Optimizing the Thermal Performance of an Ice Based Latent Heat Storage System for External Melt

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Optimizing the Thermal Performance of an Ice Based Latent Heat Storage System for External Melt

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Master of Technology in Mechanical Engineering

(Thermal Engineering)

By

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Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master of Technology in Thermal Engineering at Nirma University and has not been submitted elsewhere for a degree.
2. Due acknowledgement has been made in the text to all other material used.

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Undertaking for Originality of the Work

I, Prashant Chatterjee, Roll No. 11MMET14, give undertaking that the Major Project entitled "Optimizing the Thermal Performance of an Ice Based Latent Heat Storage System for External Melt" submitted by me, towards the partial fulllment of the requirements for the degree of Master of Technology in Mechanical Engineering (Thermal Engineering), under Institute of Technology, Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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Abbreviation

A	=	area (m^2)
Q_e	=	refrigeration capacity of compressor (kW)
h_{sf}	=	latent heat of fusion (kJ/kg)
m	=	mass of ice formed (kg)
r	=	ice radius (m)
r_i	=	inside radius (m)
r_o	=	outside radius (m)
ρ	=	density (kg/m^3)
L	=	length of an evaporator coil (m)
R	=	thermal resisitance (m^2.K/W)
T_f	=	freezing temperature of water (ºC)
T_e	=	evaporator temperature (ºC)
T_c	=	condensor temperature (ºC)
h_i	=	inside heat transfer coefficient (W/m^2.K)
h_o	=	outside heat transfer coefficient (W/m^2.K)
F	=	fouling factor for the inside of the pipe wall(m^2.ºC/W)
k_l	=	thermal conductivity for the pipe (W/m.K)
k	=	thermal conductivity for the ice (W/m.K)
R	=	overall thermal resistance (m^2.k/W)
U	=	overall heat transfer coefficient (W/m^2.K)
μ	=	dynamic Viscosity (Pa.S)
C_p	=	specific heat (kJ/kg.K)
ν_f	=	specific volume(m^3/kg)
σ	=	surface tension (N/m)
Nu	=	Nusselt number
Pr	=	Prandtl number
Re	=	Reynolds number
Fr	=	Froude number
Co	=	convection number
F_{cb}	=	convection boiling factor
F_o	=	enhancement factor
h_{cb}	=	covective boiling heat transfer coecient(W/m^2.K)
q”	=	heat flux(W/m^2.K)

T_{WONB}	=	wall temperature for the onset of nucleate boiling(^{o}C)
T_w	=	wall temperature (^{o}C)
x	=	quality of vapour
P_e	=	power consumption by compressor (kW)
SPC	=	specific power consumption
LHS	=	latent heat storage (kg)
t_s	=	ice thickness (mm)
ρ	=	density (kg/m^3)
d_h	=	hydraullic diameter (m)
V_c	=	volume occupied by coil in tank (m^3)
V_c	=	volume of tank (m^3)
V_c	=	volume available for flow (m^3)
D_e	=	equivalent diameter (m)
h_f	=	enthalpy of saturated fluid (kJ/kg)
h_f	=	enthalpy of saturated vapor (kJ/kg)
G	=	mass velocity (kg/m^2.s)

Abstract

The reasons for the use of an Ice Based Latent Heat Storage System are mainly of economical and environmental nature. Ice Based Latent Heat Storage System systems will improve the energy systems as it reduces load during on-peak hours. It also increase environmental standards as it contributes to the phase-out of synthetic refrigerants. It uses the lower electricity tariff rates available during off-peak hours for state electricity boards of different states. Application of ice based latent heat storage systems are for refrigeration installations which are used in many industries, mainly food industries and HVAC systems.For these applications it is important that they reduce the energy losses and that too should be cost effective improvement.

In the present study theoretical investigations of ice based latent heat storage system for an external melt are carried out.The system consists of tank with tubes submerged in water, a condenser, compressor, accumulator and primary as well as secondary refrigeration cycles. A case study is performed to optimize the ice based latent heat storage system. Optimization of the ice that can be formed around the tubes during on-peak hours is done using the enegy saving that is achieved by operating the system during off-peak hours. The optimum ice thickness achieved is 30 mm. This amount of ice can be built during on-peak hours of running without any extra cost. The saving achieved by operating during off-peak hours is 30.5% with respect to on-peak hours.

To find the energy efficient and economic option and a more efficient design is proposed, having better efficiency then the conventional system. Proposed design will lead to reduction in work of compression and increment in coefficient of performance of the system. Later, experimental verification is performed on ice based latent heat storage system for industrial applications.

Keywords: ice based latent heat storage system, ice, ice bank tank system, external melt, heat transfer analysis, optimization,cooling load, power consumption.

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Chapter 1

Introduction

The aim of using ice storage systems is mainly to reduce peak power requirement and use of lower electricity tariff rates available. They are most cost effective systems and provides a steady source of low temperature fluid for process cooling applications. Ice latent heat storage systems remove heat from a storage medium for use at another time. In case of conventional systems, they are operated whenever there is a requirement of cooling. The cooling requirements are mostly occurs during peak periods resulting in high national electricity demands. Ice latent heat storage system is an alternative strategy that can be used to avoid the high peak load demand charges for cooling. These systems are also environmental friendly because they help lower energy demand and reduce greenhouse gas emission. Ice latent heat storage is an economically attractive approach to meet the cooling loads if one or more of the following conditions apply[12]:

• Loads are of short duration

• Loads occur infrequently

• Loads are cyclical in nature

• Loads are not coincident with the availability of the energy source

• Energy costs are time-dependent (e.g., time-of-use energy rates)

• Charges for peak power demand are high

• Utility rebates, tax credits, or other economic incentives are provided for the use of load-shifting equipment

• Energy supply from the utility is limited, thus preventing the use of full-size non storage systems

1.1 History of Thermal Energy Storage

Thermal Energy Storage (TES) is the term used to refer to energy storage that is based on a change in temperature. TES can be hot water or cold water storage where conventional energies, such as natural gas, oil, electricity, etc. are used (when the demand for these energies is low) to either heat or cool the storage water. The energy is basically transferred, from conventional energy sources, to a temperature differential in the storage water that can be utilized during high energy demand periods. The typical domestic hot water heater is an example of thermal hot water storage that is popular throughout the world. Thermal hot water storage and thermal chilled water storage applications are very common, and are used for both process and comfort heating and cooling systems. In the 1930's, dairy farmers began using ice storage to cool the daily batches of fresh milk. Normally, the milk cooling required large chillers that cooled for only a few peak hours (twice a day). Ice storage offered the advantage of using much smaller refrigeration equipment that could build and store ice over a 10 to 12 hour period. Each batch of fresh milk could be cooled quickly using ice melt, and the thermal ice storage system could be recharged in time for the next milking. Thermal Ice storage still provides a considerable amount of milk cooling in the dairy industry.[13]

Ice has played a major role in comfort cooling applications as well. Even the definition of a ton of cooling is derived using ice. The latent heat of fusion (phase change of water to ice or ice to water) is 334 kJ/kg. One ton of ice is 907.185 kg. Therefore, the energy required to change 907.185 kg of water to ice would be 334 kJ/kg $\times$ 907.185 kg = 303000 kJ. To accomplish this in a 24 hour period, the hourly energy rate would be 12625 kJ per hour. This energy rate is defined as a ton of air conditioning. In the late 1970's, a few creative engineers began to use ice storage for air conditioning applications. During the 1980's, progressive electric utility companies looked at thermal energy storage as a means to balance their generating load and delay the need for additional peaking power plants. These utility companies offered their customers financial incentives to reduce their energy usage during selected on-peak hours. On peak electric energy (kWh) and electric demand (kW) costs increased substantially, yet non-peak energy costs remained low.

1.2 Fundamental of Ice Based Latent Heat Storage System

The principle of this system is that cooling is produced in the form of ice and the ice is stored in a well-insulated storage tank. The coolness of ice is then extracted from it and is used for cooling during peak load requirement and results in reduced peak demand.

Ice thermal storage systems uses the latent heat of fusion (phase change of water to ice or ice to water) i.e. 334 kJ/kg. Ice is used as a phase change material to store the latent heat. Latent heat storage offers higher energy storage density and higher energy density allows compact system design resulting in capital cost savings while temperature stability means preserving the quality of the product. Thermal energy is stored in ice at 0°C the freezing point of water. The plant model of the ice based latent heat storage can be easily explained through the Figure 1.2:

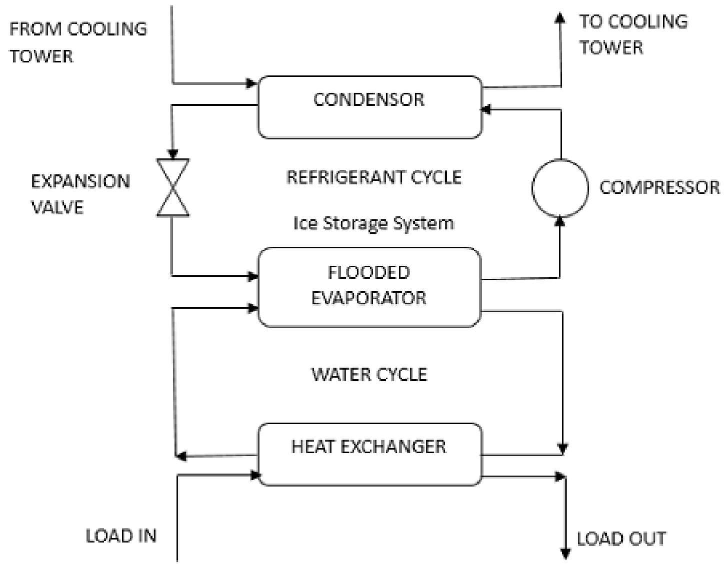

Figure 1.1: Plant model of ice thermal storage system

It is a plant model and is realistic in commercial applications. Generally, ice storage systems can be used as "Full Ice-storage" and "Partial Ice-storage" systems, depending on the amount of load transferred from the on-peak to the off-peak period.

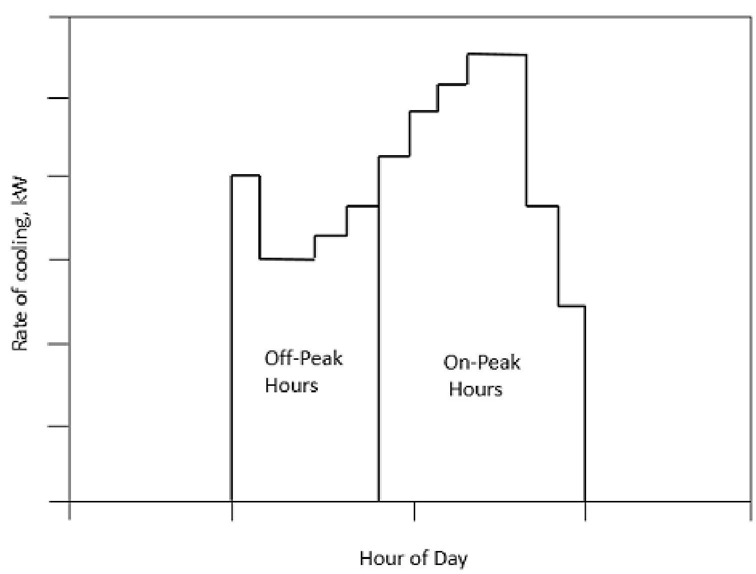

Figure 1.2: General load profile

Figure 1.2 shows the general load profile formed for different load available at different hour of the day. In a "Full Ice-storage" system, the refrigeration compressor does not operate during on-peak periods, and all the cooling is supplied from the ice stored as shown in Figure 1.3. Ice is built during the off peak hours and used later during peak hours.

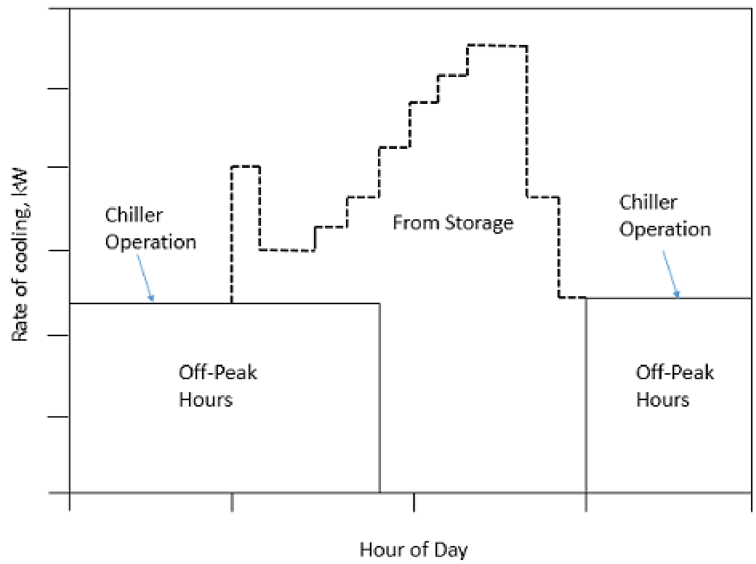

Figure 1.3: Load profile-full storage

In a "Partial Ice-storage" system, some or all of the refrigeration compressors operate during the peak period to supplement the cooling supplied by the stored ice as shown in Figure 1.4. The "Partial Ice-storage" system reduces the size and cost of the ice storage tanks and the refrigeration compressors. However, the saving in electricity costs is not as significant as using the full storage because of the need to operate compressors during the peak period. The optimum amount of storage is achieved by maintaining a minimal equipment cost while maximizing electrical savings.

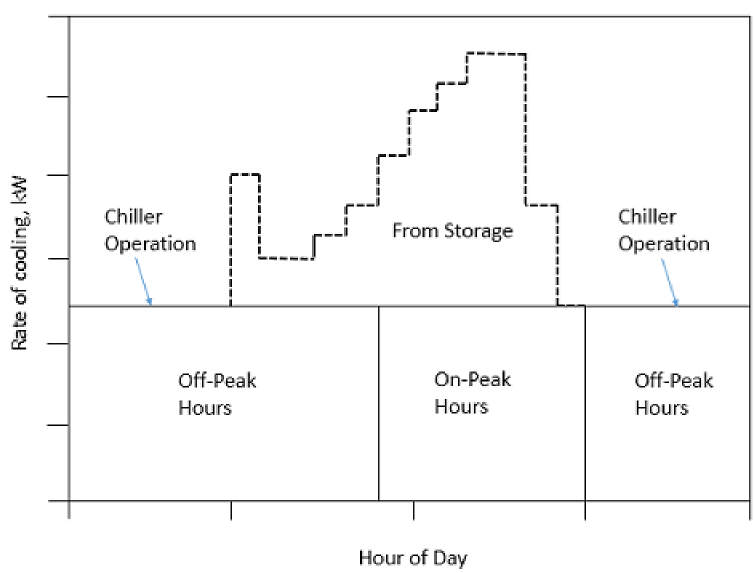

Figure 1.4: Load profile- partial storage

1.3 Advantages of Ice Based Latent Heat Storage System

The primary reasons for using ice latent heat storage are typically economic. The following are some of the key benefits of storage[17]:

1.3.1 Lowest First Cost

Systems having ice thermal storage system can be installed at lower first cost than traditional systems when the colder supply of water is available from ice. The savings that results from the use of ice thermal storage are:

1. Smaller Chillers and Heat rejection: By designing the system around 24-hour per day chiller operation, the size of the chillers and cooling towers or air cooled condensers required for an ice storage system is significantly reduced, when compared to conventional chillers and heat rejection equipment sized for the instantaneous peak load.

2. Reduced Pump and Pipe Size: Saving in the chilled water distribution loop are realized when the system design incorporates reduced flow rates that result from using a larger temperature range in the water loop. Use of larger temperature range results in reduction in pipe size. Condenser water pipe sizes are reduced due to lower flow requirements for the smaller chiller. Pump savings due to reduced chilled water and condenser water flow rates are also realized.

3. Reduced Electrical Distribution: Smaller chillers, heat rejection equipment and pumps require less horsepower than a traditional system, which results in smaller transformers, switchgear, wire sizes and starter panels.

4. Reduced Generator Size: If a facility has a generator for daily or back-up-power, the size of the generator will be significantly reduced when the peak electrical load of the facility is reduced using ice storage.

1.3.2 Reduced Energy Cost

An ice based latent heat storage system reduces peak demand, shifts energy usage to non-peak hours, saves energy, and reduces energy costs.

1.3.3 Variable Capacity

The ice based latent heat storage system will maintain a constant supply temperature regardless of the variations in instantaneous cooling demand. The flow and entering water temperature set the instantaneous capacity.

1.3.4 Reduced Maintenance

The ice storage coils have no moving parts, so very little maintenance is required. Because the chillers, pumps and heat rejection equipment are smaller, ice storage system will have less maintenance than a traditional system. The ice storage system also allows a chiller to undergo routine maintenance during the day when the ice storage can handle the system load.

1.3.5 Environmentally Friendly

Reducing energy consumption and using electricity at night will reduce global warming. The electricity generated at night generally has a lower heat rate (lower fuel use per power output), and therefore lower carbon dioxide and greenhouse gas emissions resulting in less global warming.

With smaller chillers, an ice storage system reduces the amount of refrigerant in a system. Hence help in reducing global warming.

1.4 Applications

1.4.1 HVAC cooling

Comfort air conditioning systems are ideal candidates for thermal ice storage. Large horsepower cooling compressors operate during peak summer energy periods. Latent heat storage can transfer all or part of this energy to non-peak hours. Cooling may be required year round in some locations, while only seasonally in others. Typical applications are: office buildings, hospitals, large retail stores, schools (elementary - university), sports arenas and stadiums.

1.4.2 Process cooling

This is a broad category with various applications including:

• Batch cooling where added heat must be removed (such as food processing)

• Batch cooling where internal heat is created that must be removed (such as chemical reactions or pharmaceutical processing)

• Cooling systems where a constant temperature must be maintained during all loads.

• Data centres where energy management plays a part along with multiple layers of equipment redundancy.

1.4.3 Turbine - inlet air cooling

Peak turbine design efficiency occurs when the inlet air temperature is between 4.44°C & 10.0°C. However, on hot summer days, when maximum generating capacity is essential, the ambient air temperature may be above 32.2°C and the turbine efficiency may be substantially reduced. Many utilities have used thermal ice storage systems to lower the inlet air temperature entering the turbine. By building ice during off-peak hours, and using the latent heat ice storage during on-peak hours, the inlet air, delivered by the air handling units, can be cooled below 10.0°C. Thus, the turbine efficiency and output can be maintained at design levels.

1.4.4 District cooling plants

District cooling plants for university campuses or metropolitan areas will have a variety of cooling applications: office buildings, hotels, dormitories, classrooms, labs, sports arenas and data centres. Ice storage takes advantage of this diversity of cooling loads and enhances the performance of the entire cooling plant.

1.4.5 Developing energy sources - solar & wind

The technologies for both solar and wind energy are improving every year. These energy sources supply electric energy that can be used throughout the system, and are not limited to cooling. However, the creation of these energies often relies on weather conditions that may cause fluctuations in the energy flow. Ice based latent heat storage is a perfect complement for these systems. Ice storage will reduce the amount of solar or wind energy required by taking the responsibility for a large portion of the air conditioning load.

1.5 Ice Based Latent Heat Storage System

In ice based latent heat storage systems as shown in Figure 1.5. Refrigerant/secondary coolant is flown inside the coils and a layer of ice is formed as the refrigerant/secondary coolant keeps on taking the heat from the water filled inside the tank. Most of the systems uses a coil of 42.4 mm outer diameter. The process of forming the ice is called charging process. Afterwards, the chilled water is used to meet the load. The chilled water is circulated generally through the heat exchangers and returns back to the tank, this starts melting the ice as water gains heat while providing cooling to load. The process of ice melting is called discharging process. The method of ice melt can be divided into two types as described below:

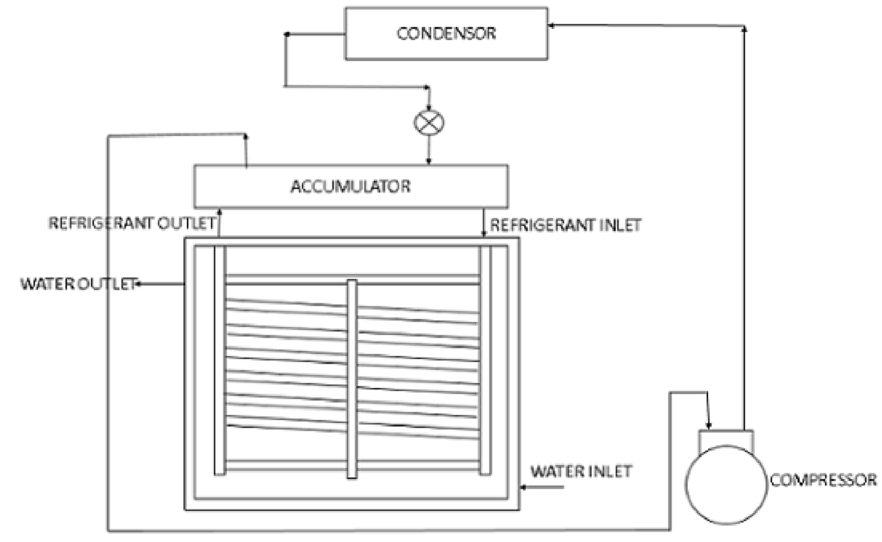

Figure 1.5: Ice based latent heat storage system

1.5.1 A. External Melt Ice-on-Coil Storage.

The oldest type of ice storage is the refrigerant-fed ice builder, which consists of refrigerant coils inside a storage tank filled with water. Refrigerant is passed through inside of the coils and evaporates by taking heat from water. The tank water on the outside of the coils starts forming ice. Ice around the coils can be formed upto a thickness of 63.5 mm.This is called as the charging process. And after this ice is melted from the outside of the formation (hence the term external melt) by circulating the return water through the tank, whereby it again becomes chilled.This process is called as discharging process. Agitator is used to agitate the water to promote uniform ice build-up and melting. Instead of refrigerant, a secondary coolant (e.g., 25% ethylene glycol and 75% water) can be pumped through the coils inside the storage tank. Figure 1.6shows the extermal melt ice-on-coil storage system.

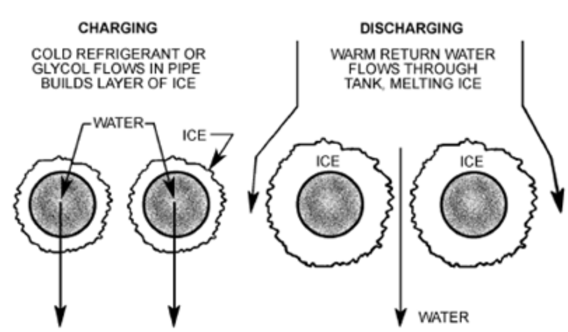

Figure 1.6: External melt ice-on-coil storage

Major concerns specific to the control of ice-on-coil storage are: (1) limiting ice thickness (and thus excess compressor energy) during the build cycle and (2) minimizing the bridging of ice between individual tubes in the ice bank. Bridging must be avoided because it restricts the free circulation of water during the discharge cycle. Though not physically damaging to the tank, this blockage reduces performance, allowing a higher leaving water temperature because of the reduced heat transfer surface.

1.5.2 Internal melt ice-on-coil storage

In this type of storage device, coils or tubing are placed inside a water tank. The coils occupy approximately 10% of the tank volume; another 10% of the volume is left empty to allow for the expansion of the water upon freezing; and the rest is filled with water. A secondary coolant solution (e.g., 25% ethylene glycol and 75% water) is cooled by a liquid chiller and circulated through the coils to freeze the water in the tank as shown in Figure 1.7. The thickness of ice on the coils and the percentage of the water in the tank that is frozen depend on the coil configuration and on the type of system. During discharge, the secondary coolant circulates to the system load and returns to the tank to be cooled again by the coils submerged in ice.

A standard chiller can provide the refrigeration for these systems. During the charging cycle for a typical system, the chilled secondary coolant exits from the chiller at a constant –3.5°C and returns at –0.5°C. When the tanks are 90% charged, the chiller inlet and outlet temperatures fall rapidly because there is little water left to freeze. When the chiller exit temperature reaches approximately –5.5°C, the chiller is shut down and locked off for the remainder of the charging period so that it does not short-cycle or recirculate because of convection flow through the pipes. As a result, the chiller remains fully loaded through the entire cycle and keeps the system running at its maximum efficiency because the exit-temperature thermostat, set at –5.5°C, prevents operation beyond that necessary to fully charge the storage tanks. The temperatures for a given system may vary from this example.

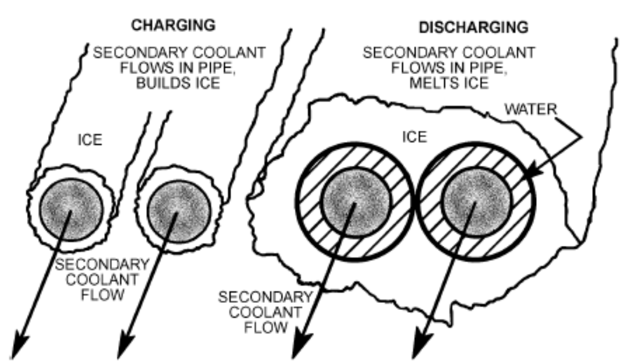

Figure 1.7: Internal melt ice-on-coil storage

Because the same heat transfer surface freezes and melts the water, the coolant may freeze the water completely during each charge cycle, minimizing loss in efficiency. A temperature-modulating valve at the outlet of the tanks keeps a constant flow of liquid to the load . Under a full-storage control strategy, the chiller is kept off during discharge, and the modulating valve allows some fluid to bypass the tanks to supply the load as needed. If a partial-storage control strategy is followed, the chiller thermostat is reset from the –5.5°C set point required for charging, up to the design temperature of the cooling coils (e.g., 6.7°C), during the discharge cycle. If the load on the building is low, the chiller operates to meet the 6.7°C setting without depleting the storage. If the load is greater than the chiller's capacity, the exiting temperature of the secondary coolant rises, and the temperature-modulating valve automatically opens to maintain the design temperature to the coils. Because water increases 9% in volume when it turns to ice, the water level varies directly with the amount of ice in the tank as long as all of the ice remains submerged. This water displaced by the ice must not be frozen, or it will trap ice above the original water level. Therefore, no heat exchange surface area can be above the original water level.

1.6 Operating Modes and Strategies

Operating mode describes which of several possible functions the system is performing at a given time. This should be defined by a specific control sequence.

The five most common operating modes are described in Table1.1. The available operating modes differ for individual thermal storage systems. Some systems may include fewer than the five basic modes. For example, the option to meet the load while charging may not be available. In some installations, operation to meet loads may be defined by a single operating mode that includes discharging-only at one end of a continuum and direct equipment-only at the other. In fact, many systems operate with just two modes: daytime and night time operation.

Many systems also include other operating modes. Some examples include

- Charging cool storage from free cooling
- Charging cool storage while recovering condenser heat
- Charging heat storage with recovered condenser heat
- Discharging at distinct supply temperatures
- Discharging in conjunction with various combinations of available equipment

Table 1.1: Operating modes for cool storage

Operating Mode	**For Cool Storage**
Charging Storage	Operating cooling equipment to remove heat from storage
Charging Storage while meeting loads	Operating cooling equipment to remove heat from storage and meet loads
Meeting loads, from discharging storage only	Discharging (adding heat to) storage to meet loads without operating cooling equipment
Meeting loads, from discharging storage and direct equipment operation	Discharging (adding heat to) storage and operating cooling equipment to meet loads
Meeting loads, from direct equipment operation only	Operating cooling euipment to meet loads (no fluid flow to or from storage)

1.6.1 Control sequences

An operating mode is defined for a given system by its control sequence. The control sequence defines what equipment is running, including the values of their set points, and what actions should occur in response to changes in load or other variables.

1. **Charging storage** - Control sequences for the charging mode are generally easily defined. Typically the generation equipment operates at full capacity with a constant supply-temperature set point and a constant flow through the storage. This operation continues until the storage is fully charged or the period available for charging has ended. Under this basic charging control strategy, the entire capacity of the equipment is applied to charging storage.

2. **Charging storage while meeting load** - A control sequence for charging storage while meeting load generally also operates the generation equipment at its maximum capacity. Capacity that is not needed to meet the load is applied to charging storage. Depending on system design, the load may be piped either in series or parallel with storage under this operating mode. Some systems may have specific requirements for this mode. For example, in an ice storage system with a heat exchanger between glycol and water loops, the control sequence may have to address freeze protection for the heat exchanger.

