COMPUTER SIMULATION OF THE LONG-TERM PERFORMANCE OF A SUMMER AIR CONDITIONING SYSTEM WITH A MAN-MADE SEASONAL THERMAL ENERGY STORE

A MASTER'S THESIS

in

Mechanical Engineering
University of Gaziantep

By
Murtaza YILDIRIM

February 1991

Approval of the Graduate School of Natural and Applied Sciences.

Prof. Dr. Mazhar ÜNSAL
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science in Mechanical Engineering Department.

Assoc. Prof. Dr. Sedat BAYSEÇ
Chairman of the Department

I certify that I have read this thesis and that in my opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science in Mechanical Engineering Department.

Prof. Dr. Mazhar ÜNSAL
Supervisor

Examining Committee in Charge:

Prof. Dr. Ömer T. GÖKSEL ____________

Prof. Dr. Mazhar ÜNSAL ____________

Assist. Prof. Dr. Refik KAYALI ____________

Assist. Prof. Dr. Şükrü KARATAŞ ____________

Assist. Prof. Dr. A. Rıza TEKİN ____________

ABSTRACT

COMPUTER SIMULATION OF THE LONG-TERM PERFORMANCE OF A SUMMER AIR CONDITIONING SYSTEM WITH A MAN-MADE SEASONAL THERMAL ENERGY STORE

YILDIRIM, Murtaza

M. S. In Mechanical Engineering

Supervisor: Prof. Dr. Mazhar ÜNSAL

February 1991, 97 Pages

Modelling and computer simulation of the long-term performance of a summer air conditioning system with a man-made seasonal thermal energy store is the subject of this investigation. The performance of two different models as hot and cold time active systems were investigated for daily and seasonal storage. In both cases, winter air is the natural source for cooling. For hot time active system, the cold storage tank is cooled by low temperature atmospheric air via an air/water heat exchanger and the air conditioning system condenser is cooled fully or partially by the water in the cold storage tank during hot time when the air conditioning system is operational. In cold time active system, the air conditioning system is operational at cold time. The condenser is cooled by cold time atmospheric air. Water is cooled and stored during cold time.The water is then circulated between the tank and fan-coils. The purpose of this study is the estimation of the effect of systems components on the coefficient of performance (COP) by an interactive computer program that is prepared in the Fortran 77 language. Results of these two models are compared with the results of a conventional system.

Key words: thermal storage, energy storage, thermal energy store, cold storage, seasonal thermal energy store.

ÖZET

SEZONLUK YAPAY ISIL ENERJİ DEPOLAMALI YAZ KLİMA SİSTEMİNİN UZUN DÖNEM PERFORMANSI BİLGİSAYAR SİMULASYONU

YILDIRIM, Murtaza

Yüksek Lisans Tezi, Makina Mühendisligi Bölümü

Tez Yöneticisi: Prof. Dr. Mazhar ÜNSAL

Şubat 1991, 97 Sayfa

Bu çalışmada, sezonluk yapay ısıl enerji depolamalı yaz klima sisteminin modellenmesi ve bilgisayar simulasyonu yapılarak sistemin uzun dönem performansı incelenmiştir. Sıcak veya soguk zamanlarda sogutma unitesinin çalıştıgı iki ayrı sistem modellenmistir. Modellerin performansları günlük ve sezonluk depolama için incelenmiştir. Her iki modelde de, soguk hava dogal soguk enerji kaynagı olarak kullanılmaktadır. Sıcak zamanlarda çalışan modelde, soguk hava ile depodaki su sogutulmuş ve bu su klimanın çalıştıgı zamanlarda klimanın kondanserinin tamamen veya kısmen sogutulmasında kullanılmıstır. Diger modelde ise sogutma unitesi soguk havada çalıstırılarak depodaki su sogutulmuştur. Böylece kondanser dogrudan soguk hava ile sogutulmuş oldu. Klimatize edilen ortam ise, gerektiginde depodaki soguk su depo ile fan-koiller arasında dolaştırılarak sogutuldu. Bu çalışmada, Fortran 77 bilgisayar dilinde bir program hazırlandı, ve bu program kullanılarak, her iki modelin gerek birbirleri ile gerekse klasik sistemle karşılaştırılmaları Isıl Verimlilik Katsayıları (COP) yardımı ile yapıldı. Ayrıca, her iki sistemin elemanlarının COP'lari üzerine etkileri incelendi.

Anahtar kelimeler: ısıl depolama, enerji depolama, ısıl enerji depolama soguk enerji depolama, sezonluk ısıl enerji depolama.

ACKNOWLEDGEMENTS

I would like to express my gratitude to Prof. Dr. Mazhar Ünsal for his inspiring and illuminating supervision and continual encouragement throughout the progress of this thesis.

This thesis was produced on Apple Macintosh and printed on an Apple Imagewriter. Software applications were Microsoft Word for processing, Mac Draw for rendering the line diagrams and Microsoft Chart for plotting the graphs. I am also grateful to Prof. Dr. Mazhar Ünsal for supplying this possibility.

I wish to express my warmest gratitude to the assistants of Mechanical Engineering Department. My very special thanks are due to my family for their moral support and continuous help.

I would like to tank Assoc. Prof Dr. Celal Koraşlı and persons of IBM Computer Center and Natural and Applied Science School for their helps.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

A	Overall heat transfer area (m^2),
A_{st}	Total surrounding area of the storage tank (m^2),
a, b, c	Coefficient of second order polynomial equation,
a_{st}, b_{st}, c_{st}	Dimensions of the storage tank (m, m, m),
C_{max}	Maximum energy capacity (kw/C),
C_{min}, $mC_p)_{min}$	Minimum energy capacity (kW/C),
C_p	Specific heat capacity (kJ/kgC),
C_{pr}	Specific heat capacity of air (kJ/kgC),
C_{pw}, C_w	Specific heat capacity of water (kJ/kgC),
COP	Coefficient of performance, defined on page
dT_r	Differential change in room temperature (C),
dt	Differential change in time (hr),
dT_w	Differential change in water temperature (C),
h_a	Convective heat transfer coefficient of air (kJ/m^2C),
h_w	Convective heat transfer coefficient of water (kJ/m^2C),
k_{ins}	Conductive heat transfer coefficient of the insulation material (kJ/m^2C),
k_{st}	Conductive heat transfer coefficient of storage tank material (kJ/m^2C),
m	Mass flowrate of fluid (kg/sec),
m_a	Mass flowrate of air (kg/sec),
m_r	Mass flowrate of air in the room (kg/sec),
$mCp)_r$	mCp value of the room (kJ/C),

N_c	Dimensionless parameter, defined on page 22,
N_e	Dimensionless parameter, defined on page 22,
N_{fc}	Dimensionless parameter, defined on page 22,
N_{he}	Dimensionless parameter, defined on page 26,
P	Dimensionless parameter, defined on page 26,
R	Dimensionless parameter, defined on page 23,
$R_{c,fc}$	Dimensionless parameter, defined on page 22,
$R_{e,fc}$	Dimensionless parameter, defined on page 22,
$R_{fc,s}$	Dimensionless parameter, defined on page 26,
$R_{he,fc}$	Dimensionless parameter, defined on page 26,
$R_{1,2,3...}$	Thermal resistance of the material 1,2,3...
Q_c, Q_h	Condenser energy (kW),
Q_e, Q_l	Evaporator energy (kW),
Q_{fc}	Fan-Coil energy (kW),
Q_{he}	Heat exchanger energy (kW),
Q_{ld}	Cooling load of the air conditioned space (kW),
Q_r	Internal energy of the room (kW),
Q_s	Defined on page 25 and 31,
q	Dimensionless energy defined on page 21,
q_c	Dimensionless condenser energy,
q_e	Dimensionless evaporator energy,
q_{fc}	Dimensionless fan-coil energy,
q_{he}	Dimensionless heat exchanger energy,
q_{ld}	Dimensionless cooling load,
q_{ls}	Dimensionless energy lost from the tank,
q_s	Defined on page 26 and 33

q_{st}	Dimensionless energy change of water,
S	Dimensionless parameter, defined on page 22,
t	Time (hr),
T_a	Ambient air temperature (C),
T_c, T_h	Condenser temperature (C),
T_e, T_l	Evaporator temperature (C),
T_r	Room temperature (C),
T_s	Surrounding temperature of the tank (C),
T_w	Water temperature (C),
T_{wi}	Inlet water temperature of the fan-coil (C),
T_{wo}	Outlet water temperature from the fan-coil (C),
U	Overall heat transfer coefficient (kJ/m2C),
$UA)_c$	UA value of the condenser (kW/C),
$UA)_e$	UA value of the evaporator (kW/C),
$UA)_{fc}$	UA value of the fan-coil (kW/C),
$UA)_{he}$	UA value of the heat exchanger (kW/C),
V_r	Volume flowrate of the air for the room (m^3/sec),
V_w	Volume of the storage tank (m^3),
w	Dimensionless compressor work defined on page
W	Compressor work (kW),
X	Defined parameter on page 21,
$d\phi_r$	Dimensionless, deferential room departure change,
$d\phi_w$	Dimensionless, deferential water temperature change,
$d\tau$	Dimensionless, deferential time change,

β, γ	Dimensionless parameter of COP,
δ_1	Dimensionless room temperature control parameter,
δ_2	Dimensionless water temperature control parameter,
Δt	Time increment (hr),
ΔT_r	Room temperature difference (C),
ΔT_w	Water temperature difference (C),
ΔX_{ins}	Insulation material thickness of the storage tank(m),
$\Delta\phi_r$	Dimensionless room temperature difference,
$\Delta\phi_w$	Dimensionless water temperature difference,
ε	Effectiveness, defined on page
$\varepsilon C_{min})_c$	Effectiveness times minimum energy capacity of the condenser (kW/C),
$\varepsilon C_{min})_e$	Effectiveness times minimum energy capacity of the evaporator (kW/C),
$\varepsilon C_{min})_{fc}$	Effectiveness times minimum energy capacity of the fan-coil (kW/C),
$\varepsilon C_{min})_{he}$	Effectiveness times minimum energy capacity of the heat exchanger (kW/C),
ϕ	Dimensionless temperature defined on page 21,
ϕ_a	Dimensionless ambient temperature,
ϕ_c	Dimensionless condenser temperature,
ϕ_e	Dimensionless evaporator temperature,
ϕ_r	Dimensionless room temperature,
ϕ_w	Dimensionless water temperature,

ϕ_s Dimensionless surrounding temperature,

ϕ_{wi} Dimensionless inlet water temperature to the fan-coil,

ϕ_{wo} Dimensionless outlet water temperature from the fan-coil,

ρ_r Density of air (kg/m^3),

ρ_w Density of water (kg/m^3),

τ Dimensionless time

CHAPTER 1

INTRODUCTION

Human beings have struggled to make their lives more comfortable, through control of the immediate environment. Although there is an evidence of the use of evaporative effects and ice for cooling in very early times, it was not until the middle of the ninetieth century that a practical refrigerating machine was built. By the end of the ninetieth century the concept of central heating was fairly well developed and early in the twentieth century, cooling for comfort got its start. Developments since that time have been rapid.

Summer air conditioning applications increased in the last few years in our country. A comfortable environment is now a necessity during the summer (hot days) season. Maintenance of a cool environment in the warm months is important, not just for the comfort of the space occupants, but also for the utilization of human resources and for productivity. The need for cooling is often extended beyond the summer months with the ever increasing use of lights, machines, and equipments in buildings and heat transfer to spaces. It is not uncommon to find nonresidential buildings requiring cooling for more than six months of a year even in the cooler regions of the world [2].

A comfortable environment is now considered as necessity and many modern processes and products need precise control of environmental conditions. Almost all homes, offices, and industrial spaces must be designed with a means of controlling the indoor environment throughout the year.

High electrical energy demand by a summer air conditioning systems creates a field to study for decreasing this high electrical energy consumption by using cold storage. All these considerations have placed more emphasis on the design and the simulation of thermal environmental systems.

Cooling for human comfort, product development and food preservation etc. have been realized by mechanical or absorption refrigeration systems. Such cooling can also be accomplished through the long-term (seasonal) storage of winter coolness. For a given annual cooling load ice production and storage requires the least volume, due to high latent heat of ice. Cold energy storage medium selection will be dealed within the another chapter.

Natural production of ice, cold water or snow in winter months and their storage for summer usage are very old concepts. In the old region of the world, ice was removed from the lakes and the rivers and was kept in insulated storage for summer usage. In the arid region of Iran, ice was produced gradually in shallow ponds during clear winter night time [2]. The ice blocks were then removed and stored in a deep underground storage. Snow was kept for the summer usage in natural pits during old times in Gaziantep by insulating its top surface. Snow was also stored in Kahramanmaraş for making ice-cream.

Seasonal storage of water chilled naturally during winter for summer has recently received and renewed interest due to increases in energy costs and need for air conditioning. This interest has motivated technical studies on the feasibility of chilled water seasonal storage.

Most of the storage systems beings considered today are designed for saving electricity. Several other concepts of seasonal cold storage have been proposed. Heat pipe ice maker, the ice box, aquifer storage, chilled water storage, underground storage with insulation and without insulation are examples. All of these storage schemes are in the development stage and their costs and performance are still uncertain. Furthermore, these approaches are not equally suitable for all applications [1].

The purpose of this thesis is the thermal analysis of the long-term performance of a summer air conditioning system with a man-made seasonal thermal energy store (TES) by using the potential between hot and cold time temperature difference. Two different models considered are illustrated in Figure 1.1.

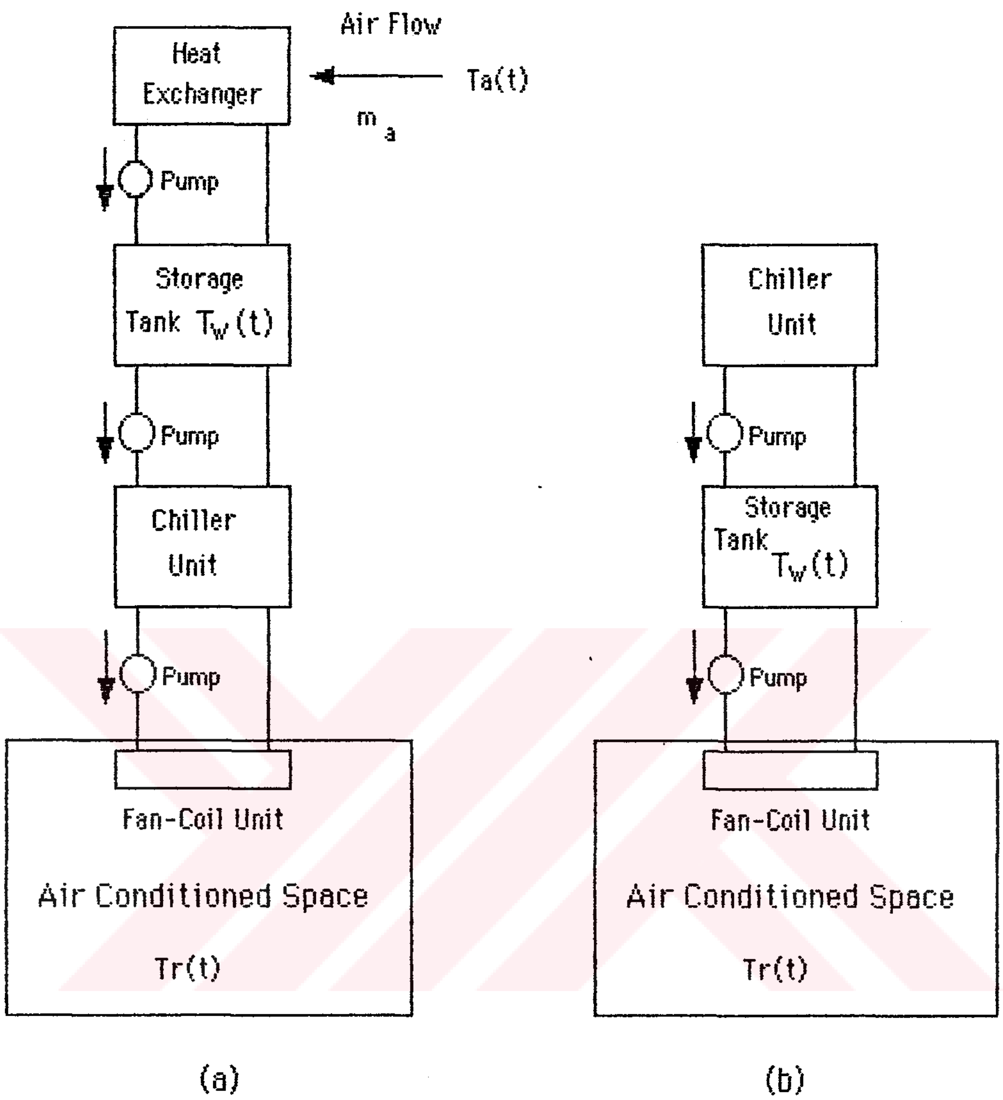


Figure 1.1 Two Different Arrangements for an Air Conditioning System with A Thermal Energy Storage Tank : a) Hot Time Active b) Cold Time Active

Figure 1.1 a and b show us the model systems as hot time active and cold time active respectively. Winter air is the natural sources for storing cold energy for the hot time active model. The heat exchanger shown in Figure 1.1a cools water which is circulated with a pump when water temperature is greater than the outside air temperature. Water in the storage tank is cooled by winter air and stored for use during the summer by air conditioning system.

In the second model shown in Figure 1.1b winter air is directly used for cooling the condenser of the chiller unit. Chiller unit is not functional during hot times. Water is cooled by the chiller unit when outside temperature is low. Then, chilled water is circulated between fan-coils and the storage tank when air conditioning is necessary.

Mathematical formulation of the two models are realized in Chapter 3 by using the effectiveness-NTU method and an energy balance equation. The equations are then converted into dimensionless forms. The ordinary differential equations are solved by using finite difference and Newton methods.

The program developed is used to compute the temperatures of system components, energy charged and discharged to the each component of the system (evaporator, condenser, heat exchanger, fan-coil, storage tank and air conditioned space).

Finally, the results of all calculations for temperature and energy distribution and Coefficient of Performance (COP) are given in graphical form. Long-term performances of a summer air conditioning systems with a man-made seasonal thermal energy store are discussed for the two models.

CHAPTER 2

LITERATURE SURVEY

D. L. Kirkpatrick, M. Masoera, A. Rable, C. E. Roedder, R. H. Socolow and T. B. Taylor [1] studied seasonal storage of ice using an ice-pond. The ice pond is a scheme of making ice in winter and storing it for use in summer. A large excavated reservoir is filled with a snow/ice mixture during winter by a snow machine of the type used in ski resorts. At the end of cold season the ice is covered with insulation. When cooling is needed, chilled water is pumped from the pond and after being warmed by the load, the water is returned to the pond.

With a stored cooling energy of 80000 kWh_{chill} the COP of ice making was 10.1. During the winter of 1980/81 the seasonal average COP of ice making was only 5.9. The main difference between the two years was the presence of the cover which inhibited the flow of fresh cold air to the snow machine.

M. N. Bahadori [2] studied natural production, storage and utilization of ice in deep ponds for summer air conditioning. A covered ice pond was analyzed for the gradual production, storage and utilization of ice for air conditioning of a building with an annual cooling load demand of 1.2 TJ. Winter ambient air is blown into the pond for ice production, provided the major source of coolness. The total heat gain by the ice pond was estimated by a two-dimensional heat conduction analysis, employing a finite difference technique.

In summer, or whenever cooling is needed, the water coming from the air conditioning unit is circulated through the buried heat exchanger pipes, melting the ice block from the bottom. The system considered in their study fails to operate in area where the winter ambient air temperature does not drop below $0^{o}C$.

H. Akbari and A. Mertol [3] investigated the current and potential use of the thermal energy storage systems for cooling commercial buildings. A general overview of the technology is presented and the applicability and the cost-effectiveness of this technology for both developed and undeveloped countries are discussed.

In this article, the advantages and disadvantages of storage medias, such as; chilled water, ice and phase change material and also advantages and disadvantages of the operational strategy as full storage partial storage and demand limited storage are discussed.

The cool storage is increasingly being used in developed countries because of the current utilities rate structures and the additional incentives offered by the utility companies to accelerate the penetration of this technology. The number of cool storage installations in the U.S. has been doubling each year. The penetration of the technology has been higher in regions where a significant day and night time differential in the price of electricity exists. Most thermal storage installations are chilled water and ice storage systems.

E. L. Morofsky [4] studied an innovative seasonal ice storage technology development. Through preliminary design, experiment and a pilot study to detailed design of a demonstration project. Experiments and field trials have developed a very efficient, fully automated ice formation, storage and utilization system. The ice block is formed by freezing a large number of thin water layers successively by introducing cold outside air onto the water surface. Control is based on freezing degree minutes accumulated by microprocessor. Major assumptions are that the designs are reliable enough to replace part or all of the conventional chilling capacity and that no additional maintenance or operational expenses are required.

A. E. McGarity, D. L. Kirkpatrik and L. K. Norford [5] studied large-scale seasonal ice storage cooling system using an ice pond has been designed and built, and operated at a commercial office building complex in Princeton, NJ. Ice is produced as artificial snow during winter using ski-slop snow machines and naturally cold outside air. During the cooling season, cold melt water is circulated through the cooling system of a three office building. The annual coefficient of performance is 10, based on total cooling energy delivered and total electricity consumed by snow machines in the winter and by circulating pumps in the summer. By adding the electricity consumed during the ice-making season for space heating, lighting and instruments to the direct consumption, the annual overall COP was about 8.

C. E. Francis and R. T. Tamblyn [6] studied an annual cycle ice production and storage system. The storage mass, located underground, will support a parking lot. A frozen Earth Annual Storage Ice System has been developed and tested at an Illinois University. Annual storage ice systems involve the use of cold winter air for the purpose of producing a large amount of ice to be stored for use during the following summer. This system has been been under development for three years and has gone through nearly two years of testing. First year testing indicates that the system operates reasonably close to projections, while optimization will increase the COP substantially.

Based upon this work, this system will provide seasonal savings of $51841 and a simple payback of 3.5 years. It was assumed that land adjacent to the building to be cooled would be available for the storage mass and that a parking lot could be located on top; therefore, no charge was made for the land nor do any of the calculated costs include parking lot expenses.

C. W. Sohn and J. J. Tomlinson [7] studied duirnal ice storage cooling systems in Army Facilities. The U.S. Army's experience with duirnal ice storage (DIS) cooling system is discussed. A few favorable characteristics of an Army post for the application of storage cooling systems are identified. DIS system designs must be based on reducing the entire post peak demand as measured by the master meter rather than the peak demand of an individual building, as in the case of a non-military commercial or residential building. The building has a controlled occupancy schedule; 9:30 a.m. to 7:30 p.m. Monday through Saturday and 11:00 a.m. to 5:00 p.m. on Sunday.

The water remains in the tanks at all times and is frozen and melted by a glycol brine that flows through the spiral heat exchanger. Data were collected at 10 minute intervals. An Army post is an excellent candidate for storage cooling systems because of its sharp peak load characteristic. A 900 ton-hr ice-in-tank DIS cooling system was successfully installed at a post exchange building at Ft. Stewart, GA, to prove the effectiveness of the DIS cooling system in reducing the peak electrical demand by an Army post.

N. Tran, J. F. Kreider, and P. Brothers [8] investigated the performance of several chilled water storage systems in commercial buildings. Their study includes eight storage systems of four different designs located in the U.S. and Canada. Storage effectiveness was evaluated for each charging and discharging cycle and for several cycle per site. In this test program, four types of anti-blending methods were studied;

1. Natural stratification-relies on density difference to reduce blending.
2. Diaphram-relies on a flexible barrier between the chilled water beneath and the warmer water above to minimize blending.
3. Empty tank- blending is reduced by never allowing warm water and chilled water to coexist in the same tank.
4. Labyrith-blending is reduced by causing a long and circuitous flow path to exist between the cold and warm parts of the storage system.

The majority (six of eight) of the chilled water storage systems tested were not operated according to the designer's intent. Reliance on a manual fine-tuning by system operators was widespread, resulting in less-than-optimal performance. This is felt to be an important finding of this test program.

