Steam Accumulators, Heat Storage in hot rocks & Concrete, Ice Storage Air Conditioning and many others to name.

The seasonal thermal storage significantly reduces the energy required for heating during the winter & cooling during the summer. The technology that Stores freely available energy to meet the requirements of a later season is “seasonal storage”. In most cases cold or hot water is stored under the earth. In order to rectify the subsidence problem, it is necessary to inject the ground water back into the earth as a mandatory measure in most developed countries. In order to store cold water under earth aquifers are used. This technology enables to store cold water for months at the cold temperature for district cooling systems. This is usually termed as aquifer thermal energy storage (ATES). The system efficiency of the ATES is reduced due to the reinjection of warm water to mitigate the problem of subsidence.

2.2 Fundamentals of Cold Thermal Storage and Phase Change Materials (PCM)

The Cold Thermal Storage system simply enables shifting of Air Conditioning loads into the different time period of the day. The required ice storage is made during the off peak time where the electricity tariff is substantially low compare to the actual air conditioning load demand time period.

There are numerous economic advantages could be obtained from cold thermal storage based air conditioning systems, in addition to the direct tariff discounts for electrical energy. Usually air conditioning equipment are designed for maximum peak demands but under cold thermal storage based systems, refrigeration equipment sizing could be done for time average load and significant reduction in capital expenditure is inevitable. In order to optimize the investment and to make the best economic benefits in cold thermal storage, optimum design should be done based on the outcome of a carefully analysis of the cooling load requirement, equipment efficiency, cost of electrical energy during the off peak period plus other influencing factors related to the project. The optimum design paves the way to decide the most appropriate operational scenario which decides when to charge and discharge the cold thermal storage. There exists proven operational strategies for cold thermal storage systems and those will be discussed in detail in later part of this report. Cold thermal storage systems are designed with wide variety of engineering technologies. But simply the storage mechanism depends on one of the followings method;

A. Sensible Heat Storage uses the principle of sensible heat transfer characteristic of solid or liquid medium.
B. Latent heat removal based storages uses the principle of heat exchange during the change of phase of a material. These materials are called “Phase Change Materials” (PCM)
C. Thermo Chemical based storages uses principle of heat exchange during chemical reactions of materials.

2.2.1 Sensible heat storage based systems

The sensible heat storage based systems use the energy store/release due to the change of temperature of any given medium such as water, brine, rock or earth. Sensible heat storage capacity simply depends on mass and the specific heat of the medium.

Sensible heat storage capacity

Q_{stored} – Energy Stored / Released during the process

T_1 - Initial Temperature

T_2 – Final Temperature

$Q_{stored} = M * C_p*(T_1 - T_2)$

or else

$Q_{stored} = p * V * C_p*(T_1 - T_2)$

The total energy exchange is decided by the M, C_p and heat difference. According to the above expression, it is evident that the specific heat and density are key characteristic of any medium those used in sensible heat storage systems. It is obvious that larger the specific heat and density of any given medium, they are good candidates to be used for sensible heat storage systems.

2.2.2 Latent heat based systems

The latent heat storage systems employs the energy absorb/release due to change of phase during which the given medium change from liquid to solid and vice versa. Usually the materials like water, ice, brine or polymers are being used. Latent heat storage has capacity to store large volume of energy compare to its sensible heat counterpart due to very high heat exchange per given unit mass

.

Latent Heat Storage Capacity

Δh – Energy stored during phase change per unit mass for given medium

Q stored = $M*\Delta h$

Phase change thermal storage associated with change of phase to effectively store/release energy. The energy density of latent storage systems is usually higher than its sensible storage systems counterpart. Latent heat storage systems are generally popular due to its capacity to provide higher thermal energy storage with minimum foot print.

Amongst above thermal heat storage techniques, latent heat thermal energy storage is particularly attractive due to its ability to provide high-energy storage density and its characteristics to store heat at constant temperature corresponding to the phase transition temperature of phase change material (PCM). Phase change can be in the any of following forms

- Solid –liquid
- solid–gas
- liquid–gas

and vice versa. During the solid–solid phase change, heat is stored or released as the material is transformed from one crystalline to another. These transitions generally have small latent heat and small volume changes compare to the solid–liquid transitions. Solid–solid PCMs offer the advantages of flexibility in container requirements during the design. [16]

Pentaerythritol (m.p. 188.8 ^{o}C, latent heat of fusion 323 kJ/kg),
pentaglycerine (m.p. 81.8 ^{o}C, latent heat of fusion 216 kJ/kg),
Li2SO4 (m.p. 578 ^{o}C, latent heat of fusion 214 kJ/kg)
KHF2 (m.p. 196.8 ^{o}C, latent heat of fusion 135 kJ/kg) [1].

Solid–gas and liquid–gas transition associate with very high latent heat but they are not considered to be good candidates for thermal storage systems due to the huge volume change during their phase change to the gas state. Usually solid–liquid phase change associates with small volume change and fairly high latent heat exchange during it phase transition. Its volume change easily be contained without any difficulty compare to solid-gas or liquid-gas phase transition. Due to these reasons solid-liquid phase change is proven to be the very promising and economically viable for thermal storage systems compare to it’s all other counterparts. Due to the very low thermal diffusivity, the heat transfer process is very slow and delayed. During the design of heat exchanges for those systems, due attention should be given to mitigate the problem of low thermal diffusivity.

2.2.3 PCM Material

(PCM) Phase change materials can simply be called as ‘‘Latent’’ heat absorb/release materials. They absorb thermal energy when its phase is changed from solid to liquid and release thermal energy during liquid phase is changed to solid. The temperature remains unchanged until the phase is completely changed. Every material change its phase at a specific temperature which depends on the pressure. In general, building services related thermal storage systems are operated under atmospheric pressure. Until any given PCM reaches its phase changing temperature, it absorbs or releases thermal energy and its energy exchange depends on the specific heat of the material and temperature difference. However, for their employment as latent heat storage materials these materials must exhibit certain desirable thermodynamic, kinetic and chemical properties[17]. Moreover, economic considerations and easy availability of these materials has to be kept in mind. The PCM to be used in the design of thermal-storage systems should have desirable thermo physical, kinetics and chemical properties which are as follows;

Thermal properties

(i) Adequately high latent heat.

(ii) Appropriate phase-changing temperature.

(iii) Fairly good thermal diffusivity to ensure proper heat transfer during the phase change.

Physical properties

(i) Favorable phase equilibrium to ensure the system stability.

(ii) High density to assure high capacity at low foot print.

(iii) Small volume change is highly desirable during the design of container.

(iv) Low vapor pressure minimize the problems in containment.

Kinetic properties

(i) No super cooling is greatly ensure the stable operating temperature and system stability.

(ii) Sufficient crystallization rate is necessary to ensure proper phase change which is necessary for desired heat extraction.

Chemical properties

(i) Should not decompose or change its Chemical properties for a long period.

(ii) Should not react with associated construction materials that is used in the thermal storage system.

(iii) Should not be toxic.

(iv) Should not pose any threat of fire hazard.

Classification of PCMs

A large number of PCMs are available in the market which changes its phase at various temperatures. PCMSs could be recognized under three main categories.

- Organic
- Inorganic
- Eutectic

Even though the PCMs available in the market do not meet all the requirements discussed above, in most cases they show stability in their phase changing temperature at the given pressure and the latent heat associated with the phase transition. Due to this reason, the designing engineers of the thermal storage systems need to take every possible step to satisfy the other requirement through a smart design approach. For example if the given PCMs do not exhibit desired thermal diffusivity, designer should increase the number of heat exchanger units submerged inside the PCMs. Inorganic compounds exhibits higher latent heat capacity compare to organic compounds.

Organic phase change materials

Organic materials are further divided into two groups as paraffin and non-paraffin. Organic materials do have a characteristic of stability during repeated melting and freezing without phase segregation and subsequent degradation of latent heat of fusion. They also exhibit self-nucleation which ensure crystallize without super cooling. Further they are non-corrosive in nature which offers flexibility in designing containers for such thermal storages systems.

Paraffin

Paraffin wax is a mixture of straight chain alkenes. During the crystallization of (CH3)-chain, it removes a large amount of latent heat. Usually the melting point as well as latent heat of fusion is increased with the chain length. Paraffin is a good candidate as thermal storage materials due to its availability in a wide temperature spectrum. Paraffin is considered to be safe, reliable and non-corrosive. It also exhibits stability in latent heat of fusion as well as its phase changing temperature. Paraffin wax is chemically inert and does not show considerable volume changes on melting. It also have very lower vapor pressure in liquid form. While having all these favorable qualities, paraffin show few undesirable properties as explained below:

(i) low thermal diffusivity & conductivity
(ii) Poses fire hazards as paraffin is flammable.

There are methods to modify the paraffin wax to mitigate those undesirable characteristics. The storage unit & heat transfer circuit design has a considerable role to play in this context.

Non-paraffin

The non-paraffin organic are the most abundant of the PCM with diverse properties. They have properties very much unique to each and unlike the paraffin's, which have very alike properties. Simply this can be considered as the largest category of PCM group for thermal storages systems. Abhat et al and Buddhi and Sawhney have conducted a very detail survey on organic materials and identified a number of esters, fatty acids, alcohol's and glycol's very much appropriate for thermal storage applications. They can further be classified under fatty acids and other non-paraffin organic. These materials are flammable and should not be exposed to high temperature or flames. Further they should be kept away from oxidizing agents as well. Here are the common characteristics of non-paraffin organic materials:

(i) They do have high heat of fusion
(ii) Cannot exposed to high temperature or flame due to high inflammability
(iii) Low thermal conductivity demands for perfectly designed heat exchangers
(iv) Varying level of toxicity limit the use in various applications

Fatty acids exhibit high heat of fusion values comparable to that of paraffin's. Fatty acids also show stability during repeated melting and freezing and without degradation or super cooling. The general formula describing all the fatty acid is given by CH3(CH2)2n_COOH. Usually fatty acids are qualified as good PCMs. They are expensive than its counterpart technical grade paraffin by 2 to 3 times. They are ideal for low temperature latent heat related applications.

Inorganic phase change materials

Inorganic materials are further classified into two groups as salt hydrate and metallic. These PCM's do not super cool significantly and their heats of fusion do not degrade due to the cycle of melting & freezing.

Salt hydrates

Salt hydrates may be regarded as alloys of inorganic salts and water forming a typical crystalline solid of general formula AB_nH2O. The liquid-solid-liquid transformation of salt hydrates is simply a process of dehydration of hydration of the salt and vice versa. This process resembles the melting or freezing thermodynamically. A salt hydrates usually melts to either to;

$$AB.\ nH_2O \rightarrow AB.\ mH_2O + (n-m)H_2O$$

or

to its anhydrous form

AB.nH2O →AB þ+ nH2O

At the melting point the hydrate crystals breakup into anhydrous salt and water, or into a lower hydrate and water as illustrated above [17]. One problem with most salt hydrates is that of non consistent melting caused by the fact that the released water of crystallization is not sufficient to completely dissolve all the solids phase to release the required amount of thermal energy. Further due to density difference, the lower hydrate (or anhydrous salt) usually settles down at the bottom of the container. Most salt hydrates doe exhibit very poor nucleating properties resulting in super cooling of the liquid which prevent crystallization. Usually these situation are managed by adding nucleating agent, which provides the required nucleion on which the crystal formation is initiated. Even though they are slightly toxic, salt hydrates exhibits very attractive PCM properties [18]:

(i) high latent heat of fusion per unit volume

(ii) Comparatively high thermal conductivity

(iii) Insignificant volume changes on melting.

(iv) Salt hydrates are not corrosive and compatible with plastics

(v) Many salt hydrates are sufficiently inexpensive for the use in storage.

The major problem in using salt hydrates, as PCMs is the most of them, which are judged suitable for use in thermal storage, melts incongruently. As n moles of water of hydration are not sufficient to dissolves one mole of salt, the resulting solution is supersaturated at the melting temperature. The solid salt, due to its higher density, settles down at the bottom of the container and is unavailable for recombination with water during the reverse process of freezing. This results in an irreversible melting–freezing of the salt hydrate goes on decreasing with each charge–discharge cycle. Another important problem common to salt hydrates is that of super cooling. At the fusion temperature, the rate of nucleation is generally very low. To achieve a reasonable rate of nucleation, the solution has to be super cooled and hence energy instead of being discharged at fusion temperature is discharged at much lower temperature. Other problem faced with salt hydrates is the spontaneous of salt hydrates with lower number of water moles during the discharge process. Adding chemicals can prevent the nucleation of lower salt hydrates, which preferentially increases the solubility of lower salt hydrates over the original salt hydrates with higher number of water moles. The problem of incongruent melting can be tackled by one of the following means[18]:

(i) by mechanical stirring

(ii) by encapsulating the PCM to reduce separation

(iii) by adding of the thickening agents which prevent setting of the solid salts by holding it in suspension

(iv) by use of excess of water so that melted crystals do not produce supersaturated solution

(v) by modifying the chemical composition of the system and making incongruent material congruent

2.3 Operational Basics of Cold Thermal Storage: [21]

2.3.1 Modes of Operation

1. Charging storage.

   This involves charging (cooling) the storage when no cooling loads in the system.

2. Simultaneous charging of storage and live-load conditioning.

   Here charging (cooling) the storage is done while the cooling loads of the system are catered. Whatever the excess cooling capacity is used to charge (cool) the storage.

3. Discharging storage.

   Here the cooling load is catered by discharging the storage without operating any refrigeration equipment.

4. Simultaneous discharging of storage and live-load conditioning. The refrigeration equipment are run at their full capacity and whatever the excess demand is served through discharging (cooling) the cold storage

2.3.2 Operational Strategies & cost benefits of Cold Thermal Storage System

The prime objective of the cold thermal storage is to cut down the electrical energy cost by shifting the electrical demand for cooling loads from peak time to off peak period. Full storage for load shifting and partial storage for either load leveling or limiting the demand. These operational strategies are presented in following diagrams. Here the 24 hour period is considered and cheaper electrical tariff periods are utilized to charge the system. The low tariff structure depends on the decision of the electricity companies that is decided based on the demand curve of electrical company.

- Full Storage

  Here the entire cooling load is catered with discharging. In certain cases cooling load and charging of storage will be over lapping for a short period during 24 hour cycle as shown in the diagram.

- Partial Storage
  - Load Leveling

    The chiller operation during the cooling load demand is maintained at a constant level and the excess cooling demand is served through discharging the storage. This will help zeroing the fluctuation of chiller operation and enable designers to cut down the capital expenditure on chillers by reducing the capacity of chillers. The chiller is running at the same capacity during the charging of storage as well as catering the cooling demand in day time.
  - Load Limiting

    The chiller operation during the cooling load demand is maintained at a constant level and the excess cooling demand is served through discharging the storage. This will also zeroing the fluctuation of the chiller operations and enable designers to cut down the capital expenditure on chillers by reducing the capacity of chillers. The chiller is running at the higher capacity during the charging of storage than the actual cooling load demand time.

2.3.3 Measuring of stored cold thermal capacity

Stored cold thermal capacity is expressed in term of Ton-Hours. 1 Ton-Hour = 1 ton of cooling capacity for 1 hour period. Similarly 20 Ton Hours can be represented as any of followings;

- 20 tons of cooling for 1 hour
- 10 tons of cooling for 2 hours
- 5 tons of cooling for 4 hours
- 1 ton of cooling for 20 hours

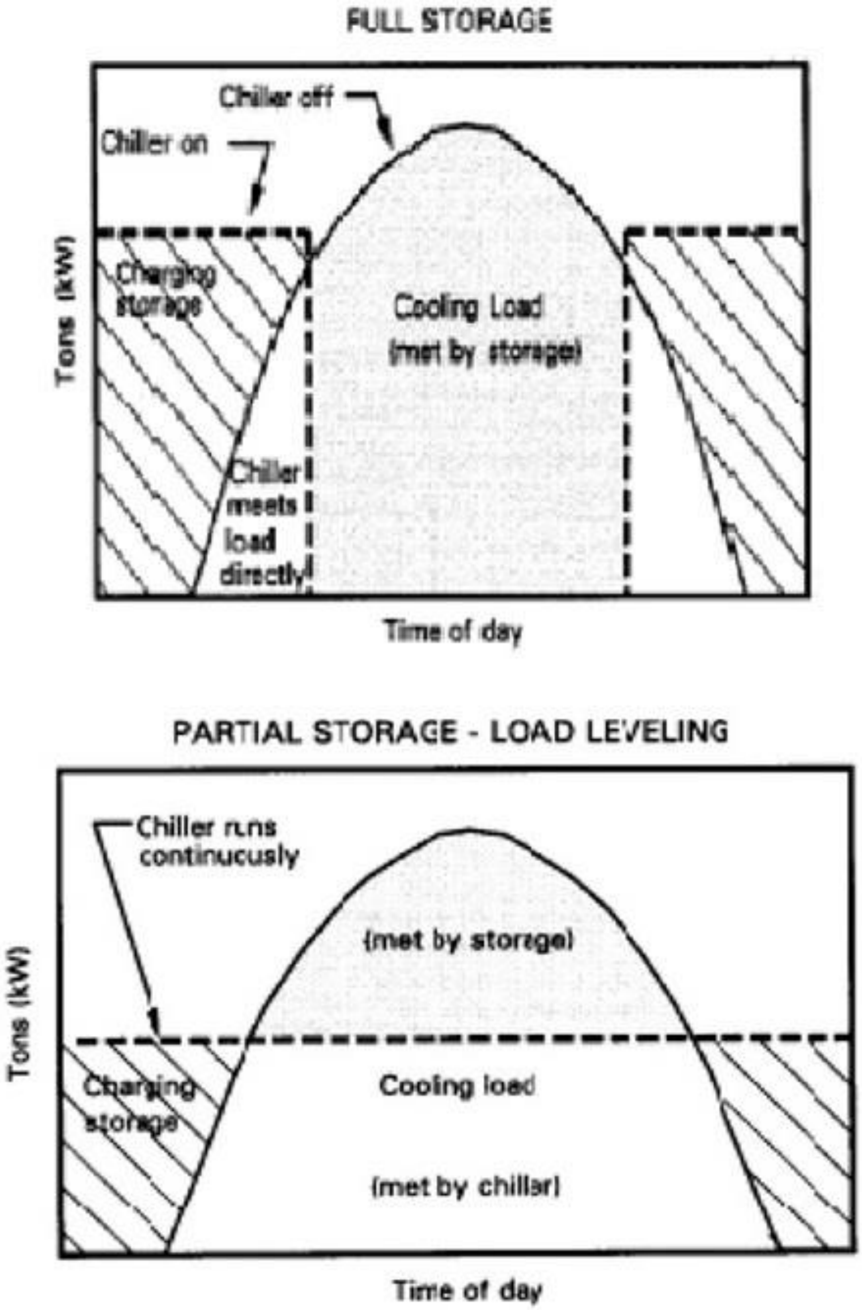

Fig 2.3.3.1 : Full Storage - Load Shifting & Partial Storage - Load Leveling [21]

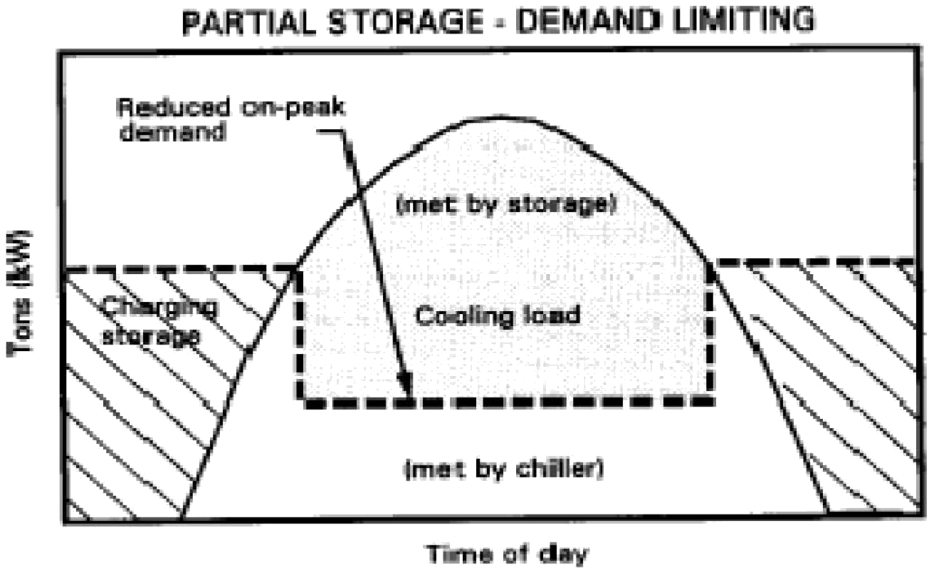

Fig 2.3.3.2: Partial Storage - Demand Limiting [21]

2.4 Applications of cold thermal storage systems

2.4.1 HVAC Cooling

Comfort air conditioning systems are ideal candidates for thermal ice storage. This prevent operation of high capacity chillers during peak electricity demand period of the day and enable them to run during the off peak time slots. This simply shift the electricity energy load from peak to off peak period of the day. The system designers have a prime role to play when selecting the storage sizes and chillers for such systems. In certain cases cooling may require year around while others may need space cooling only during the summer. Typical applications are: office buildings, apartments, hospitals, large retail stores, schools, universities, shopping malls.