3. **Meeting load from discharging only** - A control sequence for the discharging-only mode (full storage or load-shifting operation) is also straightforward. The generating equipment does not operate and the entire load is met from storage.

4. **Meeting load from discharging and direct equipment operation** - Control sequences for this mode are more complex and must regulate what portion of the load,

at any time, will be met from storage and what proportion will be met from direct generation. These partial-storage sequences have been primarily developed for and applied to cool storage. Although they could also be applied to heat storage, the following discussion is in terms of cool storage. Three common control sequences are chiller-priority, storage-priority, and constant-proportion or proportional.

5. **Chiller priority control** - Operates the chiller, up to its available capacity, to meet loads. Cooling loads in excess of the chiller capacity are met from storage. If a chiller demand limit is in place, the available capacity of the chiller is less than the maximum capacity. Chiller-priority control can be implemented with any storage configuration. However, it is most commonly applied with the chiller in series upstream of storage. A simple method of implementing chiller-priority control is to set both the chillers on board supply-temperature set point and the temperature set point downstream of storage to the desired chilled-water supply temperature. When the load exceeds the chiller capacity, the chiller discharge temperature exceeds its set point, and some flow is diverted through storage to provide the required additional cooling. Sensing errors in the storage downstream measurement

1.6.2 Control strategies

A control strategy is the sequence of operating modes implemented under specific conditions of load, weather, season, etc. For example, different control strategies might be implemented on a design day in the summer, a cool day in the summer, and a winter day. The control strategy further defines the actions of individual control loops and the values of their set points in response to changes in load or other variables.

1.6.3 Operating strategies

The operating strategy defines the overall method of controlling the ice storage in order to achieve the design intent; it determines the logic that governs the selection of operating modes such as charging, meeting the load from chiller(s) only, or meeting the load from discharging storage. The operating strategy also determines which control strategy is implemented within each mode. It is important to distinguish between the operating strategy, which defines the higher-level logic by which a system will be operated, and the various control strategies, which implement the operating strategy through selection of operating modes.

1.7 Motivation for the Present Study

Refrigeration systems in industry and air conditioning in commercial and residential buildings is the largest single contributor to electrical peak demand specially during summer daytime. This requires the electric suppliers to bring additional, more costly generating equipment on line or to import the required energy to handle this increased demand. Commercial users, whose large cooling loads greatly contribute to the need for these seldom used generating stations, are charged more for this on-peak energy, in the form of demand charge which is based on their highest on-peak demand for electricity.

Implementation of a Ice Based Latent Heat Storage System enables to shift electric load to off-peak hours which significantly lower energy charges and reduces peak loads in power system during the cooling period. Besides, it can also lower total energy usage as well. Generally, the electric supplier's generating capacity is under-utilized at night and, consequently, its rates are lowest then. Whenever the maximum cooling load is higher than the average load or the electric billing schedule includes high demand charges with difference between on- and off-peak rates, application of the latent heat ice storage may be economically justified. The higher the load factor the more attractive is ice based latent heat storage system to the customer. Particularly well suited to ice based latent heat storage are sports and entertainment facilities, convention centres, churches, airports, schools and universities, military bases and office buildings as well as various industrial applications.

Office building cooling loads often peak at a level of two or more times higher than the daily average load. Likewise, process cooling in various industrial applications such as food processing in food industry (milk cooling, carcass spray cooling, fish processing, beer and vine production) and drugs treatment in pharmaceutical industry have load peaks that rise much higher than the average load. Dairies are a good example where large short loads often are required. Typically once a day, in the morning, for a limited duration, a large quantity of incoming fresh milk must be quickly cooled. Such a high cooling demand requires either a large capacity refrigeration unit, which would be idle for the most of the day, and would consume high (electrical) power when busy, or a smaller refrigeration unit that would operate rather uniformly during low-load periods and accumulate the cooling energy to meet the peak load when needed.

Moreover, Ice Based Latent Heat Storage System not only can significantly cut operating costs but they can also substantially reduce capital outlays by downsizing the cooling equipment when systems are suitably designed for new commercial and industrial application.If implemented, Ice Based Latent Heat Storage Systems can increase environmental standards of the refrigeration system as it contributes to the phase-out of synthetic refrigerants.Today the ozone depletion potential (ODP) and global warming potential (GWP) of the synthetic refrigerants used in the majority of airconditioning and refrigeration installations are the major environmental concerns. By implementation of the indirect cooling to Ice Based Latent Heat Storage System, the refrigerant charge and related emissions could be reduced by a factor of ten compared to a complete direct expansion installation. Besides, indirect systems are more suitable for use of environmentally friendly, natural refrigerants such as ammonia (R717), carbon dioxide (R744) or propane (R290).

Chapter 2

Literature Review

A literature review was conducted to consider the previous work related to using of ice as thermal energy storage system and different analytical and numerical analysis performed of ice freezing and melting.

2.1 Previous Studies Conducted

1) Modelling of an ice-on-coil thermal energy storage system, by Alex H. W. Lee and Jerold W Jones[1] at The University of Texas, Department of Mechanical Engineering, U.S.A, in 1995 formed a stand-alone analytical models for ice-on-coil thermal energy storage(TES) system for both charging and discharging modes . Basic heat transfer analysis is performed and characteristics of the charging and discharging modes are determined from the model. Experimental results were used to validate the models. The performance predicted by models tended to be closer to that of ideal systems. In the experimental tests, there were errors and uncertainties associated with uneven ice building and melting, heat losses, and other instrumentation and human errors. On the other hand, the models used heat transfer coefficients which generally had accuracies in the order of 20 to 30%. As such, errors of 10 to 15% in the modelling results should be considered acceptable.

A. Cooling Charge Model

An analytical model developed was used to predict the ice-building performance of the evaporator/ice storage tank, with external build and melt. The output results of the cooling charge model were the volume of ice in the TES tank and cooling charge rate. Fig.2.1, 2.2and 2.3compare the predicted ice volume and the experimental results for the outdoor temperature 16 °C,25 °C and 32 °C, respectively. The model prediction showed excellent agreement with the experimental data. The slight discrepancy for time, t $>$12h, is due to non-uniform freezing of ice or experimental errors. An error analysis of the ice volume indicated that errors in the model predictions were within 5% of the experimental values.

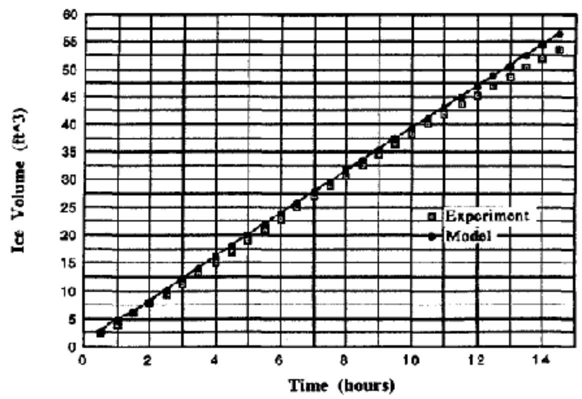

Figure 2.1: Comparison of ice built on coils for outdoor temperature of 16 °C[1]

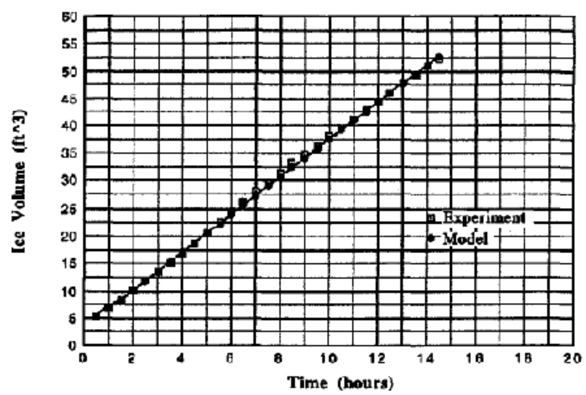

Figure 2.2: Comparison of ice built on coils for outdoor temperature of 25 °C[1]

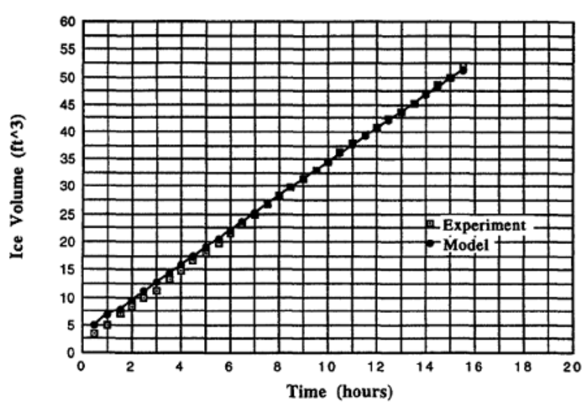

Figure 2.3: Comparison of ice built on coils for outdoor temperature of 32 °C[1]

Fig.2.4, 2.5 and 2.6 compare the predicted cooling charge rate and the experimental results for outdoor temperatures of 16 °C, 25 °C, and 32 °C, respectively. The experimental data indicated some inconsistent trends due to the variation of refrigerant mass flow rate associated with surging or "hunting" of the thermal expansion valve. To obtain the accurate measurement, the flow rate reading was taken 35 times during each of five sweeps of data sets. An energy balance was performed on all the refrigeration/ ice storage components to confirm errors within reasonable limits of less than 3%. Another reason for the discrepancies between the experimental and predicted results could be the non-uniform freezing of ice on the coils. In any case, the model prediction were at most within 12% of the experimental values.

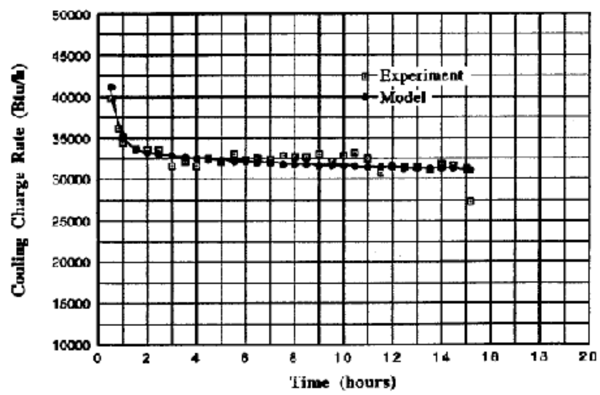

Figure 2.4: Comparison of cooling charge rate for outdoor temperature of 16 °C[1]

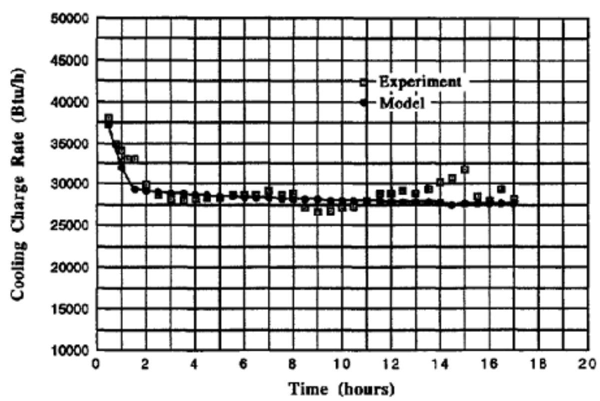

Figure 2.5: Comparison of cooling charge rate for outdoor temperature of 25 °C[1]

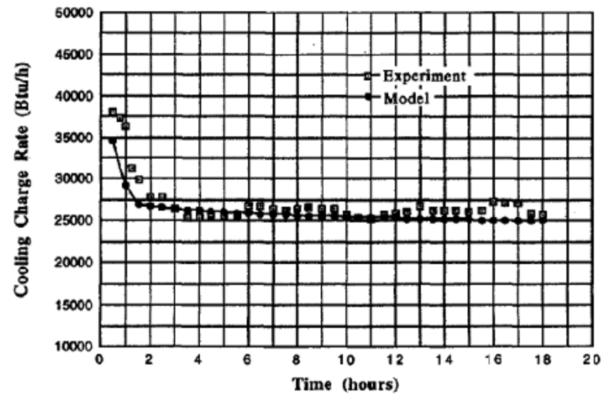

Figure 2.6: Comparison of cooling charge rate for outdoor temperature of 32 °C[1]

Sensitivity analysis of cooling charge model

Sensitivity of the solution to different variables is checked, a factorial design based on the pioneering work of Fisher was used. The sensitivity analysis was conducted by changing all the variables simultaneously, rather than changing one variable and keeping others constant. This method of analysis allowed one to inspect the interaction effects of the variables in a very small number of trials. Table 2.1 presents the results of the sensitivity analysis for the cooling charge model. Case 8 is the base case of the sensitivity analysis. The – and + values were those used in the base and change cases, respectively. The + values of the variables T_f – T_{max}, h_i, and h_o indicated increases of 25% from the base case. The – and + values of F indicated the presence and absence of fouling respectively. The error values varied slightly with time. The values shown were averages over the entire time period.

The results suggest that the cooling charge rate and ice radius were very sensitive to the variable, Tf – Tmax, temperature difference between the freezing and maximum evaporator temperature with net errors of -24.5 and -11.3%, respectively. In order to study the sensitivity of the variables and their associated interactions, the system procedure REGRESSION from MINITAB was used. The statistical analysis showed that the variable Tf – Tmax and the interactions (Ti- Tmax) × ho, hi × F, and hi × ho × F had significant effects on both cooling charge rate, Q_{ice}, and the ice radius. The variable T_f – T_{max} should be carefully monitored in the cooling charge model. Although the statistical analysis showed that the variables h_i, h_o and F were not sensitive to the model output.

Table 2.1: Results of sensitivity analysis for cooling charge model[1]

Case	T_f-T_{max}	h_i	h_o	F	% error in Q_{ice}	% error in ice radius
1	+	+	+	+	-9.4	-4.4
2	+	+	-	-	-19.2	-9.2
3	+	-	+	-	-17.2	-8.2
4	+	-	-	+	-19.2	-9.1
5	-	+	+	-	11.7	5.1
6	-	+	-	+	9.2	4.0
7	-	-	+	+	12	5.2
8	-	-	-	-	0	0

where, T_f-T_{max} is temperature difference between the freezing and maximum evaporator temperature.

h_i is internal heat transfer coefficient

h_o is external heat transfer coefficient

F is fouling factor

B. Cooling Discharge Model

An analytical model was developed to predict the ice melt performance of the evaporator/ ice storage tank. The model treated the water in the ice tank (excluding the ice) as the control volume and adopted the global energy balance approach. Ice were melt externally and no refrigerant flow was involved in this case. In order to predict the ice melt performance, a quasi-steady state energy balance was performed for the control volume. This resulted in the calculation of a new bulk temperature for the tank water. The bulk temperature was assumed to be uniform throughout the tank. The rate of change of internal energy in the ice tank can be expanded as the sum of the heat transfer between the ice and water and the sensible change in the water.

The output result of the cooling discharge model was the tank bulk temperature. Fig.2.7, 2.8, and 2.9 compare the predicted tank bulk water temperature and the experimental results for the indoor temperatures of 25°C, 27°C, and 30 °C, respectively. Model prediction showed excellent agreement with the experimental data. An error analysis of the tank bulk water temperature indicated that the model prediction were within 5% of the experimental values.

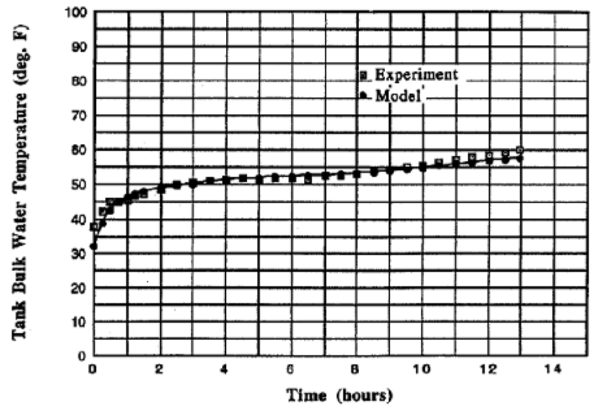

Figure 2.7: Comparison of tank bulk water temperature for indoor temperature of 25 °C[1]

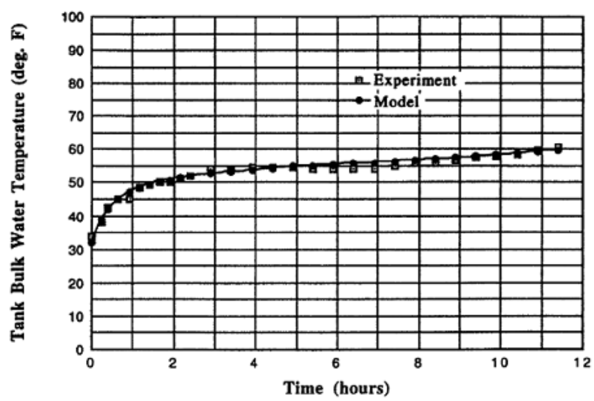

Figure 2.8: Comparison of tank bulk water temperature for indoor temperature of 27 °C[1]

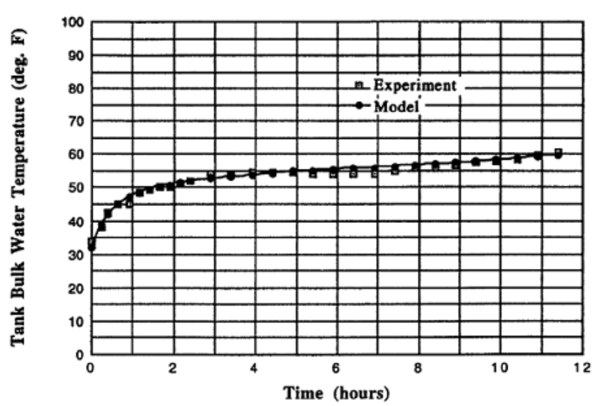

Figure 2.9: Comparison of tank bulk water temperature for indoor temperature of 30 °C[1]

Sensitivity analysis of cooling discharge model

The sensitivity analyses were conducted for one case at an indoor temperature of 25°C. Table2.2 presents the result of sensitivity analysis for the cooling discharge model. Case 8 was the base for the sensitivity analysis. The + values of the variables m, %ice, and Tci indicated increses of 5% from the base case. The + values of ho indicated increses of 25% from the base case.Among all variables ho and Tci affect the model more with net errors of – 2.0 and 3.8%, respectively. For a more precise analysis, the statistical system procedure REGRESSION was used to show variables h_o and T_{ci}, as well as the interactions $\dot{m}$ × %ice × T_{ci} and $\dot{m}_w \times h_o \times$ %ice, had significant effects on the bulk water temperature.

Table 2.2: Results of the sensitivity analysis for cooling discharge model[1]

case	$\dot{m}_w$	h_o	%ice	T_{ci}	% error T_{wtr}
1	+	+	+	+	2.4
2	+	+	-	-	-1.4
3	+	-	+	-	0.5
4	+	-	-	+	4.3
5	-	+	+	-	-1.9
6	-	+	-	+	1.7
7	-	-	+	+	3.9
8	-	-	-	-	0

where, $\dot{m}_w$ is mass flow rate of water to indoor heat exchanger

h_o is external heat transfer coefficient of water

% ice is percentage of ice in the TES tank

T_{ci} is temperature of return water from the indoor heat exchanger

2) Maximization of heat transfer in a coil in tank PCM cold storage system by A. Castell, M. Belusko, F. Bruno, and L.F. Cabeza[2]. In 2011, they performed experimental study of a PCM tank for cold storage applications. Two different configurations and different flow rates of the heat transfer fluid were studied. The effectiveness of the PCM storage system was defined as that of a heat exchanger. The results showed that the heat exchange effectiveness of the system did not vary with time, decreased with increasing flow rate and increased with increasing heat transfer area. The effectiveness was experimentally determined to only be a function of the ratio $\dot{m}/A$. This equation was found to be adequately be used to design a PCM storage system, and a case study is presented. It was shown that the tube in tank design together with a low temperature PCM is suitable as a thermal storage facility for cold storage.

Two different tank design was analysed and tested, one having higher packing factor (PF) and other having high heat transfer surface area (HTS). For each configuration different flow rates were tested for the same inlet temperature. Heat gain measurements were conducted of the PCM tank at low temperatures, and this value was found to be 12 W, or 1–3% of the heat transfer. The effect of the flow rate on the pressure drop was studied, Fig.2.10 shows the relation between pressure drop and flow rate for the present designs. Although the pressure drop increases when increasing the flow rate, this issue is not critical for the tested flow rates and the present application. For all the test performed the outlet temperature increases rapidly initially, achieves a constant temperature for a long period and then begins to increase at the end of the process towards the inlet temperature.

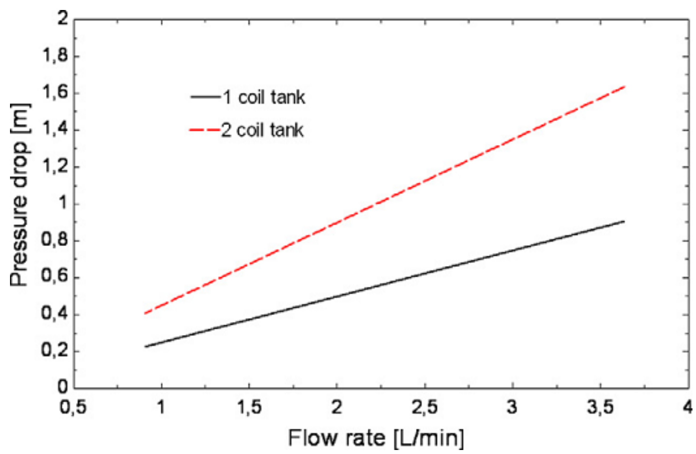

Figure 2.10: Pressure drop as function of flow rate for the PCM tanks[2]

Fig.2.11is a typical temperature–time curve graph of a melting process. The initial period represents the sensible heating of the PCM as a solid, the flat section represents the melting process, and the final stage represents the sensible heating of the PCM as a liquid. The thermal resistance to heat transfer is a function of the resistance due to convection in the HTF and the resistance due to conduction in the PCM. During the melting process the phase change interface moves away from the wall of the tube, and therefore the thermal resistance due to conduction can be expected to increase. This increasing resistance results in an increasing outlet temperature during the melting process. With increasing flow rate the temperature difference between the inlet and outlet temperature decreased. For the same flow rate the outlet temperature is lower than for the high PF design.

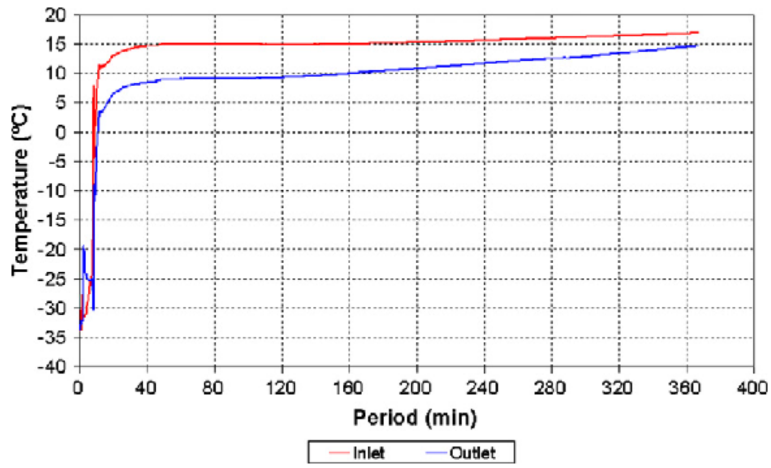

Figure 2.11: Inlet and outlet temperatures for the high PF design for a flow rate of 0.026 kg/s[2]

The effectiveness is described as a ratio of the actual heat discharged over the theoretical maximum heat that can be discharged. In using a PCM most of the energy is stored as latent heat around the phase change temperature, and so the sensible energy storage component is ignored. A PCM tank for cooling applications with coil in tank configuration has been studied. The storage tank was analysed as a heat exchanger between one fluid and a constant temperature heat sink/source at the phase change interface in the PCM. Two different designs were tested in order to characterize their behaviour and determine the main parameters for its design and optimization.The Fig.2.12 shows the variation of effectiveness of the PCM storage tank over the operating mass flow rate.

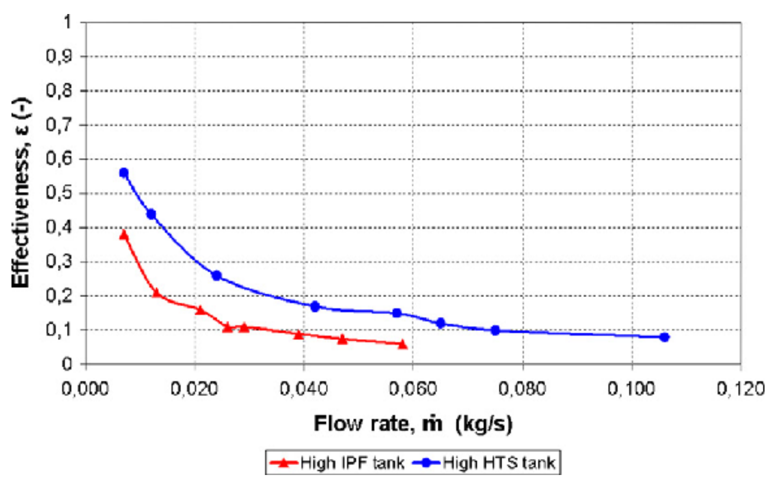

Figure 2.12: Effectiveness of the PCM storage tank over the operating mass flow rate[2]

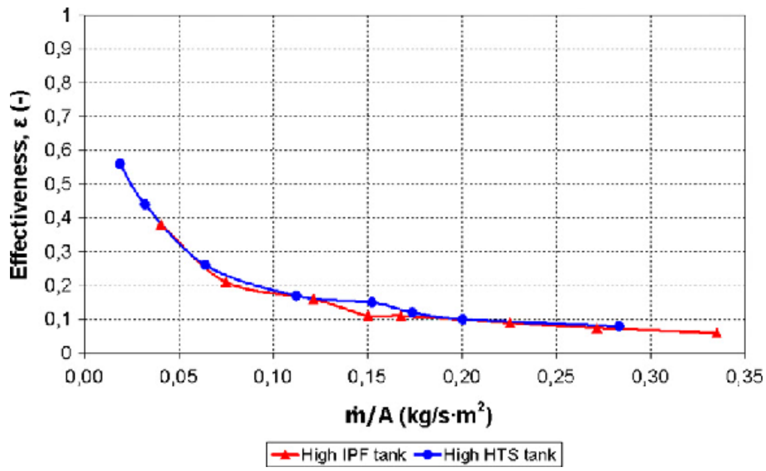

Figure 2.13: Effectiveness of the PCM storage tank over the ratio $\dot{m}/A$[2]

It was also demonstrated that coil in tank designs are effective at delivering a constant outlet temperature and effective heat transfer with large surface areas, with a high PF. The constant inlet and outlet temperatures during the phase change process demonstrate that the effectiveness is constant over the phase change period. This effectiveness describes an important design specification of the storage such as the temperature level achieved at the outlet and increases with lower flow rates. The representation of the effectiveness as a function of the ratio of the mass flow rate over the heat transfer surface results in a new parameter that considers both the working conditions and the design of the tank. A single equation was obtained for this new representation, which is useful for design. However the equation will underestimate the performance for larger tanks where the effect of natural convection is higher. Finally, the influence of the packing factor (PF) was analysed.

3) Experimental Evaluation of commercial heat exchangers for use as PCM thermal storage systems by M. Medrano, M.O. Yilmaz, M. Nogues, I. Martorell, Joan Roca, and Luisa F. Cabeza [3] in the year 2009. They investigated experimentally the heat transfer process during melting (charge) and solidification (discharge) of five small heat exchangers working as latent heat thermal storage system. Average thermal power values are evaluated for various operating conditions and compared among the heat exchangers studied.

When the comparison is done for average power per unit area and per average temperature gradient, results show that the double pipe heat exchanger with the PCM embedded in a graphite matrix (DPHX-PCM matrix) is the one with higher values, in the range of 700–800 $W/m^2.K$, which are one order of magnitude higher than the ones presented by the second best.