R. K. Tackett [9] studied a HVAC system of a large office building, which has integrated ice storage, heat recovery and cold air distribution. After several design iterations, the team produced an HVAC scheme that incorporated:

Partial Ice Storage, Glycol chillers operate at night to make ice, which is then used the next day to provide a portion of the cooling. Shifting to nighttime chiller operation eliminates about 800 kW from the daytime electric demand and saves an estimated $55000 each year in demand charges. Overall the ice system added a net ($0.25 per m^2) to the project's cost. However, it will save more than $0.19 per m^2 each year resulting in a simple payback of 11/3 years.

D. H. Spethmann [10] studied optimal control in cool storage. Many electrical utilities are considered for promoting the use of cool storage, to reduce peak demand and to increase the use of off-peak electricity. Building owners installing cool storage can reduce utility bills by shifting electricity consumption away from expensive peak periods and into off-peak periods when electricity costs and demand charges are lower. Data collected by EPRI indicate that many systems are operated at less than obtainable efficiency because of the lack of proper controls. This led to the development of an optimal controller for cool storage systems.

G. Meckler [11] studied cold air distribution option with ice storage. Cold air/water HVAC systems (4.4^{0}C) have been developed for use with ice thermal storage because (1) they require one-third to one-half as much primary air when compared with (4.4^{0}C) all-air HVAC system, (2) they control humidity and temperature separately and they can be applied in a much broader range of facilities without overdrying the air and (3) they are lower in both first cost and energy cost than all-air systems thus making ice storage equipment a more practical option.

4.4^{0}C air/water VAV systems compared with the following HVAC system, in terms of electric demand, energy usage, energy cost and first cost:
4.4^{0}C air/water VAV system with partial ice storage
4.4^{0}C air/water VAV system with full ice storage
4.4^{0}C all-air VAV system with partial ice storage
4.4^{0}C all-air VAV system with full ice storage

Conclusions include the following: (1) where there is only a small spread between the utility's day and night energy usage rates, a partial ice storage air/water system can produce a lower annual energy cost than a the most cost-effective of the five HVAC system compared, in each of four utility rate scenarios.

B. Cordaillat and R. T. Tamblyn [12] describes a new office tower with thermal storage in Paris. It has the first commercial thermal storage in Europe, and this facility was installed with savings in both initial and operating costs.

D. L. Grumman and A. S. Butkus [13] studied ice storage application to an Illinois hospital. An 8450 kWh ice storage system was designed and installed in a northern Illinois hospital as the best and the most cost effective means of alleviating an existing cooling tower noise problem over which litigation was threatened and an Illinois EPE complaint had been lodged. Selection of the ice storage option followed an intensive examination of numerous alternative methods to alleviate the problem. The system has been in operation since, not only solving the noise problem but saving the hospital increasing amounts every year as electric rates rise.

D. E. Knebel [14] studied the economics of harvesting thermal storage system for a 240000 sq. ft. merchandise distribution center located in Columbia. The reduction in electrical demand and economic benefits are demonstrated.

Harvesting ice generators separate the function of making ice and storing ice. The ice generator will operate as a chiller when water warmer than 32 F is supplied to the plates. The chilled water is pumped from the storage tank to the load and returned to the ice generator. A low head recirculation pump is used to provide optimum flow over the heat exchanger as required.

Ice thermal storage systems are an effective tool in a utility's load management plan. By reducing installed refrigeration capacity, or ensuring that equipment does not run on peak.

J. M. Ayres and H. Lau [15] studied the economics of ice storage systems in a university engineering building. A conventional chilled water plant with electric chillers and cooling tower was compared to an ice storage system with submerged coil ice makers, screw compressors and evaporative condenser under two operating modes. The analysis demonstrated that the ice storage system with two compressors designed for load leveling had the lowest life-cycle-cost worth. By investing $314540 in the ice storage system, the on-peak demand was reduced by 252 kW and the annual electric cost by $16426. With the 20 years life of the equipment, the university will save $221000.

E. I. Mackie [16] studied a thermal storage with a large volume chilled water storage incorporated into the systems serving Rabson Square and Courthouse in Vancouver. The chilled water storage facility is a 3300 m^3 atmospheric pressure concrete tank which is used to reduced electrical demand and to allow reduction of installed mechanical cooling capacity. The tank design was based on a stratification principle developed in model testing in a hydraulics laboratory. the tank is in operation and field results indicate an installed cooling capacity which is more than adequate to handle the peak loading.

L. K. Rawlings [17] studied ice storage system optimization and control strategies. Ice storage systems reduce the electrical demand charges resulting from air-conditioning. In a glycol-type ice storage system, the sizing of the chiller and the ice tanks can be optimized algebraically or by computer simulation programs. Low temperature air systems can lower the cost of the primary fans and duckwork in the cooling system. Brine type ice storage systems have the potential to provide mechanical cooling for buildings with installed costs similar to those of conventional systems, with lower operating costs due to these systems demand reducing capabilities.

M. W. Wildin and C. R. Truman [18] investigated the alternate methods of water tank stratification in order to gain an understanding of the factors that influence the thermal storage performance. Due to growing interest in thermal storage for electrical load management, stratified chilled water tanks are being more widely used for storage of cooling capacity.

Rather than operate a water chiller to meet the cooling load as its occurs, the chiller is operated either partially or solely during off-peak hours, and the chilled water it produces is stored in tanks adjacent to or under the building to be cooled. This water is used during on-peak hours, either to supplement or to replace the chiller output.

The results indicate that chilled water tanks consistently stratify well for a range of operating conditions, including normal design conditions, even when an operating strategy that is obviously less than is employed. Investment in thermal storage is always made in anticipation of realizing lower electric utility costs.

W. P. McNeil and J. D. Mathey [19] analyzed the performance of an ice storage system. This particular installation is a full storage system 100% of the daily cooling requirements between 9:00 a.m. and 5:00 p.m. are stored in the ice builders. An alternative cooling system analyzed for the building for a basis of comparison which was a conventional chilled water system, modeled using the temperature bin method described in ASHRAE(1983)

By eliminating any compressor contribution to the billing demand, the "idealized" case could be achieved. Cooling cost savings would increase from 22% to 54%. in addition, the overall load factor increases to about 58%. (Compared to 35% for a conventional system).

C. E. Francis [20] studied the production of ice with long-term storage. Cold air during winter is used for producing relatively large amounts of ice in long-term underground storage. The stored cooling capacity of the ice is recovered during the following summer. Initial research indicates that, this type of system could provide summer air conditioning, refrigeration or process cooling in regions the winter is sufficiently cold and sufficient duration.

T. A. Gilbertson [21] studied the ice cools office-hotel complex. The system is sized to operate over a seven day cycle. It is fully charged with ice over the weekend. During the period of Monday through Thursday, ice is made during off-peak periods to replace partially what was melted during the on-peak cooling period. The chiller do not operate between the hours of 11 a.m. and 6 p.m. on week days per the utility company's time-of-day rate requirements. During the hours, the entire cooling load is handled entirely by the stored ice.

Eleven steel ice storage tanks were installed under the ramps in the third (lowest) parking level under the building. The ice storage system will reduce peak building demand by 1700 kW by shifting much of the air conditioning ton-hr generation to off-peak periods with their lower power charges.

R. T. Tamblyn [22] studied the chilled water storage of a college.The water storage option is likely to provide still better economic, however and greater flexibility when linked to a central chilled water system. An example to illustrate this point is the hypothetical Casey Stengel Institute for Advanced Studies in Elocution.

When new buildings are added to a central chilled water loop it may be cheaper in first cost to add full storage with chilled water to the new buildings than to expand the chiller capacity at central plant. This preserves the capacity of the distribution loop and central pumps while reducing the electric demand cost for chillers.

R. A. Wistort and D. L. Nurisso [23] studied the chilled water storage. The project designed was a 500000 sq. ft. electronic assembly plant having a fully air conditioned manufacturing plant, office, cafeteria etc. with a minimum 350000 gal. water storage tank for fire protection water storage. The calculated peak load of 1360 tons occurred at approximately 4:00 p.m.

The chiller are activated to charge the storage tank for the next day's usage. The basic promise herein is that the thermal storage tank would be charged with chilled water during off-peak and partial peak hours only. The chiller would not be allowed to run during on-peak periods. Chilled water required during the on-peak periods would be drawn only from the tank. That has a significant economic justification.

D. P. Gatley [24] studied the cooling thermal storage. Buildings with load profiles conductive to thermal storage. Office buildings are excellent candidates. They have low cooling requirements in the morning followed by a high peak around 3:00 p.m. and have an afternoon period of four or five hours with very high loads and a morning period of four or five hours with medium to low loads.

Hotels are not generally good candidates for full storage because of 24 hours cooling loads and a long peak cooling loads demand extending from approximately 11:00 a.m. to 8:00 p.m.They can be candidates for partial storage.

Edison Electric Institute [25] designed an ice storage system for innovative bank for savings at Livermore facility. The refrigeration plant operation is completely automatic. Electronic time controls allow the plant to operate, at maximum, eighteen hours a day during off-peak and mid-peak hours of 6:00 p.m. to 12:00 noon, Sunday night through Friday morning.

The chilled water is system is an open tank, flat plate ice builder. A bypass valve is provided across the chilled water supply and return lines to limit the temperature difference on the ice bank to 8 F or less.The system is not using any more energy than would be expected using conventional cooling equipment.

T. W. Brady [26] made a thermal storage for the Merchandise Mart building in Chicago, Illinois. The thermal storage system may be the world's largest ice thermal storage system. It is capable of produce in 2200000 pounds of ice each day. Ice thermal storage systems make ice at night, the ice helps to cool the water for air conditioning in the building HVAC system during the peak demand hours.

instantaneous cooling needed during peak hours comes from parallel evaporators working with two screw compressors functioning as standard chillers.

Ice thermal storage systems are an effective means of shifting cooling loads from on-peak to off-peak electrical periods, thereby saving operating costs. First costs are comparable or less than the costs of conventional systems when properly designed.

D. A. Knebel and S. Houston [27] retrofitted a thermal storage system to the Fort Worth's Wartington Hotel, with 525000 sq. ft. of air conditioned space. The original cooling system consisted of three 400 HP open screw compressor heat-pump chiller packages and seven direct expansion 50000 lb ice on pipe thermal storage tanks. The heat pump chillers were to operate in several modes.

Mode 1- direct chilling; the chiller directly served the building load.
Mode 2- heat recovery; when building cooling was needed, the chiller meet the cooling load directly. When heating was needed, it would be rejected from the condenser into the building heating loop.
Mode 3- ice storage; one compressor would operate at night to charge the ice on the pipe thermal energy storage system while the other two compressors met direct building load.

The actual building electrical peak demand during the on-peak period was, 1880 kW the partial thermal storage system effectively reduced the peak by 604 kW from the pre-storage level of 2484 kW.

By removing all chiller compressors, icemaker compressors cooling tower and condenser, water pumps from the operating during the on-peak period, it was determined that the building electrical demand could be reduced to 1101 kW.

CHAPTER 3

MODELLING OF AN AIR CONDITIONING SYSTEM WITH AND WITHOUT A THERMAL ENERGY STORE

A brief description of the models for summer air conditioning with a man-made seasonal or daily thermal energy store was given in Chapter 1. This chapter includes a description of the processes in the two models considering different working strategies. The problem formulation to be given later will be based on the description given in this chapter. Formulation will be obtained for two different models with four different working strategies.

Two different models considered are hot and cold time active systems. Four different working strategies considered are conventional, full and partial storage for a hot time active system and full storage for a cold time active model.

First Law of Thermodynamics is used to derive energy balance equations for each component of the systems under study. Effectiveness-NTU method is used for heat exchangers. Finite differences and Newton's method are used to solve the ordinary differential equations.

3.1 HOT TIME ACTIVE AIR CONDITIONING SYSTEM WITH A THERMAL ENERGY STORAGE TANK

3.1.1 System Description

A conventional air conditioning system, shown in Figure 3.1, has a cooling tower, a chiller unit and fan-coils. Chiller unit is composed of a condenser, a compressor and an evaporator. The cooling water of the condenser is cooled by a cooling tower, at hot times when the chiller unit is operational.

An air conditioning system with a thermal energy storage tank is shown in Figure 3.2. It has an extra storage tank between the chiller unit and heat exchanger. An air/water heat exchanger is used inplace of a cooling tower. This system dissipates heat from condenser to a sink that has a lower temperature than temperature of cooling tower water in a conventional system. The low temperature at the storage tank is obtained by circulating condenser water to a heat exchanger when the outside air temperature is lower than the water temperature. Working part of the system is schematically demonstrated with a dashed line box. The cold water is stored in an insulated storage tank. The stored coldness is used for cooling the condenser at hot time when air conditioning is needed. The other part of the system works when air conditioning is necessary and is shown by a full line box.

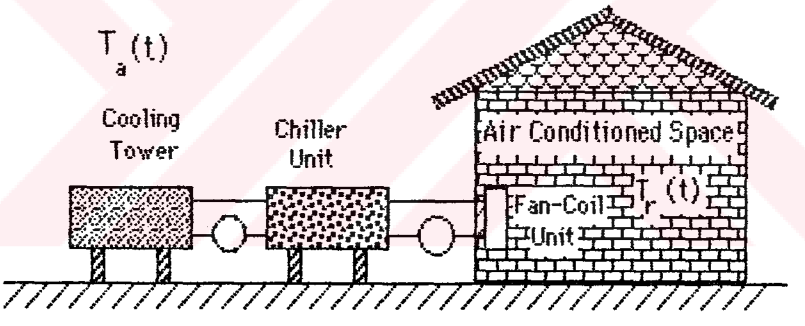


Figure 3.1 A Conventional Air Conditioning System

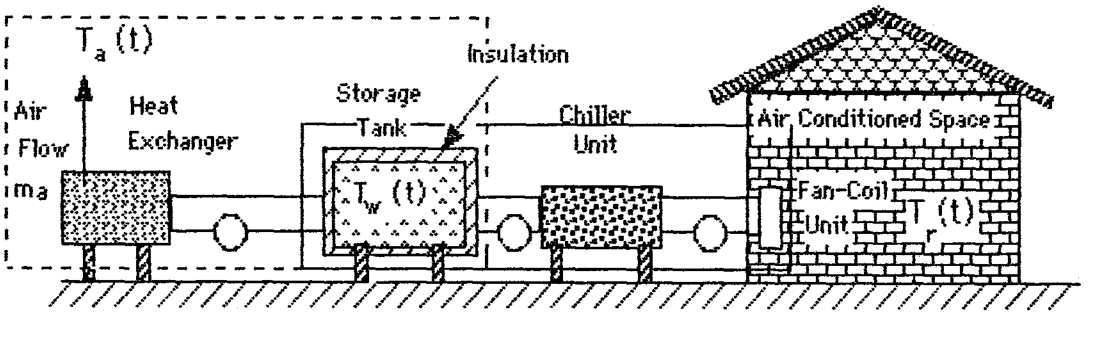


Figure 3.2 A Hot Time Active Air Conditioning System With a Thermal Energy Storage Tank

3.1.2 Formulation for a Conventional Air Conditioning System:

First Law of Thermodynamics, LMTD and effectiveness-NTU methods are used for modelling of the system. Energy balance equation is applied locally for each component of the system.

The heat losses from pipes which are used for connecting the components of the system are assumed to be negligible.

The system understudy is shown in Figure 3.1. Operation of the compressor is controlled by a temperature sensor located in the air conditioned space.

Energy balance equations for each component of the system are obtained by using the effectiveness-NTU method as follows [30].

For the fan-coil unit in the air conditioned space:

$$Q_{fc} = (\varepsilon C_{min})_{fc}(T_r - T_{wi}) \qquad (3.1.2.1)$$

$$\varepsilon = \frac{1}{X}\,\frac{T_{wo} - T_{wi}}{T_r - T_{wi}} \qquad (3.1.2.2)$$

where

$$C_{min} = mC_p)_{min} \qquad (3.1.2.3)$$

$$X = 1 \qquad \text{if} \qquad C_{pr}\Delta T_r \leq C_{pw}\Delta T_w \qquad (3.1.2.4)$$

$$X = (C_{min}/C_{max})_{fc} \qquad \text{if} \qquad C_{pr}\Delta T_r > C_{pw}\Delta T_w \qquad (3.1.2.5)$$

For the evaporator of the chiller unit:

$$Q_e = (\varepsilon C_{min})_e(T_{wo} - T_e) \qquad (3.1.2.6)$$

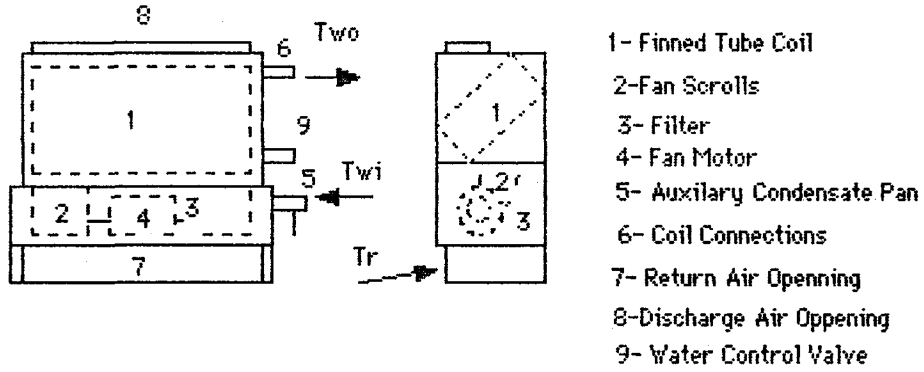


Figure 3.3 A Typical Fan-Coil Unit

For the compressor of the chiller unit:

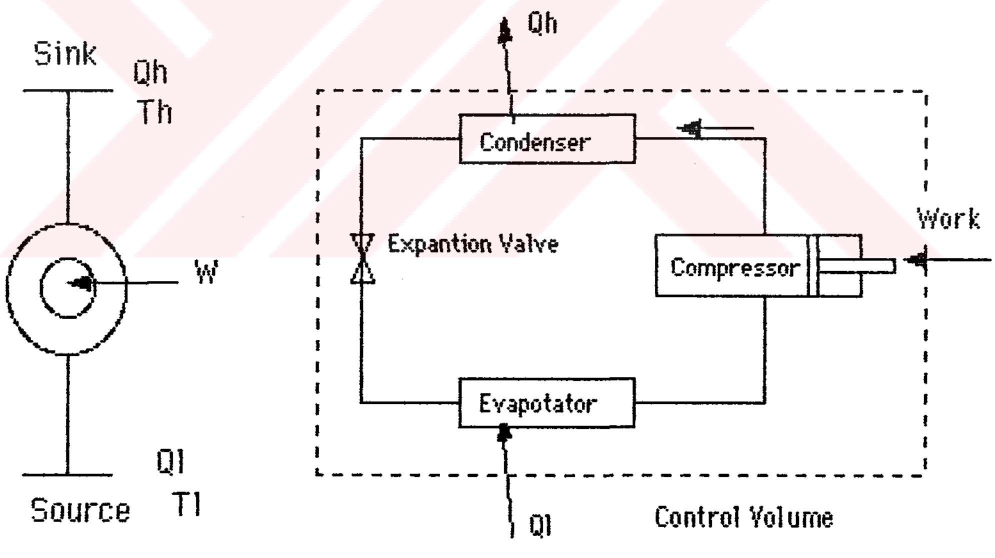


Figure 3.4 Schematics of a Refrigeration Cycle

$$Q_h = Q_l + W \quad \text{or} \quad Q_c = Q_e + W \tag{3.1.2.7}$$

The coefficient of performance, COP is [33];

$$COP = \frac{Q_e(\text{energy sought})}{W(\text{energy that costs})}$$

$$= \frac{Q_e}{Q_c - Q_e} = \frac{1}{Q_c/Q_e - 1} = \frac{1}{T_c/T_e - 1} \qquad (3.1.2.8)$$

The following equation is used for calculating COP of system

$$COP = \frac{\beta}{T_c/T_e - \gamma} \qquad (3.1.2.9)$$

For condenser of the chiller unit:

$$Q_c = -(\varepsilon C_{min})_c (T_a - T_c) \qquad (3.1.2.10)$$

Energy balance equation for the air conditioned space is:

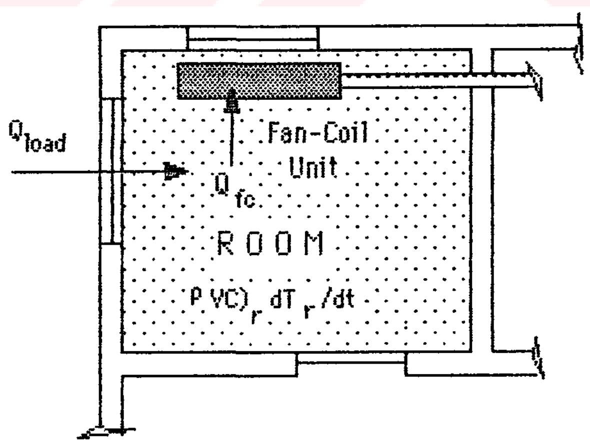


Figure 3.5 Schematics of an Air Conditioned Space

$$Q_{ld} = Q_r + Q_{fc} \qquad (3.1.2.11)$$

$$Q_{ld} = \rho_r V_r C_{pr} \frac{dT_r}{dt} + \delta_1 Q_{fc} \qquad (3.1.2.12)$$

Equations given above are converted to dimensionless forms by using the following parameters.

$$\phi = \frac{T-T_{inf}}{T_{inf}} \qquad (3.1.2.13)$$

$$q = Q/UA)_{fc}T_{inf} \text{ and } w = W/UA)_{fc}T_{inf} \qquad (3.1.2.14)$$

$$\tau = \frac{t}{24} \qquad (3.1.2.15)$$

Equations 3.1.2.1 to 3.1.2.12 now become:

For a fan-coil unit:

$$N_{fc}q_{fc} = \phi_r - \phi_{wi} \qquad (3.1.2.16)$$

$$\varepsilon = \frac{1}{X}\frac{\phi_{wo}-\phi_{wi}}{\phi_r-\phi_{wi}} \qquad (3.1.2.17)$$

where

$$X = 1 \quad \text{if} \quad C_p\Delta\phi_r \le C_{pw}\Delta\phi_w \qquad (3.1.2.18)$$

$$X = (C_{min}/C_{max})_{fc} \quad \text{if} \quad C_p\Delta\phi_r > C_{pw}\Delta\phi_w \qquad (3.1.2.19)$$

For the evaporator of the chiller unit:

$$R_{e,fc}N_e q_e = \phi_{wo} - \phi_e \qquad (3.1.2.20)$$

For the compressor of the chiller unit:

$$q_c = q_e + w \qquad (3.1.2.21)$$

$$COP = \frac{q_e}{w} = \frac{\beta}{\frac{\phi_c+1}{\phi_e+1} - \gamma} \tag{3.1.2.22}$$

For the condenser of the chiller unit:

$$R_{c,fc}N_c q_c = -(\phi_a-\phi_c) = \phi_c-\phi_a \tag{3.1.2.23}$$

Energy balance equation for the air conditioned space:

$$q_{ld} = \frac{S d\phi_r}{d\tau} + \delta_1 q_{fc} \tag{3.1.2.24}$$

where Nfc, Ne, Nc, Refc, Rcfc, and S are defined as follows:

$$N_{fc} = UA)_{fc}/\varepsilon C_{min})_{fc} \tag{3.1.2.25}$$

$$N_e = UA)_e/\varepsilon C_{min})_e \tag{3.1.2.26}$$

$$N_c = UA)_c/\varepsilon C_{min})_c \tag{3.1.2.27}$$

$$R_{e,fc} = UA)_{fc}/UA)_e \tag{3.1.2.28}$$

$$R_{c,fc} = UA)_{fc}/UA)_c \tag{3.1.2.29}$$

$$S = mC_p)_r/UA)_{fc}24 \tag{3.1.2.30}$$

and the multiplication factor δ_1 is defined as follows:

$\delta_1 = 1$ if $w \neq 0$

$\delta_1 = 0$ if $w = 0$

where w is the dimensionless compressor work.

