THERMAL ENERGY STORAGE APPLICATION FOR LOAD SHIFTING AND ELECTRICAL DEMAND MANAGEMENT IN SAUDI ARABIA

by

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Thesis submitted in partial fulfilment of the requirements for
the Degree of Doctor of Philosophy
Faculty of Technology, Kingston University UK
May 2005

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ABSTRACT

Energy plays a major role in the economic prosperity of the Middle Eastern countries. Since the region is the largest oil producer of the world, it is less anticipated that these countries would ever be face with an energy crisis similar to the one experienced by the rest of the world during 1970s.

The region was going through a chronic electricity demand supply crises with the demand for electrical energy in the rapidly expanding towns, cities and industries, far exceeding the power being made available. The relatively low electrical tariff also contributed to the increasing power demand due to wastage and uneconomical usage of electrical energy.

The power generating companies and the Government authorities in the Middle East encouraged scientists and engineers to engage in ambitious Demand Side Management (DSM) programmes to develop novel ideas and new technologies to improve system efficiencies and to reduce energy consumption specifically in the field of refrigeration and air conditioning.

The researcher began analysing the potential and possible applications of cool storage as a tool for Demand Side Management (DSM) in central air conditioning systems in the Middle East in 1991. The coupling of a refrigerated water storage tank or an ice storage tank to an air-cooled chiller plant, operated at night for load shifting, electrical peak demand reduction and energy conservation has been the major interest of investigation.

The model project commissioned in 1996 used as a typical example to investigate electrical demand management for an office building in Riyadh, Saudi Arabia.

The aim of this research was to develop new modified comfort cooling system coupled with a cool storage or commonly known as Thermal

Energy Storage (TES) network. The research was expected to establish certain favouring conditions in relation to technical, economical and environmental criteria to make the TES application a viable option in comfort cooling systems in commercial buildings in the Middle East for electrical demand reduction, load shifting and energy conservation.

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FOREWORD

The research work commenced in 1995, and the model project was commissioned during the summer of 1996. System monitoring was carried out for a further period of two years before the political unrest in the region brought an abrupt end to the monitoring and data gathering programme.

However, after a year of prolonged delay the researcher gained access to the project again under strict conditions such as the signing of a Confidential Non-disclosure of Information agreement.

In 2000 the researcher was provided with unconditional access to the project for data gathering and monitoring programme.

Understandably, the tragic events of September 11th brought an end to such a programme.

Since then, Saudi Arabia has been under a continuous and heightened threat of terrorist attack and all access to the project is now completely revoked by the bank. At one stage, the researcher's office was the target of an arson attack. In a nut shell the research was conducted under extremely difficult political conditions.

Occasional visits to observe the system function and operation are now permitted under strict access and security control.

Such incidents delayed the finalization of the research and the subsequent development of the thesis by at least three to four years.

The research was conducted under difficult conditions, however considering the political situation of the region the client provided adequate support and cooperation by intermittent access to the project for data gathering and system monitoring for research purposes.

Adequate information was collected to study, evaluate and finalize the thesis, although it took slightly longer than the anticipated completion time due to the numerous aforementioned difficulties.

The success of this project can be measured by the awareness and acceptance of Thermal Energy Storage (TES) as the preferred electrical Demand Side Management (DSM) tool in Kingdom of Saudi Arabia. The number of projects either commissioned or under construction to accept TES as the preferred DSM tool in the Kingdom is now standing at 350,000-kWh_c load shifting.

PREFACE AND ACKNOWLEDGEMENT

Several people and institutions who deserve a note of thanks and acknowledgment for this unique project; at the time of installation was the first of it's kind in the Middle East.

For ***Saudi French Bank*** (BSF), have trusted my proposal and invested 500,000 sterling pounds in the model project without knowing whether the out come of this project would justify such large investment. However, the outcome benefited them to their fullest satisfaction. My sincere thanks to BSF.

The contractors and engineers specially ***Control Technology*** and ***CALMAC*** for their support and continuous advise and assessment on the system performance and results.

Mark Mackraken, the researcher and ASHRAE Technical Committee Chairman for his support and continuous encouragement and comments and reviews of the results.

SCECO, the local power company for it's support and encouragement to realize this first DSM project in the Kingdom of Saudi Arabia.

Kingston University for accepting this project as a useful research programme in the interest and benefit of Middle East engineers, power generators and consumers.

Pauline Stephenson, my supervisor for her tireless support in reviews, assessments, publishing technical papers, and mainly her valuable time to go through the system design, review of results, and the structure of this thesis; without her support and help this thesis wouldn't be completed within the time frame.

Professor Andrew Self of Kingston University for finding time to visit the model project in Riyadh and contributed valuable comments and suggestions.

David Corneliius for his help in formatting and printing this thesis.

My wife and son; for their patience and co-operation.

My parents; my heart-felt gratitude for providing me with the valuable education to reach this point in my life.

My loyal and loving companion, my cat ***Mamalade***, who stayed awake during those long nights and kept me company when I was writing this thesis.

And, finally for all those who I couldn't mention in this acknowledgement, but some way or the other directly or indirectly contributed to the development of this research project and thesis.

CHAPTER-1

THE NEED FOR LOAD SHIFTING AND ELECTRICAL DEMAND MANAGEMENT IN SAUDI ARABIA

1.1 Introduction

Energy plays a major role in the economic prosperity of the Middle Eastern countries. Since the region is the largest oil producer of the world, it is has not been anticipated that these countries would ever be faced with an energy crisis similar to the one experienced by the rest of the world during 1970s.

However, the region is currently going through a chronic electricity demand supply crisis with the demand for electrical energy in the rapidly expanding towns, cities and industries, far exceeding the power being made available. The relatively low electrical tariff also contributes to this increasing power demand due to wastage and uneconomical usage of electrical energy.

This model project addresses electrical demand management for an office building in Riyadh, Saudi Arabia.

As an example of the pressure on electricity supply, during the summer of 1995, at the initiation of this research project, the power generating company of the central region of Saudi Arabia (SCECO) was faced with the immense task of meeting an average day time peak power demand of 5000 MVA as opposed to an average night time peak demand of 3500 MVA[1] SCECO had to bring more than thirty 50 MVA gas turbines on line during the daytime to meet this variation in demand. The major culprits are the comfort cooling systems of buildings and facilities that consume a large amount of electrical

energy during the daytime. The electricity requirements of these systems are variable based on the cooling demand of the building.
The existing conventional chilling plant design for buildings in the Middle East cannot readily accept energy management systems. Many of the plants in operation, suffered from what is known as Plant Starvation Syndrome (PSS)[2]. PSS is a hydraulic design deficiency that required more chillers to be in operation than necessary, these chillers are operated at a 50-70 percent load; Thus, PSS is clearly consuming large blocks of electrical energy in the form of chiller production in efficiency, and pumping system.

In addition selection of air-cooled chillers is a requirement by local legislation aiming to save water. Water is in short supply and an expensive commodity in the Gulf, and cooling towers are prohibited by the authorities in air conditioning system for commercial and office building; hence the use of water-cooled chillers could not be considered as an option.
Air-cooled chillers are the commonly used chillers in the Middle East though they are very inefficient in comparison to the water-cooled chillers. The average energy requirement for an air cooled chiller is approximately 0.5-0.7 kW per kW cooling compare to a water cooled chiller that only requires 0.15 - 0.17 kW per kW cooling [3].

The power generating companies and the Government authorities in the Middle East encouraged scientists and engineers to engage in ambitious Demand Side Management (DSM) programmes to develop novel ideas and new technologies to improve system efficiencies and to reduce energy consumption specifically in the field of refrigeration and air conditioning.
Cool Storage commonly known as Thermal Energy Storage (TES) is a potential DSM technology [7]; Cool Storage is widely accepted in several countries around the world due to one or more of the following favouring conditions:

- Low cost of night time electrical energy
- High electrical demand charges
- Utility rebate from the electricity companies

Cool Storage is has been used around the world to take advantage of the variable electricity tariff structure, therefore to reduce electricity costs, by shifting the consumption time from a higher to a lower tariff band of the day. [10]

These features are neither applicable nor in use in the Middle East. In the Middle East there is no low cost electricity at night, demand charges and rebates are not applicable.

However, the peak load during the summer leads the electricity company SCECO to restrict the use of electricity by requiring consumers to reduce their electricity consumption during peak period. Since majority of the electrical load is air-conditioning, this effectively means shutting down of part of the chiller plant or use alternative energy management programmes, such as onsite generation to manage electrical demand. [6]

The researcher began analysing the potential and possible applications of cool storage as a tool for DSM in central air conditioning systems in the Middle East in 1991. The coupling of a refrigerated water storage tank or an ice storage tank to an air-cooled chiller plant, operated at night for load shifting, electrical peak demand reduction and energy conservation has been the major interest of investigation.

The aim of this research is to develop new modified comfort cooling system coupled with a cool storage or commonly known as Thermal Energy Storage (TES) network. The research was expected to establish certain favouring conditions in relation to technical, economical and environmental criteria to make the TES application a viable option in comfort cooling systems in commercial buildings in the Middle East for power demand reduction, load shifting and energy conservation.

1.2 Research project background review

Fig.1.1 Saudi French Bank Head Quarters -Riyadh

A model DSM project has been designed by the researcher and built for the Saudi French Bank. It develops a load shifting and electrical demand reduction programme at their headquarters building in Riyadh. Saudi French Bank and Energico International jointly financed the project. It is worthwhile to note that the model project represents "new knowledge" to the air conditioning and refrigeration industry in the Kingdom of Saudi Arabia, as this was the *first project in the Middle East installed to provide ice storage based electrical Demand Side Management System.* [8]

At the initiation of the project an intensive literature search on the subject of TES for Demand Side Management in comfort cooling systems was conducted over a period of nine months from the date of registration at Kingston University. The literature search was a continuation to the investigation and initial system development work conducted by the researcher in the field of cool storage application from 1991. The researcher for information gathering and data collection purposes visited several cool storage projects around the world.

1.3 Aim of the Research and Expected Outcomes

The aim of the research is to demonstrate the feasibility of TES for electrical load management in the Middle East (particularly within the tariff regime represented by SCECO in Saudi Arabia) and to establish a set of new **" favouring conditions"** based on technical, economical and environmental issues that would substantiate the application of TES for Demand Side Management in comfort cooling systems in the Middle Eastern countries. The design development techniques of an energy efficient comfort cooling system that provides substantial power demand reduction and energy conservation in commercial buildings will be investigated analysed and presented.

The experimental results obtained from the proposed ice storage model will be used to develop and validate these favouring conditions during the course of the research work. Furthermore, based on the collected data, the research would establish actual data of **"de-rated"** performance of chillers during ice making conditions in the Middle East. The out come of the above investigations is expected to contribute ***original knowledge*** towards the TES application for Demand Side Management in comfort cooling systems in the Middle Eastern countries.

The TES application was investigated as an Electrical Demand Side Management (DSM) programme that would offer opportunity for chiller capacity reduction, load shifting, capital outlay reduction in power plants, and energy consumption savings in a retrofit application.

DSM using cool storage was investigated as a useful tool to curtail the Plant Starvation Syndrome (PSS) in chilling plant system. [2]

In this model system design, the installed chiller capacity would be reduced considerably in relation to the conventional design practice and therefore reduce the connected electrical load of the building.
As the chiller capacity is reduced below the peak cooling demand, this will provide the opportunity for the undersized chillers to operate on full capacity, assisted with the make up cooling from the TES system. This also would allow the chillers to operate at their highest efficiency and consume less electrical energy compare to a part load operation under conventional system design.

1.4 Electrical Demand And Cooling Requirements In Buildings In The Middle East

The cooling demand of commercial buildings in the Middle East is approximately 125 - 150 W/m2 compared to 75-100 W/m2 in Europe and North America. [10] In electricity terms this is equivalent to 90 -100 VA/m^2, above 30% of the prevailing average power requirement for comfort cooling systems in commercial buildings elsewhere in the world. In many parts of the Gulf countries the demand for electrical energy in the rapidly expanding towns, cities and industries, far exceeds the power being made available. In these countries the power utility companies are in a position where electrical demand will exceed safe capacity before a new plant can come on-line.

According to the statistics published by the Ministry of Industry and Electricity in Saudi Arabia the electrical energy generated by the utility companies for the year of 1996 has reached 85 million MWH, approximately 16 times the generated energy in the year of 1975[11] with an average annual increase of 15%, which is far higher than elsewhere in the world.

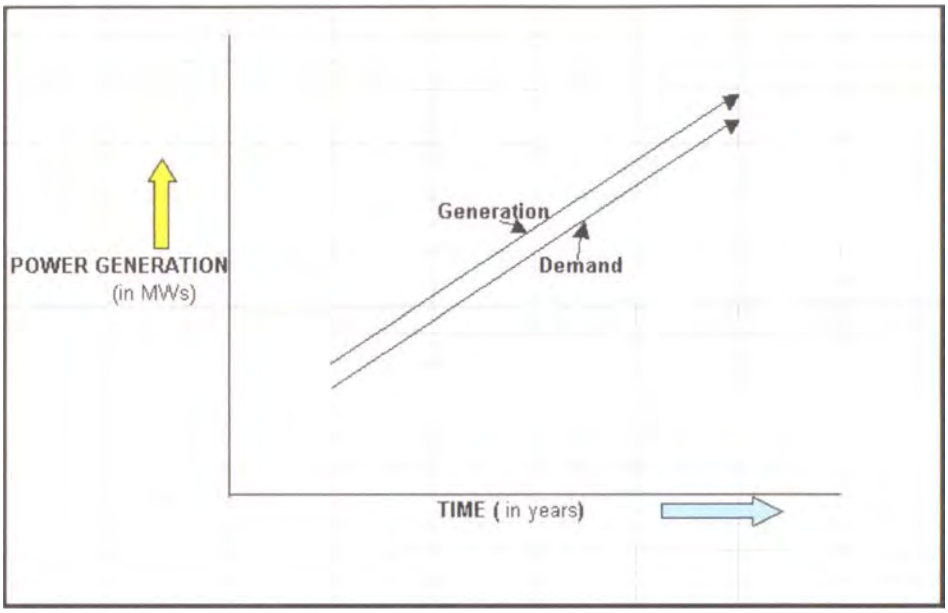

Fig.1.2 ideal Situation - power generation and demand

Fig.1.2 illustrates the ideal situation in power generation and power demand and Fig. 1.3 shows the present situation in the Kingdom of Saudi Arabia. The time that demand exceeds the production occurs between 1 p.m. and 5 p.m.

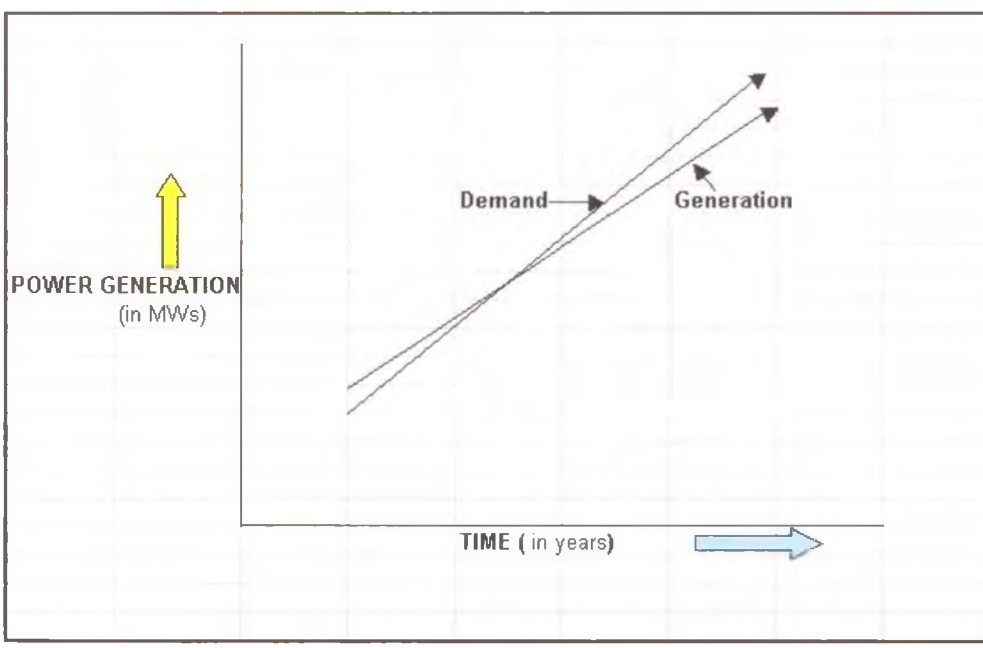

Fig.1.3 Critical Situation between Generation and Demand that calls for DSM

The summer time ambient temperatures in the region of 45-50°C call for carefully monitored close control comfort cooling systems for each and every building and facilities in the Middle East. More than 65% of the connected electrical energy of these buildings is mainly used for air-conditioning applications. The efficiency of the air-cooled chillers reduces as the ambient temperature increases during the daytime. The increase in ambient temperature also reduces the power efficiency of gas turbines. Since all cold generating equipment in buildings is powered by electricity generated by gas turbines, the combined efficiency loss of power generating and the power using equipment during peak hours is extremely high in these countries.

At the beginning of this research in 1995, SCECO's electricity tariff structure was based on maximum rate of 3 pence per kWh, and for the industries with a subsidised rate of 1 pence per kWh. [1] There was no daytime / night time variable tariff structure available, and no peak demand charges were in use, hence removed any financial incentive to operate load shifting systems.

To manage demand at peak times some member countries in the Gulf started to impose strict condition on the electricity users to reduce the power consumption during peak hours. For example in Saudi Arabia, a usage restriction was introduced in 1995, which would limit the electricity usage for air-conditioning applications at certain time of the day. [6]

In Riyadh the building owners are required to turn off or reduce their chiller capacity in the following manner:

1-2pm 10% capacity reduction
2-3pm 20% capacity reduction
3-4pm 30% capacity reduction
4-5pm 40% capacity reduction

New building developments were only given approval for power connection subject to satisfactory confirmation that certain energy conservation measures are included in the system design, and provisions were in place to reduce the on line chiller capacity by 50% between 1-5pm.

1.5 Critical Review investigation and Evaluation of the research topic

During the initial stage of the research programme past research work in the field of Demand Side Management in the Kingdom of Saudi Arabia mainly by SCECO [11] and other research institutions such as King Abdulaziz Centre for Science and Technology (KACST) were reviewed [6]. In addition the basic principles and favouring conditions of various cool storage systems specifically the concepts and design methodologies of ice and chilled water storage for Middle East application were reviewed and analysed. Their merits and limitations related to storage capacity, system efficiency and economics were investigated and validated. A thorough review was conducted on previous research works on cool storage applications in commercial buildings in Gulf countries.

During late eighties and early nineties, at the time of the commencement of this research work several investigations were conducted mainly in the US to establish efficient means to use TES techniques for load shifting in comfort cooling systems. According to MacCracken [5] TES, a 'sleeping giant' with unlimited potential in load and energy management and the efficient use of such systems yet to explored and validated. Wendland emphasized the need for optimum design and further investigation on comfort benefits in various Energy Technology conferences [40]. Dorgan, and Elleson highlighted the need for research of use of TES in high ambient environments in their technical publication 'New Design Guide for Cool Thermal Storage' [10].

However, the adaptation of cool storage in comfort cooling applications is not straight forward in the Gulf countries as such systems are mainly used in countries with certain favouring conditions such as low night time charges, peak demand charges and utility rebate. In the Gulf countries absence of such favouring conditions impose severe obstacle on designers to adopt the concept of cool storage economically and efficiently in comfort cooling systems. The research was aimed to establish some new and novel favouring technical and economical parameters that would make the cool storage a viable option for electrical demand reduction and energy management in commercial buildings.

1.6 Literature Search and Publications

A detailed literature search (see Appendix-1.1 and 1.2) on the subject of TES was undertaken from the date of registration for the intended research programme at Kingston University. American Society of Heating Refrigerating and Air Conditioning Engineers (ASHRAE), Chartered Institution of Building Services Engineers (CIBSE) and the Institute of Energy (IE) provided the main sources of information related to research work. Prior to the commencement of this research work, the researcher conducted several seminars and published papers in the Kingdom of Saudi Arabia emphasising the importance of conducting detailed research in TES applications in large comfort cooling systems. Notably, the presentation at the meeting of American Military Engineers in Riyadh in May 1994, article written in the National News papers, paper in the Gulf Industry Journal in Oct 1994 were paved way for the engagement in this important research work in the chosen subject.

Several site visits were made to the existing projects in USA and Switzerland; meetings and discussions were held with prominent researchers in the filed of Thermal Energy Storage Systems.

This includes MacCracken, inventor of the first internal melt ice bank system and winner of several technology awards of ASHRAE, Mather, developer of a unique stratified chilled water system with less than 60cm thermocline, Wait, developer of industrial control system for cool storage application, and Hill, the past president of ASHRAE.

Several papers on the research topic were published in International Conferences and regional technical forums.

Technical papers presented at the American Society of Heating Refrigerating and Air-conditioning Engineers (ASHRAE) and International Institute of Refrigeration (IIR) Joint Conference at the Al Ain University in United Arab Emirates (UAE) in April 1996 [76] and another paper presented at the International Conference organized by the Charted Institution of Building Services Engineers (CIBSE) at Harrogate Conference Centre /UK on October 1, 1996 [77].

In addition to the above several technical papers were presented at the local technical forums such as Saudi Symposium on Energy Management, at the King Abulaziz Centre for Science and Technology (KACST) in Riyadh, Saudi Arabia, and the Conference of Institute of Electrical Engineers in Riyadh, Saudi Arabia.

CHAPTER-2
DEMAND SIDE MANAGEMENT

2.1 Introduction

Demand Side Management (DSM) can be defined as the planned and programmed changing of the rate of electrical usage in a given time period. [4] The type of DSM applicable for the Middle Eastern conditions is peak load management because of the extremes in the demand for power between the daytime peak and the nighttime trough are varying large. In 1995, at the initiation of this research project, the ratio of average daytime peak to the average nighttime peak was about 30%. In buildings and large facilities 60-65% of the peak power requirement is consumed by comfort cooling systems. [11]

In the Kingdom of Saudi Arabia new buildings, or refurbishment requiring increased electrical power, now have conditions imposed by the electricity supplier SCECO that has the overall objective of managing the national electrical demand. [6]

SCECO has declared a 4 hours period from 1300 hours until 1700 hours as the critical period, which requires some sort of demand management in buildings.

Turning off certain equipment within the facility during the critical period can control the timing and duration of the peak demand (kWD) in a building or facility or region. This process is referred as “load shedding”. This implies that some means of understanding with the utility company s to when the peak periods will occur. The main point is that there is a time-related pattern of the electrical usage of the facility that has to be modified or adjusted to meet the utility company’s load shedding requirements. By understanding how this pattern develops and the effect of this to the building users, the peak demand can be reduced on the facility and thus on the grid.

In developing an efficient DSM programme, it is necessary to understand the energy operation of the facility and the constraints placed by the electricity supplier. Few facilities have knowledge of or understand the "where, how, quantities and efficiencies" of the electricity used in their facilities.
The basic usage from a historic standpoint must be understood in order to develop a suitable DSM programme for that specific facility/building.

In assessing a facility for a DSM programme have it is necessary to identify the energy usage and potential areas of demand reduction.

1. Compare the available technical options for DSM
2. Quantify and select the best technical options.
3. Design and finally Verify the DSM System

In this research project the first step to DSM was to compile a load survey of all equipment within the facility that consume electricity. (See Chapter- 4.4)
The survey included the following:

1. Equipment operating time
2. The kW draw of the equipment
3. Operating conditions and limitations of the equipment, whether it can be turned off during a particular period.
4. Whether certain equipment can be operated on partial load so that the kW can be reduced.
5. The constraints associated with the shutdown and re-start.

The above information can provide the kW values of the equipment that can be shed, and the knowledge of when to shed.
This would lead to the needed analysis that relates to "DSM".

2.2 Method For Peak Load Shaving And Demand Management

There are varieties of methods that could be used to achieved effective electrical load management in buildings/facilities. [4].

Some of the realistic options available for the Middle East conditions to manage the power requirement for cooling are as follows:

2.2.1 Peak Shaving with existing stand-by generators

Many facilities and large buildings have existing or new stand-by engine driven power generators, which are normally used during main power failure to provide continuous electricity to essential services. During short period of demand peaks, these generators could be used to supply power to the specific equipment ear marked for DSM. This option will allow the building owners to switch the power from the grid to stand-by generator through and automatic transfer switch controlled by a timer or BMS, during the critical period.

This is a good economic option in Saudi Arabia as the average kW production cost is around 12 halalas per kW against the 28 halalas per kW from the grid. In addition the operation will comply with the mandatory DSM requirement. However, several negative factors such as noise, vibration, pollution, fuel transport and storage etc. discourage the building owners from using this DSM option in their facilities.

2.2.2 Capacity Reduction:

In the Gulf, during summer period, air conditioning is the major building load. Hence, capacity reduction would mainly rely on shutting down of the air-conditioning system. Shutting down of the air-conditioning system in buildings during peak hours will result in intolerable high internal temperature/humidity conditions to the occupants in those areas.

Furthermore, the “pulling down” period required to re-cool these areas would be in hours, and these areas cannot be occupied immediately after the DSM period.

a. *Selective Control:* -Large and new cold generating equipment such as chillers are provided with ‘liner capacity controllers’. During DSM period these equipment can be speed/capacity controlled to achieve the required power reduction in these facilities. Selected areas/rooms can be vacated and the air-handling units that handle these areas can be shut down. Alternatively room set temperature can be increased by say 2-3°C, to match the reduction in chiller capacity.

b. *Equipment Up-grading*: - This would involve replacing the inefficient equipment with energy efficient equipment. In addition replacing constant speed motors with variable speed motors etc. will provide efficient but limited load reduction. However, this option involved in system retrofitting and may not provide the necessary DSM requirement such as the 50% shut-down requested in Saudi Arabia. [6]

c. *Thermal Energy Storage*(TES) *[10]* : - In facilities, or in countries of high ambient temperature conditions, which require large air-conditioning systems, Thermal Energy Storage (TES) would be an ideal choice for DSM programme. TES would allow shifting from full load up to any size partial load of the chillers during DSM period. There would no penalty on indoor environmental conditions.

2.3 Tariff Structure in Saudi Arabia

In contrast to complex tariff structures in other parts of the world, the power generating company SCECO only charges the customers based on kWh. [1] At the time of this research, peak demand charges, reactive demand charges, contract demand or billing demand charges were not used in the Kingdom.

However, to protect the interest of domestic power users from the large commercial users, SCECO has developed their own method of billing. The domestic and locally owned industrial (factory) users are charged a lower tariff, whereas the commercial power users such as offices, hotels etc. where air-conditioning is widely used charged a higher tariff.

In addition, the Government of Kingdom of Saudi Arabia has recently announced an increase in electrical tariff and this has brought serious awareness among large property developers and building owners and forced them to consider novel means and methods to conserve and manage energy usage.

There are certain restrictions on “connected load’; for example for new buildings the electrical supply is capped by 25% of the requested connected load, forcing the developer to seek conservation techniques to manage the building power requirement within the provided 75% of the needed power for the building/facility.[6]

Also time related power usage is imposed, such as mandatory reduction in consumption during predefined peak demand period during summer, commencing from June until end of September. Load reduction is specifically targeted on air conditioning systems.

In case of large facilities, SCECO has installed remote shutdown controls on chillers to make sure that the specified DSM are strictly followed.

The peak demand period is based on a 4 hours reduction period occurring between 1.00 p.m. until 5.00 p.m. with the start time is staggered depending on the distribution zone.

The tariff is tiered according to the consumption in kWh.

Band	**kWh Range** kWh/month	**Old rate (1999)** halalas / kWh	**Present Rate (2000)** halalas/kWh
Band-1	1- 2000	5	5
Band-2	2001-4000	10	10
Band-3	4001-6000	10	13
Band-4	5001-6000	15	18
Band-5	6001-7000	15	23
Band-6	7001-8000	15	28
Band-7	8001-9000	15	32
Band-8	9001-10000	15	36
Band-9	10001 & >	15	38

Figure 2.1-Electricity Tariff Structure in Saudi Arabia in 1999-2000

[1Pence= 6 halalas 1 US Cent = 3.75 halalas *source [SCECO-2000]*]

The Bands 1-3 are the average monthly consumption of domestic usage; i.e. the domestic power users are protected from this massive increase in electricity charges.
However, Bands 7-9 are the average consumption in commercial application such as office building, shopping malls, hotels and other commercial developments. Major commercial users with a peak demand in excess of 10 MW will be paying their electricity bills at the rate listed in band-9.
This massive increase in tariff was brought in during early 2000 but after several deliberations and meetings with owners of large industries and commercial developers SCECO has reduced the band-9 charges to 28 halalas.[11]

Some member countries in the Gulf have started to impose strict condition on the electricity users to reduce the power consumption during peak hours. For example in Saudi Arabia, a usage restriction was introduced in 1995, which would limit the electricity usage for air-conditioning applications at certain time of the day.

In Riyadh the building owners are expected to turn off or reduce their chiller capacity in the following manner:

1-2pm 10% capacity reduction
2-3pm 20% capacity reduction
3-4pm 30% capacity reduction
4-5pm 40% capacity reduction

New building developments are only given approval for power connection subject to satisfactory confirmation that certain energy conservation measures are included in the system design, and provisions are made to reduce the on line chiller capacity by 50% between 1-5pm. For existing buildings, SCECO and the building owners jointly establish specific capacity reduction programmes based on the size of the building and peak demand.[6]

The above restrictions institute the need for demand side management (DSM) in Gulf countries. The building-cooling load contributes to 60-65% of power demand. Hence energy conservation in cooling system provides the basis for the design of cool storage application.

In addition SECO regulations reinforce the need to target the cooling systems for load management. The other options such as stand-by generation, capacity reduction were considered but not preferred due to their obvious disadvantages.

Stand-by generation is barred in the Gulf due to noise and air pollutions and fire hazard.

Capacity reductions have minor effect on the peak load and will have negative effect on continuous occupancy and comfort conditions.

The TES option provides attractive option as this would allow the building owners to reduce the power demand substantially but maintaining the internal comfort conditions unaffected.

TES is selected for this model project to provide the necessary DSM for a typical office building occupied by a bank in Riyadh.

TES is an efficient method on managing the cooling demand with an average peak load chillers. Storage allows the operators to spread the chiller run period throughout the 24 hours.

CHAPTER-3

THERMAL ENERGY STORAGE TECHNOLOGIES FOR DEMAND SIDE MANAGEMENT

3.1 Introduction

The following chapter illustrates the Cool Storage or commonly known as Thermal Energy Storage (TES) systems in the market.

TES may be economical if one or more of the following conditions exist [8]:

- High energy demand costs
- Energy time-of-use rates
- High daily load variations
- Short duration loads
- Infrequent or cyclical loads
- Capacity of cooling equipment has trouble handling peak loads

Effective applications of thermal energy storage include:

- Electrical power use management by shifting the cooling load to off-peak hours and reducing peak load (Demand Side Management)
- Reducing required capacity of building and process cooling systems, or helping existing cooling equipment to handle an increased load.

Water storage systems are often used in new large cooling system applications in conjunction with cogeneration and/or district energy systems.[30] Water-ice storage is the most common cooling storage in smaller applications. Because latent heat storage (phase change between water and ice) has a smaller volume, it is often chosen for retrofit applications with limited space.

In general, the buildings that offer the highest potential are offices, retail, and medical facilities.

Thermal energy storage systems are installed for two major reasons: lower initial project costs and lower operating costs. Initial cost may be lower because distribution temperatures are lower and equipment and pipe sizes can be reduced. Operating costs may be lower due to smaller compressors and pumps as well as reduced time-of-day or peak demand energy costs.
In general three major TES technologies can be identified.

1. Ice Storage
2. Chilled Water Storage
3. Eutectic Phase Change Storage

In this chapter the general features of the above three forms of thermal storage systems are described and reviewed, with major focus on ice and chilled water storage.
Eutectic Phase change storage systems are normally not considered in building air-conditioning applications.

3.2 **Ice Storage**

The concept of using ice for cooling goes back to the Egyptians use of evaporative cooling to form ice at night in clay casks covered in wet cloths. Ice has long been used for space comfort conditioning. In the early nineteenth century, ice was placed in air ducts in theatres to cool and dehumidify warm air blown by fans or melted and used as in air washing/cooling systems in movie studios. Many such systems are still in use at the Paramount Studios in Hollywood, CA. after more than 70 years of service.[12]
Ice storage systems are available for end users in two different configurations, ice builders, and ice harvesters. [54] Both systems consist of an ice making chiller, ice storage tank, thermal transfer fluid, pumps and control system.
Ice builders use the circulation of low temperature glycol solution to build ice remotely from the chillers either on storage-coils or to freeze water to produce ice within a storage tank. The ice can be formed around coils, within capsules or be in the form of slush depending on the type of technology. [51]
Ice harvesters are another form of cool storage systems where the evaporator of the chillier is the ice-building surface but ice is periodically

removed from the evaporator surface and stored remotely from the ice making system.[40]

The ice builders use glycol solution as 'transfer fluid' to make ice remotely whereas harvesters uses the direct refrigeration cycle for the ice making process. In both cases air cooled chillers are normally used. In general the ice storage systems have to operate at lower chillier evaporative temperatures than chilled water storage systems (discussed below) and this usually decreases the efficiency and performance of the chillers compared with conventional air conditioning and chilled water storage systems. In both cases the ice making chillers are de-rated to operate at temperatures below freezing, in the range of –5 to –6°C.[34]

The important feature of ice storage system is that the amount of ice made in each tank is determined by the insulating effect of the ice. Larger surface area with limited space between the coils is the preferred choice of the manufacturers to increase the storage capacity in ice banks. On the other hand ice thickness controllers are to be used to avoid bridging and excessive ice building.[55]

While discharging the ice the system can either use the same ice making circuit to discharge the energy or a separate water circuit is used.

In the first instant the system is called internal melting; and in the second case the system is referred as external melting system.

In the ice harvester system the ice making and ice storage are separated; water is circulated through the storage tank to meet the cooling demand. The system is only used for external melting applications.

Ice storage system allows the designer to supply lower temperature water to the air handling and fan coil units; thus allow the designer to use a larger temperature gradient between supply and return temperatures of the hydronic and air moving system.[44]

This will help the designer to reduce the sizes of pipe, pumps, ducts and air moving equipment. The savings on pumps and fan energy will normally offset the energy penalty associated with chillers due to lower condensing temperatures.

The following discusses the details of different ice storage systems available for commercial applications.

3.2.1 Ice on Coil - Internal Melt

Ice on coil internal melting system is the most popular ice build system used in the building industry.[36] A totally closed pressurized system that uses the same pipe network for charging and discharging.

The same pipe work used for charging the ice tank is used for melting the ice. A single pipe work and pumping circuit is used thus reduce capital cost of the system. Special valving and piping arrangements are used , so that the same circuit can be used for charging and discharging . Thus this type of system provides cost effective equipment utilization, compared with other ice storage systems listed below.

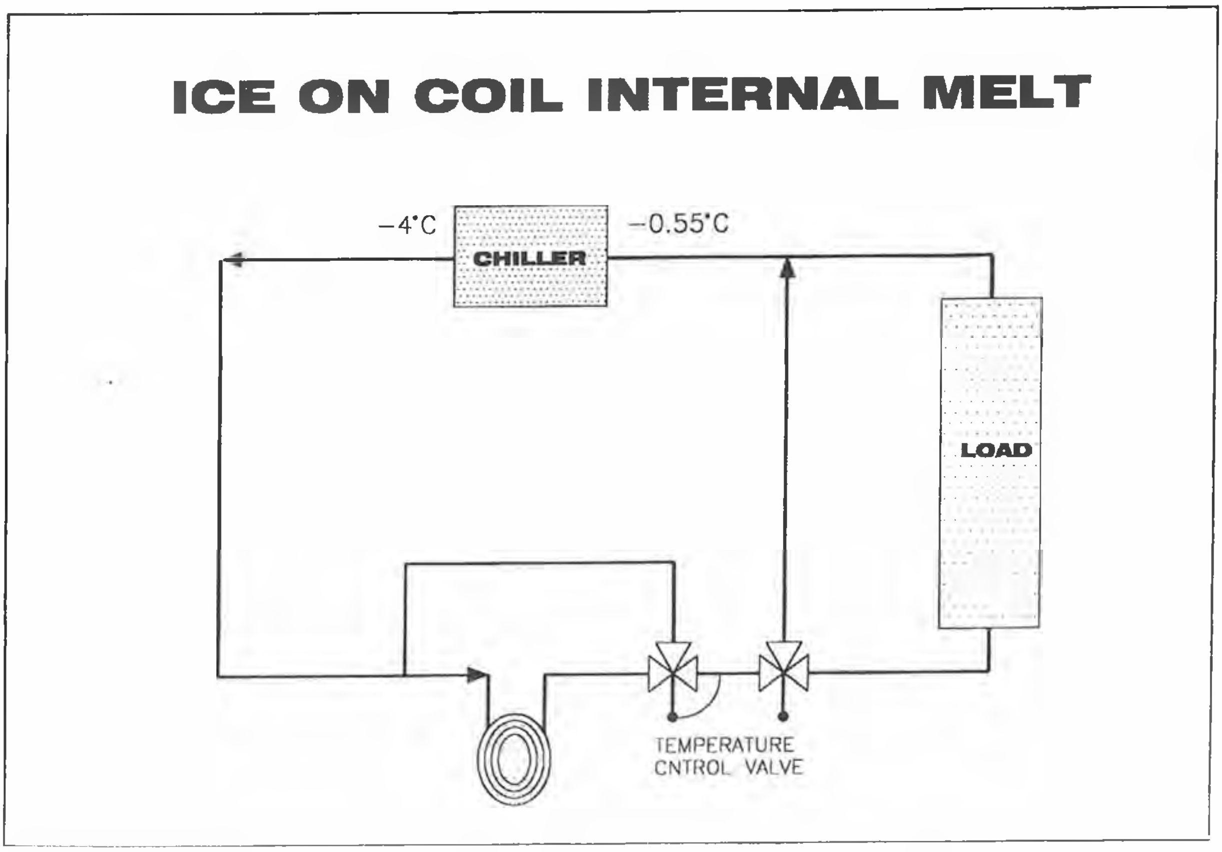

Fig.3.1 Ice on Coil Internal melt – schematic of system principle

Optimised size and length of internal heat exchangers in the tank allows 95% of the water in tank to be frozen solid within a defined time, with lower chiller capacity and energy consumption.[63]

The ice is built uniformly throughout the tank through temperature averaging method by using closely spaced counter-flow heat exchanger tubes.