On the other hand, the compact heat exchanger is by large the one with the highest average thermal power (above 1 kW), as it has the highest ratio of heat transfer area to external volume. Melting and solidification of PCM RT35 were experimentally investigated in this work, using five different heat exchangers as heat storage systems and working at two different flow rates and two different water inlet temperatures. Results show that Reynolds numbers in the turbulent regime are desirable for faster phase change processes, reducing the phase change time in about half. The increase of the driving force for heat transfer, i.e. the difference between water inlet temperature and PCM phase change temperature, from 15 to 25 ^{o}C results in a considerable decrease of the phase change time (between 30% and 60% decrease) and consequently, an increase of the average phase change power

4) Energy and exergy analysis of an ice-on-coil thermal energy storage system by Mehmet Akif Ezan, Aytunc Erek, and Ibrahim Dincer[4]. In the year 2011, they carried out energy and exergy analysis for the charging period of an ice-on-coil thermal energy storage system. The model is developed using a thermal resistance network technique. First, the time-dependent variations of the predicted total stored energy, mass of ice, and outlet temperature of the heat transfer fluid from a storage tank are compared with the experimental data.

Afterward, performance of an ice-on-coil type latent heat thermal energy storage system is investigated for several working and design parameters. The results of a comparative study are presented in terms of the variations of the heat transfer rate, total stored energy, dimensionless energetic/exergetic effectiveness and energy/exergy efficiency. The results indicate that the thermal and flow parameters of the HTF(heat transfer fluid) become key factors for determining the performance of an ice-on-coil LHTES(latent heat thermal energy storage) system. With decreasing the inlet temperature and increasing the flow rate of the HTF(heat transfer fluid), the heat transfer rate, energy efficiency and energetic/exergetic effectiveness values increases. For the current parameters, the inlet temperature of the HTF(heat transfer fluid) is determinative parameter on the storage performance. By decreasing the inlet temperature of the HTF(heat transfer fluid) from - 5 °C to -15 °C, the total processing time decreases nearly by half. Both energetic and exergetic assessments are conducted to determine the optimum working and design parameters of an LHTES system. Even though the effectiveness and total stored energy values of the system increases with decreasing the inlet temperature of the HTF(heat transfer fluid) from T_{in} = -5 °C to T_{in} = -15 °C, the exergy efficiency decreases significantly with decreasing the inlet temperature of the HTF(heat transfer fluid). Although lower inlet temperatures and higher flow rates provide faster storage capability, decreasing the inlet temperature and increasing the flow rate decrease the exergy efficiency. The irreversibility rate increases with increasing the temperature difference between the melting temperature of the PCM and the inlet temperature of the HTF(heat transfer fluid). Hence, the working parameters should be determined by considering both exergy efficiency and the storing time.

5) Experimental investigation of tubes in a phase change thermal energy storage system by N.H.S. Tay, M. Belusko and F. Bruno [5] in 2012. An experimental investigation was performed for a thermal energy storage system with coils of tube inside a PCM filled cylindrical tank. From this study, it was found that the tube-in-tank design can deliver a high energy storage density with compactness factors above 90%. Furthermore, with sufficient heat transfer area the useful energy that can be stored can be more than 70% as identified by the average heat exchange effectiveness.

This average effectiveness was found to be a function of the mass flux, representing the average NTU of the storage system. An experimentally derived equation was formulated based on this representation, which can be directly used for design purposes. However, this equation is only validated for an average tube spacing of up to 70 mm which is the maximum average tube spacing tested, and consideration needs to be made for the difference between the average and local effectiveness near the end of the phase change process. When the average tube spacing is larger than 70 mm, the equation will no longer be valid. This is because when the tube spacing increased; the thermal resistance will also increase, decreasing the average effectiveness. This equation is also useful for larger tank sizes that are having the same average tube spacing as the system tested. The tube spacing determines the thermal resistance and will have direct impact on the NTU. Furthermore for larger tanks, natural convection is expected to increase, which will increase the effectiveness that can be expected. The equation was independent of the freezing/melting processes and between two PCMs suggesting that the difference in resistance in the PCM in all the processes was small, relative to the change in resistances caused by increasing the flow rate or surface area.

6) Review on sustainable thermal energy storage technologies, part II: Cool Thermal Storage by S. M. Hasnain [6]at the Energy research institute, Saudi Arabia in 1998. This paper describes the inherent pros and cons of the two common (i.e. chilled water and ice storage) commercially available thermal energy storage (TES) technologies for off-peak air conditioning applications. Case studies on cool thermal storage have demonstrated not only savings in energy and other operation and maintenance costs but also significant savings in initial capital costs. This paper also examines the use of cool thermal storage equipment for gas turbine inlet air cooling, which can positively enhance its efficiency. Systems with cool storage shift all or part of the electricity requirement from peak to off peak hours and also take advantage of reduced demand charges and/or off peak rates. Cool storage technology has been used in commercial buildings and for turbine inlet air cooling. With the application of the technology, electricity is used to charge a reservoir (or a storage tank) when it is cheapest. The reservoir is then used to cool the building during the time when electricity is ex-pensive.

7) Performance assessment of some ice TES system by David MacPhee and Ibrahim Dincer [7] at university of Ontario, Canada in 2009. In this study, a comparison of four main types of ice storage techniques for space cooling purposes is conducted. The systems studied include ice slurry systems, ice-on-coil systems (both internal and external melt), and encapsulated ice systems.

A detailed analysis, coupled with a case study based on values found in the literature follows this review. The four ice storage and retrieval techniques are compared on the basis of energy and exergy efficiencies according to the charging, storage and discharging process. The results indicate that the energy efficiencies are misleadingly high, and only when the exergy analysis is undertaken is a more realistic view of the system performance achievable.

8) Thermal modeling of a packed bed thermal energy storage system during charging by David MacPhee and Ibrahim Dincer[8] at university of Ontario, Canada. In this paper a comprehensive study of heat transfer and thermodynamic aspects of the charging process in an ITES system is conducted. Some new performance models for energy and exergy efficiencies for system performance analysis, design and improvement is developed.

Main remarks of the study are, that the temperature profile can be accurately depicted when neglecting stream-wise conduction, assuming a one-dimensional, radial conduction model, The temperature profile was found not to appreciably change when assuming a linear or quadratic profile, so that it was deemed reasonable to assume a linear temperature profile with respect to the radius. This greatly simplified the resulting energy and exergy analyses. Due to overwhelmingly long charging times, the case where the flow rate is 0.00033 m3/s will be discarded, since it is unreasonable to assume it can be used in a real ITES system. For the other two flow rates, the inlet temperature was varied from -33 °C to -1 °C and the energy and exergy analyses were performed. The following conclusions can be made from the results: The energy efficiencies varied from 90% to over 99%, though in cases where the charging time is reasonable the energy efficiency was over 99%, The thermal exergy also varied, but at a much lesser value; from around 40% to 93%, respectively. The flow exergy was found to vary from 48% to 88%, respectively. Maximum flow exergy efficiencies were found at inlet temperatures of 269.7 K (84.3%) and 270.7 K (87.9%) for flows of 0.00164 m3/s and 0.0033 m3/s, respectively.

9) Mathematical modelling of thermal storage systems for the food industry by A. Lopez and G. Lacarra[9] in the year 1999. In this paper dynamic mathematical models of two thermal storage systems used in the food industry to produce chilled water are presented; an ice-bank system and a holding tank system. The variability of the refrigeration demand with time was taken into account in the model. A zoned approach using mass and energy balances was applied. Heat transfer phenomena in the evaporator were modelled using empirical correlations.The experimental validation of the mathematical models on an ice-bank system at pilot plant scale, and a centralized refrigeration system with a holding tank in a winery, showed accurate prediction. Simple models are adequate to predict the dynamic behaviour of these refrigeration systems under variable heat loads.

In this work two dynamic mathematical models of thermal storage systems commonly found in the food industry to produce chilled water were developed. The experimental validation of the mathematical model for a pilot scale ice-bank system, and a centralized refrigeration system with a holding tank in a winery, showed the accurate prediction. Comparison of the simulation and experimental results for a 6 h period of winery operation is shown in Figure 2.14.

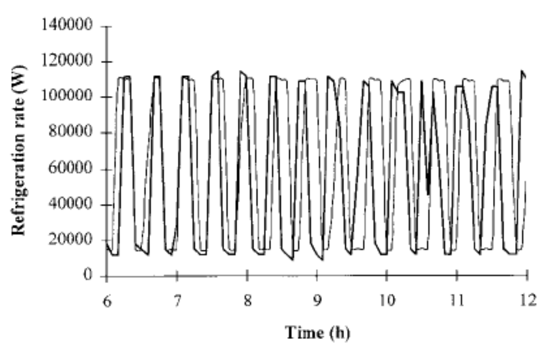

Figure 2.14: Comparison of the simulation and experimental results for a 6 h period of winery operation[10]

The models are potentially useful not only to optimize the design of these refrigeration systems (as the correct size of the refrigeration system, the ice-bank and the holding tank), but also to establish the best control strategies of the equipment performance to achieve the correct chilled-water temperature levels to be used in the food plant.

10) Mathematical modelling of the thermal performance of a phase-change material (PCM) Store: Cooling cycle by A. Kurklu, A. Wheldon and P. Hadley [10] in the year 1995. In this paper mathematical model for the prediction of the thermal performances of a PCM store containing 1 m long and 38 mm diameter polypropylene tube has been developed. Air was utilised in the store as the heat transfer fluid. The model was based on an energy balance or the 'conservation of energy principle'. The results indicate that the agreement between the predicted and observed temperature or heat transfer data is generally good. The amount of energy used in increasing the temperature of the PCM at any time during the phase-change process is predicted to be about 3.5% of the total energy stored.

The model was validated by using experimental data. The percentage of the energy released in decreasing the temperature of the PCM during freezing was found to be 3.8%. The thermal properties of the PCM tube in this study were also found to have an important effect on the heat transfer, since they represented a higher thermal resistance than the PCM, due to its lower thermal conductivity. Thus consideration of the tube and container thermal properties needs to be made separately from the PCM, whether the container be plastic or metal.

The model suggested that it was possible to eliminate some complex assumptions, such as the change of thermal properties of the PCM with temperature and the existence of convection in the liquid phase, while maintaining acceptable accuracy. The agreement between the experimental and predicted data by the model for both the PCM temperature and the energy released was good. The accuracy of the model could be increased by considering the changes in the thermal properties of the air with temperature.

11) Review of mathematical modeling on latent heat thermal energy storage systems using phase-change material by Prashant Verma, Varun and S.K. Singal [11]in the year 2008. They used mathematical model of a latent heat thermal enegy storage system for the optimum material selection and to assist in the optimal designing of the systems. Two types of models are mainly discussed, on the basis of first law and second law of thermodynamics. The important characteristics of different models and their assumptions used are presented and discussed, the experimental validation of some models are also presented.

Latent heat thermal energy storage systems are used for the storage of excess energy available by the solar or any other sources. There are many methods for the energy storage but the latent heat thermal energy storage is good among all the storage systems. There are many researchers who have worked on first law of thermodynamics and most of them verified their results by experimental investigations, which prove their acceptability. In contrast to first law a very little work has been done on the second law of thermodynamics. The second law (exergy) analysis develops a good understanding of the thermodynamic behavior or efficiency of thermal energy storage system contrary to first law of thermodynamics, as it does not take into account the time duration by which heat is supplied. There is requirement of some experimental work by which the acceptability of second law analysis can be substantiated.

2.2 Summary of Literature Review

A summary of literature reviewed related to the ice based latent heat storage system is provided in Table 2.3and 2.4 below

Table 2.3: Summary of literature reviewed

Sr. No.	Reference	Authors	Remarks
1.	Modelling of an ice-on-coil thermal energy storage system	Alex H. W. Lee and Jerold W Jones(1995)	Analytical models were developed and compared with experimental values, Charging model volume of ice in TES tank and cooling capacity and discharging model- tank bulk temperature Regression analysis is performed
2.	Maximization of heat transfer in a coil in tank PCM cold storage system	A. Castell, M. Belusko, F. Bruno, and L.F. Cabeza(2011)	Two designs of storage tank is considered to characterize its behaviour, Coil in tank designs are effective at delivering a constant outlet temperature, e-NTU technique is used.
3.	Experimental Evaluation of commercial heat exchangers for use as PCM thermal storage systems	M. Medrano, M.O. Yilmaz, M. Nogues, I. Martorell, Joan Roca, and Luisa F. Cabeza(2009)	Heat transfer process during melting(charge) and solidication(discharge) is analyzed of ve small heat exchangers working. Average thermal values are evaluated various operating conditions & compared.Comparison for average power per unit area and per average temperature gradient is done.
4.	Energy and exergy analysis of an ice-on-coil thermal energy storage system	Mehmet Akif Ezan, Aytunc Erek, and Ibrahim Dincer(2011)	Thermal and flow parameters are key factors for this system, low inlet temperatures and higher ow rates provide faster storage capability but decreases exergy efficiency
5.	Experimental investigation of tubes in a phase change thermal energy storage system	N.H.S. Tay, M. Belusko and F. Bruno(2012)	PCM used is salt hydrate and water Average heat exchanger eectiveness of tank is determined and characteristic design curve is developed
6.	Review on sustainable thermal energy storage technologies, part II: Cool Thermal Stor- age	S. M. Has- nain(1998)	Pros and cons of two systems i.e. chiller and ice storage are reviewed, case study shows saving in energy as well as saving in cost in using storage system

Table 2.4: Summary of literature reviewed

7.	Performance assessment of some ice TES system	David MacPhee and Ibrahim Dincer(2009)	Performance assessment of ice slurry system, ice-on-coil system and encapsulated ice system. Ice making techniques are compared on the basis of energy and exergy performance Case study analysis is performed for full storage and partial storage
8.	Thermal modeling of a packed bed thermal energy storage system during charging	David MacPhee and Ibrahim Dincer(2009)	Temperature profile can be accurately depicted when neglecting stream-wise conduction, assuming a 1-D radial conduction model.The energy efficiencies varied from 90% to over 99% and exergy varied from around 40% to 93%
9.	Mathematical modelling of thermal storage systems for the food industry	A. Lopez & G. Lacarra(1999)	The experimental validation of the mathematical model for a pilot scale ice-bank system, and a centralized refrigeration system showed the accurate prediction. Models are useful to optimize and establish the best control strategies.
10.	Mathematical modelling of the thermal performance of a phase-change material (PCM) Store: Cooling cycle	A. Kurklu, A. Wheldon and P. Hadley(1996)	The model was based on an energy balance or the "conservation of energy principle". The results indicate that the agreement between the predicted and observed temperature or heat transfer data is generally good.
11.	Review of mathematical modeling on latent heat thermal energy storage systems using phase-change material	Prashant Verma, Varun and S.K. Singal(2008)	The second law (exergy) analysis develops a good understanding of the thermodynamic behavior or efficiency of thermal energy storage. There is requirement of some experimental work by which the acceptability of second law analysis can be substantiated.

2.3 Dairy Industry Processes and Application of Ice Based Latent Heat Storage System in Dairy Industries

2.3.1 Milk production and processing

The milk plant performs certain functions on the raw milk received from the dairy farm which includes blending, processing, packaging and distribution to make it safe for the human consumption. The milk received from various dairy farms is cooled, mixed, and then processed by separation and blending to distribute the butter fat uniformly as required by the various items. It is followed by the pasteurization which is a part of processing and its purpose is to destroy the undesirable bacteria and prolonging the keeping quality of the milk. Pasteurization kills all the pathogenic types of bacteria and nearly all other types of organisms, but there are some heat resistant bacteria that will keep on multiplying rapidly unless the milk is cooled quickly below 10°C after pasteurization. After milking is done from the cow it should be cooled to 4.4°C or below for prolonged keeping. If proper refrigeration and processing is done the milk can retain its acceptable flavour for 15 days or longer after milking.

2.3.1.1 processes [30][30]

Different processes done during milk production are:

1. ***Handling milk at the dairy farm***

   After milking is done, most of the dairy farms use cans or bulk tanks to receive, cool, and hold the milk. Tank capacity ranges from 0.5 to 10 m^3, with a few larger tanks. Different farm uses different methods according to their need to cool the milk. The main aim behind cooling is to keep the raw milk below 10°C, so that the bacterial growth remains under control. If cans are used to store the milk, cooling is done in the cans only, by either partial immersion in a refrigerated water bath or spraying with re-circulated chilled water. The cans remain in chilled water coolers until transported to the milk plant.If farm bulk milk cooler tank is used, the milk is directly stored into it after milking. The tank is well insulated and arranged with refrigerating surfaces built into the stainless steel lining of the tank. Tank should have sufficient refrigerated surface at the first milking to cool to 10°C or less within 4 h of the start of the first milking and to 7°C or less within 2 h after completion of milking.Some large dairy farms may use a plate or tubular heat exchanger for rapid cooling. Cooled milk may be stored in an insulated tank.

2. ***Receiving and storing milk***

   Milk is generally received more rapidly than it is processed, so ample storage capacity is needed. A holdover supply of raw milk at the plant may be needed for start-up before arrival of the first tankers in the morning. Storage may also be required for non processing days and emergencies. Storage tanks vary in size from 2 to 100 m^3. The tanks have a stainless steel lining and are well insulated.

Agitation is essential to maintain uniform milk fat distribution. Milk held in large tanks, such as the silo type, is continuously agitated with a slow-speed propeller driven by a gearhead electric motor. The tank may or may not have refrigeration, depending on the temperature of the milk flowing into it and the maximum holding time.If refrigeration is provided for milk in a storage tank, the temperature of milk is to be reduced from nearly 10°C to 4.0°C.

3. ***Standardizing***

Before pasteurizing, milk and cream are standardised and blended to control the milk fat content within the required limits. The standards of different classes and designations of milk shall be as given in the Table2.5 below. Milk shall conform to both the parameters for milk fat and milk solids not fat, independently.

Table 2.5: Requirements for milk fat and non-fat solids in milks and creams

Class of Milk	Milk Fat	Minimum percent milk non fat solids
Raw Milk	6.0 to 3.5	9.0
Mixed Milk	4.5	8.5
Standardized Milk	4.5	8.5
Recombined Milk	3.0	8.5
Toned Milk	3.0	8.5
Double Toned milk	1.5	9.0
Skimmed Milk	Not more than 0.5 percent	8.7
Full Cream Milk	6.0	9.0

The desired fat standards are attained by separating the milk in the separator which divides the milk into part of skim milk and cream. The required amount of skim milk or cream is returned to milk to control final desired fat content. To increase non fat solids, condensed milk or non fat dry milk may be added.

4. ***Pasteurization***

During this process the milk is heated to a specific temperature for a predefined length of time and then immediately cooled after it is removed from the heat. This process slows spoilage caused by microbial growth in the milk. Pasteurization relies on the principle that most harmful bacterial can be killed by heat. The most effective way to kill bacteria is boiling, but this compromises the flavor of the liquid. Pasteurization is not intended to kill all micro-organisms in the milk. Instead, it aims to reduce the number of viable pathogens so they are unlikely to cause disease. Pasteurization kills all the pathogenic types of bacteria and nearly all other types of organisms, but there are some heat resistant bacteria that will keep on multiplying rapidly unless the milk is cooled quickly below 10°C after pasteurization.

There are two primary methods of pasteurization: the milk can be heated to 63°C and held there for not more than 30 minutes, or the milk can be pasteurized at high temperature i.e. 72°C for a minimum of 15 seconds. Pasteurization can be done using a continuous method, where the milk flows through a pasteurization system, or by using a batch method, where one batch of the milk is pasteurized at a time. Continuous pasteurization is popular for large producers, because it does not slow the supply line as much as batch pasteurization does.After heating cooling is continued in a heat exchanger (e.g., a plate or tubular unit) to 4.0 °C or lower and then packaged.

5. ***Equipment cleaning***

Several automatic clean-in-place (CIP) systems are used in milk processing plants. These may involve holding and reusing the detergent solution or the preparation of a fresh solution (single-use) each day. Programming automatic control of each cleaning and sanitizing step also varies. Tanks, vats, and other large equipment can be cleaned by using spray balls and similar devices that ensure complete coverage of soiled surfaces. Tubing, HTST units, and equipment with relatively low volume may be cleaned by the full-flood system.

Solutions should have a velocity of not less than 1.5 m/s and must be in contact with all soiled surfaces. Surfaces used for heating milk products, such as in batch orHTST pasteurization, are more difficult to clean than other equipment surfaces. Other surfaces difficult to clean are those in contact with products that are high in fat, contain added solids and/or sweeteners, or are highly viscous.

The usual cleaning steps for this equipment are a warm-water rinse, hot acid-solution wash, rinse, hot-alkali-solution wash, and rinse. Time, temperature, concentration, and velocity may need to be adjusted for effective cleaning. Surfaces in contact with product should be sanitized with chemical solution, hot water, or steam. During clean-in-place (CIP), the cooling section is isolated from the supply of chilled water or propylene glycol to minimize parasitic load on the refrigeration system.

6. ***Packaging milk products***

Cold product from the pasteurizer cooling section flows to the packaging machine and/or a surge tank 2 to 20 m^3 or larger. These tanks are stainless steel, well insulated, and have agitation and usually refrigeration. Milk and related products are packaged for distribution in paperboard, plastic, or glass containers in various sizes. Fillers vary in design. Gravity flow is used, but positive piston displacement is used on paper machines. Filling speeds range from roughly 16 to 250 units/min, but vary with container size.

Paperboard cartons are usually formed on the line ahead of filling, but may be pre-formed before delivery to the plant. Semirigid plastic containers may be blow-molded on the line ahead of the filler or preformed. Plastic pouches (called bags) arrive at the plant ready for filling and sealing. Filling dispenser cans and bags is a semi-manual operation.

7. ***Milk storage and distribution***

   Cases containing packaged products are conveyed into a cold storage room or directly to delivery trucks for wholesale or retail distribution. The temperature of the storage area should be between 0.6 and 4.4°C, and for improved keeping quality, the product temperature in the container on arrival in storage should be 4.0°C.

   The refrigeration load for cold-storage areas includes transmission through the building envelope, product and packaging materials temperature reduction, internally generated loads (e.g., lights, equipment motors, personnel), infiltration load from air exchange with other spaces and the environment, and refrigeration equipment related load (e.g., fan motors, defrost).The floor space required for cold storage depends on product volume, height of stacked cases, packaging type, handling, and number of processing days per week. A 5 day processing week requires a capacity for holding product supply for 2 days. A very general estimate is that 490 kg of milk product in paperboard cartons can be stored per square metre of area. Milk product may be transferred by conveyor from storage room to dock for loading onto delivery trucks. In-floor drag-chain conveyors are commonly used, especially for retail trucks. Refrigeration capacity must be sufficient to maintain products at 7.2°C or less.

The Process flow diagram is given below in Figure 2.15,

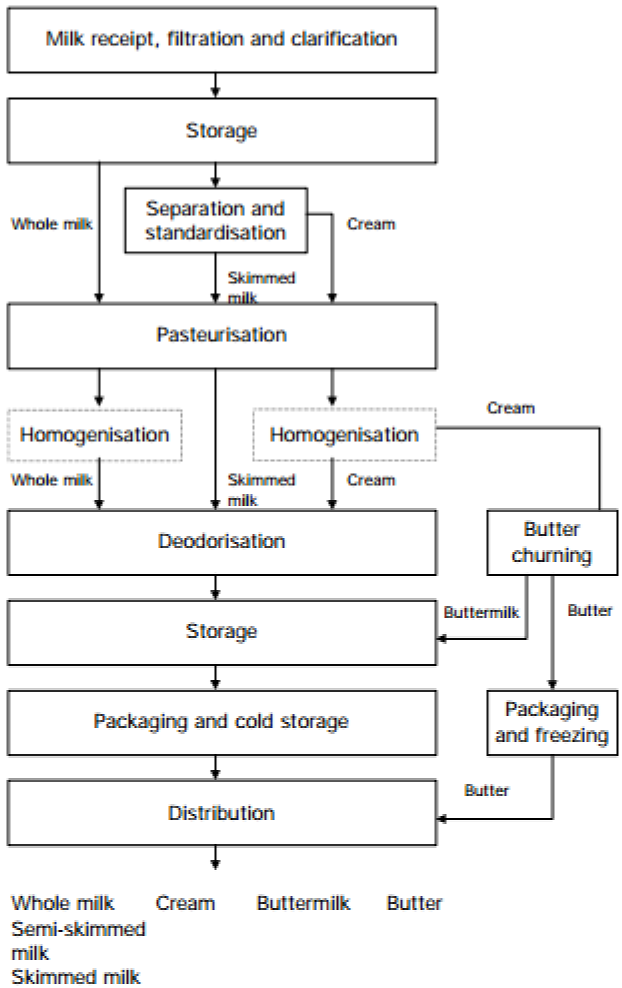

Figure 2.15: Flow diagram for processes occurring at a typical milk plant

2.3.1.2 heat exchange processes during milk production and processing

1. ***Cooling of the milk after milking***

   The cooling is done to keep the raw milk below 10°C, so that the bacterial growth remains under control. This can be achieved by different methods according to the requirement. Different methods are:

   - If cans are used to store the milk, cooling is done in the cans only, by either partial immersion in a refrigerated water bath or spraying with re-circulated chilled water. The cans remain in chilled water coolers until transported to the milk plant.
   - If farm bulk milk cooler tank is used, the milk is directly stored into it after milking. The tank is well insulated and arranged with refrigerating surfaces built into the stainless steel lining of the tank. Tank should have sufficient refrigerated

surface at the first milking to cool to 10°C or less within 4 hour of the start of the first milking and to 7°C or less within 2 hour after completion of milking.

- Some large dairy farms may use a plate or tubular heat exchanger for rapid cooling. Cooled milk may be stored in an insulated silo tank (a vertical cylinder 3 m or more in height).

2. ***During standardizing***

- For hot milk seperator their will be requirement for cooling of milk after the seperation process, if the milk is to be hold for more than 20 minute. It is beacause it separates the milk at a temperature between 40 °C.
- Cold milk separators don't require the heating or cooling process. In this method the milk is separated at the temperature 4.0 °C at a rate of 100 to 1000 litre per hour input for machine. Its capacity and efficiency increases as milk temperature increases.

3. ***Pasteurization process***

Pasteurization can be done through two metods, i.e. Batch Pasteurization and Continuous Pasteurization

- ***Batch pasteurization***: The batch pasteurization is used for relatively small quantities of liquid milk products. The minimum feasible processing rate for continuous systems is about 500 liter per hour. Less than this processing rate, batch pasteurization will be used .The product is heated in a stainless steel-lined vat to not less than 63 °C and held at that temperature or above for not less than 30 minutes.

(a) ***Heating processes***

Pasteurizing vats are heated with hot water or steam vapour in contact with the outer surface of the lining.

• For hot water heating, hot water is sprayed around the top of the lining and flows to the bottom where it is collected in a sump, reheated by steam injection and pumped to the distributor again.

• For steam vapour heating, steam at about atmospheric pressure is introduced by an automatic control valve into the annular space between the lining and the casing of the vat.

(b) ***Cooling processes***

After 30 min the milk will be cooled to 4.0 °C or below before it is packed. The milk may be cooled either in the vat itself or through the plate heat exchanger or surface cooler. Cooling will be done in two stages, firstly, using cooling tower water to a temerature around 33°C and then refrigerated chilled water or propylene glycol solution is used to cool to a temperature of 4.0 °C.

- **High Temperature-Short Time Pasteurization**: High-temperature short-time (HTST) pasteurization is a continuous process in which milk is heated to at least 72 °C and held at this temperature for at least 15 sec. The complete pasteurizing system usually consists of a series of Plate heat exchanger (PHE), a milk balance tank, one or more milk pumps, a holding tube, flow diversion valve, automatic controls, and sources of hot water or steam and chilled water or propylene glycol for heating and cooling the milk, respectively. Homogenizers are used in many HTST systems as seperators.

  The milk and the cooling medium flows in counter flow. Terminal plates are inserted to divide the press into three sections (heating, regenerating, and cooling) and arranged with ports for inlet and outlet of milk, hot water, or steam for heating, and chilled water or propylene glycol for cooling.

  To provide a sufficient heat-exchange surface for the temperature change desired in a section, milk flow is arranged for several passes through each section. The capacity range of a complete HTST pasteurizer is 13 g/s to about 13 kg/s. Shell-and-tube and triple-tube HTST units may also be used, but the plate type is by far the most prevalent. HTST pasteurization is shown in Figure2.16.

(a) ***Heating process***

The milk is clarified and homogenized or directly pumped to the heating section by a timing pump. From the heating section, the product continues through a holding tube to the flow diversion valve. If the product is at or above the preset temperature, it passes back through the opposite sides of the plates in the regeneration section and then through the final cooling section. The flow diversion valve is set at 72°C or above; if the product is below this minimum temperature, it is diverted back into the balance tank for re-pasteurization.