Equation 3.1.2.17 will be solved for ϕ_{wo} and substituted into *equation 3.1.2.20 in order to eliminate* ϕ_{wo} *from the problem.* The heat load of air conditioned space, q_{fc} is equal to the heat of the evaporator, q_e. q_e will be replaced by q_{fc} in equation 3.1.2.16. ϕ_{wi} from equation 3.1.2.16, will be substituted into equation 3.1.2.20, and the following expression will be obtained.

$$Rq_e = \phi_r - \phi_e \tag{3.1.2.31}$$

The following is obtained when equation 3.1.2.21 is solved for q_e and substituted into equations of 3.1.2.22, 3.1.2.24 and 3.1.2.31.

$$q_c - w = \beta w \frac{\phi_e + 1}{\phi_c + 1 - \gamma(\phi_e + 1)} \tag{3.1.2.32}$$

$$q_{ld} = \frac{S d\phi_r}{d\tau} + \delta_1 (q_c - w) \tag{3.1.2.33}$$

$$R(q_c - w) = \phi_r - \phi_e \tag{3.1.2.34}$$

where

$$R = R_{e,fc} N_e + (1 - \varepsilon_{fc} X) N_{fc} \tag{3.1.2.35}$$

The following is obtained by solving equations 3.1.2.34, 3.1.2.23 and 3.1.2.21 for ϕ_e, ϕ_c, and q_e respectively are inserting resulting expressions into equation 3.1.2.22.

$$q_c - w = \beta w \frac{\phi_r - R(q_c - w) + 1}{R_{cfc} N_c q_c + \phi_a + 1 - \gamma(\phi_r - Rq_c + Rw + 1)} \tag{3.1.2.36}$$

A second order polynomial is obtained when equation 3.1.2.36 is solved for q_c. This polynomial is a function of β, γ, ϕ_r, ϕ_w, N and R.

Equation 3.1.2.36 may be expressed as:

$$aq_c^2+bq_c+c = 0 \qquad (3.1.2.37)$$

where

$$a = R_{cfc}N_c+\gamma R \qquad (3.1.2.38)$$

$$b = 0.5(\phi_a+1-\gamma\phi_r-2\gamma Rw-\gamma-R_{c,fc}N_c w+Rw\beta) \qquad (3.1.2.39)$$

$$c = w^2R(\gamma-\beta)-w(\phi_a+1-\gamma\phi_r-\gamma+\beta\phi_r+\beta) \qquad (3.1.2.40)$$

The solution of equation 3.1.2.37 is given by following formula.

$$q_c = -\frac{b}{a}+\left\{\left(\frac{b}{a}\right)^2-\frac{c}{a}\right\}^{1/2} \qquad (3.1.2.41)$$

3.1.3 Formulation for a Hot Time Active Air Conditioning System With a Thermal Energy Store.

3.1.3.1 Formulation for Full Storage:

A hot time active air conditioning system with a thermal energy storage tank is illustrated in Figure 3.2. Components in this system were previously explained. Full storage is explained at this section.

What is meant by full storage is; cooling of the condenser by water stored in the cold energy in tank during all times, when air conditioning is necessary.

Equations will be derived for extra parts of the system (heat exchanger, storage tank) for modelling of a hot time active air conditioning system with a thermal energy store.

For the heat exchanger:

$$Q_{he} = (\varepsilon C_{min})_{he}(T_w-T_a) \qquad (3.1.3.1)$$

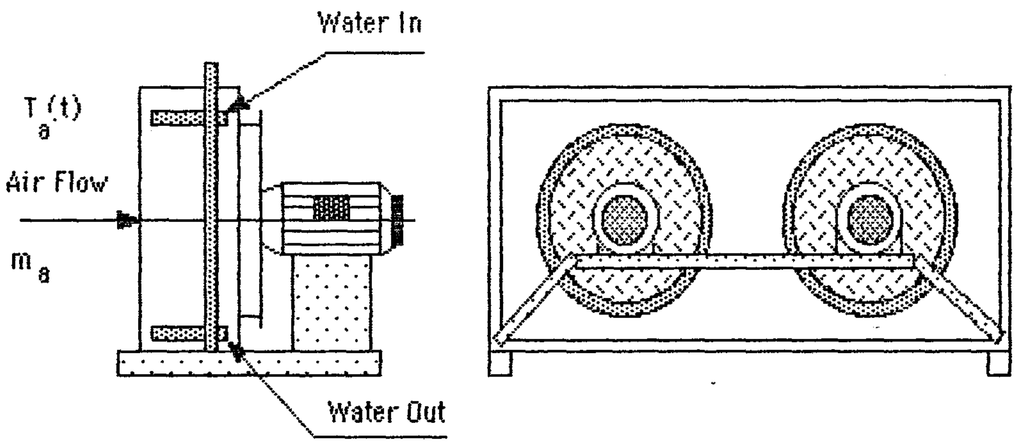


Figure 3.6 A Typical Heat Exchanger

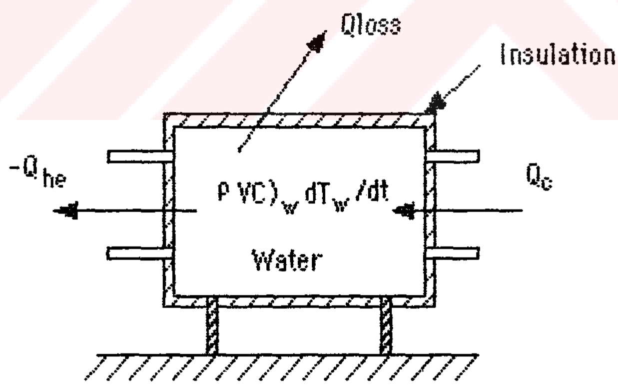


Figure 3.7 Illustration of Energy Balance of The Storage Tank

Energy balance equation for the storage tank is:

$$Q_s = Q_c + Q_{he} \tag{3.1.3.2}$$

$$Q_s = \rho_w V_w C_w \frac{dT_w}{dt} + UA)_s(T_w - T_s) \tag{3.1.3.3}$$

Dimensionless forms of equations 3.1.3.1, 3.1.3.2 and 3.1.3.3 are as follows:

For the heat exchanger:

$$R_{he,fc}N_{he}q_{he} = \phi_w - \phi_a \tag{3.1.3.4}$$

Energy balance equations for the storage tank is:

$$q_s = q_c + q_{he} = q_{st} + q_{ls} \tag{3.1.3.5}$$

$$q_s = P\frac{d\phi_w}{\phi\tau} + R_{fc,s}(\phi_w - \phi_s) \tag{3.1.3.6}$$

where $R_{he,fc}$, $R_{fc,s}$, N_{he}, and P are defined as follows:

$$R_{he,fc} = UA)_{fc}/UA)_{he} \tag{3.1.3.7}$$

$$R_{fc,s} = UA)_s/UA)_{fc} \tag{3.1.3.8}$$

$$N_{he} = UA)_{he}/\varepsilon C_{min})_{he} \tag{3.1.3.9}$$

$$P = \rho_w V_w C_w/UA)_{fc}24 \tag{3.1.3.10}$$

Condenser of chiller unit is not cooled directly by the outside air temperature in this model. It is cooled by water that must have a lower temperature than the outside temperature during air condition is operational. This is very important for full storage in order to cool condenser by a sink that has a lower temperature.

Effectiveness-NTU method is reapplied to the condenser;

$$Q_c = -(\varepsilon C_{min})_c(T_w - T_c) \tag{3.1.3.11}$$

The difference between equations 3.1.2.10 and 3.1.3.11 is that T_w and it is replaced by T_a. The dimensionless form of this equation is as follows:

$$R_{c,fc}N_c q_c = -(\phi_w-\phi_c) = \phi_c-\phi_w \qquad (3.1.3.12)$$

Now, the equation 3.1.3.12 is solved for ϕ_c, and equation 3.1.2.36 is rewritten by substituting ϕ_c, from equation 3.1.3.12 to obtained the following.

$$q_c-w = \beta w\frac{\phi_r-R(q_c-w)+1}{R_{cfc}N_c q_c+\phi_w+1-\gamma(\phi_r-Rq_c+Rw+1)} \qquad (3.1.3.13)$$

The following second order polynomial expression is obtained when equation 3.1.3.13 is solved for q_c.

$$aq_c2+bq_c+c = 0 \qquad (3.1.3.14)$$

The solution is given by equation 3.1.3.15.

$$q_c = -\frac{b}{a}+\left\{\left(\frac{b}{a}\right)^2-\frac{c}{a}\right\}^{1/2} \qquad (3.1.3.15)$$

a, b, and c are now given by:

$$a = R_{c,fc}N_c+\gamma R \qquad (3.1.3.16)$$

$$b = 0.5(\phi_w+1-\gamma(\phi_r+2Rw+1)-w(R_{c,fc}N_c-R\beta)) \qquad (3.1.3.17)$$

$$c = w^2R(\gamma-\beta)-w(\phi_w+1-\gamma\phi_r-\gamma+\beta\phi_r+\beta) \qquad (3.1.3.18)$$

Equation 3.1.3.6 will be written considering heat transferred to the storage tank.

$$\delta_1 q_c - \delta_2 q_{he} = p\frac{d\phi_w}{d\tau} + R_{fc,s}(\phi_w - \phi_s) \qquad (3.1.3.19)$$

ϕ_s is the dimensionless temperature of surroundings, of the space occupied by the storage tank.

δ_1 and δ_2 are defined as follows:

$\delta_1 = 1$ if $w \neq 0$

$\delta_1 = 0$ if $w = 0$

and

$\delta_2 = 1$ when $T_a < T_w$

$\delta_2 = 0$ when $T_a \geq T_w$

3.1.3.2 Formulation for Partial Storage:

System for partial storage is illustrated in Figure 3.8. Partial storage is defined as a combination of a conventional air conditioning system and a thermal energy storage [26]. This model is suitable for followings: a) a thermal energy storage system retrofitted to an old conventional air conditioning system and b) when space is restricted for a full storage.

Therefore, after a certain time, water gets a higher temperature than outside air temperature, due to a small storage tank. So that, condenser is cooled by the water until it reaches to outside air temperature, after that, it is cooled directly by air, in partial storage model.

For partial storage; equations of conventional air conditioning and thermal energy storage systems are combined with a check such as;

system is thermal energy storage when $T_w < T_a$

system is conventional air cond. when $T_w \geq T_a$

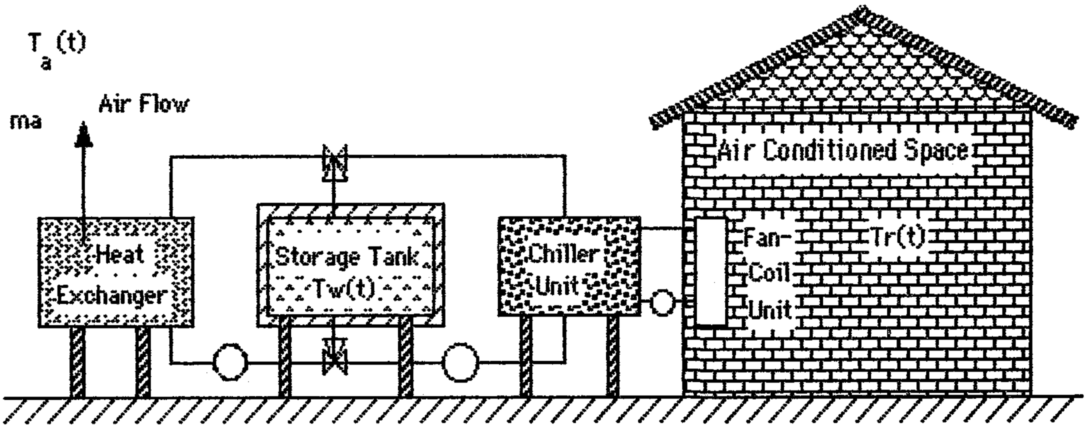


Figure 3.8 A Partial Storage Model

3.2 COLD TIME ACTIVE AIR CONDITIONING SYSTEM WITH A THERMAL ENERGY STORAGE TANK

3.2.1 System Description:

Second model illustrated in Figure 3.9, is for cooling the condenser by a sink that has a lower temperature than the outside air temperature when a conventional air conditioning is necessary. This system is composed of a chiller unit, a storage tank, and fan-coil units. There is no heat exchanger in this system. The differences between these hot and cold time active models are the place of the storage tank and working time of the chiller unit. Storage tank is placed between heat exchanger and storage tank in hot time active system. But, it is placed between chiller unit and fan-coil units in cold time active system [7], [9].

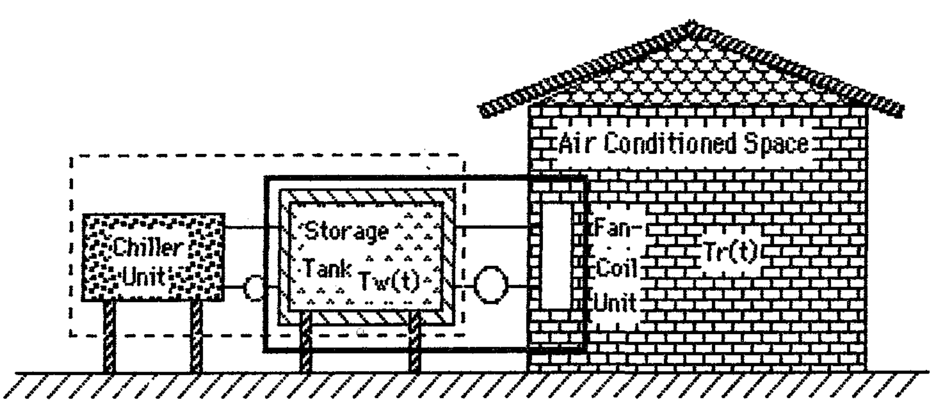


Figure 3.9 A Cold Time Active Model

Cold air is also used as a cold energy source. Temperature difference between hot and cold time is wanted to be useful in two models. But, natural cold energy is not stored in a storage tank in this system. Cold air is directly used to cool the condenser when the chiller unit is operational (at night or winter). The chiller is activated to charge the storage tank for hot time's usage during cold time. The chiller is not be allowed to run during on-peak period. Because the thermal storage thank is charged with chilled water during off-peak time [23]. The working part of the system during cold time is schematically demonstrated with a dashed line box. Then, the chilled water is stored into an insulated tank for using at hot time [5]. The chilled water storage tank must has a volume that can store enough energy to keep air conditioned space between comfort temperature zone. Because, when air conditioning is necessary, the chilled water is circulated between the storage tank and fan-coils as demonstrated in Figure 3.9 with a full line box.

3.2.2 Formulation for a Cold Time Active Air Conditioning System With a Thermal Energy Store

For mathematical modelling of the cold time active air conditioning system with thermal energy store, same theoretical methods are used for hot time active model would be also used. They are energy balance equation, LMTD and ε-NTU methods.

The energy balance equation for each component of cold time active model by using effectiveness NTU method are also written as follows.

For the fan-coil units in the air conditioned space:

$$Q_{fc} = (\varepsilon C_{min})_{fc}(T_r - T_{wi}) \qquad (3.2.2.1)$$

$$\varepsilon_{fc} = \frac{1}{X}\frac{T_{wo}-T_{wi}}{T_r-T_{wi}} \qquad (3.2.2.2)$$

where

$$C_{min} = mC_p)_{min} \qquad (3.2.2.3)$$

and

$$X = 1 \qquad \text{if} \qquad C_{pr}\Delta T_r \leq C_{pw}\Delta T_w \tag{3.2.2.4}$$

$$X = (C_{min}/C_{max})_{fc} \qquad \text{if} \qquad C_{pr}\Delta T_r > C_{pw}\Delta T_w \tag{3.2.2.5}$$

For the evaporator of the chiller unit:

$$Q_e = \varepsilon C_{min})_e(T_{wo}-T_e) \tag{3.2.2.6}$$

For the compressor of the chiller unit:

$$Q_c = Q_e + W \tag{3.2.2.7}$$

$$COP = \frac{Q_e}{W} = \frac{Q_e}{Qc-Qe} = \frac{1}{Q_c/Q_e-1} = \frac{1}{T_c/T_e-1} \tag{3.2.2.8}$$

$$COP = \frac{Q_e}{W} = \frac{\beta}{T_c/T_e-\gamma} \tag{3.2.2.9}$$

For the condenser of the chiller unit:

$$Q_c = \varepsilon C_{min})_c(T_a-T_c) \tag{3.2.2.10}$$

Energy balance equation for the storage tank is:

$$Q_s = \rho_w V_w C_{pw}\frac{dT_w}{dt}+UA)_{st}(T_w-T_s) \tag{3.2.2.11}$$

$$Q_s = Q_{fc}+Q_e \tag{3.2.2.12}$$

Energy balance equation for air conditioned space is:

$$Q_{ld} = m_r C_{pr}\frac{dT_r}{dt} + \delta 1 Q_{fc} \qquad (3.2.2.13)$$

Equations given above are converted into the dimensionless forms by using the parameters in equations 3.1.2.13, 3.1.2.14 and 3.1.2.15 as follows:

For the fan-coil:

$$N_{fc} q_{fc} = \phi_r - \phi_{wi} \qquad (3.2.2.14)$$

$$\varepsilon_{fc} = \frac{1}{X}\frac{\phi_{wo}-\phi_{wi}}{\phi_r-\phi_{wi}} \qquad (3.2.2.15)$$

where

$$X = 1 \qquad \text{if} \qquad C_{pr} d\phi_r \leq C_{pw} d\phi_w \qquad (3.2.2.16)$$

$$X = (C_{min}/C_{max})_{fc} \qquad \text{if} \qquad C_{pr} d\phi_r > C_{pw} d\phi_w \qquad (3.2.2.17)$$

For the evaporator of the chiller unit:

$$R_{e,fc} N_e q_e = \phi_{wo} - \phi_e \qquad (3.2.2.18)$$

For the compressor of the chiller unit:

$$q_c = q_e + W \qquad (3.2.2.19)$$

$$COP = \frac{Q_e}{W} = \frac{\beta}{\dfrac{\phi_c+1}{\phi_e+1} - \gamma} \qquad (3.2.2.20)$$

For the condenser of the chiller unit:

$$R_{c,fc} N_c q_c = \phi_a - \phi_c \qquad (3.2.2.21)$$

Energy balance equation for the storage tank in dimensionless form:

$$q_s = P\frac{d\phi_w}{d\tau} + R_{fc,s}(\phi_w - \phi_s) \qquad (3.2.2.22)$$

$$q_s = q_{fc} + q_e \qquad (3.2.2.23)$$

For the air conditioned space energy balance equation is:

$$q_{ld} = S\frac{d\phi_r}{d\tau} + \delta_1 q_{fc} \qquad (3.2.2.24)$$

where S, P, Rfc, s, Nc, Nfc, Rc, fc, Re, fc, Ne are defined previously, in the part of hot time active system.

The following energy balance equation is obtained for the storage tank when it is written by considering the working conditions of fan-coil and chiller unit.

$$\delta_1 q_{fc} - \delta_2 q_e = P\frac{d\phi_w}{d\tau} + R_{fc,s}(\phi_w - \phi_s) \qquad (3.2.2.25)$$

δ_1, and δ_2 in equations 3.2.2.24 and 3.2.2.25 are defined as follows:

for δ_1

$\delta_1 = 1$ if $T_r' \leq T_r$ or $T_r' > T_{rmax}$ and $T_r' > T_r$

$\delta_1 = 0$ if $T_r' > T_r$ or $T_r' < T_{rmin}$ and $T_r' < T_r$

where T_{rmax} and T_{rmin} are maximum and minimum limits of comfort temperature range.

for δ_2

$\delta_2 = 1$ if $w \neq 0$

$\delta_2 = 0$ if $w = 0$

Equation 3.2.2.14 will be solved for ϕ_{wi} and substituted into equation 3.2.2.15 in order to eliminate ϕ_{wi} from the problem. ϕ_{wo} from equation 3.2.2.15 will be substitute into equation 3.2.2.18. Finally, the following expression will be obtained for ϕ_e is obtained.

$$\phi_e = \phi_r - N_{fc}q_{fc}(1-X\varepsilon_{fc}) - R_{e,fc}N_e q_e \tag{3.2.2.26}$$

and equations 3.2.2.21 and 3.2.2.19 are solved for ϕ_e and ϕ_c respectively, they are;

$$\phi_c = \phi_a + R_{c,fc}N_c q_c \tag{3.2.2.27}$$

$$q_e = q_c - W \tag{3.2.2.28}$$

The following COP equation is obtained, when ϕ_e, ϕ_c, and q_e are substituted into equation 3.2.2.20.

$$COP = \frac{q_e}{W} = \frac{q_c - W}{W}$$

$$= \frac{\beta(\phi_r - N_{fc}q_{fc}(1-X\varepsilon_{fc}) - R_{efc}N_e(q_c - W) + 1)}{\phi_a + R_{cfc}N_c q_c + 1 - \gamma(\phi_r - N_{fc}q_{fc}(1-X\varepsilon_{fc}) - R_{efc}N_e(q_c - W) + 1)} \tag{3.2.2.29}$$

The following second order polynomial expression is obtained when equation 3.2.2.29 is solved in terms of q_c.

$$aq_c^2 + bq_c + c = 0 \tag{3.2.2.30}$$

The solution is given by equation 3.26.

$$q_c = -\frac{b}{a} + \left\{\left(\frac{b}{a}\right)^2 - \frac{c}{a}\right\}^{1/2} \tag{3.2.2.31}$$

a, b, c are now given by

$$a = R_{c,fc}N_c + \gamma R_{e,fc}N_e \tag{3.2.2.32}$$

$$b = 0.5[\phi_a + 1 - \gamma\phi_r + \gamma N_{fc}(1 - x\varepsilon_{fc})q_{fc} - 2\gamma R_{e,fc}N_e W$$
$$-\gamma - W(R_{c,fc}N_c - \beta R_{e,fc}N_e)] \tag{3.2.2.33}$$

$$c = -W(\phi_a + 1) + W(\gamma - \beta)(1 + \phi_r) + W N_{fc}(1 - x\varepsilon_{fc})(\beta - \gamma)q_{fc}$$
$$+ R_{e,fc}N_e W^2(\gamma - \beta) \tag{3.2.2.34}$$

3.3 CALCULATION OF THE OVERALL HEAT TRANSFER COEFFICIENT OF THE STORAGE TANK

Construction of the storage tank for storing cold energy by water is illustrated in Figure 3.10. The tank which is a regtangular prismatic is insulated with glass wool and it would be placed open to the outside air or buried underground by soil. Especially for seasonal storage, tank is sometimes placed under the parking lot of building [6]. So that, model can be applicable with a less placement arranged for the system.

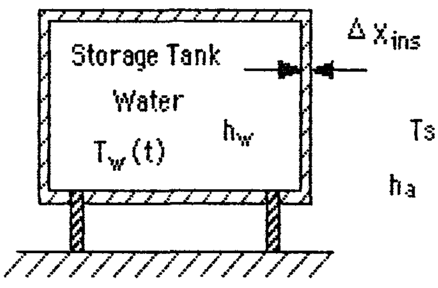


Figure 3.10 A Typical Storage Tank

The overall heat transfer coefficient

$$U_{st} = \frac{1}{R_1 + R_2 + R_3 + \ldots + R_n} \tag{3.3.1}$$

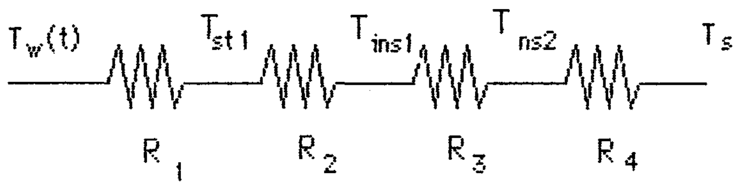


Figure 3.11 Thermal Resistance Between Water and Space

$$R_1 = \frac{1}{h_w A_{st}} \tag{3.3.2}$$

$$R_2 = \frac{\Delta X_{st}}{k_{st} A_{st}} \tag{3.3.3}$$

$$R_3 = \frac{\Delta X_{ins}}{k_{ins} A_{st}} \tag{3.3.4}$$

$$R_4 = \frac{1}{h_{air} A_{st}} \tag{3.3.5}$$

$$U_{st} = \frac{1}{\dfrac{1}{h_w A_{st}} + \dfrac{\Delta X_{st}}{k_{st} A_{st}} + \dfrac{\Delta X_{ins}}{k_{ins} A_{st}} + \dfrac{1}{h_{air} A_{st}}} \tag{3.3.6}$$

where A_{st} is the total outside surface area of the storage tank as;

$$A_{st} = 2(a_{st}b_{st}+b_{st}c_{st}+c_{st}a_{st}) \tag{3.3.7}$$

and a_{st}, b_{st}, and c_{st} are the dimensions of the storage tank.