Advantages:

Operates on single circuit

Ice banks are factory manufactured, tested and proven

Lower capital cost

Simplified operation

Uses less quantity of glycol

Useful on defined capacity discharge situation

Disadvantages:

Ice banks are unitary and only available in smaller sizes

70 TRH UPTO 300 TRH

Poor discharging efficiency after 50% melt down due to reduced thermal conductivity

Not effective for larger instantaneous discharge

3.2.2 Ice On Coil External Melt

The external melt system is similar to the internal melt system pressurized for ice making. However, ice discharging is carried out through a separate pumping and piping networks from a tank open to the atmosphere.

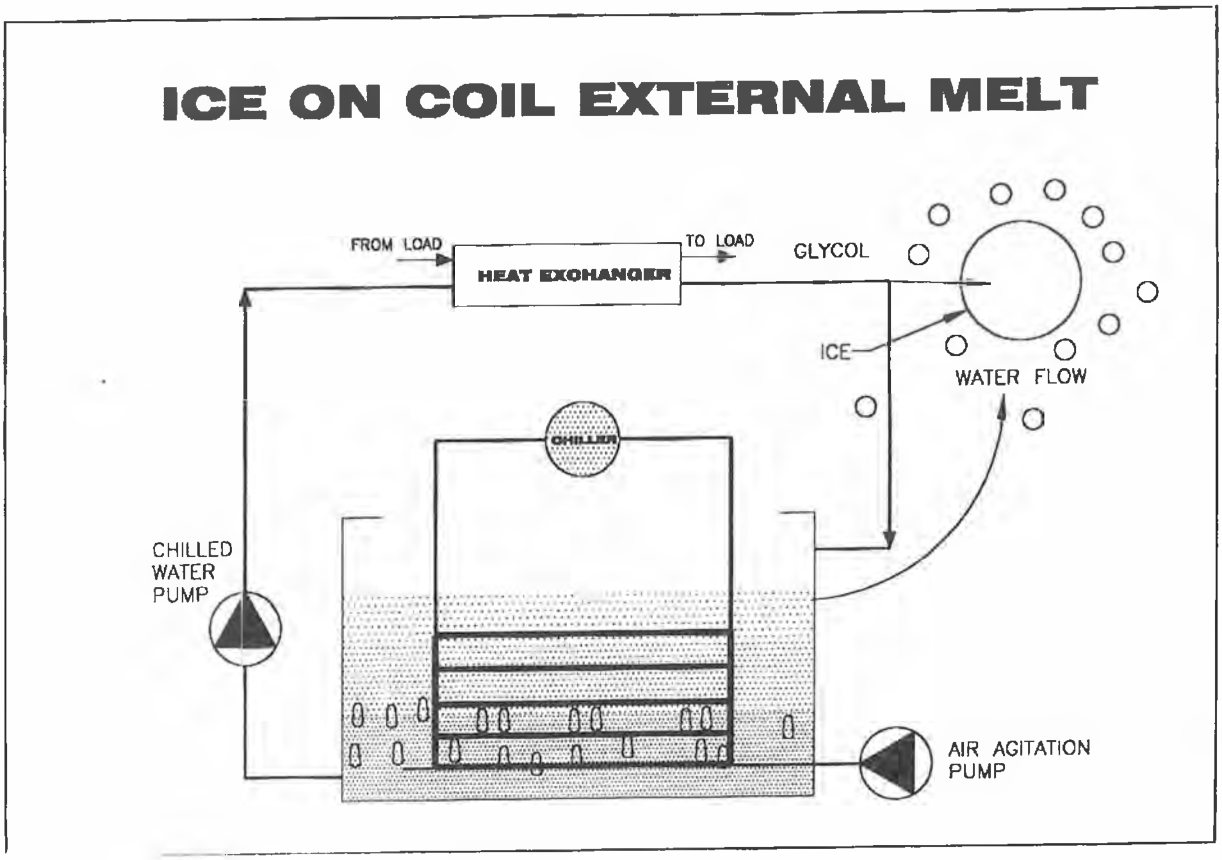

Fig.3.2 Ice on Coil External melt – schematic of system principle

The cooling circuit water is directly pumped onto the ice from one end of the tank and discharged from other end of the tank.[40]

The ice on coil external melt system makes ice on prime steel or galvanised steel coils. The refrigerant system can use all practical refrigerants including R-22, R-134a [24] and ammonia and a secondary fluid such as ethylene glycol.[23,24,25,26,28,29]

The coils are submerged in an open water bath contained in a metal or concrete tank. In the ice making mode the ice build medium is pumped inside the coil to build ice around the coil to a thickness between 32 –50mm and is terminated with an ice thickness controller. To melt the ice , water is drawn from the water bath directly to the load or an intermediate heat exchanger. The warm return water is returned to the top of the storage tank where it is cooled by the ice coils where the ice melts from the outside surface in towards the pipe. The ice builder melts the ice furthest from the mechanically chilled ice making fluid.

Air agitation is used to ensure an even ice build and avoid ice bridging between tubs which can cause uneven charging and or water channelling through the tank reducing usable capacity.
The technology has two coolants (ice building and ice melting), the partial storage strategy is inefficient and expensive due to additional piping and equipment.
Ice on coil - external melt systems were one of the first commercially applied off peak air conditioning ice storage systems. When air conditioning was first installed ice on coil external melt systems were applied to theatres and some churches.
Although ice on coil external melt system are not the most efficient system available today, they are still applied to industrial applications such as dairy cooling where a large amount of cooling needed at a short period of time. I.e discharging of stored energy quickly is more important than energy efficiency and energy consumption.
Air injection [[60] could increase heat transfer but, this system is always associated with inefficient and no-uniform discharging in relation to energy and temperature and has limited application in buildings.
Advantages:
Ice coils are factory manufactured, tested and proven
Lower capital cost
Fast discharge
Uses less quantity of glycol
Useful on un-defined capacity large discharge situation

Disadvantages:
Operates on dual circuits
Poor discharging efficiency after initial melt down due to tubing effect
Not useful for smaller ice storage systems
Need more controls and pumps

3.2.3 Slush Systems

The system is the newest ice storage system a modified version of the external melting system [67]. The system contains a pressurized ice making loop and an ice melting loop open to the atmosphere. The ice-making loop features an orbital rod evaporator, a condensing unit and a storage tank.

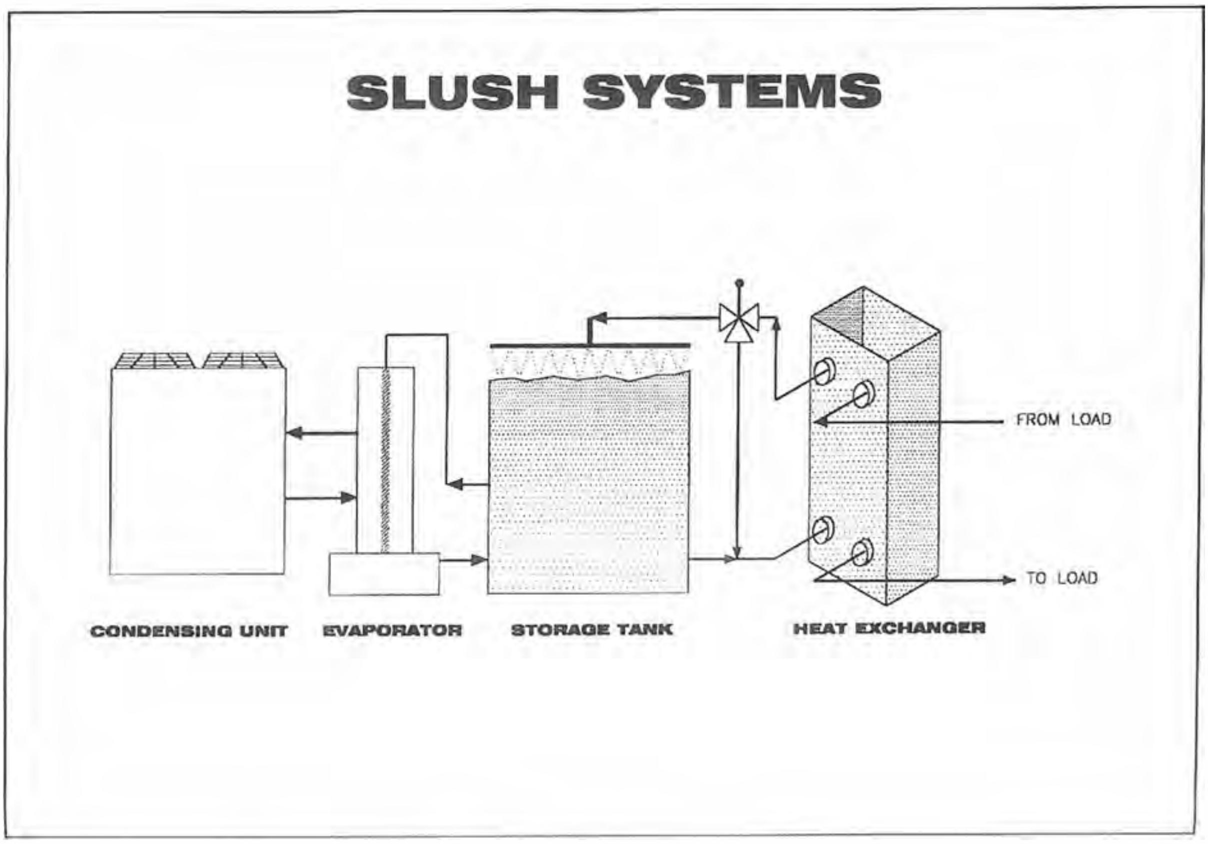

Fig. 3.3 Slush – schematic of system principle

Within the ice making loop, the solution in the storage tank is converted from a liquid to ice slurry as it is continuously pumped through the evaporator and back to the storage tank. This system is sometimes referred as 'pumpable ice' system.

The ice-melting loop contains the cooling load and the storage tank. Solution is pumped from the bottom of the storage tank to the cooling load, where the air conditioning equipment uses the cooling energy.

Advantages & disadvantages:

The system allows for flexible placement of equipment. The condensing unit can be located outside, the evaporator inside, the storage tank outside and the heat exchanger inside.

However, the slush system is a new technology that has not proven successful in the past. The storage tank stratification has been a big problem with slush systems. The slush tends to harden over time becoming a big ice chunk that is hard to melt causing uneven discharge fluid temperatures.
In addition this system is more complicated that of the internal melt static- ice build system. The slush system has more moving parts to maintain and to integrate with several other systems.
In addition the slush compressor must always operate at ice making temperatures to avoid discharging warmer than 0°C fluid in the tank melting ice storage in the partial storage mode.

3.2.4 Encapsulated Systems

The system contains the phase change medium encapsulated in several small capsules (plastic balls) stored in a large tank . A brine or ethylene or propylene glycol solution flow around the ice balls, rectangular lenses or bottles until phase change material is frozen inside the container. The container forms part of the pipe network and is not open to the atmosphere. Plastics balls, rectangular lenses even recycled soda bottles are used as the encapsulation device.[41]
Containers generally plastic, house a phase change material such as water or eutectic salts are placed into a large storage tank. Coolant is circulated through the tank and around the containers to freeze and melt them.

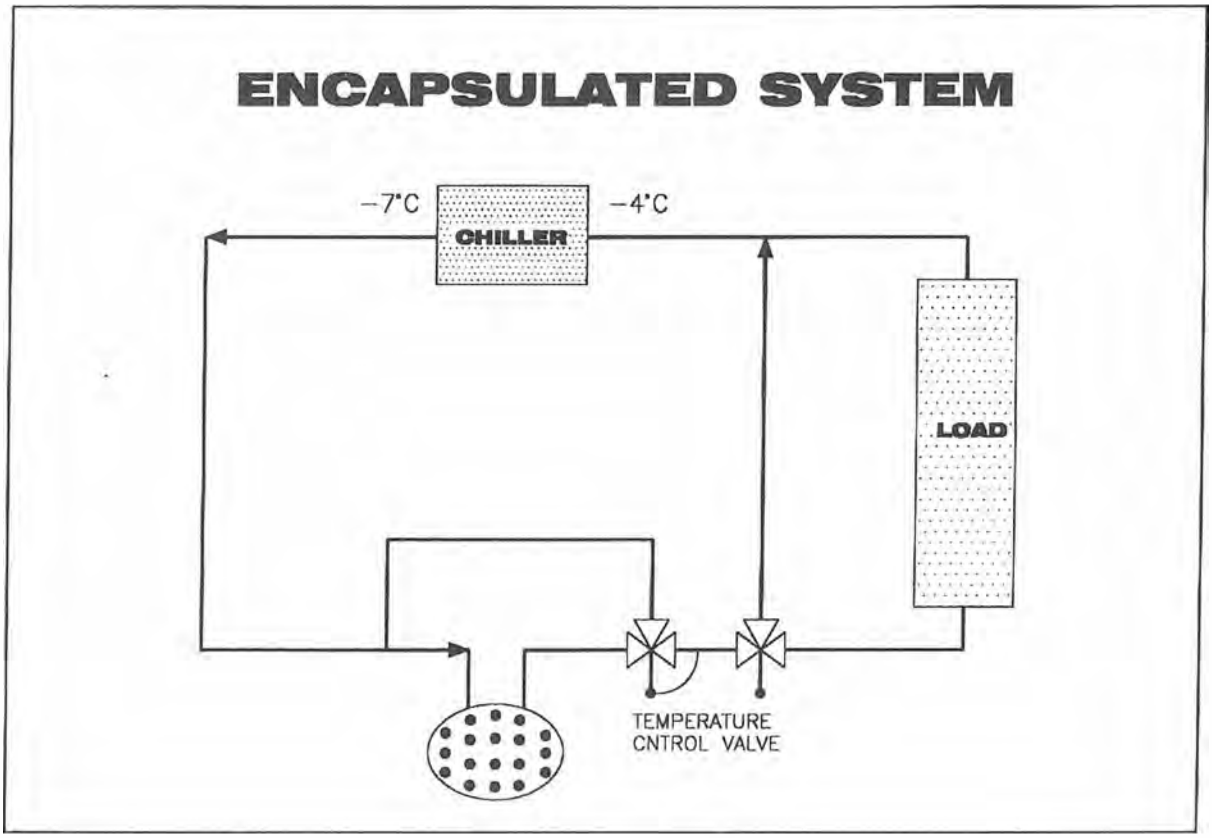

Fig. 3.4 Encapsulated system – schematic of system principle

Advantages & disadvantages:

Because encapsulated systems have one piping system and the chiller is separate from the ice maker the systems have several modes of operation such as ice making, ice making and cooling, cool, with ice, cool with chillers and cool with chiller and ice.
This flexibility allows the designer and owner to change operation of the ice system as operational schedules, rates, and or weather conditions dictate.

Encapsulated systems require a large heavy industrial style tank, which lend themselves to stress of direct burial. Direct burial tanks are out of sight and could be located under parking facilities.

The encapsulated devices are small and thus can be conform to many unusual shapes and they could potentially be put into an existing tank saving installation cost.

However, the encapsulated ice tank system is not factory assembled prior to shipment. Installation of encapsulated devices is difficult and time consuming. The quality , performance and reliability is dependent upon the experience of the mechanical contractor who is not an expert in ice storage tank assembly.

The encapsulated container is placed in a large tank and the heat transfer fluid allows flowing freely through the tank. As fluid always flow through the path of least resistance, the flow path cannot be controlled or managed. As encapsulated containers shift and move due to freezing, thawing and pressure changes flow channelling can occur through the tank. The non-uniform flow channelling may reduce capacity and ice making performance. The larger tank can require up to five-time glycol solution adding to installation and maintenance cost. The larger tank also reduces reliability of the ice system.

3.2.5 Dynamic Ice Harvesters

Ice harvesters also called ice shuckers, are refrigeration systems where the evaporator is the ice building surface and ice is periodically removed from the evaporator surface and stored remotely from the ice making system.[15] This system is a derivative of commercial ice makers (bag ice or food processing ice). Cold refrigerant is pumped into stainless steel plates. Return water is mixed with tank water and distributed over the plates. Ice is made in about a thirty-minute cycle to a thickness of about 10mm. The ice is harvested in a defrost cycle when mechanical reversing valves send hot gas into the plates. The ice falls into the atmospherically open tank below.

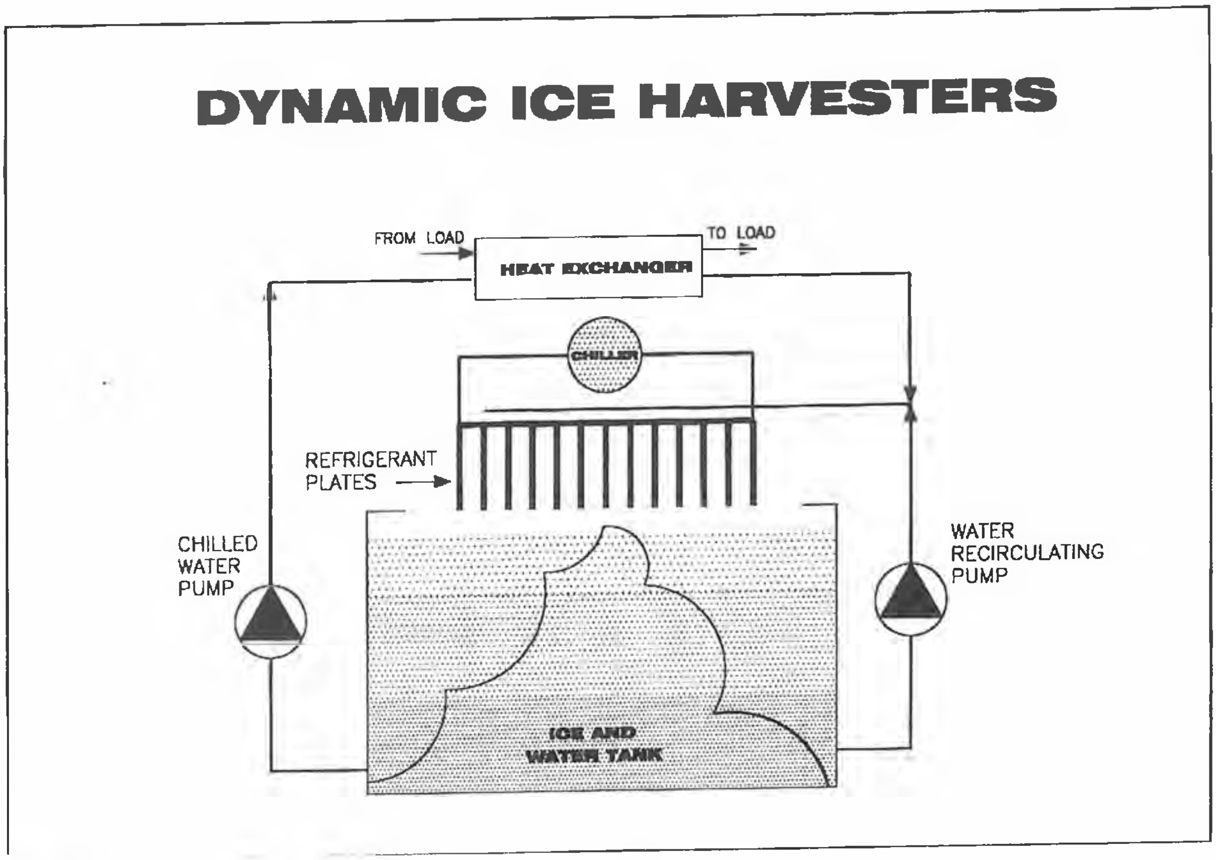

Fig. 3.5 Dynamic Ice Harvesters – schematic of system principle

Advantages & Disadvantages:

Water is circulated by a pump through the tank to the load and returned to the tank. A re-circulating pump is required to mix cold tank water with return water in order to lower the water temperature enough so ice can be made in the cycle time. This pump can be several time larger than the system pump.

Efficient refrigerant to water heat transfer and thinner ice building thickness is more efficient than ice on coil outside in melt systems.

Harvested ice has a large surface area and is in direct contact with cooling water allowing a quick burn capability at lower temperatures.

However, ice harvester technology is an expensive system to be used in comfort cooling systems. Partial storage, which requires the refrigeration system, and the ice storage to cool the load at the same time, is not a good application on ice harvesters. If a harvester system is to operate in the conventional mode producing 6°C water the ice storage could be melted. To avoid melting storage the harvester refrigeration system cools water as close

as possible to 0°C., thus wasting energy. Ice harvesters are complex refrigeration systems that must perfect operating condition to perform well and operate efficiently and reliability. Consideration must be given to how ice will stack during harvesting to eliminate the tank from flow channelling and the ice harvester from jamming during defrost. The defrost cycle, which send hot gases to plates, is required to harvest the ice from the evaporator into the tank. Energy losses from this cycle can be from 7 –10%.
In addition ice harvester system requires large amount of refrigerant.

3.2.6 Comparison of various ice storage options

Open ice storage systems (external melt) duplicate the pipe and pumping system. In addition it limits the pumping pressure and there is high possibility that air bubbles can be carried into the network. Advantage of an open ice storage system is that it can handle large discharge load in comparison to a closed system by flooding the formed ice with circulating chilled water.
Encapsulated system operates on the principle of 'flooding ' the tank with glycol, an expensive process. Establishing a uniform 'contact' space between glycol and the capsules is a difficult task as the tank is initially filled/stack with the ice balls.
The system is flexible as the storage tank is site built and can be designed for specific projects.

The internal melt system allows maximum quantity of the water in tank to be frozen solid rapidly and uniformly, with lower chiller capacity and energy consumption in comparison to open system. External melts have to charge through ice layers whereas internal melt only replenish the used ice by charging through water adjacent to the tube. The packing of tubes in an internal melt ice tank is such that the distance between them is the optimum to prevent over charging. The ice thickness formed around tubes is limited to the optimum for melting. Therefore internal melts are more efficient on charging cycle.
The system operates on single circuit giving simplified operation and lower requirement of glycol in comparison to external melt and encapsulated

systems. Disadvantage of an internal melt is that discharge rates are relatively small in comparison to open type ice storage or chilled water system and used in systems with a discharge requirement of up to 300TRH.

Compactness of an ice storage system outweighs the benefit of stratified chilled water storage in small storage applications. Chilled water storage is a useful TES tool in large storage projects requires a 10,500 kWh_c storage and above, which would provide the minimum aspect ratio of the storage tank for efficient operation. This would be approximately 8mD x 12mH. The increase of storage capacity would reduce the base area of the storage system as chilled water storage take disadvantage of the height of the storage tank, where as the maximum height of an ice storage system is limited to 6m, due to complexity in system construction for distribution of glycol circuit and system maintenance. In addition large instantaneous discharge is difficult in ice storage system

3.3 Chilled water system

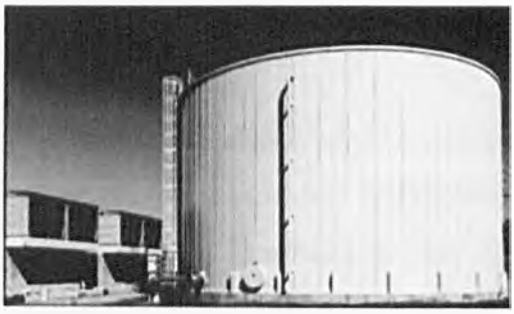

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Fig.3.6 Typical chilled water Storage projects

Chilled water storage systems are relatively simple in design, and differ primarily in how to keep stored chilled water from mixing with warm return water inside the storage tanks.[30]
Initial trial projects utilised multiple tanks to keep return water from the system mixing with stored chilled water by filling an empty tank with return water while supplying chilled water from the storage tank.[12]
This system was associated with several disadvantages such as large tanks, additional piping and valving, sophisticated control system and the possibility of introducing oxygen in water system. The maintenance issue also become a burden to the end users.
Using diaphragms to separate stored cold water and the return warm water was tires at later stages. The diaphragms moves either vertically or horizontally as the volume of the chilled water in the tank increases or decreases.[10]
The major problem associated with this method was controlling of leaks through and around the diaphragm. The weight of water imposes pressure on thin diaphragms and cause ruptures and the movement of the diaphragm also cause wear and tear.

The most accepted and commonly used chilled water storage system is based on "stratification " principles. The system often described as a Nozzle Matrix system. Stored cold water is withdrawn from the bottom of the tank through a series of hydraulically balanced nozzles; warm water enters the top of the tank through another series of nozzles.[70]
The water is distributed or drawn at an extremely low velocity often referred as "dimensionless" velocity. The nozzles keep turbulence in the tank to a minimum.

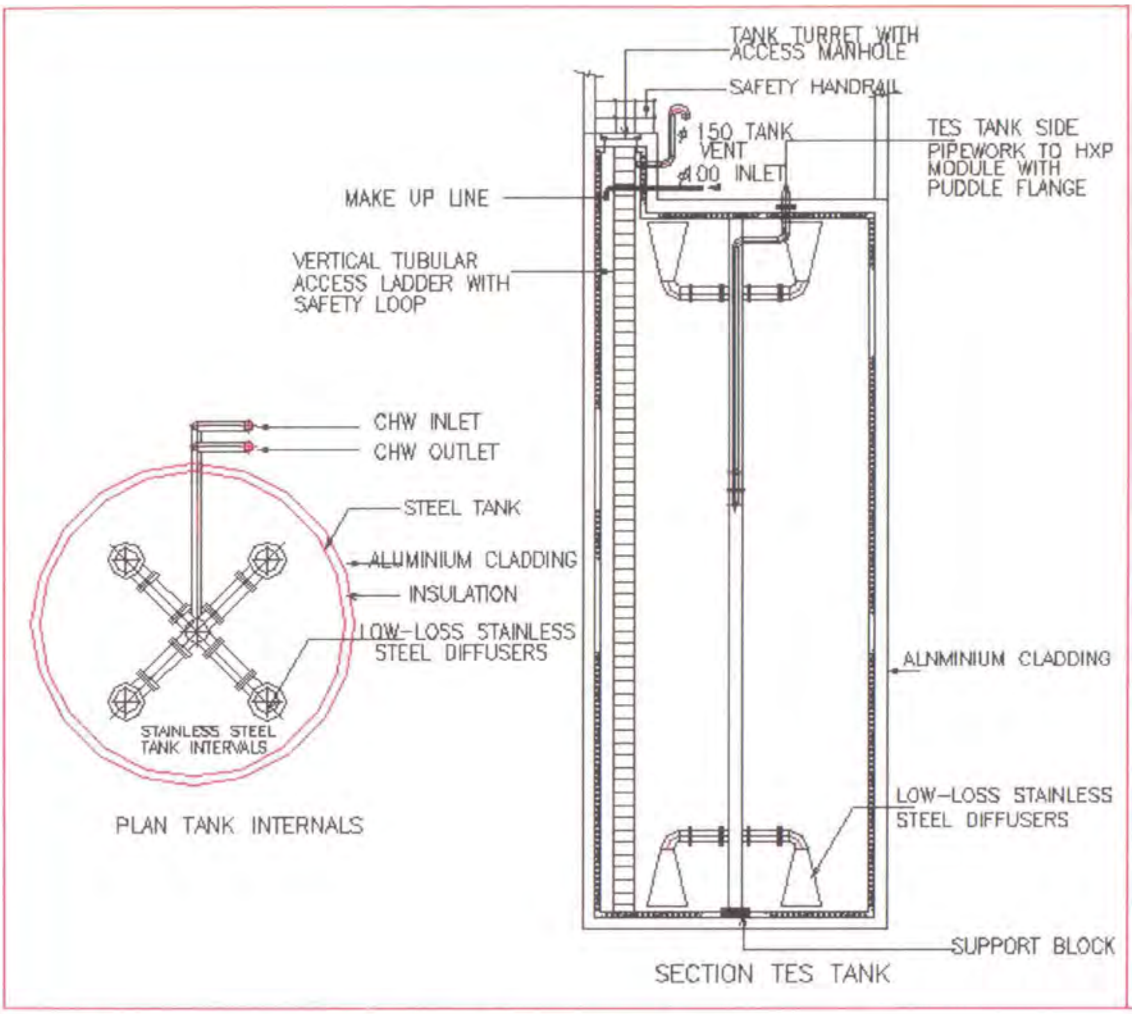

Fig.3.7 (Stratified Chilled Water Storage – Tank internals)

The mechanisms that operate in a stratified water storage system are related to both the physical properties of fresh water and hydraulic characteristics of stratification including pipe work.

The two physical properties of water that are of special interest to the researcher are:

1. The density as it varies with temperature
2. The kinematic viscosity as it varies with temperature

These two properties provide the basic mechanism for successfully stratifying water of different temperature within a single containment vessel. They also indicate to the researcher the location within the vessel where the warm water and cool fluids should be stored.[71]

It seems intuitively obvious that the cool liquid should be stored below the warm liquid within the vessel. The density difference between two liquids at different temperature creates buoyancy forces where the warm liquid is literally floated on top of the cool liquid. The relatively large difference in kinematic viscosity of liquids separated only by a few degrees in temperature suppresses any mixing of the two fluids, due to flow disturbances and free convection at the vessel walls.[70]

The strength of buoyancy forces in keeping the two fluid volumes separated is strong. This point is illustrated by a typical start up procedure that the researcher warm starts chill down cycle.

A new system that being started for the first time usually contains water that is substantially above the desired inlet temperature for the chillers. In Saudi Arabia this temperature might be as high as 40°C as in most cases well water is used for comfort cooling application. The chilling of the warm water needs special attention from the commissioning team with respect to demand limiting the chillers, to avoid over loading.

The researcher favours minimising the operational period with high inlet water to the chillers, as such portion of the tank, usually less that 20% is chilled in the typical charging mode of operation. The tank water volume is then rolled over by reversing the discharge mode. The removal of cold water from bottom of the tank and inserting it in the top creates strong mixing forces as the cold water sinks through the warm water back to bottom of the tank. The process progresses with remarkable speed and a thoroughness of mixing in the warm region but shows little penetration into the cold region. The vessel temperature sensors recording the event confirm this fact.

The two properties of water of interest both vary in the direction of increased temperature that make water the ideal candidate for TES system. That is density and kinematic viscosity both decrease with increased temperature above 4°C.

The design of stratified water TES system that successfully keep the warm and cool water volumes separated by a minimum volume of mixed water is a serious concern for all system designers. The performance of thermal energy storage vessels is defined as the Factor of Merit. The term is a measure of the Thermodynamic Availability of the fluid in the tank, to the system that it serves. A typical example is used to illustrate the above.

Presume that an air handling system has been installed such that it can properly process that air using an inlet refrigerated water temperature of 6°C. Water inlet temperatures above this level create operational difficulties for the air system and are thus to be avoided. The 6°C inlet water temperature provides the required Thermodynamic Availability for the air handlers to properly operate. It has been determined, through experience; the temperature below 6°C also provides Thermodynamically Available cooling, to the Air Unit.

Further, presume that a thermal energy storage vessel has a thermodynamic Availability of 90 percent, that is, the internal pie work design for stratification induction, is such that as the beginning of the discharge cycle, ten percent of the tank volume is raised above 6°C as the pie work system seeks to establish the thermo-cline. This wasted volume is a parasitic energy consumer with no benefit to the system, and must be reprocessed during each charge cycle.

The factor of Merit is a function of both stratification header design and the aspect ration of the storage vessel, again consider an example; Presume that a stratification header can create a thermo cline during discharge that is 1 metre thick. Further, consider this one-metre thick thermo cline as a percentage of the tank height. A one metre high tank would have a factor of merit of zero percent, while a 100 metre high tank would have a factor of merit of ninety nine percent. Large diameter tanks with small height to diameter ratios are inherently less efficient that small diameter high vessels. The process of the variable temperature water in the thermo cline presents some operational challenges for the process equipment and its control system. The thicker the thermo cline the longer the challenge continues.[20]

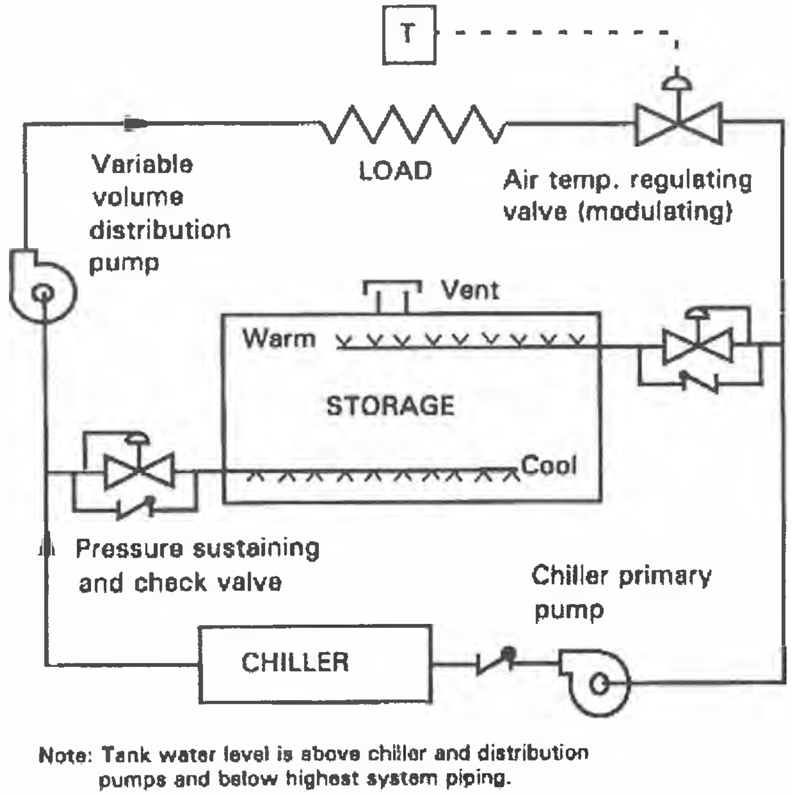

Fig. 3.8 Typical Chilled water Storage System Design (*courtesy of ASHRAE*)

3.3.1 Stratification Header Design

- Thermo Cline Thickness:

The thickness of the thermo cline is a function of the stratification header design and the flow rate through the header. Typical thickness varies from 0.7 to 1.5 metres.

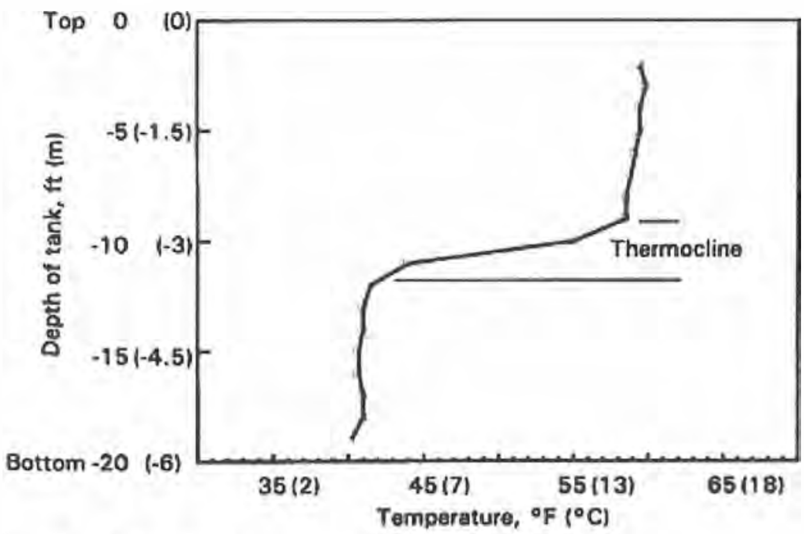

Fig. 3.9 Typical Thermo-cline (*courtesy of ASHRAE*)

- With a good header system, the harder the system is pumped, the narrower the thermo cline becomes, up to the point where the water collision velocity with the tank shell equals the fall rate of the thermo-cline. Beyond the critical velocity sag is introduced into the thermo-cline causing ripple action and thermo-cline thickness growth. A good thermo-cline is 0.5 –0.8 metres.
- Hydraulic Pressure Drop:

The pressure drop in a good stratification header system should be within 50-75 Pa/m of pipe at rated flow. The design utilises a directed flow splitting approach to evenly distribute the water to the expulsion nozzles. The nozzles are distributed radically around the area centriod of the vessels and the directed flow dividing elements of the system assure that each nozzle is fed with the same volume of water. The flow velocity is gradually reduced in the header system from 1.5m/s in the feeder pipe work to 0.1m/s at the nozzle exit. The large circular nozzles expel the water parallel to the water surface or the tank base, inducing secondary water movement only in a parallel layer. The results will produce superior hydraulic and thermo-cline performance.

3.3.2 Design and Operation Considerations

- Dissolved Oxygen:

The stored water with the TES tank contains dissolved gases, with the potentially most destructive, being oxygen. The effect of the untreated dissolved oxygen on the heat exchanger surfaces and the pipe work is to dramatically shorten the services life of the system.

- Dissolved Solids:

The stored water within the tank constrains large quantities of dissolved solids. These solids tend to precipitate out of solution in the heat exchangers that raise temperature of water, such as refrigerant condensers and air handlers.

- Biological Growth:

The stored water within the tank potentially host a wide variety of micro-organisms. The temperature of the stored water has a different effect on the seriousness of the microbe problems. In classical TES system using refrigerated water at 4 – 12°C the microbe growth has proven to be virtually none existent. The storing of condenser water at 25-35°C, will undoubtedly foster organic growth.

- Evaporation:

The air cushion above the water level at the top of the tank will in the case of a condenser water system provide a large opportunity for evaporation due to the high relative humidity in this area when compared to the outside air. The resultant vapor pressure differential will drive the evaporation mechanism. The TES system using refrigerated water typically produces water in high humidity environment.[1]

- Flow Establishment:

The establishment of flow in a TES system, during the discharge mode in particular, is critical to creating a minimal thickness thermo-cline. It is preferred that flow be established smoothly from no flow to rated flow over several minutes. It must be remembered that the entire volume of the tank water is being accelerated from rest to some vertical velocity. Establishing flow too rapidly suppresses the pump suction pressure that can cause hydraulic problems.

- Dual Tank arrangements:

The charging and discharging of a dual tank (or multiple tank system) should be performed simultaneously. The charging or discharging sequentially causes a pressure gradient to exist between the two vessels when one tank is discharges and the other one charged. The pressure gradient equalises when the second tank is opened to discharge, often creating hydraulic shock in the system.

- Pressure Sustaining:

The chilled water storage system operates on open tank principle and pressure sustaining becomes necessary when the tank water is to be pumped above the tank water level. The difficulty with pumping the water above the tank level is caused by the auto siphon created in the return pipe work. The auto siphon creates an area within-in the pipe work of low pressure where dissolved gases can escape from the solution, and potentially inducing hydraulic shocks into the piping net work. The static lift up to the higher elements must be accounted for in the pump head selections.

- Benefits over ice storage

Chilled Water storage system allows instantaneous pumping as the system works on sensible energy storage; whereas ice storage needs melting which requires a reasonable contact/time as the system works on latent energy storage.
In addition the de-rating of chillers, separating production from distribution with large heat exchangers can impose serious financial burden on the cost of the system.