2.4.2 Process Cooling

This is a very wide category with numerous applications. For eg:

- Batch cooling where added heat must be removed, during the food processing
- Batch cooling where internal heat is created that must be removed.
- Maintain of constant temperature

2.4.3 District Cooling Plants

District cooling plants for universities or urban areas will have a variety of cooling applications which includes, office buildings, hotels, dormitories, classrooms, labs, sports arenas and data centers etc. Thermal ice storage takes advantage of this diversity of cooling loads and enhances the efficiency of the entire cooling plant.

2.4.4 Developing Energy Sources - Solar & Wind

The technologies for both solar and wind energy are being improved rapidly. These energy sources supply electric energy that can be used throughout the system, without limiting it to cooling. But these energy sources highly depends on the conditions of the weather and fluctuates accordingly. These systems are not considered to be stable due to this high dependency. Thermal ice storage is a perfect complement for these systems. So thermal ice storage / chiller cooling can be tuned and harmonized to offset the effect of any electric load irregularities caused by varying weather conditions.

2.5 Ice Storage Systems

Ice is a common phase change storage material used for cooling. There are several types of ice storage systems are available which includes ice harvesting, external melt ice-on-coil, internal melt-on-ice, and encapsulated ice. Simply ice storage system are suitable to cater larger cooling loads with small footprint of installation due to the fact that higher heat of fusion of ice at the temperature of 32°F or 0 °C for the change of phase plus the safe and desirable properties of water.

2.5.1 Ice Harvesting Systems

Ice harvesting systems consist of an ice producing facility and an ice and cold water storage tank. Often the ice-maker is mounted on top of the storage tank. Water from the storage tank is pumped up to the ice-maker where it flows over the evaporator surface of the refrigeration equipment. The water freezes and sticks onto the evaporator surface. After a predefined period of time during which the ice is building on the evaporator surface, the ice making process is temporarily stopped and reversed which allows the evaporator surface to be warmed and the ice to freely falls into the tank. This is called ice harvesting that gives this technique its name.

While the water is pumped from the tank to the ice-maker, chilled water from the tank is pumped to serve the building load. The returned warmed water is sent back to the tank by flowing over the evaporator. As a far as the water temperature is cold enough, it will make ice at the evaporator, or if water is highly warm, it will be chilled at the evaporator. Either way, the ice or chilled water falls into the tank. The ability of the ice-maker to function as an ice-maker or chiller as needed is important because it allows the system to take advantage of the better efficiency of the chiller mode.

Here are the features of the ice-harvesting system

- The ice-harvester consists of a factory-assembled ice-making plant mounted on top of a site-built storage tank that contains a mixture of ice and chilled water.
- The system can change mode automatically from ice generation to chilled water generation in accordance with load conditions.
- The water in equilibrium with ice in the tank is much colder than the usual chilled water temperature, and this can result in colder air and smaller ducts for the air distribution system.

The ice harvesters has low thermodynamic efficiency compare to the chilling water to the usual chilled water temperature. Also the ice making equipment are relatively expensive than that of the chilled water equipment in any given cooling capacity.

2.5.2 Ice On Coil Systems

2.5.3 Internal Melting

For ice-on-coil systems, the ice is formed on the outside of the coil. Ice is formed on the outside of coils in which glycol solution is circulated. The warn glycol solution circulated through the coil is cooled indirectly by melting ice. Usually the melting of the ice start from the surface of the pipe to the outside, so it is called internal melting.

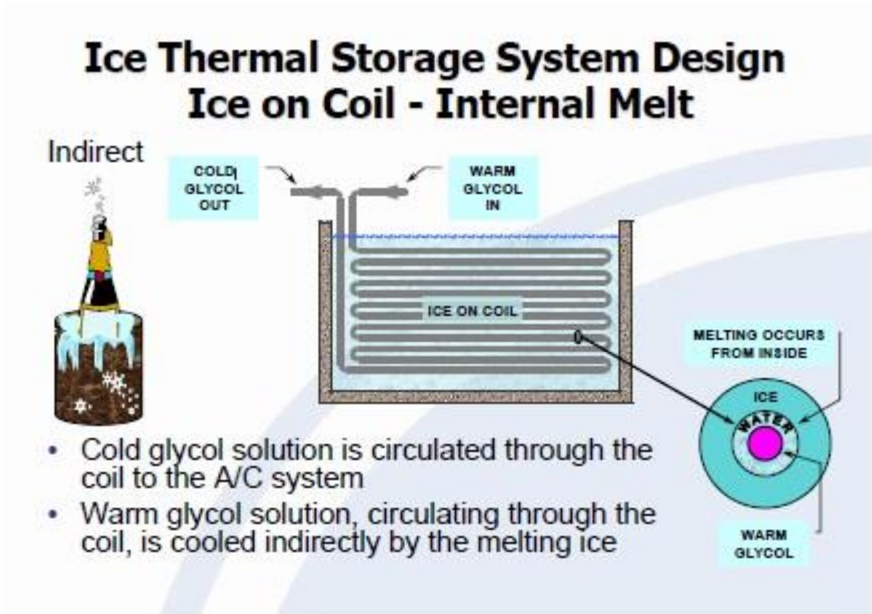

Fig 2.5.3.1: Ice on coil internal melt system. [13]

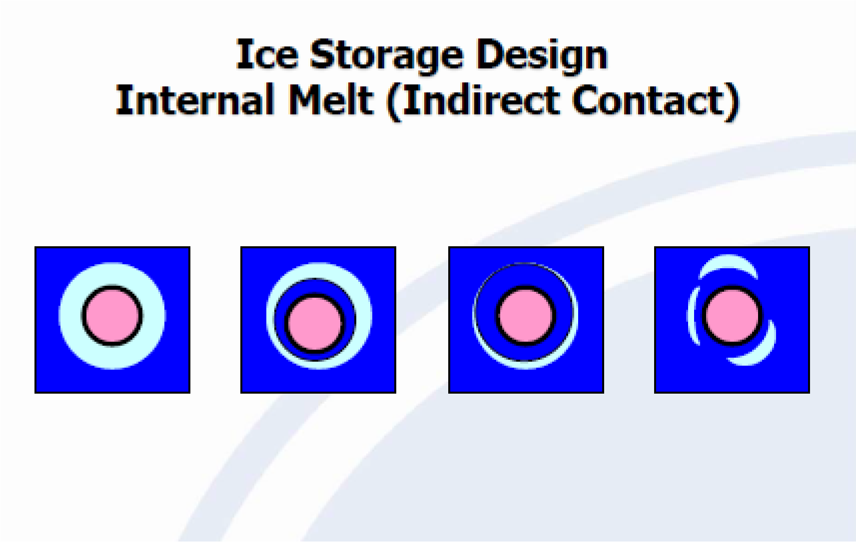

Fig 2.5.3.2. **: Ice Thermal Storage Systems [13]**

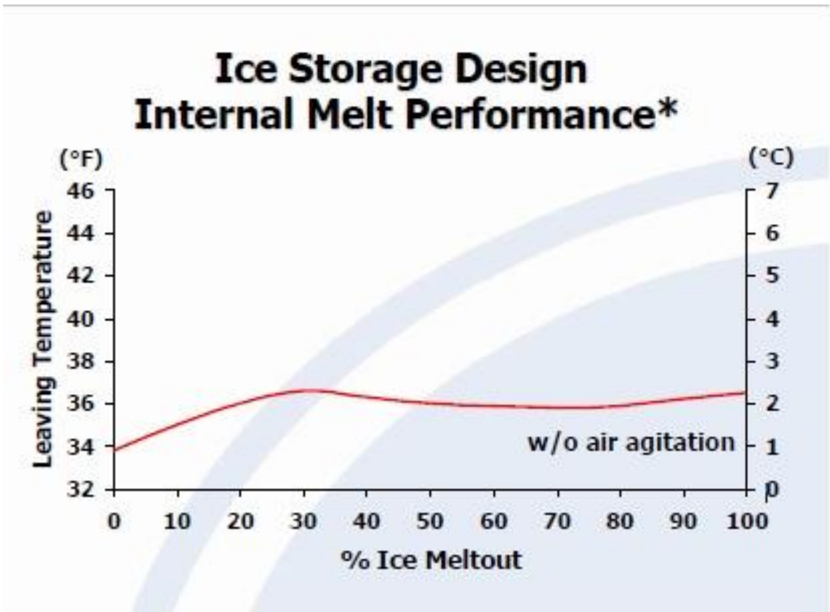

Fig 2.5.3.3 **: Ice Thermal Storage Systems. [13]**

Advantages

- Simple design and easy to operate
  - Simple control for various operating modes
  - Closed pressurized loop
- Stable & cold discharge temperatures
  - 36 ^{o}F to 38 ^{o}F (2.2 ^{o}C to 3.3 ^{o}C)
- Durable steel construction
  - 150 to 300 psi (10.3 to 20.7 bar) design pressure rating

    - Tested at 190 to 375 psi (13.1 to 25.8 bar)
- Flexible layout will encourage modular tanks or vault designs

Disadvantages

- Heat exchanger required for chilled water in building loop
- Not able to discharge as quickly as direct contact cooling due to the fact that ice melt is limited by flow through coil

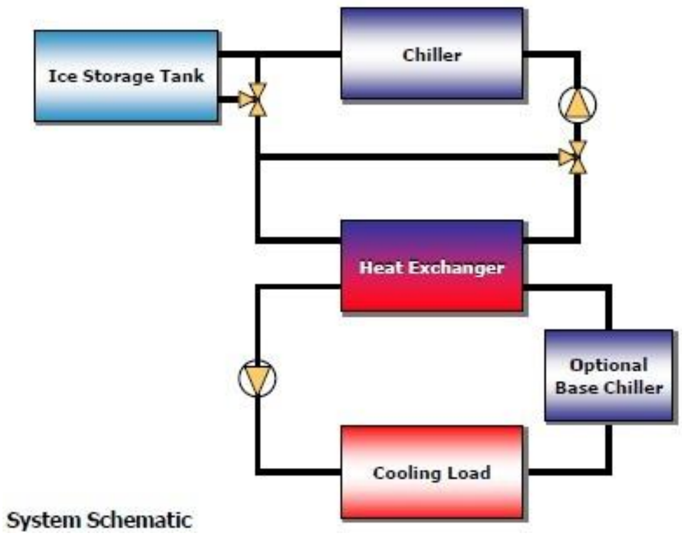

Fig 2.5.3.4: Ice Thermal Storage Systems [13]

2.5.4 External Melting

The separate coiling system with refrigerant is running inside the ice storage to make ice. In ***ice on coil, external melting systems***, the water is circulated through the ice storage tank to the air conditioning system. The warm return water is circulated through the tank will be cooled by the direct contact with ice. Here the warm water running on ice will start melting and transfer the cooling to the warm water. The melting start from the external part of the coiling system as the warm water is in contact with ice made on the coil.

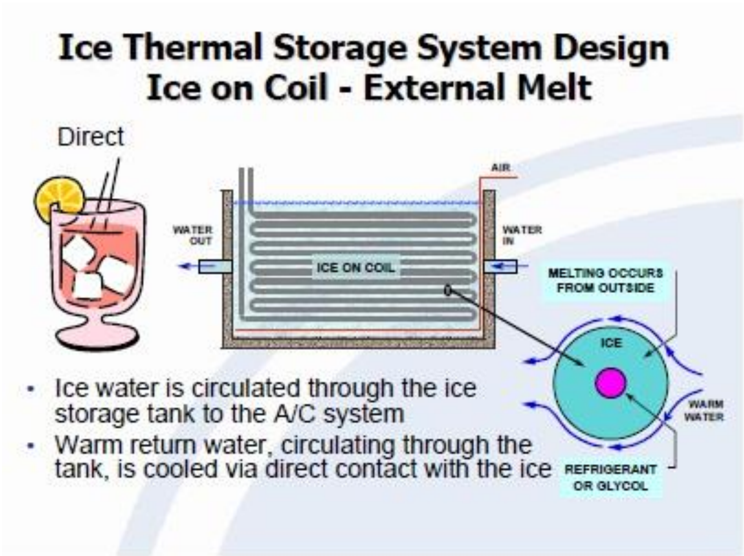

Fig 2.5.4.1: Ice Thermal Storage Systems [13]

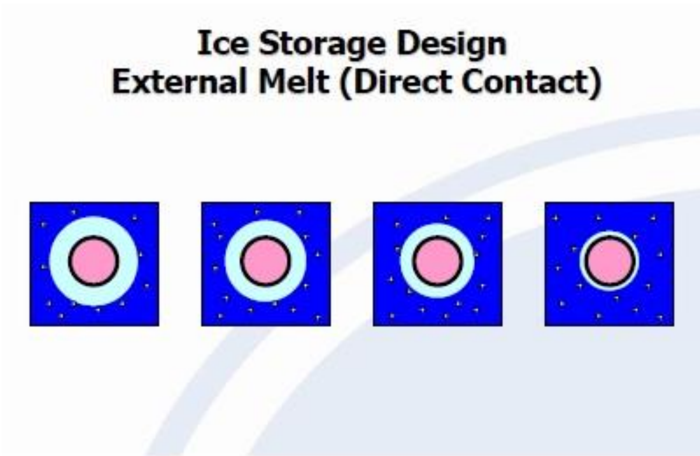

Fig 2.5.4.2 : Ice Thermal Storage Systems [13]

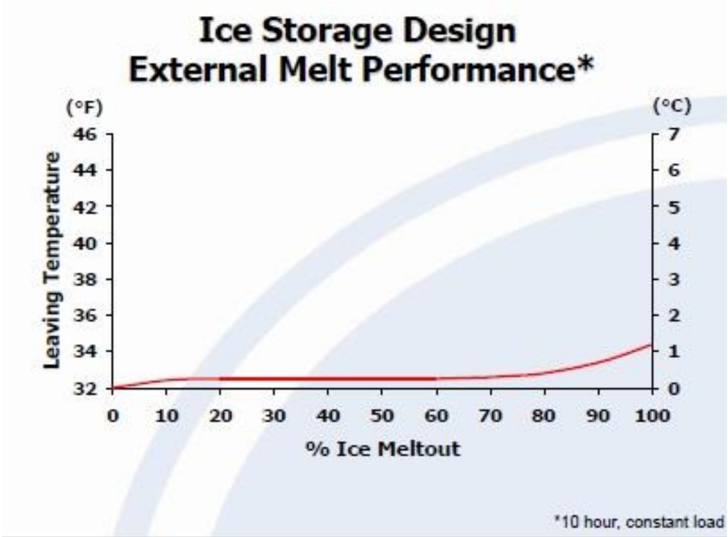

Fig 2.5.4.3 : Ice Thermal Storage Systems [13]

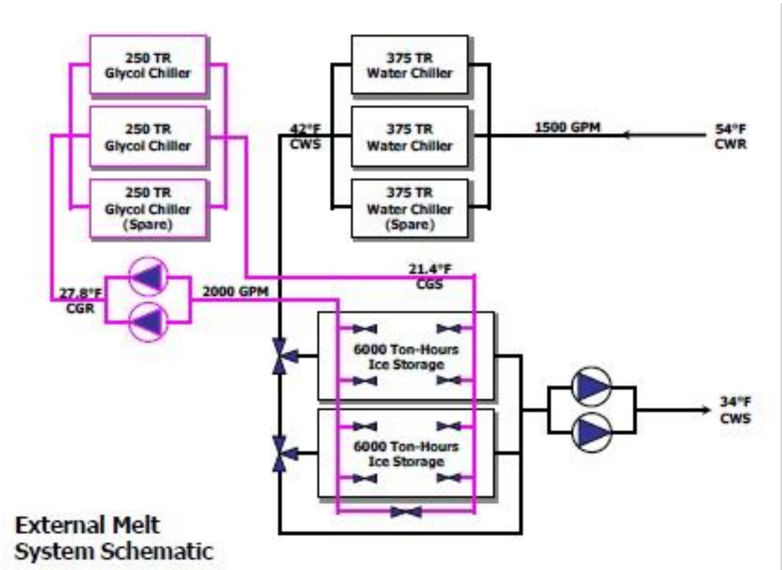

Fig 2.5.4.4 : Ice Thermal Storage Systems [13]

Advantages of External Melting (Direct Contact)

- Low chilled water supply temperature
- Quickest discharge capability
- Eliminate glycol from chilled water loop

Disadvantages of External Melting (Direct Contact)

- Chilled with lower temperature capability generally required
- Glycol control valves required on larger systems
- Heat exchanger may require to manage static head of open system

- More difficult to monitor amount of ice in inventory

2.5.5 Cost Benefits of Ice Storage Systems

Thermal energy storage is the most appropriate answer to the shifting of electrical loads from peak to off peak period. Smart engineers apply the thermal ice storage concept to cooling applications. Buildings of small elementary schools to large university complexes, commercial buildings, hospitals, arenas plus district cooling plants are common candidates for ice storage based air conditioning systems. When the thermal ice storage based air conditioning systems are analyzed in detail by giving due focus to the engineering economics, it becomes obvious that thermal ice storage offers many additional installation, operational and cost benefits to the owners.

The chiller operates during non-peak hours cooling a glycol solution to sub-freezing temperatures which is then circulated through the ice storage coils to build the ice storage.

Ice forms around the external surfaces of the coils (in which the glycol is circulated), and a full storage capacity depends on the size of the ice storage container, glycol temperature, distance between adjoin glycol circuit and number of glycol circuits. The ice is ultimately melted and used as a cooling agent.