The heating section usually has ample surface so that the temperature of hot water entering the section is no more than 1 to 3 K higher than the pasteurizing, or outlet, temperature of the product. On larger units, steam may be used for the heater section instead of hot water.If the temperature of the milk obtained in the homogenizer as 59 °C, which is the temperature of the milk coming out of the regeneration section and the pasteurizing temperature to be attain in the heating section is 72 °C.

The temperature of the hot water leaving the heating section can be calculated using heat balance.

$$Q = m_h \times C_{p_h} \times (T_{h_i} - T_{h_o}) = m_c \times C_{p_c} \times (T_{c_o} - T_{c_i})$$

The flow rate of hot water is 4 times the flow rate of milk.

$$4 \times m \times 4.18 \times (75 - T_{h_o}) = m \times 4.9 \times (72 - 59)$$

$$T_{ho} = 71.2\,^{\circ}C$$

Where,

m_h= flow rate of the hot water

m_c = flow rate of the milk
C_{p_h} = Specific heat of hot water in kJ/kg.K
C_{p_c} = Specific heat of hot milk in kJ/kg.K at 60 °C

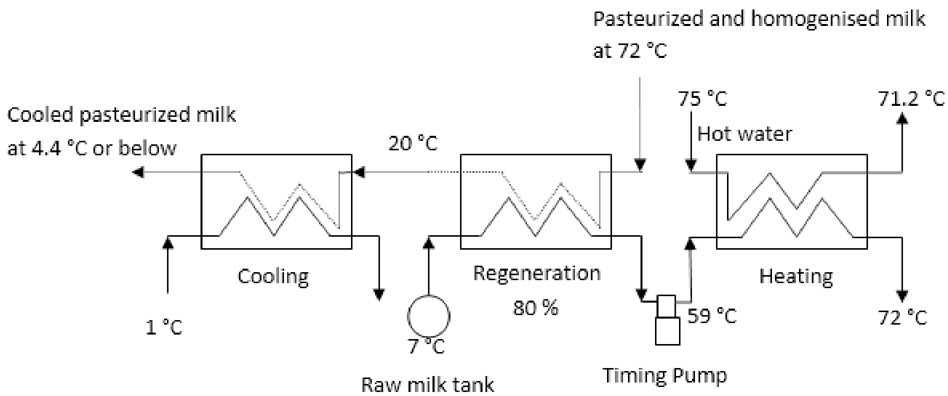

Figure 2.16: HTST Pasteurization

(b) ***Regerneration***

Heat exchange in the regeneration section causes cold raw milk to be heated by hot pasteurized milk going downstream from the heater section and flow diversion valve.Most HTST heat exchangers achieve 80 to 90% regeneration. The cost of additional equipment to obtain more than 90% regeneration should be compared with savings in the increased regeneration to determine feasibility.

If the regeneration temperature, raw milk temperature and pasteurization temperature is known. The percentage of regeneration may be calculated as follows for equal mass flow rates on either side of the regenerator:

$$\frac{59\,°C\,(Regeneration) - 7\,°C\,(Raw\,Milk)}{72\,°C\,(Pasteurization) - 7\,°C\,(Raw\,Milk)} = \frac{52}{65} = 80\%$$

The temperature of a product going into the cooling section can be calculated if the percent regeneration is known and the raw product and pasteurizing temperatures are determined. If they are 80%, 7°C, and 72°C, respectively,

$$(72 - 7) \times 0.80 = 52\,°C$$

$$(72 - 52) = 20\,°C$$

20 °C is the temperature of the milk going from the regenerator in to the cooling section.

(c) ***Cooling process***

The cooling process uses either chilled water or propylene glycol solution to cool the milk in the plate heat exchanger. The product should be cooled to at least

4.4°C, preferably lower, to compensate for the heat gain while in the sanitary pipelines and during the packaging process (including filling, sealing, casing, and transfer into cold storage).

The advantage of using propylene glycol solution to cool the milk is that it is possible to attain much lower temperature than the temperature which can be attained using chilled water. But, this requires an additional section in the plate heat exchanger, a glycol chiller, a pump for circulating the glycol solution, and a product-temperature-actuated control to regulate the flow of glycol solution and prevent product freezing. The temperatures at inlet and outlet section are:

i. The temperature of the milk entering from the regeneration section into the cooling section is 20 °C.
ii. Temperature of milk leaving the cooling section is 4.4°C or below.
iii. Temperature of the chilled water entering the section is 1 °C. If glycol solution is used, temperature of -1 °C or – 2°C can be achieved.

Milk is usually cooled with propylene glycol to approximately 1°C, and then packaged. The lower temperature allows the milk to absorb heat from the containers and still maintain a low enough temperature for excellent shelf life. Milk should not be cooled to less than 0.8°C because of the tendency toward increased foaming in this range. The flow rate of chilled water is 4 times the flow rate of milk and the specific heat of milk at 20 °C is 3.927 kJ/kg.K and chilled water is 4.187 kJ/kg.K.If Chilled water is used at 1°C, the temperature of the chilled water leaving the cooling section after gaining temperature from the milk can be calculated using heat balance.

$$m \times 3.927 \times (20 - 1) = 4 \times m \times 4.187 \times (T_{c_o} - 1)$$

$$T_{c_o} = 5.45\,^{\circ}C$$

If glycol solution is used at -2°C, flow rate of chilled water is 4 times the flow rate of milk and the specific heat of milk at 20 °C is 3.927 kJ/kg.K and 10% glycol water solution 4.10 is kJ/kg.K, the temperature of the glycol solution leaving the cooling section will be,

$$m \times 3.927 \times (20 - 1) = 4 \times m \times 4.10 \times (T_{c_o} - (-2))$$

$$T_{c_o} = 2.54\,^{\circ}C$$

The number of plates in the pasteurizing unit is determined by the volume of product needed per unit of time, desired percentage of regeneration, and temperature differentials between the product and heating and cooling media. The cooling section is usually sized so that the temperature of pasteurized product leaving the section is about 2 to 3 K higher than the entering temperature of chilled water or propylene glycol. All the heat exchange process used for cooling in milk processing is shown in Table 2.6.

Table 2.6: Heat exchange processes used in milk processing

	Processes	Temperature	Heat Exchanger Used
1)	**Handling milk at dairy farm**: After milking is done, milk is cooled rapidly and stored in silo tank	35 °C →Below 10 °C	1)Plate Heat Exchanger, 2)Tube in tube 3)Bulk milk cooler
2)	**Receiving and Storing Milk in Dairy Plant:** Milk from dairy farm is stored in the storage tank and refrigeration may be required in this process.	10-8°C→4°C or less	1) A refrigerated jacket around the interior lining of the tank 2) A distributing pipe at the top for chilled liquid to flow down the lining and drain from the bottom. 3) Some plants pass milk through a Plate Heat Exchanger to keep all milk directed into the storage tanks at 4.4°C or less.
3)	**Standardizing:** If hot milk separator is used milk will separate at 40 °C and if it is to be held for more than 20 min. it must be cooled.	40°C →4°C or less	Refrigerated water Jacketed vat. or Plate Heat Exchanger
4)	**Pasteurization**		
	Batch Pasteurization: Milk is heated to 62.8 °C and held for 30 min.	4°C → 62.8 °C	Stainless steel lined jacketed vat, around which hot water or steam is passed
	After 30 min. milk is cooled to a temperature of 4.0 °C or below	62.8°C→33°C→4°C	1) Plate heat exchanger with 1st stage cooling from cooling tower water and 2nd stage cooling from propylene glycol solution. 2) Stainless steel lined jacketed vat
	HTST Pasteurization: Milk is heated to at least 72 °C and held at this temperature for at least 16 sec. If 80 % regeneration is used.	4°C→ 59°C→ 72°C	Plate Heat Exchanger, shell-and-tube Or triple-tube HTST
	Milk will be cooled will be from 72 °C to 4.0 °C or below.	72°C→20°C→4°C	Plate Heat Exchanger, shell-and-tube Or triple-tube HTST

2.3.2 Butter manufacturing

Butter is a dairy product made by churning fresh or fermented cream or milk. It is generally used as a spread and a condiment, as well as in cooking, such as baking, sauce making, and pan frying. Butter consists of butterfat, milk proteins and water. Butter is a water-in-oil emulsion resulting from an inversion of the cream, an oil-in-water emulsion; the milk proteins are the emulsifiers. Butter remains a solid when refrigerated, but softens to a spreadable consistency at room temperature, and melts to a thin liquid consistency at 32–35 °C. The density of butter is 911 g/L.

2.3.2.1 *processes* [29][30]

Different processes done during Butter production are:

1. ***Seperation***

   The separation process is performed on the milk to separate the cream from the milk. The separation temperature is nearly 40 °C for hot milk separator. Cream from cold milk separation does not need to be re-cooled except for extended storage. Cream is received, weighed, sampled, and, in some plants, graded according to flavour and acidity. It is pumped to a refrigerated storage vat and cooled to 7°C if held for a short period or overnight.

2. ***Pasteurization***

   During this process the cream is heated to a specific temperature for a predefined length of time and then cooled after it is removed from the heat.There are two primary methods of pasteurization: the cream can be heated to 68.8°C to 79°C and held there for not more than 30 minutes, or the milk can be pasteurized at high temperature i.e. 85°C to 95°C for a minimum of 15 seconds.

3. ***Churning***

   To maintain the yellow colour of butter from cream in spring and early summer, yellow colouring can be added to the cream to match the colour obtained naturally during other periods of the year. After cooling, pasteurized cream should be held a minimum of 2 hours and preferably overnight in a covered stainless steel vat. Most butter is churned by continuous churns, but some batch units remain in use, especially in smaller butter factories. Batch churns are usually made of stainless steel. The inside surface of metal churns is sandblasted during fabrication to reduce or prevent butter from sticking to the surface. A cream pump is unnecessary with a metal churn because the cream from the vat can be drawn into the churn when a vacuum is maintained in it. The butter worked under partial vacuum results closer texture, a brighter appearance and lower air content. Churns have two or more speeds, one for churning and other for working.

   The fat content of the cream to be churned must be between 35 to 40 % for better efficiency. The temperature of the cream should be such that an exhaustive churning can take place in 40 to 45 minutes. Satisfactory amount of cream in the churn is about

40 % of the volume of churn barrel. The buttermilk generated should not have milk fat more than 1%, which is drained. Churning temperature during spring and summer generally ranges from 7.2 °C to 11.1°C and during fall and winter from 11.1 °C to 15.5 °C.

4. ***Washing***

   The butter may or may not be washed. The purpose of washing is to remove buttermilk and temper the butter granules if they are too soft for adequate working. Wash water temperature is adjusted to 0 to 6 K below churning temperature. The preferred procedure is to spray wash water over granules until it appears clear from the churn drain vent. The vent is then closed, and water is added to the churn until the volume of butter and water is approximately equal to the former amount of the cream. The churn revolves slowly 12 to 15 times and drained or held for an additional 5 to 15 min for tempering so granules will work into a mass of butter without becoming greasy.

   The butter is worked at a slow speed until free moisture is no longer extruded. Free water is drained, and the butter is analyzed for moisture content. The amount of water needed to obtain the desired content (usually 16.0 to 18.0%) is calculated and added. Salt may be added to the butter. The salt content is standardized between 1.0 and 2.5% according to customer demand. Working continues until the granules are completely compacted and the salt and moisture droplets are uniformly incorporated. Moisture droplets should become invisible to normal vision with adequate working. Butter must be thoroughly worked so that there will be no brine droplets on cut surface of butter after it has been kept in a refrigerator for some time. The butter should have close texture and waxy body; it must not be greasy, sticky or crumbly. The water used for washing must be cooled to a temperature of 0 K to 6 K below the churning temperature.

5. ***Removing the butter from the churn***

   Butter is removed from the by mechanical means. It is dumped out of the churn onto aluminium or stainless steel trays rolled under the churned. The butter is stored in these trays in the refrigerator until the time of printing. A hydraulic lifter is used for lifting the tray with butter to the level of the hopper of the butter printer. Another method used for removing the butter from the churn, consist of pumping the butter from the outlet gate of stainless steel churn by means of a specially designed pump through stainless steel pipes direct to either the bulk butter containers or to butter printer.

6. ***Printing butter***

   Printing is the process of forming (or cutting) butter into retail sizes. Each print is then wrapped with parchment or parchment coated foil. The wrapped prints may be inserted in paperboard cartons or overwrapped in cellophane, glassine, and so forth, and heat-sealed. For institutional uses, butter may be extruded into slabs. These are cut into patties, embossed, and each slab of patties wrapped in parchment paper. Most common numbers of patties are 105 to 158 per kg.

7. ***Storage of butter***

Butter keeps better if stored in bulk. If the butter is intended to be stored for several months, the temperature should not be above −18°C. For short periods, 0 to 4°C is satisfactory for bulk or printed butter. Butter should be well protected to prevent absorption of off-odours during storage and weight loss from evaporation, and to minimize surface oxidation of fat. The butter temperature when removed from the churn ranges from 13 to 16°C.

The process flow diagram of the butter industry is shown in Figure2.17.

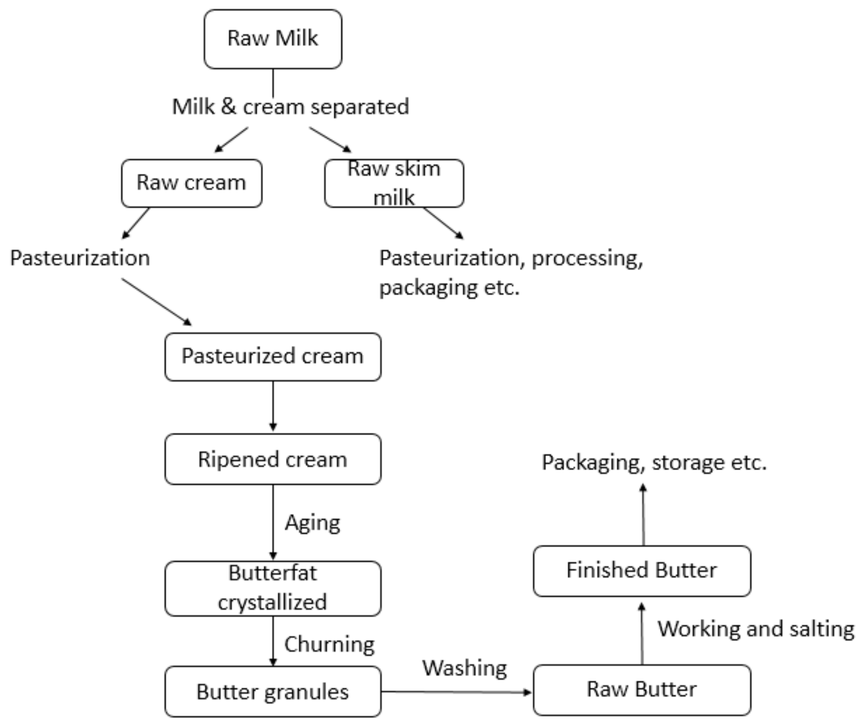

Figure 2.17: Butter manufacturing process

2.3.2.2 Heat exchange processes used in butter manufacturing processing

1. ***Cooling of cream after seperation***

   The Cream will be seperated at a temperature of 40 °C for hot milk separator and must be cooled to a temperature of 7 °C. Cream from cold milk separation does not need to be re-cooled except for extended storage.

2. ***Pasteurization***

   Cream is warmed to a temperature of 27°C to 32°C.Pasteurization process can be divided into two types:

   - ***Batch pasteurization***

Batch Pasteurization is generally used for smaller plants, in which certain fixed amount of cream is heated to certain temperature and then cooled immediately.

(a) ***Heating process***

In batch pasteurization cream is usually heated to 68 to 79°C for 30 min., depending on intended storage temperature and time. The cream is heated in a vat to the desired temperature. The hot water or steam is supplied in the jacketed vat to heat the cream. Then the cream is cooled immediately to temperature from 4 to 13 °C.

(b) ***Cooling process***

Cooling process which is carried out immediately after the required pasteurizing heating temperature is achieved for sufficient time. The Cooling process can be done in two methods:

i. Using Plate heat Exchanger, in which the cream is cooled in two stages. First, tap water is used to reduce the temperature to between 25 to 35°C and second, refrigerated water or brine is then used to reduce the temperature to the desired level.

ii. The cream may be cooled by passing the cooling medium through a revolving coil in the vat or through the vat jacket.

- ***HTST pasteurization***

In HTST continuous pasteurization cream is heated to a temperature of 85 to 95 °C for at least 15 seconds. After pasteurization, the cream is immediately cooled to temperature ranges between 4 to 13°C, depending on the time that the cream will be held before churning, whether it is ripened, season and churning method.

HTST pasteurization may be divided into three zones i.e. heating, regeneration and cooling.

(a) ***Heating***: Cream will be heated from 7 °C to 85-95 °C and this will be done through regeneration section and heating section. If the cream is to be heated to 95 °C, heat gain through regenerative section will be from 7°C to 98°C. Remaining heating will be done in the heating section, using hot water or steam.

(b) ***Regeneration***:If the regeneration temperature, cream temperature and pasteurization temperature is known. The percentage of regeneration may be calculated as follows for equal mass flow rates on either side of the regenerator,

$$\frac{91\,^{\circ}C\,(Regeneration) - 40\,^{\circ}C\,(Raw\,Cream)}{95\,^{\circ}C\,(Pasteurization) - 40\,^{\circ}C\,(Raw\,Cream)} = \frac{44}{55} = 80\,\%$$

The temperature of a product going into the cooling section can be calculated if the percent regeneration is known and the raw cream and pasteurizing temperatures are determined.

$$(95 - 40) \times 0.93 = 51°C$$

$$(95 - 51) = 44°C$$

44 °C is the temperature of the cream going from the regenerator in to the cooling section.

(c) ***Cooling*:** The temperature of the pasteurized cream entering the cooling section is 44°C. This cream needs to be cooled to a temperature of 4 °C in the cooling section. The cooling is achieved using chilled refrigerated water or propylene glycol solution.

All the heat exchange process used for cooling in butter processing is shown in Table 2.7.

Table 2.7: Heat exchange processes used in butter manufacturing processing

	Processes	Temerature	Heat Exchangers Used
1)	**Cooling after Separation**: Cream is separated at temperature of 40°C which is cooled to a temperature of 7°C	40 °C→4 °C	1)Refrigerated jacketed vat 2) Plate Heat Exchanger 3) Tube in tube Heat exchanger
2)	**Pasteurization**		
	Neutralization: Cream develops acidity and is warmed to 32°C and neutralize to 0.12 to 0.15 % acidity.	4 °C →40 °C	Stainless steel lined jacketed vat
	Batch Pasteurization: Cream is heated to 68 to 79 °C and held for 30 min.	40 °C →68 to 79°C	Stainless steel lined jacketed vat, around which hot water or steam is passed
	After 30 min. cream is cooled immediately to a temperature of 4-13 °C.	68 -79°C →25-34 °C→ 4 -13 °C	1) Plate heat exchanger with 1st stage cooling from cooling tower water and 2nd stage cooling from propylene glycol solution. 2) Tube in tube heat exchanger 3) Stainless steel lined jacketed vat
	HTST Pasteurization: Cream is heated to temperature of 85 to 121°C and held at this temperature for at least 15 sec.	40°C→91°C→95°C	Plate Heat Exchanger, shell-and-tube Or triple-tube HTST
	Cooled immediatey	95°C→44 °C→4 °C	Plate Heat Exchanger
3)	**Washing:** washing water is cooled	30°C→7-13°C	Plate Heat Exchanger

2.3.3 Ice Cream Mix Processing

Mix processing begins with combining the ingredients into an homogeneous solution that can be pasteurized, homogenized, cooled, aged, flavoured and frozen. These processes can be further discussed in detail below.

2.3.3.1 processes [31][29]

Different processes done during ice cream mix processing are:

1. ***Preparation of the mix***

   Preparing the mix involves moving the ingredients from the storage areas to the mix preparation area, weighing, measuring or metering them, and mixing or blending them. Undissolved components must be kept in suspension until they are fully hydrated or are dispersed in such small sizes that they remain suspended in the finished mix.Formulation of ice cream mix involves firstly a decision as to the composition of the final product.

   The general procedure includes:

   - Placing all liquid ingredients i.e. milk, cream, concentrated milk, liquid sweeteners, etc. in the vat then the agitation and heating is started at once. The liquid ingredients mix is then warmed to 45°C to 50°C because at this temperature there is rapid dissolving and mixing of two groups of components.
   - Amounts of liquid ingredients can be measured with a calibrated measuring stick, pumped through a volumetric or mass flow meter, pumped into tanks on load cells, or directly added as predetermined volumes or weights.
   - The dry ingredients are then added, with suitable precaution to prevent lump formation. Dry ingredient includes stabilizers, non-fat dry milk solids, powdered eggs and cocoa.
   - Proper suspension to avoid lumpiness of the dry ingredients can be obtained by mixing them with part of the crystalline sugar before adding it slowly to the liquid.
   - The liquid should be cooled to less than 30°C, when non-fat dry milk solids (NMS), cocoa or similar ingredients are added.
   - For adding stabilizer and emulsifier blends to the mix there are different temperature which must be achieved. It will be depending on the manufacturer. Some blends are capable of dissolving at relatively lower temperature, whereas others should not be added to a mix until the temperature reaches about 65°C.
   - Dry ingredients can be blended into liquid materials with an emulsifying agitator that is mounted inside the mix tank.

2. ***Pasteurization of the mix***

Pasteurization is important process and is required for all mixes. Main features of this process are:

- This process destroys all pathogenic microorganisms, thereby safeguarding the health of consumers.
- Most hydrolytic enzymes, even the natural ones of raw milk that could damage flavour and texture are destroyed by pasteurization.
- Pasteurization adds little additional expense, because it is necessary to heat mix to dissolve or hydrate dry ingredients.
- Homogenization can be best accomplished at temperatures near to those of pasteurization.
- Brings solids into solution. • Helps in blending by melting the fat and decreasing the viscosity.
- Improves flavour of most mixes.
- Extends keeping quality to a few weeks.
- Increase the uniformity of product.

Proper pasteurization consist of rapidly heating the mix to a definite temperature, holding at that temperature for minimal time and then rapidly cooling to $<$5°C.

There are two basic methods of pasteurization:

- ***Batch pasteurization***

  In batch system the mix is usually compounded in the vat. Heat is applied, by circulating hot water between the double walls of the vat, while the ingredients are being added and blended. Once all ingredients have been added to the vat and minimal temperature of 69°C has been reached, timing of pasteurization is started. As soon as the minimal time of heating of 30 min. has elapsed, mix is pumped to an homogenizer then to a continuous cooling device such as Plate Heat Exchanger (PHE).Heating and cooling in a vat increases the total heat treatment by large amount resulting in a relatively high intensity of cooked flavour in the mix. However, cooked flavour is not usually objectionable in ice cream.

- ***Continuous pasteurization***

  Continuous pasteurization can be done at several combinations of temperature and time. These types of pasteurizers facilitate use of regenerative heating and cooling with consequent large saving in costs for energy. Most continuous type pasteurizers consist of a series of parallel plates with grooved or waffled surfaces. Heat is exchanged from warmer liquid passing in one direction of flow on one side of the plates to cooler liquid passing in the reverse direction of flow on the opposite side of the same plates.

  Other continuous flow heat exchangers include tube-in-tube, triple tube and steam injection designs. Holding tubes must be designed to expose the mixes to the minimal time that matches the temperature according to the regulations. Holding time depends on the rate of mix flow, length and diameter of the tube and amount of mixing of the fluid in the tube. Continuous-flow pasteurizers must have either a positive displacement type timing pump or a meter based system to control the rate of flow through the holding tube. The simultaneous temperature difference between the hottest and coldest product in any cross-section of flow at any time during the holding period will not be greater than 0.5°C, and the average velocity through the holding tube shall not be less than 1ft/s.

3. ***Homogenization***

   Ice cream mix is homogenized during or immediately after pasteurization before mix is cooled. The main purpose of homogenization is to make a stable and uniform suspension of the fat by reducing the size of fat globules present as large as 10 or 12 µm to less than about 2µm.

   When a mix is properly homogenized, the fat will not rise and form a cream layer nor will the frozen product have a greasy or buttery appearance. Homogenized fat churns very slowly in the freezer so that emulsifiers are usually required to provide the amount of controlled churning that result in a frozen product that is dry in appearance and slow to melt.

   Homogenization is usually accomplished by forcing the mix through a very small orifice under suitable condition of pressure and temperature, using a positive displacement pump to provide the pressure. Homogenizers are piston type pumps of one or two valves and can be used a timing pump in HTST systems.Since several homogenizer valve design exist and mixes vary in their fat content and other components, pressure and necessary to produce adequate dispersion of fat also vary. Commonly accepted pressures for plain mix with 10% milk fat in two stage homogenizer are 13.8 MPa for first stage and 3.45 MPa for the second stage.Homogenization efficiency increases with increase in temperature up to about 80°C.

4. ***Cooling***

   After Pasteurization and homogenization the mix is to be cooled to temperature generally 5°C to 6°C. In smaller plant, to achieve this either surface cooler or plate type heat exchanger is used, with pre cooling and final cooling section. The pre cooling will be done with city, well, or cooling tower water and mix leaving the pre cooling section will be about 35°C to 40°C. Final cooling of the mix is done using chilled water, brine, or direct expansion refrigerant to achieve the temperature of 5°C to 6°C.The low temperature brine solution is achieved through the chillier plant, which operates a refrigeration cycle to achieve brine solution of about -1°C to -2°C.

   For large ice cream plants, where low mix temperature is desired, and where plate type continuous pasteurizing equipment is being used, it may be desired to use separate equipment for final cooling. It is generally consist of three sections heating, regeneration and cooling. Heating is done to achieve pasteurization temperature up to 80°C. The pre cooling will be done in the regenerative section and to the final cooling unit to achieve temperature $<$5°C.

5. ***Aging mixes***

   Aging is a process of keeping mix in jacketed vessel for 4 to 24 hrs at the same temperature of 4°C or less. The average rate of cooling may be estimated as one degree per hr. Purpose of this stage is to allow hydrophilic colloids to hydrate and reduced in size, to increase viscosity, texture to become finer, increase resistance to melting, to improve whip ability, fats to crystallize out and aroma to develop.

   Mixes made up with gelatine as a stabilizer should be aged 24 hr to allow time for the full set of gelatine to develop. Mixes made with sodium alginate or other vegetable type stabilizer develop maximum viscosity on being cooled, and can be used in the freezer immediately.Cooling of mixes to 0-2°C increases the rate of crystallization of fat, increase capacity of freezers, and almost completely eliminates the possibility that micro organisms can grow in the mix. Cooling of mixes to such low temperatures is done efficiently in direct-expansion type swept surface heat exchangers.

6. ***Ice cream freezer***

   Two type of ice cream freezer in general use, the batch type, which freezes a measured quantity of mix at one time, and a continuous type, which takes in continuous flow of ice cream mix and discharge out a continuous flow of partly frozen ice cream. It is arranged with a freezer cylinder having either an annular space or coils around the cylinder where cooling is accomplished by either a flooded arrangement eith an accumulator or direct expansion controlled by a thermostatic expansion valve. The freezer cylinder is arranged with a dasher having blades attached. The dasher revolves within the cylinder. The sharpened blades scrap the frozen film of ice cream as it forms on the inner surface of cylinder. Some freezers have beaters built into the dasher to aid the blades in mixing and whipping to produce the desired overrun. Batch freezer size may vary from 2 to 40 litres of ice cream per batch, and the batch may take 15-20 minutes.

   Continuous ice cream freezers are a more recent development and range in sizes from 150-3000 litres per hour ice cream output. It is use almost exclusively in commercial ice cream plants, where large capacity are required. In operation, the ice cream mix is continuously pumped to the freezer cylinder by a positive displacement pump. Air is supplied by either a separate air compressor or drawn in with the mix through the mix pump. The mixture of ice cream and air enters the rear of the ice cream freezer. The partially frozen ice cream is scrapped out from the inner surface of the freezing cylinder and it moves to the front of the cylinder and is discharged.