3.4 Characteristics of chilled water and an ice storage system

- Ice cool storage systems :
    - Uses latent energy storage
    - Compact size factory built tanks
    - Uses separate cool energy production and distribution circuits
    - uses glycol and requires complicated control system
    - uses special chillers
    - requires comparatively small volume of storage

- Characteristics of Chilled Water Storage System include:
    - Uses sensible energy storage
    - Large size site built tanks
    - Uses a single cool energy production and distribution circuit
    - Uses water and operate as an extension of the hydronic circuit.
    - Uses standard chillers
    - Uses larger volume of storage

3.4.1 Comparison

- **Compactness**

For example:

ice storage – (210kWh$_c$ per m^3) - Calmac 1190A ice storage tank 1.2mD x 2mH(210kWh$_c$ per m^3 of storage) ; has a dischargeable latent storage of 420 kWh$_c$ of cooling.

Chilled water storage - (11 kWh$_c$ /m3), a 10mDx12mH chilled water storage contains a dischargeable sensible storage at 4^{0}C of 10,500 kWh$_c$

Using ice for 10,500 kWh$_c$ would require 25 x Calmac 1190 or 50m^3 of ice storage tanks ; however using a single chilled water storage tank requires a volume of 1000m^3.

- **Foot Print, Aspect Ratio and Modularity**

A custom build chilled water storage tank has to maintain a minimum aspect ratio to create effective stratification. Most ice storage tanks are factory build maintain an average height of 2m allows modular installations gives flexibility on selection, installation and operation of the system. Encapsulated ice storage requires site built storage tanks.

In addition to maintaining an aspect ratio for suitable stratification, to avoid auto siphoning the height of a chilled water storage tank height has to be above that of the building (in case of SFB this would be 30m, translates to a diameter of 2m, technically not a feasible solution.)

3.5 Relation to the model project

The model project requires storage of approximately 3500kWh$_c$ to be discharged on full and partial mode. The system has to be coupled with the existing system through an injection circuit. As this project is the first of its kind in the Middle East the client preferred to use self-contained factory assembled and tested modules. The design brief was such that the thermal energy storage system has to be simple in design, easy to install, operate and to maintain. Also the system should provide maximum flexibility to change operation according to operational schedules imposed by the local power generating Authorities. As the project is a retrofit, the system should be able to integrate into the existing conventional air conditioning system without major difficulties. In addition the client also requested for a cost effective TES system for this project.

The chilled water storage is based on open system, need to have all hydronic components below the water level of the tank to avoid 'run-down' or auto siphoning of system water into the tank. At BSF most of the existing Air handlers, pumps and chillers were installed at the roof level. To create a balanced hydronic open loop with the required system pressure the minimum height of the tank to be at least 34 meters. This would relate to a tank diameter of 2m to hold the minimum required storage of 3500kW_c.

3.6 Selection of the Suitable TES system for the Model Project

All the above constraints imposed by the model project favours an ice storage system. Considering all the above the researcher has selected the factory build ice on coil internal melt system for this project which gives a balanced benefit of all the above requirements. At the time of undertaking this project there were no ice storage project in use in the Middle East.

Considering the location of the ice banks at the project, the researcher has selected the externally insulated ice tanks to minimize any heat gain and also to eliminate condensation.

The selected storage tank is an externally insulated polyethylene tank with counter-flow heat exchangers. (refer Chapter –6)

CHAPTER-4

Model Project - Load Management with Cool Storage for an office building in Saudi Arabia

Fig.4.1 Saudi French Bank Headquarters

4.1 Introduction

The headquarters of Saudi French bank located in the central district in Riyadh, Saudi Arabia, was constructed during late eighties (Fig.4.1). This NW facing, elegant glass façade multistory building of approximately 12,000 m^2, has eight upper floors of offices occupied by the bank staff and two basements mainly used for car parking and storage. The average occupied area per floor is around 1250m^2. This concrete building has external marble cladding and a fenestration ratio of approximately 45%. All floors are provided with suspended ceiling with an average void of 750 mm, and the void is used as return air plenum.

Height	No. of Floors	Fenestration	Average Floor Area	Plenum
30m	9(F) + 2(B)	45%	1250m^2	750mm

Table: 4.1 General Building Data

4.2 The Comfort cooling system and the accumulated growth of cooling demand

The existing Central Utility Plant (CUP 1) chillers at Saudi French Bank was approximately fifteen years old , consist of two reciprocating air cooled chillers (both on duty) rated at 700 kW each, and three heat pumps each of 210kW with a nominal total on line rating of 1800 kW cooling. The building is maintained at positive pressure through a balanced airflow between exhaust and fresh air supply. A dedicated plant room on the ninth floor houses the air handling units, chillers and the necessary pumping, power and control modules for the air conditioning system. The existing air conditioning system was design in a manner that it incorporates all necessary energy management features such as variable air volume system, total building management, localized temperature controllers and night time temperature offsets.

4.3 New Constraints on electrical Power Consumption

Saudi French Bank wanted to expand their chilled water production plant to cope with the additional cooling requirements resulting from the expansion and occupation of the building.

In addition, in 1995 several new limitations on power supply and utilization have been imposed by SCECO specifically in centrally air-conditioned commercial and office buildings. To curtail the peak demand SCECO imposed a DSM program and it was necessary to reduce the building power demand by 40-50% based on the building cooling demand, by turning off at least 50% of the operating chillers between 1.00 p.m. until 5.00 p.m., the peak power demand period.

4.4 Historical growth of Cooling Load at Saudi French Bank

The building is totally air conditioned with a central system utilizing a Variable Air Volume (VAV) network. Shut-off type VAV units are used for the perimeter areas and Constant Air Volume (CAV) units are used for the core areas. The central HVAC system consists of 2 x 700 $kW_{cooling}$ air cooled chillers together with nine VAV air handling units and necessary pumping and control units and a high velocity circular ductwork system is used for air distribution. A Building Management System (BMS) is used for the control and monitoring of the HVAC system. The HVAC system is normally turned on in the morning around 7 a.m. and turned off at 8 p.m. One chiller is used at night to control the cooling base load of approximately 150 $kW_{cooling}$. The computer center is provided with a stand-alone cooling system with a chiller capacity of 150$kW_{cooling}$. Average power consumption of each chiller was around 0.5$kW_{power}/kW_{cooling}$.

Since 1987, the occupancy, use of space and supporting facilities such as computer rooms has increased substantially and the cooling demand has exceeded the maximum capacity of the original comfort cooling system. Several split a/c units are also used for certain areas such as battery rooms, Automatic Teller Machine (ATM) room, maintenance offices etc. These units have been added as retrofits approximately 3-4 years after the initial occupancy. These additions have utilized all the pre-existed spare transformer capacity on the electrical sub-station of the building. Continuous retrofit works on the building were carried out including conversion of un-conditioned areas for office space and the addition of a large computer centre in the building. These major retrofits caused a major demand penalty on the building cooling system. Finally the existing HVAC system was unable to handle this additional cooling load mainly during peak summer period.

(Courtesy of BSF)

Year	Cooling Load ($kW_{cooling}$)
1987	1000
1992	1250
1996	1500
Expected in 2001	1800

Table: 4.2 - Historical Growth

Based on the new occupancy load and additional computer centre etc., the building cooling system had to meet a peak cooling load of approximately 1400 kW, where as only 1000 kW net cooling was available, after up-grading of the chillers.

BSF has expressed a need to establish a new Central Utility Plant (CUP) with a new control system to couple the new CUP 2 with the existing CUP 1. The control selection of all CUP 1 and CUP 2 chillers and other hydronic equipment will be carried out from the central control panel provided in the vicinity of the existing CUP 1 chillers. This request was closely investigated by the researcher with control specialists and was included as part of the CUP-2 program.

A separate location has been identified for the CUP 2 chiller plant. It has been agreed that a set of new chillers with all needed hydronic components will be installed at the roof area. Inter-linking the CUP 1 and CUP 2 chillers was considered appropriate, as it would provide operational flexibility and will allow the CUP 1 cooling demand to be transferred progressively to the CUP 2 chillers. It is expected that the existing CUP 1 chillers will be partially taken out of service and only allow them to operate as either follow-on or stand-by chillers.

The researcher carried out a new building load analysis by using Carrier E20-II load analysis programme (Refer to Chapter-6)

It has been established that the combined capacity of the cooling system must be able to handle a maximum peak load of 1800 $kW_{cooling}$.

Essentially, a new Central Utility Plant (CUP 2) had to be established to add at lest 400kW additional cooling to the existing maximum demand and CUP 1 and CUP 2 together must be able handle a cooling demand of 1800 kW by 2001.

BSF anticipate that there will be major problem obtaining electrical power for the new development. In anticipation of introduction of cheaper tariffs for night-time electricity, the researcher recommended Thermal Energy Storage. TES system principles are discussed and in analysed further in Chapter-5 The task of the CUP-2 program was to develop a parallel cooling circuit with all necessary chillers, pumping and control modules, but within the available power limit of $100kW_{power}$.

4.5 Design Constraints - Retrofit Cooling And Electrical Load Management

The researcher observed that there was a fundamental design error in the selection of the existing chillers. The original HVAC design for the existing building was carried out in Switzerland and the chiller selection was based on ARI* rating (30°C Ambient), where as it should have been selected for an ambient condition of 50°C. This basic error caused a drastic de-rating effect on the chillers during hot summer months where the ambient condition reaches around 50°C, and the building cooling demand reached a maximum. i.e. the building demand peaks while the chiller production capacity at it's lowest. To bring down the inlet air temperature the maintenance department installed a spray cooler around the chiller to induce some evaporative cooling. The approach produced some improvement in the chiller performance at the early stage. However, over a period of time hard water used for evaporative cooling caused scale formation on the condenser coils resulted in further de-rating on chiller capacity.

Researcher approached the manufacturer to supply and install a high-ambient kit for the chillers with new condensers. Considering the age and model of the chillers, the manufacturer declined to install such up-grading system. Hence, to obtain the best performance out of these chillers, the condenser coils were replaced only with high ambient kits which allows the chillers to operate without shut-down at high ambient conditions like 42°C and above. New spray coolers were provided with a reverse osmosis plant and a standard iron filer to reduce the TDS of the spray water from 750 to 350, thus avoid formation of scale on the condensers. The up grading of spray cooler resulted in increased performance of the chillers, but still below the designed capacity.

In addition, in 1995 several new limitations on power supply and utilization have been imposed by SCECO specifically in centrally air-conditioned commercial and office buildings. To curtail the peak demand SCECO imposed a DSM program and it was necessary to reduce the building power demand by 40-50% based on the building cooling demand, by turning off at least 50% of the operating chillers between 1.00 p.m. until 5.00 p.m., the peak power demand period.[6]

The following options were considered to find a solution to meet both the additional cooling demand and to comply with SCECO's new Demand Side Management of electrical load.

- Install an on-site prime power generating equipment to energize a new air-conditioning system .
- Install a Thermal Energy Storage (TES) system by utilizing the available spare capacity on the transformer.

A check on the existing building transformer capacity revealed that the limit has already reached due to the retrofit works outlined above and the available spare capacity was limited to a maximum of 100kVA.A new transformer of approximately 250kVA would needed to energize a new cooling circuit. However, The blanket DSM program introduced by SCECO has totally eliminated all opportunities of increasing the building power demand. To curtail the peak demand SCECO imposed this DSM programme it was necessary to reduce the building power demand by 300 kW_{power} by turning off at least on large chiller between 1.00 p.m. until 5.00 p.m.

On the other hand, to meet the present cooling load an increase in power demand in the region of 200 kW_{power} was foreseen. The increase in power demand was needed to incorporate a new air-cooled chiller and the necessary hydronic and air distribution system. Water cooled chillers would have been an option as they consumed less than 35% power compare to air-cooled chillers. As water is in short supply and an expensive commodity in the Gulf, cooling towers are prohibited by the authorities in air conditioning system for commercial and office building; hence the use of water-cooled chillers was not considered as an option.

Initial load estimate showed that the increase in cooling load during peak summer period is around 350 $kW_{cooling}$ over and above the available cooling capacity of the system.

As outlined earlier the validity of the two solutions were evaluated; either an on-site prime power generating set to energize the new air conditioning system or a Thermal Energy Storage (TES) system. Due to the relatively small requirement for cool storage in the range of 6700 kWh (nominal) and the limited space availability (volumetric space being a critical issue for this building) the chilled water storage option was found to be uneconomical compared to the ice storage system.

Careful consideration was given to on-site power generation. However, the building owners have eliminated this option due to the associated problems in relation to noise, vibration, environmental pollution, and operation and maintenance aspects. On site fuel storage also considered as fire hazardous. At the time of this proposal, on site turbine power generation would have cost the 30% less than of grid power per kWh, but the capital cost of installing the system would have coasted GBP:700,000/- (far higher than the ice storage option finally chosen). Furthermore the operation and maintenance costs needed to be considered, whereas the ice storage requires minimal maintenance and practically no operating costs.

Ice Storage had not previously been considered as an option in Saudi Arabia until this time due to lack of confidence and available of expertise in system design and installation. Historically there has been a lack of incentive for use of cool storage due low electricity cost, and non-existence of demand charges, utility rebate or day and night time rates.

However, the researcher proposed ice storage as a design solution for the Saudi French Bank as it provides valuable load shifting and demand reduction features. The cost effective ice storage option was implemented, and provided a good outcome with the necessary additional cooling, DSM compliance and savings on electricity bills. (Refer to end of chapter-3)

CHAPTER-5

Basic Principles of Thermal Energy Storage (TES) System and Cooling Load Calculation Procedures.

5.1 Introduction

Comfort cooling for buildings is based on a simplified heat transfer process between refrigerant- air, refrigerant-water and water-air. A simplified schematic diagram (Fig 5.1) illustrates the basic concept of air conditioning. A chiller is a cold generator that operates on Carnot principle, through a cycle of compression and expansion together with a condensing process. At the end of the cycle, the chiller cools the water flowing through the evaporator.

The chiller is part of a hydronic network that consists of circulating pumps , air handlers and fan-coil units.

The pump circulates the cold water through the heat exchanger of the air handling units; as the air flow across the coils and the cold water flow through the coils, heat transfer take places cools the return air from the air handling to a pre set level. The air supplied through a network of sheet metal ducts, grilles and diffusers, pick up the heat gain from the occupied areas, while maintaining the resultant room temperature at a comfort level say, 24°C.

The process continuously repeats several times around the hydronic and air circuits as long as the system in operation, and there is a demand for cooling from the occupied areas.

5.2 Basic Principle of Thermal Storage (TES) (Also refer to Chapter –3)

Consider a conventional comfort cooling system is in operation for a period of time. Refer to Figure-5.1. While the system in operation, if the chillers are turned-off but allowing the pumps and the air-handling units to operate without interruption, one would expect the resultant temperature of the occupied area increases as soon as the chillers are turned off.

In reality, it would take approximately ten to fifteen minutes before the occupants of the building to notice any change in the performance of the comfort cooling system.

This is due to the fact that a large amount of cold water has been stored in the up-stream hydronic line of the chiller, which could continue cool through heat transfer for a period of time, based on the amount of water available in the net work and the number of air handling units are in operation.

If we can store a large amount of cold water in the chiller upstream network, on shut down of chillers the air handing units can operate for an extended period of time based on the system storage capacity.

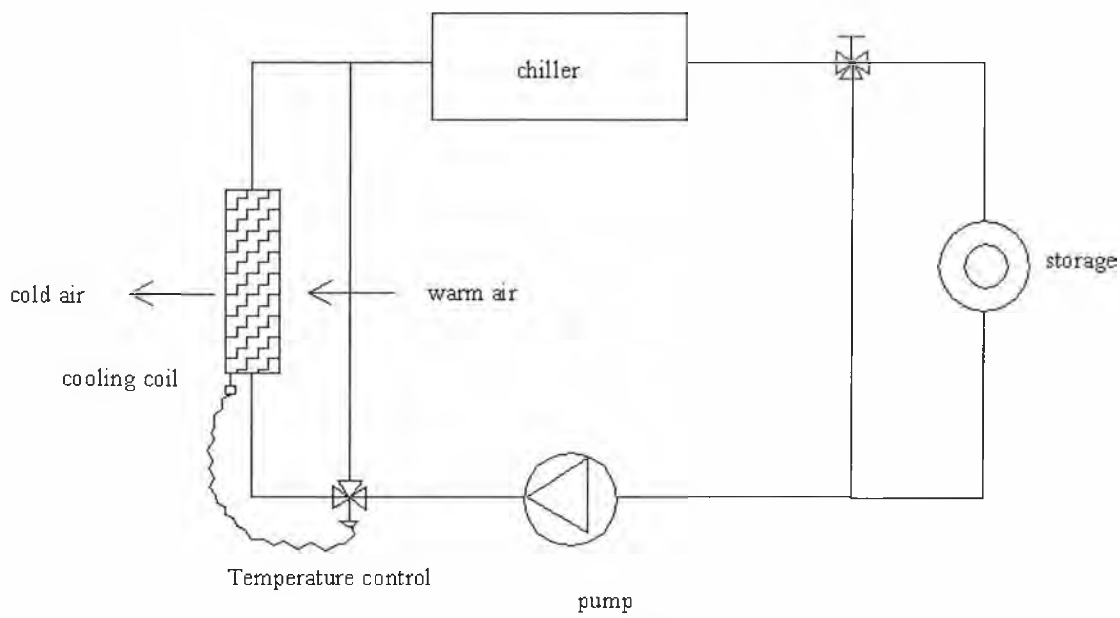

Fig.5.1 Simplified hydronic network of a TES based cooling system

This is the basic principle of cool storage or Thermal Energy Storage (TES) system.

The cool storage system uses the sensible or latent heat capacity of water to store energy. Storing the energy at night the installation not only does it not compete during the day for peak demand but at the same time utilises the idling excess capacity of the power plants. In doing so the demand profile becomes evened out thereby assisting the power generating authorities in demand management. Demand Side Management (DSM) utilising cool storage will de-couples the time of production from the time of consumption. The de-coupling of the two functions, creates the opportunity to demand manage a major portion of a facility's electrical demand. Furthermore, the shifting of the time of production and the elimination of production plant's output modulation requirements, create an opportunity for reducing power consumption.

Cool Storage has been used successfully for a long time in the North American and European counties for the specific reason of energy cost savings in comfort cooling systems. Cool storage became very economical as the storage tanks are normally charged during the night time where the electricity tariff is almost half of the daytime. Demand charges are higher in these counties and the peak demand reduction could save a large amount of money. Furthermore, the utility companies also provide cash incentive to the developers if they reduce the peak demand and shift most of the cooling electrical demand to the night time.

However, these favouring conditions vary from country to country and the conditions that would favour the use of cool storage in the North American countries, may not be acceptable nor in practice in elsewhere in the world.
In the absences of the above benefits in the Middle Eastern countries, it is important to establish alternate favouring conditions suitable for this specific region to make the cool storage as an acceptable concept in the air-conditioning industry.

To introduce cool storage successfully in the Middle East, some favouring conditions have to be established based on technical, economical, and environmental criteria.

5.3 Cool Storage Operating Strategies

Under investigation are the following two types of cool storage system operating strategies:

5.3.1 Load shifting

Load shifting involves producing and storing 'cool energy' in the night and use this stored energy during the peak hours of the day.[5] This would allow the user to shut off the chillers during the peak hours. The extended application of this strategy would be the *full storage*, [13] which involves in storing the entire daily cooling requirement to achieve maximum load shifting. Full storage systems, also known as load shifting systems, are designed to meet all on-peak cooling loads from storage. On the peak demand day, the chiller in a full storage system operates at its capacity during off-peak hours to charge storage and meet cooling loads occurring during off-peak hours. This type of system results in larger and, therefore, more expensive chiller and storage units compared to partial storage systems. However, full storage also captures the greatest savings possible by shifting electricity demand from on peak to off-peak. Full storage systems are relatively attractive when demand charges are high; the differential between on-peak and off-peak energy charges is high and/or when the peak demand period is short, such as dairy plants. This would be an expensive cool storage option for buildings in the Gulf since the high ambient temperatures of the region would require a very large storage volume.

5.3.2 *Load levelling*

On a Load levelling, *partial storage* or peak-shaving cool storage application, some active chiller load is required to satisfy the building load during the daytime.[42] When there is no or minimum cooling load in the night the chiller will charge the storage tank. During the day time the system will start to operate as a conventional system . As the building load progressively increases beyond the chiller capacity the required additional cooling will be supplemented by the depletion of stored cooling from the storage tank to meet the load requirements.

This would be the most suited cool storage application for the Middle East region. Partial storage involves down sizing both refrigeration and storage to the point where a combination of the two will just meet the peak demand. A major potion of the conventional system will reduce the installed chiller capacity, and the storage tank will be sized to hold several hours of the peak-cooling load.

Demand limiting is an extension to partial load shifting programme. In this application the chiller capacity further reduced during the DSM or prescribed peak load period. This option is utilized at the model project.

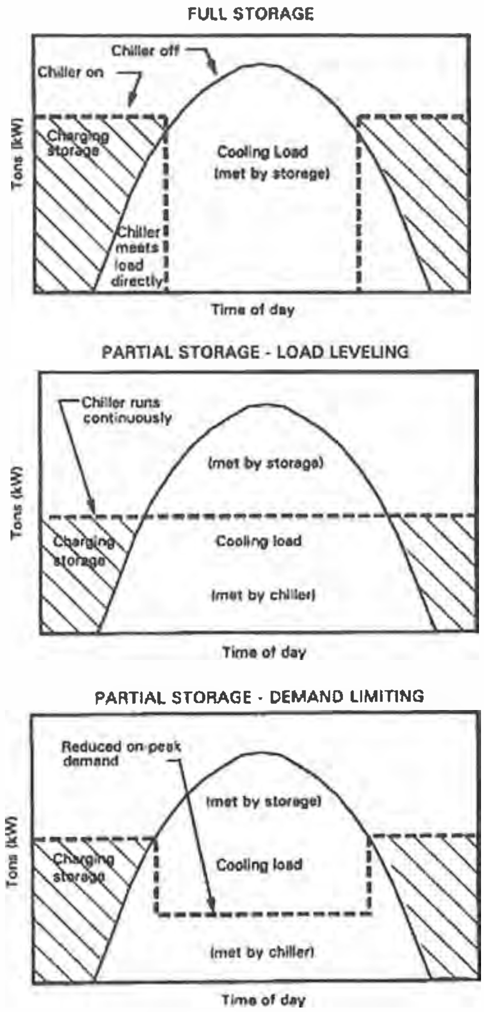

Fig.5.2 Operating Strategies

5.3.3 *Operating priorities*

The first option would be to make use of all the stored ice or chilled water on daily basis, and operate the chiller only as a backup - this is often referred as *ice or chilled water priority*. The second option would be to use the chiller to lead the cooling operation and only use the stored ice or chilled water for topping up purposes - *chiller priority*.

The most preferred strategy in North America & in Europe is the chiller priority; i.e. to use the stored chilled water as little as possible, only for topping-up purposes. The charging hours will be reduced and the stored chilled water will be used as a backup whenever the chillers cannot handle the load.

In the Gulf, where there is no variable electricity tariff, the chilled water priority becomes more attractive since it reduces the operating cost. The control system is complex compare to the chiller priority since it is involved in weather prediction, intelligent usage of stored energy to avoid the chances of running out of chilled water too soon and have to rely on undersized chillers to handle a larger load at a later stage of the day. The introduction of computer based storage management control systems are currently available in the market to overcome this management problem related to cool energy discharging.

By introducing cool storage in a conventional hydronic system the researcher studied and investigated the possibility of reducing the installed chiller capacity and to establish the economical combination between active chillers and the cool storage in air-conditioning system

As water is a precious commodity in the Middle East, in certain member countries the use of water-cooled chillers are prohibited. Air cooled chillers are the most preferred chillers in the Middle East though they are very inefficient in comparison to the water cooled chillers. The average energy requirement for an air cooled chiller is approximately 0.45-0.5 kW per kW cooling compare to a water cooled chiller that only requires 0.14 - 0.17 kW per kW cooling.

If considerable reduction in installed chiller capacity can be achieved by utilising a cool storage system, this would result in major electrical demand reduction in buildings.

The average temperature swing between day and night in the Middle East is around 16°C. In Riyadh, the capital of Saudi Arabia the temperature swing is around 20°C.

The researcher is investigating in taking advantage of the improved chiller efficiency due this temperature swing, hence, saving electrical energy by shifting the chiller operation from peak day time to cooler night time.

The energy conservation option would allow the researcher to investigate the option of shifting the cooling electrical load from peak day time to off peak night time. This would allow the power generating companies to level the electrical load demand evenly over the 24 hours period.

In a conventional cooling application the installed chiller capacity will represent the peak load and a percentage of the peak load to provide some standby capacity. In a cool storage application the researcher will study the option of down sizing of chillers and thereby, down sizing of transformers and associated electrical services.Fig-12. If the chiller capacity reduction can be achieved, then a major cost savings can be also achieved on building power connection charges due the reduction in the electrical peak demand.

The improvement in chiller efficiency by shifting the cooling load production from day to night will result in a considerable savings on total electrical consumption. The charging of tanks normally takes place in the night when the ambient temperature is far below its daytime peak. Thus an improved kW electrical energy per kW cooling ratio is expected.

Maintenance is a major aspect in most of the Gulf counties. A large sum of money is spent every year on preventive maintenance of the air conditioning equipment. The average cost of maintaining a liquid chiller is approximately 5 to 7 Sterling Pound per kW cooling per year taking into account of the useful chiller life of approx. 20 years. as the cool storage is expected to reduce the installed chiller capacity , this would yield in substantial savings on maintenance charges.

In the new DSM system design the chillers will be running on full capacity on a cool storage application. Intermittent cut-offs and inefficient partial load operations will be minimised. Hence, the chillers will last longer in comparison to their life expectancy in a conventional air conditioning system.

The Montreal Protocol to which most of the Middle Eastern countries are signatories, called for a ban of CFCs in 1995, and a phase out of all HCFCs by year 2030; Replacing the existing CFC & HFC based refrigerants with the new refrigerants such as R-123 & R-134, could result in reduced chiller efficiency. In larger facilities, this could cause a major reduction in system capacity. The research will also investigate the option of incorporating a cool storage system that would upgrade the system capacity up to 35% without adding any extra chillers.

5.3.4 *Plant Starvation Syndrome*

Plant Starvation Syndrome (PSS) is a hydraulic design deficiency that requires more chillers to be in operation than necessary, these chillers are operated at a 50-70 percent load; Thus, PSS is clearly consuming large blocks of electrical energy in the form of chiller production inefficiency, and pumping system.

Even though the chiller sections are based on full load capacity, the chillers are normally operated at partial load; the chillers consume higher electrical power when it operates under the specified optimum operating conditions. For example (a typical chillier selection data-reference)

The Chiller/TES matching of this project formatted in a manner that chillers always operate on full capacity, and the TES is used to handle the variable load over and above the operating chiller capacity thus minimizing any PSS in the system.

5.4 Cooling Load Calculations Methodologies

Prior to establishing any cooling loads it is important that an extensive survey be conducted to affirm proper evaluation of the load constituents.

Cooling load is established based on a mathematical analysis of series of heat gains into the specific zone.

Analyzing cooling load for a building is always an approximation as several factors used in the calculation are based on historical weather data and properties of material. In addition several set and recommended data of equipment, occupants etc be used in this analysis that are averaged or approximated. Hence the calculation of cooling load of a building should be established on good engineering practice rather than just following a guidance and using set data recommended by a standard or guide. Also it is important

to understand the result is an estimate and the completeness and accuracy of the calculation is based on the proper application of the above variable factors for the specific location and area where the building is located.
The data available for the M.E. is established within a shot period of time and still require up-dating and validation. Several data were established remotely from the location and validation and confirmation of these data is extremely important.

The data used in the analysis of cooling load for the Saudi French Bank were mainly derived from ASHRAE –Fundamental 1997, however several good engineering practice and knowledge gained from field experience are also used to establish the load.
ASHRAE has improved the calculation procedures and established three advanced cooling load methodologies. These are,
1.Total Equivalent Temperature Differential (TETD) Method
2. CLTD –CLF Method or ASHARE method
3. Transfer Function Method

5.4.1 *Total Equivalent Temperature Differential (TETD) Method*

The conventional method widely used by engineers for a long period of time. The methodology uses the sole air temperature as the datum and to use several response factors for representative walls and roofs assemblies and various heat gain components are added to average an instantaneous cooling load by time averaging. The time averaging involve in establishing the radiant portion of the heat gain load by relating it to the preceding hours. The engineers due to its complexity, poor approximation and assumptions did not prefer the methodology.

5.4.2 *CLTD –CLF Method or ASHARE method*

Rudoy and Duran conducted the initial research in 1975 by comparing the TETD/TA and TFM.

Using the data derived from the TFM they have generated a set of cooling load temperature differential data for direct one step calculation.
ASHRAE sponsored the research and the first set of data was published in 1989.
Compare to the other two methods CLTD-CLF method allows the designer to use scientifically developed factors to establish the cooling load. However, there are several limitations that caused dissatisfaction with the users and ASHRAE has decided not to invest further into the development of this method and to leave this method at its present level and to target the research towards more promising methodologies.

5.4.3 *Transfer Function Method (TFM)*

The methodology was developed in 1972 by Mitals and adopted by ASHRAE as an alternative method to TETD. The methodology is based on set data called Conduction transfer function (CTF) and room transfer functions (RTF) using several weighting factors.
Several compromises are made in the development of RTF. The engineer has to use several set data developed by a third party to establish the cooling load, giving him little authority in the way the load factors are used in the calculation. Alternatively the engineer has to establish his own TFM data, which involves in extensive computer modelling to establish the heat balance. However, TFM is now become an easy task with the advanced desk top computer, thus allowing engineers to develop a more precise heat balance analysis.
ASHRAE endorsed TFM as the preferred method of computing cooling load for commercial, industrial and residential buildings.

5.5 *Commercially developed Software for Load analysis*

Introduction of desktop computers encouraged several individuals and organizations to develop software for cooling load analysis. Several such software are readily available in the market and the most common software are listed below:

5.5.1 *Trace*

Developed by Trane Company from the US. This is mainly a energy analysis software allowing the designer to establish annual energy cost and life cycle analysis of a cooling system. To develop the energy model the designer has to use the cooling load analysis program to establish the 24 hours / 365 days load profile. The option of using CLTD-CLF, TETD-TA or TFM is provided in the selection menu. However, the equipment selection is limited to Trane equipment only.

5.5.2 *E20-II*

M/s Carrier Co developed the software E20II used in the analysis. Carrier the father of air conditioning and the inventor of the centrifugal chillers established the company almost 70 years ago. He also developed the psychometric chart and the first format for establishing cooling load for a building. This method was called "Carrier Cooling method" and was widely used in the industry for a long period of time. This software is developed by Carrier Corporation to conduct engineering design and analysis. The programme also provides established data necessary to size and select cooling equipment based on TFM.

5.5.3 HEVACOMP

This software is developed by a group of engineers from the UK. The software was developed based on CIBSE design methodology. The software is widely used in the UK, and extensively applied in heating load analysis.

5.5.4 TAS

Another load analysis programme developed in the UK. Mainly used for thermal modelling of building. The analysis gives the option of using various methodologies including CIBSE and ASHRAE methods for cooling load calculations.

5.6 Selection of Appropriate Software Package

E-20II was selected as the appropriate software for the load analysis at the Saudi French Bank for the following reasons.

- The software is based on the most preferred Transfer Function Method (TFM)
- The software has been used in the industry over a period of 16 years and proven of its accuracy.
- The software was well suited for the specific application at the Saudi French Bank as the analysis required was to establish the block load of the building rather than the room-to-room analysis.
- The software provides detailed design parameters established for Saudi Arabia.

5.7 Project Specifics

- **Location and Climate**: The building is located in Riyadh, Saudi Arabia. The design data was established according to the 1997 ASHRAE Handbook of Fundamentals (SI Edition, Chapter 26, Table 3A).

- **Building layout**: The bank is a freestanding, multi story, rectangular building. A simple layout schematic is shown in appendix and the dimensions are identified in the load analysis report appendix(refer to room data sheet)
- **Exterior Wall construction**: A single type of wall construction is used for all exterior wall. Wall consists of concrete structure with marble cladding. The front elevation of the building is facing north, with other walls facing south, east and west. According to the 1997 ASHRAE Handbook of Fundamentals, SI Edition, Chapter 28, Tables 11, 12 and 13, the thermal properties, and transfer function methods are established.
- **Roof Construction**: A single roof construction is used, consists of built-up roofing, insulation, and concrete and suspended ceiling. According to the 1997 ASHRAE Handbook of Fundamentals, SI

Edition, Chapter 28, Tables 11, 12 and 13, the thermal properties, and transfer function methods are established.

- **Window / Glazing:** The building has 60% of the façade constructed of tinted, reflected double-glazing. According to the 1997 ASHRAE Handbook of Fundamentals, SI Edition, Chapter 27, Table 5, appropriate shading co-efficient and U value for this glass is established. Chapter 28, Table 25 is used to establish transfer function co-efficient. In addition the manufacturer's technical data also used.
- **Occupancy:** Maximum occupancy of $10m^2$/person is assumed. This value is in the range appropriate for office buildings. From ASHRAE Handbook of Fundamentals SI edition 1997, Chapter 26 Table 3 , people heat gain of 75W/person sensible and 55W/person latent are obtained. These values are for light work category, which ASHRAE recommends for office buildings. Finally, no people will be present in the building during the night time.
- **Lighting:** A lighting intensity of $22W/m^2$ is used for the office lights are fluorescent with an efficient ballast starter device. A wattage multiplier of 1.08 is used to account for the starter device. The light fixtures are recessed into the acoustic ceiling and are vented. During the night shutdown period, 10% of the lights are left on for security purposes.
- **Electrical appliances**: The office contains several other electrical appliances such as computers, printers, etc. However, the major load factors in certain critical areas were induced by the internal heat gain generated by a large number of these appliances. In general, 15W/m2 is used for electrical appliances, but in certain areas up to 50W/m2 is used where a large number of computers are installed in one room or area.
- **Infiltration**: Fresh air brought into the building by the air handlers and exhausted via ex-filtration through toilet extracts, and doors. The windows are airtight and no ex-filtration is considered through windows. The building is maintained at positive pressure to keep dust and hot air infiltration.

- **Slab**: The office is built with two basements and on a raft foundation. The dimensions of the slab are the same as those of the building.
- **Equipment**: Variable Volume air handling units, air-cooled chillers, and in certain areas split type air conditioners are used. There are three heat pumps installed for providing heating during wintertime by reversing the cooling cycle.
- **Set Points**: For cooling a set point of 24°C is used throughout the office areas. For supply air sizing, a supply temperature of 14°C is used. For ventilation 10L/s per person is used. (See appendix)

5.8 General Components Load Analysis

The cooling load is the sum of a number of transmission, infiltration, solar and internal load components as defined above.

$Qs = [Qw + Qg + Qr + Qp + Qslr + Qpe + Qeq + Qms + Qi] \{ 1 +Fss/100]$.(Eq. 5.1)

Where,

Qs = Total sensible load (W)

Qw = Wall transmission load (W)

Qg = Glass transmission load (W)

Qr = Roof transmission load (W)

Qp = Partition transmission load (W)

Qslr = Solar gain load (W)

Qlt = Lighting load (W)

Qpe = People sensible load (W)

Qeq = Equipment load (W)

Qms = Miscellaneous sensible load (W)

Qi = Sensible infiltration load (W)

Fss – Sensible cooling safety factor.

The transfer function methodology takes into account the transient heat transfer characteristics of all components loads in the building. The summary of transfer function load calculation procedure is defined in table 10, Chapter 28, 1997 ASHRAE Handbook of Fundamentals, SI Edition.

5.9 Transfer Function Method For Block Load Analysis

The following describes design load calculations and equipment-sizing procedures used in the development of "Block Load Analysis" for Saudi French Bank Headquarters building based on the Transfer Function Method (TFM) described in ASHRAE Fundamental 1997 Chapter-28 pages 17-39.
This procedure was chosen for the model project because it takes into account the transient heat transfer characteristics of all component loads in the building. All other available load analysis methodologies set to calculate the instantaneous peak cooling load, rather than the time averaging 24 hours cyclic load.
Currently, Transfer Functions are recommended by ASHRAE as the preferred means of computing hourly loads. This method has several parallel similarities with the CIBSE load analysis methodology.
The method consider several weighting factors, including applying conduction transfer function (CTF) and considering the difference between the sol air temperature and inside mean average temperature, which includes all effects due to reflection conduction, radiation etc.

Maximum cooling loads are determined by using an hour-by-hour calculation approach. Loads for a number of hours specified by the user are examined. Maximum cooling loads are identified from among this group of hours and are then used to size the cooling system. A summary of TFM load calculation procedures is listed in ASHRAE Fundamental 1997 Chapter-28 page-18.
However, as Saudi French Bank Headquarters is an existing building and the up grading of the cooling system is mainly examining the difference in the actual peak load and the maximum capacity of the HVAC system, a check on the block load of the cooling system would be adequate. Hence as art of the TFM analysis a simple straightforward single-zone load analysis was also conducted. The total building is considered as a single zone for the block load analysis, with a time averaging factor. This 24 hours weighted average allows

the designer to establish the equipment load including chillers and ice banks. The chillers represent the night time maximum charging capacity whereas the ice banks represent the total daytime discharge capacity.

Cooling analyses for single-zone and multiple-zone systems differ slightly.

For single-zone HVAC systems, the analysis involves two stages:

5.9.1 Zone Load Calculations. First, zone sensible loads were computed for the month of July and for hour 1600 normally the peak load time in Riyadh without considering the lag time or building thermal mass. A “zone load” is the amount of heat, which must be removed in order to maintain zone air at the cooling set point temperature.

5.9.2 Cooling Coil Load Calculations. Next, system operation was analysed in order to determine the system airflow rate and the sensible and latent cooling coil loads. Airflow and coil loads are determined for all the months and hours specified by the user.

Then the sensible and latent coil loads will be combined to determine the total cooling coil load.

5.9.3 In the first step in the analysis zone sensible cooling loads were calculated. A zone load is the amount of heat, which must be removed from a zone in order to maintain zone air at the cooling set-point temperature. This load is the sum of a number of transmission, infiltration, solar and internal load components:

$$Qs = [Q_w + Q_g + Q_r + Q_p + Q_{slr} + Q_{lt} + Q_{pe} + Q_{oe} + Q_{ms} + Q_i] [1 + F_{ss}/100] \; ..... \text{(Eqn 1-1)}$$

Where:

Q_s = Zone sensible load (W)

Q_w = Wall transmission load (W)

Q_g = Glass transmission load (W)

Q_r = Roof transmission load (W)

Q_p = Partition transmission load (W)

Q_{slr} = Solar load (W)

Q_{lt} = Lighting load (W)

Q_{pe} = People sensible load (W)

Q_{oe} = Other electrical load (W)

Q_{ms} = Miscellaneous sensible load (W)

Q_j = Sensible infiltration load (W)

F_{as} = Sensible cooling safety factor (%)

It is important to note that a "zone load is different from a "coil load". A coil load is the amount of heat removed at the cooling coil. Heat removed at the coil is a combination of heat removed from the zone plus fan and ventilation heat gains plus portions of the plenum heat gains.