The most significant benefit of thermal ice storage is the reduction of on-peak electric demand and the shift of energy use to non-peak hours. However, there are additional benefits that may not be as obvious to those designing a thermal ice storage system for the first time. Depending on the method of ice melt, the cooling fluid temperature can be substantially reduced when compared to a conventional chilled water system. Leaving temperatures are often as low as 34°F to 36°F (1.1°C to 2.2°C) in thermal ice storage based systems[19]. This colder fluid offers many design benefits which lead to reduce the overall system's capital investment while improving HVAC and process cooling performance. The benefits of the colder fluid are often unnoticed when evaluating thermal ice storage systems. These benefits include:

2.5.6 Capital cost savings

A. Reduced chilled water distribution flow:

Colder supply water allows the engineers to increase the system delta-T without effecting the performance of the system. Thermal ice storage systems typically are designed with an 10.0°C to 11.1°C delta-T distribution loop while conventional systems maintain 5.53°C to 6.63°C delta-T . The 20°F (11.1°C) versus the 12°F (6.63°C) delta-T simply reduce the chilled water flow approximately 35%, thus reducing the required distribution pipe sizes and pumping energy [19].

B. Reduction of chilled air distribution flow;

The colder distribution fluid will allow the designer to use a larger air side delta-T as well. Conventional air-conditioning systems are designed for a leaving air temperature of 55°F (12.8°C) that result in a 50% relative humidity in the conditioned space. The colder fluid temperature available from the thermal ice storage system would allow for very cold air distribution temperatures (45°F) (7.22°C). Designing for the lowest possible air temperature will provide the most first cost and energy cost savings. However, these low temperatures would require a very tight (no leak) duct system, extra insulation and special terminal units. A more conservative 51°F (10.6°C) still offers considerable cost savings and results in 40% - 45% relative humidity in the conditioned space. By lowering the supply air temperature 4°F (55°F to 51°F) (2.2°C)(12.8°C - 10.6°C), the air flow can be reduced by 20%. This results in a corresponding 20% reduction in the distribution duct sizes. The cost of the duct system is often considered to be the largest single HVAC construction expense, and the first cost savings will be significant[19].

C. Reduced fan motors and pump motors sizes:

Designs using colder fluids and larger delta-Ts reduce air and water flow requirements. Therefore, fan and pump sizes, including the electric motor size, can be minimized, thus capital investment for equipments can be reduced and saved by significant percentage. Reduced electrical distribution cost - Smaller components such as chillers, fans and pump motors reduce the connected system's energy consumption capacity (kW) and save the electrical distribution costs which may include the building's main transformer cost plus all the way to the starter panels. Reduced connected horsepower (kW) can also reduce the size of any emergency generation equipment that may be required[19].

D. Building space savings:

Smaller supply and return ducts require very less floor area compare to its conventional counterpart. The result is less construction costs and more usable floor space[19].

2.5.7 Long term benefits in ice storage systems

A. The combined electric demand of the fan and pump motors often exceed the electric demand of the chiller system. Low temperature fluid distribution designs will reduce the pump horsepower (kW) by approximately 40%. Low temperature air distribution designs will reduce the fan horsepower (kW) by approximately 50%. This is a substantial electric demand and energy use reduction that will provide operating energy cost savings for the life of the system.

B. The chillers will always operate at or near 100% capacity when charging the storage system and typically at a higher capacity factor when compared with a conventional chilled water system during the daytime cooling. This reduces the amount of time chillers will operate at low load conditions, as well as on / off cycling based on cooling loads. When chillers are operated at or near full capacity most of the time, they require less maintenance. Any fluctuation in the cooling load will be satisfied by the ice storage.

C. Process operations often have high cooling spikes of short duration. Chillers are not designed to load and unload quickly. Thermal ice storage can respond quickly and efficiently to load variations and can track rapid cooling spikes.

D. Utility Electric Peak rate periods vary considerably. Demand periods may be 8 -10 hours or a series of shorter 2-3 hour segments. Thermal ice storage is very effective with any utility time-of day demand schedule and can be easily modified should rate periods change in the future.

Today, there are thousands of thermal ice storage systems installed and operating world wide and many of these have taken advantage of the benefits offered from lower temperatures.

2.5.8 Encapsulated Phase Change Materials & Ice storage systems

An encapsulated PCM storage system consists of sealed plastic containers of water immersed in a tank of secondary coolant, such as ethylene glycol. To charge the storage, cold coolant is circulated through the tank and the encapsulated PMC in the containers freezes. To discharge the storage, warm ethylene glycol is circulated through the tank and the encapsulated PMC will change its phase by absorbing the energy in the warm ethylene glycol. The encapsulated PCM products has the ability to accommodate the volume increase when phase is changed from solid to liquid. Same way the alternatively encapsulated to ice is also used. Plastic containers, filled with de –ionized water and ice-nucleating agent, are immersed in a secondary coolant ethylene glycol solution in a steel tank. The ice is charged and store when the secondary coolant is at a temperature between -6 ^{0}Cand -3 ^{0}C circulated through the tank. Ice is melted when the warm coolant returned from AHU is circulated through the tank. Chiller can also provide direct cooling at a coolant temperature from 2 ^{0}C to 6 ^{0}C. An encapsulated ice storage system consists of the following components. Chillers, steel tank, encapsulated containers, pumps, air system controllers, piping and accessories. Encapsulated ice balls and typical encapsulated ice configuration are shown in following figures.

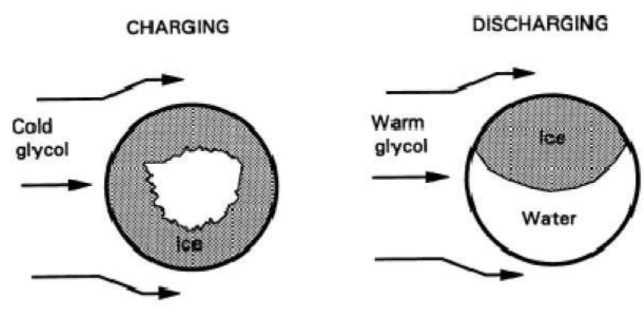

Fig 2.5.8.1 : Phase changing within encapsulated ice balls [13]

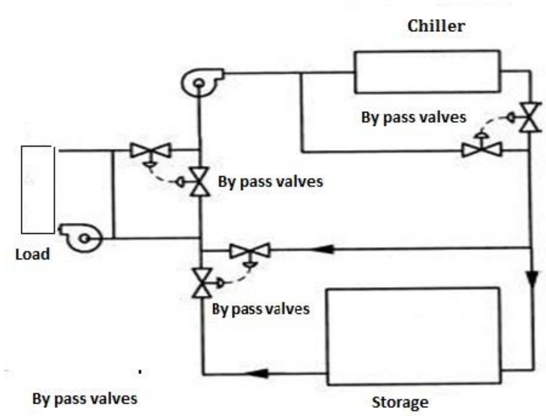

Fig 2.5.8.2 : Equipment configuration of encapsulated PCM/Ice storage system [13]

2.6 Design Guideline for Cold Thermal Storage Systems

2.6.1 ASHRAE Design guidelines for Cold Thermal Storage Systems [20]

ASHRAE (American society of Heating, Refrigeration and Air conditioning Engineers) has published "Design guide for Cool Thermal Storage" in response to the growing need of complete and comprehensive cold thermal design guide for engineers to evaluate the potential of cold thermal storage systems for specific applications, to select the most appropriate cool storage technology for the given application and to design, implement and commissioning a successful cool storage technologies. This guideline further describes a rational procedure for designing cool storage systems based on various early research work carried under this technology. This design guide covers the following areas.

- Basic background of cold storage concepts and its associated technologies.
- Fundamental theory and design considerations.
- Comparison of various currently available storage technologies.
- Step by step design process.

This guide line further explains how to evaluate the valuation of cool thermal system from engineering economics perspective for a given application based on the capital cost, operational cost and return on investment with that of the non storage HVAC systems. The step by step explanation provides an excellent guide for system designers which include followings;

1. Calculating load profiles
2. Screening initial economics
3. Selecting storage type
4. Selecting operating strategy
5. Determining storage interface parameters
6. Sizing cooling plant and storage
7. Evaluating economics
8. Finalizing design
9. Commissioning

As each application is unique and has its own characteristics, the design process will not necessarily be conformed to the sequence of steps given in this guide. In practice, the design procedure may pass through each step but in different. As a common practice, in real world, the intermediate design steps will be repeated more than one times for the sake of comparison or improvement of the accuracy. The complete cool storage design process includes development of specifications for equipment, installation, startup, commissioning, and operation. The end point is not reached until the system meets or exceeds its intended design performance. The commissioning provides a framework for integrating all the phases from pre design through the first year of operation.

The design guide recommends to calculate the design load profile for the applications using the methods available in this manual and insist to have a accurate load profile, whether it is a hypothetical or actual load. The validity of all subsequent engineering calculations and design decisions, as well as the successful implementation, commissioning and operation of the cool storage system, entirely depends on the accuracy of the calculated load profile. The selection of design weather conditions has a big impact on calculation of load profiles. This has an influence to calculation of load profiles, as well as equipment performance. These are discussed in details under the chapter 10 of the design guide.

The engineering economics of cold thermal storage is given in clause 10.7 of Chapter 10 with following highlights.

(1) When system design parameters and equipment sizes have been determined, detailed first cost and operating cost estimates can be developed. First cost estimates are developed using the same procedures as for non-storage systems.
(2) Cost estimates for storage tanks recommended to obtain from manufacturers and site-built tanks estimation should be obtained from contractors experienced in the construction of tanks for cool storage or similar systems.
(3) The operating cost estimates can be done with varying levels of detail and accuracy. In some cases, simple on peak demand basis estimation is adequate. If necessary detailed an hour-by-hour analysis including energy consumption of chillers and other equipment can be calculated and established. Usually the operating cost saving is from reduction in demand during the peak period time and as well as the drop of peak demand cost. If the saving is calculated for on day basis, this can be repeated for the year and the estimated load profile with reduced load capacity could be obtained.
(4) For certain cases , when the difference between the peak and off peak energy cost are high, more saving can be expected from running chillers during off peak period.
(7) Estimates of energy savings can be developed based on the design operating profile developed for detailed sizing. Energy savings for a single storage cycle are calculated from the applicable utility rates and from the differences in on- and off-peak energy consumption between the storage system and the non-storage base case.
(8) Energy savings can often be represented by the number of ton-hours per day shifted from on-to off-peak periods. It may be assumed that the energy savings for the peak day is a fully achieved on any given day if the on-peak cooling load capacity is lesser than or equal to the storage capacity. During lower load days the complete on-peak load is shifted and saving is fully proportional to the peak load. Estimation done by this method is not fully account for the variations in part load efficiency of non-storage chillers. The annual energy savings can be assessed by repeating detailed daily comparisons of storage and non-storage performance over a given range of load and ambient temperature conditions.

2.6.2 How to decide whether Ice Storage or Chilled Water Storage is suitable for a given project.

2.6.2.1 Introduction

One challenge that plagues professionals managing large facilities, from k-12 schools, colleges and offices to medical centers, stores, military bases and data centers, is finding a more cost-effective, environmentally friendly strategy for using and consuming energy. This problem is compounded by the fact that the cost of electricity is at its highest during the day, when demand for power is at its peak. As a result, storing less expensive off-peak electricity has been a viable solution for many firms and institutions.

The National Institute of Building Science noted that nearly 40 percent of the electricity consumed by commercial facilities in the United States is dedicated toward heating, ventilation and air conditioning systems. As a result, the resource emphasized that addressing the costs associated with cooling a facility has huge potential for generating cost-savings.

2.6.2.2 Cool storage

Cool storage offers a reliable, cost-effective means of controlling electricity costs while ultimately helping to limit greenhouse gas emissions. The technology allows facilities to take advantage of less costly electricity available at night and functionally save that energy for use at a later time. Cool storage achieves this performance by using ice or chilled water as a medium for storing and deploying energy.

The right time to decide on cool storage varies, but may be most attractive when at least one of the followings is present:

- It’s time to invest in a chiller plant.
- Back-up cooling is needed.
- Increased cooling loads
- Reaching electrical service limit
- Limited space for fans and ducts require colder air temperature applications.
- Storage of water for fire protection is required.

- The facility is seeking to reduce its environmental impact.
- The building has a poor load factor (peak load is higher than the average load).

2.6.2.3 Chilled Water and Ice Storage each offer unique benefits

Each facility has its own unique energy needs and challenges to contend with. That's why there are multiple strategies for cool thermal energy storage solutions on the market. Ice storage and chilled water make up the two most prominent strategies available - taking a closer look at the advantages of each strategy will reveal which application is the best fit for an organization interested in deploying energy storage.

Ice Storage and Chilled Water have plenty in common. Both are reliable energy storage solutions that have been deployed for years, and both are capable of making it easier for facilities to efficiently operate their cooling systems. Both have superior benefits over traditional cooling. They offer energy costs savings, back up cooling capacity, may extend an existing system capacity at less cost than conventional non-storage, make renewable more viable, reduce greenhouse gas emissions and lower transmission line losses. Up to 14 percent of the electricity traveling across the grid can be lost by inefficiencies in the system, especially during the summer daytime periods. Storing ice at night puts less pressure on the grid - the system is most vulnerable during peak demand periods when an entire regions attempt to draw power at the same time.

The two technologies also have a long list of differences with regards to the installation process, design parameters, expected long-term maintenance and operating cost-savings. Facility managers should think through a few of the following questions as they evaluate the advantages that each cool thermal storage technology can bring to the table.

Markets- Both chilled water and ice storage work for large facilities such as schools, hospitals and offices. If the building has loads with a very short duration (30 minutes to 2 hours); then Chiller Water storage may be a better choice due to the quicker discharge rates. Data Centers and very large (6,000 tons and up) systems would be a great fit for Chilled Water storage. Whereas, ice storage would be a better applied on large and taller multi-story buildings Chilled water storage tanks operate at atmospheric pressure, complicating the interface with pressurized building piping.

Site Locations - Ice storage may be buried, put on roofs, placed indoors, in the basements or outdoors. Chilled Water storage however may not be applied on roofs due to its mass and largely due to atmospheric pressure limits.

Expansions and Permanence of Installation – If so, it makes sense to compare the installation permanence of Ice Storage and Chilled Water applications. Ice storage tanks like CALMAC's Ice Banks are modular and re-deployable, making it a simple task to change their location with respect to the needs of the business while conveniently staying as a permanent structure for the life of the system if needed. Modular ice storage tanks can be easily added to an existing ice storage facility.

A Chilled Water tank, on the other hand, is only designed to be a permanent structure and may be considered if storage locations are expected to stay consistent for the foreseeable future.

Retrofits – Most screw and scroll chillers can produce ice making temperatures with very few exceptions. If the facility only has centrifugal compressors that have NOT been designed to produce ice making temperatures and has no plans to add new chillers, the energy storage design would lean towards Chilled Water storage. Any type chiller can be used with Chilled Water storage including existing centrifugals.

Tank Size - Chilled Water storage tanks are significantly larger than those used for ice applications like CALMAC's IceBank tanks. Typically, Chilled Water installations are sized anywhere from 8 to 10 times larger than ice tanks, often approaching 30' or taller in height. By comparison, CALMAC's IceBank tanks stand at just 8 ½' tall, providing much more flexibility during the installation process. The shorter tank size is preferable for indoor applications as well as outdoors.

Short and Long-Term Value – The first costs of ice and water systems should be compared. In smaller (< ~ 10,000 tons), ice storage can be installed for less $/ton-hour than water storage. With larger installations, water storage can be less expensive. Once at the end of the product life cycle, large water storage tanks can be a stranded asset, ie not used at another location. Ice storage may be reused and installed at different facilities.

Performance and Reliability - When selecting a company, it is necessary to select one with diverse portfolio of projects they have completed to date which makes it simple for building project implementers to identify a frame of reference and estimate the potential advantages of thermal storage applications.

Ease and Cost of Maintenance - Building managers are tasked with keeping HVAC systems performing at optimal levels, and will be responsible for keeping an eye on the organization's energy storage applications as well. The simpler a system is the better, as there are fewer opportunities for components to age or become damaged. Also, a company with a single source provider eases not just the purchase but any commissioning and maintenance.

In the case of ice thermal storage, readymade storage tanks are commonly available from various credible suppliers from around the world. To achieve this goal, the system separates water meant to store cooling from the volume dedicated to cooling the building. Chilled Water storage tanks instead utilize the same volume of water to store and cool. The result is a system that requires 8 to 10 times the water and significantly more treatment to maintain than ice storage. Other maintenance considerations may include the number of tanks and size. With one tank, if the tank is unusable due to maintenance or repairs, all of its stored energy is unavailable. Secondly, in larger tank sizes, leaks may be difficult to find or repair.

System Design – Chilled Water storage systems operate the ice and chiller at the same time with flow for charging and discharging traveling back and forth within the same piping. This flow reversal for charge/discharge is awkward and may be complex to design. Whereas, in an ice storage system flow travels in only one direction and series configurations are possible allowing for flexible configurations to take advantage of utility rate structures.

For systems designed for a larger delta T, ice storage will offer the greatest benefit. Low temperature coolant from ice systems can provide wider delta T's, increasing pumping efficiency. Installed costs for piping and pumps are significantly less due to less flow and extra energy used to make the ice. Plus extra energy needed to make ice may be offset by less pumping energy. In a Chilled Water storage system, low ice temperatures are not feasible, however wide delta T's are still possible. For optimal efficiency it is critical to maintain system delta T at all load levels. Any water that returns to tank below design temperatures represent lost storage capacity.

Two sides of the same coin

Both Ice and chilled water storage have their merits. Chilled Water storage can be incorporated into a fire protection system and is more suitable for data centers due to fast discharge rates. Also, Chilled Water storage plants don't need to produce ice making temperatures, so existing centrifugal machines that aren't ice ready can be used. On the other hand, Ice storage arrives factory assembled and installs in far less time than Chilled Water storage. In larger installations, CHW storage can be less expensive, even though it takes longer to install. Ice storage pays back in as little as 3-5 years, requires less space, is modular, reusable and simple to maintain and control. You have the flexibility to run the ice and chiller at the same time or run just one or the other and at the end of the system life, it's a non-stranded asset.

Yet despite all their differences, no matter which solution is chosen, cool storage shares many powerful benefits and fundamentally changes how and when facilities draw electricity from the grid. More efficient lights and equipment, day lighting and fans are all great ways for facilities to improve efficiency however, reducing peak demand by storing energy for when you need it is just as important. In fact, cool storage is such a novel solution because it allows building managers to fundamentally change how and when the facility draws electricity from the grid, slashing cooling costs and leveling loads for greener, smarter buildings.

2.7 Mathematical model of ice forming inside a horizontal rectangle chamber with constant temperature at the top & bottom walls.

Reference to the paper published by P.Bhargavi, Dr. Radha Gupta & Dr. K.Rama Nirasimha, a mathematical model was established on forming of ice inside a closed enclosure. The problem of phase change could be analyzed as moving boundary problems which relates to the formation of ice in the water. Here the top & bottom surface temperatures of rectangle enclosure filled with water is maintained at constant value. When this surface temperature is quickly raised & maintained at a value which is below the freezing temperature of the water, the heat is transferred from the water inside, to the top & bottom surfaces due to the temperature difference. The water at the top & bottom surfaces inside the enclosure will start reducing its temperature until it reaches the freezing point and thereafter the ice forming is started.

In order to make a mathematical model for this case following approach is considered. Here only one surface is considered for heat transfer analysis.