   Due to the presence of air in the mix and agitation action by the dasher, air is entrapped within frozen ice cream and gives the overrun. The variation of ice cream discharge temperature can be obtained by regulating the speed of pump or evaporator temperature around the freezer cylinder by means of suction pressure regulating valve. The average discharge temperature of ice cream from continuous freezer is about -5°C, when operating at about -30°C evaporating temperature of the refrigerant. At this temperature, only 40% of total water available in ice cream mix is frozen. The flavoured mix is run through continuous freezer and then passed through a fruit feeder, which automatically mixes nuts or fruits into the ice cream.

7. ***Hardening of ice cream***

   After ice cream leaves the freezer, it is in semi-solid state and must be further refrigerated to become solid enough for storage and distribution. The ideal serving temperature of ice cream is about -13°C and it is considered hard at -18°C.It is necessary to freeze the remaining water content rapidly, so that the ice crystals formed will be small. For this reason most hardening chambers are maintained at the temperature of -30 to -35°C and in some cases as low as -40°C. Hardening chambers are equipped with forced air circulation using unit cooler. The ice cream containers are so arranged so air will circulate around them. It is desirable to maintain the air velocity in the range of 500-700m/min.

   The hardening systems include cabinets, hardening room, hardening tunnels and contact plate freezers. The cabinet design may be straight through with tray or spiral belt freezer. The spiral type hardening tunnel typically receives soft product at the bottom and discharges hardened product at the top. Operating speeds can be varied to match manufacturing speeds and time required for hardening the product.

8. ***Storage and distribution***

   Under proper conditions, ice cream can be stored for several weeks. It is recommended to store ice cream at -25°C in the cold store. However prolonged storage is costly and the chemical, physical and microbiological qualities of the product do not maintain during storage. Ice cream must be protected from surface evaporation, from shrinkage or loss of overrun, and from sugar crystallization. Proper packaging will provide considerable protection against surface deterioration.

   Ice cream must be protected from melting during distribution. The required protection can be provided in several ways. For short period of an hour or two, wrapped in closed packages of hardened ice cream can be placed inside the insulated container which protects rapid changes of the temperature. Extensive use of eutectic pads is done these days for distribution of ice cream. Solution of sodium chloride or mixture of salt is stored in eutectic pads at -18°C. These pads provide refrigeration to the vans used for distribution of ice cream. For long periods mechanical refrigeration must be supplied the preserve the frozen characteristics of the product. The temperature of the frozen product in the distribution system should never get above the desired temperature in the retail counter. i.e. -20°C. Therefore, truck bodies should be well insulated, cooled prior to loading, and kept cold by mechanical refrigeration.

The process flow diagram for ice cream manufacturing is shown below in Figure 2.18.

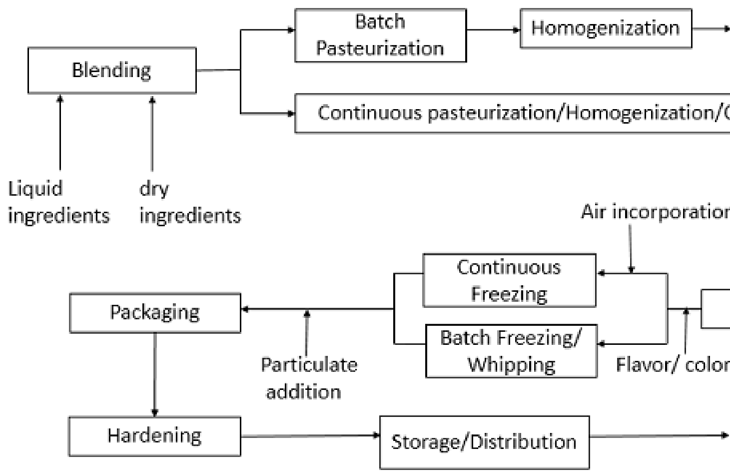

Figure 2.18: Process flow diagram for ice cream manufacture

2.3.3.2 Heat exchange processes used in ice cream manufacturing processing

All the heat exchange process used for cooling in ice cream processing is shown in Table 2.8

Table 2.8: Heat exchange processes used in ice cream manufacturing processing

	Processes	Temperature	Heat Exchagers Used
1)	**Mixing**: liquid ingrediants are heated in vat to temperature 35°C to 40°C	30°C→40°C	Hot water jacketed vat
	Dry ingrediants are then added- liquid should be cooled to 30°C for adding	40°C→30°C	Hot water jacketed vat
	Adding emulsifier and stabilizer	30°C→65°C	Hot water jacketed vat
2)	**Pasteurization**		
	Batch Pasteurization: Heating is done from 65°C to 70°C held for 30 minutes	65°C→70°C	Stainless steel lined jacketed vat, around which hot water or steam is passed
	HTST Pasteurization: heating is done to temperature of 80°C and held at this temperature for at least 25 sec.	65°C→80°C	Plate Heat Exchanger, shell-and-tube Or triple-tube HTST
	Cooling:After 30 min. cream is cooled immediately to a temperature of 5-6 °C.	70-80°C→5-6°C	1) Plate heat exchanger with 1st stage cooling from cooling tower water and 2nd stage cooling from propylene glycol solution. 2) Tube in tube heat exchanger 3) Stainless steel lined jacketed vat
3)	**Aging Vat:** Aging is a process of keeping mix in jacketed vessel for 4 to 24 hrs at the same temperature of 4°C or less.	6°C→4°C	Stainless steel lined jacketed vat
4)	**Continuous Freezer:** Ice cream mix is cooled to -5°C to become partly frozen ice cream	4°C→-5°C	Jacketed Cylinder-Evaporator Shell and tube - Condenser

2.4 Objective of Present Study

Based on the literature reviewed it was found that there is a need for further theoretical investigation to determine the optimum condition of a ice based latent heat storage system for an external melt. This will be done by performing heat transfer analysis to calculate and compare different thermal parameters at different operating conditions for a case study. The main parameters on which the optimization depends are, thickness of the ice formed around the tubes, compressor power and the overall heat transfer coeffiecient. The refrigeration load does not remain constant. The ice built up leads to reduced heat transfer and other circulation problems in the system. However, the ice build-up, if regulated, can save energy in terms of compressor power and also heat transfer rate can be maintained.The system consists of tank with tubes submerged in water, a condenser, compressor, expansion valve and accumulator.The optimization for the ice formed around the tubes during charging process is proposed to find the energy efficient and economic option.

Hence,objectives of the present study are summarized as under:

- To perform heat transfer analysis on the ice based latent heat storage system at different operating conditions.
- To find an optimum ice thickness around the tubes.
- To find an energy efficient design.
- To validate the results with the industrial application.

Chapter 3

Heat Transfer Analysis

Heat transfer analysis of ice based latent heat storage system was performed to calculate the overall heat tranfer coefficient, area and total length of tube. These calculations requires dealing with the two phase heat transfer problems. The calculations are performed for different ice thickness around the tube at different time durations and the results obtained are used to optimize the system. The evaporator system here is flooded type and condition of liquid at the entry of evaporator will always be saturated. this system is explained below.

3.1 Flooded Evaporator System

A flooded evaporator type[27] with float control valve is shown in Figure 3.1. The liquid refrigerant enter from lower end of evaporator and boils due to heat absorption from warmer liquid as it move upwards. The resulted vapor so formed due to boiling gets collected in a accumulator.The vapor remains at the top of accumulator where as the liquid remains at the bottom. Separated vapor passes to compressor, and liquid flows back to the evaporator. The accumulator also collects the vapor obtained in the expansion device. In a flooded type evaporator refrigerant liquid level is maintained. Float valve is used as throttling device.

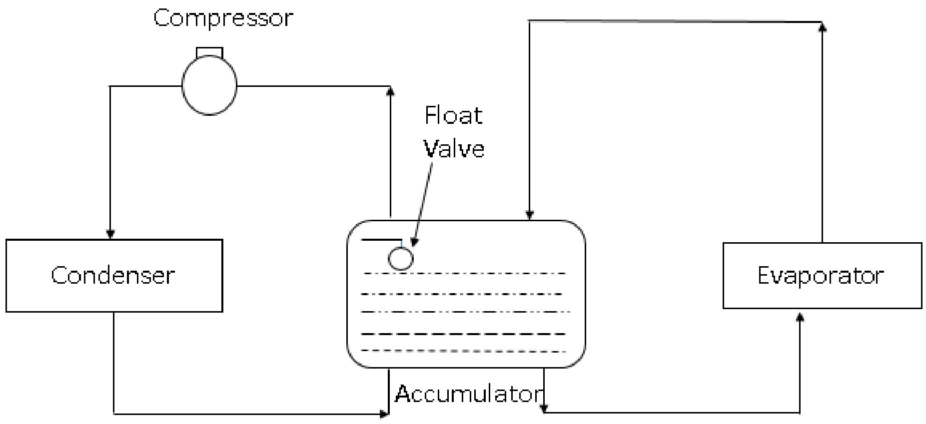

Figure 3.1: Ice storage charging system

The heat transfer efficiency increases because the entire surface is in contact with the liquid refrigerant. But the refrigerant charge is relatively large as compared to dry expansion type.The accumulator is used to prevent liquid carry over to compressor. The evaporator coil is contacted to accumulator and the liquid flow from the accumulator to the evaporator coil is generally by gravity. The vapor formed by the vaporizing of the liquid in the coil being lighter rises up and passes on to the top of the accumulator from where it enters the suction line. In some cases liquid eliminators are provided in the accumulator top to prevent the possible carry over of liquid to suction line.

3.2 Ice Formation Rate for a Thermal Storage System

A model for predicting the ice formation rate around an evaporator coil is presented in this section. A direct approach to the formulation of an equation for ice formation rate is to equate the heat transfer per unit length of evaporator coil to the extraction of the latent heat of fusion necessary to form the ice around the evaporator coil.[28]

The mass rate of ice formation on a surface is given by

$$\frac{dm}{dt} = \frac{Q_e}{h_{sf}} \tag{3.1}$$

where $Q_e$is the heat transfer rate to the surface and h_{sf} is the latent heat of fusion. The mass of ice formed on an evaporator coil of length L immersed in water is given by

$$m = \pi(r^2 - r_o^2)\rho L \tag{3.2}$$

where,

ρ= density of ice

The evaporator output is given by,

$$Q_e = Q_i + Q_s \tag{3.3}$$

where,

Q_s= rate of heat tranfer from the surrounding to the thermal storage tank

The evaporator output is a function of the evaporator temperature, T_e,as indicated in figure. for a typical condensing unit. This function can be approximated by a straight line give as,

$$Q_e = Q_f - b(T_f - T_e) \tag{3.4}$$

where,

Q_f= evaporator output extrapolated to an evaporator temperature equal to the freezing temperature of water, T_f

The average slope of the $Q_e(T_f$-$T_e)$ function is given by

$$b = \frac{Q_{max} - Q_{min}}{T_{max} - T_{min}} \tag{3.5}$$

where $Q_{max}$and $Q_{min}$are the maximum and minimum rated output corresponding to $T_{max}$and T_{min}.

The evaporater output is inversely prortional to the thermal resistance R, between the refrigerant and the water. Thus,

$$Q_e = \frac{A_s}{R}(T_f - T_e) \tag{3.6}$$

where T_e is the evaporator temperature and T_f is freezing temperature of water i.e 0°C

$$A_s = 2\pi r L \tag{3.7}$$

The thermal resistance term is give by

$$R = \frac{r}{r_i}\left(\frac{1}{h_i} + F + C\right) + \frac{r}{k_l} ln\left(\frac{r_o}{r_i}\right) + \frac{r}{k} ln\left(\frac{r}{r_i}\right) + \frac{1}{h} \tag{3.8}$$

where,

h_i, h_o= internal and external heat transfer coefficients

F = fouling factor for the inside of the pipe wall

k_l, k = thermal conductivity for the pipe and ice, respectively.

Since ice radius increases with time, the thermal resistance, R, increases with time. The minimum value of R occurs at time = 0 when r = t_o, and is given by the following expression:

$$R_{min} = \frac{r_o}{r_i}\left(\frac{1}{h_i} + F + C\right) + \frac{r_o}{k_l} ln\left(\frac{r_o}{r_i}\right) + \frac{1}{h_o} \tag{3.9}$$

where $h_o = h$ and r = r_o

Overall heat tansfer coefficient can be calculated using equation,

$$Q_e = UA\Delta T \tag{3.10}$$

Multiplying equation by r_o/r and using equation , then

$$\frac{r_o}{r}\frac{R}{R_{min}} = 1 + \frac{R'}{R_{min}} \tag{3.11}$$

where

$$R' = \frac{r_o}{k} ln\left(\frac{r}{r_o}\right) + \left(\frac{r_o}{rh} - \frac{1}{h_o}\right) \tag{3.12}$$

Assuming that the evaporator output at time zero (t = 0) is Q_{max}(with corresponding temperature T_{max}), then the evaporator length can be obtained from eqution

$$2\pi r_o L = \frac{Q_{max} R_{min}}{T_f - T_{max}} \tag{3.13}$$

From equation 3.1, 3.2, and 3.3 , the radial growth rate of ice formation is given by

$$\frac{dr}{dt} = \frac{Q_e - Q_s}{2\pi r_o L \rho h_{sf}\left(\frac{r}{r_o}\right)} \tag{3.14}$$

substituting for $2\pi r_o L$ from equation 3.13, gives

$$\frac{dr}{dt} = \frac{T_f - T_{max}}{\rho h_{sf}\left(\frac{r}{r_o}\right)}\frac{1}{R_{min}}\left(\frac{Q_e}{Q_{max}} - \frac{Q_s}{Q_{max}}\right) \tag{3.15}$$

from equation , 3.4, 3.6, 3.7, 3.11and 3.13, it can be shown that,

$$\frac{Q_e}{Q_{max}} = \frac{1}{1 + \frac{R'}{R_{min}}(1 - \frac{Q_{max}}{Q_f})} \tag{3.16}$$

Sustituting into equation 3.15 gives,

$$\frac{dr}{dt} = \frac{T_f - T_{max}}{\rho h_{sf}\left(\frac{r}{r_o}\right)}\left[\frac{1}{R_{min} + R'(1 - Q_{max}/Q_f)} - \frac{Q_s/Q_{max}}{R_{min}}\right] \tag{3.17}$$

Non dimensionalizing the above equation,

$$r^* = \frac{r}{r_o} \tag{3.18}$$

$$t^* = \frac{(T_f - T_{max})k}{(1 - Q_{max}/Q_f) r_o^2 \rho h_{sf}}.t$$

$$\frac{1}{R^*} = \frac{kR_{min}}{r_o(1 - Q_{max}/Q_f)} \tag{3.19}$$

equation 3.17 can be written as

$$\frac{dr^*}{dt^*} = \frac{1}{r^*}\left[\frac{1}{\frac{1}{R*} + lnr^* - \frac{k}{r_o h_o}(1 - \frac{h_o}{r^* h})} - R^*\frac{Q_s}{Q_{max}}\right] \tag{3.20}$$

3.3 Case Study for Heat Transfer Calculations

A case study is performed for an ice based latent heat storage system. The conditons for which calculations are performed are given below.

1. The area of the tank used is 39 m^2.
2. Total length of the coil is 1000 m.
3. Compressor Model is of kirloskar KC4 series which is a single stage compressor with 4 cylinders
4. Operating conditions are -15°C/+40°C, 0°C Subcooling, 0°C Superheating and 1000 rpm.
5. Flooded type evaporator system is used.
6. Refrigerant used is ammonia (R717).
7. Coil is of 32mmNB type with outside diameter of 42.4mm and wall thickness of 4.05 mm.

3.3.1 Heat transfer analysis

Heat transfer calculations is performed for the heat exchange process taking between the refrigerant R717 and the water[16]. The calculations will be performed for different ice thickness (t_s) around the tube from 0 mm i.e. for bare tube to 63.5 mm. The calculations will give the value overall heat transfer coefficient. To find the overall heat transfer coefficient different evaporator temperature will be taken from between -5 ºC to -15 ºC and then their average will be done to find the exact value.

At time t=0, the ice thickness will be t_s= 0 mm. For refrigerant evaporator temperature of -5 ºC and water outlet temperature and the inlet temperature of 9 ºC and 1 ºC respectively calculations are performed.

- Operating Conditions of the system are:

$$Swept\, volume\, at\, 1000\, rpm = 530.8\, m^3/hr$$

$$Length\, of\, Tube = 1000\, m$$

$$Refrigeration\, capacity,\, Q_e = 354.65$$

$$Evaporator\, temperature,\, T_e = -5°C$$

$$Condensor\, temperature,\, T_c = 40°C$$

$$Power\, required\, shaft,\ P_e = 92.1\, kW$$

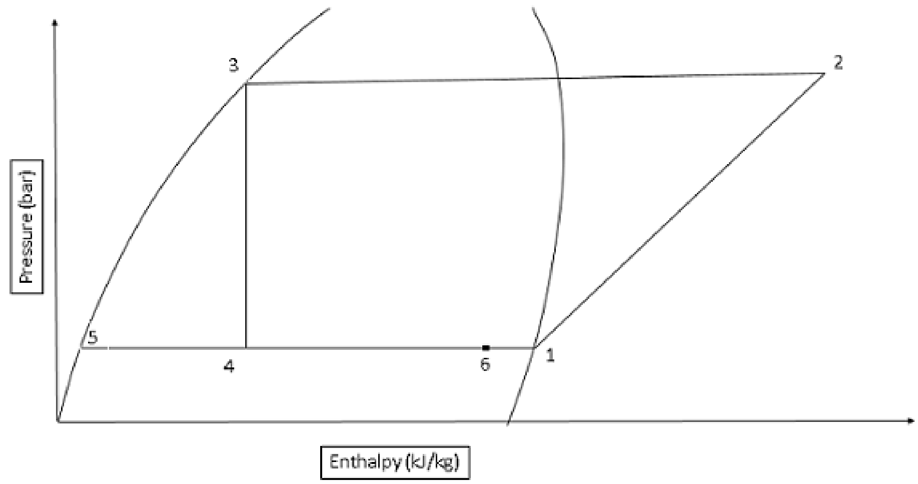

Figure 3.2: Pressure-enthalpy diagram

- **Processes**

  1-2: Compression

  2-3: Condensation

  3-4: Expansion

  4-5: To low pressure liquid refrigerant accumulator

  5-6: Evaporation

  6-1: To low pressure liquid refrigerant accumulator

Quality of vapour after expansion process can be calcuated as:

$$Evaporator\, Pressure\, at\ -5°C = 3.55\, bar$$

$$Condensor\, Pressure\, at\, 40°C = 15.57\, bar$$

$$Entahlpy\, at\, point\, 1,\ h_1 = 1456.15\, kJ/kg$$

$$Entahlpy\, at\, point\, 2,\ h_2 = 1666\, kJ/kg$$

$$Entahlpy\, at\, point\, 3,\ h_3 = 390.58\, kJ/kg$$

$$Entahlpy\, at\, point\, 4,\ h_4 = 390.58\, kJ/kg$$

$$h_4 = (1 - x_4)h_{f_e} + x_4.h_{g_e}$$

$$390.58 = 176.96 \times (1 - x_4) + x_4 \times 1456.15$$

$$x_4 = 0.16$$

3.3.1.1 External side heat transfer coefficint[?]

Properties of water at bulk mean temperature $T_b = 5^oC$ are

$$\rho = 999.9\,kg/m^3$$
$$k = 0.571\,W/m.K$$
$$Pr = 11.2$$
$$C_p = 4.18\,kJ/kg.K$$
$$\mu_l = 1.53 \times 10^{-3} kg/m.s$$
$$\mu_v = 0.93 \times 10^{-5} kg/m.s$$
$$\beta = 0.015 \times 10^{-3}$$
$$v = 0.5\,m/sec$$

Hydraullic Diameter,

$$d_h = 4 \times \frac{Net\,free\,flow\,area}{Wetted\,perimeter} \tag{3.21}$$

$$d_h = 4 \times \frac{[(3.9 \times 2.5) - \frac{\pi}{4}(0.0424)^2]}{[(2 \times 3.9) + (2 \times 2.5) + (\pi \times 0.0424)]}$$
$$d_h = 3.01\,m$$

Reynolds Number,

$$Re = \frac{\rho v d}{\mu} \tag{3.22}$$

$$Re = \frac{999.9 \times 0.5 \times 3.01}{1.53 \times 10^{-3}}$$
$$Re = 998269$$

Length of Coil is 1000 meter, so volume occupied by coil in the tank is,

$$V_c = \left(\frac{\pi}{4} d_o^2\right) \times L \tag{3.23}$$

$$V_t = \frac{\pi}{4} \times (0.0424)^2 \times 1000$$
$$V_t = 1.41\,m^3$$

Volume of tank,

$$V_t = L \times W \times H \tag{3.24}$$
$$V_t = 4 \times 3.9 \times 2.5$$
$$V_t = 39\,m^3$$

Volume available for flow,

$$V_f = 39 - 1.41 = 37.58\, m^3$$

Equivalent Diameter,

$$D_e = \frac{4 \times V_f}{\pi \times d_0 \times L} \tag{3.25}$$

$$D_e = \frac{4 \times 37.58}{\pi \times 0.0424 \times 1000}$$

$$D_e = 1.12m$$

For Re > 10000, correlation used is:

$$\frac{h_o D_e}{K} = 0.36\,(Re)^{0.55}\,(Pr)^{\frac{1}{3}}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{3.26}$$

$$\frac{h_o \times 0.55}{0.568} = 0.36(998269)^{0.55}(11.2)^{\frac{1}{3}}(\frac{1.53 \times 10^{-3}}{1.99 \times 10^{-3}})^{0.14}$$

$$h_o = 774.75\, W/m^2K$$

3.3.1.2 Tube side heat transfer coefficient

Saturated liquid properties of refrigerant R717 at -5°C are

$$p = 3.55bar$$

$$\Delta h_v = 1279200\, J/kg$$

$$h_f = 176.9\, kJ/kg$$

$$h_g = 1456.1\, kJ/kg$$

$$\rho_v = 2.89\, kg/m^3$$

$$\rho_l = 645.4\, kg/m^3$$

$$C_{p_l} = 4589\, J/kg.K$$

$$K_l = 0.5505\, W/m.K$$

$$\mu_v = 9.218 \times 10^{-6}\, kg/m.s$$

$$\mu_l = 2.003 \times 10^{-4}\, kg/m.s$$

$$\sigma = 0.02786\, N/m$$

The mass flow rate of refrigerant can be calculated using overall energy balance,

$$Q = m_r(h_{out} - h_{in}) \tag{3.27}$$

$$Q = 354.65kJ/sec$$

$$354.65 = m_r \times (1456.15 - 176.9)$$

$$m_r = 0.277\, kg/sec$$

Prandtl number can be calculated by:

$$Pr = \frac{\mu_l . c_{pl}}{k_l} \tag{3.28}$$

$$= \frac{(2.003 \times 10^{-4}) \times 4589}{0.5505} = 1.66$$

Reynolds number for the liquid phase can be calculated by:

$$Re_{lo} = \frac{G.di}{\mu_l} \tag{3.29}$$

$$G = \frac{m_r}{\frac{\pi}{4}(d_i)^2}$$

$$G = \frac{m_r}{\frac{\pi}{4}\left(d_i\right)^2} \tag{3.30}$$

$$G = \frac{0.28}{\frac{\pi}{4}(0.0343)^2} = 300.19\, kg/m^2.sec$$

$$Re_{lo} = \frac{300.19 \times 0.0343}{2.003 \times 10^{-4}} = 51484$$

Nusselt number can be calculated using Gnielinski correlation:

$$Nu = \frac{\left(\frac{f}{2}\right)(Re_{lo} - 1000)Pr_l}{1 + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr_l^{\frac{2}{3}} - 1\right)} \tag{3.31}$$

where

$$f = (1.58 ln\, Re_l - 3.28)^{-2} \tag{3.32}$$

$$f = (1.58 ln(51484) - 3.28)^{-2}$$

$$f = 5.20 \times 10^{-3}$$

$$Nu = \frac{\left(\frac{5.20 \times 10^{-3}}{2}\right)(51484 - 1000)(1.66)}{1 + 12.7\left(\frac{5.20 \times 10^{-3}}{2}\right)^{\frac{1}{2}}\left(1.66^{\frac{2}{3}} - 1\right)} = 173.87$$

Heat transfer coefficient is calculated by

$$h_{lo} = \frac{Nu_{lo}.k_l}{d_i} \tag{3.33}$$

$$= \frac{173.87 \times 0.5505}{0.0343}$$

$$= 2790\, W/m^2k$$

By using the shah method, whether the effect of stratification is important or not. For this Froude number is calculated,

$$Fr_{lo} = \frac{G^2}{\rho_l^2 g d_i} \tag{3.34}$$

$$Fr_{lo} = \frac{(300.19)^2}{(645.4)^2 \times 9.81 \times 0.0343} = 0.64$$

Fr>0.04, therefore stratification effects are negligible

Shah's method at x = 0.01

Convection number is calculated by

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} K_{FR} \tag{3.35}$$

$$Co = \left(\frac{1-0.01}{0.01}\right)^{0.8} \left(\frac{2.886}{645.6}\right)^{0.5} \times 1 = 2.64$$

Convection boiling factor

$$F_{cb} = 1.0 + 0.8\, exp[1 - (Co)^{0.5}] \quad for\, Co > 1.0 \tag{3.36}$$

$$F_{cb} = 1.428$$

Enhancement factor

$$F_o = F_{cb}(1-x)^{0.8} \tag{3.37}$$

$$F_o = 1.428(1-0.01)^{0.8} = 1.428$$

Covective boiling heat transfer coefficient

$$h_{cb} = F_o.(h_{lo}) \tag{3.38}$$

$$h_{cb} = 1.428 \times 2790 = 3985\, W/m^2k$$

Overall heat transfer coefficient with neglgible wall resisitance and fouling:

$$U = \left(\frac{1}{h_{cb}} + \frac{1}{h_o}\right)^{-1} \tag{3.39}$$

$$U = \left(\frac{1}{3985} + \frac{1}{774.75}\right)^{-1} = 648.63\, W/m^2k$$

Heat flux,

$$q" = U \times (T_H - T_S) \tag{3.40}$$

$$q" = 648.63 \times (5 - (-5)) = 6486.33\, W/m^2k$$

The wall temperature for the onset of nucleate boiling can be determined,

$$T_{WONB} = \left(\frac{8\sigma q" T_s}{k_l \triangle h_v \rho_v}\right)^{0.5} + T_s \tag{3.41}$$

$$T_{WONB} = \left(\frac{8 \times 0.02786 \times 6486.33 \times 268}{0.5505 \times 1279200 \times 2.89}\right)^{0.5} + 268 = 268.43K$$

The wall temperature can be calculated,

$$T_w = \frac{q"}{h_{cb}} + T_s \tag{3.42}$$

$$T_w = \frac{6486.33}{3985} + 268 = 269.62\,K$$

$T_w > T_{WONB}$ and nucleate boiling is present,

Boiling number is calculated as,

$$Bo = \frac{q"}{G.\triangle h_v} \tag{3.43}$$

$$Bo = \frac{6486.33}{300.19 \times 1279200} = 1.6 \times 10^{-5}$$

This is the approximate value of the boiling number and Bo $<$ 1.9 $\times 10^{-5}$, therefore nucleate boiling is negligible.

The Overall thermal resistance,

$$R = \frac{r}{r_i}\left(\frac{1}{h_i}\right) + \frac{r}{k_l} ln\left(\frac{r_o}{r_i}\right) + \frac{r}{k} ln\left(\frac{r}{r_i}\right) + \frac{1}{h} \tag{3.44}$$

In this case, minimum value of R occurs at time = 0 when thickness of ice is 0 mm, and is given by the following expression:

$$R_{min} = \frac{r_o}{r_i}\left(\frac{1}{h_i}\right) + \frac{r_o}{k_l} ln\left(\frac{r_o}{r_i}\right) + \frac{1}{h_o} \tag{3.45}$$

where $h_o = h$ and $r = r_o$

$$R = R_{min} = \frac{0.0212}{0.01715}\left(\frac{1}{3985}\right) + \frac{0.0212}{53.6} ln\left(\frac{0.0212}{0.01715}\right) + \frac{1}{774.75}$$

$$R = R_{min} = 0.001684\,m^2.k/W$$

$$U = 593.55\,W/m^2.k$$

Similarly many more iteration are performed taking different evaporator temperature between -5°C and -15°C, to find the overall heat transfer coefficient which will be taken as an average. These calculations will be performed for every 5 mm increase in ice thickness over the coil. By using trial and error method and refrigeration capacity, evaporator temperature can be found.Iterations will be performed using Microsoft Excel and the results obtained are shown in the Table 3.1.