3.4 SELECTION OF THE STORAGE MATERIAL

Two storage concepts standpoised to help level electric demand and save money for customers and utilities alike: ice and water. The technique for using each is somewhat different. Which is likely to dominate on the other in the long run and what are the characteristics of each?

It is early to announce a winner but, if the number of active promoters is a factor, ice will easily win over water. With no champion for chilled water storage, but many motivated manufacturers trying to sell ice storage, the latter is liable to dominate future designs. Water storage has greater flexibility for storing waste heat, so there is potential for future development. Meanwhile, both systems fulfill the national purpose of energy optimization, so competition will be healthy for each [33].

In a temperature change, 1 kg of water stores 4186 J for every degree C of temperature change. In a phase change, 1 kg of water stores 335 kJ in changing from 1 kg at $0^{o}C$ liquid to 1 kg at $0^{o}C$ solid. When changing water from a liquid to gas 1 kg of liquid stores 2.44 MJ in changing to a gas at atmospheric pressure.

Thus, steam is not a practical storage medium because of the huge volumes involved, but water remains practical in changing temperature or changing phase to ice.

The current argument is over which is better for storage, water with a temperature changes or ice. There is no clear-cut answer. The merit of water is that storage can be accomplished with existing water chilling equipment simply by adding a large water volume to the system.

The merits of ice are; it takes up less volume than water and it is cheaper if it is a new construction project. Thus it depends on what's being sold. One argument used by those with a vested interest in water chiller is that it takes more energy to make ice than it does to chill water. This isn't necessarily true since it depends on the refrigeration technique used. If the same systems are used, it takes more energy to make ice to cool air to $13^{o}C$ than it does to chill water to cool air to $13^{o}C$. It also takes more energy to boil refrigerant in a direct system to cool air to $13^{o}C$ [28].

Water is also a the cheapest and available refrigerant, it isn't dangerous for surrounding.

Finally, water is selected as a storage material of cold energy in the storage tank and it is also used as a carrier of energy between evaporator and fan-coil units.

CHAPTER 4

DESCRIPTION OF THE COMPUTER PROGRAM

An interactive computer program is developed to analyze the long-term performance of a summer air conditioning system with a man-made seasonal thermal energy store (TES). This program is named as "COLD". Flowchart and listing of the program are given in Appendix 1. The program is applicable for both conventional air conditioning system and air conditioning system with thermal energy store. If the program is wanted to be run for air conditioning system with and without thermal energy store, the type of storage must be defined as whether daily [7] or seasonal [1], [4] storage. At both cases working strategies must be defined as hot or cold time active. Hot and cold times are summer and winter respectively for seasonal storage. They are taken as day and night for daily storage. That conditions are schematically illustrated in Figure 4.1.

Four types of inputs must be entered to run this program. They can be classified as, variable inputs given from computer terminal, fixed and variable inputs in data files. Last type is fixed inputs in the program. Lists of the inputs are given as follows according to their classifications.

1. Variable inputs given from terminal:
   - Type of storage as seasonal or daily, SD
   - Type of model as cold or hot time active, NDY
   - Type of working strategies, as conventional, full or partial storage, ITAC
   - Number of year for seasonal or day for daily storage to get periodic results, NY
   - Print hour for seasonal storage, TIPR

2. Variable inputs in data file:
   - Condenser capacity , Q_c
   - Volume of storage tank, A_s, B_s, C_s

- Heat exchanger capacity, Q_{he}
- Evaporator capacity , Q_e
- Fan-coil capacity , Q_{fc}

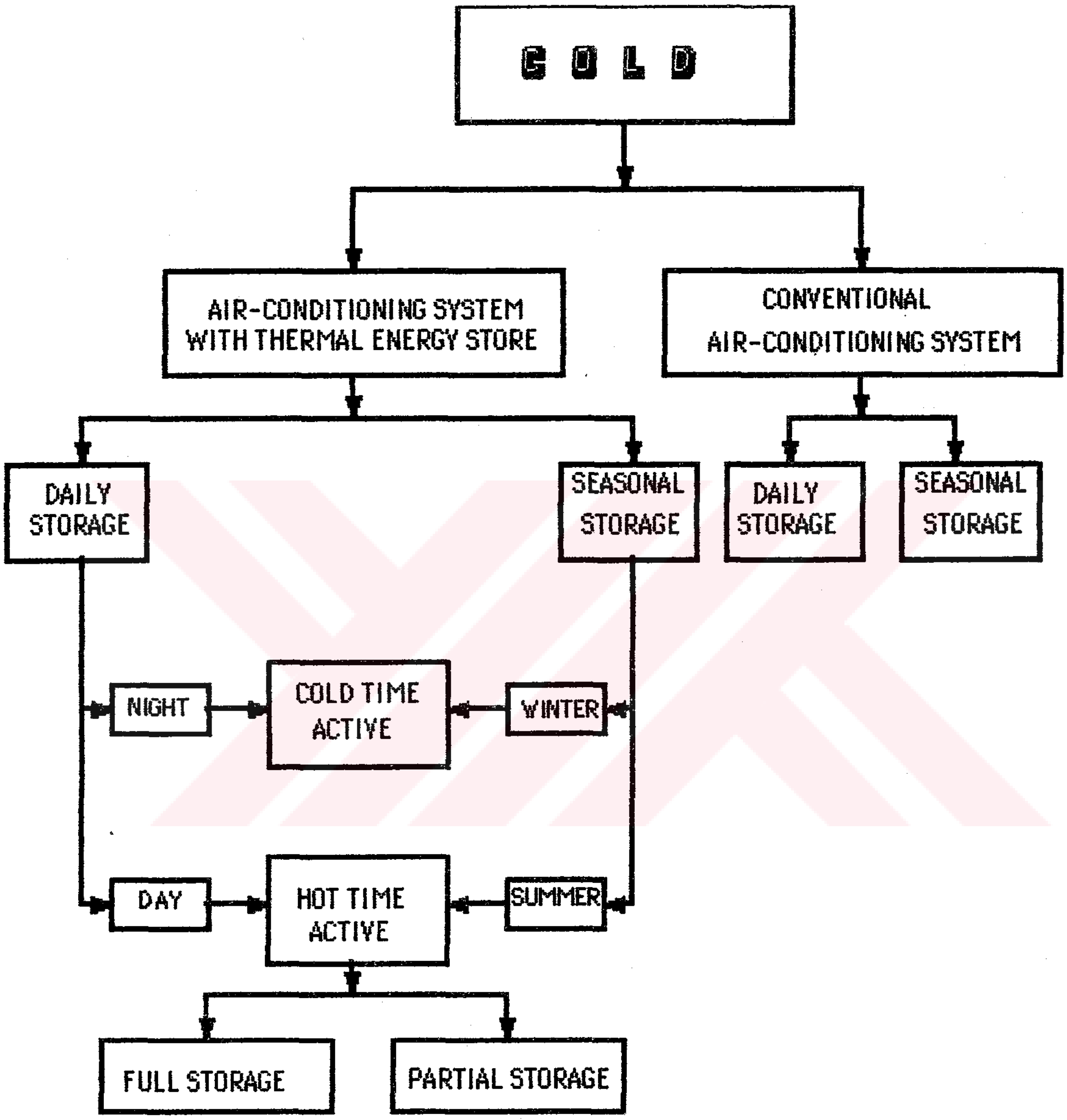


Figure 4.1 Main structure of the COLD program

3. Fixed inputs in data files:
   - Compressor work, W
   - Highest and lowest cooling capacity of compressor, Q_{lh}, Q_{ll}
   - Highest and lowest cooling temperature of compressor, T_{lh}, T_{ll}
   - Condensing temperature of chiller unit, T_h

- Input and the output fluid temperature of fan-coil, T_{fi}, T_{fo}
- Dimensions of air conditioned space, A_r, B_r, C_r
- Input and the output fluid temperature of evaporator, T_{ei}, T_{eo}
- Input and the output fluid temperature of condenser, T_{ci}, T_{co}
- Heat transfer coefficient of insulation material for storage tank, k
- Thickness of the insulation material, x
- Initial water temperature in the tank, T_w
- Initial room temperature of the air conditioned space, T_r
- Maximum and minimum limits of comfort temperature range for room, T_{max}, T_{min}
- Dimensionless coefficient of COP, β and γ
- Constant soil temperature, T_{inf}
- Time increment, Δt
- Cooling load distribution of the air conditioned space, Q_{load}
- Outside air temperature in Gaziantep, T_a

4. Fixed inputs in the program:
   - Specific heat capacity of air, C_{pair}
   - Density of air, ρ_{air}
   - Specific heat capacity of water, C_{pw}
   - Density of water, ρ_{air}
   - Volume flowrate of air, V_{air}
   - Mass flowrate of water, m_w
   - Upper and lower time limits for the compressor work at winter time (hour of year's), TIM1,TIN2
   - Upper and lower temperature limits for water stored in the tank, TEM1, TEM2

Technical data for all components of the models are given as variable and fixed inputs in data files. They are taken from manufacturer's catalogues [31]. Hourly cooling load distribution of the air conditioned space is assumed to be known. And it is constant for all day that makes the results to be periodic. The meteorological data for hourly ambient air temperature for Gaziantep given by Ref.[32] are available in data files. It is also constant for more than one year. Cooling load has hourly and monthly variation but no daily variation.

The program has two subroutines as QCON1 and INPT1. Subprogram QCON1 is used to solve equation 3.24 that is a second order polynomial equation. Other subprogram INPT1 is used to convert the inputs into dimensionless inputs as N, P, S, $\Delta\tau$ etc. by using their equations in Chapter 3.

All inputs are converted to dimensionless form by subprogram INPT1. After taking dimensionless inputs the ordinary differential equations given in Chapter 3 are solved by Finite Difference and Newton's methods. Such as results are made to be periodic and temperature at the end of each time increment for water and room is taken as an initial temperature for the following time increment. Energy output from the condenser is computed when room temperature passes the maximum limit through decreasing to minimum limit of comfort range. Energy of the heat exchanger is computed when ambient air temperature is lower then water temperature. Finally water and room temperature and COP of the model selected are computed as a function of cooling load, condenser, heat exchanger energy. The list of results computed by simulated program "COLD" is given below;

5. Outputs computed by program COLD:
 - Evaporator energy, Q_e
 - Condenser energy, Q_c
 - Heat exchanger energy, Q_{he}
 - Fan-coil energy, Q_{fc}
 - Condenser temperature, T_c
 - Evaporator temperature, T_e
 - Fan-coil temperature, T_{fc}
 - Water temperature in the storage tank, T_w
 - Temperature of air conditioned space, T_r
 - Coefficient of Performance of the selected model, COP

The results for different variable inputs are given in graphical form and discussed in the following chapter.

CHAPTER 5

RESULTS AND DISCUSSION

Results from the analysis of the summer air conditioning system with a man-made seasonal thermal energy store are given in Chapter 3 for different models. The models were used to develop an interactive computer program "COLD" to estimate the coefficient of performance (COP) of these systems and effects of the system components on their COP. The coefficient of performance is the ratio of evaporator energy to total energy demand of a system. The demands are for the compressor, pumps, heat exchanger, and the fan-coils.

Results presented in this part are obtained for both daily and seasonal storage. Outputs include results for the conventional, full and partial storage of the hot time active system. Hot time active and cold time active systems have been studied. Constant inputs given in Table 5.1 and variable inputs given in Table 5.2 and 5.3 were used for all cases.

β and γ which are coefficients of COP of the used chiller unit are obtained as 0.183 and 1.04 respectively with the help of the thecnical data of the chiller unit [31]. They are constants for all computations of daily and seasonal storage.

The surrounding temperature Ts for the tank is taken as the outside air temperature for daily storage. The difference between results for water temperature of a buried tank under the soil with a constant temperature and an unburied tank is negligibly small as shown in Figure 5.1.2. A large tank is necessary for seasonal storage. A large tank has big energy loss due to a large surface area. The tank is assumed to be placed underground with insulation and the surrounding temperature is assumed to be constant during seasonal storage.

Table 5.1 Constant Inputs for the Air Conditioning System with and without TES

Compressor work	$W = 10.0$ kW
Highest Cooling Capacity of the Compressor	$Q_{lh} = 29.3$ kW
Lowest Cooling Capacity of the Compressor	$Q_{ll} = 9.8$ kW
Highest Cooling Temperature of the Compressor	$T_{lh} = 5.0^{0}C$
Lowest Cooling Temperature of the Compressor	$T_{ll} = -35.0^{0}C$
Condensing Temperature of the Chiller Unit	$T_h = 25.00C$
Input Fluid Temperature of the Fan-coil	$T_{fi} = 10.0^{0}C$
Output Fluid Temperature of the Fan-coil	$T_{fo} = 22.0^{0}C$
Soil Temperature	$T_{inf} = 15.0^{0}C$
Dimensions of Air the Conditioned Space	$A_r = 20.0$ m.
	$B_r = 20.0$ m.
	$C_r = 3.0$ m.
Time Increment	$\Delta t = 60.0$ sec.
Inlet Water Temperature to the Evaporator	$T_{wi} = 10.0^{0}C$
Design Room Temperature	$T_r = 22.0^{0}C$
Design Evaporator Temperature	$T_e = 0.0^{0}C$
Input Fluid Temperature of the Evaporator	$T_{ei} = 10.0^{0}C$
Output Fluid Temperature of the Evaporator	$T_{eo} = 6.0^{0}C$
Input Fluid Temperature of the Condenser	$T_{ci} = 20.0^{0}C$
Output Fluid Temperature of the Condenser	$T_{co} = 10.0^{0}C$
Heat Transfer Coefficient of the Insulation Material	$k_{ins} = 0.035 \frac{kW}{m^{0}C}$
Thickness of the Insulation Material	$\Delta X_{ins} = 0.10$ m
Initial Water Temperature to the Storage Tank	$T_w = 15.0^{0}C$
Initial Room Temperature	$T_r = 25.00C$
Maximum Limit of the Comfort Temperature	$T_{max} = 26.0^{0}C$
Minimum Limit of the Comfort Temperature	$T_{min} = 22.0^{0}C$
Upper and Lower Time Limits for the Compressor Wort at Winter Time (hour of year's)	TIM1=6000 hr
	TIM2=8040 hr
Upper and Lower Temperature Limits for Water Stored in the Tank	$TEM1 = -20.0^{0}C$
	$TEM2 = -10.0^{0}C$

Table 5.2 Changed Capacities of the System Components

Number of Capacity Increament	Q_c (kW)	Q_{he} (kW)	Q_{fc} (kW)	Q_e (kW)
1	50	1.8	40	40
2	100	3.6	80	80
3	150	5.4	120	120

Table 5.3 Changed Volumes of the Thermal Energy Storage Tank

	Volume of the Storage Tank (m^3)		
		Seasonal Storage	
Number of Volume Increament	Daily Storage for Night and Day Time Active System	for Summer Time Active System	for Winter Time Active System
1	24	120	512
2	36	210	720
3	48	336	900

Periodic solutions were obtained by using Finite difference for daily storage with a sixty second time interval, and twelve days total time span. For seasonal storage, the time interval was sixty second and the time span was four years.

Results are depicted in three graphs for each cases of daily and seasonal storage. These are graphs for temperature distribution versus time, energy distribution versus time, and COP variation with the different component sizes. The capacity of each component was altered one at a time in order to investigate effects of capacity changes on the COP of the systems.

5.1 Discussion of Results for Daily Storage:

Results for daily storage (full and partial) of a hot time active system, a cold time active system and a conventional system are given in graphical form in Figures 5.1.3 to 5.1.14 . These results were obtained using inputs given in Tables 5.1, 5.2 and 5.3.

Results for daily storage were obtained with the temperature and cooling load distributions given in Figure 5.1.1. As seen from Figures 5.1.7 and 5.1.9; the heat exchanger starts to cool the water stored in the tank when the water temperature is greater than the ambient air temperature. The compressor starts functioning when the room temperature is greater than the maximum limit of the comfort temperature range. It stops when the room temperature falls below than the minimum limit.

Energy distribution graphs of all systems show the times, when the compressor of the chiller unit and heat exchanger is functional or not, and charge and discharge times of the storage tank.

Condenser temperature plotted is equal to the ambient air temperature for conventional and night time active systems, when the compressor is not functional. It is equal to the temperature of water stored in the tank for full and partial storage systems.

Room temperature plotted, is the last temperature obtained when the air conditioning is needed for all system, if the compressor is not functional. Temperatures of the evaporator and fan-coil are equal to T_{inf} which is used to convert the equations into dimensionless forms. T_{inf} is given as 15^{0}C in Table 5.1

Full storage air conditioning system with thermal energy store has a lower COP than the conventional air conditioning system as seen in Figures 5.1.5, 5.1.8, 5.1.11 and 5.1.14. This is because water stored in the tank reaches temperature higher than the ambient air temperature. This can be created by a small tank or a small heat exchanger. When the water reaches a temperature higher than the ambient temperature, the condenser of the chiller unit is cooled directly by ambient air. This is named as "partial storage". COP of the partial storage system is greater than the COP of the full storage and conventional systems. The highest COP is obtained by a night time active system. The limited working time interval at night as seen in Figure 5.1.13 is to be eight hours. It is seen that the room temperature reaches to the comfortable temperature range by the night time active system.

Effect of condenser size on the COP of conventional air conditioning system is large while effects of the evaporator size is low as shown in Figure 5.1.5

Effects of heat exchanger, condenser and fan-coil sizes on the COP of full and partial storage systems are high. Effect of the storage tank volume is negligibly small as shown in Figure 5.1.8 and 5.1.11.

It is seen in Figure 5.1.14 that, the condenser size and the evaporator size are important for the night time active system, while increasing the fan-coil size has a negative effect on the COP. Because, chilled water is circulated between the fan-coil and the storage tank.

COP of the night time active system has a greater value than COP of the partial storage system. COP of the conventional system is greater than COP of the full storage air conditioning system. But it is lower than COP of the partial storage system. COP of the full storage system becomes greater than the COP of the conventional system when capacities of the heat exchanger and storage tank are increasing.

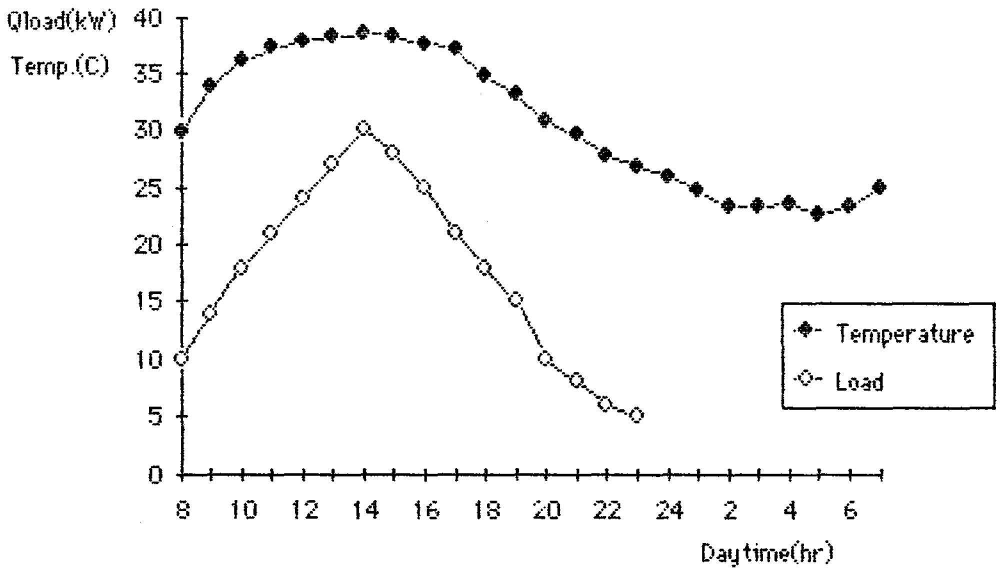


Figure 5.1.1 Daily Highest Temperature Distribution on August 2,1985 and Assumed Cooling Load Variation

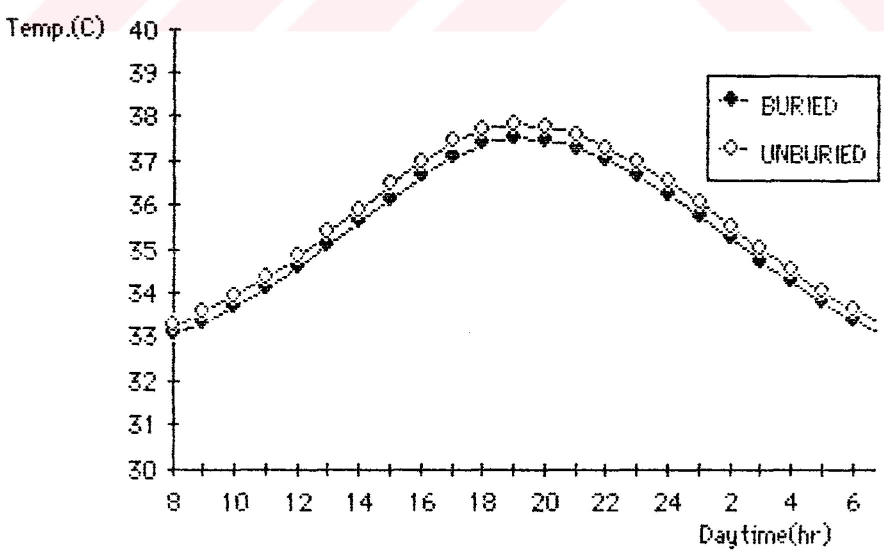


Figure 5.1.2 Temperatures Comparison Between Buried and Unburied Tank

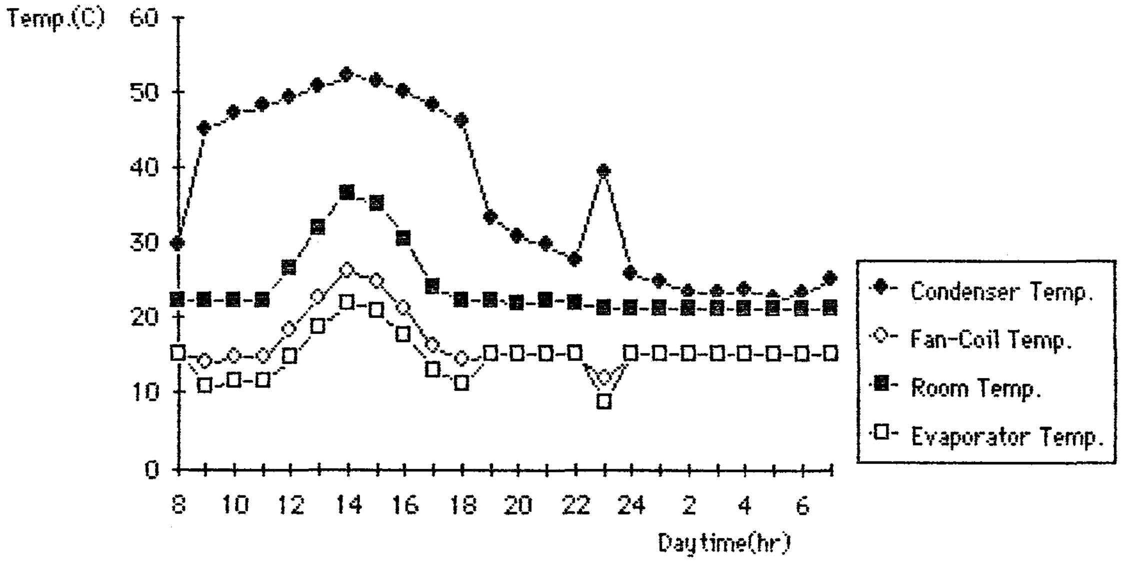


Figure 5.1.3 Temperature Distribution of a Conventional Air Conditioning System

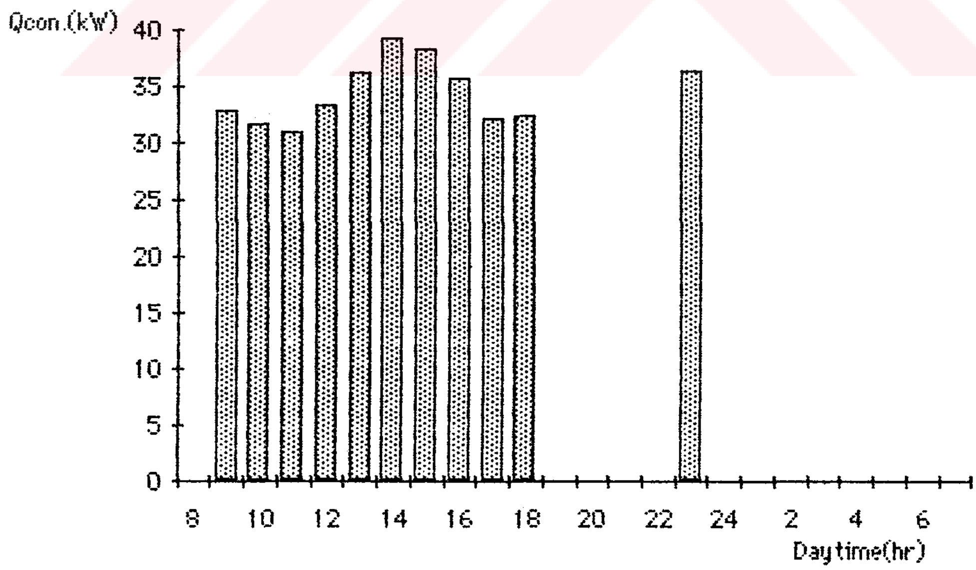


Figure 5.1.4 Energy Distribution of a Conventional Air Conditioning System

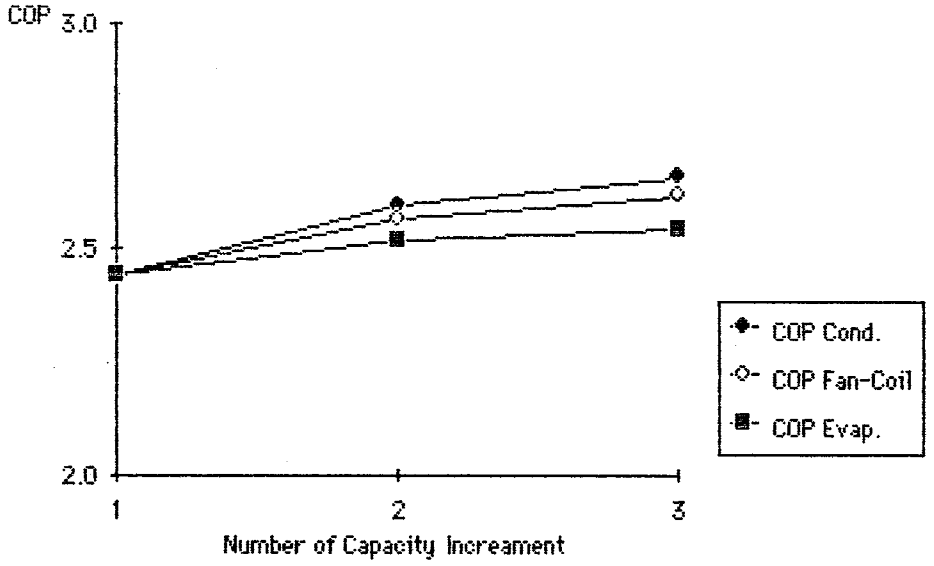


Figure 5.1.5 Components Effects of a Conventional Air Conditioning System on Its COP