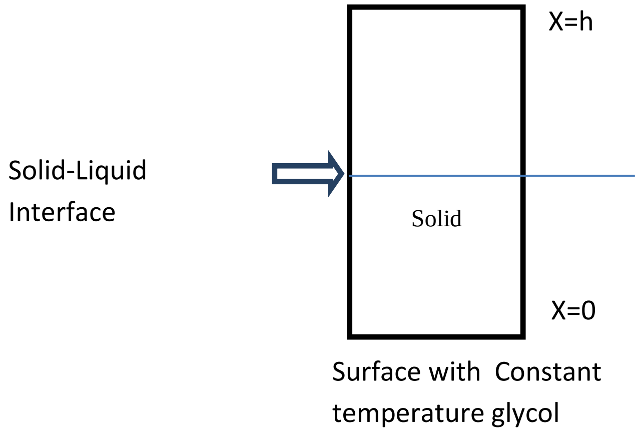

Fig 2.7.1
Diagram for developing mathematical model for heat transfer during ice making

This model concerns the phase changing process and it involves physical properties for each phase separately. The following assumptions are made during the development of the mathematical model for this case.

1. Law of conservation of energy is satisfied.
2. The thermo-physical systems is inside a closed system.
3. The values of different thermo physical properties do not change significantly during the phase changing process.
4. The initial temperature of water is known
5. All the thermo physical properties are continuous functions of time including the phase change, and are valid for entire temperature range associated during the freezing.

6. Heat is transferred isotopically by conduction and omits a possible heat transfer by convection and radiation.
7. Neither super cooling effects nor the gravitational, elastic, electromagnetic effects are considered.

The governing equations for heat transfer encompassing the initial and boundary conditions are defined as follows;

Consider solid region

$$k_s \frac{\partial^2 T_s}{\partial x^2} = \rho_s c_s \frac{\partial T_s}{\partial t} \quad 0 \le x < y$$ Eq. 2.7.1

Consider liquid region

$$k_l \frac{\partial^2 T_l}{\partial x^2} = \rho_l c_l \frac{\partial T_l}{\partial t} \quad y < x \le h$$ Eq. 2.7.2

Boundary conditions are given by:

$$T_s = T_b \text{ at } x = 0$$

$$\frac{\partial T_l}{\partial x} = 0 \quad \text{at } x = h$$

The rate at which the building of ice is determined by the energy balance at the ice – water interface. The energy balance is expressed with using the following equation

At the solid – liquid interface, the conditions are as follows;

$$T_s = T_l = T_m; \quad \text{at} \quad x = y$$

$$\rho_s H_s \frac{\partial Y}{\partial t} = k_s \frac{\partial T_s}{\partial x} - k_l \frac{\partial T_l}{\partial x} \quad x = y$$ Eq. 2.7.3

Here the boundary conditions are

$$T_s = T_b \text{ at } x = 0$$

$$\frac{\partial T_l}{\partial x} = 0; \text{ at } x = h$$

The initial condition is

$T_l(x,0) = T_m$ at t = 0

After introducing non dimensional as follows;

$$\theta = \frac{T - T_b}{T_m - T_b}$$ Eq. 2.7.4

$$X = \frac{x}{h}$$ Eq. 2.7.5

$$S = \frac{y}{h}$$ Eq. 2.7.6

$$\tau = \frac{\alpha_s t}{h^2} = \frac{k_s t}{\rho_s c_s h^2}$$ Eq. 2.7.7

$$Ste = \frac{c_s (T_m - T_b)}{H_s}$$ Eq. 2.7.8

After substitution of these quantities following final equations are obtained

For solid region

$$\frac{\partial^2 \theta_s}{\partial X^2} = \frac{\partial \theta_s}{\partial \tau} \quad 0 \le X < S$$ Eq. 2.7.9

For liquid region

$$\frac{\partial^2 \theta_l}{\partial X^2} = \frac{\alpha_s}{\alpha_l} \frac{\partial \theta_l}{\partial \tau} \quad S < X \le 1$$ Eq. 2.7.10

Boundary conditions change to

$$\theta_s = 0 \qquad \text{at } X = 0$$ Eq. 2.7.11

$$\frac{\partial \theta_l}{\partial X} = 0 \qquad \text{at } X = 1$$ Eq. 2.7.12

The initial conditions changes to

$$\theta_l = 1; \quad \text{at} \quad \tau = 0$$ Eq. 2.7.13

The solid-liquid interface condition become

$$\theta_s = \theta_l = 1; \text{at } X = S$$ Eq. 2.7.14

$$\frac{1}{Ste}\frac{\partial S}{\partial \tau} = \frac{\partial \theta_s}{\partial X} - \frac{k_l}{k_s}\frac{\partial \theta_l}{\partial X} \text{ at } X = S$$ Eq. 2.7.15

$$\int_0^s \frac{\partial \theta_s}{\partial \tau} dX = \int_0^s \frac{\partial^2 \theta_s}{\partial X^2} dX = \frac{\partial \theta_s}{\partial X}|_s - \frac{\partial \theta_s}{\partial X}|_0$$ Eq. 2.7.16

$$\int_s^1 \frac{\alpha_s}{\alpha_l}\frac{\partial \theta_l}{\partial \tau} dX = \int_s^1 \frac{\partial^2 \theta_l}{\partial X^2} dX = \frac{\partial \theta_l}{\partial X}|_1 - \frac{\partial \theta_l}{\partial X}|_s$$ Eq. 2.7.17

From Eq. 2.7.16 and Eq. 2.7.17,

$$\frac{\partial \theta_s}{\partial X}\Big|_s = \int_0^s \frac{\partial \theta_s}{\partial \tau} dX + \frac{\partial \theta_s}{\partial X}|_0$$ Eq. 2.7.18

$$\frac{\partial \theta_l}{\partial X}|_s = \frac{\partial \theta_l}{\partial X}|_1 - \int_s^1 \frac{\alpha_s}{\alpha_l}\frac{\partial \theta_l}{\partial \tau} dX = -\int_s^1 \frac{\alpha_s}{\alpha_l}\frac{\partial \theta_l}{\partial \tau} dX$$ Eq. 2.7.19

Substituting,Eq. 2.7.18 and Eq. 2.7.19 in Eq. 2.7.15

$$\frac{1}{Ste}\frac{\partial S}{\partial \tau} = \int_0^s \frac{\partial \theta_s}{\partial \tau} dX + \frac{\partial \theta_s}{\partial X}|_0 + \frac{\rho_l c_l}{\rho_s c_s}\int_s^1 \frac{\partial \theta_l}{\partial \tau} dX$$ Eq. 2.7.21

Using the relation $\frac{\partial \theta}{\partial \tau} = \frac{\partial \theta}{\partial S}\frac{\partial S}{\partial \tau}$ Eq. 2.7.22

Eq. 2.7.21 , changes to

$$\frac{1}{Ste}\frac{\partial S}{\partial \tau} = \left[\int_0^s \frac{\partial \theta_s}{\partial S} dX + \frac{\rho_l c_l}{\rho_s c_s}\int_s^1 \frac{\partial \theta_l}{\partial S} dX\right]\frac{\partial S}{\partial \tau} + \frac{\partial \theta_s}{\partial X}|_0$$ Eq. 2.7.23

$$\frac{\partial S}{\partial \tau}\left[\frac{1}{Ste} - \int_0^s \frac{\partial \theta_s}{\partial S} dX - \frac{\rho_l c_l}{\rho_s c_s}\int_s^1 \frac{\partial \theta_l}{\partial S} dX\right] = \frac{\partial \theta_s}{\partial X}|_0$$ Eq. 2.7.24

The linear approximation $\theta_s = a + bX$ gives

$$\int_s^1 \frac{\partial \theta_l}{\partial S} dX = 0$$ Eq. 2.7.26

The interface velocity as

$$\frac{\partial S}{\partial \tau} = \frac{2Ste}{(2+Ste).S}$$ Eq. 2.7.27

2.1 cubic Approximation

Let the cubic approximation for the temperature distribution in the solid region be

$$\theta_s = a + bX + cX^3$$ Eq. 2.7.28

Using the boundary conditions

When $X = 0, \theta_s = 0$ and

When $X = S, \theta_s = 1$

$a = 0$

$$bS + cS^3 = 1$$ Eq. 2.7.29

From Eq. 2.7.15

$$\frac{1}{Ste}\frac{\partial S}{\partial \tau} = b + 3cS^2$$ Eq. 2.7.30

Substituting Eq.2.7.27 in Eq. 2.7.30

$$bS + 3cS^3 = \frac{2}{(2+Ste)}$$ Eq. 2.7.31

Solving Eq.2.7.29 and Eq.2.7.31

$$b = \frac{(4+3Ste)}{2(2+Ste)S}$$ Eq. 2.7.32

$$c = -\frac{Ste}{2(2+Ste)S^3}$$ Eq. 2.7.33

Substituting

a = 0

Eq. 2.7.32 and Eq. 2.7.33 in

Eq. 2.7.28

$$\theta_s = \frac{(4+3Ste)}{2(2+Ste)}\left(\frac{X}{S}\right) - \frac{Ste}{2(2+Ste)}\left(\frac{X}{S}\right)^3$$ Eq. 2.7.34

From Eq. (29)

$$\frac{\partial \theta_s}{\partial X}\Big|_0 = \frac{(4+3Ste)}{2(2+Ste)}\frac{1}{S}$$ Eq. 2.7.35

$$\frac{\partial \theta_s}{\partial S} = -\frac{(4+Ste)}{2(2+Ste)}\frac{X}{S^2} + \frac{Ste}{2(2+Ste)}\frac{3X^3}{S^4}$$ Eq. 2.7.36

Substituting Eq. 2.7.34 and Eq. 2.7.35 in Eq. 2.7.24

$$\frac{\partial S}{\partial \tau} = \frac{4Ste}{(4+Ste)}\frac{1}{s}$$ Eq. 2.7.37

Integrating Eq. 2.7.37

$$S = \left(\frac{8Ste}{(4+Ste)}\right)^{1/2}\tau^{1/2}$$ Eq. 2.7.38

From Eq. 2.7.38

$$\frac{\partial S}{\partial \tau} = \frac{1}{2}\left(\frac{8Ste}{4+Ste}\right)^{1/2}\tau^{-1/2}$$ Eq. 2.7.39

When the solidification is final (S=1)

$$\tau_f = \frac{(4+Ste)}{8Ste}$$ Eq. 2.7.40

Rate of change of solidification thickness is given by the following equation

$$S = \frac{y}{h}$$

$$\tau = \frac{\alpha_s t}{h^2}$$

$$t = \tau\frac{h^2}{\alpha_s}$$

$$\frac{\partial t}{\partial \tau} = \frac{h^2}{\alpha_s}$$

$$\frac{\partial s}{\partial \tau} = \frac{-}{h}\frac{\partial y}{\partial t}\frac{\partial t}{\partial \tau}$$

$$\frac{\partial s}{\partial \tau} = \frac{-}{h}\frac{\partial y}{\partial t}\frac{h^2}{\alpha_s}$$

$$\frac{\partial y}{\partial t} = \frac{\alpha_s}{h}\frac{\partial S}{\partial \tau}$$ Eq. 2.7.41

$$\frac{\partial y}{\partial t} = \frac{\sqrt{2 \text{ Ste } \alpha_s}}{\sqrt{(4+\text{Ste})t}}$$ Eq. 2.7.42

$$S = \left(\frac{\sqrt{8 \text{ Ste}}}{\sqrt{(4+\text{Ste})}}\right)\sqrt{\tau}$$ Eq. 2.7.43

$$y = \frac{\sqrt{8 \text{ Ste } \alpha_s \ t}}{\sqrt{4+\text{Ste}}}$$ Eq. 2.7.44

$$\Theta s = \frac{\text{Ts}-\text{Tb}}{\text{Tm}-\text{Tb}}$$ Eq. 2.7.45

$$\theta_s = \frac{(4+3Ste)}{2(2+Ste)}\left(\frac{X}{S}\right) - \frac{Ste}{2(2+Ste)}\left(\frac{X}{S}\right)^3$$ Eq. 2.7.34

$$\text{Ts} = (\Theta s)\ (\text{Tm} - \text{Tb}) + \text{Tb}$$ Eq. 2.7.46

2.8 Prospects of utilization of cool thermal storage for buildings in Sri Lanka

In Sri Lanka at present the base electricity demand is met with coal thermal Power plants and hydro power plants while the peak electricity demand is met by gas turbines & diesel power plants due to non-availability of sufficient capacity of hydro power reservoirs.

Any industry falling under the category of Industrial Time of Day Tariff (Two Part) and (Three Part) eligible to be benefitted if cold thermal storage is operated to shift the cooling loads from peak time to off peak period. The magnitude of the cost savings depends on the mode of operation. If the entire cooling load during the peak time is shifted to the off peak period, maximum possible operational cost benefits could be obtained but type of the capital cost will be a decisive factor when the return on investment is estimated. Depending upon the period during which the estimated cooling load is stored during the off peak period will further help the industries to reduce the maximum demand charge for electrical energy cost. Further shifting of partial cooling load from peak period to off peak period will also help to reduce the maximum demand charge of electrical energy bill.

The difference between tariff between peak and off peak period is moderate and not very high. If Sri Lanka build its next coal power plant at "Sampur", there is a great possibility that the off peak tariff could further be reduced because two coal plants will be catering the base load of national grid. Under such situation Ceylon Electricity Board has to look for all possible avenue to increase the demand for its base loads during the non-peak period with the aim of replacing peak time generation from coal power plants. Ceylon Electrical Board need to take more active steps to popularize the cold thermal storage in Sri Lanka to tactically cater the growing demand for electrical energy for Air Conditioning systems.

Apart from the direct operational cost there are numerous benefits could be expected from capital cost point of view if the cold thermal storage system is properly designed, built and commissioned. Those additional benefits are as follows.

(1) Shifting of electrical load generated from cooling loads from peak time to off peak period.
(2) Downsizing of chiller plants and other equipment in HVAC system. Downsizing of cables, Transformer, switch gears of the associated electrical system of the HVAC system.
(3) Downsizing of Stand by Generator for the building.
(4) Reduction of chilled water pipe sizing.
(5) Due to the lower chilled water temperature, it is possible to down size the AHUs, Fan coil units of HVAC system.
(6) Ability to provide Air Conditioning for a building during Grid power failure with a help of low capacity generator to run the set of pipes to maintain the chilled water flow.

(7) Reduction of maintenance cost of electrical systems due to downsizing of the equipment.

However cool thermal storage for building MVAC systems is having following drawbacks.

- The initial cost of cold thermal storage based MVAC systems is comparatively high.
- When ice is built by the chiller, its rated capacity is degraded. Simply the required chiller capacity has to be increased and this also contributes to increase the capital cost.

The government of Sri Lanka should make reasonable and fair attempt to introduce and promote cold thermal storage systems in Sri Lanka. Due to the promotion of solar energy, electricity companies may further losing the day time electricity demand and the gap between day time and peak demand time will be becoming wider. If the proper measures are not taken on time electricity companies may have to heavily depend of expensive solutions like diesel and gas turbine power plants to meet the peak demand.

When considering taking decisions on cold thermal storage systems, careful estimation of the cooling load is very prime important. The cold thermal storage is proven and promising technology and but need to be designed, built & commissioned properly with the help of experts.

Due to the massive popularity of cold thermal storage technologies, more research has been conducted and published with the focus on large scale systems. But this technology could still be used for air conditioning systems of small scale. There is a huge demand for split type air conditioning systems due to easiness of implementation. These systems constitute to a fairly large cooling load and ultimately contribute to the major portion of electrical loads. This these is focused to assess engineering economics of developing ice storage based air conditioning for individual cottage type house units for resort hotels of Sri Lanka. This would be a major mile stone in development of cold thermal storage technologies.

Even though the required “Time of Day Tariff” was introduced in 2008, still the use of cold thermal storage is so far has not yet been successfully implemented for building air conditioning systems in Sri Lanka. The gap between peak and off peak is considerably high but due to the lack of real initiatives taken to promote the cold thermal storage facilities, the demand for cold thermal storage systems very low in Sri Lanka.

The Government Gazette notification No 1572/25 dated 24th October 2008 has introduced new Tariff Charges which includes the “Time of Day Tariff” that offers certain benefits for implementation of cool thermal storage especially for Industry and Hotels sectors with the aim of shifting cooling loads from peak time to off peak period.

3. RESEARCH APPROACH OF DESIGNING A SMALL SCALE ICE STORAGE BASED AIR CONDITIONING UNIT

3.1 Introduction

No attention is yet being paid to the use of cold thermal technology for small scale domestic air conditioning systems. Due to the uplifting of life standards & urbanization in Sri Lanka more demand was created for domestic air conditioning units. This chapter deals with the research approach of the designing a small scale ice storage based air conditioning system. The rate of ice forming and built ice thickness is calculated based on the mathematical modeling of the water freezing inside a closed enclosure with constant temperature glycol circulation system at the top & bottom walls. Three designs are reviewed based on 3 different glycol temperatures at the walls of the ice forming chambers. The capital cost and the operational cost are calculated for 3 designs.

3.2 Design parameters

The design will focus to build a cold thermal storage based air conditioner unit with capacity of 32 Ton hours which is equal to 384,000 BTU hours.

a.) Capacity: 32 Ton hours (384,000 BTU hours)

b.) Ice Making Process uses secondary heat transfer fluid, Propylene Glycol

c.) Room space cooling is implemented through standard fan coil unit

d.) Refrigeration is done by glycol chiller unit, capacity of which has to be decided based on the analysis

e.) Ice storage: 1000 liters

f.) Glycol pump will circulate propylene Glycol inside a closed loop

3.3 Design configuration & operational modes

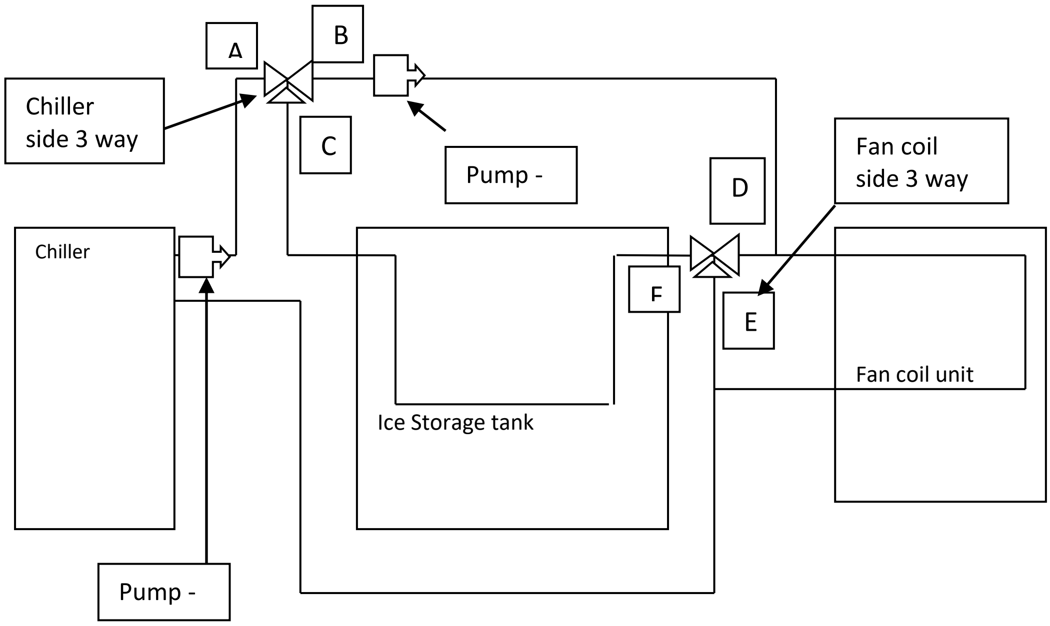

Fig 3.3.1: Schematic diagram of the proposed cold thermal storage based air conditioning unit

Operational Modes

1. Ice storage charging (Making the ice storage)
2. Discharge ice storage and provide cooling effect to the room space
3. Direct cooling of room space by the chiller

The above indicated operational modes are carried out with using the 2 units of three way valves and two pumps installed in the system.