5.10 Cooling Coil Load Calculations

In the second step of the process, system operation was analyzed in order to determine cooling coil loads. In this procedure airflow rates, dry-bulb temperatures and humidities at all key points in the system were computed. The ultimate goal is to determine the coil inlet and outlet conditions so that the coil load can be calculated. Coil loads are determined for a number of hours and the maximum coil load among these hours is then identified.

This analysis is performed in two parts. The first involves a sensible analysis of the system, which leads to a calculation of the sensible cooling coil loads. In the second part, system humidities are determined in order to compute the latent cooling coil load. Temperatures, humidities and airflow rates referred to in these discussions are tabulated in Appendix-2.

5.10.1 *Sensible Load Analysis*

- Calculate Supply Air Characteristics. First, the zone sensible load is used to determine either the supply airflow rate or the supply temperature. Different calculations are required depending on which supply characteristics are defined during HVAC system inputs.

- Supply Airflow Rate Defined as L/s or L/s/sqm. In this case the specified airflow rate will be used for all hourly cooling coil load calculations. For each hour, a required supply air temperature is computed by solving equation 1.2 for Ts.

$$Q_s = P_a V_s C_{pa} F_u (T_c - T_s) \; ..... \; \text{(Eqn 1-2)}$$

Where:

Q_s = Zone sensible load (W)

P_a = Density of air. Value is adjusted for site elevation.

= $P_{sl} P_{ba} / P_{sl}$

V_s = System supply airflow rate (L/s). For the case in which supply air is specified as L/S/sqm this value is multiplied by the total floor area served by the HVAC system to obtain the system supply air flow rate.

C_{pa} = Specific heat for air (1004.832 J/(kg-K)).

F_u = Units conversion factor.

= m3/(1000 L)

T_c = Cooling thermostat setpoint temperature (C)

T_s = Supply air temperature (C)

P_{sl} = Density of air for standard sea level conditions (1.201 kg/m3)

P_{ba} = Standard atmospheric pressure at site elevation (kPa)

= 101.3 (1-2.25569x10-5E)5.2561

P_{sl} = Standard atmospheric pressure at sea level (101.3 kPa).

- Supply Temperature Defined. In this case the temperature will be used for all hourly cooling coil load calculations. For each hour the required supply airflow rate is computed by solving Equation 1.2 for V_s.

5.10.2 *Calculating the Supply Fan Heat Gain*:

Based on the system supply air flow rate, the fan heat gain can be computed next. Fan heat gain is due to friction between air and the fan blades, energy added to the air by compression, energy lost in the fan drive mechanism, and energy losses in the fan motor. Depending on data supplied by the manufacturer, fan heat gain is calculated in one of three different ways:

a. If a total pressure is defined for the fan, heat gain is determined using the equation.

$$Q_r = F_{fu} V_s T_{sp} / nt \quad \text{..... (Eqn 1-3)}$$

where:

Q_r = Fan heat gain (W).
V_s = Supply airflow rate (L/s)
T_{sp} = Total static pressure across fan (Pa).
N_f = Combined fan drive, mechanical and motor efficiency (dimensionless). A value of 0.54 (i.e. 54%) is assumed.
F_{fu} = Units conversion factor.
= m3/(1000 L)

b. If a fan brakehorsepower is specified , heat gain is computed using the equation:

$$Q_f = F_{fu}\ BHP / N_m \quad \ldots\ (\text{Eqn } 1\text{-}4)$$

Where:

Q_f	=	Fan heat gain (W)
F_{fu}	=	Units conversion factor.
	=	745.7 W/bhp
BHP	=	Fan brakehorsepower specified by user.
N_m	=	Fan motor efficiency (dimensionless). A value of 0.90 (i.e. 90%)

c. Finally, if fan kW is directly specified , heat gain is computed using the equation.

$$Q_f = F_{fh}\ P_f \quad \ldots\ (\text{Eqn } 1\text{-}5)$$

Where:

Q_f	=	Fan heat gain (W)
F_{fu}	=	Units conversion factor.
	=	1000 W/kW
P_f	=	Fan kW specified by user.

5.10.3 *Calculating Coil Outlet Temperature:*

Based on the values computed thus far, a coil outlet temperature can be determined next. The coil outlet temperature will be used later to compute the coil sensible load. Two different calculation procedures are used depending on whether the supply fan configuration is blow-thru or draw-thru; In the case of Saudi French Bank the supply fan is a draw through type and equation 1-6 was used.

a. Blow-Thru Fans. For this fan configuration, the coil outlet temperature is lower is equal to the supply air temperature, T_s.

b. Draw-Thru Fans. For this configuration, the coil outlet temperature is lower than the supply temperature due to fan heat gain. The coil outlet temperature is computed using the equation:

$$T_{co} = T_s - Q_r / (p_a C_{pa} V_s F_u) \ldots \text{(Eqn 1-6)}$$

Where:

T_{co} = Coil outlet temperature (C)
T_s = Supply air temperature (C)
Q_f = Supply fan heat gain (W)
P_a = Air supply (kg/m3). Values are adjusted for elevation.
C_{pa} = Specific heat of air (1004.832 J/(kg-K)).
V_s = System supply airflow rate (L/s)
F_u = Units conversion factor.
= m3/(1000 L)

5.10.4 ***Calculating Ventilation Airflow Rate:***

This and the following four steps are used to determine the cooling coil inlet temperature. This process involves working forward from the point air exits from the zone to the point where air enters the cooling coil, solving for airflow rates and temperatures at each point along the way. First the ventilation airflow rate must be determined. Airflow is calculated in four different ways depending on how the user has defined ventilation air:

a. When the ventilation air per unit floor area (L/s/sqm) has been

$$V_v = V_{vf} A_{tot} \text{ .... (Eqn 1-7)}$$

Where:

V_v = Ventilation airflow rate (L/s)

V_{vf} = Ventilation airflow per unit floor area (L/s/sqm).

A_{tot} = Total floor area served by system (sqm)

b. When the ventilation airflow rate (L/s) is directly specified, no calculation is necessary.

c. When ventilation is specified as a percentage of the supply airflow rate, airflow is calculated as:

$$V_c = V_s F_v/100 \text{ ... (Eqn 1-8)}$$

Where:

V_v = Ventilation airflow rate (L/s)

V_s = Supply airflow rate (L/s)

F_v = Percentage value specified by user (%)

d. Finally, when ventilation is specified on an airflow per person basis (L/s/person), the airflow is calculation as:

$$V_v = V_{vp} P \text{ .... (Eqn 1-9)}$$

Where:

V_v = Ventilation airflow rate (L/s)

V_{vp} = Ventilation airflow rate per person (L/s/person)

P = Total number of people occupying zone served by the HVAC system at maximum occupancy.

It is important to note that in some unusual situation, the ventilation airflow rate computed will be larger than the system supply airflow rate. This could occur when a fixed ventilation rate (L/s) and a supply air temperature are defined. For this combination of inputs, it is possible the required supply airflow rate that is calculated will be less than the ventilation airflow specified. In case of Saudi French bank load analysis an occupation load of 1 person per $10m^2$ and a ventilation rate of 10(L/s)/person was considered. However, in the actual installation process the owner has used fan coil units without any fresh air make up. The additional load considered for fresh air load was left as a safety factor in the equipment selection.

5.10.5 Calculating Direct Exhaust Airflow Rates

In office environment similar to the bank, air is directly exhausted from the zone via toilet extracts, kitchen hoods etc., before it enters the return duct or plenum. This airflow rate is determined in two different ways depending on the HVAC system inputs:

a. When the direct exhaust air is defined as a total airflow (L/s), no calculations are necessary.

b. When direct exhaust is specified as a percentage of the ventilation airflow, the direct exhaust airflow is computed using:

$$V_{de} = V_v F_{de}/100 \text{ .... (Eqn 1-10)}$$

Where:

V_v = Ventilation airflow rate (L/s)

V_{de} = Direct exhaust airflow rate (L/s)

F_{de} = Direct exhaust percentage value specified by user (%).

5.10.6 *Calculating Return Airflow Rate:*

Next, knowing the direct exhaust airflow rate, the return airflow rate can be calculated using the equation:

$$V_r = V_s - V_{de} \quad \text{.... (Eqn 1-11)}$$

Where:

V_r = Return airflow rate (L/s)

V_s = System supply airflow rate (L/s)

V_{de} = Direct exhaust airflow rate (L/s)

5.10.7 *Calculating Return Air Temperature:*

Similar to the bank project, if a return air plenum is used, portions of the roof, wall and lighting loads may be carried away by air flowing through the plenum. To calculated the temperature of air after it has passed through the plenum, the following equation is used:

$$T_r = T_c + Q_p / (pa\ C_{pa}\ V_r\ F_u) \quad \text{.... (Eqn 1-12)}$$

Where:

T_r = Temperature of air after it has passed through the return plenum (C)

T_c = Cooling thermostat set point temperature (C).

Q_p = Total sensible (kg/m^3). Values are adjusted for elevation.

C_{pa} = Heat capacity for air (1004.832 J/(kg-K)).

V_r = Return airflow rate (L/s).

F_u = Units conversion factor.

= m^3/(1000 L)

Note that if no return plenum is used, the return air temperature is assumed to be equal to the zone air temperature.

5.10.8 *Calculating Mixed Air Temperature*

Next, the air temperature resulting from the mixture of return air and outdoor ventilation air can be computed:

$$T_{mix} = [V_v\, T_a + (V_s - V_v)\, T_r] / V_s \quad \text{..Eqn 1-13)}$$

Where:

T_{mix} = Mixed air temperature (C)

V_v = Ventilation airflow rate (L/s)

T_a = Outdoor air temperature (C)

V_s = System supply airflow rate (L/s)

T_r = Return air temperature (C)

5.10.9 *Calculating Coil Inlet Temperature.*

At this point, the coil inlet temperature can be computed. Different calculations are required depending on the supply fan configuration:

a. Draw-Thru Fans. The coil inlet temperature is equal to the mixed air temperature in this case, so no calculation is necessary. (as used in BSF)

b. Blow-Thru Fans. In this case, fan heat gain must be considered so the inlet temperature is computed as:

$$T_{ci} = t_{mix} + Q_f / (pa\ C_{pa}\ V_s\ F_u) \ldots \text{(Eqn 1-14)}$$

Where:

T_{ci} = Cooling coil inlet temperature (C)
T_{mix} = Mixed air temperature (C)
Q_f = Fan heat gain (W)
P_a = Air density (kg/m3). Values are adjusted for altitude.
C_{pa} = Heat capacity of air (1004.832 J/(kg-K)).
V_s = System supply airflow rate (L/s)
F_u = Unit conversion factor
= M3/(1000 L)

5.10.10 *Calculating Sensible Cooling Coil Load*

Finally, compute the sensible cooling coil load:

$$Q_{cs} = pa\ C_{pa}\ V_s\ F_u\ (T_{ci} - T_{co}) \ldots. \text{(Eqn 1-15)}$$

Where:

Q_{cs} = Sensible cooling coil load (W)
P_a = Air density (kg/m3). Values are adjusted for altitude.
C_{pa} = Heat capacity of air (1004.832 J/(kg-K)).
V_s = System supply airflow rate (L/s)
F_u = Unit conversation factor.
= m3/(1000 L)
T_{ci} = Coil inlet temperature (C)
T_{co} = Coil outlet temperature (C)

5.10.11 **Calculating Sensible Ventilation Load**: Based on the previous calculations, the sensible ventilation load can also be determined at this point. This calculation is performed differently for cases in which direct exhaust air is used and is not used. For the case in which no direct exhaust air is used:

$$Q_{vs} = P_a C_{pa} V_v F_u (T_a - T_r) \text{ .... (Eqn 1-16)}$$

where:

Q_{vs} = Sensible ventilation load.
P_a = Air density (kg/m3). Values are adjusted for altitude.
C_{pa} = Heat capacity of air (1004.832 J(kg-K).
V_v = Ventilation airflow rate (L/s)
F_u = Unit conversion factor.
= m3/(1000 L)
T_a = Outdoor air temperature (C)
T_r = Return air temperature (C)

For the case in which air is directly exhausted from the zone before it flows through a return air duct or plenum:

$$Q_{vs} = pa\, C_{pa} V_{de} F_u (T_a - T_c)$$
$$P_a C_{pa} (V_v - V_{de}) F_u (T_a - T_r) \text{ .... (Eqn 1-17)}$$

Where:

Q_{vs} = Sensible ventilation load (W)
V_{de} = Direct exhaust airflow rate (L/s)
T_c = Cooling thermostat setpoint (C)

5.11 Latent Heat Analysis

The latent system analysis is somewhat more complicated than the sensible analysis. This is due to the inter-dependence of humidities throughout the system.

5.11.1 Background Information

Humidity calculations are frequently a source of confusion among designers. This confusion seems to result from differences between the way humidity was analysed in older hand-calculation approaches and the way it is analysed in computer programs. Typically in the older hand-methods a room relative humidity was assumed (usually around 50%) and was then used as the starting point for calculations.

It is important to recognize, however, that this assumed room RH is an approximation. For many applications the true room RH is within 5 to 10 percentage points of 50%, so only small to moderate errors are introduced by the RH assumption. For other applications, the true room RH deviates father from 50% and therefore, the RH assumption introduces larger errors.

5.11.2 General Approach

Before humidity calculations begin, a sensible load analysis for the air conditioning system must be completed. This analysis defines the airflow rates and dry-bulb temperatures at all points in the system.

Next, the basic humidity equations for the system are formulated. There are five separate equations as shown below. These define the latent ventilation load (Eqn 1-18), the latent coil load (Eqn 1-19), the zone latent load (Eqns 1-20 and 1-21), and the relationship between the bypass factor and coil humidities (Eqn 1-22).

$$Q_{vl} = pa \quad V_{vhfg}F_u(wa\text{-}wz) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

(Eqn 1-18)

$$Q_{cl} = pa \quad V_{shfg}Fu(wci\text{-}wco) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

(Eqn 1-19)

$$Q_{zl} = [Q_l+p_a \quad V_{ihfg}F_u(wa\text{-}wz)](M_{sl}) \quad \ldots\ldots\ldots\ldots\ldots$$

(Eqn 1-20)

$$Q_{zl} = pa \quad V_{shfg}F_u(wz\text{-}wco) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

(Eqn 1-21)

$$BF = (wco - wadp)/(wci\text{-}wadp) \quad \ldots\ldots\ldots\ldots\ldots\ldots$$

(Eqn 1-22)

$$BF = (T_{co}\text{-}T_{adp})/(T_{ci}\text{-}T_{adp}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

(Eqn1-22a)

Where:

BF	=	Cooling coil bypass factor (dimensionless).
F_{sl}	=	Latent cooling safety factor (%).
F_u	=	Units correction factor.
	=	m3/(1000 L)
hfg	=	Heat of vaporization for water (2.4535 x 106 J/kg).
M_{sl}	=	Latent cooling safety multiplier.
	=	(1 + Fsl/100)
Q_{cl}	=	Latent cooling coil load (W)
Q_l	=	Internal heat gains due to people and miscellaneous internal sources (W).
Q_{vl}	=	Latent ventilation load (W)
Q_{zl}	=	Latent zone load (W)
T_{ci}	=	Coil inlet temperature (C)
T_{co}	=	Coil outlet temperature (C)
V_i	=	Zone infiltration airflow rate (L/s)
V_s	=	Supply airflow rate (L/s)
V_v	=	Ventilation airflow rate (L/s)
W_a	=	Outdoor air specific humidity (kg/kg).

W_{adp}	=	Specific humidity corresponding to apparatus dew-point for coil (kg/kg)
W_{ci}	=	Specific humidity at coil inlet (kg/kg).
W_{co}	=	Specific humidity at coil outlet (kg/kg).
W_z	=	Zone specific humidity (kg/kg).
P_a	=	Density of air (kg/m3). Values adjusted for site elevation.

To solve for the system humidifies, a moisture balance for the system must be formulated. Because the total moisture added to and removed from the system must be equal for steady-state conditions, we know that:

$$Q_{vl} - Q_{cl} + Q_{zl} = 0 \quad \text{(Eqn 1-23)}$$

Using equations 1–18, 1-19 and 1-20 from above we can rewrite equation 1-23 as:

$$V_{swci} - V_{swco} + (V_i M_{sl} + V_v) w_z = Q_l M_{sl} / pa_{hfg} F_u + (V_v + ViM_{sl}) w_a \quad \text{(Eqn 1-24)}$$

This single equation contains three unknowns (w_{co}, w_{ci}, w_z) which reveals an important fact – that all system humidities are interdependent. The supply air provided with humidity w_{co} influences the room humidity; the coil outlet humidity is influenced by the coil inlet humidity; the coil inlet humidity is influenced by return air with humidity w_z. Thus, no single humidity can be calculated directly. Because of this, the only way to solve the problem is to develop a set of three equations containing the three unknowns and then solve the equations simultaneously.

Equation 1-24 serves as the first of the three equations required. To obtain a second equation, we can set Equations 1-20 and 1-21 equal to one another since both are expressions of the zone latent load, Q_{zl}:

$$[Q_l + p_a V_{ihfg} F_u (wa-wz)(M_{sl}) = p_a V_{shfg} F_u (wz-wco)$$

This equation can be rewritten as:

$$-V_{swco} + (V_s + V_iM_{sl})\,w_z = Q_lM_{sl}/(p_{ahfg}F_u) + w_aV_iM_{sl} \quad \text{(Eqn 1-25)}$$

To obtain the third equation, we can use Equation 1-22. Because the coil inlet and outlet dry-bulb temperatures are already known from a sensible analysis of the system, the apparatus dewpoint temperature, T_{adp}, can be computed. Further, because the apparatus dew-point is a saturated condition, the corresponding specific humidity, w_{adp}, can be easily determined. Then Equation 1-22 can be rewritten as:

$$(BF)w_{ci} - w_{co} = w_{adp}(BF - 1) \quad \text{(Eqn 1-26)}$$

Finally, with the set of equations 1-24, 1-25 and 1-26, we have a system of three equations and three unknowns and the equations can be solved simultaneously to determine w_{ci}, w_{co} and w_z.

First, simultaneous solution of the three equations yields the following expression for wz:

$$W_z = [wavvBF+Q_lM_{sl}/(p_{ahfg}Fu)+waV_iM_{sl}+wadpV_s(1\text{-}BF)] / [(V_iM_{sl}+V_s - (V_s\text{-}V_v)BF] \quad \text{(Eqn 1-27)}$$

Once w_z is known, it can be used in Equation 1-25 to solve for w_{co}:

$$W_{co} = [-Q_lM_{sl}/(p_{ahfg}F_u)-waV_iM_{sl}+(V_s+V_iM_{sl})wz]/V_s \quad \text{(Eqn 1-28)}$$

Finally, w_z and w_{co} can be used in Equation 1-24 to solve for wci:

$$W_{ci} = [Q_lM_{sl}/(p_{ahfg}F_u) + (V_v+V_iM_{sl})wa+V_{swco} - (V_iM_{sl}+V_v)wz] / V_s \quad \text{(Eqn 1-29)}$$

From this discussion it is evident that calculation of true system humidities involves a fair amount of mathematics and several complicated equations. This is probably whey the procedure was never used in older hand-calculation methods. Using a computer, however, it is no trouble to perform the rigorous analysis in order to determine the correct humidity values.

The procedures for using these equations to complete the latent system analysis is outlined below:

5.11.3 STEP-BY-STEP LATENT SYSTEM ANALYSIS PROCEDURE

- Compute total Internal Latent Heat Gain. The first step is to add up all the internal latent heat gains from people and miscellaneous sources for the zone served by the system. This quantity is referred to as Q_l in the humidity equations.

- Compute Room Humidity. Next, Equation 1-27 is solved to determine the room specific humidity, w_z.

- Compute Coil Outlet Humidity. Then solve Equation 1-28 for the coil outlet specific humidity, w_{co}.

- Compute Coil Inlet Humidity. Solve Equation 1-29 for the coil inlet specific humidity, w_{ci}.

- Calculate Latent Cooling Coil Load. Once the coil inlet and outlet humidities are known, the latent cooling coil load can be determined using Equation 1-19.

- Calculate Latent Ventilation Load. At this point, the latent component of the ventilation load can also be computed using Equation 1-18.

- Calculate Resulting Room Relative Humidity. Finally, the resulting relative humidity can also be determined. The relative humidity is determined using standard psychometric equations with the know values of the zone cooling set point temperature, T_c, and the zone specific humidity, w_z.

Cooling analyses for multiple-zone central air handler systems are slightly more complicated than single-zone analyses. The multiple-zone calculation procedure includes three stages:

- Zone Load Calculations. Zone sensible loads are computed for all zones in the system for all months and hours specified by the user.

- Zone Airflow Calculations. Next, required airflow quantities for the zones served by the HVAC system are determined.

- Cooling Coil Load Calculations. Finally, system operation is analyzed to determine the system airflow rate and the sensible and latent cooling coil loads for all months and hours specified by the user.

Sensible and latent coil loads are then combined to obtain the total coil load. The maximum coil load is identified from among all the months and hours considered.

A. Zone Load Calculations

Zone sensible cooling loads are computed using the same procedure described for single-zone systems . The only difference is that loads are computed separately for each zone served by the system. The total of sensible loads for all zones served by the system is also calculated for each hour.

B. Zone Airflow Calculations

Next, the required zone airflow rates are calculated. These computations, take one of three forms depending upon the supply air characteristics.

1. Given a specified supply airflow per unit floor area (L/s/sqm), zone airflow quantities are computed as shown in Equation 1-30. Note that these airflows are not related to a specific system supply temperature, or even to the zone loads. The airflow rates are therefore of questionable use for multiple-zone HVAC systems.

$$V_{sz} = V_{sf} A_z \text{ .... (Eqn 1-30)}$$

Where:

V_{sz}	=	Zone supply airflow rate (L/s)
V_{sf}	=	Supply flow rate per unit floor area (L/s-sqm)
A_z	=	Zone floor area (sqm)

2. If a total supply airflow rate is specified for the system, the zone supply air flow rates are determined using zone sensible load ratios:

$$V_{sz} = V_s (Q_{sz} / Q_s) \text{ .... (Eqn 1-31)}$$

Where:

V_{sz}	=	Zone supply airflow rate (L/s)
V_s	=	System supply airflow rate defined by user (L/s).
Q_{sz}	=	Zone sensible load, calculated for hour when maximum cooling coil load occurs (W).

Q_s = Sum of zone sensible loads for all zones, calculated for hour when maximum cooling coil load occurs (W).

Note that this calculation relies partly on data for the month and hour during which the maximum cooling coil load occurs. Therefore, the zone supply airflow rates cannot be determined in this case until after the cooling coil load analysis has been completed.

3. Finally, given the supply temperature, the zone supply flow rate is computed by solving the following equation for Vsz:

$$Q_{sz} = \text{pa}\ V_{sz}\ C_{pa}\ F_u\ (T_c - T_s) \text{ .... (Eqn 1-32)}$$

Where:

Q_{sz}	=	Maximum zone sensible cooling load (W)
P_a	=	Density of air (kg/m3). Value is adjusted for site elevation.
V_{sz}	=	Zone supply airflow rate (L/s).
C_{pa}	=	Specific heat of air (1004.832 J/(kg-K)).
F_u	=	Units conversion factor:
	=	m3/(1000 L)
T_c	=	Cooling thermostat setpoint temperature (C)
T_s	=	Supply air temperature (C)

C. Cooling Coil Load Calculations – Sensible Analysis

1. Calculate Supply Air Characteristics. First, the total zone sensible load is used to determine either the supply airflow rate or the supply temperature. Different calculations are required depending on which supply characteristics are defined.

a. Supply Airflow Rate Defined as (L/s) or (L/s/sqm). In this case, the same specified airflow rate will be used for all hourly cooling load calculations. For each hour, a required supply temperature will be computed by solving Equation 1-5. This approach is appropriate for the analysis of constant volume HVAC systems as used in Saudi French Bank.

b. Supply Temperature Defined. In this case the specified temperature will be used for all hourly calculations. For each hour, the required supply airflow rate will be computed using Equation 1-5.

For the remaining steps in the sensible analysis procedure:

2. Calculate Supply Fan Heat Gain.
3. Calculate Coil Outlet Temperature.
4. Calculate Ventilation Airflow Rate.
5. Calculate Direct Exhaust Airflow Rate.
6. Calculate Return Airflow Rate.
7. Calculate Return Air Temperature.
8. Calculate Mixed Air Temperature.
9. Calculate Coil Inlet Temperature.
10. Calculate Sensible Cooling Coil Load.
11. Calculate Sensible Ventilation Load.

All calculations are the same as for single-zone systems. In these calculations zone sensible and plenum loads represent the total for all zones served by the system.

D. Cooling Coil Load Calculations – Latent Analysis

The same latent cooling load analysis described for single-zone systems is used for multiple-zone systems. The only difference is that the internal latent heat gain. Q_l, and the infiltration airflow rate. V_i, represent totals for all zones served by the HVAC system rather than the total for a single zone.

5.11.4 IMPLEMENTATION OF TFM FOR BSF:

The TFM method was used to analyze the block load of the building. The Carrier E20II was a preferred tool for analyzing the load as it employs the TFM for load analysis and the time of analysis the building owner wanted a design software that has been tried and used is the Kingdom.

Application of TFM to BSF was based on range of fundamental building data. The block load analysis is implemented on floor-by-floor basis

The above TFM analysis for BSF Building was used to determine the block-cooling load for the building to develop the DSM and Load Shifting programme. The load analysis provided the following fundamental data for the needed retrofit system design based on cool storage to develop a DSM and load shifting circuits to full fill SCECO and Building owners requirements:

Peak Cooling Load for DSM

Peak Cooling load for FCU circuit:

From the above, the Chiller Capacity for ice making during night

Ice Storage Capacity

Discharge Profiles

The outcome of the analysis showed that a daily total requirement of approximately 4000 $kWh_{cooling}$ was needed to handle the additional load and the SCECO's DSM requirement during peak summer period.

The total nominal storage capacity has to be 5250 $kWh_{cooling}$ sensible cooling stored in adequate numbers of ice banks to discharge without any air agitation system. The system discharge also needed to meet the building (fan coil unit circuit) constant cooling load of 175$kW_{cooling}$ and a DSM discharge of approximately 350$kW_{cooling}$ for a period of 4 hours, with one of the existing large air-cooled chillers turned-off. The system also should be able to deliver an average continuous emergency load of 525 $kW_{cooling}$.

The above provided the base data for the design of cool storage for the model project.

CHAPTER-6

DESIGN APPROACH, SYSTEM ANALYSIS AND COMPONENT SELECTION FOR THE MODEL PROJECT

6.1 Design and System Analysis for the Model Project

6.1.1 Introduction to Design Basis

TFM methodology described in Chapter-5 was utilized to develop the block-cooling load for Saudi French Bank project. Complete data are presented in Appendix

A diversity factor of 10% is utilized for future expansion and further 10% was utilized in equipment selection to compensate for losses and performance reduction due to wear and tear of major equipment.

6.1.2 Cooling load requirement:

Load Analysis indicated that a total daily requirement of approximately 4000 $kWh_{cooling}$ was needed to handle the additional load and the SCECO's DSM requirement during peak summer period.

The total nominal storage capacity has to be 5600 $kWh_{cooling}$ sensible cooling stored in adequate numbers of ice banks to discharge without any air agitation system. The system discharge also needed to meet the building (fan coil unit circuit) constant cooling load of $175kW_{cooling}$ and a DSM discharge of approximately $350kW_{cooling}$ for a period of 4 hours, with one of the existing air-cooled chillers turned-off. The system also should be able to deliver an average continuous emergency load of 500 $kW_{cooling}$.

6.1.3 Design Approach:

Using the block load cooling analysis outlined in Chapter-5, a system was designed to provide the above additional cooling to the existing system.

The design approach took into account of the limitations imposed by the project specifications and the power company requirement for DSM.

Air conditioning system is the dominant power user thus any DSM programme has to be based on the Air-conditioning system.
The first step to the design approach is to establish the required chiller capacity in relation to the building load analysis. This enable to estimate the required additional power needed to operate these chillers.
The second step is to develop or upgrade the hydronic network which include the pumps, controls, valves and the pipe network. This would lead the design to couple the production plant and the hydronic network with the air distribution system.

However this approach cannot be implemented as stated since this model project has several restrictions. These restrictions are:

1. Retrofit
2. Availability of grid power; sub-station capacity could not be increased.
3. Limited space
4. Type of chillers – must be air-cooled (Saudi Requirement)
5. Access to existing air-conditioning units
6. DSM requirement

The conventional design approach, which involves in simple system up grading with new chillers, is not possible in this project due to the above restrictions specifically item-2 and 6.

Hence a new approach for retrofit has to be introduced that satisfies the key limits listed above, mainly availability of grid power; sub-station capacity could not be increased. DSM requirement.
One possible option would be of the use of localized power generation based on diesel generators. The second option would be using selective shut down of air-conditioning system in non-essential areas.
The other option would a load-shifting programme based on cool storage based on ice storage system.

The chiller capacity has to be increased by $700kW_{cooling}$. The hydronic net work has to be re-sized to accommodate the additional chilled water flow of 28 L/s. This would involved in either replacing the existing pumps and hydronic components such as valves etc., or adding an additional pump in series to provide the necessary increase in flow rate and system dynamic pressure.

The airside equipment also have to be modified. This would involve in changing the pulleys of the fans of AHUs , thus increasing the airflow in the distribution ductwork with the needed volume flow rate and system pressure.

Regarding the DSM requirement there were two options available for the bank. Selective shut down of air-conditioning system in certain non-priority areas until the required reduction in peak load is achieved, or providing a localized power generating unit with automatic transfer switch to transfer the excess load from the grid to the localized power source during the DSM period.

However, the building (BSF) has several limitations and restrictions to apply the above retrofit conventional system modification approach. SCECO wasn't prepared to provide additional power to the building nor the existing sub-station would have taken any increase in building load as it was running at 95% of it's capacity. All internal services were based on high quality interior design work, thus modifying the air distribution network would involve in dismantling architectural ceiling features. This would not only disrupt the smooth operation of the bank but also would involve in expensive interior refurbishment work.

The bank wasn't prepared to install a localized power generator as this would involved in extensive restructuring of the electrical circuits, and also would cause noise and other environmental problems to the bank and the adjacent buildings.

A new system based on the following was introduced to eliminate the above problems:

1. Charging chillers, only to operate in the nights after the office hours, while the main chillers are turned off
2. A new fan coil circuit to provide spot cooling for the areas with cooling problems

3. An injection circuit to provide DSM
4. All the above, work from a unique Cool storage system

The cool storage (TES) was unique to the Gulf and the system introduced at the BSF was the first TES system installed in a commercial building in the region to provide combined benefits of load shifting, load leveling and DSM.

A separate and stand-alone production plant was installed at the roof level comprising two low temperature glycol chiller, pumps for the glycol loop, motorized valves for the injection circuits and two heat exchangers to provide dedicated cooling to the fancily and DSM circuits. Several novel features such as injection loops, freeze protection, PLC based control system etc. were introduced in the system design. The system design was unique as it employed a novel injection circuit philosophy to reduce the number of pumps used in various circuits, which were coupled together to the mail glycol loop.

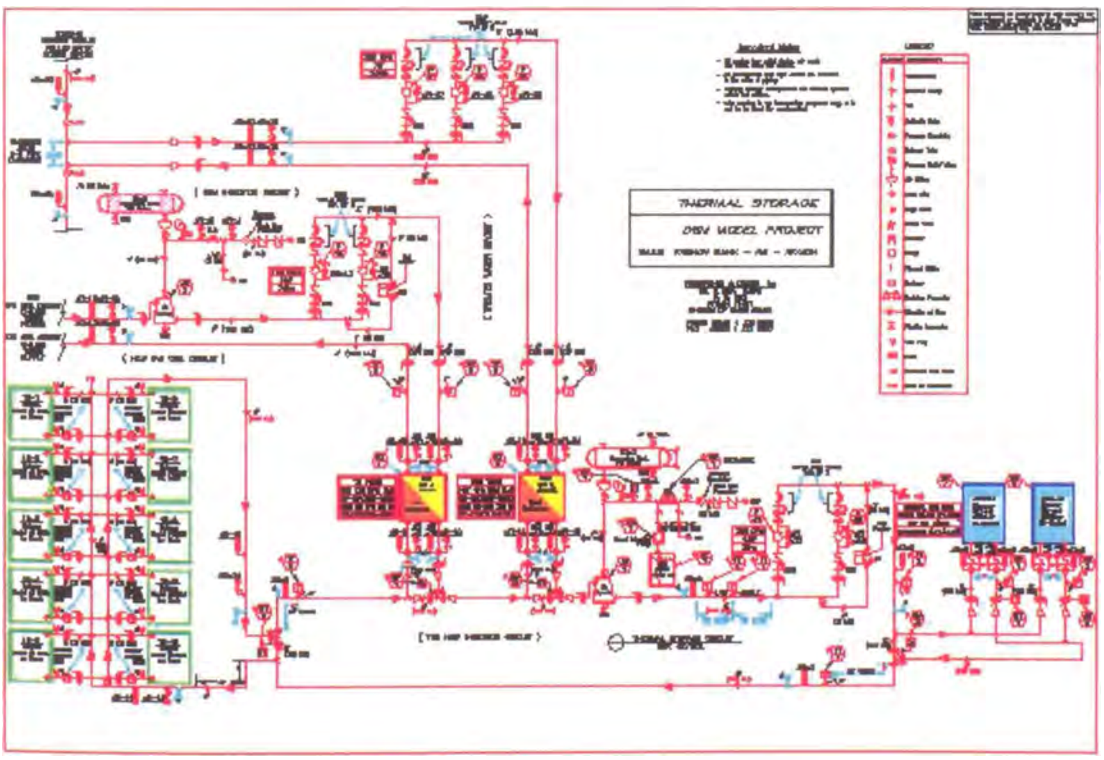

Fig.6.1 System Schematics(Refer to Appendix for a full drg.)

6.1.4 System Components:

To meet the cooling load requirement derived from the above analysis, two chillers with a total installed chiller capacity 700 $kW_{cooling}$ (2x 350 $kW_{cooling}$ nominal) were selected based on screw type glycol chillers. Screw type chillers were selected, as they operate with minimum penalty on power consumption compared with reciprocating chillers on load shedding or partial load operating conditions [n]. Centrifugal chillers were not considered as low capacity air cooled chillers are not available in the market for commercial cooling applications. The above capacity was rated according to ARI standard at 30°C condenser air entering temperature at sea level. The expected ice making capacity of these chillers was 210$kW_{cooling}$ each based on 25% ethylene glycol 75% water (by weight mixture) coolant rated at 30°C condenser air entering temperature and glycol mix. leaving temperature of -4.0°C. The chiller capacity during charging is not include the building nighttime base load and this load was handled by either one of the building chiller or the heat pump based on the load requirement.

The two new glycol chillers are electrically coupled with the two existing water 700$kW_{cooling}$ chillers allowing either the glycol chillers or one of the water chiller to operate at any time. The sequencing of these chillers by the Programmable Logic Controllers (PLC) is shown in table-6.1.

CHILLERS	7a.m. - 1p.m.	1 p.m. - 5 p.m.	5 p.m.- 8.00 p.m.	8.30 p.m. - 6.30 a.m.
Water Chiller-1	**ON** (cooling)	**OFF** (DSM-ice discharge)	**ON**	**Off**
Water Chiller-2	**ON** (cooling)	**ON** (cooling)	**ON**	**Stand-By**
Glycol Chiller-1	**OFF** (ice discharge*)	**OFF** (ice discharge*)	**OFF** (ice discharge*)	**ON** (ice making)
Glycol Chiller-2	**OFF** (ice discharge*)	**OFF** (ice discharge*)	**OFF** (ice discharge*)	**ON** (ice making)

*** ice discharge to provide additional cooling**

Table 6.1 Sequencing and chiller-Ice Matching

The operational sequence illustrated in Table 6.1 shows that between 8.30 p.m. – 6.30 p.m. the ice being made and stored. During the hours of 7.a.m.-1p.m. and 5p.m.-8 p.m. the system is melting ice to meet the additional cooling load as described in section "Comfort Cooling system" above. Between 1 p.m. – 5 p.m. the system is continuously melting ice and increasing the secondary chilled water flow rate to meet SCECO's DSM requirement in addition.

The above sequence of operation was established, so that the electrical load due to chillers is maintained within the maximum power limit available at the existing sub-station.

In addition these two glycol chillers provide valuable stand-by features in case of failure of one of the main water chillers during the daytime.