Table 3.1: Results of iterative calculation performed at different ice thickness

Ice Thickness(mm)	Overall Heat Transfer Coeff.,Uo($W/m^2.K$)
0	593.55
5	253.11
10	148.40
15	100.89
20	74.48
25	58.03
30	46.95
35	39.12
40	33.28
45	28.78
50	25.28
63.5	18.70

Chapter 4

Optimization of the System

4.1 Load Distribution in a Dairy Industry

An accurate design day cooling load profile is recommended for sizing the various system components. Before selecting the system components, it is necessary to make decisions regarding the overall system operation. Consider the following:

1. Review the electric utility company's time-of-day rate schedule as these are needed in the thermal ice storage design process.

2. Select either external melt or internal melt as the basis of design of the thermal ice storage system.

3. Determine the chilled water system design flow and delta-T. Evaluate the use of glycol / chilled water or ice water / chilled water heat exchangers.

4. Most thermal ice storage system designs will be for partial storage. However, full storage should be considered in areas where energy supplies are limited or very expensive.

5. Review the design day cooling load profile and determine if conventional chillers will be necessary for cooling during non-peak and on-peak hours.

4.2 Case Study

Operating Information of the System Selected:

1. The electric utility Company's time-of-day rate schedule is different in every state/country. But it is generally divided into peak hour period and off-peak hour period.

2. An external melt thermal ice storage system was selected.

3. The system chilled water loop is based on a 8 °C delta-T (9 °C to 1°C).

4. The system can be a partial ice storage system/full ice storage system.

5. Refrigerent on tube side will be Ammonia (NH_3).

Different processes carried out in a dairy industry require cooling which is achieved from the chilled water. The temperature of water available for cooling from ice based latent heat storge system is generally at 1°C. These processes are given below in the Table 4.1.s

Table 4.1: Different processes in dairy industries which require cooling from ice based latent heat storge system

S. No.	Process	Temperature	Quantity	Time of day
1.	Milk Chilling	30°C →4°C	40000 litre	9 am to 11 am
		30°C →4°C	30000 litre	9 pm to 11 pm
2.	Milk from chilling centre	10 °C →4°C	80000 litre	12 noon to 1 pm
3.	Milk Pasteurization	11 °C →4°C	140000 litre	9 am to 2 pm 3 pm to 7 pm
4.	Cream used for butter manufacturing	40 °C →4°C	10000 litre	10 am to 3 pm
5.	Cream Pasteurization	44 °C →4°C	10000 litre	10 am to 3 pm 4 pm to 7 pm

There will be additional cooling load for milk cold store room which will be of 35 kW and operating time will be between 10 am to 4 am.

Cooling load during different time of the day can be find using equation,

$$Q = m \times C_p \times \Delta T \tag{4.1}$$

where,

Q = Cooling load

m = mass in kg

C_p= Specific heat in kJ/kg

Properties of milk and milk derivatives are given in appendix A.

Table 4.2: Cooling load during different time of the day

Time (hours)	Mass (kg)	Cooling Load (kW)
9 am to 11 am	40000×1.025 = 41000 kg	$Q = \frac{41000 \times 3.955 \times (30-4)}{10800} = 390.37\, kW$
9 pm to 11 pm	30000×1.025 = 30750 kg	$Q = \frac{30750 \times 3.955 \times (30-4)}{10800} = 292.77\, kW$
12 noon to 1 pm	80000×1.033 = 82640 kg	$Q = \frac{82640 \times 3.9 \times (10-4)}{3600} = 537.16\, kW$
9 am to 2 pm	77778×1.035 = 80500.23 kg	$Q = \frac{80500.23 \times 3.944 \times (11-4)}{18000} = 123.46\, kW$
3 pm to 7 pm	62222×1.035 = 64399.77 kg	$Q = \frac{64399.77 \times 3.944 \times (11-4)}{14400} = 123.46\, kW$
10 am to 3 pm	10000×0.978 = 9780 kg	$Q = \frac{9780 \times 3.567 \times (40-4)}{18000} = 69.77\, kW$
10 am to 3 pm	6250×1.008 = 6300 kg	$Q = \frac{6300 \times 3.6 \times (44-4)}{18000} = 50.4\, kW$
4 pm to 7 pm	3750×1.008 = 3780 kg	$Q = \frac{3780 \times 3.6 \times (44-4)}{10800} = 50.4\, kW$

The design day performance cooling load is formed and shown below in Table 4.3:

Table 4.3: Cooling load for 24 hours

Time of day	Cooling Load, Q (kW)
9 am -10 am	514.33
10 am -11 am	669
11 am - 12 pm	278.63
12 pm -1 pm	815.79
1 pm -2 pm	278.63
2 pm -3 pm	155.17
3 pm -4 pm	158.46
4 pm -5 pm	208.86
5 pm -6 pm	208.86
6 pm -7 pm	208.86
7 pm - 8 pm	35
8 pm -9 pm	35
9 pm -10 pm	327.77
10 pm -11 pm	327.77
11 pm -12 am	35
12 am -1 am	35
1 am -2 am	35
2 am -3 am	35
3 am -4 am	35
4 am -5 am	0
5 am -6 am	0
6 am -7 am	0
7 am -8 am	0
8 am -9 am	0

4.3 Load Profile

The load Profile of the dairy plant can be formed for the cooling loads of the whole day. This data is plotted for time (hrs) v/s cooling load (kW).

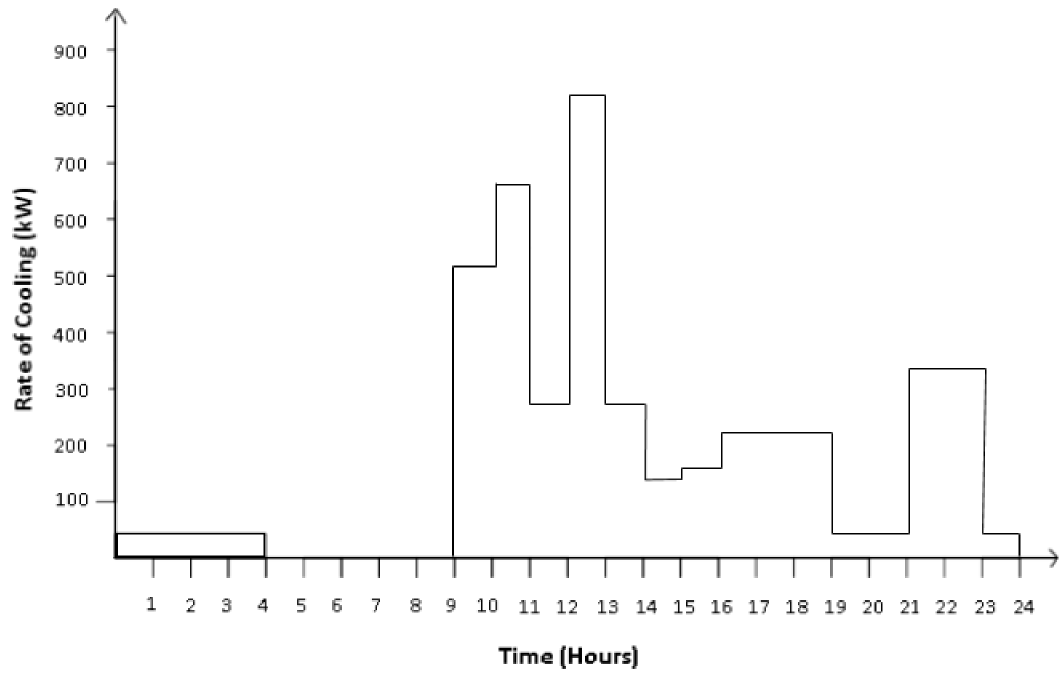

Figure 4.1: Cooling Load Profile

The Load Profile shown in Figure4.1, is for 1,50,000 litre of milk per day in the dairy industry. The electricity rate schedule is different for different states throughout the country as given in appendix G. For Gujrat (India), schedule is,

- On-Peak Hours are 07:00 AM to 11:00 AM and 06:00 PM to 10:00 PM
- Off-Peak Hours are 10:00 PM to 6:00 AM

Here, the load to be met by the refrigeration unit will be quite high, i.e Rate of Cooling required will be 815.79 kW. The average unit electric power consumption will also be high.

If ice latent heat storage system is used, the refrigeration capacity of the compressor will reduced and the peak load to be meet will be decreased. This system will reduce the operating cost because the compressor will be reduced to great amount. Using 278 kW refrigeration capacity compressor, the reduction in peak load is shown in Figure4.2.

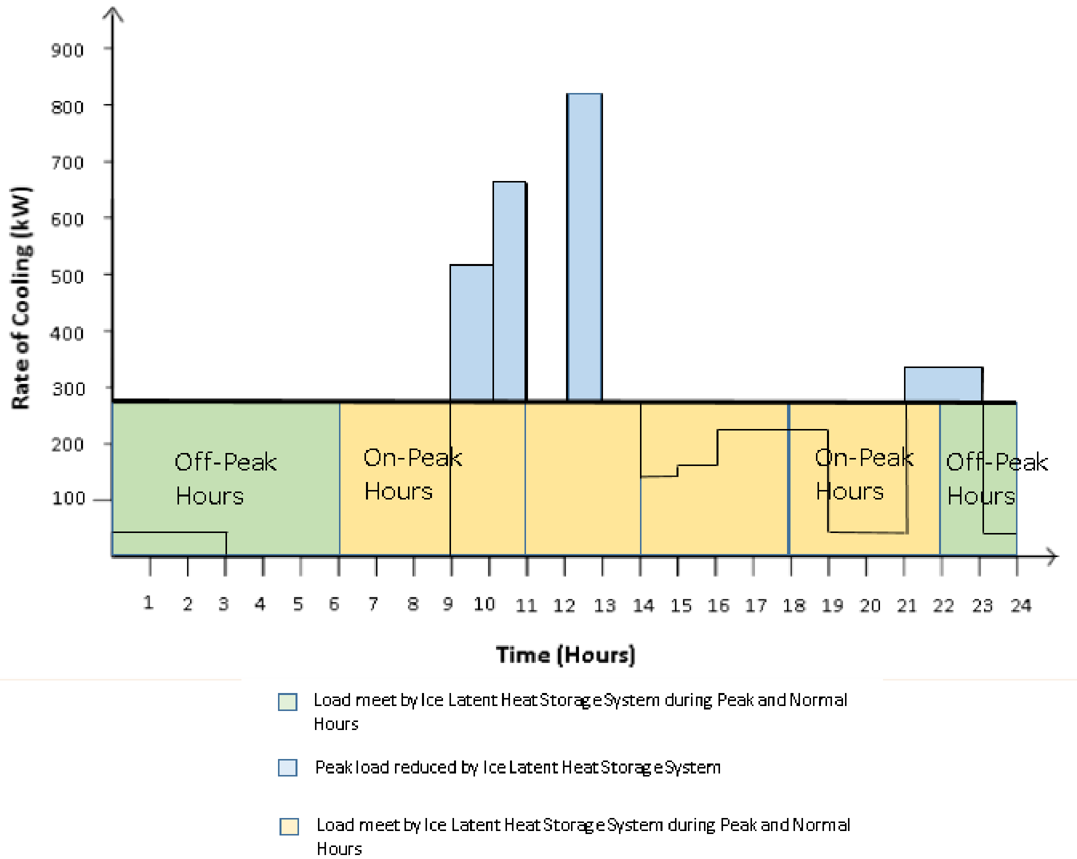

Figure 4.2: Cooling load profile- ice latent heat storage system

4.4 Optimum Ice Thickness

The Tariff charges during off peak hours are less than on peak hours, this can be used as an advantage and the ice can built in the latent heat storage system around the coil during the off peak hours. This leads to certain amuont of saving in the cost. But load required to be meet by the latent heat storage system during the entire day is generally more then the amount of of latent heat stored by the system.The maximum ice built during off-peak hours gets melt and their is a requirement of building ice during the peak hours is well. The saving in the cost made during the off peak hours can be used to built ice in on peak hours without costing any extra charges. The Tariff rates are different for different electricity company and also varies for different states troughout the country. For Gujrat (India), the tariff rates according to the Gujarat Electricity Regulatory Commission are given below in Table 4.4.

Table 4.4: Tariff rates according to the Gujarat Electricity Regulatory Commission

Time	Rate
10:00 PM - 06:00 AM	Normal Charge - 75 Paise
06:00 AM - 11:00 AM	Normal Charge + 75 Paise
11:00 AM - 02:00 PM	Normal Charge + 30 Paise
02:00 PM - 06:00 PM	Normal Charge
06:00 PM - 10:00 PM	Normal Charge + 75 Paise

- Normal charges are Rs. 4.20 per unit
- Peak hours are from 6 am to 11 am and 6 pm to 10 pm

The saving during off peak hours i.e. 10:00 PM to 06:00 AM will be used to built ice during on peak hours. This amount of saving will give the optimum ice thickness which can be built during peak hours without any extra cost.

4.5 Efficient Design

In Ice Based Latent Heat Storage System coils are used for building ice, so that constant leaving water temperature can be maintained during the discharge cycle. In the regular conventional system, ice is built around the coil called 32mmNB which is having outer diameter of 42.4mm and a wall thickness of 4.05mm.

A more efficient design is suggested which is having smaller diameter coils i.e. 20mmNB which is having outer diameter of 26.9mm and a wall thickness of 3.25mm. This will also result in thinner ice formation around the coil to meet the cooling load requirement and will have more latent heat storage then conventional system.

The proposed design will result in lesser compressor work and gain in refrigeration effect and coefficient of performance.

4.5.1 Conventional regular ice builder coil

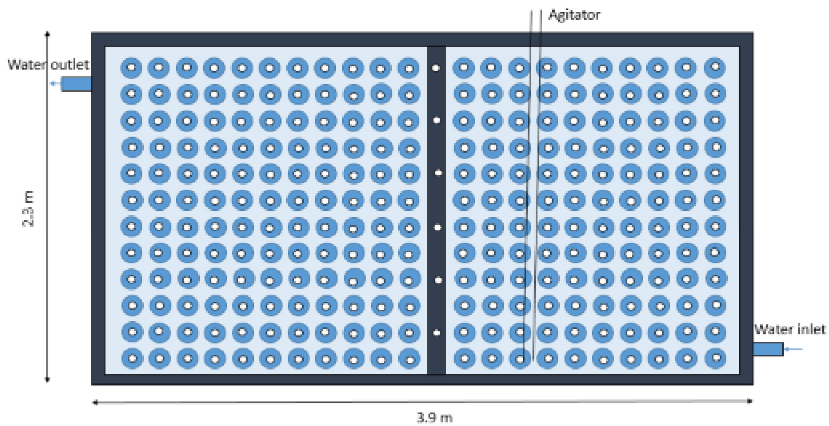

Figure 4.3: Design of ice based latent heat storage system

For Conventional system the maximum ice allowed to build over coil is 63.5 mm ice. Refrigerant temperature at this ice thickness will be calculated and according other parameters will be found. The length of the coil is 1000 m.

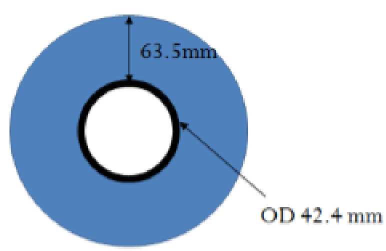

Figure 4.4: Coil of a convetional system

From the result table 5.1, the evaporator temperature can be found which will be -14.99°C. The other values will be:

$$Coil\,Type = 32mmNB$$

$$Length\,of\,Coil, L = 1000\,m$$

$$Total\,Outside\,diameter\,after\,building\,of\,ice = 0.1694\,m$$

$$Surface\,Area,\; A = \pi \times d \times l = 531.91m^2$$

$$Evaporator\,Temperature,\; T_e = -14.99°C$$

The Refrigeration capacity from KC4 chart of Kirloskar Compressors will be,

$$Refrigeration\,Capacity, Q_e = 210.7\,kW$$

$$Power\,Consumption, P_e = 75\,kW$$

$$Specific\,Power\,Consumption,\ SPC = 0.35$$

$$Latent\,Heat\,Storage,\ LHS = \frac{\pi}{4}[(0.1694)^2 - (0.0424)^2]\times 1000\times 999$$

$$= 21094.3kg\,of\,ice$$

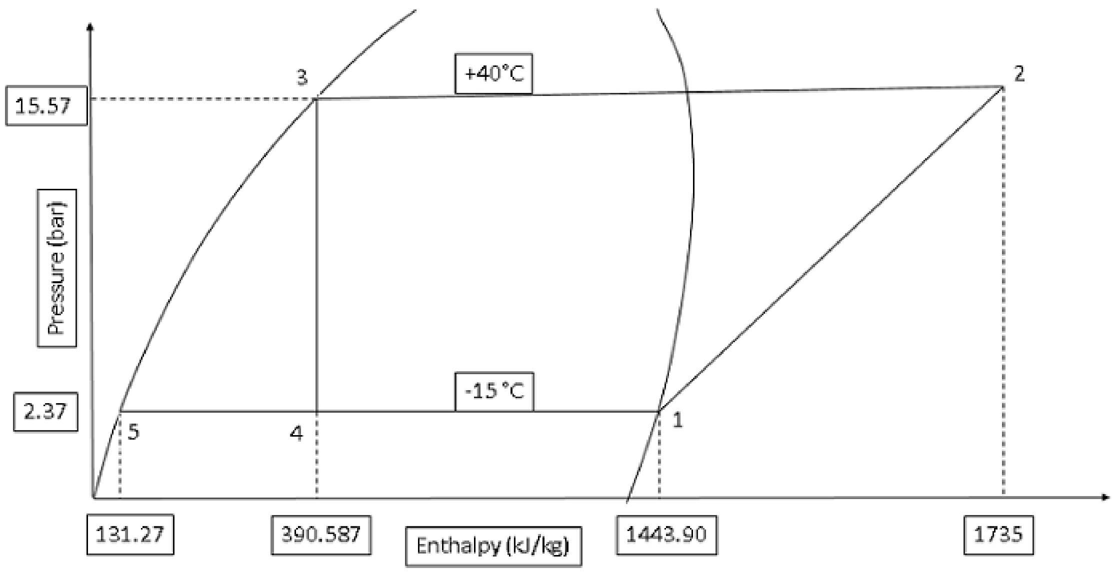

Figure 4.5: P-h diagram

$$Refrigeration\,effect = Q_1 = (Enthalpy\,at\,point\,1) - (Enthalpy\,at\,point\,4)$$

$$= (h_1 - h_4) = (1443.90 - 131.27)$$

$$= 1312.63\,kJ/kg$$

$$Work\,of\,Compression = W_1 = (Enthalpy\,at\,point\,2) - (Enthalpy\,at\,point\,1)$$

$$= (h_2 - h_1) = (1735 - 1443.90)$$

$$= 291.1\,kJ/kg$$

$$Coefficient\,of\,Performance = \frac{(h_1 - h_4)}{(h_2 - h_1)} = 4.5$$

4.5.2 Improved ice builder system coil

Many different type of standard coils are available in the market having different diameter and thickness, shown in Table 4.5.

Table 4.5: Standard available coils[32]

Heavy Steel Tube Type		
Nominal Bore (mm)	Outside diameter (mm)	Wall Thickness (mm)
15	21.3	3.25
20	26.9	3.25
25	33.7	4.05
32	42.4	4.05

4.5.2.1 Coil 25mmNB

For coil 25mmNB, we have outer diameter of 33.7 mm and wall thickness of 4.05 mm. Comparing the surface area bare coil of 25mmNB coil with 32mmNB coil, we get,

$$L_2 = \frac{\pi \times d_1 \times L_1}{\pi \times d_2} \tag{4.2}$$

$$L_2 = \frac{3.14 \times 0.0424 \times 1000}{3.14 \times 0.0337} = 1259\, m$$

Where,

d_1= Outer diameter of coil of conventional system

d_2= Outer diameter of coil of improved system

L_1= Length of coil of conventional system

L_2= Length of coil of improved system

If latent heat storage capacity of both the coils are compared, the ice thickness required by the improved coil system to meet the load can be calculated.

$$Latent\, Heat\, Stored = \frac{\pi}{4}[(D_1)^2 - (d_1)^2] \times L_1 \times \rho_1 = \frac{\pi}{4}[(D_2)^2 - (d_2)^2] \times L_2 \times \rho_2$$

$$= \frac{\pi}{4}[(0.1694)^2 - (0.0424)^2] \times 1000 \times 999 = \frac{\pi}{4}[(D_2)^2 - (0.0337)^2] \times 1259 \times 999$$

$$D_2 = 0.15\, m$$

Where,

D_1= Total outside diameter of coil of conventional system

D_2= Total outside diameter of coil of improved system

Ice thickness will be,

$$Thickness = \frac{(TotalOutsideDiameter) - (ousideDia.of coilof improvedsystem)}{2}$$

$$Thickness = \frac{(0.15 - 0.0337)}{2} = 0.058m$$

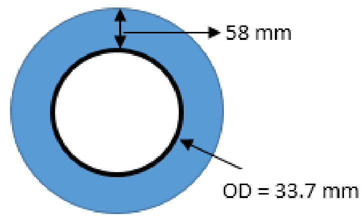

Figure 4.6: Coil of improved system with 25mmNB coil

Now, using heat transfer analysis the temperature of the refrigerant can be found. Calculations are performed using shah's correlation. Trial and error method will be used to find the refrigerant temperature[16].

The tank area will change as 1259 m coil require larger area. New tank area will be 42 m^2. Water outlet temperature and the inlet temperature of 9 ^{o}C and 1 ^{o}C respectively, calculations are performed as,

Taking refrigerant temperature as - 10 ^{o}C.

- Operating Conditions of the system are:

$$Swept\, volume\, at\, 1000\, rpm = 530.8\, m^3/hr$$

$$Refrigeration\, capacity,\, Q_e = 278.84$$

$$Evaporator\, temperature,\, T_e = -10°C$$

$$Condensor\, temperature,\, T_c = 40°C$$

$$Power\, consumption\, by\, compressor,\, P_e = 84.6\, kW$$

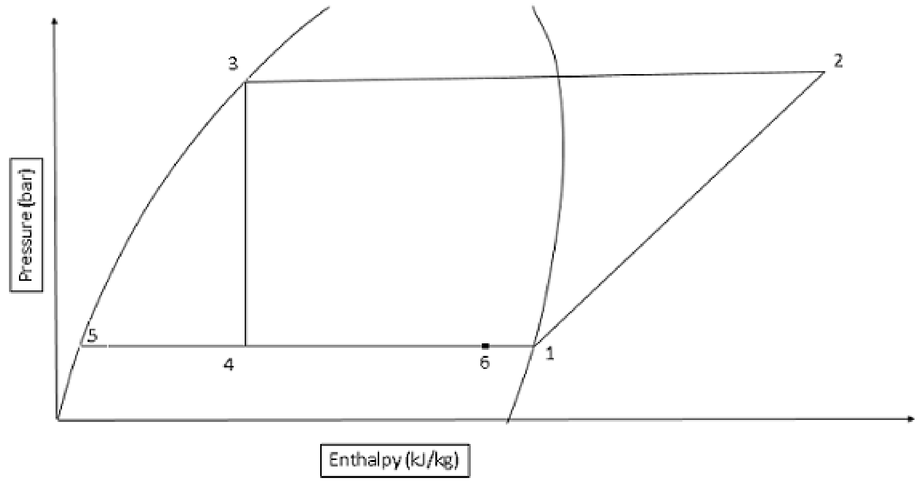

Figure 4.7: Pressure-enthalpy diagram

$$Evaporator\,Pressure\,at\,-10°C = 2.91\,bar$$

$$Condensor\,Pressure\,at\,40°C = 15.57\,bar$$

$$Entahlpy\,at\,point\,1,\,h_1 = 1450.22\,kJ/kg$$

$$Entahlpy\,at\,point\,2,\,h_2 = 1705\,kJ/kg$$

$$Entahlpy\,at\,point\,3,\,h_3 = 390.58\,kJ/kg$$

$$Entahlpy\,at\,point\,4,\,h_4 = 390.58\,kJ/kg$$
$$Entahlpy\,at\,point\,5,\,h_4 = 154.05\,kJ/kg$$

Quality of vapour at point 4 can be calculated from equation:

$$h_4 = (1 - x_4 h_{f_e} + x_4.h_{g_e}) \tag{4.3}$$
$$390.58 = 154.05 \times (1 - x_4) + x_4 \times 1450.22$$
$$x_4 = 0.18$$

- **External side heat transfer coefficint**

Properties of water at bulk mean temperature T$_b$ = 5ºC are

$$\rho = 999.9\,kg/m^3$$
$$k = 0.571\,W/m.K$$
$$Pr = 11.22$$
$$C_p = 4.18\,kJ/kg.K$$
$$\mu_l = 1.51 \times 10^{-3} kg/m.s$$

$$\mu_v = 0.93 \times 10^{-5} kg/m.s$$
$$\beta = 0.015 \times 10^{-3}$$
$$v = 0.5\, m/sec$$

Hydraullic Diameter,

$$d_h = 4 \times \frac{Net\, free\, flow\, area}{Wetted\, perimeter} \tag{4.4}$$

$$d_h = 4 \times \frac{[(4.2 \times 2.5) - \frac{\pi}{4}(0.1497)^2]}{[(2 \times 4.2) + (2 \times 2.5) + (\pi \times 0.1497)]}$$
$$d_h = 3.02\, m$$

Reynolds Number,

$$Re = \frac{\rho v d_h}{\mu} \tag{4.5}$$

$$Re = \frac{999.9 \times 0.5 \times 3.02}{1.51 \times 10^{-3}}$$
$$Re = 1000904$$

Length of Coil is 1259 meter and Total Outside Diameter, d_o= 0.0453 meter, so volume occupied by coil in the tank is,

$$V_c = \left(\frac{\pi}{4} d_o^2\right) \times L \tag{4.6}$$

$$V_c = \frac{\pi}{4} \times (0.1497)^2 \times 1259$$
$$V_c = 22.14\, m^3$$

Volume of tank,

$$V_t = L \times W \times H \tag{4.7}$$

$$V_t = 4 \times 4.2 \times 2.5$$
$$V_t = 42\, m^3$$

Volume available for flow,

$$V_f = 42 - 22.14 = 19.85\, m^3$$

Equivalent Diameter,

$$D_e = \frac{4 \times V_f}{\pi \times d_o \times L} \tag{4.8}$$

$$D_e = \frac{4 \times 19.85}{\pi \times 0.1497 \times 1259}$$
$$D_e = 0.134\, m$$

For Re > 10000, correlation used is:

$$\frac{h_o D_e}{K} = 0.36\,(Re)^{0.55}\,(Pr)^{\frac{1}{3}}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{4.9}$$

$$\frac{h_o \times 0.134}{0.568} = 0.36(1000904)^{0.55}(11.2)^{\frac{1}{3}}(\frac{1.51\times10^{-3}}{1.99\times10^{-3}})^{0.14}$$

$$h_o = 6530.28\,W/m^2K$$

- **Tube side heat transfer coefficient**

Saturated liquid properties of refrigerant R717 at -10°C are

$$p = 2.91\,bar$$

$$\Delta h_v = 1297000\,J/kg$$

$$h_f = 154.05\,kJ/kg$$

$$h_g = 1450.22\,kJ/kg$$

$$\rho_v = 2.391\,kg/m^3$$

$$\rho_l = 652.1\,kg/m^3$$

$$C_{p_l} = 4564\,J/kg.K$$

$$K_l = 0.5621\,W/m.K$$

$$\mu_v = 9.034\times10^{-6}\,kg/m.s$$

$$\mu_l = 2.117\times10^{-4}\,kg/m.s$$

$$\sigma = 0.02896\,N/m$$

The mass flow rate of refrigerant can be calculated using overall energy balance,

$$Q_e = m_r(h_{out} - h_{in}) \tag{4.10}$$

$$Q_e = 278.84kJ/sec$$

$$278.84 = m_r \times (1450.22 - 154.05)$$

$$m_r = 0.215\,kg/sec$$

Prandtl number can be calculated by:

$$Pr = \frac{\mu_l.c_{pl}}{k_l} \tag{4.11}$$

$$= \frac{(2.117\times10^{-4})\times4564}{0.5621} = 1.71$$

Reynolds number for the liquid phase can be calculated by:

$$Re_{lo} = \frac{G.di}{\mu_l} \tag{4.12}$$

$$G = \frac{m_r}{\frac{\pi}{4}\left(d_i\right)^2} \tag{4.13}$$

$$G = \frac{0.215}{\frac{\pi}{4}(0.0256)^2} = 418.16\, kg/m^2.sec$$

$$Re_{lo} = \frac{4018.16 \times 0.0256}{2.117 \times 10^{-4}} = 50734.45$$

Nusselt number can be calculated using Gnielinski correlation:

$$Nu = \frac{\left(\frac{f}{2}\right)(Re_{lo} - 1000)Pr_l}{1 + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr_l^{\frac{2}{3}} - 1\right)} \tag{4.14}$$

where

$$f = (1.58 ln\, Re_l - 3.28)^{-2} \tag{4.15}$$

$$f = (1.58 ln(48155) - 3.28)^{-2}$$

$$f = 5.2 \times 10^{-3}$$

$$Nu = \frac{\left(\frac{5.2\times10^{-3}}{2}\right)(48155 - 1000)(1.71)}{1 + 12.7\left(\frac{5.2\times10^{-3}}{2}\right)^{\frac{1}{2}}\left(1.71^{\frac{2}{3}} - 1\right)} = 174.23$$

Heat transfer coefficient is calculated by

$$h_{lo} = \frac{Nu_{lo}.k_l}{d_i} \tag{4.16}$$

$$= \frac{174.23 \times 0.5621}{0.0256}$$

$$= 3825.63\, W/m^2k$$

By using the shah method, whether the effect of stratification is important or not. For this Froude number is calculated,

$$Fr_{lo} = \frac{G^2}{\rho_l^2 g d_i} \tag{4.17}$$

$$Fr_{lo} = \frac{(418.16)^2}{(652.1)^2 \times 9.81 \times 0.0256} = 1.63$$

Fr>0.04, therefore stratification effects are negligible

Shah's method at x = 0.01

Convection number is calculated by

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} K_{FR} \tag{4.18}$$

$$Co = \left(\frac{1-0.01}{0.01}\right)^{0.8} \left(\frac{2.391}{652.1}\right)^{0.5} \times 1 = 2.391$$

Convection boiling factor

$$F_{cb} = 1.0 + 0.8\, exp[1-(Co)^{0.5}] \quad for\, Co > 1.0 \tag{4.19}$$

$$F_{cb} = 1.463$$

Enhancement factor

$$F_o = F_{cb}(1-x)^x \tag{4.20}$$

$$F_o = 6.49(1-0.18)^{0.8} = 1.463$$

Covective boiling heat transfer coefficient

$$h_{cb} = F_o.(h_{lo}) \tag{4.21}$$

$$h_{cb} = 1.463 \times 4509.88 = 5598\, W/m^2k$$

Overall heat transfer coefficient with neglgible wall resisitance and fouling:

$$U = \left(\frac{1}{h_{cb}} + \frac{1}{h_o}\right)^{-1} \tag{4.22}$$

$$U = \left(\frac{1}{5598} + \frac{1}{6530.28}\right)^{-1} = 3014.09\, W/m^2k$$

Heat flux,

$$q" = U \times (T_H - T_S) \tag{4.23}$$

$$q" = 3014.09 \times (5-(-10)) = 45211.31\, W/m^2k$$

The wall temperature for the onset of nucleate boiling can be determined,

$$T_{WONB} = \left(\frac{8\sigma q" T_s}{k_l \triangle h_v \rho_v}\right)^{0.5} + T_s \tag{4.24}$$

$$T_{WONB} = \left(\frac{8 \times 0.02896 \times 45211.31 \times 263}{0.5621 \times 1297000 \times 2.391}\right)^{0.5} + 263 = 264.25K$$

The wall temperature can be calculated,

$$T_w = \frac{q"}{h_{cb}} + T_s \tag{4.25}$$

$$T_w = \frac{45211.31}{5598} + 263 = 271.07\,K$$

$T_w >$ T_{WONB} and nucleate boiling is present,

Boiling number is calculated as,

$$Bo = \frac{q"}{G.\triangle h_v} \tag{4.26}$$

$$Bo = \frac{45211.31}{418.16 \times 1297000} = 8.341 \times 10^{-5}$$

This is the approximate value of the boiling number. Value of Bo $>$ 1.9 $\times 10^{-5}$, therefore nucleate boiling is not negligible.