Operational Mode			**Chiller side Three way valve**			**Fan coil side Three way valve**		
	Pump-1	**Pump-2**	**A**	**B**	**C**	**D**	**E**	**F**
Ice Storage charging	ON	OFF	ON	OFF	ON	OFF	ON	ON
Discharging of ice storage	OFF	ON	OFF	ON	ON	OFF	ON	ON
Direct cooling by the chiller	ON	OFF	ON	ON	OFF	OFF	OFF	OFF

Table 3.3.1 : Operation of valves and fan coil units

3.4 Research methodology

1. Build the mathematical model for ice forming inside a horizontal rectangle chamber while maintaining a constant temperature at one wall.
2. Develop 3 ice storage systems based on 3 different glycol temperatures ranges (inlet/outlet) to charge within 6.5 hours. The target storage capacity is 32 Ton hours.
3. Investigate the manufacturing cost of all three designs.
4. Operation will be investigated for hotel rooms & movie theaters under relevant electrical tariffs.
5. Recovery period of the investment will be calculated comparing the conventional approach and the savings due to the installation of proposed small scale ice storage based air conditioning system.

3.4.2 Develop 3 ice storage systems based on 3 different glycol temperatures ranges (inlet/outlet) to charge ice within 6.5 hours. The target storage capacity is 32 Ton hours.

Case 1

Set propylene glycol temperature: -12 ^{0}C (261 K)

The ice build thickness during the 20 minute time interval calculated with **Eq 2.7.44** and tabulated in table 3.4.2.1

Time (hours)	0:20	0:40	1:00	1:20	1:40	2.00	2:20	2:40	3.00	3:20	3:40
Time (minutes)	20	40	60	80	100	120	140	160	180	200	220
Ice build thickness (cm)	1.47	2.08	2.54	2.94	3.28	3.60	3.89	4.16	4.41	4.65	4.87
Time (hours)	4.00	4:20	4:40	5.00	5:20	5:40	6.00	6:20	6:40		
Time (minutes)	240	260	280	300	320	340	360	380	390		
Ice build thickness (cm)	5.09	5.30	5.50	5.69	5.88	6.06	6.23	6.40	6.56		

Table 3.4.2.1 : Ice build thickness during 6.5 hours period

$$y = \frac{\sqrt{8 \text{ Ste } \alpha_s \ t}}{\sqrt{4+\text{Ste}}} \qquad \boxed{\text{Eq. 2.7.44}}$$

$$\theta_s = \frac{(4+3Ste)}{2(2+Ste)}\left(\frac{X}{S}\right) - \frac{Ste}{2(2+Ste)}\left(\frac{X}{S}\right)^3 \qquad \boxed{\text{Eq. 2.7.34}}$$

$$Ts = (\Theta s)\ (Tm - Tb) + Tb \qquad \boxed{\text{Eq. 2.7.46}}$$

Based on the mathematical calculations, ice build thickness is calculated for three different wall temperatures. The main purpose of this approach is to find the best possible and economically viable dimensions for ice making chambers. It is a usual practice to maintain a very low below zero temperatures to build ice. The lower the set temperature results in lowering the COP of chillers. Usually chillers operated at 5^0C or 6 ^{0}C will degrade its capacity of operation when its set temperature is -15^0C or -20 ^{0}C. The ability to built ice at relatively higher temperatures (below zero ^{0}C) will ensure more economical benefits. The ice build thickness during 6.5 hours period is calculated with Eq.2.7.44 for three different wall temperatures with the aim of building three cases for analysis.

Case 1 : Glycol temperature 261 K (-12 ^{0}C)
Case 2 : Glycol temperature 267 K (-6 ^{0}C)
Case 3 : Glycol temperature 261 K (-3 ^{0}C)

The achieved ice build thickness is calculated for every 20 minutes during the 6.5 hours. Same is repeated for all three cases and results are tabulated. According to the results of ice making during 6.5 hours period, three different achievable ice build thicknesses are defined as follows.

Case 1: Maximum ice build thickness - 6.5 cm
Case 2: Maximum ice build thickness - 4.6 cm
Case 3: Maximum ice build thickness - 3.25 cm

Thereafter total energy removal during the 6.5 hours period is calculated for which primary ice making chamber unit dimensions are defined below. Each ice making chamber unit will have two walls on top & bottom through which the energy is removed from water by the cold glycol circulation system.

Case 1:
Primary ice build chamber dimensions
Height: 13 cm, Width : 10 cm, Length : 110 cm
Case 2:
Primary ice build chamber dimensions
Height: 9.2 cm, Width : 10 cm, Length : 110 cm
Case 3:
Primary ice build chamber dimensions
Height: 6.5 cm, Width : 10 cm, Length : 110 cm

In order to calculate the total energy removal during the freezing, the temperature profiles along the built ice thickness is calculated with using the **Eq.2.7.34** & **Eq.2.7.46**. The **Eq.2.7.34** defines the dimensionless temperature ratio quantity **Θs** and **Eq.2.7.46** results the dimensional temperature **Ts** along the build ice thickness.

In all 3 cases, top and bottom walls are maintained at constant glycol temperature **Ts**. For each case maximum ice build thickness that can be built during 6.5 hours period is calculated and results are obtained. For all three cases built ice thickness during every hour is calculated and tabulated. In the same time the temperature profile is calculated along the built ice thickness for every 2.5 mm distance during each 1 hour period. Based on the temperature profile the energy removed during each hour is calculated for every 2.5 mm distance and totaled.

The calculation of energy removal within every 2.5 mm is calculated based on following scenarios;

- Initial temperature of water is assumed to be 300 K or 27 ^{0}C. The total energy removed to reduce water temperature to 273 K from 300 K is calculated which is sensible energy.
- Energy removed during freezing is calculated which is latent energy.
- If the ice temperature is below 273 K, then the energy removed due to sub cooling is calculated which is sensible energy.

Based on the above criteria, total energy removed from each 2.5 mm ice thickness is calculated for the half of the primary ice making chambers of each case and tabulated. (Each primary ice making chamber unit has got two glycol circuits, one at bottom and other on top) This is repeated for 6.5 hours period for all 3 cases. The total energy removed from each chamber unit is double the results due to the fact that each ice making chamber is two time the size of maximum achievable ice thickness due to both top & bottom glycol circulations.

These 3 cases provides an insight of ice building process during 6.5 hours period and the dimensions of the each primary ice making chamber unit & number of such units will decides the cost of cold thermal energy storage for each case. Similarly glycol temperature will also influence the capital cost of the chillers as well as the operational cost.

Case 1:
Primary ice making chamber dimensions
Height: 13 cm, Width: 10 cm, Length: 110 cm
Glycol temperature: 261 K (-12 ^{0}C)

The maximum achievable ice build thickness
Time : 23,400 seconds (6.5 hours or 390 minutes)

Time (hours)	0:20	0:40	1:00	1:20	1:40	2:.00	2:20	2:40	3:.00
Time (minutes)	20	40	60	80	100	120	14	160	180
Ice Thickness (cm)	1.47	2.08	2.54	2.94	3.28	3.60	3.89	4.16	4.41

Time (hours)	3:20	3:40	4:.00	4:20	4:40	5:.00	5:20	5:40	6:.00	6:20	6:30
Time (minutes)	200	220	240	260	280	300	320	340	360	380	390
Ice thickness (cm)	4.65	4.87	5.09	5.30	5.50	5.69	5.88	6.06	6.23	6.40	6.56

Table 3.4.2.2 :Ice build thickness during 6.5 hours period

The dimensionless temperature ratio quantity Θs and temperature of ice Ts along the ice build thickness in every 2.5 mm is tabulated in Appendix A**.** The details of heat removed during each 60 minute period during 6.5 hours period is also tabulated in **Appendix A.**

Temperature profile & energy removed from ice during the freezing process
(1^{st} , 2^{nd} 3^{rd} Hours)

T : Temperature K
E1: Energy removed from water to reach 273 K (kJ)
E2: Energy removed when freezing (kJ)
E3: Energy removed due to sub cooling of ice (kJ)

Ice Thick -ness (mm)	**1^{st} hour** **Maximum ice thickness : 25.4 mm** **Total E1 = 316 kJ** **Total E2 = 933.20 kJ** **Total E3 = 31.53 kJ** **Total energy removed = 1280.81 kJ**				**2nd hour** **Maximum ice thickness : 35.9 mm** **Total E1 = 130.67 kJ** **Total E2 = 385.77 kJ** **Total E3 = 14.6 kJ** **Total energy removed = 531.04 kJ**				**3^{rd} hour** **Maximum ice thickness : 44.07 mm** **Total E1 = 101.67 kJ** **Total E2 = 300.17 kJ** **Total E3 = 11.28 kJ** **Total energy removed = 1280.81 kJ**			
	T	E1	E2	E3	T	E1	E2	E3	T	E1	E2	E3
2.5	262.2	31.11	91.85	6.24	261.85			0.2	261.69			0.09
5	263.4	31.11	91.85	5.54	262.7			0.41	262.39			0.18
7.5	264.6	31.11	91.85	4.85	263.54			0.61	263.08			0.27
10	265.79	31.11	91.85	4.16	264.39			0.81	263.77			0.36
12.5	266.98	31.11	91.85	3.48	265.24			1.01	264.46			0.45
15	268.16	31.11	91.85	2.80	266.08			1.2	265.15			0.54
17.5	269.33	31.11	91.85	2.12	266.92			1.39	265.84			0.62
20	270.5	31.11	91.85	1.45	267.75			1.58	266.52			0.71
22.5	271.65	31.11	91.85	0.788	268.59			1.77	267.21			0.8
25	272.8	31.11	91.85	0.12	269.42			2.07	267.89			0.88
25.4	272.98	4.98	14.696	0.00	270.24	26.13	77.15	1.59	268.57			0.96
27.5												
30					271.06	31.11	91.85	1.12	269.25			1.05
32.5					271.87	31.11	91.85	0.65	269.92			1.13
35					272.68	31.11	91.85	0.18	270.59			1.39
35.9					272.97	11.20	33.06	0.01	271.26	19.91	58.78	1
37.5												
40									271.93	31.11	91.85	0.62
42.5									272.59	31.11	91.85	0.24
44.07									273	19.54	57.68	0

Table 3.4.2.3

Temperature profile & energy removed from ice during the freezing process (4^{th}, 5^{th}, 6^{th} Hours)

T : Temperature K
E1: Energy removed from water to reach 273 K (kJ)
E2: Energy removed when freezing (kJ)
E3: Energy removed due to sub cooling of ice (kJ)

Ice Thickness (mm)	**4^{th} hour Maximum ice thickness : 50.8 mm Total E1 =81.86 kJ Total E2 = 247.26 kJ Total E3 = 9.6 kJ Total energy removed = 338.72 kJ**				**5th hour Maximum ice thickness : 56.8 mm Total E1 = 74.67 kJ Total E2 = 220.44 kJ Total E3 = 10.70 kJ Total energy removed = 305.8 kJ**				**6^{th} hour Maximum ice thickness : 62.3 mm Total E1 =68.44 kJ Total E2 = 202.07 kJ Total E3 = 7.43 kJ Total energy removed = 277.94 kJ**			
	T	E1	E2	E3	T	E1	E2	E3	T	E1	E2	E3
2.5	261.6			0.05	261.54			0.04	261.49			0.03
5	262.2			0.11	262.07			0.07	261.98			0.05
7.5	262.8			0.16	262.61			0.11	262.47			0.08
10	263.4			0.21	263.15			0.15	262.96			0.11
12.5	264			0.27	263.68			0.18	263.45			0.13
15	264.6			0.32	264.22			0.22	263.94			0.16
17.5	265.19			0.37	264.75			0.25	264.43			0.19
20	265.79			0.42	265.29			0.29	264.91			0.21
22.5	266.38			0.48	265.82			0.33	265.4			0.24
25.0	266.98			0.53	266.35			0.36	265.89			0.27
27.5	267.57			0.58	266.88			0.4	266.37			0.29
30.0	268.16			0.63	267.41			0.43	266.86			0.32
32.5	268.75			0.68	267.94			0.47	267.34			0.35
35.0	269.33			0.73	268.47			0.5	267.82			0.37
37.5	269.92			0.78	268.99			0.53	268.3			0.4
40.0	270.5			0.82	269.51			0.57	268.78			0.42
42.5	271.08			1.11	270.04			1.71	269.26			0.45
45.0	271.65	11.57	34.17	0.78	270.56			1.41	269.74			0.47
47.5	272.23	31.11	91.85	0.45	271.07			1.11	270.22			0.5
50.0	272.8	29.96	91.85	0.12	271.59			0.81	270.69			0.52
50.8	272.98	9.22	29.39	0								
52.5					272.1	21.16	62.46	0.52	271.16			0.54
55.0					272.61	31.11	91.85	0.22	271.63			0.57
56.8					272.98	22.4	66.13	0.01				
57.5									272.1	8.71	25.72	0.51
60.0									272.57	31.11	91.85	0.25
62.3									272.99	28.62	84.5	0

Table 3.4.2.4

Temperature profile & energy removed from ice during the freezing process (6.5th hour)

T : Temperature K E1: Energy removed from water to reach 273 K (kJ) E2: Energy removed when freezing (kJ) E3: Energy removed due to sub cooling of ice (kJ)				
Ice Thickness (mm)	6.5th hour Maximum ice thickness : 65.6 mm Total E1 = 41.07 kJ Total E2 = 121.24 kJ Total E3 = 5.75 kJ Total energy removed =168.06 kJ			
	T	E1	E2	E3
2.5	261.46			0.01
5	261.93			0.03
7.5	262.39			0.04
10	262.86			0.06
12.5	263.32			0.07
15	263.79			0.09
17.5	264.25			0.1
20	264.71			0.12
22.5	265.18			0.13
25.0	265.64			0.14
27.5	266.1			0.16
30.0	266.56			0.17
32.5	267.02			0.19
35.0	267.48			0.2
37.5	267.93			0.21
40.0	268.39			0.23
42.5	268.84			0.24
45.0	269.3			0.25
47.5	269.75			0.27
50.0	270.2			0.28
52.5	270.65			0.29
55.0	271.1			0.31
57.5	271.55			0.32
60.0	271.99			0.58
62.5	272.44	2.49	7.35	0.33
65.0	272.88	31.11	91.85	0.65
65.6	272.98	7.47	22.04	0.28

Table 3.4.2.5

Period hour	Water cooling (kJ)	Sub cooling (kJ)	latent (kJ)	Total energy removed (kJ)
1st hour	316.09	31.53	933.20	1280.81
2nd hour	130.67	14.60	385.77	531.04
3rd hour	101.67	11.28	300.17	413.11
4th hour	81.86	9.60	247.26	338.72
5th hour	74.67	10.70	220.44	305.80
6th hour	68.44	7.43	202.07	277.94
6.5 hour	41.07	5.75	121.24	168.06
Total energy removed				**3315.50**

Table 3.4.2.6 :Total energy removed during 6.5hrs period

- The total energy removed from each ice making chamber is two times the above value which is 6631 kJ.
- Total energy to be removed from 6 units of ice making chambers : 6 x 6631 kJ
- Total energy to be removed from total stack of 12 such units : 6x 6631 x 12 kJ

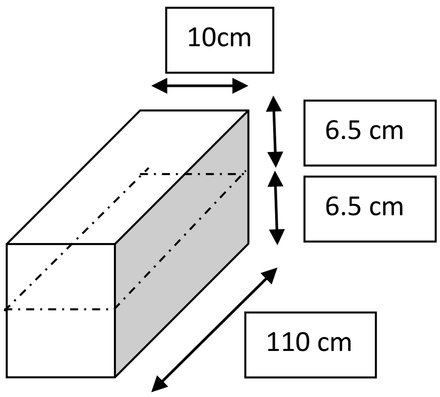

Fig 3.4.2.1 : Primary ice making chamber unit

6 ice build chambers are in series to build each composite unit as illustrated in **Fig. 3.4.2.2**

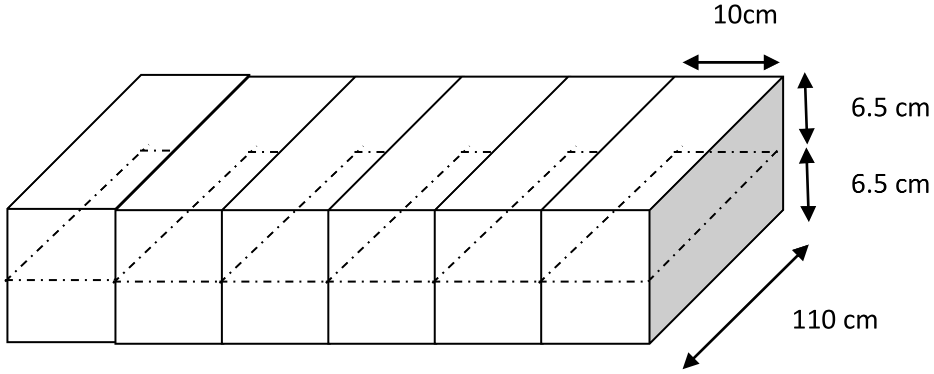

Fig4.2.2 Composite unit of Ice making chamber

Total of 12 composite ice build chambers are stacked by keeping glycol circuit in between as illustrated in **Fig 3.4.2.3**

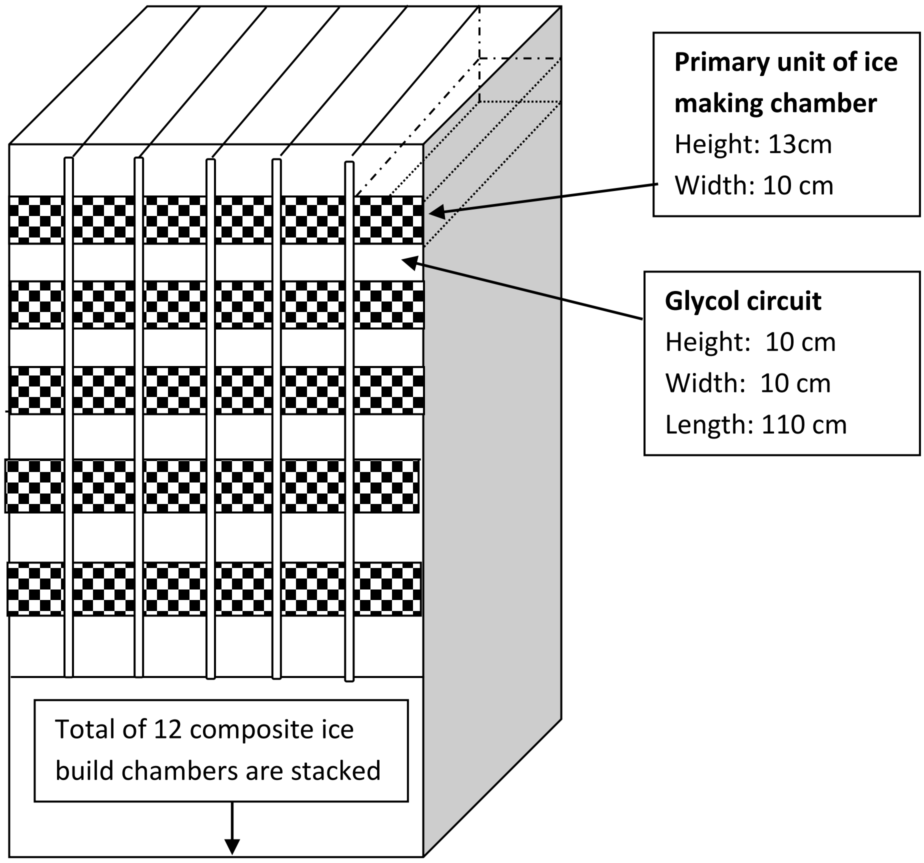

Fig 3.4.2.3 : Total of 12 composite ice making chambers are stacked together

Total capacity of ice thermal storage

Capacity of each primary ice making chamber : 6631 kJ

Capacity of composite ice making chamber : 6x6631 = 39,786 kJ

Total capacity of 12 composite ice making chambers : 12 x 39,786 = 477,432 kJ

Total capacity in BTU (1 BTU = 1.055 kJ) = 452,518.22 BTU

Total capacity in Ton Hours 1 (Ton Hours = 12000 BTU): 37.7 Ton Hours

Design of glycol circulation system for the above case

Consider primary ice making chamber unit which has dimensions of 0.1 x 0.13 x1.10 in meters and capacity of 0.0143 m^3. This unit has two walls to remove heat energy from the water during the ice forming process. In the same time each glycol circuit is sandwiched between two ice making chamber units as illustrated in **Fig. 3.4.2.3**. Apart from the top and bottom glycol circuits, all others need to remove heat energy from the top and bottom walls. Total energy to be removed from each glycol circuit is 50%

from bottom wall and the balance 50% from top wall of ice making chamber unit. Energy to be removed by each glycol circuit equals the total energy removed from each ice making chamber unit.