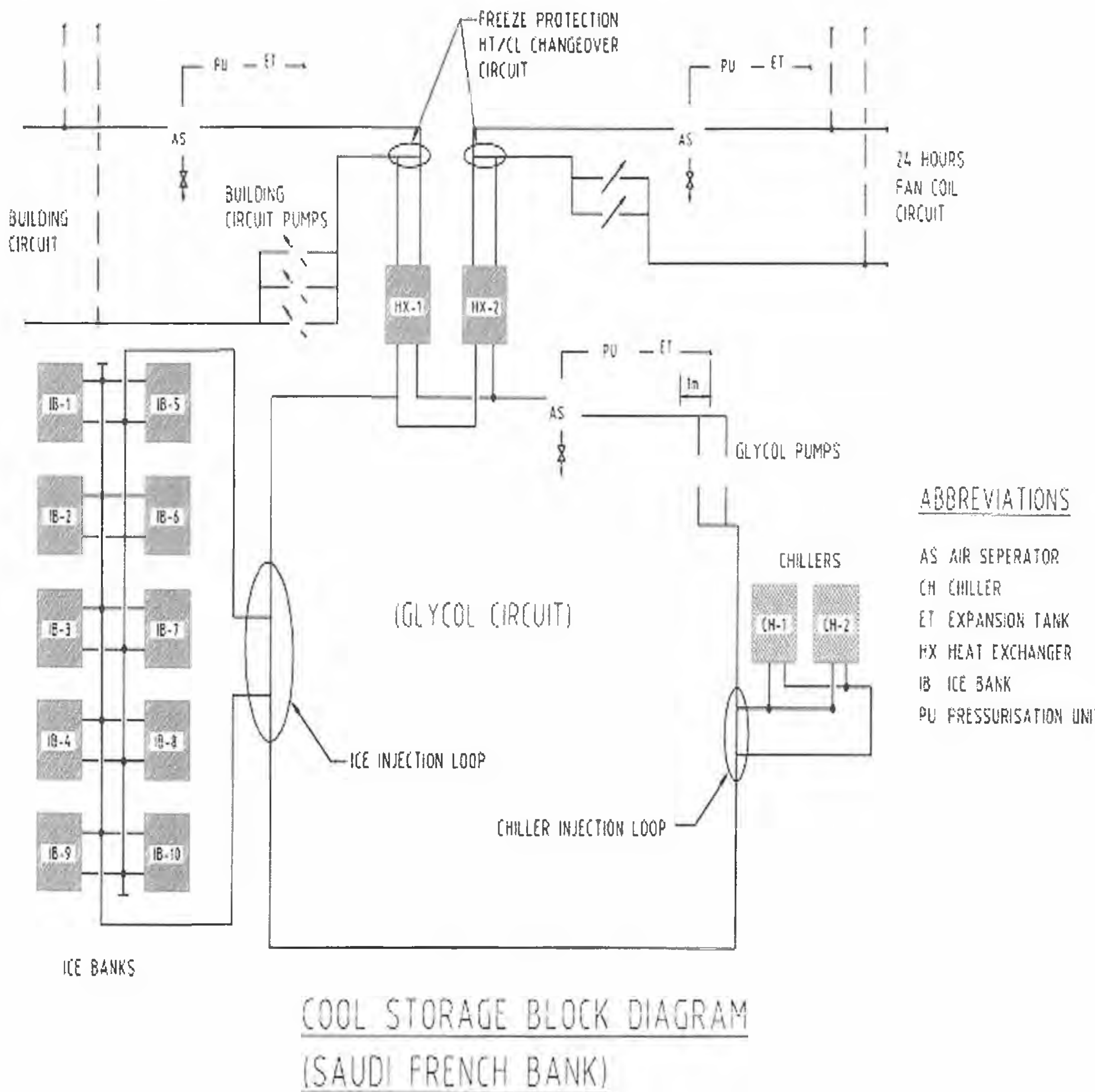

Fig.6.2 System Schematic diagram(Refer to Appendix for a full drg.)

6.1.5 Hydronic Components

A central constant volume glycol pumping and control module for the ice storage system was established. The chillers, ice banks and heat exchangers are coupled with the glycol circuit by motorized valves. (Figure-6.1)

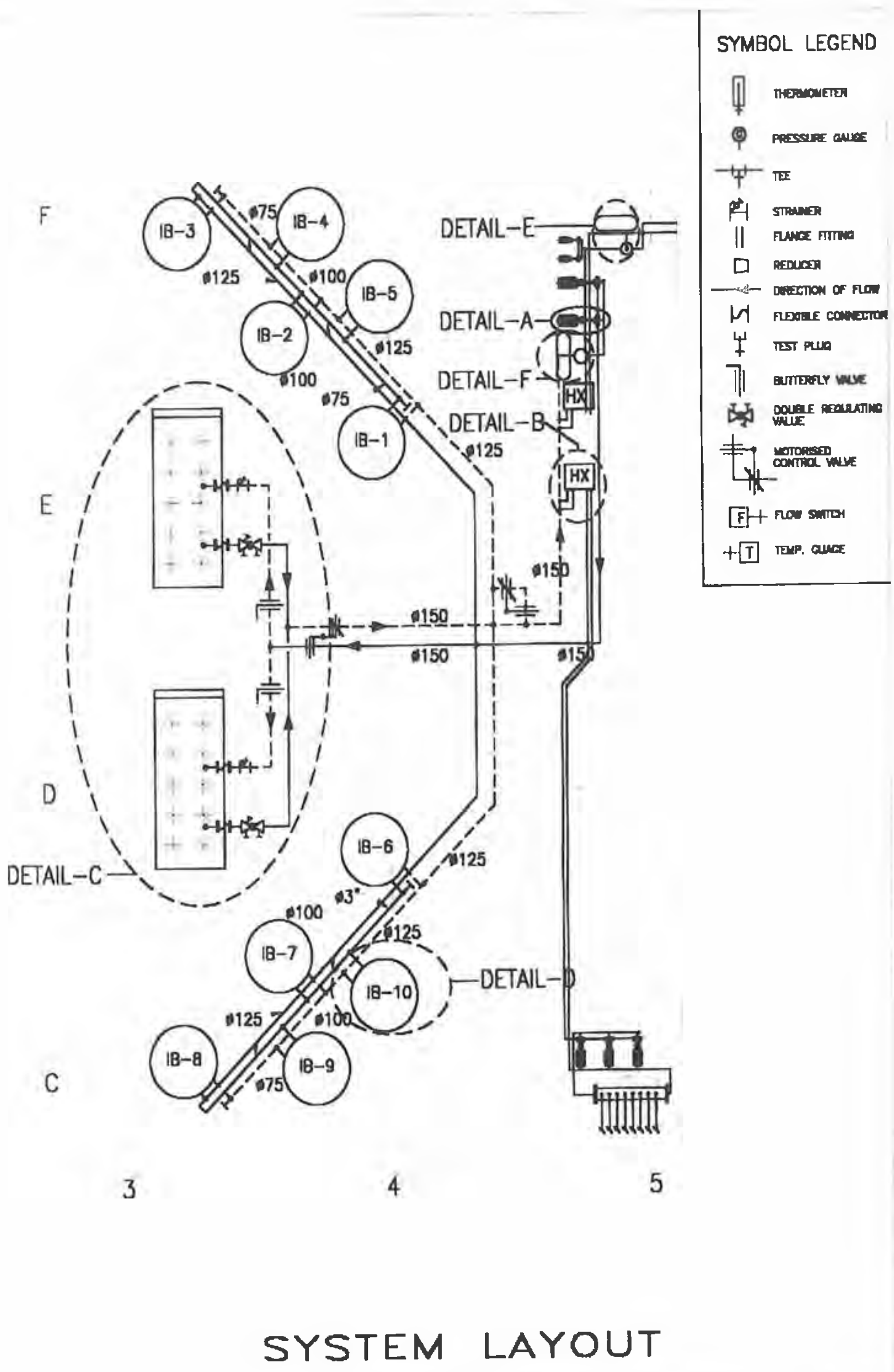

Fig.6.3 Basic System Layout(Refer to Appendix for a full drg.)

One injection circuit was established to supply chilled water for the 50 fan coil units, each of them rated at $5kW_{cooling}$, which were newly installed throughout the building where additional cooling was needed. The second injection circuit was established to provide the required cooling into the existing network during SCECO's imposed DSM period between 1 p.m. – 4 p.m.
The constant volume glycol circuit, by utilizing motorized valves, has eliminated the need for pumps on each injection circuit on glycol side, therefore having a lower demand for power to operate the circuit by 12-15%. The glycol circuit is provided with two active and one stand-by constant volume pumps electrically coupled with the glycol chillers. Coupling the injection circuits with the constant volume glycol circuit by modulating motorized valves, instead of individual pumps have resulted in lower power demand by approximately 20 kW_{power} as the motorized valves consume much less energy than pumps. Due to this design feature the operating kWh_{power} for the ice storage system is considerably reduced in comparison to a traditional coupling method utilizing pumps.
Plate and frame heat exchangers are used to separate the glycol circuit from the chilled water circuit.

To provide freeze protection during lower temperature ice making, 25% ethylene glycol was used in the primary circuit.

6.2 Selection of Ice Banks and justification for ice on coil internal melt system

The model project requires storage of approximately 3700kWh to be discharged on full and partial mode between 8 a.m. and 7 p.m.

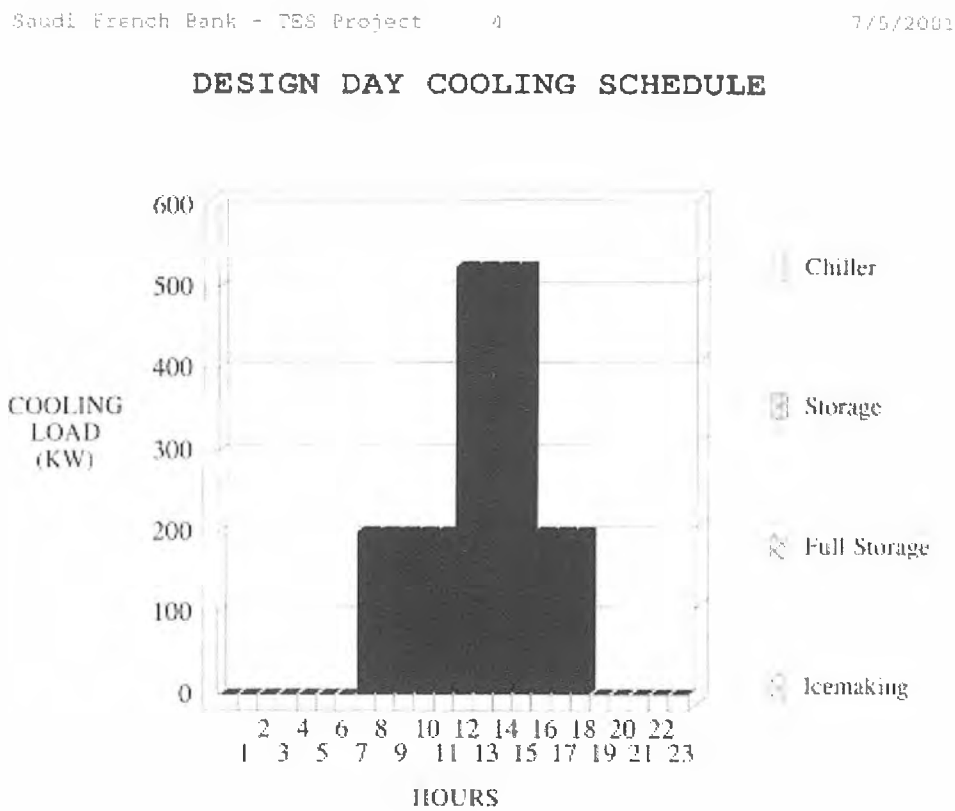

Fig. 6.4 Ice Discharge Profile for ice bank Selection

The system has to be coupled with the existing system through an injection circuit. As this project is the first of its kind in the Middle East the client preferred to use self contained factory assembled and tested modules. The design brief was such that the thermal energy storage system has to be simple in design, easy to install, operate and to maintain. Also the system should provide maximum flexibility to change operation according to operational schedules imposed by the local power generating Authorities. As the project is a retrofit, the system should be able to integrate into the existing conventional air conditioning system without major difficulties. In addition the client also requested for a cost effective TES system for this project.
Considering all the above the researcher has selected the factory build ice on coil internal melt system for this project which gives a balanced benefit of all the above requirements.
The researcher has selected the externally insulated tank to avoid heat transfer and also to eliminate condensation.
The selected storage tank is an externally insulated polyethylene tank with counter-flow heat exchangers.

Fig 6.5 Ice Banks as Installed at BSF

In the ice banks counter flow tubular heat exchangers are used to improve charging and discharging efficiencies. The charging temperature was set at –4.5°C instead of the standard –6°C used by other commercial type ice banks supplied with uni-directional flow heat exchangers. The greater heat transfer efficiency of counter flow arrangement allowed the chillers to charge the ice banks at a relatively warmer temperature and shorter period of time than an alternative system utilizing uni-directional heat exchangers.

6.2.1 Basic Controls (also refer to section 6.9.2)

A stand-alone PLC controller was installed to control the primary ice storage and the secondary cooling and DSM circuits. The PLC controller was selected in place on the usual Direct Digital Control (DDC) for system reliability and to provide fast response action such as opening and closing of control valves. The fast response feature is important for the injection circuits operating with modulating valves instead of the usual direct pumping system. The PLC controller was programmed to provide the following operational features:

1. On/Off/Changeover of chillers
2. On/Off Glycol Pumps
3. On/Off Control Valves
4. Control of glycol make-up system
5. Heat exchanger freeze protection
6. Leak detection
7. $kW_{power}/kW_{cooling}$ data
8. Charging & Discharging of ice banks during make-up 7 DSM cooling period
9. Alarms

6.3 Major Equipment and Component Data

The TES system at BSF was designed to employ the principal of ice storage based on ice build and internal melt concept storing ice in ten modular tanks and the system was integrated with the chilled water building cooling system through the Heat Exchangers and pumping modules that are part of the production plant and the TES system.

All plant and components were selected in a way that they can be installed in the allocated restricted space available in the plant room on site. The system was designed on the basis that it can operate as partial storage system from 7 a.m. till 8 p.m. according to the variable load demand of the building.

The system was provided with an ice inventory-measuring device, which will indicate the amount of ice available at any time within an accuracy of $\pm 5\%$.

To provide freeze and burst protection ethylene glycol was used in the TES system contain proper corrosion and decay inhibitors. The product used in this project was UCARTHERM. A refractometer was provided to measure the glycol concentration in the water/glycol cooling mixture by indicating the mixture's anti-freeze protection point. The set ration was 25% by weight.

The pipe work was black steel which is compatible with the TES system. All system pipe work and components were insulated with 25mm closed cell rubber insulation, vapour protected and cladded to provide thermal insulation, avoid energy loss and condensation.

The researcher imposed a special condition that all material and equipment shall be industrial grade type and compatible with glycol application.

6.3.1 Ice Banks

The ice storage tank contains a non-corrosive heat exchanger and supply and return headers. The tanks was selected in a manner that it will be capable of being individually isolated from the system so that each may be serviced without interrupting the operation of the total system.

The tank container is constructed of high-density polyethylene with an average thickness of 9mm and a minimum tear resistance of 230 kg/mm in all direction according to ASTM-D-1004. The bottom and side of the tank was insulated with a 50mm R-9 insulation and the top was insulated with a minimum of 75mm R-24 insulation. The tank was externally insulated to provide protection from extreme weather condition. The selection of insulation thickness was to match standby losses of not more than 1% of the stored capacity at 46°C environment. The sides of the tanks were covered with 0.8mm textured aluminium sheet.
The ice bank heat exchanger was selected to withstand a 6 bar maximum operating pressure. As the tanks were installed at roof level on a specially constructed elevated platform the operating weight of the tank became paramount to the researcher. The floor loading of the tank was limited to 1900 Kg /m2 when filled with water.
Connections to the tank from the main header were with 4-ply braided rubber hose having a burst pressure of 20 bar. To avoid expansion and contraction, when connected to the tanks, the connection points were installed in a way to move freely 25mm in all directions. Each and every tank was provided with circuit setters and butterfly valves for easy system balancing and isolating from the system during emergency.

Fig.6.6 Reverse Return Piping arrangement

The ice bank glycol circuit was developed on a reverse return piping arrangement.

1. The tanks were installed in away that is capable of being individually isolated from the TES system so that each may be serviced without interrupting the operation of the total system.
2. Storage of the 10 x Calmac 1190A tanks represents a cumulative storage of 6700kWh; 5300 kWh latent & 1400 kWh sensible cooling. The efficiency of discharge is depending on the inlet and outlet temperature and the amount of energy discharge per hour. Larger the delta T (temperature difference between the inlet and the out let of the ice tank) faster the depletion of energy from the tank.
3. The system discharge requirement is around 3750kWh_c
4. Full Storage of the tank takes place on the first day of commissioning or during the initial charging of the system.
5. After the initial charge, the daily charge represents the top-up energy, related to the energy depleted providing cooling on the previous day and the system losses.
6. The initial storage (data point) shown in the chart represents the carryover of stored energy from the previous day of charging, with losses accounted for freeze protection and ambient loss.
7. The charging of the system governs by the amount of depletion of energy from the previous day and the time allowed for system

charging, normally 12 hours during the night commencing from 1900 hours.

8. The discharge of the system is governs by the FCU circuit operating time of 10 hours , starting at 0800 hours and the DSM period of 4 hours commencing at 1300 hours, during such time one main chiller is turned off.
9. The peak discharge is around 525kW$_c$, related to almost 225kW$_e$ of DSM of the circuit.
10. A lapsed period of 1 hour minimum is maintained to allow changeover between charging and discharging modes.
11. This period may vary due to factors such as office opening hours, achievement of full charging of the tanks
12. The increase in demand represents the trend of the accumulation. Storage is due to progressive increase in ambient temperature during the month of August.
13. By end of August the ambient temperature beginning to fall, thus allowing the system operators to change to mode of operation from chiller priority to ice priority. Ref.....see chapter....
14. DSM only required for the peak summer period commencing from May-Sept.
15. During the winter period the TES was used during the day time as full load shifting tool to proving cooling for the internal area to offset the internal heat gain such as lighting, occupants and ancillary equpt. (computers etc.) thus receiving further discount from SECECO.
16. This allows the shut down of the bigger chillers during winter, normally operated on partial load, eliminating PSS, and provide adequate maintenance period.
17. The winter cooling load of approx. 150 TR, matches the TES system capacity.
18. The system originally designed for Chiller priority partial load operation; however, the system is now operated under the following three modes:
    Chiller Priority –Partial Load --- summer
    Ice Priority –Partial Load---- autumn & spring
    Ice Priority –Full load ---- winter

19. The system is designed for 12 hours charging and 10 hours discharging of the ice banks. The additional 2 hours allowed for changeover of the operating modes of the system (charging and discharging), and the shut down of primary chillers.

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CHILLER AND TANK SELECTION SUMMARY

LEVLOAD TANK MODEL		1190
DESIGN LOAD	(KW)	525
SYSTEM SUPPLY TEMPERATURE	(C)	3.3
SYSTEM RETURN TEMPERATURE	(C)	[illegible]
DEFAULT CHILLER COOLING CAPACITY	(% OF NOMINAL)	100
DEFAULT CHILLER ICEMAKING CAPACITY	(% OF NOMINAL)	60
NUMBER OF COOLING HOURS		[illegible]
NUMBER OF ICE-MAKING HOURS		[illegible]
TOTAL COOLING LOAD	(KWH)	[illegible]

NOM CHLR KW	COOL CAP KW	ICE CAP KW	STRG DIV	ESTMTD KWH	STRG INLET C	STRG OUT C	PEAK STRG KW	MIN # TANKS	MAX # TANKS
685.19	685.19	411.11	0.47	453.05	6.90	3.30	525.00	8.17	8.14

FLOW ANALYSIS

685 KW CHILLER
9 STORAGE TANKS MODEL 1190
25 PERCENT ETHYLENE GLYCOL

CHG dT	CHG L/S	L/S/TANK	dP(KPA)	AVG LCWT	MIN LCWT
2.0	52.3	5.8	126.8	-3.1	-4.8
2.5	41.8	4.6	89.9	-3.5	-5.1
3.0	34.9	3.9	68.9	-3.8	-5.3
3.5	29.9	3.3	55.5	-4.2	-5.6
4.0	26.1	2.9	46.3	-4.6	-5.8
4.5	23.2	2.6	39.7	-5.1	-6.1
5.0	20.9	2.3	34.6	-5.5	-6.4
5.5	19.0	2.1	30.7	-5.9	-6.6

DIS dT	DIS L/S	L/S/TANK	dP(KPA)
4.4	27.0	3.0	40.1
4.9	24.5	2.7	35.2
5.4	22.5	2.5	31.4
5.9	20.7	2.3	28.3
6.4	19.2	2.1	25.8
6.9	17.9	2.0	23.6
7.4	16.8	1.9	21.8
7.9	15.8	1.8	20.2

Table.6.2 Chiller and Tank Selection

6.3.2 Industrial grade digital controller

A stand-alone TES control system was provided to automate the thermal storage plant. Mainly, the following equipment are controlled by this stand alone controller:

1. Chillers
2. Pumps
3. Control Valves
4. TES Ice tanks and inventory system
5. Glycol make-up system

All controls and equipment provided were “industrial grade” with a mean time failure of fifteen(15) years. The PLC processor has an overall scan time of 15 milliseconds. A factory developed, tested software based on the researcher’s requirement was provided by M/s Control Technology USA. All inputs and outputs were fully isolated to 600V and Analogue-to-Digital conversion is based on 12 bit. A special cabinet type control panel was constructed (1800mm x 900mm x 400mm) epoxy painted force/filtered ventilation and lighted with a window kit to view process indicators.
All devices were wired in accordance with international standard and with cloth wire identification at both ends. Control wiring were installed in a neat manner with wire bundles, and ties.
The operator interface is a “Nema 4” panel mounted 300mm colour display with keypad. All operators, displays , set points, control modes and alarms are accessed at this location.
The following Modes of operations are provided:

1. Ice Charge
2. Chiller Building Load Only

3. Ice Discharge
4. Ice Discharge with Chiller assist
5. Ice Charge with Building load
6. System OFF

The following control features also provided:

1. Monitoring package with thermal balance cooling tons and ton/hours from each source including pump losses
2. Alarm history and status
3. Menu-driven operator displays
4. Real time clock
5. Up to 135 individual mode programming steps
6. Closed transition mode changes
7. Soft loading and unloading of chillers and equipment
8. Heat Exchanger freeze protection
9. Glycol make-up and inventory control
10. Control and reset of chillers
11. Servo-driven control valves
12. Eight (8) level of security access
13. Building Automation System interface
14. Built-in diagnostics and remote monitoring.
15. Data logger printer, stand and paper tray
16. kW/Ton monitoring

Fig.6.7 PLC based Stand alone Control System

The controls system was selected based on PLC based control system interlining cup-1 and cup-2 . A control panel with operator interface and digital display system was installed in the control room, providing total system illustration with flows and control details. The control system will be inter-linked with the building BMS for monitoring purposes.

Fast system iteration of the control system would be needed for efficient TES operation .The provide efficient scanning of the system the system was provided with a PLC processor with a overall scanning time of 15 milliseconds.

The control system will provide in excess of 1000 times faster iteration on system analysis in comparison with a digital control system during various mode of operation.

The stand-alone control system was designed to automate the total production and the TES plant.

The following equipment is controlled by the system.

- Chillers
- Pumps
- Motorized Valves

- Ice Banks
- Ice Inventory System
- Glycol Make-up system

The control system was developed to provide the following four modes of operation.

- Ice Charge
- Chiller operation for building load
- Ice discharge
- Ice charge with building load

The system is currently operated on ice charge and ice discharge modes only, and the other two modes are allowed for emergency operation only.
It is anticipated as and when BSF extend CUP-2 with additional chillers these two modes will become active and will provide useful operational modes.

The control system also selected to provide the following selective control features:

- **Thermal balance control**; providing information on cooling kW and kW/hr from each source including pump losses: This is an inventory system specially developed for the research purposes. The feature allows the researcher to monitor the total energy balance of the system between, storage, usage and losses.

- **Soft loading and unloading of chillers**: Chiller soft loading is a critical issue for low temperate application. As the fluid will be cooled through a larger temperature gradient, it is necessary that the chillers to be loaded in a step-by step manner providing adequate time for temperature pickup and valves closing and opening.
- **Heat Exchanger Freeze protection:** This is a major issue in TES system during ice making. The glycol side of the Heat exchanger will be at below

freezing temperature; this may induce freezing on the water side of the heat exchanger unless otherwise adequate thermal transfer is in process and continuos fluid circulation is available on both sides of the heat exchanger. A trace heating system is provided with adequate low temperature cut-off and alarm facility to over come this major problem associated with low temperature application.

- **Glycol Make-up and inventory Control**: As above this is another major issue faced by TES system operators. The system is provided with 25% ethylene glycol (by weight), for antifreeze during low temperature charging period. Ethylene glycol is normally depleted from the system due to unforeseen leaks and bacteria attack. If the glycol concentration falls below the set point this would result in system freezing during charging period.

- **Closed Transition Mode Changes**: It is necessary the system has to "roll-over" into several operating modes on a 24 hours cycle. This would require several components to operate under different duties. Certain valves have to be closed, certain valves have to be open, pumps to be shut off, pumps to be turned on, chillers to be shut off, chillers to be turned on etc. The closed transmission mode will allow these operations to take place in an orderly and sequential way that the system will be protected from surges and back-flows.

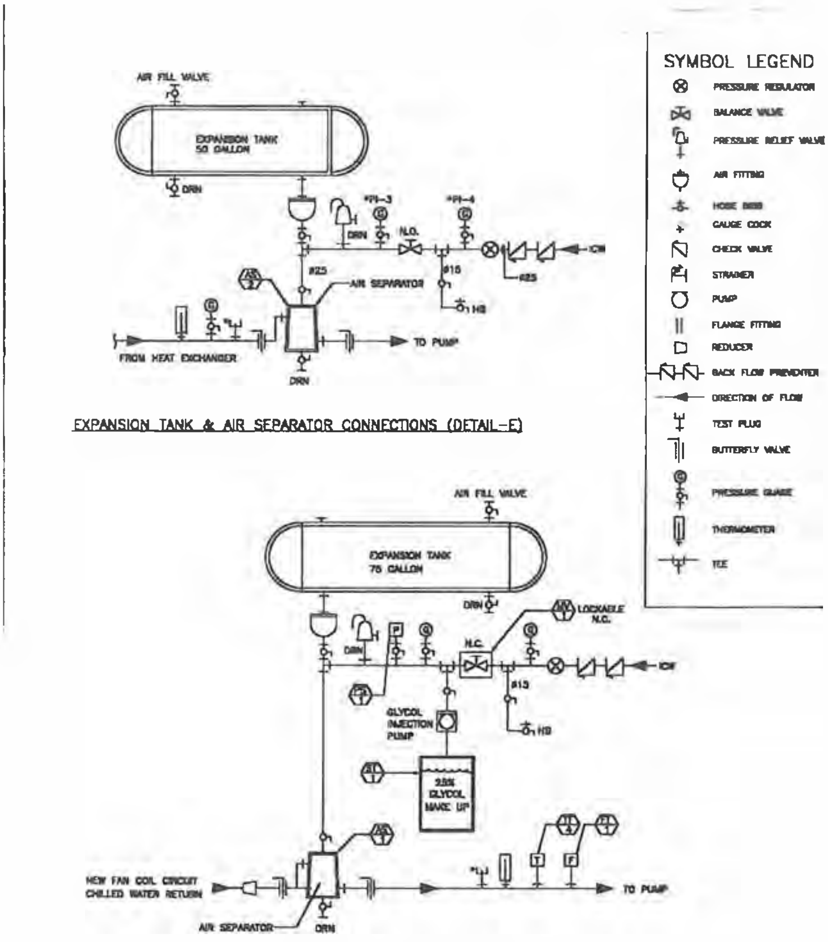
- **Resets**: It is necessary to have a re-set feature in the control system to allow the researcher to reset all the operating parameters during the first year of monitoring period to reach optimum operating conditions for all equipment to achieve the best overall performance of the system.

- **History and status:** This special feature was developed mainly to record system performance and data logging on a 30 minutes interval.

6.3.3 Control valves, sensors and flow meters

All control valves selected are industrial grade butterfly control valves and compatible for low temperature glycol operation. Control valve bodies are rated at 1000 kPa design, with a rangability of 100 to 1. All two-position valves are selected with electrical actuators. However, modulating valves are provided with servo-driven motors and position feedback. Due to the low temperature application and high ambient conditions all valves are provided with thermostatically controlled heaters with "Nema 4" enclosure. Control sensors are provided with with 4-20mA transmitters. The researcher requested for an overall accuracy of $\pm$ 0.05°C. The thermal wells are provided with O- rings for condensate seal between sensor and thermal well to prevent freeze damage. All glycol and chilled water flow meters are industrial grade type, in-line turbine flow meters. These meters will measure true mass flow with an overall accuracy of $\pm$2%.

Fig. 6.8 Expansion Tanks and air separation System



EXPANSION TANK & AIR SEPARATOR CONNECTIONS (DETAIL-F)

EXPANSION TANKS CONNECTION DETAILS

6.3.4 Glycol make-up and control

A special control system is provided to automate the glycol make-up. A factory assembled glycol make-up tank was provided with high-pressure injection pump, filters, check valves and pressure controller. Lockable water make-up valve and warning sign are provided to prevent make-up water contamination. One set of portable glycol test meter and sample kit was also provided. The purpose of this system is in two folds; to maintain the required 25% glycol concentration and also to provide make-up water as part of the glycol injection circuit.

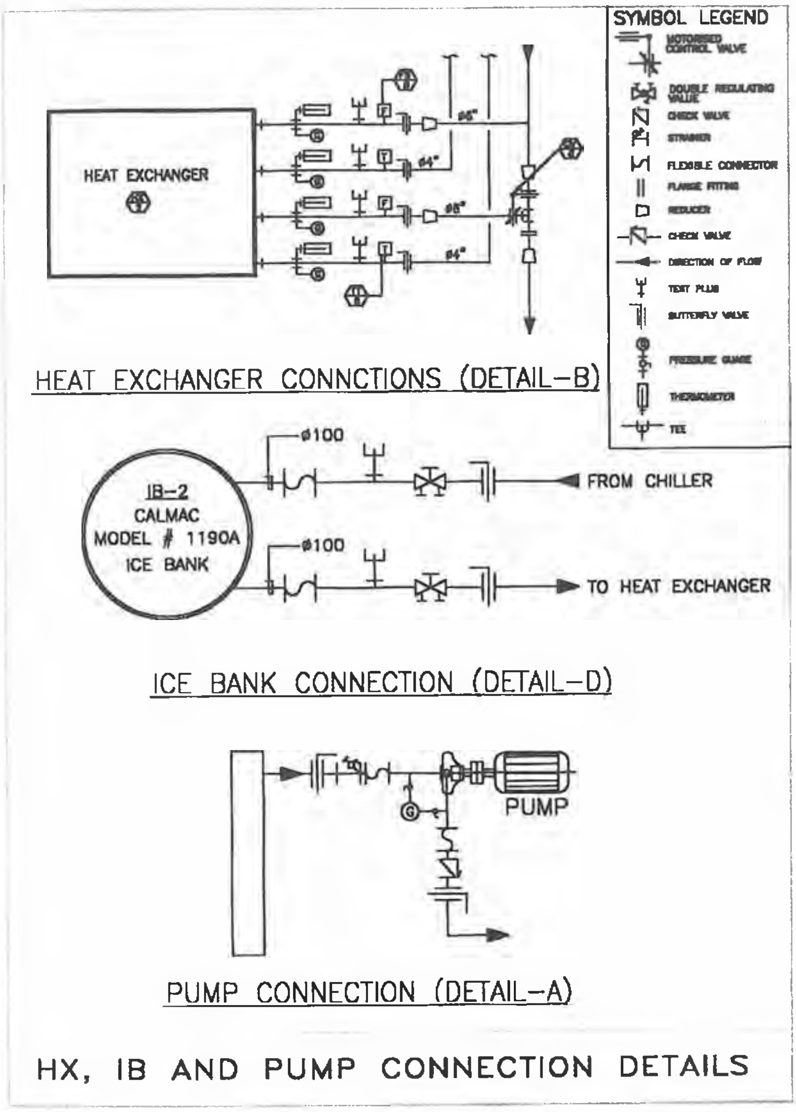


Fig. 6.9 Details of Components Connection

6.3.5 Heat Exchangers

The project was provided with two heat exchangers of the stated net capacities as shown on the drawings (see addendum) . The heat exchangers are of counter-flow type design to accommodate thermal duties with an acceptable temperature crossover and approach selected by the researcher, but not less than 1.2°C.
The heat exchanger was rated with the appropriate fouling allowance required for the specified duty.

Design Parameters:

A. Nozzle velocity is limited to 6m/s.

B. The manufacturer was requested to optimise heat transfer using the capacity specified by the researcher using the given flow rates and pressure drop conditions to heat exchanger performance requirement without reversing the flow.

C. The heat exchanger was provided with one-piece inter-plate gaskets made of material suitable for the given fluids (water and glycol mixture) and the given process conditions.

D. Due to the space restriction at the project site, the heat exchanger was selected of compact design, not requiring more than 50% installed space for servicing and maintenance.

E. The plates are selected to withstand full maximum differential pressure without any pressure on the adjacent plates.

F. All inlet and outlet connections were provided with flanged connection for easy removal for maintenance.

G. The heat exchanger frame capacity was sized to accommodate at least 20% additional plates for future purposes.

H. The heat exchanger was constructed with heat transfer plates fabricated from type 304 stainless steel with the necessary thickness of plates was sized to accommodate a 1700 kPa pressure.

I. The heat exchanger was protected by an automatic freeze protection system controlled by the industrial grade stand alone digital control system.

Fig. 6.10 Heat Exchangers at BSF (circuit 1 & 2)

6.3.6 Pumps

The pumps were selected to provide the required flow rates and working pressure as designed and calculated by the researcher. Each circuit was provided with one stand by pump of the stated capacity. The building side pumps are provided with variable speed drive. The speed of the pumps are limited to1750 rpm. The selected split case centrifugal pumps were secured to a reinforced concrete inertia pad isolated by open spring vibration isolators, to provide noise and vibration control.

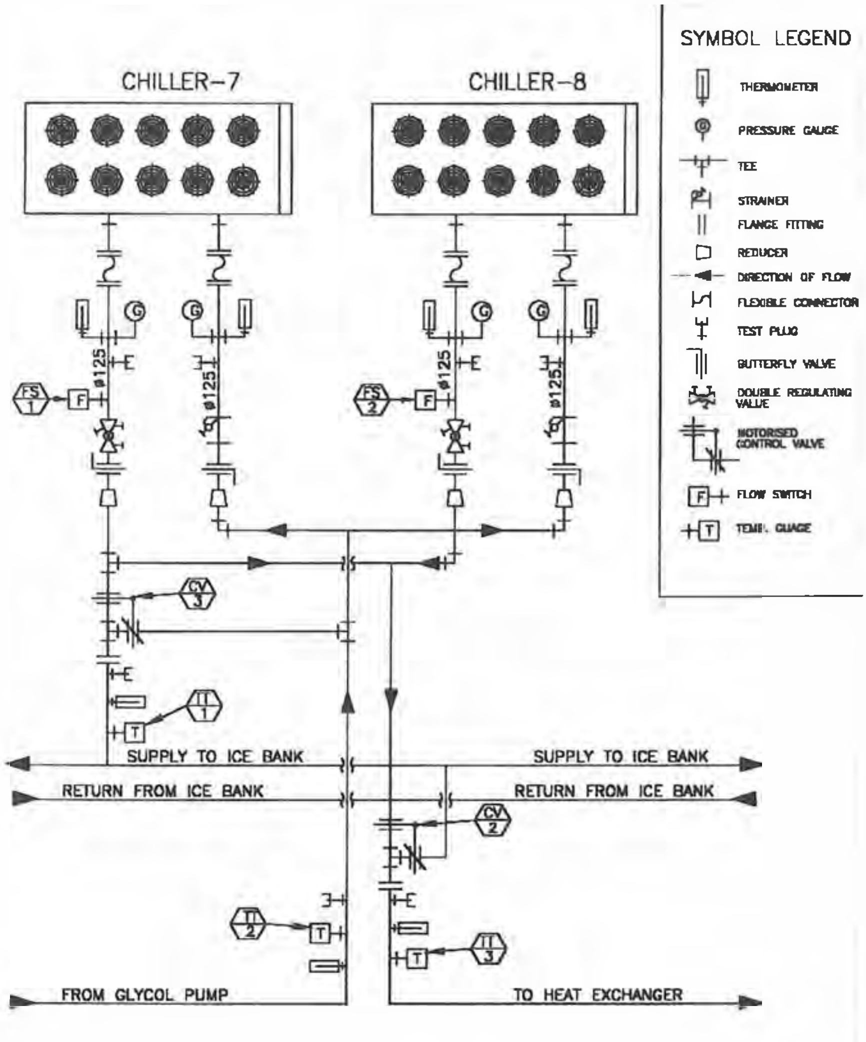
The basic construction of these centrifugal pumps was, cast iron casing, stainless steel impeller, and stainless steel shaft with bronze sleeves. All pumps are provided with mechanical shaft seals and other components specifically selected for glycol application and their treatment and pressures. The pump impeller are of the enclosed type, hydraulically and dynamically balanced and keyed to the shaft and secured by a suitable locking device.

The motor were rated according to NEMA specifications and be up to the standards required for industrial use. They are furnished with re-greasable ball bearings, adequate for the maximum load for which the motor is designed. The motors are TEFC and to be high efficiency.

Fig. 6.11 Pumps at BSF Project

6.3.7 Ice making chillers

The ice making chillers were selected to be screw (or sometimes referred as rotary type) with R-22 refrigerant, capable of producing a wider temperature lifts between a lower charging temperature of -4.$^{\circ}$C and the normal water return temperature of 12°C. The high return glycol temperature will be used only on emergency conditions with a 6°C-temperature difference. At normal ice making conditions, the return glycol temperature will be around 0°C. The chiller components shall be compatible to operate with 25 % ethylene glycol 75% water mixture by weight. The glycol chillers were selected to operate under the stand-alone automatic controller on normal mode of supply temperature of 6°C during day time operation (emergency mode only) and - 4.0°C supply temperature during ice bank charging mode. The glycol chillers were provided with factory mounted low temperature kit and automatic temperature re-set facility for the above two modes of operation.



CHILLER CONNECTION (DETAIL-C)

CHILLER CONNECTION DETAILS

Fig. 6.12 Details of Chiller Connection

The glycol chillers have two separate but equal capacity electrical connection points so that it shall have minimum of two separate circuits with the total electrical and mechanical isolating features. This feature was selected to provide a 50% stand-by facility on each chiller. The chillers are also provided with logic control system and a remote monitoring and sequencing controller.

The chiller controls have open protocol to interface with the industrial grade control system and the building BMS system. The chillers are provided with all accessories such as flow switch, vibration isolators, disconnects switch, control circuit transformer, part-wind start etc.

The chillers are rated at 46°C ambient for the normal operation and 30°C ambient for low temperature operation. The chillers are recently provided with acoustic enclosures for the compressors and mufflers for the condenser fans. However, the mufflers were later removed due to the restriction caused to the condenser airflow, thus reducing heat transfer.

Fig.6.13 Chillers at BSF Project

CHAPTER-7

MODEL PROJECT – SYSTEM IMPLEMENTATION IN SAUDI FRENCH BANK

7.1 Introduction:

A model project was built at a cost of Sterling 500,000/- , with a generous funding from Messrs. Saudi French Bank (BSF) of Saudi Arabia, to develop a load shifting and demand reduction programme at their headquarters building in Riyadh.

The researcher conducted a series of studies and investigation on system performance, data collection and system monitoring under various operating conditions. This section of the research work continued for a further period of 4 years to analyse and verify the performance of the system at various operating conditions. Data sheets are enclosed in Appendix.

Fig.7.1 The Model Project (ice banks can be seen at the roof level)

The researcher completed the initial analysis and system design in January 1996, and the project was awarded to a local contractor in February 1996. The installation work of this first cool storage project in the Middle East was completed in five months and the system was commissioned in July 1996.

The model project was established to provide the following two major features to the existing building:

Task-1: Provide additional cooling to meet the increase in the building-cooling load due to expansion, at a maximum rate of 200 KW_c for a continuous period of 10 hours without exceeding the present building peak electrical demand.