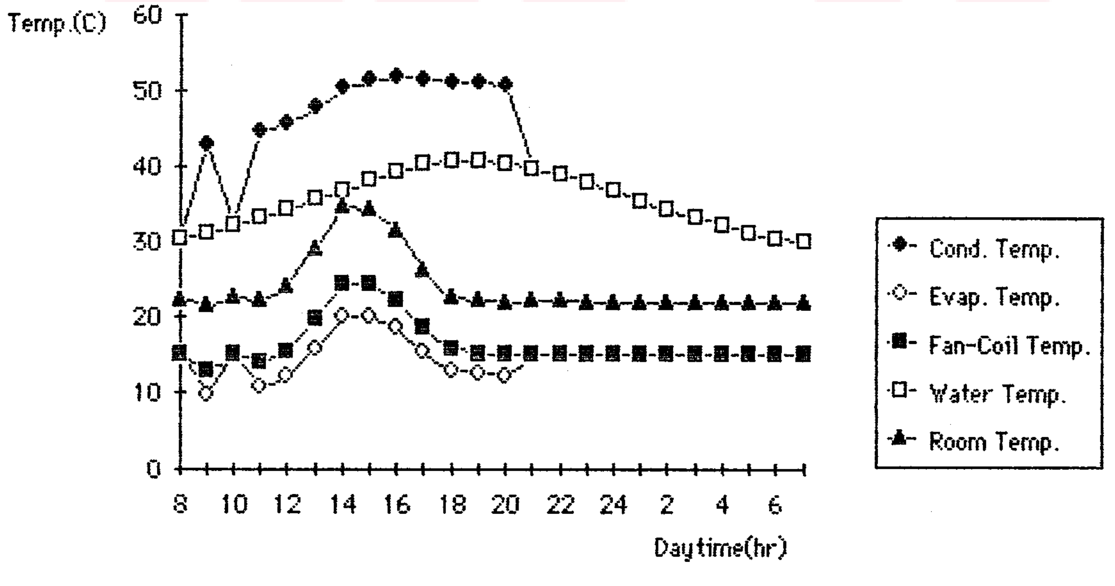


Figure 5.1.6 Temperature Distribution of an Air Conditioning System with a Thermal Energy Storage (Full Storage)

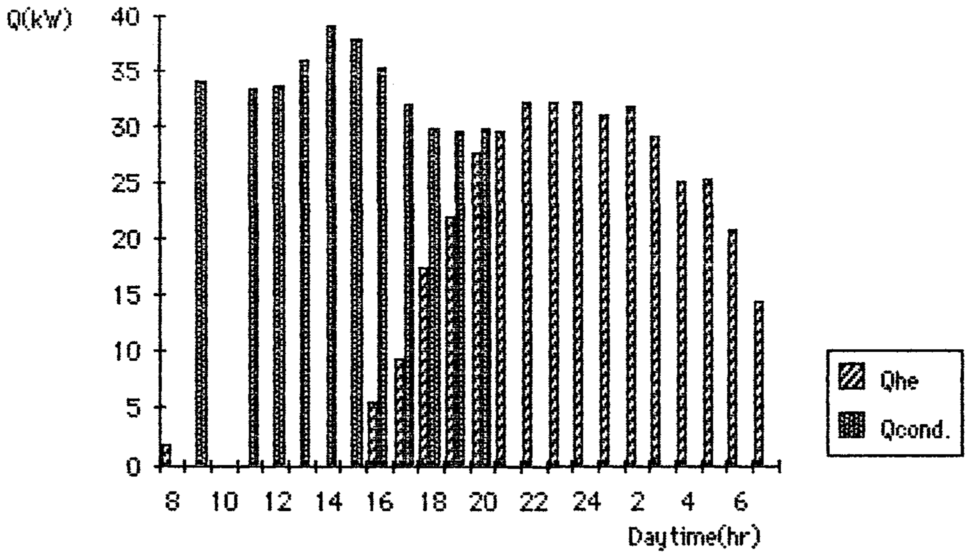


Figure 5.1.7 Energy Distribution of an Air Conditioning System with a Thermal Energy Store (Full Storage)

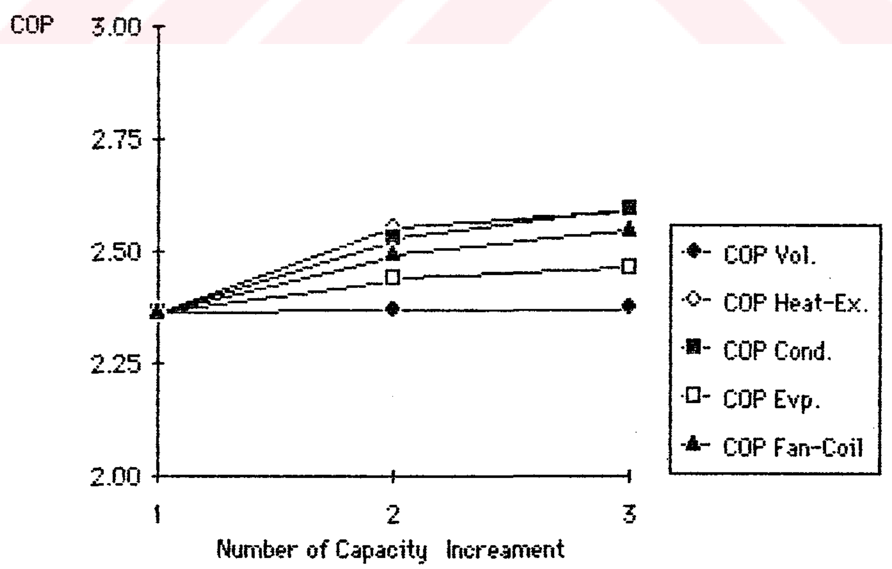


Figure 5.1.8 Components Effects of an Air Conditioning System with a Thermal Energy Store on Its COP (Full Storage)

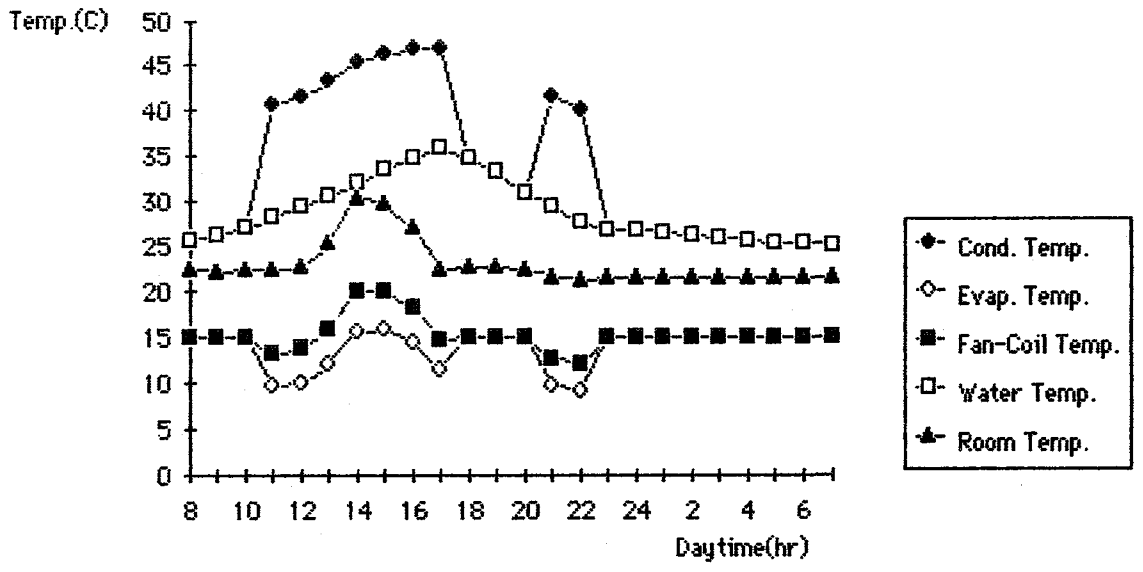


Figure 5.1.9 Temperature Distribution of an Air Conditioning System with a Thermal Energy Storage (Partial Storage)

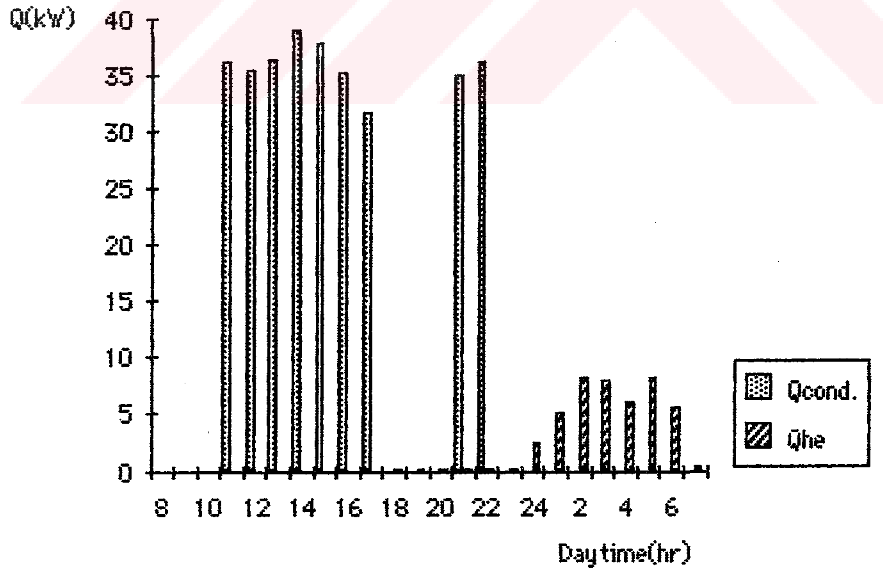


Figure 5.1.10 Energy Distribution of an Air Conditioning System with a Thermal Energy Store (Partial Storage)

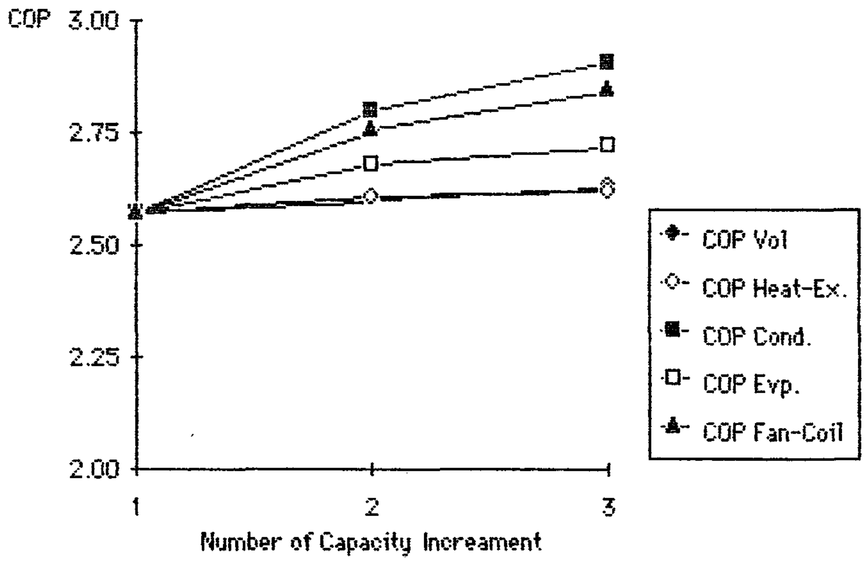


Figure 5.1.11 Components Effects of an Air Conditioning System with a Thermal Energy Store (Partial Storage) on Its COP

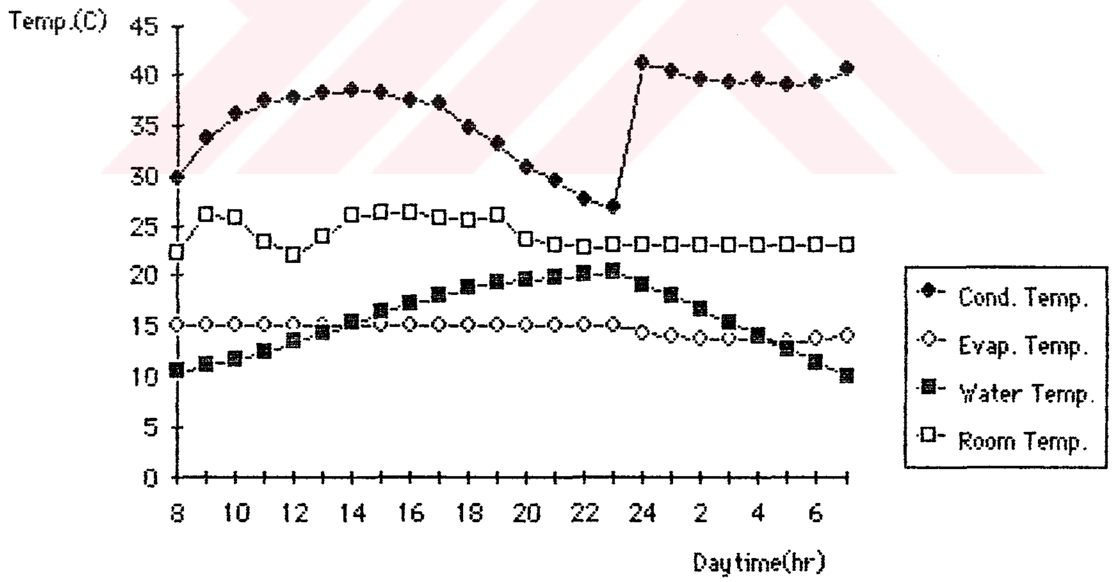


Figure 5.1.12 Temperature Distribution of an Air Conditioning System with Thermal Energy Storage (Night Time Active)

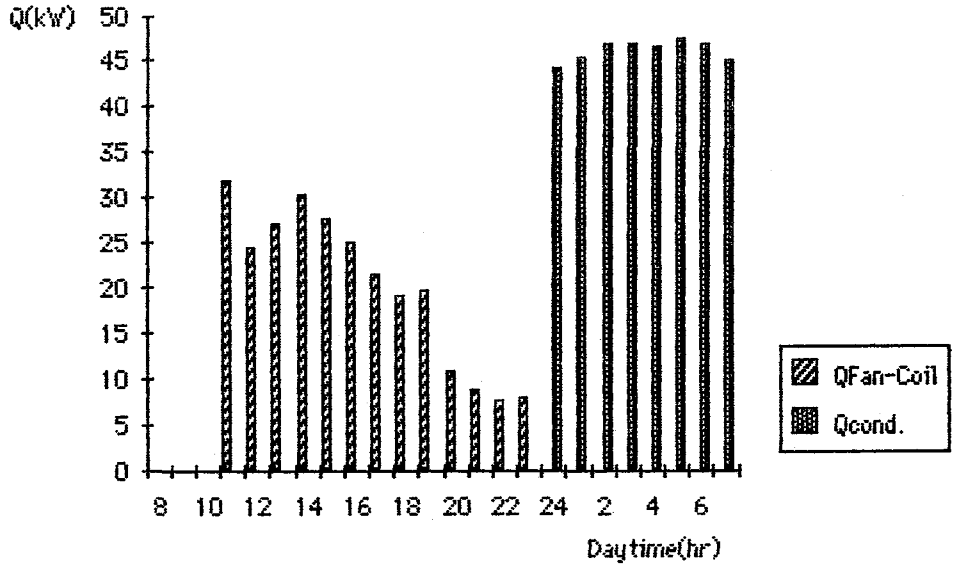


Figure 5.1.13 Energy Distribution of an Air Conditioning System with Thermal Energy Store (Night Time Active)

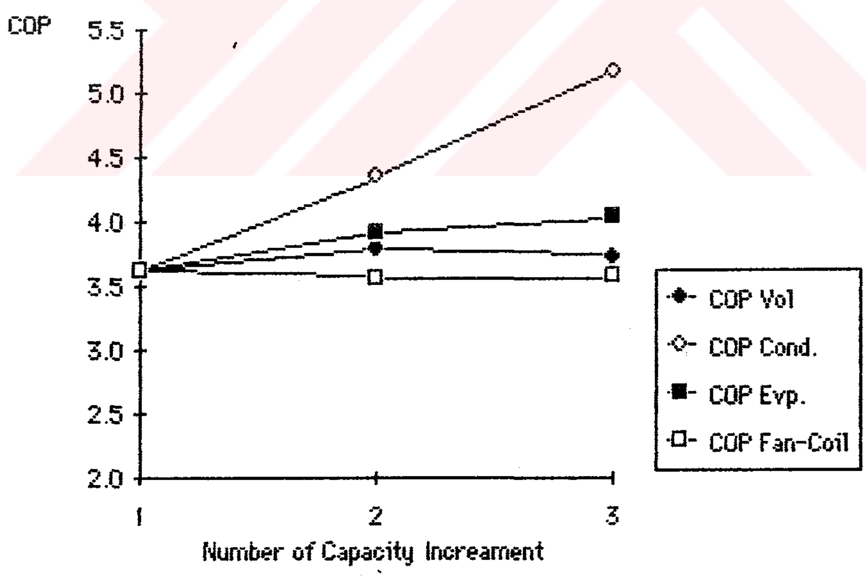


Figure 5.1.14 Components Effects of an Air Conditioning System with Thermal Energy Store on Its COP (Night Time Active)

5.2 Discussion of Results for Seasonal Storage:

Results for seasonal storage of all cases (conventional, full and partial storage of hot time active system and cold time active system) are also given in the graphical forms of monthly temperature and energy distribution and COP variation of the system with different component capacities.

Monthly cooling load variation is assumed for four months which are June, July, August and September. Because, it is not uncommon to find nonresidential buildings requiring for more than six months of the year even in the cooler region of the world [2]. It is assumed that cooling loads have no daily variation. Their hourly variation are shown in Figure 5.2.2 for four months. Figure 5.2.1 shows the yearly outside ambient air temperature for Gaziantep on 1985 [32].

The inputs used for daily storage were also used for seasonal storage, but only the capacity of the tank is increased to a large volume. Because, the effect of the tank volume on COP of the seasonal storage systems is more important than the daily storage system. The effects of the other component capacities on COP of the systems are generally same for daily and seasonal storage systems for all cases as seen in Figures 5.2.5, 5.2.8, 5.2.11 and 5.2.14. But, seasonal storage has a greater COP value than the COP of daily storage for all cases. The difference between two COP values is created by the ambient temperature change for the constant cooling load distributions. Because, ambient air has a constant and high temperature distribution during total time span of daily storage. But, hourly ambient temperatures of a year are used for seasonal storage [32].

Temperature and the energy of the system components are plotted for the four months. But, monthly water temperature distribution for one year is plotted. Because, water is cooled by an air/water type heat exchanger when the water temperature is greater than the ambient air temperature in the summer time active system. Water is cooled by the chiller unit during cold times for summer usage, with the winter time active system. Working duration of the chiller unit during winter is given in Table 5.1 and seen in Figure 5.2.12 and 5.2.13 as three months which are March, April and May. Time is started as hour from first day of July.

Monthly water temperature distributions of full and partial storage of summer time active and winter time active systems show that water freezes during winter months. To avoid freezing of water stored in the tank, 23% (by mass) salt (NaCl) is added to obtain brine with a density of 1175 kg/m^3, a specific heat of 3.33 kJ/kg^{0}C and a freezing temperature of -21.1^{0}C.

As seen in Figures 5.2.5, 5.2.8, 5.2.11 and 5.2.14 COP of the winter time active system has a greater value than COP of the partial storage. COP of the conventional system is higher than COP of the full storage and lower than COP of the partial storage.

In this study, COP of the winter time active system was found as 5.2 with a limited chilled water temperature and working time. But the COP can increase to 7 or over by changing the capacities of the active components.

COP of full and partial storage of the summer time active systems are already lower than the COP of the winter time active system.