Period - hours	Total energy removed (kJ)
1st hour	2561.62
2nd hour	1062.08
3rd hour	826.22
4th hour	677.44
5th hour	611.6
6th hour	555.88
6.5 hour	336.12
Total energy removed	**6631.00**

Table 3.4.2.7
Energy to be removed by the glycol circuit during 6.5 hours period

For the purpose of mathematical analysis, the wall temperature is assumed to be constant but in real scenario there is a drop in the temperature along the length of primary ice making chamber unit. The design of glycol circulation system propose 5^0C reduction of wall temperature between the input & output of primary glycol circuit. The glycol inlet will maintain 261 K (-12^0C) while outlet side maintain 266 K ($-7\ ^0$C) wall temperature. This will result two different ice thicknesses at glycol inlet & outlet ends of ice build chamber unit as illustrate in **Fig. 3.4.2.5**

Glycol inlet end : 6.5 cm
Glycol outlet end : 5.0 cm

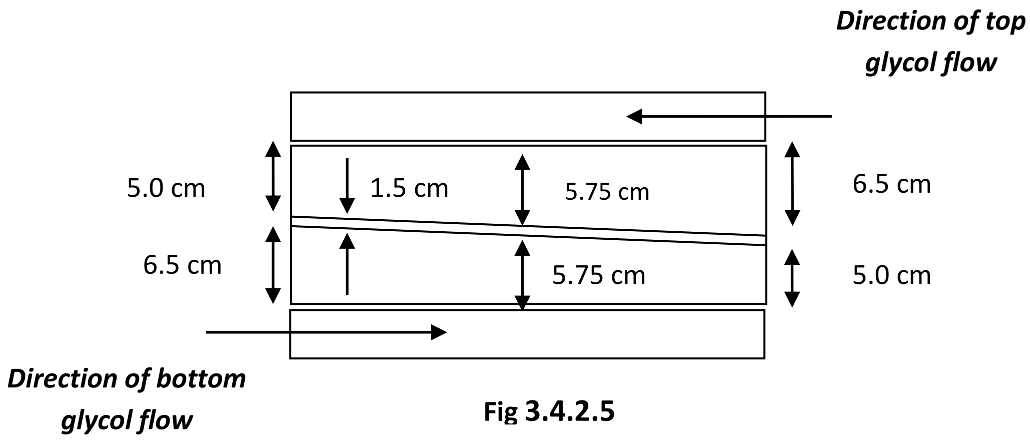

Fig 3.4.2.5

Dimensions of primary ice making chamber unit

As of the above case it could be assumed that mid of the length PIMC will maintain 263.5 K (-9.5 ^{0}C) and its final ice build thickness is 5.75 cm. These sounds that there will be a closely linear variation of ice build thickness along the length of the ice build chamber. In order to overcome effect of this, top and bottom wall glycol will flow in counter directions as illustrated in **Fig. 3.4.2.5**

Primary Ice making chamber units with top & bottom glycol circulations in counter flow direction.

The net effect of the counter flow will create a 1.5 cm gap between top & bottom ice layers inside the ice making chamber unit. This will change the total ice thermal storage capacity as follows.

Volume of Primary ice making chamber: 0.13 x 0.1x1.1 = 0.0143 m^3 = 14.3 liters
Volume lost due to variation of wall temperature: 0.015x0.1x1.1 m^3 = 1.65 liters
The total reduction of ice volume is 11.5 %

The actual capacity of the ice storage unit = 37.7 x (100-11.5)/100
= 33.36 Ton hours

Utilization of circular pipes for glycol circuit for actual product development

For the purpose of calculations and analysis, square type tubes were considered. During the actual product development, building a complete glycol circuit with square tube is more costly and difficult task compare to circular tubes. Circular tubes are easy to connect and more accessories are available from the open market with correct fittings. Due to those reasons primary glycol circuit will be developed as explained below.

Each composite ice build chamber unit consist of 6 primary ice making Chamber units as shown in **Fig 3.4.2.5**

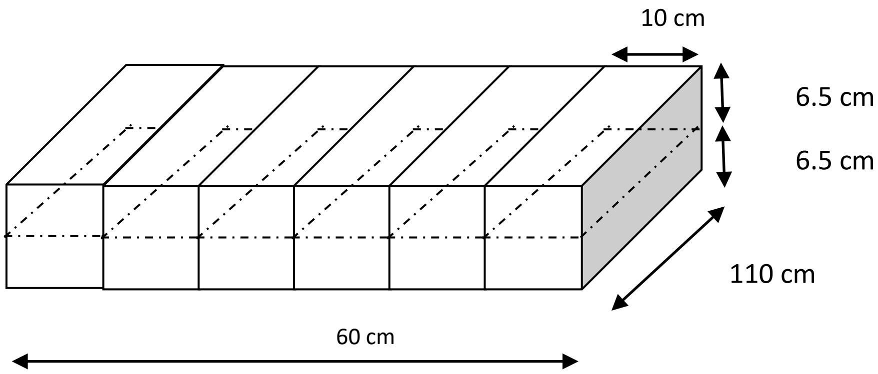

Fig 3.4.2.5; **Composite ice build chamber unit**

Consider 1st Hour

The counter flow arrangement compensate the drop in ice thickness of each other. In order to satisfy operational requirements, the maximum energy removed during an hour is considered when designing the glycol flow rate.

Maximum energy removed in a hour =2561.62 kJ

Average energy removed in a second = 2561.62 /3600 = 0.7116 kJ per second

Due to the change in thickness of ice build inside the chamber, the actual energy removed within the first hour is less than the above amount but for the purpose of maintaining safety factors the above rate of energy removal is considered for designing glycol flow rate.

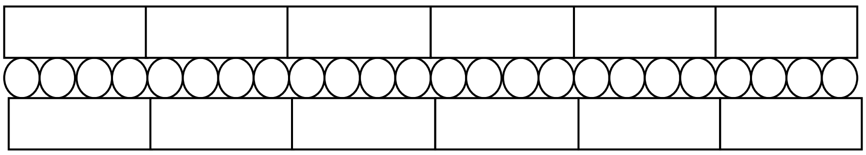

Fig 3.4.2.6 : Circular glycol circuit sandwiched between composite Ice making chamber units

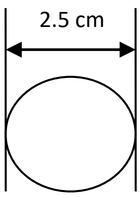

Fig 3.4.2.7 : Diameter of glycol circulation pipe

It is assumed that the top and bottom surfaces of 24 circular pipes will form equivalent flat surfaces on both sides.

Outer diameter of circular pipes : 2.5 cm

Thickness of pipe : 1 mm

Total number of pipes per composite glycol circuit : 24

Length of each pipe : 1.1 m

Energy balance equation

(Flow rate) x (Density) x Cp (Tin –Tout)= Energy removed in second

Energy removed per second from PIMC = 0.7116 kJ per second

Energy removed per second from composite Ice making chamber =0.7116 x6 =4.27 kJ

Quantity	
A - Cross sectional area	0.1x 0.1 =.01 m^2
Cp - Specific heat of glycol	3.8 kJ/K kg
Density of glycol	1030 kg/ m^3
Tin – Entrance Temperature	261 K
Tout – Exit Temperature	266 K

Table 3.4.2.8 : Parameters of glycol circulation system

(Flow rate in 24 pipes) x Density x Cp x (Tout – Tin)=Rate of Energy removal from composite ice making chamber.

(Flow rate in 24 pipes) x Density x Cp x (Tout – Tin)=4.27 kJ per second

Flow rate = 4.27/ (Density x Cp x (Tout – Tin))

Flow rate in 24 pipes = 0.00022 m^3/Sec.

Flow rate in 24 pipes =0.22 liters per second

Flow rate in each pipe = 0.0091 Liters per second

As there are 12 composite glycol circuit stacks

The flow rate of the pump = 12 x 0.22 liters per second

= 2.6 liters per second

The total energy to be removed from water is reduced in every hour, so that flow rate also to be reduced. Following table provide the required pump flow rate in each hour during 6.5 hours period

Period - hours	Total energy removed primary ICB (kJ)	Energy removal rate in composite unit (kJ/s)	Flow rate in each pipe(24 pipes) (liters/s)	Flow rate in each stack (24 pipes per stack) (liters/s)	Pump flow rate (12 stacks) (liters/s)
1st hour	2561.62	4.27	0.0091	0.22	2.62
2nd hour	1062.08	1.77	0.0038	0.09	1.09
3rd hour	826.22	1.38	0.0029	0.07	0.84
4th hour	677.44	1.13	0.0024	0.06	0.69
5th hour	611.6	1.02	0.0022	0.05	0.63
6th hour	555.88	0.93	0.0020	0.05	0.57
6.5 hour	336.12	0.56	0.0012	0.03	0.34

Table 3.4.2.9 : Pump flow rate on hourly basis

Final design dimensions of the primary ice making chamber

Due to the variation of glycol temperature along the length of the ice making chamber, there will be a 1.5 cm gap between upper and bottom ice layers if the primary ice making chamber is built with 13 cm height. In order to minimize the waste of resources, the final height of the primary ice making chamber is 1.5 cm less than the initial proposed height. The new dimensions are illustrated in **Fig 3.4.2.7**

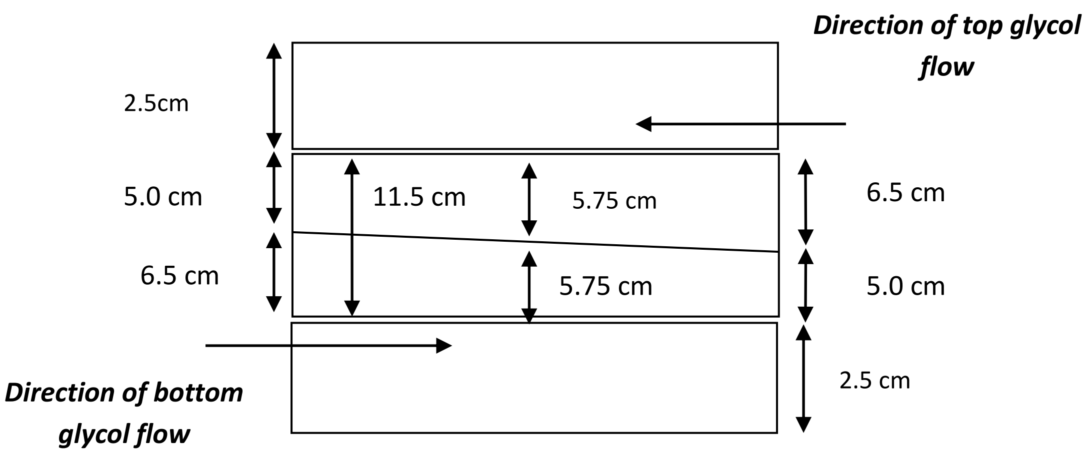

Fig 3.4.2.8 : Final design dimensions of primary ice making chamber

3.4.3 Case 2 & Case 3

The designs of Case 2 & Case 3 are presented in Appendix B & C respectively.

3.4.4 Chiller Operational Strategy

According to the above analysis and findings, there is a significant percentage of ice is built during the first hour. This is due to the fact when ice is built the thermal conductivity is lower in ice and rate of ice forming is reduced. According to the above, required chiller capacities are very high and not economical and will lead to a huge capital cost, so following chiller operational strategy is employed to cater the cooling requirement at the cost of reducing final ice storage capacity. This will enable to employ chillers with lower capacities as follows.

Case	**Without operational strategy**	**With operational strategy**		
	Required minimum chiller capacity (TR)	new chiller capacity (TR)	% of reduction of chiller capacity	Energy removal capacity **Ton hours**
Case 1	15	7	53%	33.32
Case 2	13	6	46%	30.39
Case 3	14	7	46%	32.64

Table 3.4.4.1 :Chiller capacity with and without chiller operational strategy

The actual ice thermal storage capacity is very much closer to the actual energy removal capacity of the storage system under new operational strategy.

Case 1	**Chiller operational strategy**						
Total chiller capacity	TR	7					
	BTU	84000					
Energy removal on hourly basis							
Hours	Primary IMC unit (kJ)	6 Primary IMCs (kJ)					
1st	2561.62	15369.72					
2nd	1062.08	6372.48					
3rd	826.22	4957.32					
4th	677.44	4064.64					
5th	611.6	3669.6					
6th	555.88	3335.28					
6.5th	336.12	2016.72					
Total	**6630.96**	**39785.76**					
H.C. Unit - Horizontal composite unit							
Hours	1st	2nd	3rd	4th	5th	6th	6.5th
H.C. unit 1	15369.72	6372.48	4957.32	4064.64	3669.6	3335.28	1008.36
H.C. unit 2	15369.72		6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 3	15369.72		6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 4	15369.72		6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 5	15369.72		6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 6		15369.72	6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 7		15369.72	6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 8		15369.72	6372.48	4957.32	4064.64	3669.6	1667.64
H.C. unit 9		15369.72		6372.48	4957.32	4064.64	1834.8
H.C. unit 10		15369.72		6372.48	4957.32	4064.64	1834.8
H.C. unit 11			15369.72	6372.48	4957.32	4064.64	1834.8
H.C. unit 12			15369.72	6372.48	4957.32	4064.64	1834.8
Total kJ in hour	76848.6	83221.08	80304.12	64255.8	51951.36	45281.04	20021.04
Btu	399889.14						
Ton hours	33.32						

Table 3.4.4.2 : Chiller operational strategy : Case 1

Case 2	**Chiller operational strategy**						
Total chiller capacity	TR	6					
	BTU	72000					
Energy removal on hourly basis							
Hours	Primary IMC unit (kJ)	6 Primary IMCs(kJ)					
1st	1660.94	9965.65					
2nd	795.03	4770.16					
3rd	600.43	3602.56					
4th	487.71	2926.28					
5th	421.05	2526.33					
6th	390.40	2342.39					
6.5th	319.38	1916.25					
Total	**4674.94**	**28049.62**					
H.C. Unit (Horizontal composite unit)							
Hours	1st	2nd	3rd	4th	5th	6th	6.5th
H.C. unit 1	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 2	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 3	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 4	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 5	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 6	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 7	9965.65		4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 8		9965.65	4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 9		9965.65	4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 10		9965.65	4770.16	3602.56	2926.28	2526.33	1171.20
H.C. unit 11		9965.65		4770.16	3602.56	2926.28	1263.16
H.C. unit 12		9965.65		4770.16	3602.56	2926.28	1263.16
H.C. unit 13		9965.65		4770.16	3602.56	2926.28	1263.16
H.C. unit 14		9965.65		4770.16	3602.56	2926.28	1263.16
H.C. unit 15			9965.65	4770.16	3602.56	2926.28	1263.16
H.C. unit 16			9965.65	4770.16	3602.56	2926.28	1263.16
Total kJ in hour	69,759.57	69,759.57	67,632.88	64,646.51	50,878.16	42,820.97	19,290.94
Total kJ	384,788.61						
Btu	364,728.54						
Ton hours	30.39						

Table 3.4.4.3 : Chiller operational strategy : Case 2

Case 3	**Chiller operational strategy**						
Total chiller capacity : 7 TR							
	7 TR						
BTU	84,000						
kW	21.1						
Energy removal on hourly basis							
Hours	**Primary IMC unit (kJ)**	**6 Primary IMC units - Composite unit**					
1st	950.61	5703.66					
2nd	394.29	2365.74					
3rd	348.22	2089.32					
4th	255.29	1531.74					
5th	225.11	1350.66					
6th	203.63	1221.78					
6.5th	95.25	571.5					
	2472.4	**14834.4**					
H.C. Unit - Horizontal composite unit							
Chiller capacity : 6 ton							
Hours	**1st**	**2nd**	**3rd**	**4th**	**5th**	**6th**	**6.5th**
H.C. unit 1	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 2	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 3	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 4	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 5	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 6	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 7	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 8	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 9	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 10	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 11	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 12	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 13	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 14	5703.66	2365.74	2089.32	1531.74	1350.66	1221.78	285.75
H.C. unit 15		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 16		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 17		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 18		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 19		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 20		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 21		5703.66	2365.74	2089.32	1531.74	1350.66	610.89
H.C. unit 22		5703.66	2365.74	2089.32	1531.74	1350.66	610.89

H.C. unit 23			5703.66	2365.74	2089.32	1531.74	675.33
H.C. unit 24			5703.66	2365.74	2089.32	1531.74	675.33
H.C. unit 25			5703.66	5703.66	2365.74	2089.32	1531.74
H.C. unit 26			5703.66	5703.66	2365.74	2089.32	1531.74
H.C. unit 27			5703.66	5703.66	2365.74	2089.32	1531.74
H.C. unit 28				5703.66	2365.74	2089.32	1531.74
H.C. unit 29				5703.66	2365.74	2089.32	1531.74
Total kJ in hour	79,851.24	78,749.64	76,694.70	71,408.70	47,170.50	41,420.28	17,896.98
Total kJ	413,192.04						
Btu	391,651.22						
Ton hours	32.64						

Table 3.4.4.4 : Chiller operational strategy : Case 3

Description		Case 1	Case 2	Case 3
Cold thermal storage capacity (TR)		33.32	30.39	32.64
Ice build thickness in PIMC unit (cm)		11.5	8	5.0
Dimension of Primary Ice Making Chamber unit (cm)	H	13.00	9.2	5
	W	10	10	10
	l	110	110	110
No of Primary IBCs		72.00	96.00	174.00
No of Primary IMCs per horizontal composite IMC stack		6	6	6
No of horizontal composite IMC stacks per storage system		12	16	29
Complete volume of ice storage system *Cubic meters*		1.03	0.97	0.97
Dimensions of primary glycol circuit (cm)	Out diameter	2.5	2.5	2.5
	Inner diameter	2.3	2.3	2.3
	Length	110	110	110
No of 2.5 cm pipes per composite glycol circuit		24	24	24
No of composite glycol circuit stacks per thermal storage		13	17	30
Total length of glycol circuit pipes (2.5 cm diameter, Aluminium)		343.2 meters (24x13x1.1)	448.8 meters (24x17x1.1)	792 meters (24x30x1.1)
Hazen-Williams Coefficient of Aluminium		130-150	130-150	130-150
Energy extract from each Primary IBC unit in 6.5 hours (kJ)		6631	4675	3296.50
Design energy extract in 6.5 hours (kJ)		415,361.52	448,800	477,432
Actual capacity Ton Hours		33.36	30.7	30.27
energy removal during 1st hour (kJ)		184,464.00	159,456.00	165,387.00
Required chiller capacity under normal scenario (TR)		15	13	14
Required chiller capacity under new operational strategy (RT)		7	6	7
Maximum Flow rate inside primary glycol circuit (liters per second)		0.0090	.0098	0.0067
Maximum Pump flow rate (liters per second)		2.6	3.7	4.9