Task-2: Provide cooling for the building from 1pm till 5 p.m. at an average rate of 350 kW_c to reduce the peak electrical demand.

The model project was designed on the basis of primary and secondary network interconnected by heat exchangers. The primary system is the low temperature network consist of 25% glycol and 75% water by weight, and the secondary system is a standard hydronic network.

In addition to the basic components and the primary and secondary circuits, several ancillary circuits such as glycol injection circuit, freeze protection circuit, expansion circuit , and air separation circuits are incorporated in the system.

Two independent hydronic circuits are established, one to serve the building fan coil units system and the other to sever the "demand reduction" circuit.

The system is controlled by an industrial grade Direct Digital Control (DDC) network. Furthermore an ice inventory meter also included in the network to monitor the availability of ice in the tanks.

The primary circuit consists of two 350kW screw type chillers, ten ice banks, glycol pumps and number of control valves. The positioning of the control valves in managed by the DDC, so that the charging and discharging process could be conduced by either shutting off or modulating these valves.

The system has been designed in a manner that the chillers start charging the ice banks from 6 p.m. until 6 a.m., and the total net work operates without the chillers on line from 6 a.m. until 6 p.m.

Fig. 7.2 Over View of the total System installed at the roof Level

7.2 Experimental results

The system has been under full operation from July 15 1996, and operational results/data have been collected on daily basis until the studies were completed in September 2000.

The PLC controller has been set in a manner that the following daily performance data were collected.

Chiller performance -	30 minutes interval
Storage rate -	30 minutes interval
Discharge rate-	30 minutes interval
Ambient Conditions -	hourly data

Results/Data for a typical summer day ($26^{th}/27^{th}$ July 2000) are presented in the Appendix-3, and analysed in Chapter-8

The experimental results were analysed to establish chiller de-rating capacity, chiller performance at various conditions such as glycol leaving temperature, ambient conditions, load demand etc.

7.3 System Design

7.3.1 System History and Parameters

The existing Central Utility Plant (CUP 1) chillers at Saudi French Bank are approximately 15 years old and consist of 2 x Trane Reciprocating air cooled chillers (both on duty) rated at 700 kW each, and 3 heat pumps each of 210kW with a nominal total on line rating of 1800 kW cooling.

Saudi French Bank desperately wanted to expand their chilled water production plant to cope with the additional cooling requirements resulting from the expansion and occupation of the building.

Historical growth of Cooling Load at Saudi French Bank :
(Data provided by BSF)

Year	Cooling Load (kW)
1987	1000
1992	1250
1996	1500
2001(recorded)	1800

Essentially, a new Central Utility Plant (CUP 2) had to be established to add at lest 400kW additional cooling to the existing maximum demand and CUP 1 and CUP 2 plant must handle a cooling demand of 1800 kW by 2001.

7.3.2 Preparatory Work

The researcher observed that there was a fundamental design error in the selection of the existing chillers. The design was carried out in Switzerland and the chiller selection was based on ARI* rating (30°C Ambient), where as it should have been selected for an ambient condition of 50°C. This basic error caused a drastic de-rating effect on the chillers. Without understanding the true reason for this de-rating, the maintenance department installed a spray cooler around the chiller to lower the inlet air temperature at the condenser by inducing some evaporative cooling. The approach produced satisfactory results at the early stage, but due to the fact that municipality water was used for the spry cooler, high content of calcium and SO_4 salts caused scale formation on the condenser coils resulted in further de-rating on chiller capacity.

Researcher approached the manufacturer to supply and install a high-ambient kit for the chillers with new condensers. Considering the age and model of the chillers, the manufacturer declined to install such up-grading system. Hence, to obtain the best performance out of these chillers, the condenser coils were replaced, and the spray coolers were provided with a reverse osmosis plant with a standard iron filer to reduce the TDS of the spray water from 750 to 350 ppm, thus avoid formation of scale on the condensers. The up-grading of spray cooler resulted in increased performance of the chillers, but still below the normal rated capacity.

7.3.3 CUP – 2 Approach

A separate location has been identified for the CUP 2 chiller plant. It has been agreed that a set of new chillers with all hydronic components will be installed at the roof area. Inter-linking the CUP 1 and CUP 2 chillers was considered appropriate, as it would provide operational flexibility and will allow the CUP 1 cooling demand to be transferred progressively to the CUP 2 chillers. It is expected that the existing CUP 1 chillers will be partially taken out of service and only allow them to operate as either follow-on or stand-by chillers.

The researcher carried out a new building load analysis by using Carrier E20-II load analysis programme (Refer to appendix)
Based on the new occupancy load and additional computer centre etc., the building cooling system had to meet a peak cooling load of approximately 1400 kW, where as only 1000 kW cooling was available, after up-grading of the chillers.

BSF has expressed a need to couple the CUP 1 and CUP 2 chiller plant control systems. The control selection of all CUP 1 and CUP 2 chillers and other hydronic equipment will be carried out from the control panel provided in the vicinity of the existing CUP 1 chillers. This request was closely investigated by the researcher with control Specialists and was included as part of the CUP-2 program.
BSF anticipate that there will be major problem obtaining electrical power for the new development. In anticipation of introduction of cheaper tariffs for night-time electricity, the researcher recommended Thermal Energy Storage. This topic is analysed further in this thesis.
The task of the CUP-2 program was to develop a parallel cooling circuit with all necessary chillers, pumping and control modules, but within the available power limit, not exceeding 100kW.

7.3.4 Chiller options

This Section addresses the relative benefits of screw and centrifugal chillers as applied to BSF project.

7.3.4.1 *Centrifugal Chillers*

Centrifugal chillers are the most preferred chilled water producing equipment for normal air conditioning applications for their efficient operation on energy utilisation and partial load efficiency.
However, centrifugal chillers are pressure and temperature sensitive and require additional special components such as gears and speed controllers to operate below "preferred operating conditions" such as "low temperature" application.
Centrifugal chillers operate on staging and may require up to 3-4 stages on low temperature application.
The centrifugal chillers are the most reliable chillers in the industry, however, require continuous maintenance to up-keep them, which needs skilled technical support and high financial cost.

7.3.4.2 *Industrial Screw Chillers*

Screw chillers were the preferred options in the industrial sector for refrigeration applications. They are most efficient chillers in low temperature applications. Ice storage application in the air-conditioning industry brought a new interest in screw chillers in HVAC applications. These chillers require little or less maintenance as they have very few moving parts and operate under harsh conditions such as cold stores, gas condensing etc.
Relatively these chillers are less expensive then of the centrifugal chillers also require little skill and low financial burden on maintenance.

7.3.4.3 *Absorption Chillers*

Absorption chillers are becoming preferred chillers in the commercial air-conditioning application as they operate on lower kW/TR energy utilisation.

However, there are some pitfalls in absorption chiller applications. Absorption chillers are efficient while operate under a co-generation application. Waste hear from the power plant will be recovered through heat recovery boilers for production of steam.

Alternatively direct-fired boilers are used for steam generation.

In both cases fuel storage on site becomes a necessity and require careful panning in relation to fire safety, pollution etc.

Absorption chillers require evaporative cooling towers, and consume large amount of water for evaporation.

4.4 This issue has to be clearly addressed as water is scarcely available in the Kingdom and also there are restriction imposed by the Govt. on water usage for air-conditioning applications.

7.3.4.5 *Chiller Selection*

The researcher has selected screw chillers for CUP-2 for the following reasons:-

1. Efficient operation on ice making conditions
2. Efficient operation on air-cooled application
3. Availability of required capacity
4. Cost effectiveness.
5. Local legislations and constraints

The option of using either centrifugal or absorption was omitted due to the following reasons:

1. The required capacities are not valuable in the market
2. Air cooled versions are not available in the market

7.3.4.5 *Chilled water distribution*

It is the intention that the CUP 2 chiller plant is linked to the existing CUP 1 chilled water distribution network so that eventually, the CUP 1 cooling demands could be handled by new chiller plant as lead chillers. The older plant could be retired from active service and will be used as

either stand by or as a supporting chiller circuit at full load condition once adequate chillers are installed in CUP-2 plant.
This may involve the introduction of heat exchangers to separate the "old" circuits from the new ones, if the new hydronic circuit is based on glycol, which is not recommend due to high glycol charges and loss of chiller efficiency due to de-rating and heat exchanger applications.
The following issues were also addressed:
De-coupling of Chillers
Variable Speed Pumping
Expansion/Contraction of Pipework
Thermal Insulation
Water Treatment

7.3.4.6 De-coupling of Chillers

In connecting chillers to a variable volume chilled water system, the following rules should be followed:

- Ensure that there is always a constant flow of water through each chiller evaporator by installing a de-coupling loop
- The pipe-work must be configured to eliminate any possibility of the back flow of system return water into the flow water
- The return water temperature to chillers operating in parallel must be the same for all of the chillers

De-coupling Loop
Accurate chiller control depends upon constant flow through the evaporator. This constant flow will prevent the freezing in the evaporators caused by a rapid reduction in chilled water flow. The system return water connection is a straight line into the primary circuit. The return Tee connection acts as a simple eductor and ensures that back-flow does not occur in the bypass.

Instead of using the conventional de-coupling methodology, the researcher has decided to use a new approach to couple the CUP-2 system with CUP-1 for peak load support.

A central constant flow loop was created; the pumping circuit, chiller circuit the ice bank circuit and the heat exchanger circuit all coupled with this central constant flow loop through a 3-port motorised valve (this connection could have been created through pumps, but energy consumption would have increased by at least 5 folds against using a 3-port valve)

Each circuit operates as an injection loop, either injecting from or into the main loop.

6. This approach ahs eliminated a large amount of pumping requirements.

7.4 Design Considerations

7.4.1 Variable Speed Pumping

In the last 2 or 3 decades, the cost of variable speed drives has reduced considerably.

The speed control of the circulating pumps in variable flow chilled water systems allows pumping energy to be saved and eases the operation of control valves.

For these reasons, the use of variable speed pumps has become widespread. However, speed control devices are not inexpensive and their application must be carefully considered.

For BSF, the chiller the plant was de-coupled from the secondary chilled water circuits through a heat exchanger. The chiller pumps were selected as e constant speed and matched to the chillers in a way that is acceptable to the chiller manufacturers.

The secondary pumps were selected as variable speed.

7.4.2 Expansion and Contraction of Pipe-work

Due to low temperature application and large temperature variation between day and night time operation, a full 3-dimensional pipe-work stress analysis was carried out on all the new CUP 2 chilled water distribution pipe-work to ensure that brackets, guides and anchors are adequately sized. Wherever possible, horizontal pipe-work was supported by bands and drop-rods to minimise thrust forces.

7.4.3 Thermal Insulation

The extreme low temperature application presented a high risk of moisture condensation on cold surfaces. Chilled water pipe work, ice tanks and chiller evaporator were adequately thermally insulated and vapour sealed to prevent sweating and corrosion. 25mm 60kg density closed cell insulation was used on all exposed surfaces.

7.4.4 Water Treatment

The new chilled water pipe work was subjected to a rigorous flushing, pre-commission cleaning and chemical treatment regime.
Initially, the system was flushed with main's cold water via a break tank to remove traces of loose dirt, scale and extraneous matter.
The pipework circuits was provided with facilities to enable sections of pipework to be isolated and for circuits to be individually flushed in a methodical manner. Pipework circuits serving a number of terminal units, fan coil units, cooler coils etc. was provided with valved loop returns at the ends of the circuits to thoroughly clean pipework before flushing terminal units.
The Water Treatment specialist carried out the pre-commissioning cleaning of the chilled water pipework systems.

The purpose of the cleaning process was to remove light surface mill-scale and small debris such as cement splashes from the internal pipework and material that may have entered the systems during the course of installation. It should be noted that this process would not safely dissolve pebbles, pieces of wood and metal by the use of any chemical and it is important to pay particular attention to the protection of all pipework throughout the installation period to prevent the ingress of foreign matter.

The pipework system was chemically cleaned using a pre-operational cleaner which was allowed to remain in the systems for approximately 12 – 16 hours including a minimum period of 8 hours of pumped circulation.

The systems was then be drained and flushed until tests at all available sampling points showed that all traces of suspended matter have been removed.

The system was refilled with water only and circulated for one hour and again drained down completely. When empty, the system was refilled with fresh clean water. Circulation was repeated. Constant flushing was continued until tests throughout the systems indicate that the water quality was acceptable.

The Water Treatment specialist provided a detailed water analysis report of the water supply to the CUP 2 development and recommendations for treatment of each system.

This proposal included:

1. Description of chemicals required
2. Quantity of chemicals required for one year of operation
3. Control parameters of the water treatment program
4. Test equipment to be furnished to the Owner for maintenance of proper water conditions in each system
5. The proposed treatment should be designed to provide corrosion, scale and biological fouling protection within all parts of the systems. Liquid sulphuric acid and biocides containing tin are not considered acceptable.

The chilled water piping system fill water was pre-treated with a de-ionising unit as 25% (by weight) inhibited ethylene glycol was added to the system and continuously monitored and controlled to maintain desired brine concentration and prevention of biological growth.

The water sample was analysed to confirm that the correct condition and chemical concentration was achieved, before a completion certificate was issued.

Regular monitoring and testing of the chilled water quality was carried out by BSF's in-house operation and maintenance staff and further chemicals added as required to ensure on-going protection.

The Building Research Establishment in the UK publishes comprehensive guidelines for the cleaning and treatment of chilled water systems. Reference was made to their document "Pre-commission Cleaning of Water Systems (AG08/91)".

7.4.5 Hydronic Balancing of Chilled Water Systems

It was understood that there was a reduced flow of chilled water to some of the existing CUP 1 air handling units and performance testing of the systems was carried out to enable further remedial action.

The performance testing of existing systems followed the procedures adopted for new installations, as given below:-

Commissioning of the various services was carried out strictly in accordance with the U.K. CIBSE current Commissioning Codes:

A- Air Distribution Systems – High and Low Velocity

B- Boiler Plant

C - Automatic Control Systems

R- Refrigeration Systems

W- Water Distribution Systems

The following guidance were followed:

1. Test all equipment in accordance with manufacturers recommendation, checking rotation, revs, running current on each phase of electrical motors.
2. Run HVAC systems with air filters to disperse all dust, etc. and commission the system before the introduction of the filters.
3. Measure air volumes and pressures across fans. Proportionally balance air systems
4. Check airside and waterside pressure drops across all items of equipment and all individual air handler sections
5. Measure air on/off, water on/off temperatures to all cooler coils, heat exchangers, chilled evaporators and chilled water circuits.
6. Check waterside mass-flows across chillers, pumps, cooler coils, control valves and commissioning sets. Proportionally balance hydronic systems.
7. Demonstrate the start-up, control and shut down of each chiller including the operation of safety devices. The manufacturer's engineer was present on site to pre-commission, commission, test and demonstrate the chillers in accordance with an agreed programme. The demonstrations was carried out to coincide with the demonstrations of the Building Management System so that the performance of the complete system was judged.

It was requested to demonstrate <u>every</u> function and interlock relating to the stand-alone automatic controls and BMS by BSF, including:

1. All automatic temperature control functions, including high limit and safety cut-outs
2. All start-up and shut-down functions, including those instigated by timers, manually and by emergency and fire alarm signals
3. The operation and re-setting of smoke fans and dampers
4. All BMS control strategies, including calibration of sensors

5. All controls associated with variable air volume systems demonstrated and checked at full, 2/3 and 1/3 partial load operation

Checks were made to ensure correct operation of the AHUs, VAV terminal units, including air temperature and pressure checks.
Air circulation and distribution was demonstrated after the systems had been balanced to provide the design air quantities. Pitot tube readings were taken in all ductwork mains and branches, together with terminal point readings. Smoke tests may be required to provide distribution within enclosures.
Static pressure readings, at design supply volume were taken and recorded for <u>every</u> variable Volume Terminal Unit.
Sufficient numbers of flow measuring and flow balancing devices were installed on the chilled water systems to allow regular checks of chilled water flow rates to be made. Orifice plates, venturi meters and flow measuring devices formed part of a double regulating valve assembly.
These devices were calibrated at the manufacturers works and charts were made available showing the flow rates for differing pressure drops.
The commissioning Engineer using a mercury manometer measured the pressure drops across these devices on site.
Measured pressure "rises" across pumps can also be measured on site and flow rates obtained from the pump manufacturers "certified" curves.

7.4.6 System selection - Analysis

The intentions of utilising a TES system in this project are as follows:
1. Reduction in-peak demand
2. Take advantage of the night time low ambient temperature
3. Reduce the number of chillers operating during the day time
4. Limit the power requirement
5. Increase the peak load cooling availability
6. Easy O&M approach

The TES system philosophy explained earlier allowed BSF to select from several TES technologies available in the market.
The researcher was provided with large options of TES system for this project based on their own merits.
The chilled water system was as option initially favoured by BSF due to its low initial capital cost and lower energy consumption during charging period. However, this option was discarded due to the fact that chilled water storage requires at lest 6 times more storage volume that of an ice storage system due to its warm temperature operating condition. To store 7000 kWh (nominal) a tank size of 10m dia. x 12m high was requested. There was no space available at ground level to install such large tanks. Hence, the chilled water option was omitted due to lack of space availability at the project.
Ice storage system was offered by the industry with several choices (refer to section-)

The CALMAC system was selected for the following reasons:

1. Non-corrosive material used for construction
2. High thermal heat transfer from each tank during discharging
3. Lower de-rating of chillers during charging
4. External tank insulation
5. Lower installation cost
6. Modular tank arrangements
7. Light weight
8. Lower initial cost
9. 10 years system warranty provided by the Manufacturer

7.4.7 Calmac system performance analysis

PROJECT NAME : Saudi French Bank - TES Project

ANALYSIS BY : S.S.Bavan

CALMAC MANUFACTURING CORP.

101 West Sheffield Ave.

Englewood, NJ 07631

201 569-0420

FILE NAME : BSF1

COMMENTS :

FILE DATE : 10/1/96

DESIGN DAY LOAD DATA

HOUR	LOAD	TYPE	CHILL %	HOUR	LOAD	TYPE	CHILL %
1	.00	I	60.0	13	525.00	F	.0
2	.00	I	60.0	14	525.00	F	.0
3	.00	I	60.0	15	525.00	F	.0
4	.00	I	60.0	16	525.00	F	.0
5	.00	I	60.0	17	200.00	F	.0
6	.00	I	60.0	18	200.00	F	.0
7	.00	I	60.0	19	200.00	F	.0
8	200.00	F	.0	20	.00	F	.0
9	200.00	F	.0	21	.00	F	.0
10	200.00	F	.0	22	.00	F	.0
11	200.00	F	.0	23	.00	I	60.0
12	200.00	F	.0	24	.00	I	60.0

CHILLER AND TANK SELECTTON SUMMARY

LEVLOAD TANK MODEL		1190
DESIGN LOAD	(KW)	525
SYSTEM SUPPLY TEMPERATURE	(C)	3.3
SYSTEM RETURN TEMPERATURE	(C	5.5
DEFAULT CHILLER COOLING CAPACITY	(% OF NOMINAL)	100
DEFAULT CHILLER ICEMAKING CAPACITY	(% OF NOMINAL)	60
NUMBER OF COOLING HOURS		[illegible]
NUMBER OF ICE-MAKING HOURS		[illegible]
TOTAL COOLING LOAD	(KWH)	[illegible]

NOM CHLR KW	COOL CAP KW	ICE CAP KW	STRG DIV	ESTMTD KWH	STRG INLET C	STRG OUT C	PEAK STRG KW	MIN # TANKS	MAX # TANKS
685.19	685.19	411.11	[illegible]	453.06	6.90	3.30	525.00	8.17	8.17

FLOW ANALYSIS

685 KW CHILLER
9 STORAGE TANKS MODEL 1190
25 PERCENT ETHYLENE GLYCOL

CHG dT	CHG L/S	L/S/TANK	dP(KPA)	AVG LCWT	MIN LCWT
2.0	52.3	5.8	126.0	-3.1	-4.8
2.5	41.8	4.6	89.9	-3.5	[illegible]
3.0	34.9	3.9	68.9	-3.8	-5.3
3.5	29.9	3.3	55.5	-4.2	-5.6
4.0	26.1	2.9	46.3	-4.6	-5.8
4.5	23.2	2.6	39.7	-5.1	-6.1
5.0	20.9	2.3	34.6	-5.5	-6.4
5.5	19.0	2.1	30.7	-5.9	-6.6

DIS dT	DIS L/S	L/S/TANK	dP(KPA)
4.4	27.0	3.0	40.1
4.9	24.5	2.7	35.2
5.4	22.5	2.5	31.4
5.9	20.7	2.3	28.3
6.4	19.2	2.1	25.8
6.9	17.9	2.0	23.6
7.4	16.8	1.9	21.8
7.9	15.8	1.8	20.2

Saudi French Bank - TES Project 3 7/5/2001

DESIGN DAY SYSTEM ANALYSIS

CHILLER UPSTREAM - SERIES FLOW		NOMINAL CHILLER SIZE	= 685
SYSTEM SUPPLY TEMPERATURE (C) -	3.3	NUMBER OF TANKS - 9	MODEL 1190
SYSTEM RETURN TEMPERATURE (C) -	5.5	PERCENT ETHYLENE GLYCOL	= 25.0
FLOW (L/S) : DISCHARGE =	61	DELTA P (KPA) : DISCHARGE	= 150.8
CHARGE -	61	CHARGE	= 161.7

HOUR & TYPE		LOAD KW	CHLR KW	STRG KW	TANK KW	KWH TOTAL	KWH PER TANK	% CHRG	CHLR TEMP	REQD STRG TEMP	AVLB MIN TEMP	RTRN TEMP	L/S PER TANK	PD KPA	F L G
1	I	0	411	411	45.7	1233	137.0	20.5	-1.9	-.2	-.2	-.2	6.7	159.8	
2	I	0	411	411	45.7	1644	182.7	27.4	-2.0	-.3	-.3	-.3	6.7	159.9	
3	I	0	411	411	45.7	2056	228.4	34.2	-2.0	-.3	-.3	-.3	6.7	160.0	
4	I	0	411	411	45.7	2467	274.1	41.0	-2.1	-.4	-.4	-.4	6.7	160.1	
5	I	0	411	411	45.7	2878	319.8	47.9	-2.3	-.6	-.6	-.6	6.7	160.4	
6	I	0	411	411	45.7	3289	365.4	54.7	-2.5	-.8	-.8	-.8	6.7	160.7	
7	I	0	411	411	45.7	3700	411.1	61.6	-2.8	-1.1	-1.1	-1.1	6.7	161.2	
8	F	200	0	-200	-22.2	3500	388.9	58.2	4.1	3.3	****	4.1	****	****	T
9	F	200	0	-200	-22.2	3300	366.7	54.9	4.1	3.3	****	4.1	****	****	T
10	F	200	0	-200	-22.2	3100	344.5	51.6	4.1	3.3	****	4.1	****	****	T
11	F	200	0	-200	-22.2	2900	322.2	48.3	4.1	3.3	****	4.1	****	****	T
12	F	200	0	-200	-22.2	2700	300.0	44.9	4.1	3.3	****	4.1	****	****	T
13	F	525	0	-525	-58.3	2175	241.7	36.2	5.5	3.3	2.1	5.5	4.4	78.1	
14	F	525	0	-525	-58.3	1650	183.3	27.5	5.5	3.3	2.6	5.5	5.0	95.1	
15	F	525	0	-525	-58.3	1125	125.0	18.7	5.5	3.3	2.9	5.5	5.7	113.[illegible]	
16	F	525	0	-525	-58.3	600	66.7	10.0	5.5	3.3	3.3	5.5	6.7	150.5	
17	F	200	0	-200	-22.2	400	44.5	6.7	4.1	3.3	****	4.1	****	****	T
18	F	200	0	-200	-22.2	200	22.2	3.3	4.1	3.3	****	4.1	****	****	T
19	F	200	0	-200	-22.2	0	.0	0.0	4.1	3.3	****	4.1	****	****	T
20	F	0	0	0	.0	0	.0	0.0	3.3	3.3	****	3.3	****	****	T
21	F	0	0	0	.0	0	.0	0.0	3.3	3.3	****	3.3	****	****	T
22	F	0	0	0	.0	0	.0	0.0	3.3	3.3	****	3.3	****	****	T
23	I	0	411	411	45.7	411	45.7	6.8	-1.8	-.1	-.1	-.1	6.7	159.6	
24	I	0	411	411	45.7	822	91.4	13.7	-1.9	-.2	-.2	-.2	6.7	159.7	

TANK DISCHARGE DATA

HOUR	TYPE	KW /TANK	INLET TEMP(C)	OUTLET TEMP(C)	% TANK DISCH	KWH DISCH	OUT OF RANGE
8	F	22.2	4.1	3.3	3.3	22.22	*TEMP
9	F	22.2	4.1	3.3	6.7	44.44	*TEMP
10	F	22.2	4.1	3.3	10.0	66.67	*TEMP
11	F	22.2	4.1	3.3	13.3	88.89	*TEMP
12	F	22.2	4.1	3.3	16.6	111.11	*TEMP
13	F	58.3	5.5	3.3	25.4	169.44	
14	F	58.3	5.5	3.3	34.1	227.78	
15	F	58.3	5.5	3.3	42.8	286.11	
16	F	58.3	5.5	3.3	51.6	344.44	
17	F	22.2	4.1	3.3	54.9	366.67	*TEMP
18	F	22.2	4.1	3.3	58.2	388.89	*TEMP
19	F	22.2	4.1	3.3	61.6	411.11	*TEMP

Fig.7.3 Calmac System Analysis

7.4.8 Energy Utilisation:

Chilled water storage systems are charged at 4°C, hence consume around 1.2kW/TR. However, charging an ice storage system require –5°C brine leaving temperature, results in de-rating of chillers. On the energy side the chillers may utilise approximately 1.3kW/TR, but the charging time is extended by approximately 25%.

There are two defined penalties:

1. To charge the system within the available period, the installed chillier capacity has to be increased by 25%.
2. The energy consumption is increased by average 23% for Saudi conditions.

7.4.9 Pit-falls of chilled Water Storage systems:

Even though on capital cost and energy utilisation chilled water storage become the favourite option for large TES projects, in the Kingdom due to lack of technical support and availability of resident specialists makes the system construction, operation and maintenance a difficult tasks.
The researcher's experience in the two chilled water storage projects in this country confirms the above.

The “roll-over” process to dilute the “thermocline” during partial load conditions prior to re-charging of the storage system requires technical expertise.
In addition careful control and monitoring of chemical treatment is a must in chilled water storage and failure would result in algae growth within the storage tank and eventually will be carried into the net work through chilled water circulation process which could result in major clogging problems in valves and heat exchangers.
These are critical issues to be carefully considered by the end users before embarking into a chilled water storage system project.

The above also formed as the supporting reasons to omit chilled water storage at this model project.

7.4.10 Ice storage as useful option

Ice storage may have its disadvantages in capital and operating costs. However, if certain cautions and application process can be adopted the ice storage can be used as cost efficient as a chilled water storage application.

The benefit of an ice storage option is that it would use 6 times less volume for storage compares with chilled water application.

The size of this project favours the application of factory build storage system, as the selected type of ice banks could be installed in a modular manner with a storage capacity of 670kWh capacity of each tank.

The requirement of instantaneous discharge capacity at peak load condition is around 350 kWh, favours the use of internal melt system and the system will be designed in a manner that it will take the full advantage of low temperature application.

The system was developed in a manner that it made use of standard equipment, not requiring any special operator or maintenance skills. The ice storage itself is static, requiring a minimum of maintenance such as

- Check water level and conditions
- Check operation ice inventory

7.4.11 Low Temperature Application:

The central TES distribution system will be designed to provide 3°C ice water to the network.

As, $M_f = Q/s.(t_1-t_2)$

M_f - Mass flow in kg/s

Q - Power in kW

S – specific heat of brine

(t_1-t_2)– temperature difference in °C

This allowed the designer to reduce the distribution pumps and pipe work by at least 40% based on the design condition, compare with the conventional design approach.

The system was designed to meet the following requirements:

- Thermal Storage Capacity: 3500 kWh (actual) with a maximum discharge capacity of 525 kWh
- The principle of ice on coil was used
- Internal melt principle was used to maintain a single circuit for charging and discharging
- As cooling medium, a 25% Ethylene Glycol (water mixture was used)
- Chillers - screw type.

7.4.12 Design Parameters:

- Ice build time : 10 hours
- Peak Cooling Load: 1500 kW
- Max. Peak load assist from TES system: 550 kW
- Max.Peak load assist from water Chillers : 1000 kW
- Max.Peak load assist from glycol chillers: 550 kW (emergency use)
- Chilled water loop temperatures:
  13°C - return

3°C - supply

Assuming an ice build time of 10 hours two chillers were operated with a total de-rated capacity of 425 kW at a supply glycol temperature of -5°C.

During the day, these ice-making chillers were allocated as stand-by chillers and could be operated at a positive mode to deliver an emergency load cooling capacity of 550kW.

The existing old chillers were allocated as primary chillers.

The system was provided with two heat exchangers to separate the glycol circuit from the chilled water network.

One heat exchanger was exclusively used for peak demand reduction during 1-4 p.m. The other heat exchanger was used for the fan coil unit circuit provide between 175 –200 kW load based on the building load requirement and ambient conditions.

The DSM circuit was operated with direct injection of the chilled water in the return chilled water loop at an injection capacity of 350kW.

7.5 Injection Circuit Philosophy

The system design is based on injection circuit philosophy developed by the designer; the injection circuit allows the designer to couple all production and distribution net work to a common constant flow loop. This would minimize the number of pumps to be installed in a network thus reduces the overall energy consumption of the system..

The chiller circuit will operate as an independent injection circuit. Through a 3-port motorised valve, it would cool the circulating fluid to a fixed temperature based on either charging or discharging mode of the circuit.

The temperature sensors will maintain the outlet temperature of the chiller. The charging is set at -5.5°C and the discharging is set at 3.8°C.

The ice bank circuit also operates as an independent injection circuit through another 3-port motorised valve. Based on the mode of operation the valve will act as either by pass or a mixing valve.

Similar to the above tow circuits the heat exchangers also coupled with the constant flow glycol loop through another 3-port motorized valve.

During charging or discharging conditions, these circuits either inject into or injected from the constant flow glycol loop.

This methodology eliminated four sets of pumps from the TES circuit thus reduced four sets of electrical connections to the system.

Fig.7.4 Injection circuits at BSF

7.6 Selection of Heat transfer Fluid for Ice Storage Application

This section describes the corrosion-inhibited heat transfer fluid selected for the use of ice making purposes, that provides freeze and burst protection. Automotive antifreeze, uninhibited glycol, and freeze-inhibited glycol do not meet the specific requirement needed for low temperature application in chilled water pipes.
The heat transfer fluid selected for this project is industrial grade ethylene glycol, 99.5% purity.
The selected type glycol was inhibited for ferrous and copper-based metal protection. The glycol also contains buffers to extend the life of ethylene glycol component by restricting fluid oxidation.

Coolant concentration is determined by first deciding what freeze and /burst protection is appropriate for this specific application, based on operating temperature and ambient conditions.

Glycol Concentration Needed									
Operating Temperature in °C	-7	-12	-18	-23	-28	-34	-40	-46	-51
Freeze Protection	16	25	32	39	44	48	52	55	58
Burst Protection	11	16	21	26	31	36	37	38	40

Table 7.1 Glycol Concentration Data

The heat transfer fluid was provided with 25% glycol concentration will provide –12°C freeze protection and -23°C burst protection for the system.

Based on the charging temperature of –5°C for the TES system at BSF this would be an adequate glycol concentration for both freeze and burst protection. The high concentration will allow for emergency protection during leakage and other form of glycol losses.

The following properties were used in the system design development:

1. Specific Gravity : 1.133
2. Specific Heat : 3.912kJ/(kg.K)
3. Thermal Conductivity: 0.039 W/(m.K)

As water plays a major role in the heat transfer performance, the dilution water had to be of very high quality. Poor quality water contains ions that make the fluid "hard" and corrosive and contradicts the purpose of inhibiting of glycol. Calcium and magnesium hardness ions build up as scale on the walls of the system pipe work and reduce heat transfer. These ions also react with the corrosion inhibitors in the heat transfer fluid, causing them to precipitate out of solution and rendering them ineffective in protecting against corrosion. In addition high concentration of corrosive ions, such as chloride and sulphate, will eat through any protective layer that corrosion inhibitors form on the walls of the system pipe work.

The water available in Riyadh is hard consisting of large amount of iron, sulphate and carbonate ions. The average water quality exceeds 750 TDS, normally categorised as hard water.

Hence, de-ionised water was used for dilution, since de-ionising removes both corrosive and hardness ions.

CHAPTER-8

RESULTS AND TYPICAL SYSTEM PERFORMANCE ANALYSIS

8.1 Introduction:

The researcher continuously monitored the project from 1995 until year 2000. Major changes such as up grading of the chiller condenser coil, fitting of high ambient kit, installation of the R/O plant etc. carried out to improve the efficiency of the existing chillers, thus improving system efficiency and reducing the demand on TES system. The system performance dramatically increased after the above up grading procedures were completed.

The Marketrol system was set to record system performance on a 30 minutes interval. The system reordered the chiller performance, charging and discharging status, valves opening and closing positions, storage and discharge capacities.

The results were recorded and tabulated.

Annual performance data was analysed to make adjustments and modifications in system operational procedures. The adjustments and modifications were carried out particularly to reduce the chiller dependency during the daytime and to make use of the TES during high ambient conditions. As a result in political conditions in Saudi Arabia, a significant amount of **data that were recorded at later stage of the project are not available to include in this thesis. Despite this the available data included in the thesis is adequate to complete the thesis**. This project **is a first of its kind in the Middle East.**

The following summarizes the system requirements and operational procedures.

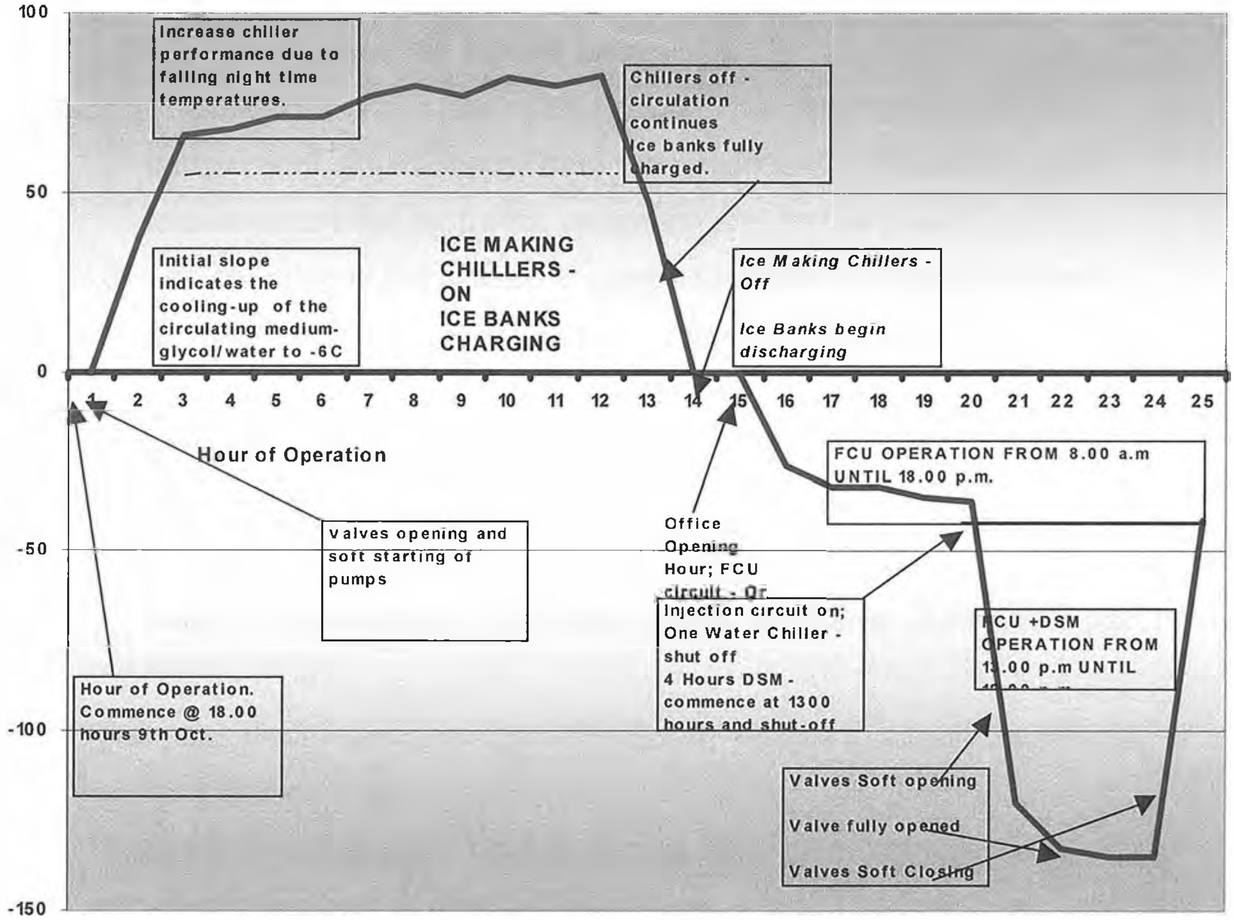


Fig. 8.1 Ice bank - rate of storage – 9th-10th August 1996

8.2 System operation Notes and Observations:

8.2.1 Storage of the 10 x Calmac 1190A tanks represents cumulative storage of 6700kWh$_c$; 5290 kWh$_c$ latent & 1410 kWh$_c$ sensible cooling. The efficiency of discharge was depended on the inlet and outlet temperature and the amount of energy discharge per hour. Larger the delta T (temperature difference between the inlet and the out let of the ice tank) faster the depletion of energy from the tank.

8.2.2 The system maximum accumulative demand from the storage was is around 3500 kWh$_c$

8.2.3 Full Storage of the tank took place on the first day of commissioning or during the initial charging of the system.

8.2.4 After the initial charge, the daily charge represented the top-up energy, related to the energy depleted providing cooling on the previous day and the system losses.

8.2.5 The initial storage (data point) shown in the chart represents the carryover of stored energy from the previous day of charging, with losses accounted for freeze protection and ambient loss.

8.2.6 The charging of the system is governed by the amount of depletion of energy from the previous day and the time allowed for system charging, normally 12 hours during the night commencing from 1900 hours.

8.2.7 The discharge of the system is governs by the FCU circuit operating time of 10 hours, starting at 0800 hours and the DSM period of 4 hours commencing at 1300 hours, during such time one main chiller is turned off.

8.2.8 The peak discharge is around 525 kWh_c/d, related to almost 225kW_e/d of DSM of the circuit.

8.2.9 A lapsed period of min 1 hour is maintained to allow changeover between charging and discharging modes.

8.2.10 This period may vary due to factors such as office opening hours, achievement of full charging of the tanks

8.2.11 The increase in demand represents in the trend of the cumulative storage is due to progressive increase in ambient temperature during the month of August.