For, combined nucleate and convective boiling effect, enhancement factor can be determined from:

$$F_{nb} = 231(Bo)^{0.5} \tag{4.27}$$

$$F_{nb} = 231(8.341 \times 10^{-5})^{0.5}$$

$$F_{nb} = 2.11$$

So, boiling heat transfer coefficient is,

$$h_{vo} = F(h_{lo}) \tag{4.28}$$

$$h_{vo} = 2.11 \times (3825.63) = 8071.2\,W/m^2k$$

The overall thermal resistance,

$$R = \frac{r}{r_i}\left(\frac{1}{h_i}\right) + \frac{r}{k_l} ln\left(\frac{r_o}{r_i}\right) + \frac{r}{k} ln\left(\frac{r}{r_i}\right) + \frac{1}{h_o} \tag{4.29}$$

$$R = \frac{0.07485}{0.0128}\left(\frac{1}{8301.61}\right) + \frac{0.07485}{53.6} ln\left(\frac{0.01685}{0.0128}\right) + \frac{0.07485}{2.27} ln\left(\frac{0.007485}{0.0128}\right) + \frac{1}{7566.86}$$

$$R = 0.0504\,m^2.k/W$$

The overall heat transfer coefficient,

$$U = 19.83\,W/m^2.k$$

Similarly many more iteration are performed taking different evaporator temperature between -5°C and -15°C, to find the overall heat transfer coefficient which will be taken as an average. By using trial and error method and refrigeration capacity, evaporator temperature can be found. Iterations will be performed using Microsoft Excel and the results obtained are shown in the Table4.6.

Table 4.6: Results of iterative calculation performed at different ice thickness

Assumed Evaporator Temperature	Overall Heat Transfer Co-eff.,Uo(W/m².K)	Refrigeration Capacity, Q_e(kW)
-5	19.85	354.65
-6	19.85	339.49
-7	19.84	324.49
-8	19.84	309.17
-9	19.83	294.01
-10	19.83	278.84
-11	19.82	265.22
-12	19.82	251.16
-13	19.81	237.98
-14	19.81	224.35
-15	19.80	210.7

The average overall heat transfer coefficient is 19.83 W/m².K.

By performing iterations and using trial and error method, the evaporator temperature found is -14.48 °C. The other values will be:

$$Coil\,Type = 25mmNB$$

$$Length\,of\,Coil,\; L = 1259\,m$$

$$Logarithmic\,mean\,temperature\,difference = 18.32°C$$

$$Total\,Outside\,diameter\,after\,building\,of\,ice = 0.1497\,m$$

$$Surface\,Area,\; A = \pi\times d\times l = 591.80\,m^2$$

$$Evaporator\,Temperature, T_e = -14.48°C$$

The Refrigeration capacity from KC4 chart of Kirloskar Compressors will be,

$$Refrigeration\,Capacity, Q_e = 217.4\,kW$$

$$Power\,Consumption\,by\,Compressor, P_e = 75.96\,kW$$

$$Specific\,Power\,Consumption,\; SPC = 0.34$$

$$Latent\,Heat\,Storage,\; LHS = \frac{\pi}{4}[(0.1497)^2 - (0.0337)^2]\times 1259\times 999$$

$$= 21094.3\,kg\,of\,ice$$

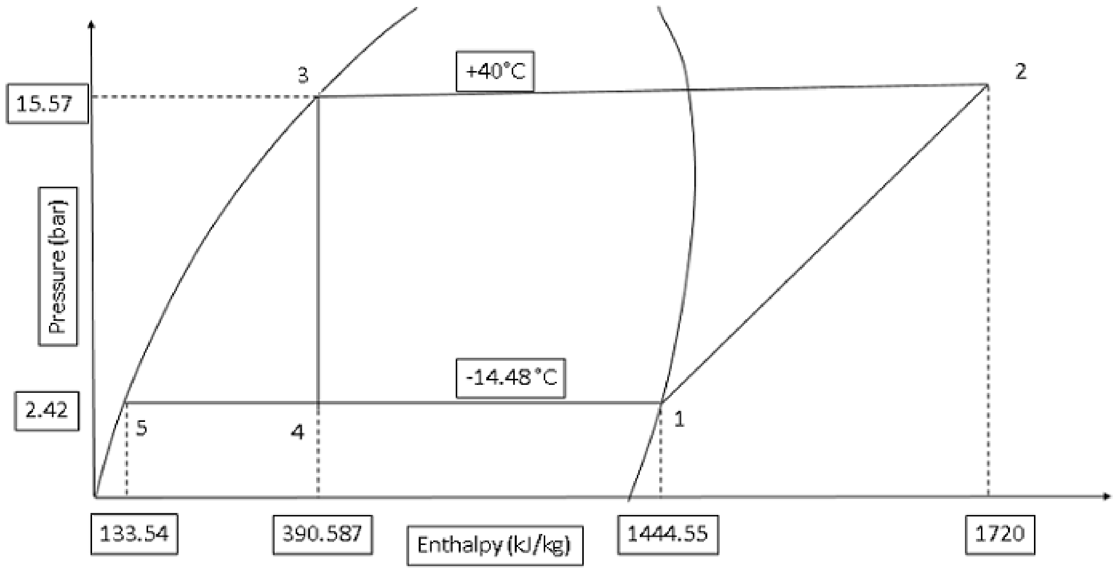

Figure 4.8: Pressure-enthalpy diagram

$$Refrigeration\,effect = Q_1 = (Enthalpy\,at\,point\,1) - (Enthalpy\,at\,point\,5)$$
$$= (h_1 - h_4) = (1444.55 - 133.54)$$
$$= 1311.01\,kJ/kg$$
$$Work\,of\,Compression = W_1 = (Enthalpy\,at\,point\,2) - (Enthalpy\,at\,point\,1)$$
$$= (h_2 - h_1) = (1720 - 1444.55)$$
$$= 275.45\,kJ/kg$$
$$Coefficient\,of\,Performance = \frac{(h_1 - h_4)}{(h_2 - h_1)} = 4.75$$

4.5.2.2 20mmNB Coil

Similarly for 20mmNB Coil following results are obtained:

$$Coil\,Type = 20mmNB$$
$$Length\,of\,Coil, L = 1577\,m$$
$$New\,Area\,of\,Tank = 46\,m^2$$
$$Evaporator\,Temperature, T_e = -12.38°C$$
$$Thickness\,of\,ice = 53\,mm$$
$$Overall\,Heat\,Transfer\,Coefficient = 23.56\,W/m^2.k$$
$$Logarithmic\,mean\,temperature\,difference = 16.19°C$$
$$Total\,Outside\,diameter\,after\,building\,of\,ice = 0.1329\,m$$

$$Surface\,Area,\; A = \pi \times d \times l = 658.0916\, m^2$$

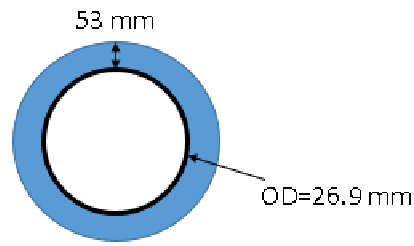

Figure 4.9: Coil of improved system with 25mmNB coil

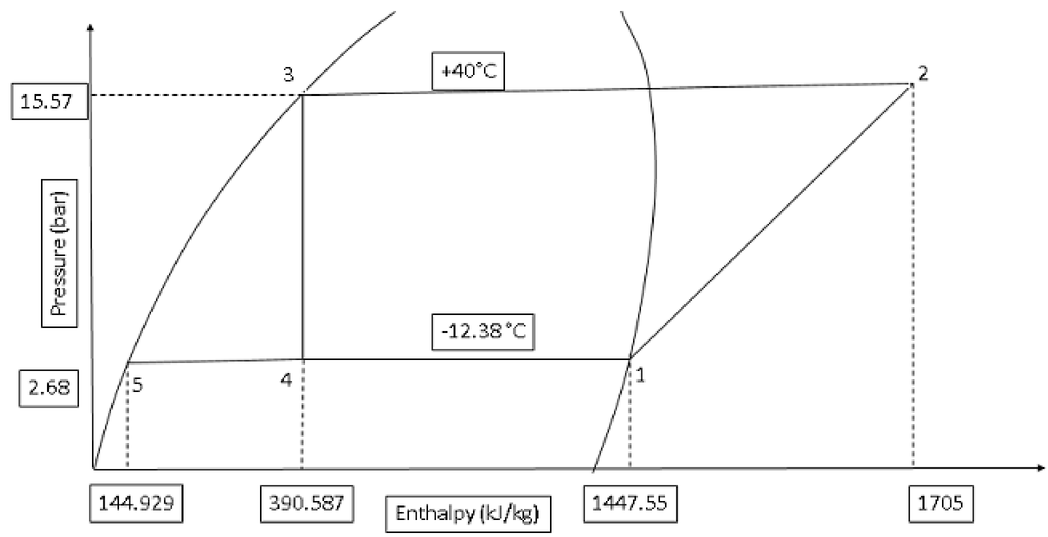

Figure 4.10: p-h diagram pressure-enthalpy diagram

From Pressure-Enthalpy Chart 4.10, we can find,

$$Refrigeration\,effect = Q_1 = (Enthalpy\,at\,point\,1) - (Enthalpy\,at\,point\,4)$$
$$= (h_1 - h_4) = (1447.55 - 144.929)$$
$$= 1302.62\,kJ/kg$$
$$Work\,of\,Compression = W_1 = (Enthalpy\,at\,point\,2) - (Enthalpy\,at\,point\,1)$$
$$= (h_2 - h_1) = (1705 - 1447.55)$$
$$= 257.45\,kJ/kg$$
$$Coefficient\,of\,Performance = \frac{(h_1 - h_4)}{(h_2 - h_1)} = 5.05$$

4.6 Experimental Verification

Experimental verification for ice based latent heat storage system is done through industrial application in Samruddh Dairy Products Pvt. Ltd, Ridrol. The system installed has following operating parameter:

1. The area of the tank used is 30 m^2 .

2. Total length of the coil is 600 m.

3. Compressor Model is of kirloskar KC2 series which is a single stage compressor with 2 cylinders

4. Operating conditions are -15°C/+40°C, 0°C Subcooling, 0°C Superheating and 1000 rpm.

5. Flooded type evaporator system is used.

6. Refrigerant used is ammonia (R717).

7. Coil is of 32mmNB type with outside diameter of 42.4mm and wall thickness of 4.05 mm.

4.6.1 Experimental setup

Figure 4.11: Ice based latent heat storage system, compressor, condenser and pressure gauges

4.6.2 Instruments required

- Outside calliper
- Scale

4.6.3 Experimental procedure

1. After the completion of the milk chilling process the compressor will be kept off and a reading will be taken for the minimum ice thickness .
2. The compressor will be started and the charging will be started.
3. At increment of every 5 mm ice thickness, reading of the following parameters will be taken:
    - Suction pressure (bar)
    - Discharge pressure (bar)
    - Ice thickness
4. Overall heat transfer coefficient, refrigeration capacity and specific power consumption will be found for each reading.
5. Graphs will be formed between thickness vs. evaporator temperature, specific power consumption vs. evaporator temperature, specific power consumption vs. evaporator temperature.
6. Optimum ice thickness will be calculated according to the electricity tariff rates.

Chapter 5

Results and Discussion

5.1 Optimization of the System

Results obtained by performing iterations for different evaporator temperature between -5°C and -15°C, to find the overall heat transfer coefficient which will be taken as an average. By using trial and error method and refrigeration capacity, evaporator temperature can be found. Iterations are performed using Microsoft Excel and the results obtained are shown in the Table5.1.

Table 5.1: Results of the iterations performed at different ice thickness and evaporator temperature

Thickness of ice (mm)	**Area (m^2)**	**Evaporator Temperature (°C)**	**Overall Heat Transfer Coefficient ($W/m^2.k$)**	**Refrigeration Capacity, Q_e (kW)**	**Power Consumption by Compressor, P_e (kW)**	**Specific Power Consumption, P_e/Q_e**
0	133.13	-4.01	593.55	372.06	94.62	0.243
5	164.53	-5.065	253.11	363.35	92	0.259
10	195.93	-7.42	148.40	324.49	89.2	0.275
15	227.33	-9.05	100.89	294.01	86.85	0.295
20	258.73	-10.67	74.48	278.84	84.6	0.303
25	290.13	-11.53	58.03	258.19	82.2	0.310
30	321.53	-12.38	46.95	244.57	80.66	0.321
35	357.96	-13.17	39.12	237.98	79.3	0.333
40	384.33	-13.81	33.28	231.17	77.88	0.337
45	415.73	-14.25	28.78	224.36	76.82	0.342
50	447.13	-14.6	25.28	215.7	76.96	0.352
63.5	536.94	-14.99	18.70	210.7	75	0.355

Power Consumption by the compressor is give in the KC 4 Compressor chart given in Appendix B. Accordingly specific power consumption can be calculated which is defined as the ratio of power consumption by compressor and refrigeration capacity.

$$Specific\,power\,consumption = \frac{Power\,consumption\,by\,compressor}{Refrigeration\,capacity} \tag{5.1}$$

Different graphs obtained from these results are give below,

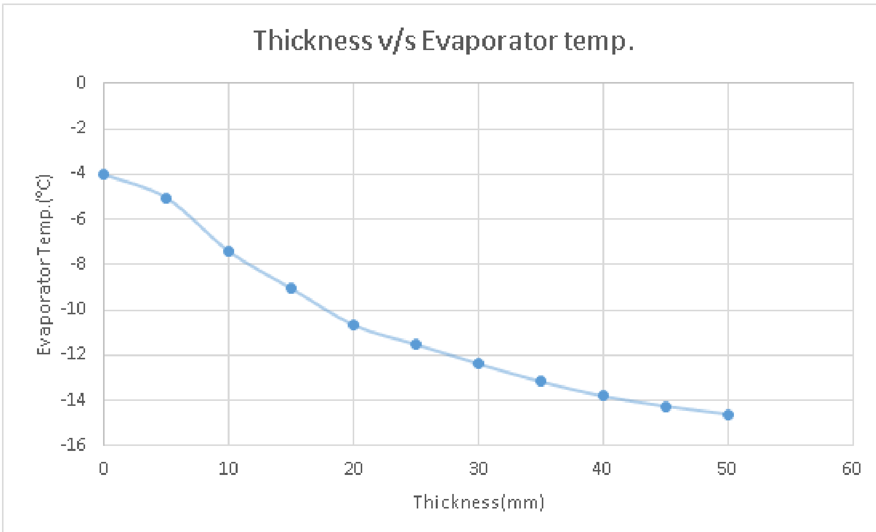

Figure 5.1: Thickness v/s Evaporator Temperature

Figure5.1 shows as the evaporator temperature inside the tube keeps on decreasing the ice thickness around the tube keeps on increasing. The evaporator temperature for ice thickness up to 50 mm is calculated here.

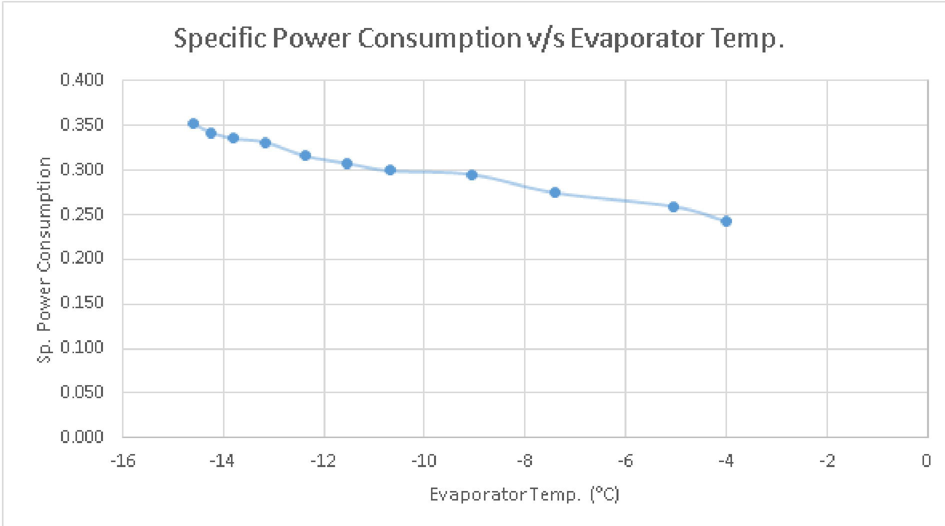

Figure 5.2: Specific Power Consumption v/s Evaporator Temp.

Figure 5.2 shows specific power consumption increases with the decrease in evaporator temperature.

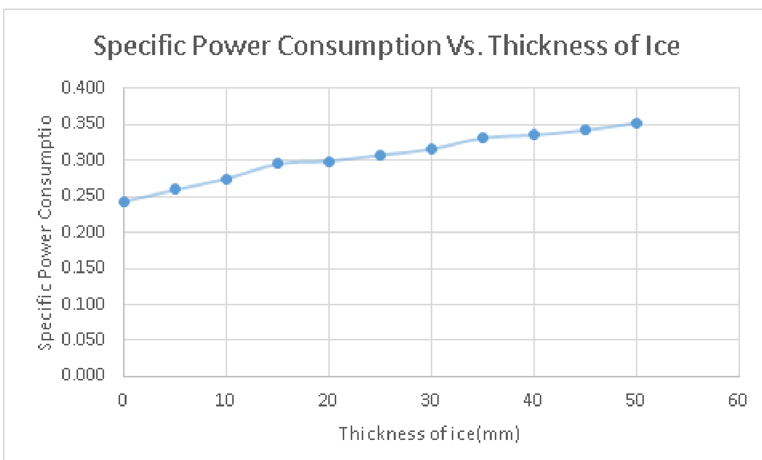

Figure 5.3: Specific Power Consumption Vs. Thickness of Ice

Figure 5.3 shows increase in specific power consumption as ice thickness increases

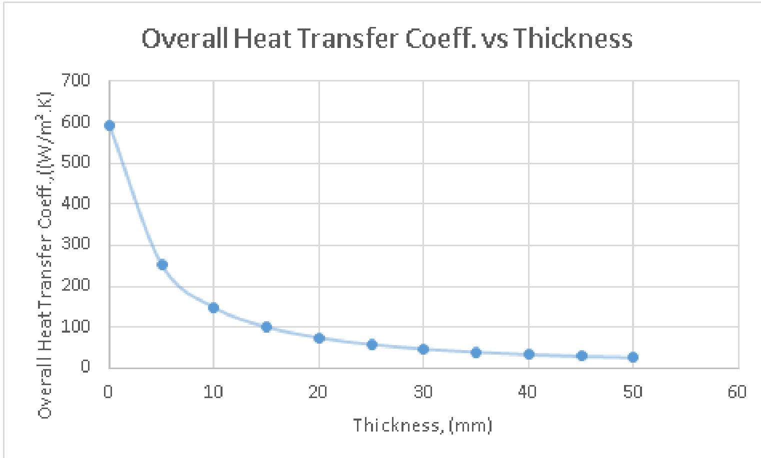

Figure 5.4: Overall Heat Transfer Coeff. vs Thickness

If refrigeration system is used to build ice around the tube during the off-peak hours, there is almost a saving of 30.5 % which can be seen in the tariff table 4.4. By using this amount of saving we built ice during on=peak hours without any extra cost for refrigeration. So the thickness of ice that we can achieved using this 30.5 % of saving in compressor power consumption will be the optimum ice thickness.

From Figure 5.1 and Figure 5.2, the optimum ice thickness achieved is 30 mm ice around the coil.

5.2 Efficient Design

The result obtained for using different diameter coil rather than using 32mmNB coil are shown belo in Table 5.2.

Table 5.2: Standard Available Coils

Nominal Bore (mm)	Compressor Work (kW)	Coefficient of Perfor- mance	Refrigeration Effect (kJ/kg)
20	257.45	5.05	1302.62
25	275.45	4.75	1311.01
32	291.1	4.50	1312.63

There is a considerable decrement in the compressor work and an increment in coefficient of performance by using different diameter coils. These are shown below in Table 5.3.

Table 5.3: Saving achieved by using different coils

Nominal Bore (mm)	Decrease in Compressor Work (kW)	Increase in Coefficient of Perfro-mance
20	11.55 %	12.22 %
25	5.37 %	5.55%

5.3 Experimental Verification

Readings are taken at increment of every 5 mm ice thickness shown below in Table 5.4

Table 5.4: Data Sheet of experiment

S. No.	**Total Outside Diameter (mm)**	**Suction Pressure (bar)**	**Discharge Pressure (bar)**
1.	70	2.57	15.2
2.	75	2.57	14.8
3.	80	2.47	14.5
4.	85	2.47	14.2
5.	90	2.37	13.8
6.	95	2.37	13.6
7.	100	2.37	13.4
8.	105	2.27	13.3
9.	110	2.27	13.1
10.	115	2.18	12.8
11.	120	2.18	12.7
12.	125	2.18	12.5
13.	130	2.08	12.3
14.	135	2.08	12.3
15.	140	2.08	12.1
16.	143	2	12

Results obtained after heat transfer calculation are given in Table.5.5and Table 5.6.

Table 5.5: Heat transfer calculations

Thickness of ice (mm)	**Total Outside Diameter (mm)**	**Area (m^2)**	**Evaporator Temperature, T_e (°C)**	**Condenser Temperature, T_c (°C)**	**Overall Heat Transfer Coefficient ($W/m^2.k$)**
3.8	50	94.2	-8	40	278.92
6.3	55	103.6	-8.6	40	208.19
8.8	60	113.0	-8.6	40	162.73
11.3	65	122.4	-8.6	40	131.64
13.8	70	131.8	-9.5	40	109.32
16.3	75	141.3	-10.6	39.9	92.7
18.8	80	150.7	-10.6	38.5	79.93
21.3	85	160.1	-11.2	37.1	69.88
23.8	90	169.5	-11.2	36.1	61.8
26.3	95	178.9	-11.9	36.6	55.19
28.8	100	188.4	-11.9	35.2	49.71
31.3	105	197.8	-13.1	35.5	45.09
33.8	110	207.2	-13.1	34	41.17
36.3	115	216.6	-13.9	33.5	37.8
38.8	120	226.0	-13.9	33	34.87
41.3	125	235.5	-13.9	33.7	32.32
43.8	130	244.9	-14.8	33.9	30.07
46.3	135	254.3	-14.8	33.9	28.08
48.8	140	263.7	-14.8	31.6	26.31
50.3	143	269.4	-14.8	31	25.34

Table 5.6: Results of the iterations performed at different ice thickness and evaporator temperature

Evaporator Temperature, T_e(°C)	**Refrigeration Capacity, Q_e (kW)**	**Power Consumption by Compressor, P_e (kW)**	**Specific Power Consumption, P_e /Q_e**
-8	157.1	42.79	0.272
-8.6	150.4	43.16	0.287
-8.6	150.4	43.16	0.287
-8.6	150.4	43.16	0.287
-9.5	143.21	43.9	0.307
-10.6	135.96	42.83	0.315
-10.6	135.96	42.83	0.315
-11.2	130.83	42.1	0.322
-11.2	130.83	42.1	0.322
-11.9	125.69	41.42	0.330
-11.9	125.69	41.42	0.330
-13.1	118.82	40.48	0.341
-13.1	118.82	40.48	0.341
-13.9	111.96	39.54	0.353
-13.9	111.96	39.54	0.353
-13.9	106.81	38.85	0.364
-14.8	106.81	38.85	0.364
-14.8	106.81	38.85	0.364
-14.8	106.81	38.85	0.364
-14.8	106.81	38.85	0.364

Graph obtained from Table 5.5and Table5.6 are shown below. Figure5.5 shows as the evaporator temperature inside the tube keeps on decreasing the ice thickness around the tube keeps on increasing. The evaporator temperature for ice thickness up to 50.3 mm is calculated here. Figure 5.6 shows specific power consumption increases with the decrease in evaporator temperature.