Table 3.4.4.5 : Summary of all 3 cases

Viscosities of Propylene Glycol Solutions

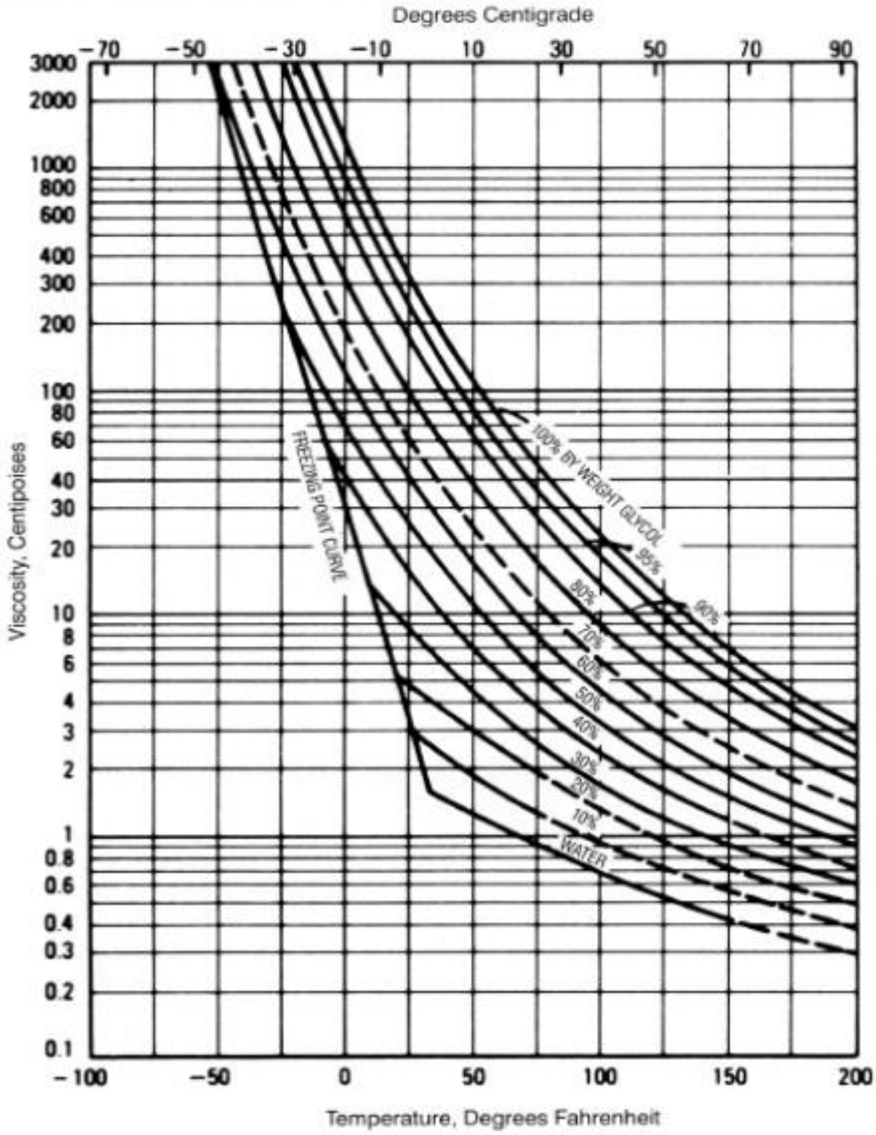

Fig 3.4.4.1

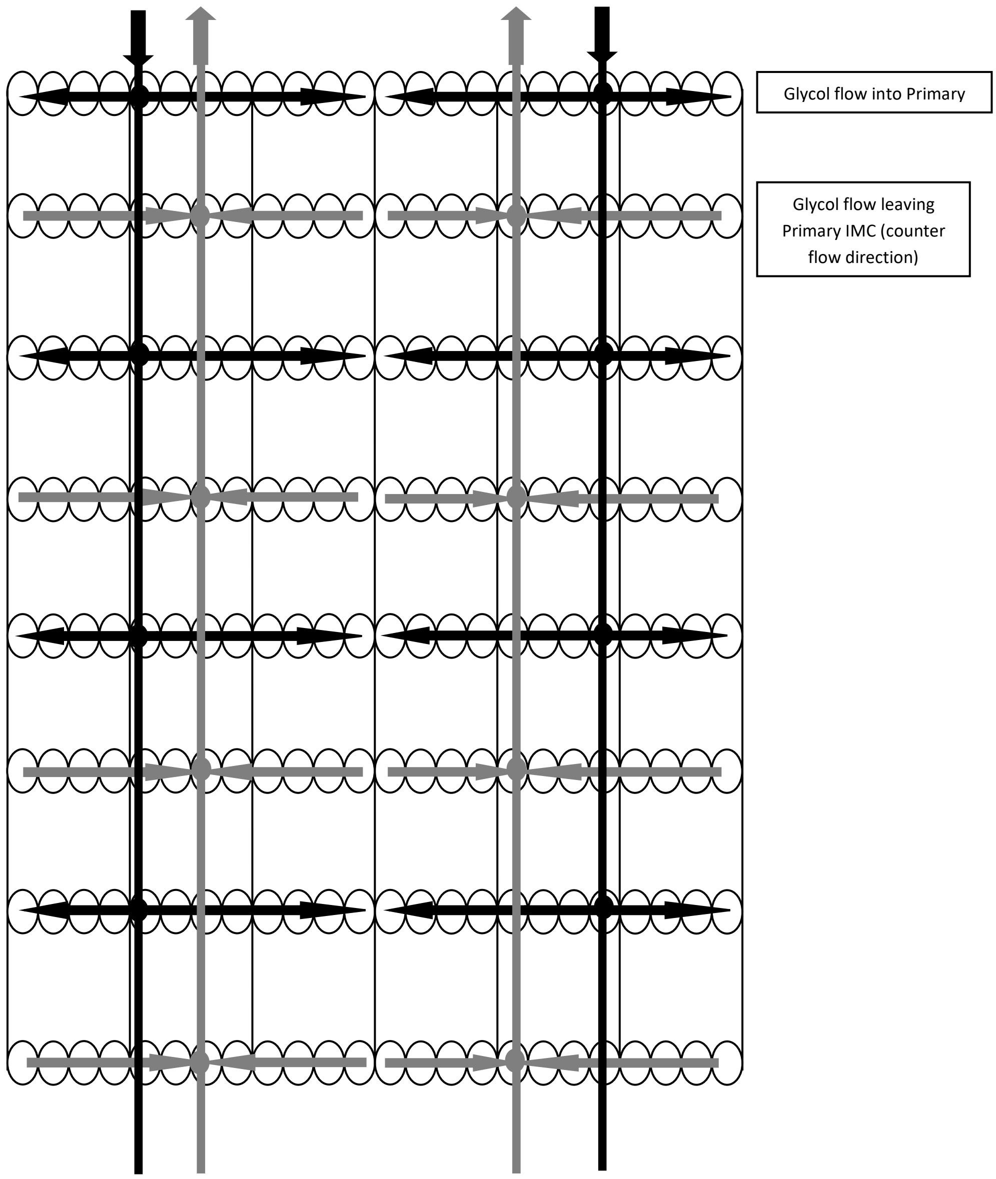

Fig 3.4.2.25: Typical glycol flow direction inside the circular pipe circuit

4 RESULTS & ANALYSIS

4.1 Estimating the power consumption of the pumps to circulate the propylene glycol

Description	Case 1	Case 2	Case 3
Maximum Flow rate: Q Inside 2.5cm diameter pipe circuit	0.009 Liters per second	0.0098 Liters per second.	0.0067 Liters per second
Pipe diameter: D	2.3 cm 0.023 m	2.3 cm 0.023 m	2.3 cm 0.023 m
Pipe cross sectional area	0.000416 m^2	0.000416 m^2	0.000416 m^2
Flow velocity: V	0.022 m/s	0.024 m/s	0.016 m/s
Dynamic viscosity: μ	0.033 Pa second (40% propylene glycol, -12 ^{0}C)	0.01. Pa second (30% propylene glycol, -6 ^{0}C)	0.0045 Pa second (20% propylene glycol, -3^{0}C)
Density of glycol : ρ	1065 kg/m^3 (40% propylene glycol, -12 C)	1050 kg/m^3 (30% propylene glycol, -6C)	1030 kg/m^3 (20% propylene glycol, -3c)
Kinetic viscosity: ν	0.000031 m^2/Sec.	0.0000095 m^2/Sec.	0.0000044 m^2/Sec.
Reynolds number Re = DV/ ν	(0.023x.022)/ 0.000031 Re 16.07 Laminar flow	(0.023x0.024)/ (0.0000095) Re : 56.94 Laminar flow	(0.023/0.0161)/ (0.0000044) Re : 84.86 Laminar flow
Reynolds number: $\mathrm{Re} = \frac{DV\rho}{\mu} = \frac{DV}{\nu}$	16.07 Laminar flow	56.94 Laminar flow	84.86 Laminar flow
Friction factor : F= 16/Re	1	0.28	0.19
Total length of glycol circuit pipes -m (Φ 2.3 cm)	343 m (24x13x1.1)	449 m (24x17x1.1)	792 m (24x30x1.1)
Head loss : h_f Due to pipe length $h_f = 2f \times (L/D) \times (V^2/g)$	1.42 m	0.62 m	0.34 m
Total power required per hour to overcome the friction inside along the pipe Power = $h_f \times \rho \times g \times Q$	0.133 kWh	0.063 kWh	0.023 kWh

Power required during 6.5 hours (Max)	0.87 kWh	0.41 kWh	0.15 kWh
Reynolds number: Re = $\frac{DV\rho}{\mu} = \frac{DV}{\nu}$	16 Laminar flow	0.28 Laminar flow	0.19 Laminar flow
No of 90^o Elbows & Tees	624 nos (24x2x13)	816 (24x2x17)	1440 (24x 2 x 30)
Friction coefficient of the fitting (Max) : K_L	2	2	2
Head loss per fitting $h_m = K_L(V^2/2g)$	0.004599529	0.005453565	0.002549048
Total head loss due to fittings	2.9 m	4.6 m	3.7 m
Power $h_m.\rho.g.Q$	0.27 kWh	0.45 kWh	0.25 kWh
Power for 6.5 hours to over come the friction loss in all fittings (Max)	1.75 kWh	2.92 kWh	1.62 kWh
Total power for pump during 6.5 hours	2.62 kWh	3.33 kWh	1.77 kWh
Rating of the pump	1.5 kW 5 meter head	1.5 kW 5 meter head	1.5 kW 5 meter head

Table 4.1.1

4.2 Investigate the manufacturing cost of all three designs.

Case 1				
Description	**Dimensions and other data**	**Material quantity**	**Cost per unit (LKR)**	**Cost (LKR)**
Glycol circuit				
2.5cm outer diameter Aluminium pipes (internal diameter 2.3 cm)	2.5cm diameter Aluminium pipes	343 m	75.00	25,725.00
Primary IMC separator Aluminium plate	110 x 10 cm x cm, 1 mm thickness Aluminium plates	84.00	50.00	4,200.00
Elbow connections	2.5 cm diameter PVC elbows	96	75.00	4,800.00
TEE connections	2.5 cm diameter PVC TEEs	96	75.00	7,200.00
Pumps	3KW pumps, 2.5 liters per second	2.00	15000.00	30,000.00
Solenoid single way valves	2.5 cm diameter	5.00	3500.00	17,500.00
Solenoid 3 way valves	2.5 cm diameter	2.00	7500.00	15,000.00
Glycol	40% propylene glycol solution	150 liters	75.00	11,250.00
Stainless steel enclosure				
size	0.025			
height - m - (0.025+0.115)x12+0.2	1.88			
Width - m - (0.1x6)	0.6			
Depth - m - 1.1	1.1			
H x W x D - 1.88 x 0.6 x 1.1 m3	1.88 x 0.6 x 1.1 $\mathbf{m^3}$	1.24	40000.00	40,000.00
Electronic controller unit				30,000.00
Cost of Glycol chiller (-12C, 7RT) Input power : 13.5 kw	(4500 US$ x 177)			796,500.00
4 units of fan coil units	12,000 Btu capacity	4.00	20000	80,000.00
Other accessories		25,000.00		25,000.00
Total				**1,087,175.00**

Table 4.2.1: Cost of production case 1

Case 2				
Description	**Dimensions and other data**	**Material quantity**	**Cost per unit (LKR)**	**Cost (LKR)**
Glycol circuit				
2.5cm outer diameter Aluminium pipes (internal diameter 2.3 cm)	2.5cm diameter Aluminium pipes	449 m	75.00	33,675.00
Primary IMC separator Aluminium plate	110 x 10 cm x cm, 1 mm thickness Aluminium plates	112.00	50.00	5,600.00
Elbow connections	2.5 cm diameter PVC elbows	128	75.00	6,400.00
TEE connections	2.5 cm diameter PVC TEEs	128	75.00	9,600.00
Pumps	3.5 KW pumps, 3.7 liters per second	2.00	17500.00	35,000.00
Solenoid single way valves	2.5 cm diameter	4.00	3500.00	14,000.00
Solenoid 3 way valves	2.5 cm diameter	2.00	7500.00	15,000.00
Glycol	40% propylene glycol solution	190 liters	75.00	12,750.00
Stainless steel enclosure				
size	0.025			
height - m - (0.025+0.115)x12+0.2	2.072			
Width - m - (0.1x6)	0.6			
Depth - m - 1.1	1.1			
H x W x D - 1.88 x 0.6 x 1.1 m3	2.072 x 0.6 x 1.1 m^3	1.37	50000.00	50,000.00
Electronic controller unit				30,000.00
Cost of Glycol chiller (-6 ^{0}C, 6 RT) Power input : 7.5 (kW)	(4000 US$ x 177)			708,000.00
4 units of fan coil units	12,000 BTU capacity	4.00	20000	80,000.00
Other accessories		25,000.00		25,000.00
Total				**1,025,025.00**

Table 4.2.2: Cost of production case 2

Case 3				
Description	**Dimensions and other data**	**Material quantity**	**Cost per unit (LKR)**	**Cost (LKR)**
Glycol circuit				
2.5cm outer diameter Aluminium pipes (internal diameter 2.3 cm)	2.5cm diameter Aluminium pipes	792 m	75.00	59,400.00
Primary IMC separator Aluminium plate	110 x 10 cmxcm, 1 mm thickness Aluminium plates	203.00	50.00	10,150.00
Elbow connections	2.5 cm diameter PVC elbows	232	75.00	11,600.00
TEE connections	2.5 cm diameter PVC TEEs	232	75.00	17,400.00
Pumps	1.75 KW pumps, 5 liters per second	2.00	10000.00	20,000.00
Solenoid single way valves	2.5 cm diameter	4.00	3500.00	14,000.00
Solenoid 3 way valves	2.5 cm diameter	2.00	5000.00	10,000.00
Glycol	40% propylene glycol solution	170 liters	75.00	12,750.00
Stainless steel enclosure				
size	0.025			
height - m - (0.025+0.115)x29+0.2	2.375			
Width - m - (0.1x6)	0.6			
Depth - m - 1.1	1.1			
H x W x D - 1.88 x 0.6 x 1.1 m3	2.375 x 0.6 x 1.1 m^3	1.57	30000.00	30,000.00
Electronic controller unit				30,000.00
Cost of Glycol chiller (-3C, 7RT) Power input : 7 (kW)	(3750 US$ x 177)			575,250.00
4 units of fan coil units	12,000 BTU capacity	4.00	25000	100,000.00
Other accessories			15000	15,000.00
Total				905,550.00

Table 4.2.3: Cost of production case 3

4.3 Power consumption for chillers & pumps during 6.5 hours period

Description	Case 1	Case 2	Case 3
Glycol circulation energy consumption for 6.5 hours (kWh)	3	3.5	2
Glycol chiller power consumption for 6.5 hours - KWh (Assume 80% of max power input is consumed in each hour)	70	40	40
Total energy per day (kWh)	73.2	42.5	41

Table 4.3.1 : Power consumption for chillers & pumps

According to the total energy consumption per day, the Case 1 is considerably very high compare to the Case 2 & Case 3. In the same time, the capital cost for Case 1 & Case 2 is more than Rs 182,000 & Rs 120,000 respectively. It is evident that when the set point of the chiller is lower, its energy consumption to produce each Refrigerant Ton is becoming higher. In the same time, it is required to consume higher percentage of glycol for lower set points, this will increase the pump energy due to higher density and viscosity of the 2nd refrigerant fluid. Operational expenditure is assessed and compare with two use cases, one for Hotel Room and other for Movie Theater. The Hotel room & Movie theaters are falling under two different electricity tariff as follows;

Customer Category H-2

Supply of electricity used for hotels approved by the Sri Lanka Tourism Development Authority. This rate shall apply to supplies at each individual point of supply delivered and metered at 400/230 Volt nominal and where the contract demand exceeds 42kVA.

Time Intervals	Energy Charge (LKR/kWh)	Fixed Charge (LKR/month)	Maximum Demand Charge per month (LKR/kVA)
Peak (18.30-22.30)	23.50	3,000.00	1,100.00
Day (5.30-18.30)	14.65		
Off-peak (22.30-05.30)	9.80		

Table 4.3.2 - [9] : Customer category H-2

Customer Category GP-2

Supply of electricity to be used in shops, offices, banks, warehouses, public buildings, hospitals, educational establishments, places of entertainment and other premises not covered under any other tariffs. This rate shall apply to supplies at each individual point of supply delivered and metered at 400/230 Volt nominal and where the contract demand exceeds 42kVA.

Time Intervals	Energy Charge (LKR/kWh)	Fixed Charge (LKR/month)	Maximum Demand Charge per month (LKR/kVA)
Peak (18.30-22.30)	26.60	3,000.00	1,100.00
Day (5.30-18.30)	21.80		
Off-peak (22.30-05.30)	15.40		

Table 4.3.3 : [9] : Customer category GP-2

Use case 1 : Energy cost for hotel room application. Operating one unit for 6.5 hours during off peak period

Description	Case 1	Case 2	Case 3
Total energy consumption per 6.5 hours per day (kWh)	73.2	42.5	41
Rupees per KWh during off peak	9.8	9.8	9.8
Total energy expenses per day (LKR)	717.36	416.50	401.80

Table 4.3.4 : Energy cost for use case of Hotel Room Application

Use case 2 : Energy cost for Movie theater application. Operating one unit for 6.5 hours during off peak period.