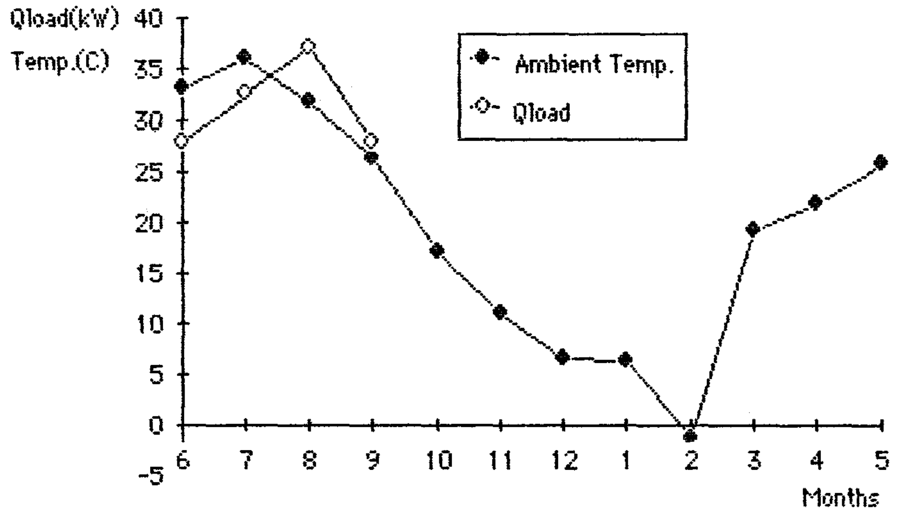


Figure 5.2.1 Yearly Temperature Distribution on 1985 and Assumed Cooling Load Variation

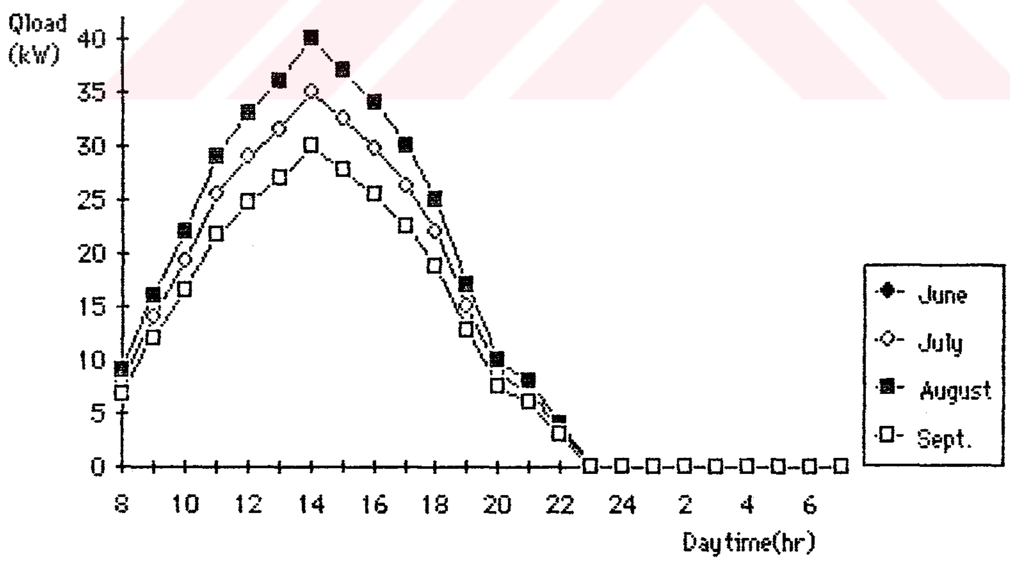


Figure 5.2.2 Monthly Load Distribution for Seasonal Storage

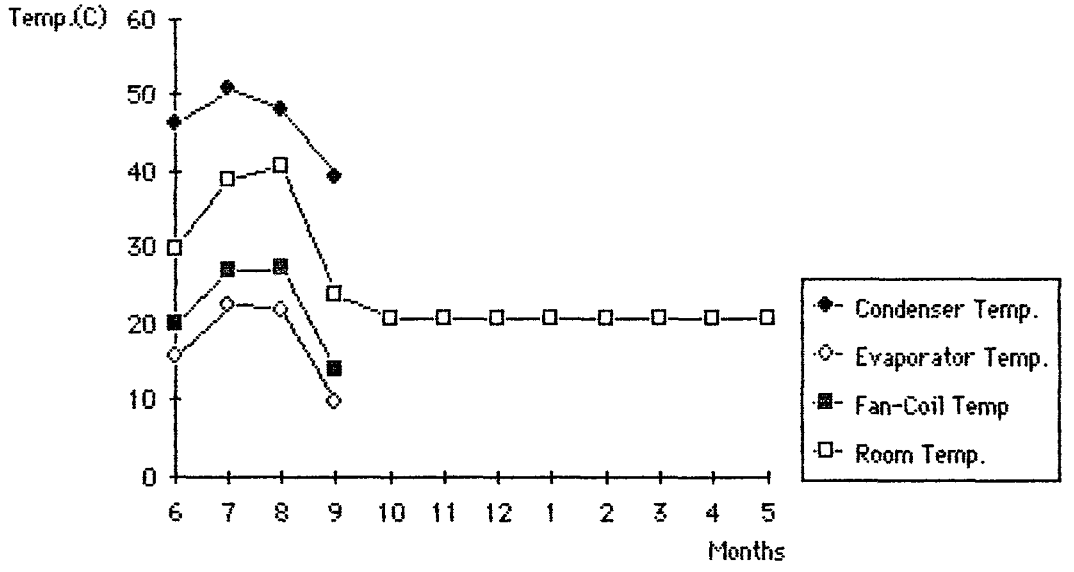


Figure 5.2.3 Temperature Distribution of a Conventional Air Conditioning System

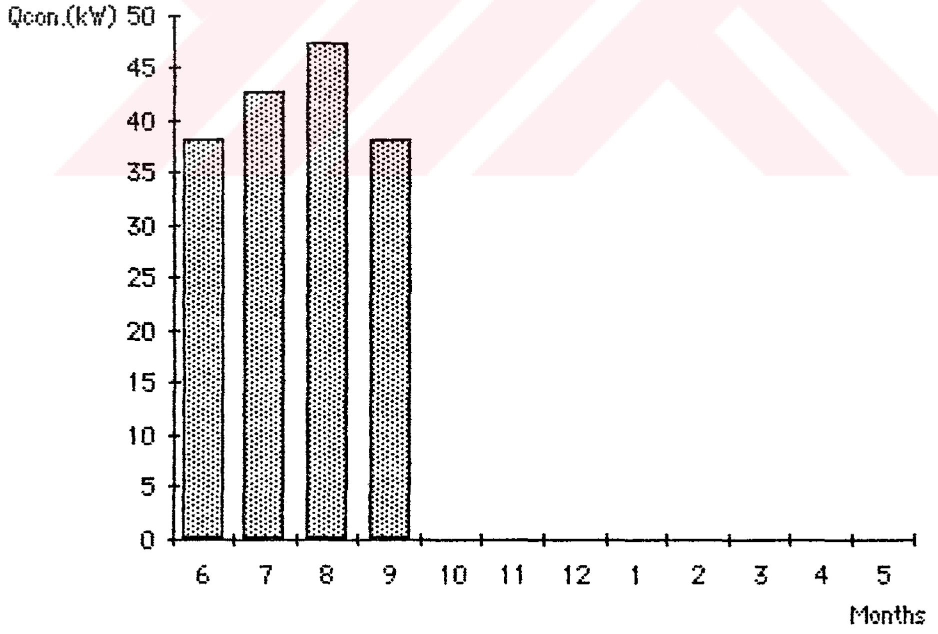


Figure 5.2.4 Energy Distribution of A Conventional Air Conditioning System

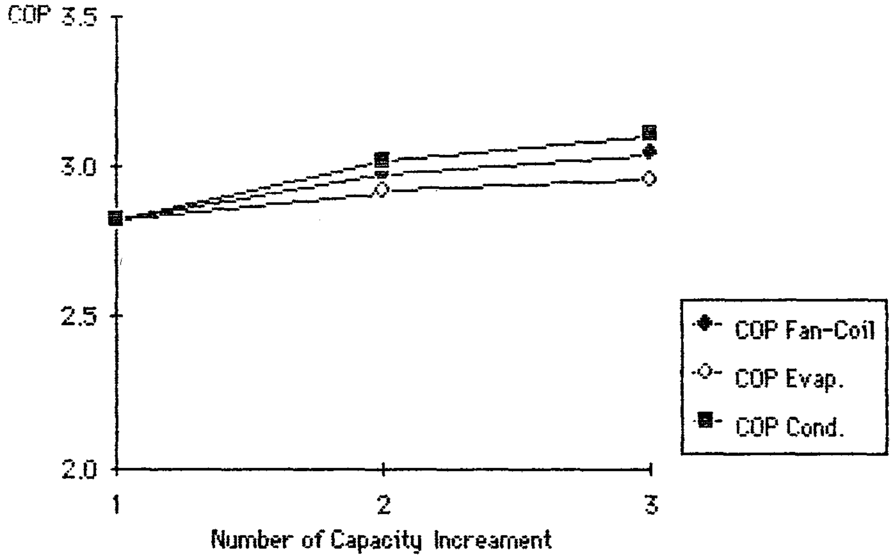


Figure 5.2.5 Components Effects of a Conventional Air Conditioning System on Its COP

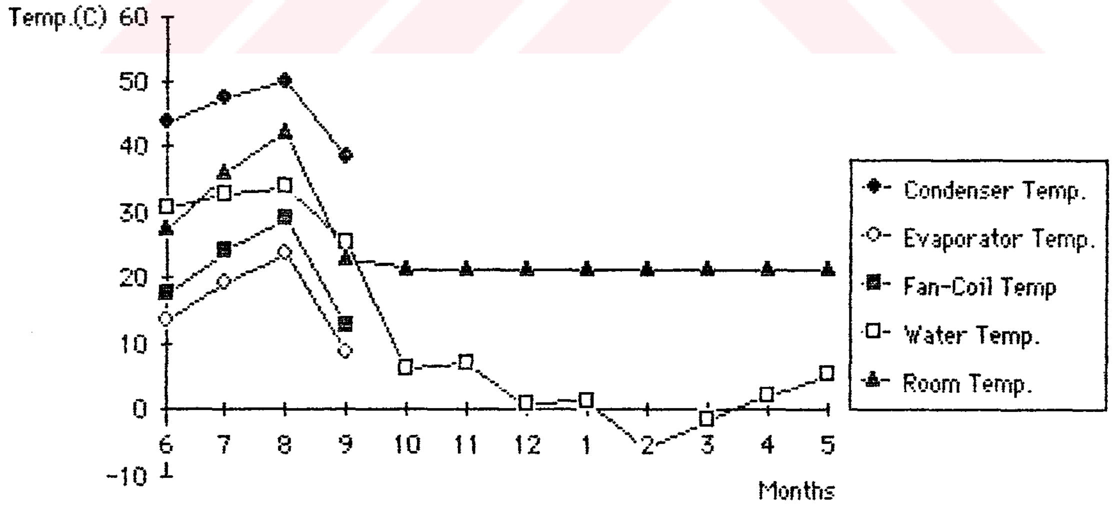


Figure 5.2.6 Temperature Distribution of an Air Conditioning System with Thermal Energy Storage (Full Storage)

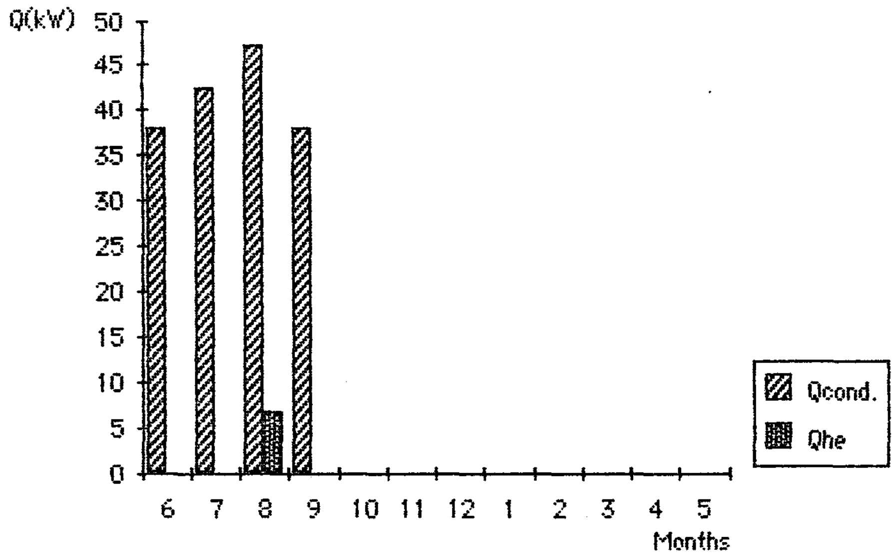


Figure 5.2.7 Energy Distribution of an Air Conditioning System with Thermal Energy Store (Full Storage)

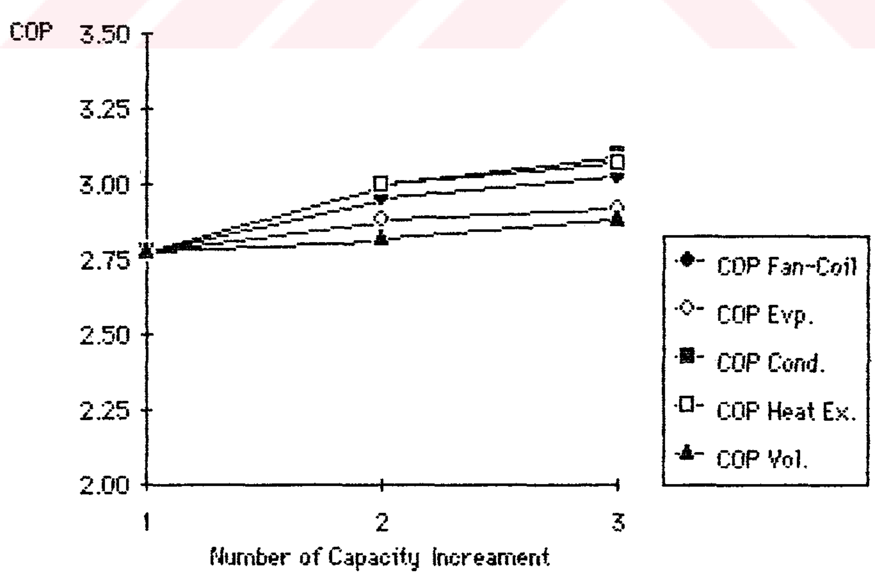


Figure 5.2.8 Components Effects of An Air Conditioning System with Thermal Energy Store on Its COP

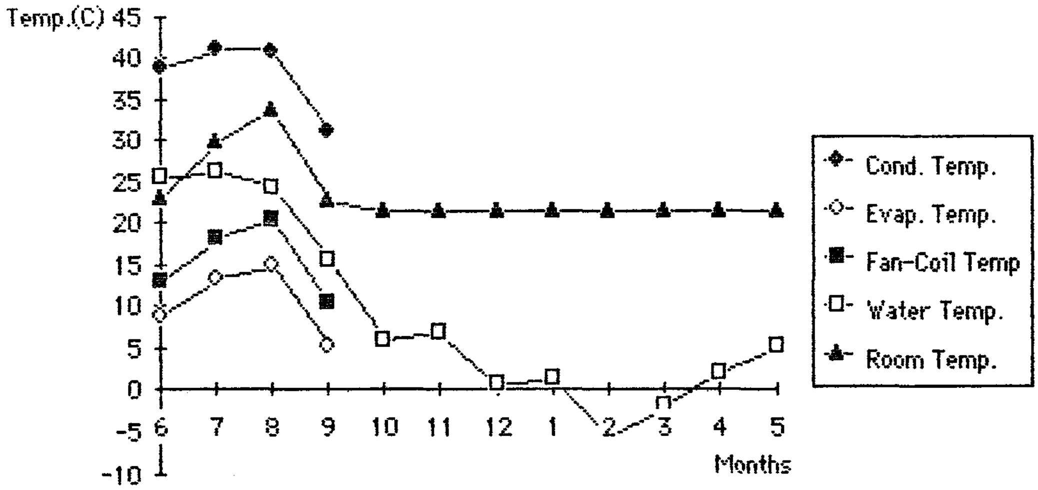


Figure 5.2.9 Temperature Distribution of an Air Conditioning System with Thermal Energy Storage (Partial Storage)

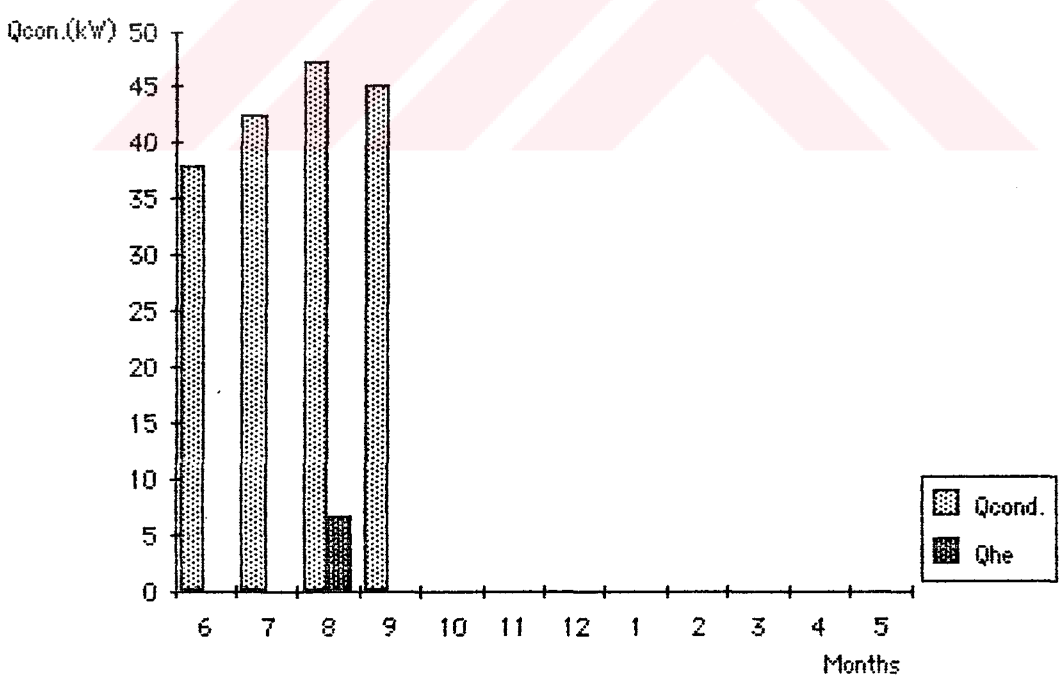


Figure 5.2.10 Energy Distribution of an Air Conditioning System with Thermal Energy Store (Partial Storage)

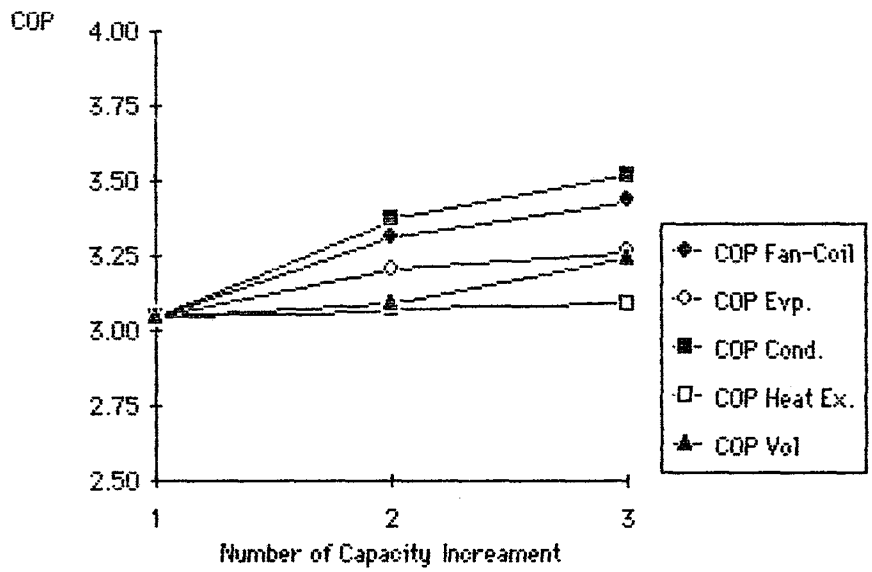


Figure 5.2.11 Components Effects of an Air Conditioning System with Seasonal Thermal Energy Store (Partial storage) on Its COP

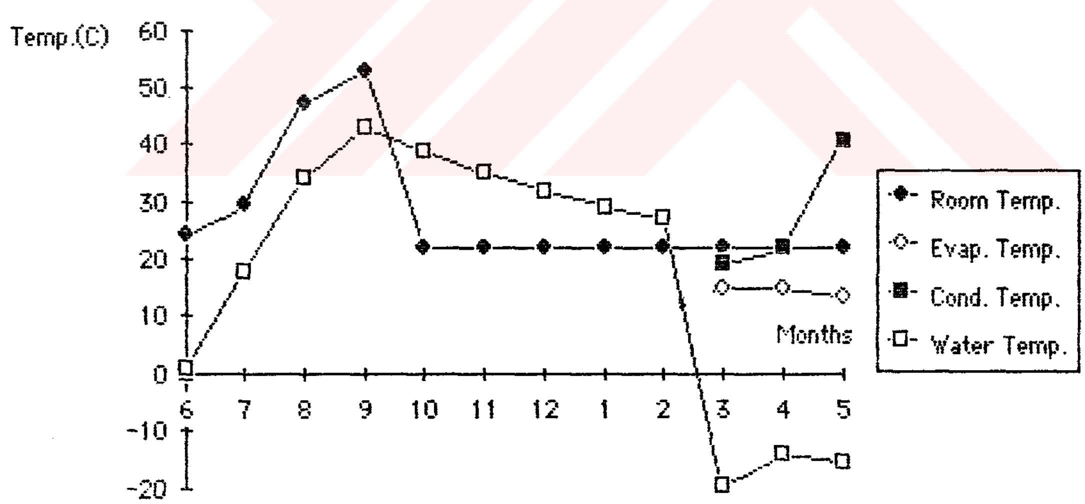


Figure 5.2.12 Temperature Distribution of an Air Conditioning System with Thermal Energy Storage (Winter Time Active)

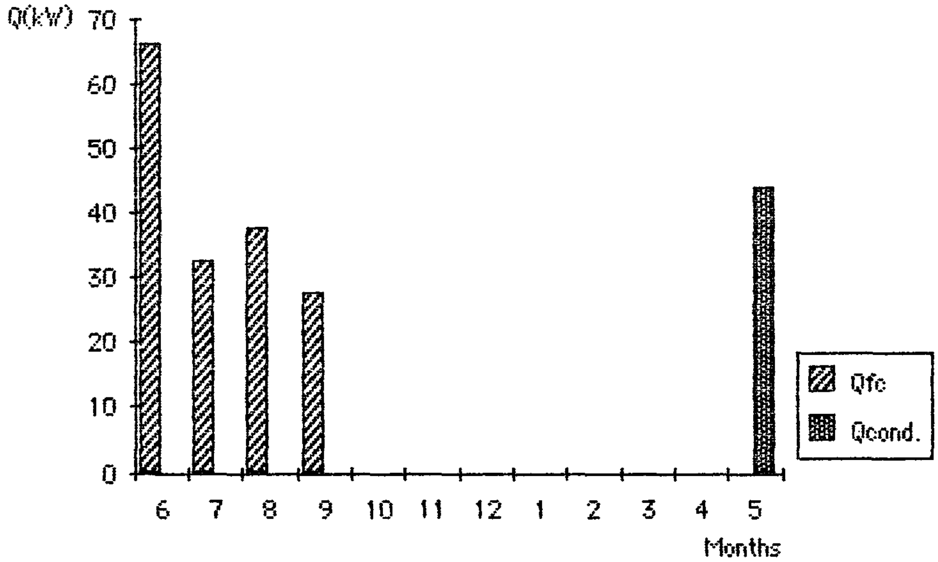


Figure 5.2.13 Energy Distribution of an Air Conditioning System with Thermal Energy Store (Winter Time Active)

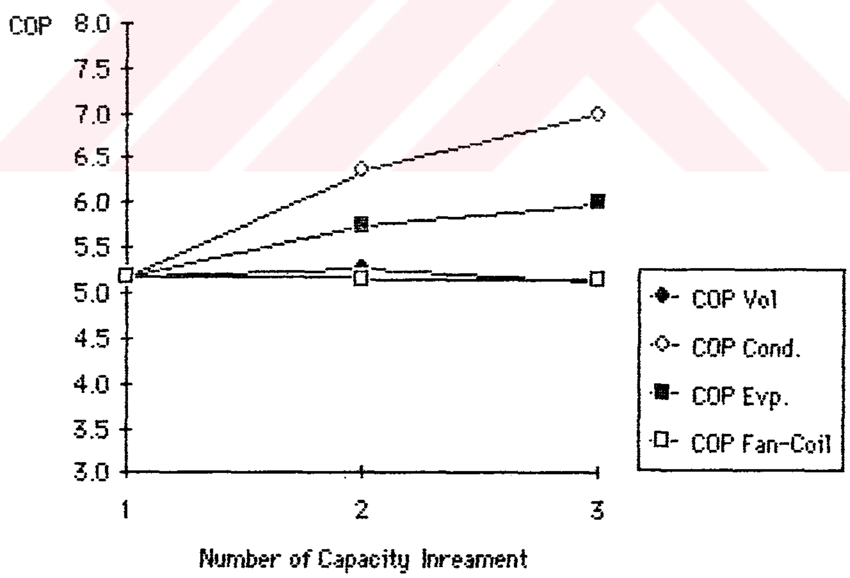


Figure 5.2.14 Components Effects of An Air Conditioning System with Thermal Energy Store on Its COP (Winter Time Active)

CHAPTER 6

CONCLUSIONS

A thermal analysis and computer simulation of a summer air conditioning system with a man-made seasonal thermal energy store are presented in this study.