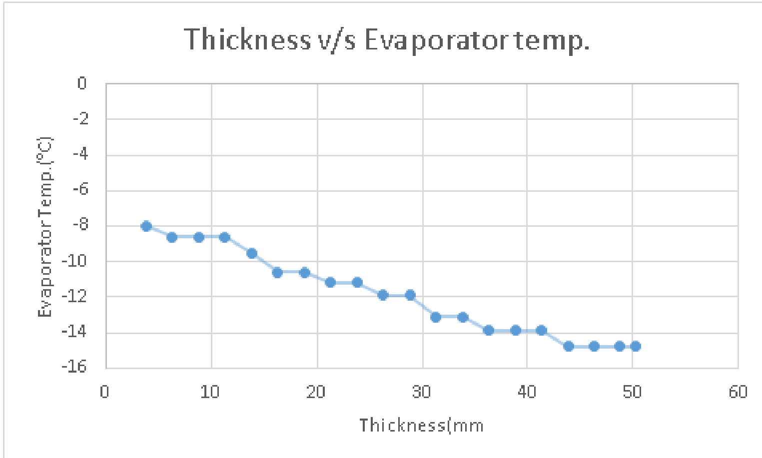

Figure 5.5: Thickness v/s Evaporator Temperature

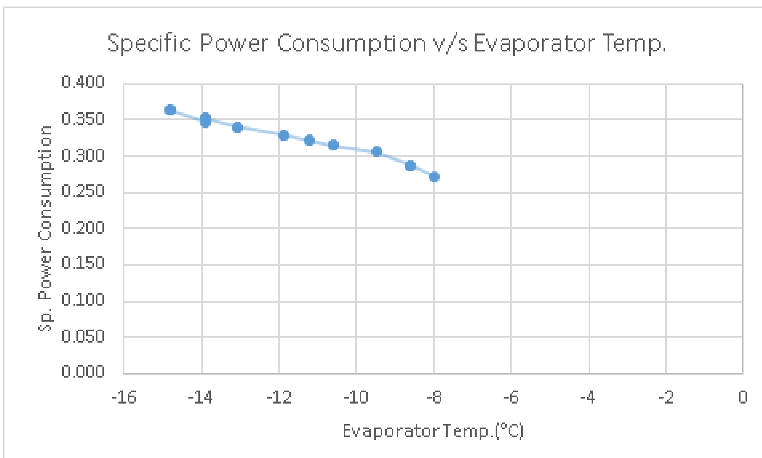

Figure 5.6: Specific Power Consumption v/s Evaporator Temp.

Refrigeration system is used to build ice around the tube during the off-peak hours, there is almost a saving of 30.5 % which can be seen in the tariff table 4.4. By using this amount of saving we build ice without any extra cost for refrigeration during on-peak hours. So the thickness of ice that we can achieved using this 30.5 % of saving in compressor power consumption will be the optimum ice thickness. The minimum ice left in the system is of thickness 3.8 mm.

From Figure 5.5 and Figure 5.6, the optimum ice thickness achieved is 36 mm ice around the coil.

Chapter 6

Conclusion and Future Scope

6.1 Conclusion

- Use of Ice based Latent Heat Storage System during off-peak hours leads to a certain amount of saving economically. The saving can be used to run the compressor during on-peak hours when load is more without any extra expense.
- The optimum ice thickness achieved that can be built during on-peak hours is 30 mm.
- The efficient design proposed, decreases work of compression upto 11.55% and increases coefficient of performance upto 12.22 %.
- Experimental verificaton is done for the theoritical analysis performed on ice based latent heat storage system.

6.2 Future Scope

- CFD analysis of the ice based latent heat storage system
- Usage of plate type heat exchanger with ice based latent heat storage system may be investigated.

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Appendix A

Properties of Milk and its derivtives is given in Table 1 and Table 2.

Table 1: Density of standard dairy product

Product	Density (kg/L)			
	4 °C	10°C	20°C	38.9°C
Raw Milk	1.035	1.033	1.030	1.023
Homogenized Milk	1.033	1.032	1.029	1.022
Skim Milk	1.036	1.035	1.033	1.026
Light Cream	1.021	1.018	1.012	1
Heavy Cream	1.008	1.005	0.994	0.978

Table 2: Specific heat of standard dairy product

Product	Temperature			
	0°C	**15°C**	**40°C**	**60°C**
Skim Milk	3.936	3.948	3.986	4.932
Raw Milk	3.852	3.927	3.984	3.844
15% Cream	3.140	3.864	3.764	3.768
20% Cream	3.027	3.936	3.684	3.710
30 % Cream	2.818	4.116	3.567	3.601
45% Cream	2.537	4.254	3.295	3.320
60% Cream	2.345	4.409	3.019	3.086

Appendix B

Technical Data for Single Stage Compressors is given in Table 3.

Table 3: Technical Data for Single Stage Compressors

Compressor Model		KC2	KC3	KC4	KC6	KC9	KC12
Cylinder Arrangement		1XV	1XW	2XV	2XW	3XW	4XW
Number of Cylinders		2	3	4	6	9	12
Cylinder Bore	mm	160	160	160	160	160	160
Piston Stroke	mm	110	110	110	110	110	110
Swept Volume at 1000 rpm	m^3/hr	265.4	398.1	530.8	796.2	1194.3	1592.4
Maximum Discharge Pressure	bar	21	21	21	21	21	21
Oil Charge Capacity	L	9	10	12	13	20	23
Weight of Compressor (Without Flywheel)	kg	435	535	665	900	1245	1585
Moment of Inertia GD2 of crank mechanism	$kg.m^2$	0.356	0.422	0.446	0.594	0.829	1.045
Ice (block) Making Capacity* with Ammonia (Tonnes/day)	TPD	19.5	29.3	39	58.5	87.8	117.1
Power Consumption	kW	40.5	59.4	78.5	116.2	172.9	229.5

Appendix C

Table 4: Rating Chart for Single Stage Compressors

Compressor Model	Refrigerant	NH_3			
	Evaporating Temp. in °C	Rating for Single Stage Comp. in °C			
		35°C		40°C	
		Q_o(kW)	P_e(kW)	Q_o(kW)	P_e(kW)
KC2/KCX2	5	283.6	46.3	271.86	51.8
	0	230.81	45.1	220.81	49.8
	-5	186.4	43.1	177.33	47
	-10	147.67	40.4	139.42	43.3
	-15	114.53	36.3	105.1	38.6
	-20	89.5	31.8	-	-
KC4/KCX4	5	566.05	90.6	543.72	101.6
	0	461.74	88.2	441.74	97.7
	-5	372.79	84.3	354.65	92.1
	-10	295.35	78.8	278.84	84.6
	-15	229.07	71.5	210.7	75
	-20	177.1	62.1	-	-

Appendix D

Iterative calculations performed in Microsoft Excel 2010

Operating Conditions			
Compressor Type			Single Stage
Speed		RPM	1000
Swept Volume at 1000 RPM		m3/hr	530.8
Refrigeration Capacity	Qe	kW	278.84
Evaporator Temperature	Te	°C	-10
Condensor Tempeature	Tc	°C	40
Evaporator Pressure	Pe	bar	2.91
Condenser Pressure	Pc	bar	15.57

TANK SIDE INPUT DATA			
Properties of water at bulk mean temperature of 5 °C			
Fluid			Water
Density of Fluid	ρ_s	kg /m^3	999.9
Thermal Conductivity	Ks	W/mk	0.571
Dynamic Viscocity (liquid)	μl	Kg/m.s	0.00151
Prandtl number	Pr		11.2
Specific Heat of Fluid	Cps	kJ/kg K	4.2
Velocity of water	V	m/s	0.5
Inlet Temperature	Thi	°C	9
Output Temperature	Tho	°C	1
Dynamic Viscocity of fluid at wall temp.	μ_w	Kg/m.s	0.00199

TANK SIDE CALCULATION			
Hydraullic Diameter	D_h	m	2.9392261
Reynolds Number	Re		973156
Volume occupied by coils	Vc	m^3	15.9180416
Volume of Tank	Vt	m^3	39
Volume available for flow	Va	m^3	23.0819584
Equivalent Diameter	De	m	0.20648714
Heat Transfer Coefficient	ho	W/m^2.K	4178.35

Appendix E

Tube Length (meter)	
L	1000

Conductivity	
Material	Conductivity (W/mK)
32mmNB	53.6
Ice	2.27

Tube Dimensions(meter)	
id	0.0343
od	0.0424
thickness	0.00405

Tank Dimensions (meter)	
Width	3.9
Height	2.5
Length	4

Thickness of Ice On Coil	
Thickness	0.05
Total O.D	0.1424

TUBE SIDE INPUT DATA			
Inlet Pressure	Pt	bar	2.91
Fluid			R717
Inlet Temperature	Tci	°C	-10
Outlet Temperature	Tco	°C	-10
Quality of vapour at outlet	x		0.00
Enthalpy of saturated liquid	hf	KJ/kg	154.06
Enthalpy of saturated vapour	hg	KJ/kg	1450.22
Entahlpy at evaporator inlet	hi	KJ/kg	154.06
Mass Flow Rate	mr	kg/s	0.215127

Appendix F

TUBE SIDE CALCULATION			
Properties of R717 at -10°C			
Density of liquid	ρ_l	kg/m^3	652.1
Density of vapour	ρ_v	kg/m^3	2.391
Specific heat of liquid	C_{pl}	J/Kg.K	4564
Thermal Conductivity	K_l	W/m.K	0.5621
Surface Tension	σ	N/m	0.02896
Dynamic Viscosity Liquid	μ_l	Kg/m.s	0.000211
Dynamic Viscosity Vapour	μ_v	Kg/m.s	0.00000903
Enthalpy of Vapourization	Δh_v	J/Kg.K	1296164
Calculated Values			
Prandtl Number liquid only	Pr		1.713225405
Mass Velocity	G	$Kg/m^2.s$	232.9363265
Reynold Number Liquid only	Re_{lo}		37865.9526
Friction Factor	f		0.005589125
Nusselt Number	Nu		137.2032688
Heat Transfer Coefficient liquid only	h_{lo}	$W/m^2.K$	2248.453567
Froude Number	Fr_{lo}		0.379212737
Correction Factor	K_{fr}		1
If Froude number, $Fr_{lo} < 0.04$			
Correction Factor	K_{fr}		0.509275897
From shah's method			
Convection number	Co		2.3913
Convective boiling factor	Fcb		1.463
enhancement factor	Fo		1.463226799
Convective boiling heat transfer coeff.	hcb	$W/m^2.K$	3290
Overall heat transfer coeff.	U	$W/m^2.K$	1840.67
heat flux	q"	W/m^2	27610.06
Onset of nucleate boiling	T_{WONB}	K	263.9827188
Wall Temperature	Tw	K	271.3921219
For x=0.17,Tw>T_{WONB} and therefore nucleate boiling is present			
Boiling number	Bo		0.00009145
Bo>0.000019, nucleate boiling enhancement should be considered			
Nucleate boiling factor	Fnb		2.209
Combined nucleate& conve. Boiling factor	Fcnb		2.12
nucleate Boiling heat transfer coeff.	hnb	$W/m^2.K$	4966.8
two phase heat transfer coeff.	htp	$W/m^2.K$	8071

Final Calculations					
Overall Heat Transfer Coefficient		Uo	W/m².K	25.62
Area		A	m^2	447.14

Appendix G

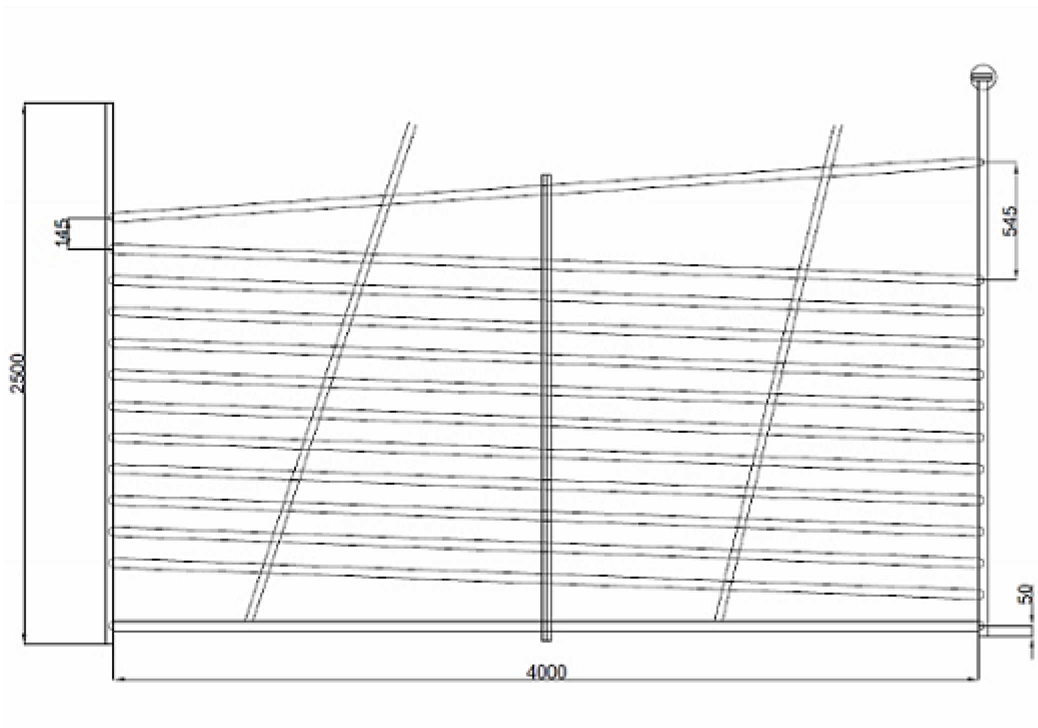
Property of saturated water and saturated ammonia

Temp. T, °C	Saturation Pressure P_{sat}, kPa	Density ρ, kg/m³ Liquid	Density ρ, kg/m³ Vapor	Enthalpy of Vaporization h_{fg}, kJ/kg	Specific Heat c_p, J/kg·K Liquid	Specific Heat c_p, J/kg·K Vapor	Thermal Conductivity k, W/m·K Liquid	Thermal Conductivity k, W/m·K Vapor	Dynamic Viscosity μ, kg/m·s Liquid	Dynamic Viscosity μ, kg/m·s Vapor	Prandtl Number Pr Liquid	Prandtl Number Pr Vapor	Volume Expansion Coefficient β, 1/K Liquid
0.01	0.6113	999.8	0.0048	2501	4217	1854	0.561	0.0171	1.792×10^{-3}	0.922×10^{-5}	13.5	1.00	-0.068×10^{-3}
5	0.8721	999.9	0.0068	2490	4205	1857	0.571	0.0173	1.519×10^{-3}	0.934×10^{-5}	11.2	1.00	0.015×10^{-3}
10	1.2276	999.7	0.0094	2478	4194	1862	0.580	0.0176	1.307×10^{-3}	0.946×10^{-5}	9.45	1.00	0.733×10^{-3}
15	1.7051	999.1	0.0128	2466	4185	1863	0.589	0.0179	1.138×10^{-3}	0.959×10^{-5}	8.09	1.00	0.138×10^{-3}
20	2.339	998.0	0.0173	2454	4182	1867	0.598	0.0182	1.002×10^{-3}	0.973×10^{-5}	7.01	1.00	0.195×10^{-3}
25	3.169	997.0	0.0231	2442	4180	1870	0.607	0.0186	0.891×10^{-3}	0.987×10^{-5}	6.14	1.00	0.247×10^{-3}
30	4.246	996.0	0.0304	2431	4178	1875	0.615	0.0189	0.798×10^{-3}	1.001×10^{-5}	5.42	1.00	0.294×10^{-3}
35	5.628	994.0	0.0397	2419	4178	1880	0.623	0.0192	0.720×10^{-3}	1.016×10^{-5}	4.83	1.00	0.337×10^{-3}
40	7.384	992.1	0.0512	2407	4179	1885	0.631	0.0196	0.653×10^{-3}	1.031×10^{-5}	4.32	1.00	0.377×10^{-3}
45	9.593	990.1	0.0655	2395	4180	1892	0.637	0.0200	0.596×10^{-3}	1.046×10^{-5}	3.91	1.00	0.415×10^{-3}
50	12.35	988.1	0.0831	2383	4181	1900	0.644	0.0204	0.547×10^{-3}	1.062×10^{-5}	3.55	1.00	0.451×10^{-3}
55	15.76	985.2	0.1045	2371	4183	1908	0.649	0.0208	0.504×10^{-3}	1.077×10^{-5}	3.25	1.00	0.484×10^{-3}
60	19.94	983.3	0.1304	2359	4185	1916	0.654	0.0212	0.467×10^{-3}	1.093×10^{-5}	2.99	1.00	0.517×10^{-3}
65	25.03	980.4	0.1614	2346	4187	1926	0.659	0.0216	0.433×10^{-3}	1.110×10^{-5}	2.75	1.00	0.548×10^{-3}
70	31.19	977.5	0.1983	2334	4190	1936	0.663	0.0221	0.404×10^{-3}	1.126×10^{-5}	2.55	1.00	0.578×10^{-3}
75	38.58	974.7	0.2421	2321	4193	1948	0.667	0.0225	0.378×10^{-3}	1.142×10^{-5}	2.38	1.00	0.607×10^{-3}
80	47.39	971.8	0.2935	2309	4197	1962	0.670	0.0230	0.355×10^{-3}	1.159×10^{-5}	2.22	1.00	0.653×10^{-3}
85	57.83	968.1	0.3536	2296	4201	1977	0.673	0.0235	0.333×10^{-3}	1.176×10^{-5}	2.08	1.00	0.670×10^{-3}
90	70.14	965.3	0.4235	2283	4206	1993	0.675	0.0240	0.315×10^{-3}	1.193×10^{-5}	1.96	1.00	0.702×10^{-3}
95	84.55	961.5	0.5045	2270	4212	2010	0.677	0.0246	0.297×10^{-3}	1.210×10^{-5}	1.85	1.00	0.716×10^{-3}
100	101.33	957.9	0.5978	2257	4217	2029	0.679	0.0251	0.282×10^{-3}	1.227×10^{-5}	1.75	1.00	0.750×10^{-3}
110	143.27	950.6	0.8263	2230	4229	2071	0.682	0.0262	0.255×10^{-3}	1.261×10^{-5}	1.58	1.00	0.798×10^{-3}
120	198.53	943.4	1.121	2203	4244	2120	0.683	0.0275	0.232×10^{-3}	1.296×10^{-5}	1.44	1.00	0.858×10^{-3}
130	270.1	934.6	1.496	2174	4263	2177	0.684	0.0288	0.213×10^{-3}	1.330×10^{-5}	1.33	1.01	0.913×10^{-3}
140	361.3	921.7	1.965	2145	4286	2244	0.683	0.0301	0.197×10^{-3}	1.365×10^{-5}	1.24	1.02	0.970×10^{-3}
150	475.8	916.6	2.546	2114	4311	2314	0.682	0.0316	0.183×10^{-3}	1.399×10^{-5}	1.16	1.02	1.025×10^{-3}
160	617.8	907.4	3.256	2083	4340	2420	0.680	0.0331	0.170×10^{-3}	1.434×10^{-5}	1.09	1.05	1.145×10^{-3}
170	791.7	897.7	4.119	2050	4370	2490	0.677	0.0347	0.160×10^{-3}	1.468×10^{-5}	1.03	1.05	1.178×10^{-3}
180	1,002.1	887.3	5.153	2015	4410	2590	0.673	0.0364	0.150×10^{-3}	1.502×10^{-5}	0.983	1.07	1.210×10^{-3}
190	1,254.4	876.4	6.388	1979	4460	2710	0.669	0.0382	0.142×10^{-3}	1.537×10^{-5}	0.947	1.09	1.280×10^{-3}
200	1,553.8	864.3	7.852	1941	4500	2840	0.663	0.0401	0.134×10^{-3}	1.571×10^{-5}	0.910	1.11	1.350×10^{-3}
220	2,318	840.3	11.60	1859	4610	3110	0.650	0.0442	0.122×10^{-3}	1.641×10^{-5}	0.865	1.15	1.520×10^{-3}
240	3,344	813.7	16.73	1767	4760	3520	0.632	0.0487	0.111×10^{-3}	1.712×10^{-5}	0.836	1.24	1.720×10^{-3}
260	4,688	783.7	23.69	1663	4970	4070	0.609	0.0540	0.102×10^{-3}	1.788×10^{-5}	0.832	1.35	2.000×10^{-3}
280	6,412	750.8	33.15	1544	5280	4835	0.581	0.0605	0.094×10^{-3}	1.870×10^{-5}	0.854	1.49	2.380×10^{-3}
300	8,581	713.8	46.15	1405	5750	5980	0.548	0.0695	0.086×10^{-3}	1.965×10^{-5}	0.902	1.69	2.950×10^{-3}
320	11,274	667.1	64.57	1239	6540	7900	0.509	0.0836	0.078×10^{-3}	2.084×10^{-5}	1.00	1.97	
340	14,586	610.5	92.62	1028	8240	11,870	0.469	0.110	0.070×10^{-3}	2.255×10^{-5}	1.23	2.43	
360	18,651	528.3	144.0	720	14,690	25,800	0.427	0.178	0.060×10^{-3}	2.571×10^{-5}	2.06	3.73	
374.14	22,090	317.0	317.0	0	—	—	—	—	0.043×10^{-3}	4.313×10^{-5}			

Temp. T, °C	Saturation Pressure P, kPa	Density ρ, kg/m³ Liquid	Density Vapor	Enthalpy of Vaporization h_{fg} kJ/kg	Specific Heat c_p, J/kg·K Liquid	Specific Heat Vapor	Thermal Conductivity k, W/m·K Liquid	Thermal Conductivity Vapor	Dynamic Viscosity μ, kg/m·s Liquid	Dynamic Viscosity Vapor	Prandtl Number Pr Liquid	Prandtl Number Vapor	Volume Expansion Coefficient β, 1/K Liquid	Surface Tension, N/m
−40	71.66	690.2	0.6435	1389	4414	2242	—	0.01792	2.926×10^{-4}	7.957×10^{-6}	—	0.9955	0.00176	0.03565
−30	119.4	677.8	1.037	1360	4465	2322	—	0.01898	2.630×10^{-4}	8.311×10^{-6}	—	1.017	0.00185	0.03341
−25	151.5	671.5	1.296	1345	4489	2369	0.5968	0.01957	2.492×10^{-4}	8.490×10^{-6}	1.875	1.028	0.00190	0.03229
−20	190.1	665.1	1.603	1329	4514	2420	0.5853	0.02015	2.361×10^{-4}	8.669×10^{-6}	1.821	1.041	0.00194	0.03118
−15	236.2	658.6	1.966	1313	4538	2476	0.5737	0.02075	2.236×10^{-4}	8.851×10^{-6}	1.769	1.056	0.00199	0.03007
−10	290.8	652.1	2.391	1297	4564	2536	0.5621	0.02138	2.117×10^{-4}	9.034×10^{-6}	1.718	1.072	0.00205	0.02896
−5	354.9	645.4	2.886	1280	4589	2601	0.5505	0.02203	2.003×10^{-4}	9.218×10^{-6}	1.670	1.089	0.00210	0.02786
0	429.6	638.6	3.458	1262	4617	2672	0.5390	0.02270	1.896×10^{-4}	9.405×10^{-6}	1.624	1.107	0.00216	0.02676
5	516	631.7	4.116	1244	4645	2749	0.5274	0.02341	1.794×10^{-4}	9.593×10^{-6}	1.580	1.126	0.00223	0.02566
10	615.3	624.6	4.870	1226	4676	2831	0.5158	0.02415	1.697×10^{-4}	9.784×10^{-6}	1.539	1.147	0.00230	0.02457
15	728.8	617.5	5.729	1206	4709	2920	0.5042	0.02492	1.606×10^{-4}	9.978×10^{-6}	1.500	1.169	0.00237	0.02348
20	857.8	610.2	6.705	1186	4745	3016	0.4927	0.02573	1.519×10^{-4}	1.017×10^{-5}	1.463	1.193	0.00245	0.02240
25	1003	602.8	7.809	1166	4784	3120	0.4811	0.02658	1.438×10^{-4}	1.037×10^{-5}	1.430	1.218	0.00254	0.02132
30	1167	595.2	9.055	1144	4828	3232	0.4695	0.02748	1.361×10^{-4}	1.057×10^{-5}	1.399	1.244	0.00264	0.02024
35	1351	587.4	10.46	1122	4877	3354	0.4579	0.02843	1.288×10^{-4}	1.078×10^{-5}	1.372	1.272	0.00275	0.01917
40	1555	579.4	12.03	1099	4932	3486	0.4464	0.02943	1.219×10^{-4}	1.099×10^{-5}	1.347	1.303	0.00287	0.01810
45	1782	571.3	13.8	1075	4993	3631	0.4348	0.03049	1.155×10^{-4}	1.121×10^{-5}	1.327	1.335	0.00301	0.01704
50	2033	562.9	15.78	1051	5063	3790	0.4232	0.03162	1.094×10^{-4}	1.143×10^{-5}	1.310	1.371	0.00316	0.01598
55	2310	554.2	18.00	1025	5143	3967	0.4116	0.03283	1.037×10^{-4}	1.166×10^{-5}	1.297	1.409	0.00334	0.01493
60	2614	545.2	20.48	997.4	5234	4163	0.4001	0.03412	9.846×10^{-5}	1.189×10^{-5}	1.288	1.452	0.00354	0.01389
65	2948	536.0	23.26	968.9	5340	4384	0.3885	0.03550	9.347×10^{-5}	1.213×10^{-5}	1.285	1.499	0.00377	0.01285
70	3312	526.3	26.39	939.0	5463	4634	0.3769	0.03700	8.879×10^{-5}	1.238×10^{-5}	1.287	1.551	0.00404	0.01181
75	3709	516.2	29.90	907.5	5608	4923	0.3653	0.03862	8.440×10^{-5}	1.264×10^{-5}	1.296	1.612	0.00436	0.01079
80	4141	505.7	33.87	874.1	5780	5260	0.3538	0.04038	8.030×10^{-5}	1.292×10^{-5}	1.312	1.683	0.00474	0.00977
85	4609	494.5	38.36	838.6	5988	5659	0.3422	0.04232	7.646×10^{-5}	1.322×10^{-5}	1.338	1.768	0.00521	0.00876
90	5116	482.8	43.48	800.6	6242	6142	0.3306	0.04447	7.284×10^{-5}	1.354×10^{-5}	1.375	1.871	0.00579	0.00776
95	5665	470.2	49.35	759.8	6561	6740	0.3190	0.04687	6.946×10^{-5}	1.389×10^{-5}	1.429	1.999	0.00652	0.00677
100	6257	456.6	56.15	715.5	6972	7503	0.3075	0.04958	6.628×10^{-5}	1.429×10^{-5}	1.503	2.163	0.00749	0.00579

Appendix H

The design of the tank/evaporator is shown below.



Appendix I

Different electricity tariff rates

- **Karnataka**

Table 5: Electricity tariff rates for Karnataka

Rate Schedule	
Demand charges	Rs.180/kVA of billing demand/month
Energy charges	380 paise/unit
TOD Tariff at the option of the Consumer	
Time of Day	Increase + / reduction (-) in energy charges over the normal tariff applicable
22.00 Hrs to 06.00 Hrs	(-) 125 paise per unit
06.00 Hrs to 18.00 Hrs	0
18.00 Hrs to 22.00 Hrs	+ 100 paise per unit

- **Andra Pradesh**

Table 6: Electricity tariff rates for Andra Pradesh

Category	HP/kW Charge(Rs/Month)	kVA Charge (Rs/Month)	Energy Charge (Rs./Unit)
11 kV	0	250	4.80
33 kV	0	250	4.37
132 kV and above	0	250	3.97
TOD Tariff at the option of the Consumer			
Time of Day	Increase + / reduction (-) in energy charges over the normal tariff applicable		
18.00 Hrs to 22.00 Hrs	(+) 100 paise per kVAh		
06.00 Hrs to 18.00 Hrs	0		
22.00 Hrs to 06.00 Hrs	- 100 paise per kVAh		

- **New Delhi**

Table 7: Electricity tariff rates for New Delhi

Month	Peak Hours	Surcharge on energy charges	Off-Peak Hours	Rebate on Energy Charges
April-September	1500-2400 Hrs	10 %	0000-0600 Hrs	10 %
October-March	1700-2300 Hrs	5 %	2300-0600 Hrs	10 %

- **West Bengal**

Table 8: Electricity tariff rates for West Bengal

Period	Summer	Monsoon	Winter
Peak (18.00 Hrs to 22.00 Hrs)	Rs.5.36	Rs.5.30	Rs.5.24
Normal	Rs.5.22	Rs.5.16	Rs.5.10
Off-Peak (22.00 Hrs to 06.00 Hrs)	Rs.4.82	Rs.4.76	Rs.4.70