Description	Case 1	Case 2	Case 3
Total energy consumption per 6.5 hours per day (kWh)	73.2	42.5	41
Rupees per KWh during off peak	15.4	15.4	15.4
Total energy expenses per day (LKR)	1127.28	654.50	631.40

Table 4.3.4 : Energy cost for use case of Movie Theatre Application

4.4 Analysis of results

Use Case 1 : Hotel Rooms	
No of rooms (Dual sharing, 15’ x15’ room)	4
Total cooling load during the day per room (Ton hour)	8
Cooling load during - day time (Ton hour)	6
Cooling load during - peak time (Ton hour)	2
Rupees per kWh during – day time	14.65
Rupees per kWh during – peak time	23.5
Each split type AC unit - cooling capacity (BTU)	12,000
Each split type AC unit - cooling capacity (TR)	1
Power capacity (kW)	1.25
Total power consumption - day time (kWh)	7.5
Total power consumption - Peak time (kWh)	2.5
Cost of energy - daytime (LKR)	109.87
Cost of energy – peak time (LKR)	58.75
Per day per room cost (LKR)	168.62
Cost per day for four (4) rooms (LKR)	674.48
Cost of split type 12000 BTU AC unit -	120,000.00
Cost of 4 units	480,000.00
Annual cost of operations	242,820.00
Number of Ice Thermal storage based AC units required	1
Cost of Ice Thermal Storage based unit cost (LKR)	905,550.00
Annual cost of operation (LKR)	144,648.00
Savings per year (LKR)	98,172.00
Recovery period (Years)	4.3

Table 4.4.1: Use case Hotel Room

Use case 2 : Movie theatre	
Seat capacity	100 seats
Area	40' x 60 ' = 2400 sqft or 224 sqm
General operating hours2.30 Pm to 10.30 PM	8 hours per day
Max cooling capacity of all AC unit	8 RT
Total per day cooling load - Ton hour capacity (Assume full load during all 8 hours)	64 Ton hours
Hours of operation - daytime (2.30 pm-6.30 pm)	4
Hours of operation - Peak time (6.30 pm - 10.30 pm)	4
LKR per kWh - daytime	21.80
LKR per kWh – Peak time	26.60
Cooling capacity of each AC unit capacity – (BTU)	24000
Cooling capacity of each AC unit capacity – (TR)	2
Input power for each unit (kW)	2.5
Total units - Nos	4
Total AC capacity (BTU)	96,000
Total AC capacity (TR)	8
Total cooling capacity - Ton hours	64
Total power consumption – day time (kWh)	40
Total power consumption – peak time (kWh)	40
Cost of energy – daytime (LKR)	872
Cost of energy – peak time (LKR)	1064
Theater AC energy cost per day (LKR)	1936
Total AC energy cost per year (LKR)	696,960.00
Cost of split type 24000 BTU AC unit (LKR)	250,000.00
Cost of 5 units (LKR)	1,000,000.00
Annual cost of operations (LKR)	696,960.00
Capacity of Ice thermal Storage based AC unit (RT)	32
Number of Ice thermal storage based AC units required	2.00
Per unit cost (LKR)	905,550.00

Cost of all 3 units (LKR)	1,811,100.00
Cost of operation per year per unit (LKR))	227,304.00
Cost of operation for 3 units (LKR)	454,608.00
yearly saving from operational expenses (LKR)	242,352.00
Recovery period (Years)	3.4

Table 4.4.2 : Use case Movie Theatre

5 DISCUSSION & CONCLUSION

The cold thermal storage technology is a matured & well established MVAC load shifting technology with the history of several decades. This is widely used in large scale air conditioning projects and no due attention has yet being paid for using this technology for small scale air conditioning systems. This thesis made a full scale investigation on all the steps that is necessary to understand the economic feasibility & technological viability of developing a small scale cold thermal storage based air conditioning system.

The literature review section of the thesis, focused on all the fundamentals and all the available & established engineering methodologies of operating cold thermal storage systems in today's world. Phase Change Materials (PCM) were further reviewed in very detail giving due attention to its thermal, physical, kinetic & chemical properties. It was further reviewed the classification of PCMs as well. Applications of cold thermal storage systems were discussed under HVAC Cooling, Process cooling, District cooling plants & developing energy sources. The ice storage systems were further reviewed by giving more attention towards all the ice storage technologies available in the market. Cost benefits of ice storage system were critically analyzed in depth under capital cost saving & long term benefits. The ASHRAE design guidelines for cold thermal storage systems were also discussed in detail. The prospects of using cold thermal storage systems were reviewed under the Sri Lankan perspectives and relevant electrical tariffs.

The ice forming is a very critical factor in cold thermal storage systems. So this thesis made a complete review of mathematical analysis of ice forming inside a rectangular closed enclosure, published by P.Bhargavi, Dr. Radha Gupta & Dr. K.Rama Nirasimha. Here the top & bottom walls of rectangular chamber is subjected to a constant temperature (Subzero) which absorb the heat energy from the water to form ice. Initially the heat in the water is transferred through the top & bottom walls and when the temperature of water reaches freezing point, the ice forming is started. As the ice is formed inside the chamber, water/ice interface is moved towards the center of the enclosure. Here the freezing process is subjected to the pure conduction is considered for analysis. With using the energy balance equations and boundary conditions, several non-dimensional equations were derived. Those equations were converted to dimensional forms for the purpose of investigating the ice forming inside the rectangular closed enclosure with known dimensions. Those derived equations were later used for further analysis of ice forming inside various chamber units under different subzero temperatures of top & bottom walls.

The chapter 3 was completely dedicated to the research approach of the developing a small scale ice thermal storage based air conditioning system. Initially the research scope was clearly set by defining the design parameters of the proposed system as follows;

a) Capacity : 32 Ton hour
b) Ice making process uses secondary heat transfer fluid propylene glycol
c) Room space cooling is implemented through standard fan coil unit
d) Refrigeration is done by chiller unit, capacity of which should be decided through the research approach & analysis
e) Ice storage capacity : 1000 liters
f) Glycol pump will circulate propylene glycol inside a closed loop

The first step of the research process is to find the achievable ice thickness of water in a closed rectangular enclosure within 6.5 hours period by maintaining a constant subzero temperature at its top & bottom walls. Mainly 3 equations were used to find the ice thickness at a given time and the temperatures along that ice thickness. It was considered 3 different subzero wall temperatures at the walls of the ice making chamber to find the most economically viable dimensions for ice making chamber. Here the height is the decisive factor which depends on the wall temperature. The width and length has no special difference under all three wall temperatures. The subzero temperatures of -3^0C, -6^0C & -12^0C were selected to be assessed under proposed research procedure. The achieved ice thickness for due to one wall was calculated at every 20 minutes during the 6.5 hours. The results of 3 different achievable ice build thicknesses are indicated in the table 4.5.1.

Case	Wall temperature (^{0}C)	Achievable ice thickness in 6.5 hours period (cm)	Primary ice making chamber dimensions (cm)		
			Height	Width	Length
Case 1	-12	6.5	13	10	110
Case 2	-6	4.6	9.2	10	110
Case 3	-3	3.25	6.5	10	110

Table 4.5.1: Summary of all 3 cases investigated

The length and width of the ice making chamber was fixed to 110 cm & 10 cm respectively. The primary ice making chamber sizes were defined for all three cases by considering both the top & bottom walls do maintain a constant temperature glycol circulation system to remove the heat energy from the water to form ice.

The total heat energy removed during the ice forming process was calculated for all 3 cases. The temperature (**Ts**) profiles along the built ice thickness were estimated. The dimensionless temperature ratio quantity **Θs** was initially estimated and thereafter temperature profile (**Ts**) along the built ice thickness was estimated. The temperature profile was calculated and tabulated for every 2.5 mm distance of built ice thickness in every 1 hour during 6.5 hours period. This procedure was repeated for all 3 cases. It was found that the rate of ice building becomes very slow as the ice is formed. This is mainly due to the low thermal conductivity and diffusivity of ice. The achievable ice thickness is high when the wall temperature is very low. Based on the temperature profiles of all 3 cases, the total heat energy removed during each hour was calculated and tabulated.

By considering the ability to manage, it was decided to have 6 units of primary IMC Ice Making Chambers) in series. This was called as composite ice making chamber. Each composite IMC is sandwiched between two glycol circuits at top & bottom. By considering the total heat energy removal, the total number of composite ice making chambers for each case was decided.

Under the ideal case, the glycol circuit was considered to be rectangular and without any temperature gradient along wall of the ice making chamber units. In order to convert this model to practical reality, temperature gradient along the glycol circuit was introduced for all three cases. The glycol circuit inlet and outlet temperatures are as follows for all 3 cases which are under investigation.

Case	Glycol circuit temperatures (^{0}C)		Temperature different (^{0}C)
	Inlet	Outlet	
Case 1	-12	-7	6
Case 2	-6	-3	3
Case 3	-3	-0.5	2.5

Table 4.5.2: Glycol circuit temperatures

Due to the temperature difference at inlet and outlet of the glycol circuit, the thickness of the ice is gradually decreased along the primary ice build chamber unit. In order to overcome this situation, counter flow arrangement was introduced. It was revealed that counter flow arrangement maintains an approximately same ice thickness along the ice making chamber but still there exists a reduction in the volume of ice compare to its initial estimated volume. Due to this finding, the primary ice making chamber heights were adjusted as follows for all 3 cases.

Case	Wall temperature (^{0}C)	Achievable ice thickness in 6.5 hours period (cm)	Primary ice making chamber dimensions		
			Height (cm)	Width (cm)	Length (cm)
Case 1	-12	6.5	11.5	10	110
Case 2	-6	4.6	8.00	10	110
Case 3	-3	3.25	4.6	10	110

Table 4.5.3 Achievable ice thickness in 6.5 hours

Another interesting finding of the investigation is that the rate of heat energy removal during each hour is decreased under all three cases as illustrated in table 4.5.4.

Period - hour	Energy removal rate in Composite Ice Making Chamber unit(kJ/s)		
	Case 1	**Case 2**	**Case 3**
1st hour	4.27	2.77	1.58
2nd hour	1.77	1.33	0.66
3rd hour	1.38	1.0	0.58
4th hour	1.13	0.81	0.42
5th hour	1.02	0.70	0.38
6th hour	0.93	0.65	0.34
6.5 hour	0.56	0.53	0.16

Table 4.5.4: Rate of energy removal in all 3 cases

This revealed that more than the 35% of the ice is formed during the 1st hour of the chiller operation. According to these findings, the required chiller sizes for all the investigated cases were found to be very large and not economical. As a remedy to this, it was investigated the possibility of introducing a chiller operational strategy. This simply allows to use low capacity chillers and limit its glycol circulation only to a selected composite ice making chamber units at a given time period. This enabled to employ chillers with low capacities as illustrated in the following table.

Case	Without operational strategy	With operational strategy		
	Required minimum chiller capacity (TR)	new chiller capacity (TR)	% of reduction of chiller capacity	Energy removal capacity of ice storage unit (Ton hour)
Case 1	15	7	53%	33.32
Case 2	13	6	46%	30.39
Case 3	14	7	46%	32.64

Table 4.5.5: Required chiller capacity with and without operational strategy

The table 3.4.4.2, Table 3.4.4.3 & Table 3.4.4.4 chiller operational strategy for all the investigated cases.

Even though the rectangular glycol circuits were used for analysis, they are not easy when it comes to real world manufacturing. The circular shape glycol circuits were introduced and it was assumed that the top & bottom surfaces of 24 circular pipes will form an equivalent to flat surface on the both sides as illustrated in the following figure.

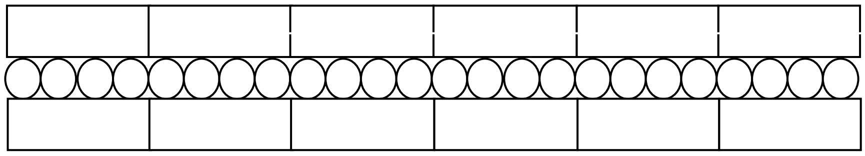

Fig 4.5.1 : Circular glycol circuit sandwiched between composite Ice making chamber units

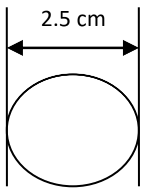

Fig 4.5.2: Glycol circulation pipe dimension

The flow rate of glycol was calculated using the energy balance at the pipe surfaces. According to the initial findings, the rate of energy removal during the ice forming is being reduced as the ice is formed. This phenomenon continues to reduce the pump flow rate during each hour while having the highest flow rate during the first hour. The pump capacity was further investigated by considering the frictions generated within the pipes and its fittings for each case. The maximum flow rate during the first hour of operation was considered for calculating pump capacities. By considering all the results of the investigation, the table 4.5.6 includes the summary developed for all the 3 cases.

Description		Case 1	Case 2	Case 3
Cold thermal storage capacity –(TR)		33.32	30.39	32.64
Ice build thickness in Primary Ice Making Chamber unit (cm)		11.5	8	5.0
Dimension of Primary Ice Making Chamber (PIMC) unit (cm)	H	13.00	9.2	5
	W	10	10	10
	l	110	110	110
No of Primary IMCs		72.00	96.00	174.00
No of Primary IMCs per horizontal composite IMC stack		6	6	6
No of horizontal composite IMC stacks per storage system		12	16	29
Complete volume of ice storage system *Cubic meters*		1.03	0.97	0.97
Dimensions of primary glycol circuit (cm)	Out diameter	2.5	2.5	2.5
	Inner diameter	2.3	2.3	2.3
	Length	110	110	110
No of 2.5 cm pipes per composite glycol circuit		24	24	24
No of composite glycol circuit stacks per Ice Thermal Storage Unit		13	17	30
Total length of glycol circuit pipes (2.5 cm diameter, Aluminium)		343.2 meters (24x13x1.1)	448.8 meters (24x17x1.1)	792 meters (24x30x1.1)
Hazen-Williams Coefficient of Aluminium		130-150	130-150	130-150
Energy extract from each Primary IBC unit in 6.5 hours (kJ)		6631	4675	3296
Total energy extract from Ice Thermal Storage unit in 6.5 hour (kJ)		415,361.52	448,800	477,432
Actual capacity Ton hours		33.36	30.7	30.27
energy removal during 1st hour (kJ)		184,464	159,456	165,387
Required chiller capacity under normal scenario (TR)		15	13	14
Required chiller capacity under new operational strategy (TR)		7	6	7
Maximum Flow rate inside primary glycol circuit (liters per hour)		32.4	35.28	24.12
Maximum Pump flow rate (liters per second)		2.6	3.7	4.9

Table 4.5.6 : Summary of all 3 cases

Thereafter all three cases under investigations were further itemized to find the manufacturing cost of each unit. The glycol chiller was found to be the most expensive item used in each case. The glycol chiller alone contributed more than 50% of the cost of manufacturing and this could further be reduced during the mass production by obtaining volume discounts for glycol chillers.

Description	Case 1	Case 2	Case 3
Glycol inlet temperature (^oC)	-12	-6	-3
Manufacturing cost (LKR)	1,087,175.00	1,025,025.00	905,550.00
Total energy consumption during 6.5 hour ice forming period (kWh)	73.2	42.5	41

Table 4.5.7 : Comparison of all 3 cases

According to the total energy consumption per day, the Case 1 is considerably very high compared to the Case 2 & Case 3. In the same time, the capital cost for Case 1 & Case 2 is more expensive than case 3. It is evident that when the set point of the chiller is lower, its energy consumption to produce each Refrigerant Ton is becoming higher. In the same time, it is required to consume higher percentage of glycol for lower set points; this will further increase the pump energy due to the higher density and viscosity. According to the above findings, the case 3 is found to be the most economically viable design from all the investigated cases.

5.1 Conclusion

Description	Case 1	Case 2	Case 3
Glycol circulation energy consumption for 6.5 hours (kWh)	3	3.5	2
Glycol chiller power consumption for 6.5 hours - KWh (Assume 80% of max power input is consumed in each hour)	70	40	40
Total energy per day (kWh)	73.2	42.5	41

Table 5.1.1: Power consumption for chillers & pumps during 6.5 hours period

According to the total energy consumption per day, the Case 1 is considerably very high compared to the Case 2 & Case 3. In the same time, the capital cost for Case 1 & Case 2 is more than Rs 182,000 & Rs 120,000 respectively. It is evident that when the set point of the chiller is lower, its energy consumption to produce each Refrigerant Ton is becoming higher. In the same time, it is required to consume higher percentage of glycol for lower set points; this will increase the pump energy due to higher density and viscosity.

Operational expenditure is assessed and compared with two use cases, one for Hotel Room and other for Movie Theater. The Hotel room & Movie theaters are falling under two different electricity tariff as follows;

	Customer Category H-2			**Customer Category GP-2**		
Time Intervals	**Energy Charge (LKR/kWh)**	**Fixed Charge (LKR/month)**	**Maximum Demand Charge per month (LKR/kVA)**	**Energy Charge (LKR/kWh)**	**Fixed Charge (LKR/month)**	**Maximum Demand Charge per month (LKR/kVA)**
Peak (18.30-22.30)	23.50	3,000.00	1,100.00	26.60	3,000.00	1,100.00
Day (5.30-18.30)	14.65			21.80		
Off-peak (22.30-05.30)	9.80			15.40		

Table 5.1.2 : Customer Category H-2 & GP-2 [9]

Customer Category H-2
Supply of electricity used for hotels approved by the Sri Lanka Tourism Development Authority. This rate shall apply to supplies at each individual point of supply delivered and metered at 400/230 Volt nominal and where the contract demand exceeds 42kVA.

Customer Category GP-2
Supply of electricity to be used in shops, offices, banks, warehouses, public buildings, hospitals, educational establishments, places of entertainment and other premises not covered under any other tariffs. This rate shall apply to supplies at each individual point of supply delivered and metered at 400/230 Volt nominal and where the contract demand exceeds 42kVA.

Description	Energy cost for hotel room application (Category – H2)			Energy cost for Movie Theatre application (Category – GP-2)		
	Case 1	**Case 2**	**Case 3**	**Case 1**	**Case 2**	**Case 3**
Total energy consumption per 6.5 hours per day (kWh)	73.2	42.5	41	73.2	42.5	41
Cost per KWh during off peak (LKR)	9.8	9.8	9.8	15.4	15.4	15.4
Total energy expenses per day (LKR)	717.36	416.50	401.80	1127.28	654.50	631.40

Table 5.1.3 : Energy cost for using proposed unit for 6.5 hours during off peak period

Use case of Hotel Room application	
Cost of split type 12000 BTU AC unit (LKR)	120,000.00
Cost of 4 units (LKR)	480,000.00
Annual cost of operations (LKR)	242,820.00
Number of Ice Thermal storage based AC units required	1
Cost of Ice Thermal Storage based unit cost (LKR)	905,550.00
Annual cost of operation (LKR)	144,648.00
Savings per year (LKR)	98,172.00
Recovery period - Years	**4.3**
Use case of Movie Theatre Application	
Cost of split type 24000 BTU AC unit (LKR)	250,000.00
Cost of 5 units (LKR)	1,000,000.00
Annual cost of operations (LKR)	696,960.00
Capacity of Ice thermal Storage based AC unit	32 Ton hours
Number of Ice thermal storage based AC units required	2.00
Per unit cost (LKR)	905,550.00
Cost of all 3 units (LKR)	1,811,100.00
Cost of operation per year per unit (LKR)	227,304.00
Cost of operation for 3 units (LKR)	454,608.00
yearly saving from operational expenses (LKR)	242,352.00
Recovery period - Years	**3.4**

Table 5.1.4: Recovery period for Hotel Room & Movie Theatre applications

The case 3 design was selected for further investigation and operational expenditure was assessed and compared for two use cases, one for Hotel Room and other for Movie Theater. Recovery period was assessed by considering the savings from operational expenses compare to conventional approach. The use case of hotel room had 4.3 years of recovery period as detailed in table 4.1. The use case of movie theater had 3.4 years recovery period as detailed in table 4.2. By reducing the cost of chillers, it is possible to further reduce the recovery period. The outcome of this thesis clearly indicates the economic feasibility of manufacturing & operating of small scale cold thermal storage based air conditioning systems. This industry segment needs more research & development with the aim of further minimizing the cost of manufacturing & optimizing design.

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