Alternative cooling systems with a thermal energy store are thermally analysed for a conventional building compared with a conventional chilled water system. The conventional system was modelled by using the ambient air as a natural cold energy source.

This study includes two storage models. Each one has three working strategies. These are full and partial storage for a hot time active system, a cold time active system and a conventional air conditioning system.

This study shows that the most important component of an air conditioning system with and without a thermal energy store is the condenser of the chiller unit. Heat exchanger has a significant effect on the COP of hot time active full and partial storage systems.

Night time active system is the most suitable for daily storage. COP of a night time active system is the highest and the room temperature by this system is within the comfort temperature range.

Room temperature closest to the comfort range is obtained by the partial storage summer time active system with seasonal storage. Effect of the storage tank volume is on the COP important for seasonal storage systems than the COP of daily storage due to long storage time of cold energy.

Fan-coil is an important component than the volume and evaporator for the full and partial storage systems. Fan-coil has a negative effect on the COP of night time active systems with daily storage. But it has no effect on the COP of winter time active system with seasonal storage.

Volume of the storage tank has no significant effect on COP of an conditioning system with a thermal energy store.

When the COP of all systems are compared, the cold (winter/night) time active system is the best.

COP of air conditioning system with and without thermal energy store is a function of the condenser evaporator, heat exchanger and fan-coil sizes. COP of a cold time active system is a function of only the condenser and evaporator sizes.

CHAPTER 7

RECOMMENDATIONS FOR FURTHER STUDY

The results presented in this study show the variation of COP with the capacities of the evaporator, condenser, storage tank, heat exchanger and fan-coil.

Effects of an air conditioning system components on the COP was studied. A performance of the systems studied in this investigation should also be estimated considering a stratified storage tank and using meteorological data for several years in order to obtain long-term evaluation of performance predictions.

Calculated cooling load and temperature instead of ambient temperature for the space occupied by the storage tank should be used, if the construction or the overall heat transfer coefficient of the walls are known.

Briefly, simulation must be realized. After the realization and economic analysis of the systems, this study needs experimental verification, by retrofitting a thermal energy storage tank to a conventional air conditioning system.

REFERENCES

1. D. L. Kirkpatrick, M. Masoero, A. Rabl, C. E. Roedder, R. H. Socolow and T. B. Taylor, "The Ice Pond-Production and Seasonal Storage of Ice for Cooling", Solar Energy 35, pp. 435-445, (1985).

2. M. N. Bahadori, "Natural Production, Storage and Utilization of Ice in Deep Ponds for Summer Air Conditioning", Solar Energy 34, pp.143-149 (1985).

3. H. Akbari and A. Mertol, "Thermal Energy Storage for Cooling of Commercial Building", Energy Storage Systems, pp. 315-347 (1989).

4. E. L. Morofsky, "Developing and Innovative Building Cooling Technology", ASHRAE Transactions 21-1, pp.1741-1748 (1987).

5. A. E. McGarity, D. L. Kirkpatrick and L. K. Norford, "Design and Operation of an Ice Pond for Cooling a Large Commercial Building", ASHRAE Transactions 21-2, pp.1749-1759 (1987).

6. C. E. Francis and R. T. Tamblyn, "Annual Cycle Ice Production and Storage", ASHRAE Transactions 21-3, pp.1760-1765 (1987).

7. C. W. Sohn and J. J. Tomlinson, "Duirnal Ice Storage Cooling Systems in Army Facility", ASHRAE Transactions 19-1, pp.1079-1085 (1987).

8. N. Tran, J. F. Kreider and P. Brothers, "Field Measurement of Chilled Water Storage Performance", ASHRAE Transactions 19-4, pp.1106-1112 (1989).

9. R. K. Tackett, "Case Study; Office Building Uses Ice Storage, Heat Recovery and Cold Air Distribution", ASHRAE Transactions 19-5, pp.1113-1121 (1989).

10. D. H. Spethmann, "Optimal Control for cool Storage", ASHRAE Transactions 22-1, pp.1189-1193 (1989).

11. G. Meckler, "Cold Air distribution Option with Ice Storage", ASHRAE Transactions 22-2, pp.1194-1205 (1989).

12. B. Cordiallat and R. T. Tamblyn, "French Office Tower Pioneers with Thermal Storage", ASHRAE Transactions 25-1, pp.1861-1865 (1988).

13. D. L. Grumman and A. J. Butkus, "Ice Storage Application to an Illinois Hospital", ASHRAE Transactions 25-3, pp.1879-1893 (1988).

14. D. E. Knebel, "Economics of Harvesting Thermal Storage Systems; a Case Study of a Merchandise Distribution Center", ASHRAE Transactions 25-4, pp.1894-1904 (1988).

15. J. M. Ayres and H. Lau, "Economics of Ice Storage Systems in a University Engineering Building", ASHRAE Transactions 25-5, pp.1905-1929 (1988).

16. E. I. Mackie, "Thermal Storage in Robson Square, Vancouver, British Columbia", ASHRAE Transactions 8-1, pp. 717-723 (1980).

17. L. K. Rawlings, "Ice Storage System Optimization and Control Strategies", ASHRAE Transactions 1-2, pp.12-23 (1985).

18. M. W. Wildin and C. R. Truman, "A Summary of Experience with Stratified Chilled Water Tanks", ASHRAE Transactions 20-2, pp. 956-976 (1985).

19. W. P. McNeil and J. D. Mathey, "Review of an Operating Ice Storage System Performance", ASHRAE Transactions 20-3, pp. 977-992 (1985).

20. C. E. Francis, "The Production of Ice with Long-Term Storage", ASHRAE Transactions 20-4, pp. 993-999 (1985).

21. T. A. Gilbertson, "Ice Cools Office-Hotel COmplex", Heating/Piping/Air Conditioning, pp. 47-52 August 1989.

22. R. T. Tamblyn, "Chilled Water Storage Goes to College", <u>ASHRAE Journal</u>, pp. 39-44 July 1987.

23. R. A. Wistort, "Chilled Water Storage", <u>Heating/Piping/Air Conditioning</u>, pp. 91-100 April 1982.

24. D. P. Gatley, "Cooling Thermal Storage", <u>Heating/Piping/Air Conditioning</u>, pp. 73-83 April 1987.

25. Edison Electric Institute, "Innovative Bank Storages Ice for Savings at Livermore Facility", <u>ASHRAE Journal</u> June,1986.

26. T. W. Bardy, <u>Thermal Storage for the Merchandise Mart</u>, ASHRAE Journal, pp. 18-21 November 1986.

27. D. A. Knebel and S. Houston, "Thermal Storage Retrofit", <u>ASHRAE Journal</u>, pp. 34-43 May 1986.

28. R. H. Stamm, "Thermal Storage Systems", <u>Heating/Piping/Air Conditioning</u>, pp. 133-140 January 1982.

29. C. D. McCracken, "Refrigeration Research: Ice vs Water", <u>ASHRAE Journal</u>, pp. 60 December 1986.

30. M. Necati Özışık, <u>Heat Transfer</u>, McGraw-Hill, (1988).

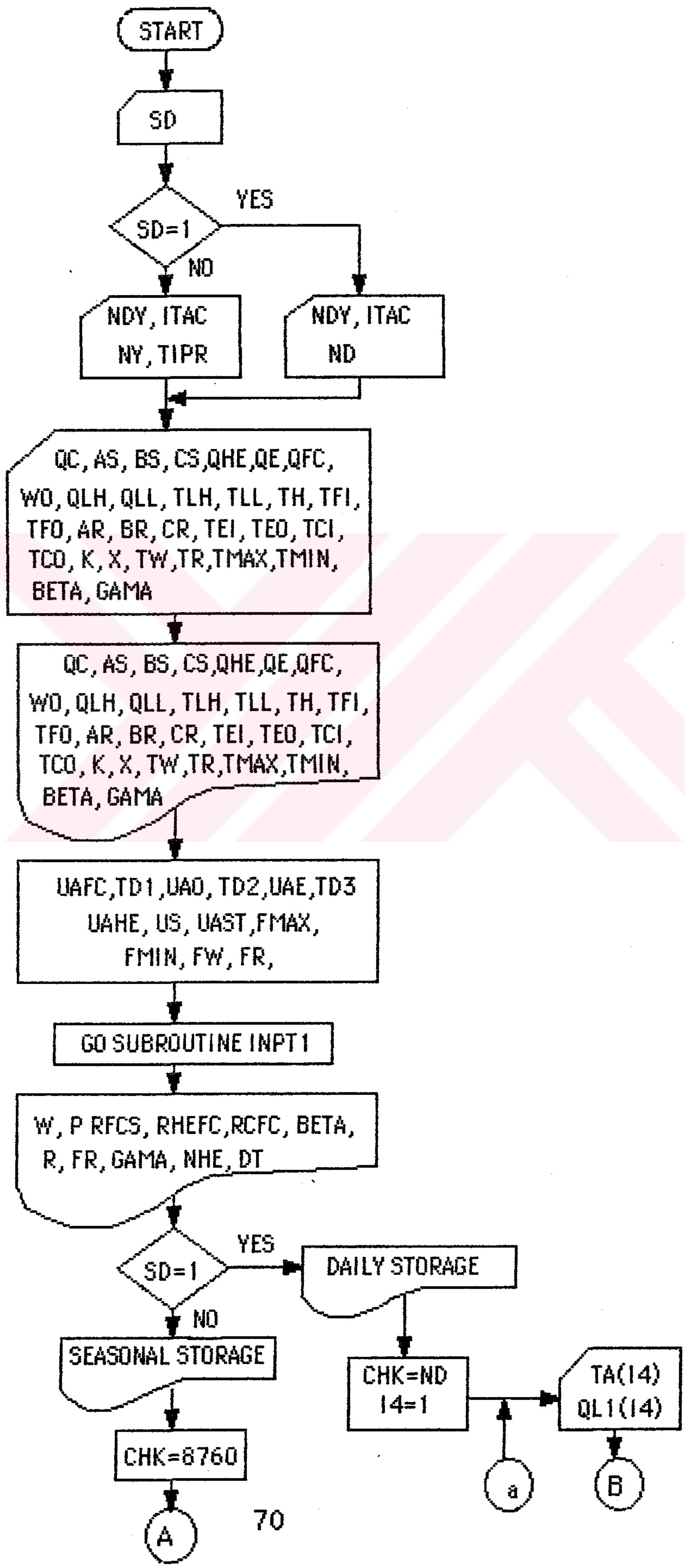
31. Air Condition Components Manufacturer's Catalogue, GENSA, (1988).

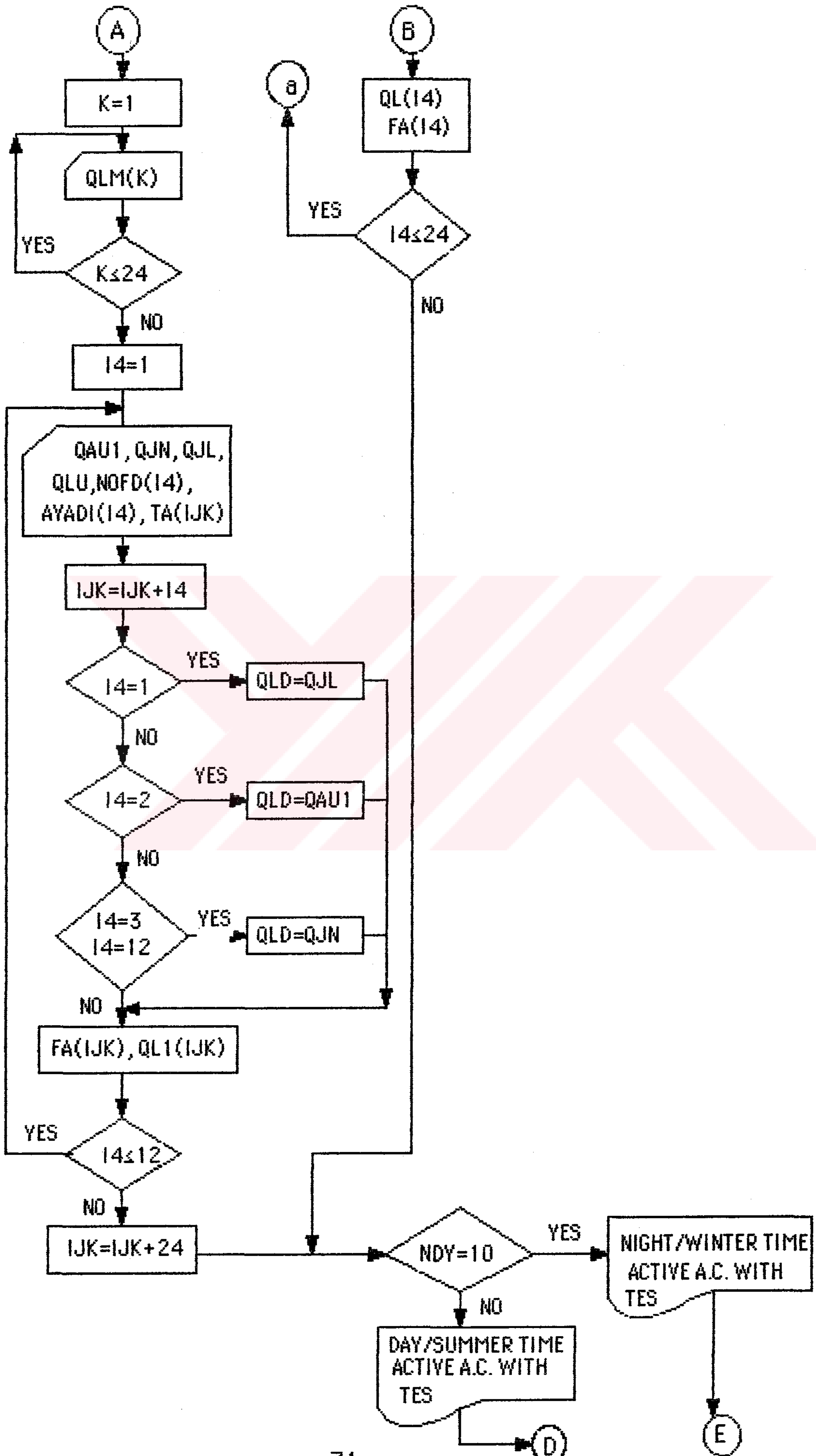
32. <u>"Saatlik Hava Sıcaklıgı (Tomograf) Tablosu"</u>, Gaziantep Meteoroloji İstasyonu, (1985).

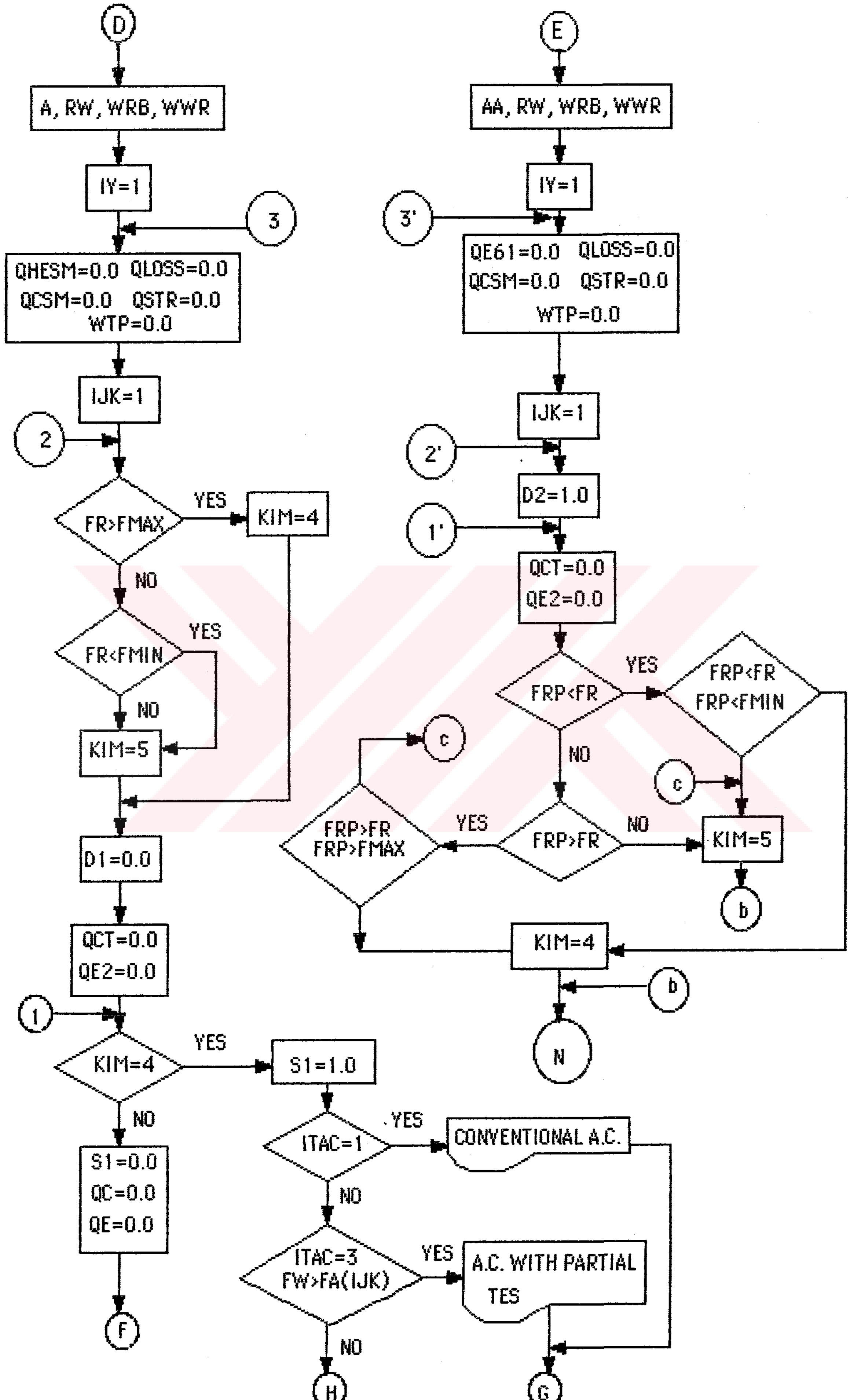
33. G. J. Van Wylen and R. E. Saintag, <u>"Fundamentals of Classical Thremodynamics"</u>, John Wiley and Sons, New York (1978)