8.2.12 By end of August the ambient temperature is beginning to fall, thus allowing the system operators to change to mode of operation from chiller priority to ice priority.

8.2.13 DSM only required for the peak summer period commencing from May-Sept.

8.2.14 During the winter period the TES was used during the daytime as full load shifting tool to proving cooling for the internal area to offset the internal heat gain such as lighting, occupants and ancillary equpt. (Computers etc.) Thus receiving further discount from SCECO.

8.2.15 This allows the shut down of the bigger chillers during winter, normally operated on partial load, eliminating PSS, and provide adequate maintenance period.

8 8.2.16 The winter cooling load of approx. 525 kWh_c, matches the TES system capacity.

8.2.17 The system originally designed for Chiller priority partial load operation; however, the system is now operated under the following three modes:

Chiller Priority –Partial Load ---	summer
Ice Priority –Partial Load----	Autumn & spring
Ice Priority –Full load ----	Winter

8.2.18 In terms of future of elec. demand management.. Access to real time pricing contracts. Total of elimination of PSS in large chiller plants.

8.2.19 It is important that in large projects two independent but operationally coupled cooling systems one based on Chillers and the other based on TES.

8.2.20 The TES system can be oversized to accommodate a possibility of trading with the power exchange market during peak hours to trade a block of MW reduction by using the TES.

8.3 Chiller Performance Observations and Discussions :

The forecasted performance of the chiller was based on 60% de-rating maximum; this forecast was established based on the following factors:

- Manufacturer's Data on chillers, and ice banks
- ARI rating in Chillers on various operating conditions
- History of ambient condition in Riyadh
- Design flow temperatures

A typical chiller when it charges the system at –5°C will be at a capacity efficiency of 60% of its nominal level. Refer to chart –1

CHILLER PERFORMANCE ANALYSIS May 1998

COOLING CAPACITY

AMBIENT	CHILLED WATER LEAVING TEMPERATURE C																	
C	-10	-8	-6	-4	-2	0	1	2	3	4	5	6	7	8	9	10	11	12
24	0.551	0.61	0.669	0.725	0.765						1.011	1.045	1.082	1.12	1.152	1.18	1.222	1.252
26	0.544	0.603	0.662	0.71	0.767						0.997	1.019	1.067	1.093	1.126	1.155	1.199	1.229
28	0.538	0.598	0.648	0.702	0.758						0.976	1.005	1.038	1.079	1.112	1.13	1.175	1.207
30	0.527	0.586	0.634	0.685	0.739						0.954	0.978	1.023	1.05	1.085	1.118	1.151	1.173
32	0.516	0.573	0.619	0.668	0.72						0.94	0.964	1	1.022	1.058	1.092	1.126	1.151
34	0.503	0.56	0.604	0.651	0.71						0.91	0.937	0.977	1.008	1.031	1.066	1.102	1.128
36		0.546	0.589	0.642	0.69						0.895	0.923	0.962	0.979	1.071	1.04	1.077	1.093
38		0.553	0.581	0.633	0.68						0.881	0.894	0.931	0.964	0.989	1.014	1.052	1.06
40			0.573	0.615	0.66						0.85	0.88	0.915	0.934	0.96	1.001	1.026	1.045
42				0.597	0.649						0.835	0.865	0.892	0.92	0.946	0.974	1.001	1.021
44					0.639						0.82	0.836	0.868	0.89	0.918	0.947		
46											0.805	0.807	0.844	0.88	0.89			

CHILLER PERFORMANCE (CAPACITY)

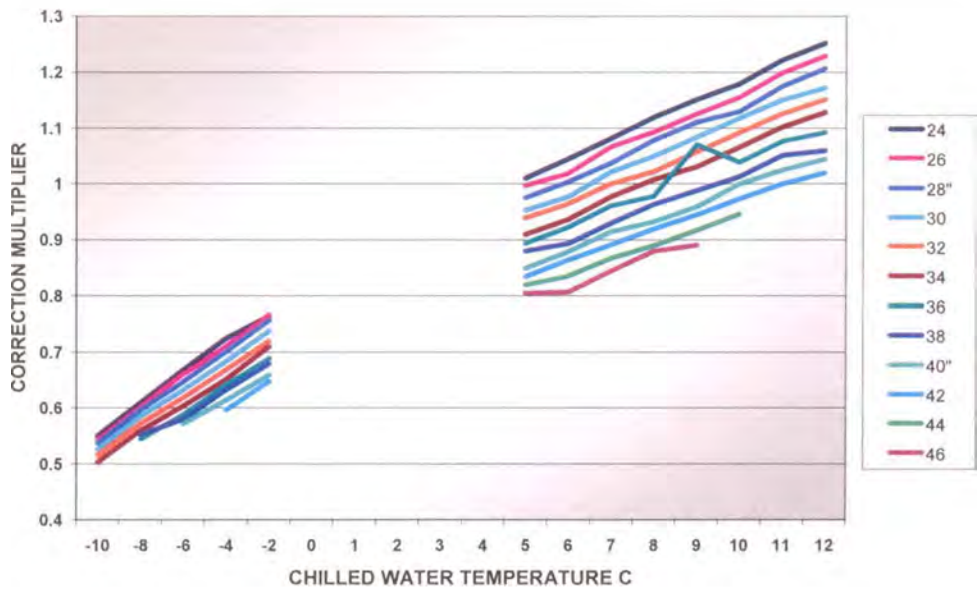


Fig. 8.2 Chiller Performance at various ambient Conditions

If allowances are made for elevation and glycol de-rating, and ambient temperature losses this might be reduced to 57% of its nominal capacity. If we can charge the same TES system by a warmer temperature of say – 4.5°C, the efficiency of the chiller will increase to 65% of its nominal capacity. However, one should not confuse the de-rating of the chiller capacity with energy consumption. When a chiller operates at lower discharge temperature the energy consumption also decreases.

						CHILLER PERFORMANCE ANALYSIS May 1998												
						COMPRESSOR POWER												
AMBIENT				CHILLED WATER LEAVING TEMPERATURE C														
C	-10	-8	-6	-4	-2	0	1	2	3	4	5	6	7	8	9	10	11	12
24	0.639	0.666	0.693	0.722	0.752						0.863	0.875	0.894	0.908	0.921	0.935	0.949	0.96
26	0.649	0.68	0.711	0.742	0.772						0.877	0.902	0.921	0.935	0.95	0.965	0.979	0.99
28	0.664	0.696	0.728	0.76	0.792						0.912	0.927	0.948	0.962	0.977	0.994	1.009	1.02
30	0.696	0.73	0.764	0.798	0.833						0.938	0.953	0.974	0.989	1.005	1.022	1.036	1.05
32	0.712	0.747	0.782	0.818	0.853						0.962	0.979	1	1.018	1.034	1.05	1.067	1.08
34	0.729	0.764	0.799	0.836	0.873						0.988	1.004	1.025	1.036	1.061	1.078	1.095	1.11
36	0.745	0.781	0.817	0.854	0.894						1.011	1.029	1.052	1.07	1.088	1.106	1.124	1.14
38		0.797	0.834	0.874	0.914						1.036	1.054	1.078	1.096	1.114	1.134	1.152	1.2
40			0.853	0.893	0.933						1.06	1.078	1.103	1.126	1.142	1.161	1.18	1.23
42				0.911	0.953						1.084	1.103	1.128	1.148	1.168	1.187	1.208	
44					0.973						1.107	1.128	1.153	1.173	1.194	1.214		
46											1.13	1.153	1.178	1.198	1.26			

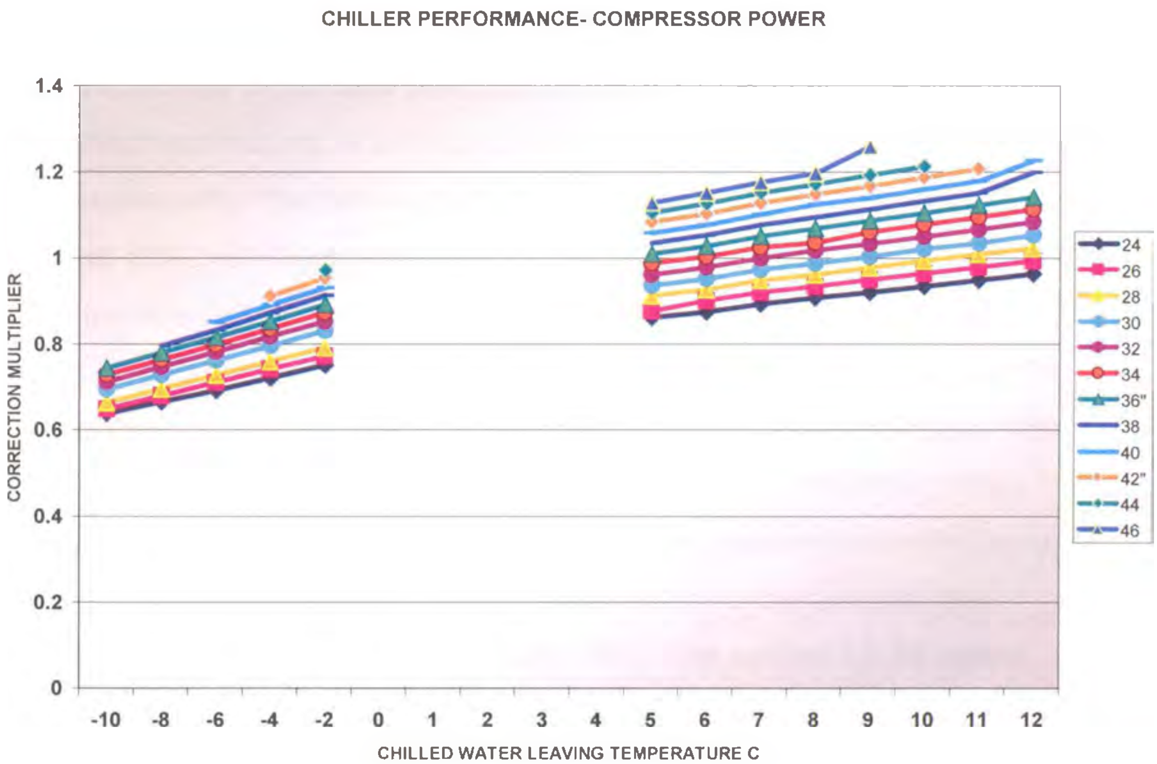


Fig. 8.3 Chiller Power Consumption at various operating Conditions

The efficiency of a TES system is based on its charging temperature; warmer the temperature the system efficiency increases.

But, if we compare the decrease of capacity against decrease of energy consumption, they are not proportional.

i.e the decrease in energy consumption is not as low as the decrease in energy losses.

The average energy losses are around 20% on a de-rating condition.
This criteria plays a major role in selecting the appropriate ice banks for this project.
The added advantage on TES application in the Middle East are as follows:
Chillers run at night with lower ambient temperatures; expected to be at least 20°C lower than of day time.

However, the recorded data shows that the de-rating of the chiller is as low as 50%. This has resulted in operating the chiller for a longer period to charge the ice banks.

Possible reasons for lower de-rating:

1. The accuracy of the data provided by the manufacturer: The data provided by the manufacturer is from an approximation derived from theoretical analysis rather than actual performance. The chiller performance is not linear with the temperature drop of leaving water temperature; at lower temperatures it drops almost exponentially. The actual performance within this area is hard to predict This item was question by the owner, as the predicted out put of chillers by the manufacturer was 490 $kW_{cooling}$ where as the recorded performance was only 350 $kW_{cooling}$ maximum. The manufacturer later accepted that the recorded data would be the accurate rather than their predicted performance. It should be noted in this particular project, this de-rating didn't affect the system performance, as it only extended the charging period. In the design the researcher has allowed adequate cushion on charging times and it has helped the chillers to fully charge the ice banks within the maximum allowed charging period.

2. The average recorded nighttime temperature is at lest 3-4 degrees higher than forecasted: this is one of the difficult parameters to control. The historical data shows that the average summer time temperature difference between the day and night time in Riyadh is around 10-12°C; However, the recorded temperature profile shows that the night time temperature during hot summer days does not fall below 36°C, giving an

average drop of 6-8°C. This higher nighttime ambient condition had a de-rating effect on chiller capacity performance and energy consumption. The following tabulated figure shown that the chillers are de-rated 20% below the manufacturer's predicated data.

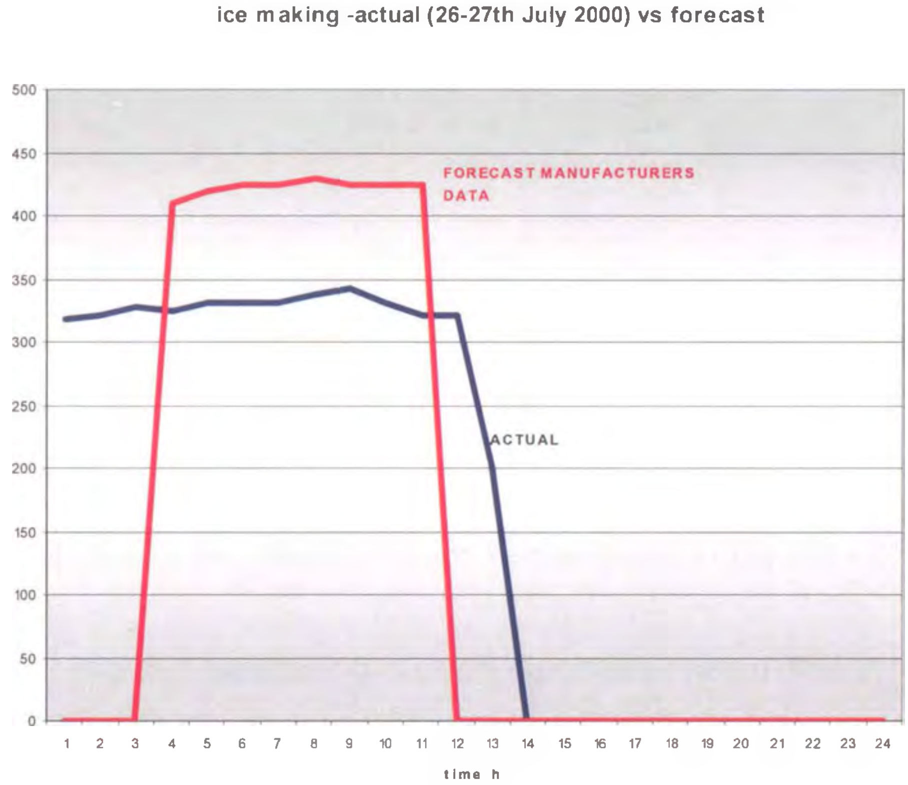


Fig.8.4 Chiller Actual Performance Against Manufacturer's Data

3. The chiller selection was based on nominal rating rather than high ambient conditions. The selection was based on the assumption that these ice-making chillers would not operate during the daytime and the nighttime average temperature will be within the nominal rating conditions. The nominal rating for Riyadh is based on 34°C; recorded temperature shows the average ambient in the night is in the range of 38°C. Refer to Fig. 8.5

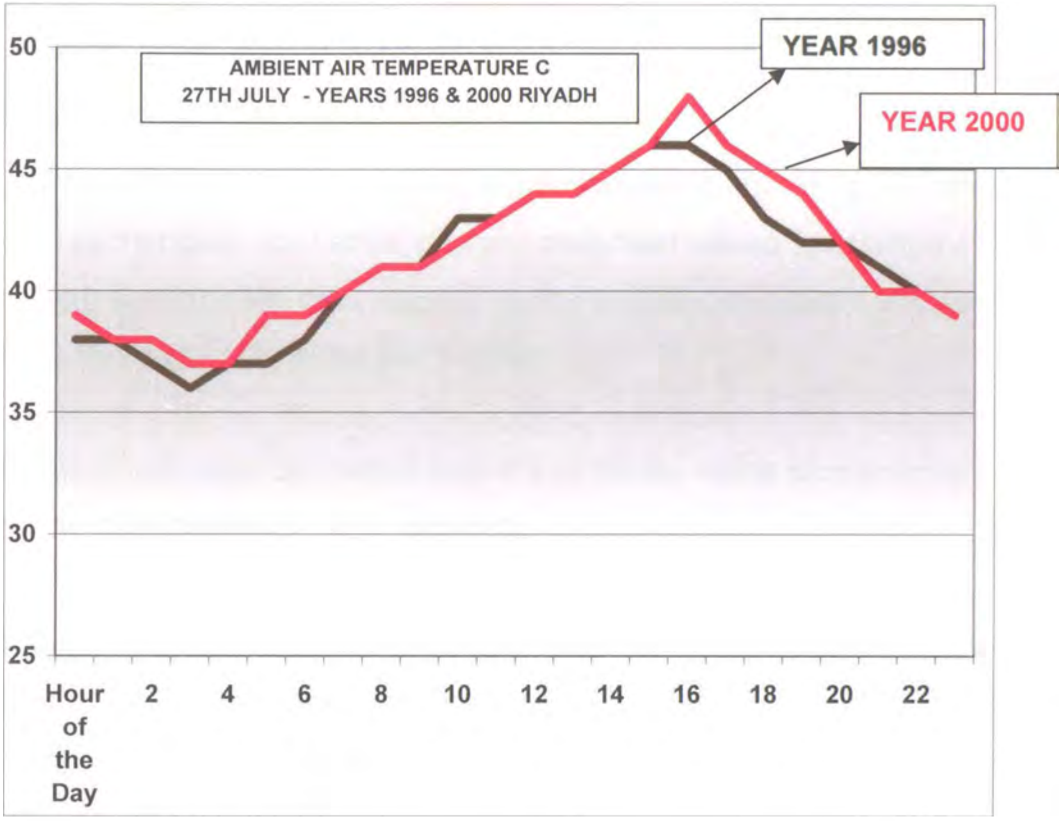


Fig.8.5 24 hours recorded Ambient Temperature in Riyadh - 2000

4. Controlling and maintaining of glycol percentage was a difficult task; an industrial refractor meter was used to establish this percentage. To safe guard the system, the operators maintained a glycol percentage well above the design conditions.

8.4 Performance Analysis

It has been clearly proven from the results that manufacturer's data on low temperature chiller performances in high ambient conditions are over estimated. This is due to the fact that the manufacturer's data are not based on actual system performance test, rather based on estimation. This statement is fairly accurate for air-cooled chillers as these chillers are rarely used in Europe and North America, and an investment by manufacturers to establish accurate data is not a worthy financial option, at least in their opinion. Hence the estimated data was provided to the end users.

The results clearly show the de-rated capacity of each chiller is around 45-50% of the nominal capacity of the chiller, where as the manufacturers data indicates this should be in the range of 70%.

A system check was carried out by all equipment/component suppliers of the TES system to confirm that all other equipment and components are operating according to the design parameter so that there chiller performance was not affected by a secondary equipment or components. All items such as heat exchangers, ice banks, pumps, motorised valves specially the control system sensors and data logging centre including hardware and software were thoroughly checked and verified.

Finally it has been concluded that the chiller de-rating performance predicted by the manufacturers were incorrect.

This finding has assisted the designers and the researcher in his other TES projects to size the chillers accurately based on the limited charging time. BSF project had a time cushion of approximately 3 hours that helped the charging process with the 50% de-rated chillers.
In addition the client has cancelled the phase-2 installation of the system, which was planned to be an extension to the existing TES system with additional 5 ice banks. If this system was implemented the two ice making chillers would not charge the total of 15 ice banks within the available 11 hours window.

However, as stated earlier the research has clearly established a de-rating data for air-cooled screw chillers for the specific operating condition in Riyadh. This new data was presented in technical forum in Riyadh for design engineers, and currently used by all designers in the TES system design developments. In addition the researcher has used the results in the followed on 9 TES projects and the results were accurately verified and confirmed.
The design temperatures for the Middle East still remain as a myth. ASHRE has modified the design data in the Fundamentals several times during the past ten years. The only recorded data available are from Aramco, but this is limited to the Eastern Province only.
There are several temperature data available for Riyadh, from different sources such as Ministry of Aviation, Airport, Ministry of Environment etc., but if act the data provided by these sources contradict each other, even on a

specific design day. This may be due to the fact certain temperatures are recorded in shade and others data logged in open space. The noticeable difference in these recorded data of mean radiant temperature is around 2-4°C. To ignore radiation effect on dry bulb temperatures Saudi meteorological society has decided standards all temperature records under "shade ' conditions. Basically the thermometers are installed outside but under a canopy so that solar radiation doesn't fall directly on the thermometer.

The recorded data at BSF model project averages a night-time temperature of approximately 38°C during charging period. The original design charging time was envisaged from 23.00 hours until 6.00 a.m. This would have given an average charging temperature of approximately 36°C, with a temperature swing of approximately 12°C.

Nevertheless, the actual recorded data is almost 6-8°C higher than the figures used in the system design development, based on the locally recorded data by the above stated institutions.

This was one of the reasons that chiller rating was far below that predicted by the manufacturer.

The above finding leads to the analysis for the third reason that contributed to the capacity loss of the ice making chillers.

The researcher has selected the chiller to be a "standard type" chiller for the following reasons:

- The ice making chillers will only operate in the night where the average temperature will be around 20°C less than the daytime temperature, thus will be operating at nominal ambient conditions. All chiller ratings were calculated based on the fact the chillers are operating at nominal ambient conditions during the night time.
- The initial cost difference between "standard type" chillers and "high ambient" chillers are around 10-15%. The selection of a standard type would have saved a substantial sum of money.

This has established knowledge that standard type chillers cannot be used for low temperature application even if the chillers are meant to operate during

cooler night time only. At BSF the chillers are now retrofitted with “high Ambient Kit” to minimise the de-rating.
Finally, the glycol concentration issue; this is a minor issue compare with the other three items discussed above. Glycol mix reduces the chiller performance by approximately 2-3%. Higher concentration will further de-rate the chillers. The glycol concentration in the system is now maintained within 5% of the design value (25% by weight) by checking the data on daily basis.

All the above contributed to a major deficiency on chiller operation at the project. It was originally anticipated that the large temperature swing and chiller operation at lower condensing temperature conditions would yield in large energy savings. The electricity cost savings as advised by BSF was in the range of 10% as shown in Fig.8.6

The researcher believes these savings are realized by an undeclared rebate by SCECO, rather than against actual energy savings. However, these savings can be compared with the nighttime low tariffs in Europe and North America.

8.5 System Analysis

8.5.1 System Brief

The TES system was installed at BSF to provide the following two services:

1. Provide up to 250kW additional cooling through a newly installed fan coil unit system to the building for a continuous period of 10 hours without increasing the maximum allowable electrical peak demand of the building.

2. Provide make up cooling (DSM) to the building during peak hours, to comply with SCECO’s requirement in the following manner:
   1-5 p.m. – 350kW (50% main chiller load shifting)

3. Investigate that installing a TES system can provide energy conservation.

8.5.2. Reduction in Energy cost/consumption

The results confirmed that there is defined benefit of using air-cooled chillers during the cooler period for charging the ice banks. The average recorded 10-12°C reduction in ambient conditions resulted in some form of energy conservation in chiller operation.

The following analysis shows that to produce 4000kW cooling instantaneously the chillers would have consumed:

$0.45\ kW_{power}/\ kW_{cooling}$

i.e, total consumption = 4000 x .45 x 28 =50,400 kWh per month (based on 28 working days)

= 50,400 x 7.5 /100 =US$ 3780 per month approximately. (Based on 7.5 cents /kWh)

BSF confirmed that compare to the above forecast they have paid 20-30% less charge for the corresponding months. Due, to the company privacy policy they couldn't disclose the actual kWh reduction in the monthly power consumption. This has raised a concern that the energy cost reduction may not necessarily contributed by reduction of energy consumption alone. There may be some sort of rebate was given by the utility company to reward BSF for proving the required DSM at their project.

ELECTRICITY CHARGES STATISTICS FOR HEAD OFFICE

MONTH	1995	% save	1996	% save	1997
JAN	93,661		128,161	7%	119,622
FEB	97,111		138,561	14%	122,022
MAR	122,961		130,761	7%	121,822
APR	134,561		135,961	22%	111,622
MAY	126,161		121,961	6%	114,622
JUNE	128,361		161,361	3%	157,302
6 Months	702,815		816,766	9%	747,012
JUL	184,361	11%	165,962		
AUG	169,961	9%	156,562		
SEP	172,181	8%	158,962		
OCT	171,961	4%	165,962		
NOV	154,361	-1%	156,362		
DEC	146,961	10%	135,381		
6 Months	[illegible]	7%	939,191		-
12 Months	1,704,552		1,755,957		747,012

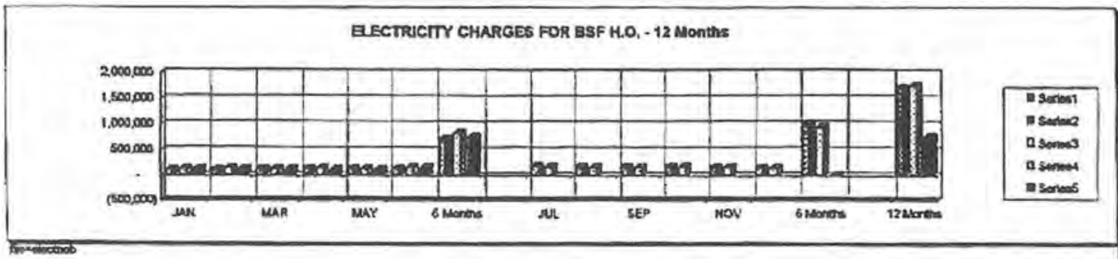


Fig.8.6 Comparison of Electricity Charges

The result related to energy conservation and energy charges were satisfactory. It should be noted that in Europe and US, where TES is operated with water-cooled chillers the net difference in energy consumption is around 20% higher than of a conventional system.

This is due to the fact that water-cooled chillers consume $0.15kW_{power}/kW_{cooling}$ and also the wet bulb temperature remains almost constant through out the day.

8.5.3. Load Expansion

From the results it can be noted that the building system-cooling load was expanded in two separate ways.

The fan-coil unit circuit was provided with an average 175 $kW_{cooling}$ load without increasing the existing chiller capacity during daytime operation, even though the system is capable of handling a total of $250kW_{cooling}$.

If we also consider the DSM load, which is approximately another 2000 $kWh_{cooling}$, the building cooling demand could be expanded to an average of 650 $kW_{cooling}$, or 40% of its existing capacity with the TES system. If the DSM requirement is relaxed by SCECO, the building owner could use this additional capacity to expand the cooling circuit without increasing the peak power demand. However, as the pipe and pumps are already installed the expansion of the circuit can only be achieved through low temperature chilled water application as described earlier in previous sections.

8.5.4. Load Shifting

The second injection circuit provided an average 650 $kW_{cooling}$ to the existing building HVAC circuit. The shifting was an average 50% of the existing cooling demand of the building. The requirement of SCECO was fulfilled neither increasing the power consumption nor interrupting office work.

8.5.5. Demand Side Management

A successful Demand Side Management result was achieved at this model project. The chiller capacity was increased by 33%, but the power demand was reduced by 25%. As one of the chillers was turned off during the peak load conditions an approximate chiller load of 300kW was reduced. The new injection circuit and the fan-coil circuits were provided with additional 100kW power.

Hence a net demand reduction of 200kW was achieved during the peal load condition. In addition, the chiller operation was managed in a manner that at anytime of the day the main water-cooled chillers or the ice making chillers are not allowed operating simultaneously.

8.5.6 Middle East Application

The results confirmed that a TES system could be successfully operated in the Middle East in the benefit of the end user. Different favoring conditions allow the building owners to use TES for energy conservation and Demand side Management.

The approach to the above benefits differs from the reasons identified in other countries for using TES as a DSM and Energy conservation tools.

The model project illustrated the proper approach to TES in the Middle East for successful operation. The relation between air-cooled chiller application and the temperature swing is to be considered. The local mandatory DSM requirement is another favoring factor for TES application in the Middle East. The reduction in installed chiller capacity thus reducing connected load also becomes a major favoring factor for power produces.

8.5.7 Load Profile Analysis

The project was commissioned in 1996 and has been in continuous operation. Several adjustments were made to optimize the system performance.

The analysis presented below is carried out based on the results obtained on 23rd July 2000, the hottest summer day for that particular year in Saudi Arabia, with a reordered dry bulb temperature of 49°C.

The first data set illustrates the design parameters, whereas the second shows actual results.

The predicted results were based on the data provided by the client in 1995. It should be noted that the actual results recorded for analysis in July 2000 accommodate all progressive changes in occupancy load and other expansion load in the building over the period of 4 years from system commissioning.

BUILDING/CHILLERS/ICE BANKS LOAD PROFILE

DESIGN PARAMETERS

(FORECASTED)

Time	Building FCU Circuit Load	Building DSM Load	Ice Making Chillers	Ice Storage	Ice Discharge
19	0	0	0	0	0
20	0	0	0	0	0
21	0	0	0	0	0
22	0	0	410	410	0
23	0	0	420	830	0
24	0	0	425	1255	0

1	0	0	425	1680	0
2	0	0	430	2110	0
3	0	0	425	2535	0
4	0	0	425	2960	0
5	0	0	425	3385	0
6	0	0	420	3805	0
7	0	0	0	0	0
8	175	0	0	0	175
9	175	0	0	0	350
10	175	0	0	0	525
11	175	0	0	0	700
12	175	0	0	0	875
13	175	350	0	0	1400
14	175	350	0	0	1925
15	175	350	0	0	2450
16	175	350	0	0	2975
17	175	0	0	0	3150
18	175	0	0	0	3325

Table 8.1 Load Profile - Forecasted

The TES system operation can be envisaged as follows:

1. From 7.00 a.m. till 8.00 p.m.
2. The building will be cooled by water chiller during day time operation
3. The water chillers are rated at 700kW each (nominal capacity)
4. A spare capacity of 500kW is available in the storage as stand-by when actual load exceeds the design conditions

Fig8.7. Shows the predicted charging (black line) and discharging cycle (red) with the chiller shutdown period for two hours from 6.00 a.m. until the commencing of the next cycle of charging on the next day (21.00 hours) and the fig shows the predicted cumulative discharge of the stored energy from 7.00 a.m until 18.00 hours, the business hours. The fig. also illustrates the two different type of discharging modes. From 7.00 a.m until 18.00 hours the discharge is 175kW from the ice banks via the fan coil unit circuit . The change of slope at 12.30 p.m shows the soft opening of the DSM valve to allow discharge cooling to meet the greater cooling load required to compensate for the shut down of one main chillers according to SCECO regulation to reduce electric power consumption between 13.00 – 17.00 hours. The DSM circuit shuts off at 1700 hours, but the fan coil circuit continues to discharge until 18.00 hours at a rate of 175 kW per hour.

BUILDING/CHILLERS/ICE BANKS LOAD PROFILE (ACTUAL 26th & 27th July 2000)

	TIME Hour 0	BUILDING Fan Coil Unit KWh	BUILDING DSM load KWh	ICE making Chillers KWh cooling	ICE STORAGE KWh stored (Cumulative)	
26-Jul	19	0	0	317	286	Charging commence
	20	0	0	321	603	
	21	0	0	328	927	
	22	0	0	324	1252	
	23	0	0	331	1580	
	24	0	0	331	1883	
27-Jul	1	0	0	331	2250	
	2	0	0	338	2585	
	3	0	0	342	2927	
	4	0	0	331	2909	
	5	0	0	321	3593	
	6	0	0	321	3794	
	7	0	0	201	3780	
	8	95	0	0	3780	Discharging commence
	9	92	0	0	3685	
	10	162	0	0	3593	
	11	197	0	0	3431	
	12	197	0	0	3233	
	13	197	56	0	3036	
	14	197	254	0	2782	DSM Commence
	15	190	251	0	2331	
	16	176	251	0	1883	
	17	159	2[illegible]9	0	1449	

	18	**148**	**0**	**0**	**1061**	Base storage
	19	0	0	335	Charging commence	
	20	0	0	337	600	
	21	0	0	340	923	
	22	0	0	340	1246	
	23	0	0	344	1572	
	24	0	0	344	1906	
28th July	1	0	0	344	2240	
	2	0	0	348	2573	
	3	0	0	351	2913	
	4	0	0	348	3246	
	5	0	0	340	3576	
	6	0	0	0	3776	
	7	91	0	0	0	
	8	204	0	0	0	
	9	165	0	0	0	
	10	200	0	0	0	
	11	196	0	0	0	
	12	200	60	0	0	
	13	199	250	0	0	
	14	200	250	0	0	
	15	185	250	0	0	
	16	160	230	0	0	
	17	150	0	0	0	
	18	0	0	0	0	

Table 8.2 Sample - Actual Load Profile July 27-28th 2000

BUILDING/CHILLERS/ICE BANKS LOAD PROFILE
DESIGN PARAMETERS(kWh)

Typical Day analysis: 30th July 2000

Time	Building Load	Water Chillers	Ice Making Chillers	Ice Storage	Ice Discha
0	150	150	345	2027	0
1	140	140	345	2373	0
2	130	130	349	2725	0
3	125	125	353	3078	0
4	140	140	349	3431	0
5	150	150	342	3776	0
6	250	250	550	5310	0
7	400	400	0	0	150
8	500	500	0	0	300
9	700	700	0	0	460
10	800	800	0	0	635
11	850	850	0	0	810
12	900	900	0	0	985
13	950	350	0	0	1585
14	1000	350	0	0	2235
15	1100	350	0	0	2985
16	1150	350	0	0	3785
17	1100	1100	0	0	3960
18	1050	1050	0	0	4135
19	950	950	0	0	4310
20	900	900	0	0	4485
21	250	250	500	500	0
22	200	200	500	1000	0
23	175	175	510	1510	0

Table 8.3 Sample - Actual Load Profile July 30th 2000

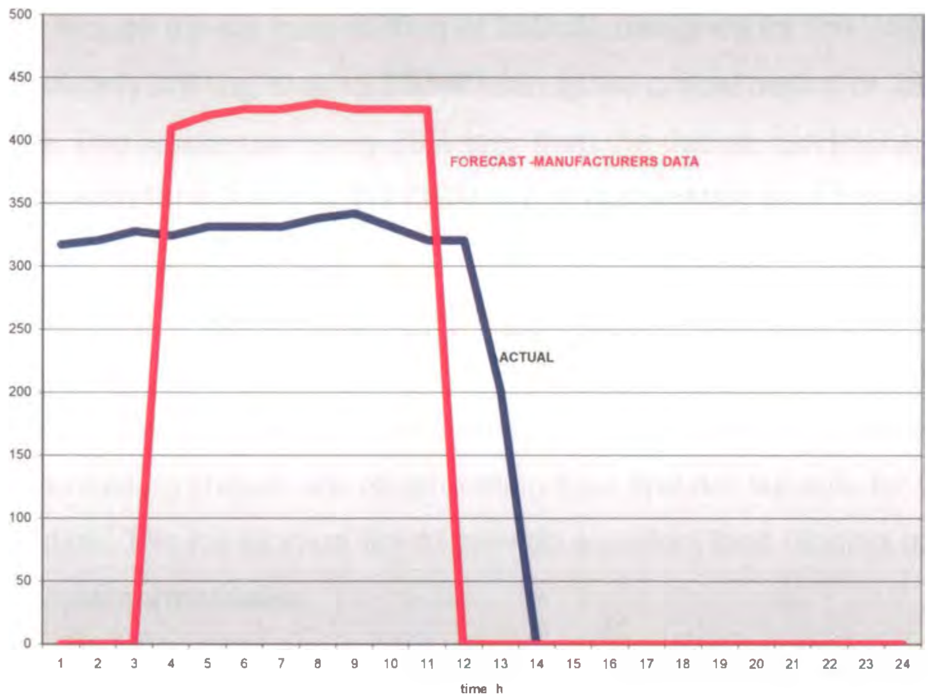
Tables 8.2 and 8.3 shows measured data for a combined ice bank charging and discharging cycle on a peak summer day, July 26/27th 2000. The charging commences at 18.00 hours and continues until 06.00 hours on the following day. The discharging cycle begins at 0700 hours with the fan coil unit circuit, discharging from the ice banks to provide the necessary additional cooling for the building from 07.00 hours until 17.00 hours, the normal office working hours. At 12.30 hours the soft opening of the DSM valve shows the increase ice discharge rate to provide cooling to compensate for the shut down of one main chiller.

The discharge rate for the fan coil circuit reached the maximum allowed capacity of 200kW or 100% valve opening position. On the other hand the maximum discharge recorded on the DSM circuit is around 250kW at 71.5% of valve opening position. The valve opening position almost represents the percentage of the design maximum hourly demand of the ice bank system. On the DSM circuit the ice storage system operates as a top-up cooling circuit, or the system operates under chiller priority. This allows the primary chiller to operates on full capacity thus eliminates the PSS (Chapter-1). Both circuits are sharing the discharge from the same storage system; this provides an added benefit or flexibility that each system can compensate for each other at critical peak load conditions.

The following items were noted:

1. The soft-opening 3-port valve start opening @ say 12.40 hours; this is illustrated by the 60kW energy utilized by the DSM circuit.
2. The FCU circuit is operating on maximum design capacity of 200kW
3. The soft-closing 3-port valve start closing @ say 16.40 hours; this is illustrated by the 20kW less energy utilization during the 4th DSM hour.

ice making -actual (26-27th July) vs forecast



kWh discharge
Comparison of forecast and actual (26-27th July 2000) (peak day)

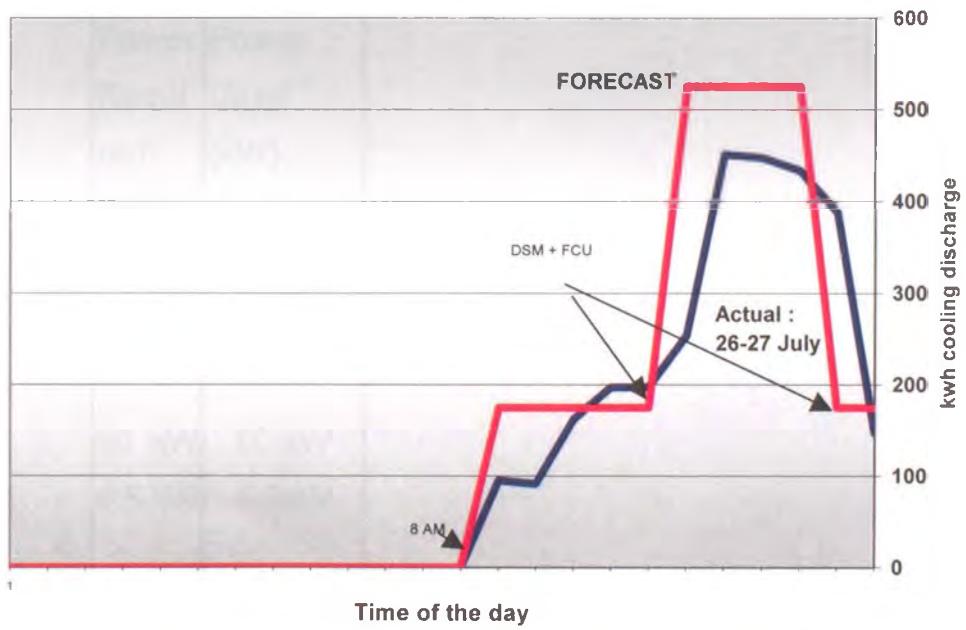


Fig 8.7 Actual System Performance in relation to design forecast:

Even though a peak load shifting of 350kW designed for the DSM circuit the system only shifting around 250kW during the critical period of 1300 –1700 hours. This is approximately 29% less than the design conditions.

The reason for this is that the DSM circuit is operating as a top-up circuit to the main water chiller on line. This is providing a “chiller priority” operation. The building load is satisfied with an average 250kW additional load from the DSM circuit. In similar cases an “ice priority” would be a better option. However, the ice priority option cannot be implemented in this project for DSM as the existing chillers are reciprocating type and not suitable for liner load reduction. The ice storage would provide excellent load topping up through the 3-port control valve.

8.5.8 Realization of Load Management

Due to the above design features the addition electrical load to the building transformer was reduced as shown in table 8.4.

Items	Power Required (kW)	Power Used (kW)
2x 350 $kW_{cooling}$ Chillers	300	0
Pumps	50 kW	50 kW
Fan Coil Units	6.5 kW	6.5kW
AHUs	5 kW	5 kW
Total	**401.5**	**61.5**

Table 8.4 Equipment Rating Analyses

The load management program electrically couples the glycol chillers with the existing water chillers as described above. A large electrical diversity factor for the chillers has been established as the glycol chillers only operate in the nighttime during the shut-down of one of the large water chillers (see table-8.4) . The total power requirement for the two glycol chillers is less than that of the connected load of a single large water chiller, it was not necessary to provide additional connected load for the glycol chillers. The overall additional electrical demand was limited to a mere 61.5 kW_{power}, thus, accommodated well within the available capacity of the existing sub-station.

8.6 Capital investment and payback analysis

This section further analyses the economics of the proposed system for the production plant and the TES system of CUP-1 of BSF. It is now clear from the recorded results that based on present electrical tariff structure in the Kingdom of Saudi Arabia, the installation of TES system would neither save energy nor reduce electrical charges. Hence, a payback analysis on the TES cannot be considered at this stage, nor is justifiable.
Hence, the payback period for any investment can be only analysed based on the following condition: "***Comparing air cooled against water cooled chiller application***"

To conduct "Payback Analysis" a "datum economics" has to be established. The recommended datum for payback period study would be the cost of "***Water-cooled Centrifugal system with TES***" system. This cost is established as SR 3.2 Million.

All other system cost comparisons whether screw type or centrifugal type is not useful for payback analysis, as they are not varying on energy utilization during operation.
However, the absorption technology may result in major payback, and analyzed separately.

8.6.1 Payback Analysis

A. Centrifugal / air-cooled Vs Water Cooled option

1. Datum System cost: SR 3.2 Million.(Centrifs/water-cooled & TES)

   SCECO Tariff: 26 Halals/kWh

   Fuel Cost : 10 Halalas/TR (or 33 Halalas / liter diesel)

   R/O Water Cost: SR 2 /m^3 (Municipality water considered)

2. **Assumptions:**

   Loading factor: 70% for TES and 85% for Chillers

   Capacity Utilization factor: 100% for TES and 80% for chillers

   Daily operation factor : 100% for TES and 90% for chillers

3. **Energy Consumption: (Air Cooled)**

   Chiller Operation: 200 x 1.25x 0.85 x0.8 x 0.90x 24 x 365

   = 1340 MWh

4. **Annual Energy Cost (Air Cooled)** : 1340MWh x 0.26 Halalas/kWh

   = SR 350,000/-

5. **Energy Consumption: (Water Cooled)**

   Chiller Operation: 200 x 0.75 x0.85 x 0.8 x 0.90x 24 x 365 = 805 MWh

Annual Energy Cost (water Cooled) 805x 0.26 Halalas/kWh : SR 210,000

Allow, for cooling tower, Cooling water pump operational cost : SR 50,000

= SR 260,000

6. Operational Cost savings

Annual gross savings on energy cost = SR 90,000/- (Approximately)

Additional Expenses on Water-cooled systems:
Allow 15% of the gross annual savings for chemicals, R/O water, and other consumable services.

Net savings on operating cost: SR 76,500/- per annum.

Common system operation & maintenance costs:
Certain common operation and Maintenance charges are not considered in the analysis, as they are almost equal on all applications

7. Payback Period

The cost difference between an air-cooled and water cooled system is around SR 600,000/-. If we can ignore discounted rates, and inflation the additional cost related to switching from air-cooled to water-cooled system will be payback within 7-8 years period.
A substantial portion of the initial capital cost will be recovered within the useful lifetime of the system, which is 25-30 years.

8. Advantages and Disadvantages

Moving from air cooled to water-cooled chillers would have produced the following benefits:

1. Smaller capacity Transformers
2. Low energy consumption
3. Lower refrigerant application

Also would result in the following disadvantages:

1. Large water usage & storage
2. Additional Piping system
3. Additional pumping system

4. Additional chemical usage
5. R/O Plant
6. Lower efficiency during high humid conditions

B. Absorption Vs Centrifugal Chiller Option

There is no air-cooled type absorption chiller option available, Hence the water cooled centrifugal chillers/TES system cost is used for payback comparison analysis.

The absorption chillers consume approximately 10% electrical energy compare to a centrifugal chiller.
The fuel option is compared with a co-gen. turbine operation analysis;
The system will consume approximately 0.33 litre of fuel per TR production.
Based on 33halals/litre fuel cost (diesel)

CENTRIFS/SCREWS	ABSORPTION WITH DIESEL	ITEMS
700 kW $_{cooling}$	700kW$_{cooling}$	Capacity
160	10	Power
16	20	Cooling Tower
14	20	Cooling Water Pump
190	50	Total power (kW)
160m3/Hr	2,00m3/hr	Condenser Water Flow
1.6m3/hr	2.0m3/hr	Make up Water
SR 50	SR 13	Power cost/Hr
	SR 20	Fuel cost/Hr
SR3	SR 4	Water cost/Hr
SR53	SR 37	Hourly Operating costs

5360	5360	Total Operating hrs with diversity Factors
SR 284,080	SR 198,320	Operating cost per year
SR 25,000 SR 5,000 SR 5000 SR 90,000	SR 30,0000 0 SR 5000 SR 90,000	Maintenance Cost per year Refrigerant Charges/year Chemical Charges/year Operation Man power charges
SR 130,000	SR 125,000	Total Annual Operating Cost

Table 8.1 Chillers – Performance and related items/Comparison

The above results clearly indicate that an absorption chiller application is the most economical solution as the annual savings on O&M will be around SR 91,000/-.

There is no additional investment over the centrifugal option, and the overall investment payback will be within the system lifetime.

8.6.2 Analysis

The life cycle analysis indicates the Absorption chiller system would be the best economical system, however, there is an existing Royal Decree that water-cooled chillers are not allowed in the central region of Saudi Arabia for air conditioning applications. This is due to the fact that the production cost of water is lot more expensive than of the production of electricity in the Kingdom of Saudi Arabia.

In addition the smaller air cooled centrifugal chiller or the absorption chillers are not capable of producing low temperature chilled water for ice making operation.

Hence, the installed system, *TES with air-cooled screw chillers* became the economical and efficient system for this project. The system was installed with reasonable capital investment and operational benefits. The chillers selected for this option has excellent track record and reliable after sale support within the Kingdom. The screw chillers are efficient for low temperature applications, and in addition, these chillers are normally used in medium size TES size projects.

CHAPTER-9

DISCUSSION, CONCLUSIONS, RECENT DEVELOPMENTS AND FUTURE RESEARCH

9.1 Aims And Achievement Of Aims

The aim of the research was to develop a cool storage system to manage electrical demand in large air-conditioning systems within Middle Eastern environment. A model cool storage project was developed for an existing building in Riyadh. A new retrofit design based on ice storage was used to reduce the dependence on the installed chillers, thus reducing connected power load (building user's requirement) and reduce peak demand (Utility requirement). The ice storage was used for the expansion of the building usage without adding further chillers, or transformers or supplementary onsite generation to comply with the electrical company, SCECO's DSM constraints and restrictions.

The aim also extended to identify the specific benefits of using cool storage in air-conditioning systems.

The research has delivered new knowledge in cool storage system design and application that would confirm and validate the use of cool storage system in the Middle East for load shifting peak shaving and demand side management.

The project influenced new/future projects and brought up new legislations related to HVAC system design, specially related production plants that uses chillers.

Climate of thinking have changed due to the research out comes in the form of production plant design. The novel path finding approach inspired several designers in the Gulf and changed their way of designing production plant. The system design was original, distinct from conventional and traditional design philosophies for building cooling systems used previously. In particular the ice storage and the injection circuit principle were departures from the traditional cooling system design methodology in the Gulf region, in particular in the Kingdom of Saudi Arabia.

In this retrofit application, the ice production plant and the building distribution network were separated and coupled through a heat exchanger. The approach gives the building user confidence in the system design as the existing water circuits are maintained without major changes.
In the absence of traditional favouring conditions , the installed system provided new potential benefits for cool storage application in the Gulf countries.
The compulsory use of air-cooled chillers and the variation in chiller efficiency due to changes in daily ambient temperature becomes the focal point for energy conservation. The cooler nighttime charging provided considerable reduction in chiller power consumption. The saving in electricity charges shown above resulted from this operational benefits.
Considerable savings in the massive connection charges imposed by SCECO compared with the utility rebate provided for Cool Storage application. At BSF the savings in connection charges was amount to US$ 70,000/-.

Thus, the expected design benefits of power management and enlargement of the cooling capacity without exceeding the limits imposed on peak power were realized.
The research application was for a specific building with particular climatic conditions and restriction on electrical load dictated by local legislations in relation to peak demand imposed by such climatic conditions.

9.2 Summary of Key Results and Outcomes

9.2.1 Cooling Load Demand on TES system

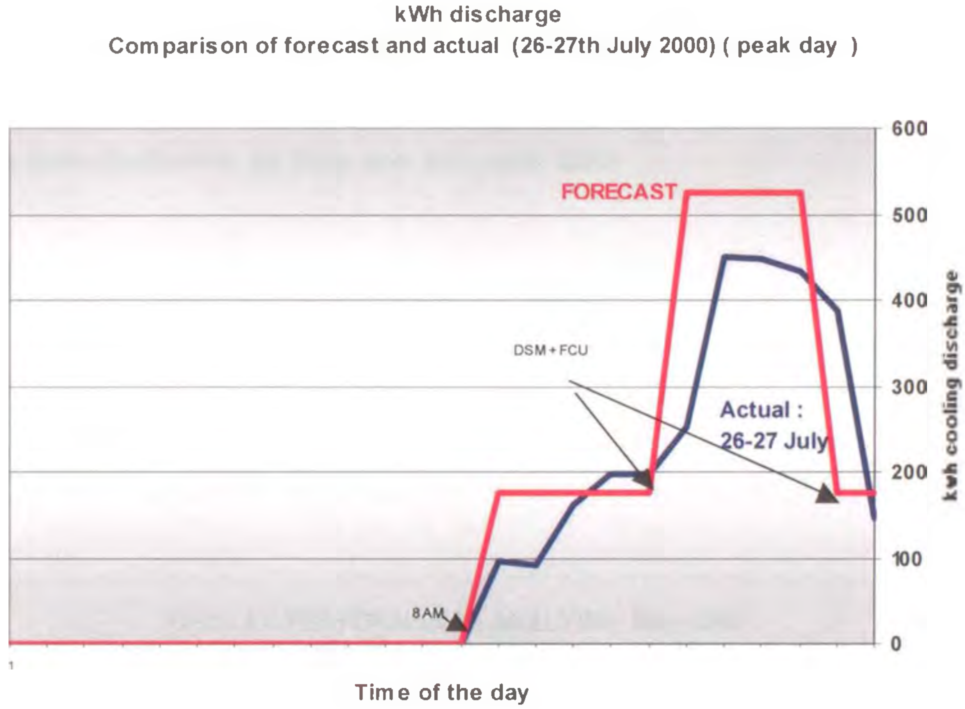


Fig. 9.1 Comparison of Designed and Actual Demand

The calculations and subsequent design was based on a peak TES load of 525kW$_c$. The reordered load profile illustrates the average peak load was 475kW$_c$. The reduction in ice discharge demand is mainly due to the elimination of PSS from the chiller operation.

This unexpected reduction in TES demand was fully utilized by reducing the run time of the second chiller, thus improving on the PSS deficiency of the system.

9.2.2 The chiller de-rating

Chiller de-rating during ice making was a major concern to the researcher, as the manufactures couldn't provide reliable de-rated data for the chillers operating under severe ambient conditions in the Gulf. Even though the chillers are not expected to operate during high ambient daytime, the de-rating data for ice making is paramount information in designing a single circuit charging and cooling chiller applications.

Between 1997-1999, the researcher conducted a series of data collection for chiller performance (at subzero)under various ambient and discharging conditions.

The results are tabulated below.

The chiller performance under normal operating conditions were obtained from the manufacturers as they are accurate data

CHILLER PERFORMANCE ANALYSIS May 1998																		
COOLING CAPACITY																		
AMBIENT	CHILLED WATER LEAVING TEMPERATURE C																	
C	-10	-8	-6	-4	-2	0	1	2	3	4	5	6	7	8	9	10	11	12
24	0.551	0.61	0.669	0.725	0.765						1.011	1.045	1.082	1.12	1.152	1.18	1.222	1.252
26	0.544	0.603	0.662	0.71	0.767						0.997	1.019	1.067	1.093	1.126	1.155	1.199	1.229
28	0.538	0.598	0.648	0.702	0.758						0.976	1.005	1.038	1.079	1.112	1.13	1.175	1.207
30	0.527	0.586	0.634	0.685	0.739						0.954	0.978	1.023	1.05	1.085	1.118	1.151	1.173
32	0.516	0.573	0.619	0.668	0.72						0.94	0.964	1	1.022	1.058	1.092	1.126	1.151
34	0.503	0.56	0.604	0.651	0.71						0.91	0.937	0.977	1.008	1.031	1.066	1.102	1.128
36		0.546	0.589	0.642	0.69						0.895	0.923	0.962	0.979	1.071	1.04	1.077	1.093
38		0.553	0.581	0.633	0.68						0.881	0.894	0.931	0.964	0.989	1.014	1.052	1.06
40			0.573	0.615	0.66						0.85	0.88	0.915	0.934	0.96	1.001	1.026	1.045
42				0.597	0.649						0.835	0.865	0.892	0.92	0.946	0.974	1.001	1.021
44					0.639						0.82	0.836	0.868	0.89	0.918	0.947		
46											0.805	0.807	0.844	0.88	0.89			

Table 9.1 Chiller De-rating Performances

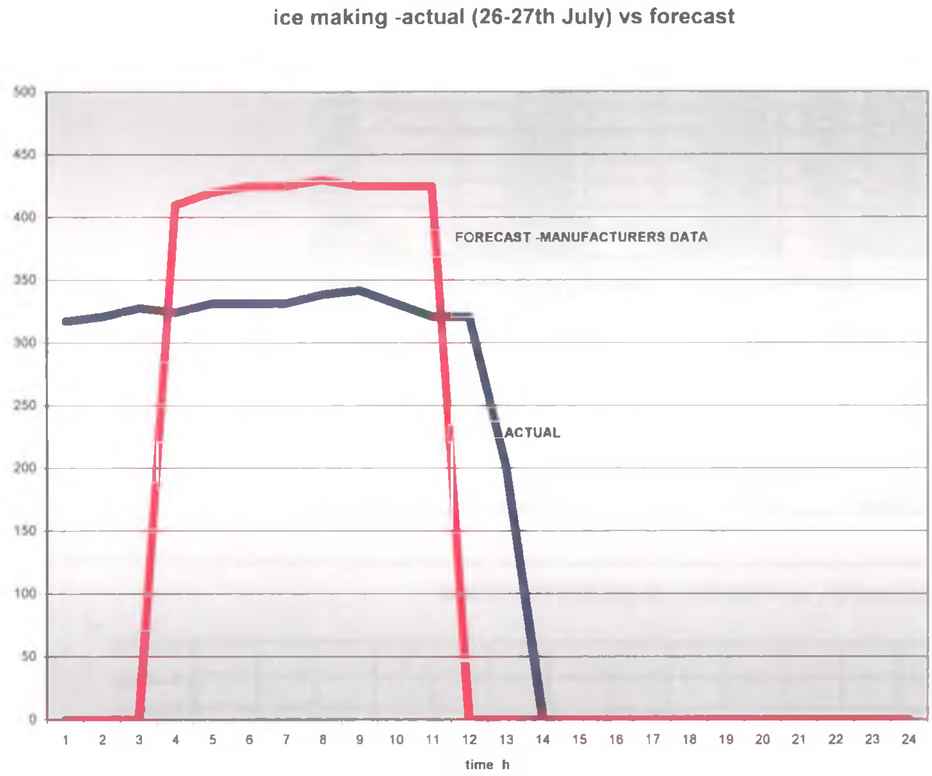


Figure 9.2 Comparison of predicted and actual chiller rating

9.2.3 Energy Conservation:

The following confidential data was provided by Saudi French Bank indicating the changes in electricity charges and clearly showing a net reduction in energy charges after the installation of the cool storage system. The savings were achieved due to the fact that the chillers were operated during cooler night times thus tremendously improving chiller efficiency. Hence, operating the chillers for TES in cooler night time is considered as a “favouring condition” to use TES in the Gulf region.

ELECTRICITY CHARGES STATISTICS FOR HEAD OFFICE

MONTH	1995	% save	1996	% save	1997
JAN	93,661		128,161	7%	119,622
FEB	97,111		138,561	14%	122,022
MAR	122,961		130,761	7%	121,822
APR	134,561		135,961	22%	111,622
MAY	126,161		121,961	6%	114,622
JUNE	128,361		161,361	3%	157,302
6 Months	702,816		816,766	9%	747,012
JUL	184,361	11%	165,962		
AUG	169,961	9%	156,562		
SEP	172,161	8%	158,962		
OCT	171,961	4%	165,962		
NOV	154,361	-1%	156,362		
DEC	148,961	10%	135,381		
6 Months	1,001,766	7%	939,191		-
12 Months	1,704,582		1,755,957		747,012

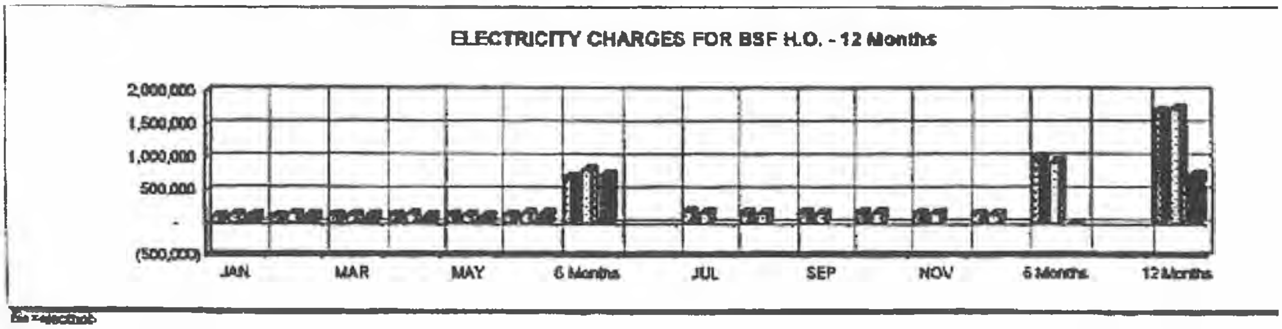


Figure 9.3 Comparison of Electricity charges between 1995 and 1996

9.3 Realization of Design Goals

A successful ice storage system has been designed and implemented within the system design constraint outlined above, providing additional cooling and DSM without exceeding the maximum capacity of the existing sub-station. The system has been operating continuously since 1996, and has delivered targeted benefits such as Demand management, additional cooling, and savings of electricity costs.

The project was presented to the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) Research Committee for the International Technology Award (ITA). The researcher and the project Director of studies, Mrs. Pauline Stephenson, jointly developed the case for submission. In addition to the above the researcher has participated in several technical forums such as Second Saudi Symposium on Energy Management, KACST-Energy Institute of Riyadh annual Conference presenting his arguments and expressing his views on the subject of demand management and cool storage applications in the Middle East The model project was visited by Prof. Andrew Self, Head of School, MAP, Kingston University.

The ice storage system was the first of its kind in the Kingdom of Saudi Arabia and has paved the way for a number of further applications in commercial developments. This innovative ice storage application has provided a considerable reduction in peak electrical demand and additional cooling for the Saudi French Bank headquarters building. This has been achieved without exceeding the maximum capacity of the existing on-site electrical sub-station. This pioneering project has changed the climate of opinion concerning ice storage for load management and encouraged others to develop similar projects. BSF has used TES at their Jeddah regional office and twelve further projects were undertaken and completed by the researcher for clients such as banks, museum, high-rise office buildings, shopping malls, and institutions. The installed TES system at the above projects successfully shifted in excess of 125 $MWh_{cooling}$ or 200 $MWh_{electrical}$ load during the critical peak demand period of 1p.m. – 4 p.m. The system continues in beneficial operation at the time of writing (May 2005)

This pilot project at the Saudi French Bank has influenced SCECO to review their tariff structure in the light of practical demonstration of load shifting provided by the TES system. Subsequent to the introduction of the ice storage system in the Saudi French Bank , new electrical tariffs, peak demand charges and night rates are expected to be introduced in the near future which would make the economics of the ice storage application even more attractive. The ice storage system at the Saudi French Bank is expected to lead to additional financial benefits due to this anticipated tariff changes.

SCECO has accepted TES system as their preferred mode of load shifting in large buildings during their peak demand period.

9.3.1 System Selection:

The intention of the system design was to achieve electrical load management during DSM period specified by the electricity supplier plus additional cooling as energy efficiently as possible. The DSM requirement was focused around shutting down of one of the primary chillers during the specified peak period. This reduction in the primary chiller operation was met by the ice storage system.

The decision to go for ice storage was driven by space restriction and using the existing equipment in the airside. (Also see Chapter 3.5) The ice storage allowed the researcher to maintain the operating temperature difference on the airside equipment by heat exchangers.
Selection of air-cooled chillers for ice making was necessary due to the restrictions imposed by local legislation aiming to save water.
Selecting the roof as the location for ice making chillers and ice banks may seem at first sight to be an unwise decision, considering the ambient temperature of 55°C during summer period. However, the selection caused by space restrictions in this retrofit project was justified by the good performance of the system within the intended design parameters. The location dictated the selection of a particular type of ice bank; the researcher had to select the tank with external insulation and lower kg/m^2 load bearing and warmer charging temperature. The above parameters dictated the selection of Calmac ice banks for this project.
The innovative glycol injection circuit obviated the needs for pumps on secondary circuits by using motorized valves, which provided a unique energy efficiency benefit. The soft opening and closing of valves may have an effect on instantaneous discharging of ice compare with pumps. Early opening and late closing of valves during and after DSM period compensated the time delay. However, the use of bypassing and diverting control valves avoided the need of additional power for numerous pumps in the injection circuits.

9.3.2 Chiller Coupling

The FCU circuit for additional cooling was designed to be totally independent of active chillier support to cool selected areas in the building during the daytime. The FCU circuit was totally depended on the static chiller or ice during the daytime, thus provided a full load shifting system.

In addition to the full load shifting the FCU also operates as a DSM circuit in the building during the prescribed period thus elevating the total DSM level.

Electrically coupling the new ice chillers with the existing water chillers is another novel idea implemented in this project. By using an “either or” switching programme the ice chillers were energized during the night without exceeding the existing power demand at the transformers.

The selection of ice storage resulted in far reaching benefits in future projects.

9.3.3 Using TES to realize reduction in Electrical Peak Load Requirement (DSM)

Cold Storage has delivered electrical load shifting benefits. After installing the TES, the primary chiller operation was reduced by 50% during the DSM period (from 1300 hrs until 1600 hrs). The reduction of primary chiller operation was only possible due to the TES system, which compensated for the offline chiller during this period. This application was a new knowledge in the Gulf and now used widely in major projects for DSM programme.[4]

9.3.4 Using TES to achieve additional cooling load in retrofits

The TES system became a valuable tool to handle the added cooling load at the BSF. The additional cooling for various part of the building was provided with TES, through a FCU network. The TES working in parallel to the primary chiller provided the needed additional cooling to various areas of the building. This was achieved by creating a stand-alone parallel FCU based hydronic circuit. As a result of the experience in BSF this approach is now used in complexes where high short time loads such as conference rooms, prayer areas, canteens etc. Application of TES in building now reduced the need for large chiller plants in complexes and facilities such as universities, hospitals, commercial complexes etc, where large short time cooling loads are expected.

9.4 Discussion on novel Operational Features

Chiller matching:

If we can define the electric chiller as 'active chiller' then we can identify TES as 'static chiller'. In the night during off peak period the active chiller (electric chiller) charges the static chiller (TES), thus allowing the static chiller to work in series or parallel as required with the active chiller during the peak load period (daytime) based on building cooling demand. The static chiller becomes the energy manager of the system whereas it can be used as either, lead, follow-up or top-up energy provider. The system operates similar to grid power and stored battery power in a prime and secondary power application in a computer room application with UPS facilities.

The term chiller matching in conventional design was related to selecting the chillers to handle the maximum building load at any operating conditions.

In the past this selection process was conducted without giving much consideration for energy management or energy conservation. This concept caused the major energy management problems defined under PSS.

The static chiller or the TES allowed the designer to match between the active and passive chillers within a very narrow margin of accuracy, that all instantaneous loads are met with the just needed energy supply from the production plant without exceeding the demand of the building. The TES become the most flexible load-matching tool in hydronic system.[18]

9.5 Innovations, Observations and Additional Benefits

The innovative outcomes of the project have subsequently shown wider applicability.

9.5.1 Operating Priorities:

For example, air-conditioning in office buildings is a year round requirement in the Gulf; even though the ambient conditions during spring, autumn and winter are substantially lower than of the summer months. The operation sequence of the system would be the typical example of this analogy. [17]

During the first year of operation, the chiller/TES sequencing applications used in the project were based on peak load operational conditions, i.e. summer load conditions.

However, the system is expected to operate throughout the year under various operating conditions, weekend and night load, spring, winter and autumn loads, winter load etc.

When the system operates on below design load it is important that the chiller operations are carefully selected so that partial load operation of a chiller is eliminated and system efficiency is maintained.

The following sequencing was selected to achieve the above, which was only possible due to the introduction of the TES in the network.

A load monitoring data has been developed by BSF over the years, and the data file was used to establish an innovative load management programme. A basic logarithm has been established and the daily data recorded by the PLC Controller is fed into the programme.

Circuilt-1 was used as a full load shifting circuit throughout the years. However, the design of Circuit-2 was based on chiller priority; i.e. the cool storage was used for top up load management during summer DSM period.

During Off DSM period (autumn, winter and spring), the building cooling circuit is turned around to ice priority. Cool storage is used as the primary cooling system whereas chillers are used as make-up cooling. The switchover allowed BSF to run air-conditioning system without chillers on line throughout winter and partially during spring and autumn.

The system originally designed for Chiller priority partial load operation; however, the system operation is now innovated to extend to the following three modes:

Chiller Priority	Partial Load	summer
Ice Priority	Partial Load	Autumn & spring
Ice Priority	Full load	Winter

This combined application of active and passive chillers with alternative priority modes are totally new and innovative application in the Gulf.

9.5.2 Elimination of Plant Starvation Syndrome Due to Cool Storage

At the model project, the chillers were allowed to run on full capacity during summer and the ice storage was used for “top-up” purposes. This was achieved by allowing one chiller (instead of two) to run on full capacity. In addition screw chillers were introduced in this project (first time in the Gulf), which have excellent linear load shedding characteristics. A combination of intelligent use of TES and chiller matching, and the usage of new type of chillers with liner load shedding characteristics, has totally eliminated the traditional PSS problem at this project.[2]
During winter the ice was used as the primary cooling system and the operation of chillers for cooling during daytime was fully eliminated. Hence at all operating conditions the chillers were allowed to operate at their rated full capacity, which provided a considerable energy benefit to the operator (eg data measured for kWh/Th).

9.5.3 Flexibility on Delta T:

As a result of operational experience, for *new projects*, the delta T (differential temperature) of supply and return fluid temperatures is increased from the conventional selection of 5.5 up to 9 deg. C. The increase in the delta T has resulted in reduction in mass flow . This provided substantial capital cost saving on major hardware of cooling systems, such as pumps, valves, pipes, transforms, switching gears, cables etc. The limiting factor on selection of delta T is the dew point; in Riyadh due to lower humidity level delta T can be stretched up to 8-9C. This is applicable on charging and discharging conditions. Thus providing TES in a project would provide the benefit of containing the overall cost in comparison with a conventional system. [46]

9.6 Future Work

9.6.1 Turbine Inlet Cooling:

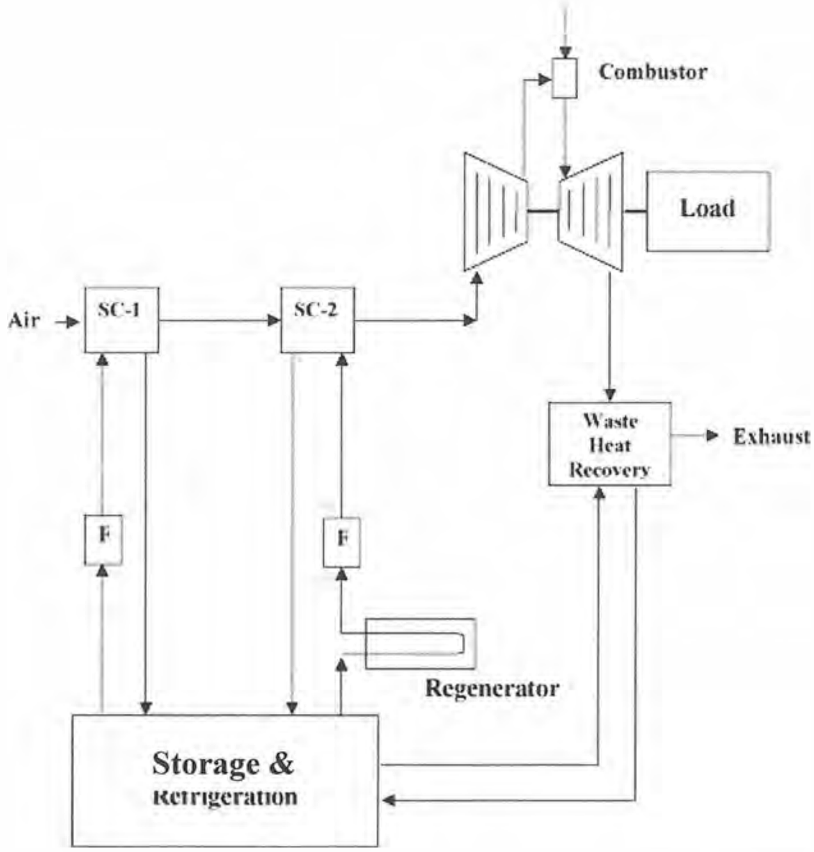


Fig. 9.4 Suggested Inlet Cooling System Diagram

In addition to load shifting and DSM, the researcher currently is working with SCECO to introduce the TES system for turbine inlet cooling in the Kingdom. As stated in previous chapters the load fluctuation between day and night at power-generating centres in the central region has reached an abnormal level. During summer it reaches a ratio up to 3:2 thus forcing SCECO to engage several turbines into spinning reserves during the night time.

A system is proposed that using these turbines during the night time to run large chillers to produce cooling, and store the energy in the form of chilled water, or ice and during the peak time use this stored energy to cool the turbine inlet air to increase the turbine efficiency.

Theoretical dada illustrates the turbine efficiency can be improved up to 25% if an inlet cooling system is installed to reduce the entering air temperature from 46°C to 12°C. [65]. The first test project in Riyadh installed in year 2000, achieved an average turbine efficiency improvement of 17.5% over a period of three years.

This subject would be an interesting subject for future research students in the filed of TES application.

9.6.2 CFC Conversion:

Montreal protocol called for the phase out of the refrigerants R-12 in 1995 and R-22 in 2000. This cases a major problem for large centrifugal chiller users mainly in the Middle East. The change of refrigerant de-rates the chiller performance by 20% or more. To compensate this loss the owner has to install additional chiller, which require additional power and all electrical ancillary services.

A cool storage in parallel with the existing chillers will solve this problem without installing additional chillers and ancillary electrical services.

The chillers will be used in the off-peak night time to charge the ice/chilled water storage system. During the peak time the ice storage will be used to top-up the required additional cooling load due to make up for the reduced chiller capacity. [52]

This would be a useful subject for research students and would bring upon new knowledge in load management field.

9.6.3 Monitoring and validating chiller Efficiency:

The full load operation of chiller due to cool storage application should be further researched. [33] The cool storage allows the chillers to operate on full load thus partial load application is eliminated due to undersized chiller in the network. This represents an analogy of urban and high way usage of a car. The chiller life will be extended hence provide financial benefits to the end users.

9.6.4 Data Base of chiller Operation:

At Saudi French bank, during autumn and winter months the chiller operation is now limited to nighttime charging only. The storage is adequate to provide cooling for the internal areas during any time without calling the chillers to be on line. A research can be conducted to establish database for a period of time and by using the database establishing an algorithm for chiller operation. The algorithm could be fed to the control program thus by monitoring the external temperatures for few days, they mode of operation can be adjusted automatically.

9.6.7 Real time Pricing & TES

Speed of TES system response against demand is another major area of interest for researchers; If a TES system can provide instantaneous response to demand call by the building cooling system, TES can play a major role in the novel 'Real Time Pricing' programmes for electrical demand and supply during critical periods. [61]

9.6.8 Co-Generation and Other Areas of Interest:

The research also brought interest in other DSM technologies. *Co-Generation*, Geothermal techniques, Solar Energy Utilisation, are some of the areas under studies and reviews among the designers and system developers.[59]

The profile below is of an actual processing plant designed and installed by the researcher in Riyadh operating in a daily summer profile taking electrical service at transmission level power (110kV+), Combining thermal storage with cogeneration.

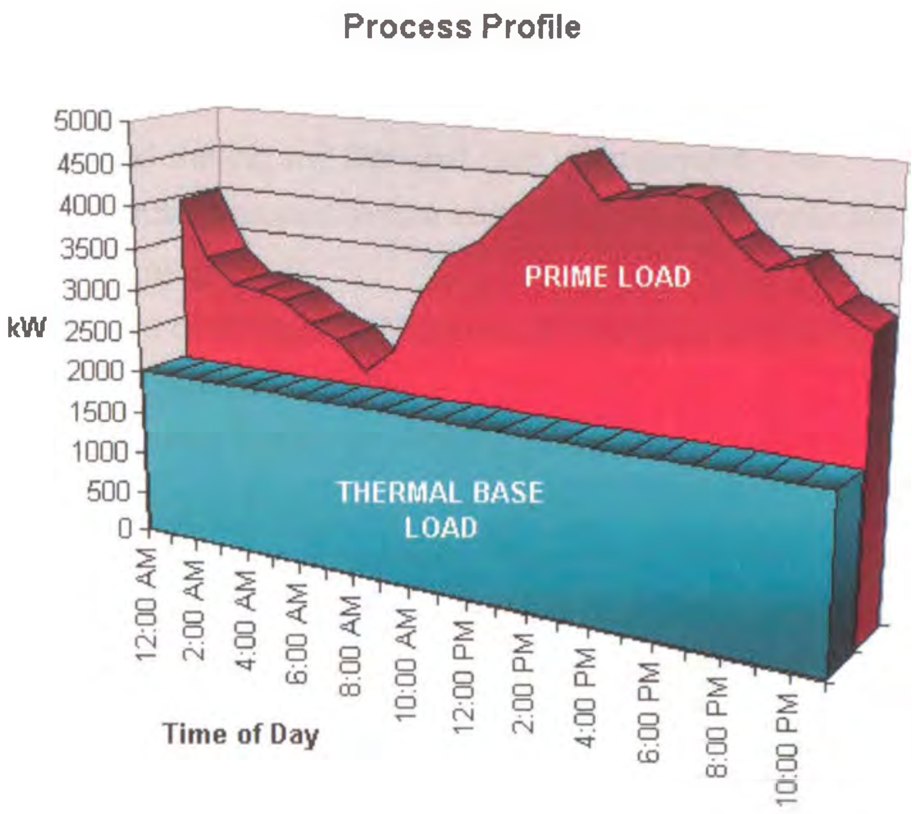


Fig. 9.5 Using TES as the base load

9.7 End Note

The research project has brought serious awareness of DSM and Load shifting in commercial building to the Kingdom of Saudi Arabia.

Traditional refrigeration systems are designed to satisfy the maximum peak cooling demand, which occurs only a few hours per year, and thus spend their operational life working at reduced capacity and low efficiency. Thermal Storage is suitable for any air-conditioning system or refrigeration plant, allowing installed chiller capacity (and size of other components) to be significantly reduced , typically between 50 and 60%. The TES system provides the shortfall of the energy when demand is higher than the chiller capacity. Thus chiller operation is continuous and its efficiency is at a maximum thus eliminating potential PSS in the system. Thermal Storage allows real management of the cooling energy according to the demand.

In 1996 after completion of the model project a total load of $500kW_{power}$ was shifted during peak hours.

However, by year 2004 a total of approximately 350,000 $KW_{cooling}$ [86] was shifted during peak hours through chilled water and ice based TES systems. The Central region of Saudi Arabia has seen an unprecedented growth in the TES application for load shifting and Demand Side Management. The researcher designed and developed almost all the projects that contributed to this major load-shifting program.

The pilot project brought confidence in TES system among project developers and mainly SCECO as useful load shifting and demand side management tool that would provide a balance benefit for both the power producer and end user.

After the successful completion and monitoring this system over five years, SCECO brought in a ***mandatory requirement*** that any building development with more than 2 MW_{power} connected load must be provided with a TES system for DSM and load shifting.

The application of Thermal Energy Storage in the Kingdom of Saudi Arabia is going to see a major growth in the future as the power generating companies increased the cost of power by 260% in 2002. In addition they are also considering introducing a lower nighttime tariff to encourage the property developers and building users to install TES system for full load shifting during peak hours. This would encourage the old buildings to install TES system as there will be defined pay back period of capital investment associated with any type of load shifting programme.

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