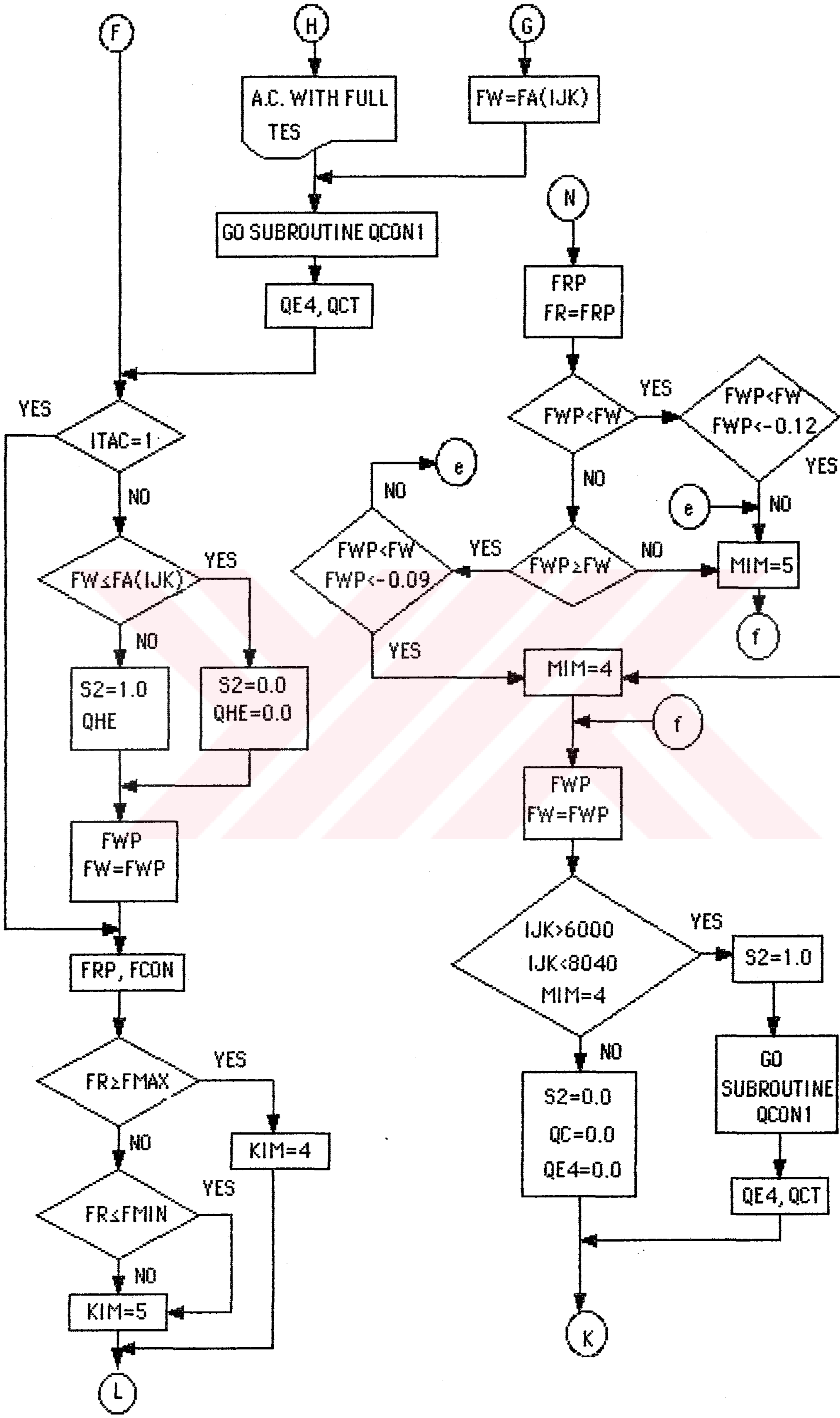
APPENDIX-1

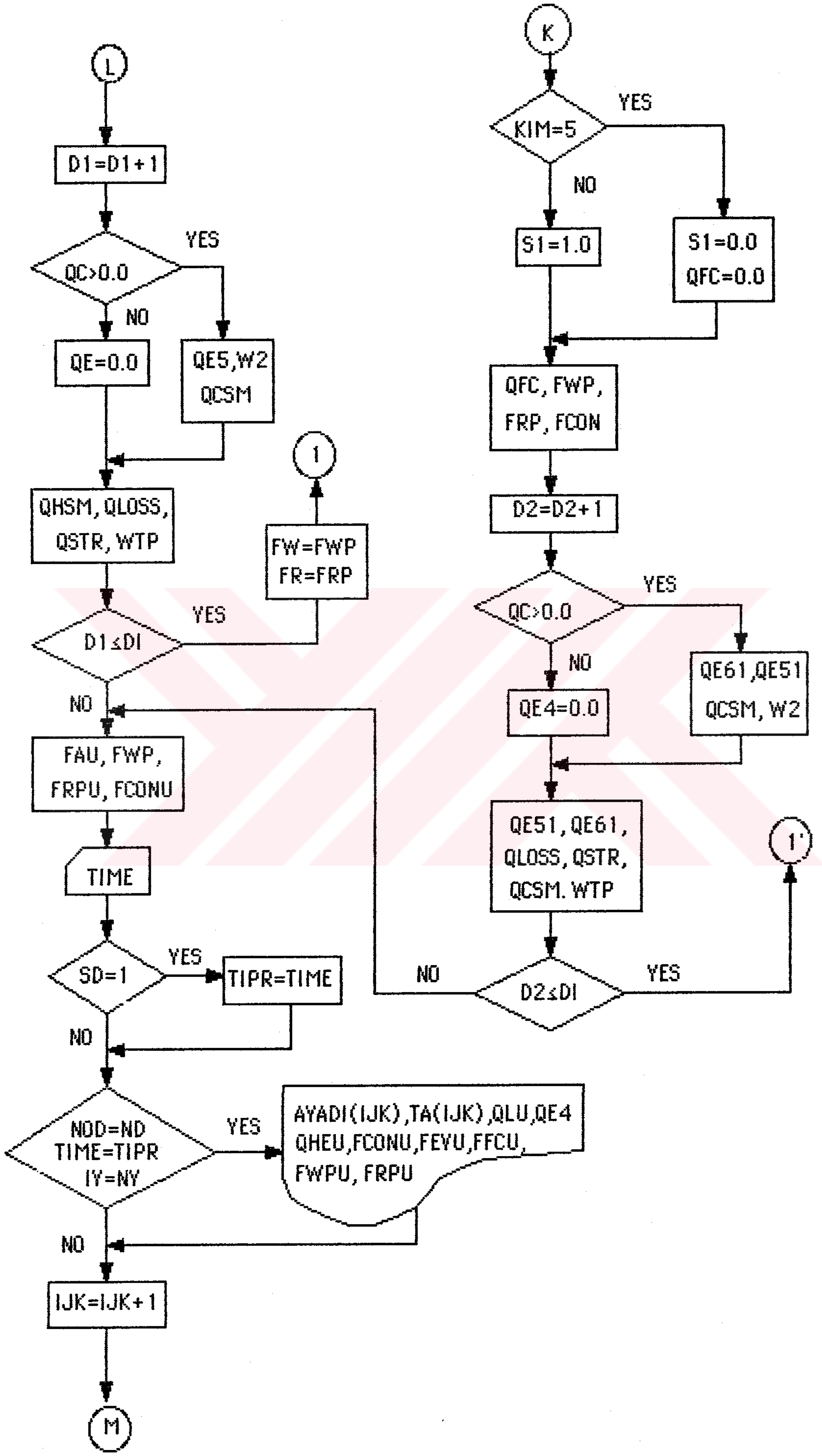
FLOWCHART FOR COLD



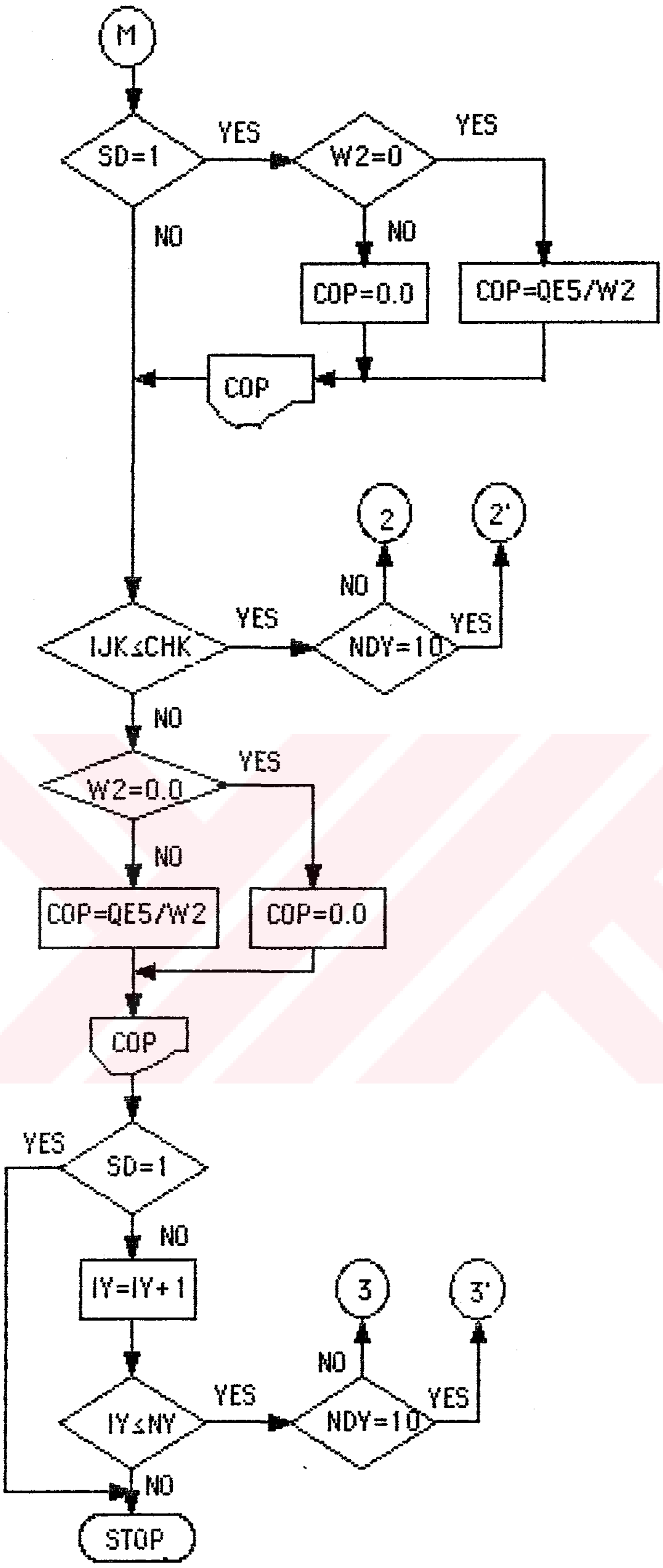

A
K=1
QLM(K)
K≤24
YES
NO
I4=1
QAU1, QJN, QJL,
QLU, NOFD(I4),
AYADI(I4), TA(IJK)
IJK=IJK+I4
I4=1
YES
QLD=QJL
NO
I4=2
YES
QLD=QAU1
NO
I4=3
I4=12
YES
QLD=QJN
NO
FA(IJK), QL1(IJK)
I4≤12
YES
NO
IJK=IJK+24
B
QL(I4)
FA(I4)
I4≤24
YES
a
NO
NDY=10
YES
NIGHT/WINTER TIME
ACTIVE A.C. WITH
TES
E
NO
DAY/SUMMER TIME
ACTIVE A.C. WITH
TES
D


D
A, RW, WRB, WWR
IY=1
3
QHESM=0.0 QLOSS=0.0
QCSM=0.0 QSTR=0.0
WTP=0.0
IJK=1
2
FR>FMAX
YES
KIM=4
NO
FR<FMIN
YES
NO
KIM=5
D1=0.0
QCT=0.0
QE2=0.0
1
KIM=4
YES
S1=1.0
NO
S1=0.0
QC=0.0
QE=0.0
F
ITAC=1
YES
CONVENTIONAL A.C.
NO
ITAC=3
FW>FA(IJK)
YES
A.C. WITH PARTIAL
TES
NO
H
G
E
AA, RW, WRB, WWR
IY=1
3'
QE61=0.0 QLOSS=0.0
QCSM=0.0 QSTR=0.0
WTP=0.0
IJK=1
2'
D2=1.0
1'
QCT=0.0
QE2=0.0
FRP<FR
YES
FRP<FR
FRP<FMIN
NO
c
c
FRP>FR
FRP>FMAX
YES
FRP>FR
NO
KIM=5
b
KIM=4
b
N

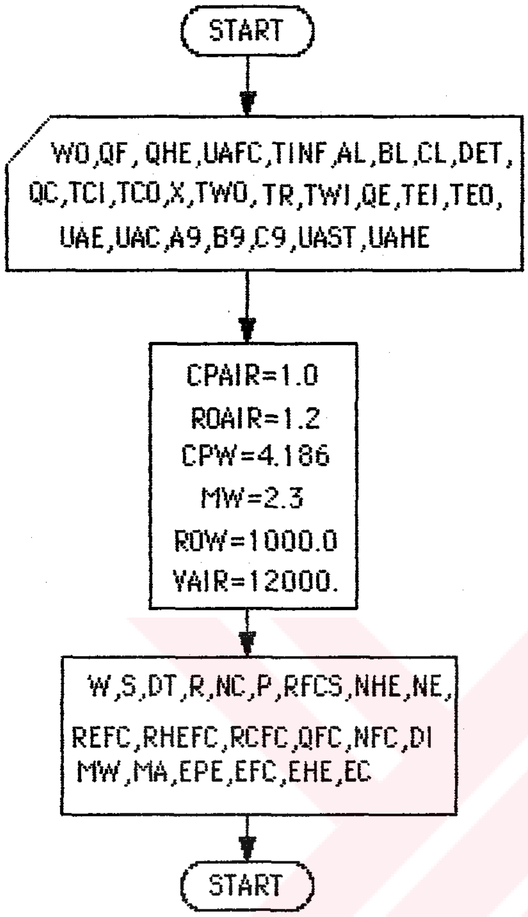

F
H
G
A.C. WITH FULL TES
FW=FA(IJK)
GO SUBROUTINE QCON1
QE4, QCT
N
FRP
FR=FRP
ITAC=1
YES
NO
FWP<FW
FWP<FW
FWP<-0.12
FW≤FA(IJK)
FWP<FW
FWP<-0.09
FWP≥FW
MIM=5
S2=1.0
QHE
S2=0.0
QHE=0.0
MIM=4
FWP
FW=FWP
IJK>6000
IJK<8040
MIM=4
S2=1.0
FRP, FCON
FR≥FMAX
KIM=4
GO SUBROUTINE QCON1
FR≤FMIN
S2=0.0
QC=0.0
QE4=0.0
QE4, QCT
KIM=5
L
K
e
f

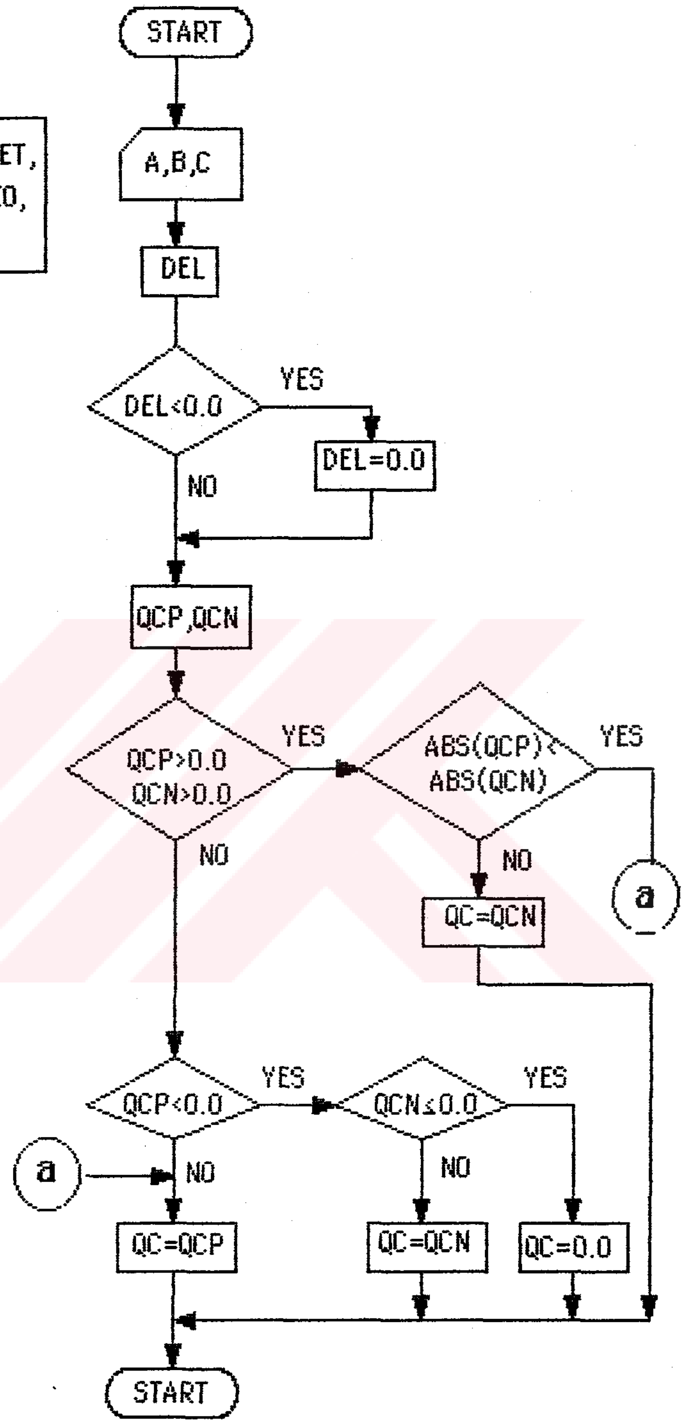


L
D1=D1+1
QC>0.0
YES
NO
QE=0.0
QE5,W2
QCSM
QHSM, QLOSS,
QSTR, WTP
1
FW=FWP
FR=FRP
D1≤DI
YES
NO
FAU, FWP,
FRPU, FCONU
TIME
SD=1
YES
TIPR=TIME
NO
NOD=ND
TIME=TIPR
IY=NY
YES
AYADI(IJK),TA(IJK),QLU,QE4
QHEU,FCONU,FEVU,FFCU,
FWPU, FRPU
NO
IJK=IJK+1
M
K
KIM=5
YES
NO
S1=1.0
S1=0.0
QFC=0.0
QFC, FWP,
FRP, FCON
D2=D2+1
QC>0.0
YES
NO
QE4=0.0
QE61,QE51
QCSM, W2
QE51, QE61,
QLOSS, QSTR,
QCSM. WTP
1'
NO
D2≤DI
YES


M
SD=1
YES
W2=0
YES
NO
NO
COP=0.0
COP=QE5/W2
COP
2
2'
NO
YES
IJK≤CHK
YES
NDY=10
NO
YES
W2=0.0
NO
COP=QE5/W2
COP=0.0
COP
YES
SD=1
NO
IY=IY+1
3
3'
NO
YES
IY≤NY
YES
NDY=10
NO
STOP

FLOWCHARTS FOR SUBPROGRAMS OF INPT1 AND QCON1





P R O G R A M : 'C O L D'

```
|--------------------------------------------------------------------------|
|                                                                          |
|     DAILY AND SEASONAL SIMULATION OF LONG TERM PERFORMANCE               |
|                                                                          |
|       OF A SUMMER AIR-CONDITIONING SYSTEM WITH A MAN-MADE                |
|                                                                          |
|                 SEASONAL THERMAL ENERGY STORE                            |
|                                                                          |
|                 PREPARED BY: MURTAZA YILDIRIM                            |
|                                                                          |
|--------------------------------------------------------------------------|

      DIMENSION CP(50),CP1(50),QL1(8760),TA(8760),FA(8760),NOFD(12),
     &QLM(25),AYADI(12)
      REAL NC,NHE,NE,MFC,MW,MA,MCPR,MCPA,MCPW
      CHARACTER*6 AYADI

      WRITE*,' ENTER 1 FOR DAILY STORAGE'
      WRITE*,' ENTER 2 FOR SEASONAL STORAGE'
      READ*, SD
      IF(SD.EQ.1.) GO TO 1001

      MT=2
1002  WRITE(6,152)
 152  FORMAT(' IF COMPRESSOR WORKS ON SUMMER TIME ENTER 5'/
     %' IF COMPRESSOR WORKS ON WINTER TIME ENTER 10'/
     %' WHAT IS THE TYPE OF AIR CONDITIONING SYSTEM AS'/
     %'   FOR AIR-CONDITIONING SYSTEM WITH NO STORAGE ',5X,
     %'ENTER --> 1 WITH 5'/'   FOR AIR-CONDITIONING SYSTEM WITH STORAGE'
     %,9X,'ENTER --> 2'/'   FOR AIR-CONDITIONING SYSTEM WITH PARTIAL',
     %' STORAGE ENTER - > 3'//' ENTER THE NUMBER OF YEAR'/
     %' ENTER THE PRINT TIME FROM 1 TO 24')
      READ(5,*) NDY,ITAC,NY,TIPR
1003  WRITE(6,*) ' ENTER THE WORK OF COMPRESSOR IN KW W=?'
      WRITE(6,*) ' ENTER THE HIGHEST COOLING CAPACITY OF COMPRESSOR'
      WRITE(6,*) '          AS QLH IN KW QLH=?'
      WRITE(6,*) ' ENTER THE LOWEST COOLING CAPACITY OF COMPRESSOR'
      WRITE(6,*) '          AS QLL IN KW QLL=?'
      WRITE(6,*) ' ENTER THE HIGHEST COOLING TEMPERATURE OF COMPRESSOR'
      WRITE(6,*) '          AS TL1 IN KELVIN TL1=?'
      WRITE(6,*) ' ENTER THE LOWEST COOLING TEMPERATURE OF COMPRESSOR'
      WRITE(6,*) '          AS TL2 IN KELVIN TL2=?'
      WRITE(6,*) ' ENTER THE CONDENSING TEMPERATURE OF CHILLER UNIT'
      WRITE(6,*) '          AS TH IN KELVIN TH=?'
      WRITE(6,*) ' ENTER THE COOLING CAPACITY OF FAN-COILIN KW QF=?'
      WRITE(6,*) ' ENTER THE INPUT AND OUTPUT TEMPERATURE OF FAN-COIL'
      WRITE(6,*) '          IN KELVIN TFI=?, AND TFO=?'
      WRITE(6,*) ' ENTER THE TEMPERATURE OF SURRAUNDING IN K TINF=?'
      WRITE(6,*) ' ENTER THE DIMENSIONS OF AIR-CONDITIONED SPACE '
```

```
      WRITE(6,*) ' ENTER THE DIMENSIONS OF AIR-CONDITIONED SPACE '
      WRITE(6,*) '        AS AR=? (IN M.), BR=? (IN M.), CR=? (IN M.)'
      WRITE(6,*) ' ENTER THE TIME INCREAMENT IN SECOND DET=?'
      WRITE(6,*) ' ENTER THE OUTLET WATER TEMPERATURE IN C TWO=?'
      WRITE(6,*) ' ENTER THE ROOM TEMPERATURE IN C TR=?'
      WRITE(6,*) ' ENTER THE INLET WATER TEMPERATURE IN C TWI=?'
      WRITE(6,*) ' ENTER THE COOLING CAPACITY OF EVAPORATOR IN KW QE=?'
      WRITE(6,*) ' ENTER THE INPUT AND OUTPUT TEMPERATURE OF EVAPORATOR'
      WRITE(6,*) '        IN KELVIN TEI=?, AND TEO=?'
      WRITE(6,*) ' ENTER THE INPUT AND OUTPUT TEMPERATURE OF COMPRESSOR'
      WRITE(6,*) '        IN KELVIN TCI=?, AND TCO=?'
      WRITE(6,*) ' ENTER HEAT TRANS. COEFF. AND THICKNESS OF INSULATION'
      WRITE(6,*) '        FOR STORAGE TANK IN KW/M*C K=? AND IN M. X=?'
      WRITE(6,*) ' ENTER THE MAXIMUM  AND MINIMUM CONTROL TEMPERATURE'
      WRITE(6,*) '        OF THE AIR-CONDITIONED SPACE IN CELSIUS AS'
      WRITE(6,*) '        TMA AND TMI=?'
      WRITE(6,*) ' ENTER THE COOLING CAPACITY OF CONDENSER IN KW QC=?'
      WRITE(6,*) ' ENTER THE DIMENSIONS OF STORAGE TANK '
      WRITE(6,*) '        AS AS=? (IN M.), BS=? (IN M.), CS=? (IN M.)'
      WRITE(6,*) ' ENTER THE COOLING CAPACITY OF HEAT EXCH. IN KW QHE=?'

136   READ(3,135,END=134) QC7,A9,B9,C9,QHE7,QF,QE,QD,TDO,TDI
135   FORMAT(10F5.1)
      VOL=A9*B9*C9
      IF(MT.EQ.1) GO TO 137
      WRITE(7,133)
133   FORMAT(/22X,'THE CHANGED INPUTS ARE:'/21X,25('-'))
      WRITE(7,124) QC7,A9,B9,C9,VOL,QHE7,QF,QE
124   FORMAT(/'  THE COOLING CAPACITY OF CONDENSER',12X,' QC =',F7.3,
     %' KW'/'  THE DIMENSIONS OF STORAGE TANK AS',13X,'AS =',F7.3,
     %' M.'/48X,'BS =',F7.3,' M.'/48X,'CS =',F7.3,' M.'/
     %'       VOLUME OF THE STORAGE TANK IS,',12X,'VOL=',F7.3,' M**3'
     %/'  THE COOLING CAPACITY OF HEAT EXCHANGER',8X,'QHE=',F7.3,' KW'
     %/'  THE COOLING CAPACITY OF FAN-COIL',14X,'QFC=',F7.3,' KW'
     %/'  THE COOLING CAPACITY OF EVEPORATOR',12X,'QE =',F7.3,' KW'/)
137   REWIND 2
      READ(2,*) WO,WR,QLH,QLL,TL1,TL2,TH

      IF(MT.GT.1) GO TO 140
      WRITE(7,80)
 80   FORMAT(//12X,'* + * + * INPUT FOR THE SIMULATION * + * + *'/
     %21X,26('=')/)
      WRITE(7,71) WO,QLH,QLL,TL1,TL2,TH
 71   FORMAT('  THE WORK OF COMPRESSOR',23X,' W  =',F7.3,' KW'/
     %'  THE HIGHEST COOLING CAPACITY OF COMPRESSOR',7X,' QLH=',
     %F7.3,' KW'/'  THE LOWEST COOLING CAPACITY OF COMPRESSOR ',7X,
     %' QLL=',F7.3,' KW'/'  THE HIGHEST COOLING TEMPERATURE OF',
     %' COMPRESSOR TL1=',F7.3,' K.'/
     %'  THE LOWEST COOLING TEMPERATURE OF COMPRESSOR',
     %'  TL2=',F7.3,' K.'//
     %'  THE CONDENSING TEMPERATURE OF CHILLER UNIT',
     %'     TH =',F7.3,' K.')
140   READ(2,*) TFO,TFI,TINF,AL,BL,CL,DET,TWO
```

```
140 READ(2,*) TFO,TFI,TINF,AL,BL,CL,DET,TWO
    IF(MT.GT.1) GO TO 141
    WRITE(7,72) QF,TFO,TFI,TINF,AL,BL,CL,DET,TWO
 72 FORMAT('  THE COOLING CAPACITY OF FAN COIL',14X,'QF =',F7.3,' KW'
   %/'  THE INPUT AND OUTPUT TEMPERATURE OF FAN-COIL',
   %'  TFI=',F7.3,' C.'/48X,'TFO=',F7.3,' C.'//
   %'  THE TEMPERATURE OF SURRAJNDING',15X,'TINF=',F7.3,' K.'/
   %'  THE DIMENSIONS OF AIR-CONDITIONED SPACE AS',4X,'AR =',F7.3,
   %' M.'/48X,'BR =',F7.3,' M.'/48X,'CR =',F7.3,' M.'/
   %'  THE TIME INCREAMENT',26X,' DET=',F7.2,' SECOND'//
   %'  THE OUTLET WATER TEMPERATURE',17X,' TWO=',F7.3,' C.')
141 READ(2,*) TR,TWI,TEI,TEO
    IF(MT.GT.1) GO TO 142
    WRITE(7,73) TR,TWI,QE,TEI,TEO
 73 FORMAT('  THE ROOM TEMPERATURE',25X,' TR =',F7.3,' C.'/
   %'  THE INLET WATER TEMPERATURE',18X,' TWI=',F7.3,' C.'/
   %'  THE COOLING CAPACITY OF EVAPORATOR',12X,'QE =',F7.3,' KW'/
   %'  THE INPUT & OUTPUT TEMPERATURE OF EVAPORATOR',
   %'  TEI=',F7.3,' C.'/48X,'TEO=',F7.3,' C.')
142 READ(2,*) TCI,TCO,CK,XX,TFW,TFR,TMA,TMI,ND
    IF(MT.GT.1) GO TO 143
    WRITE(7,74) QC7,TCI,TCO,A9,B9,C9,VOL,CK,XX,QHE7
 74 FORMAT(/'  THE COOLING CAPACITY OF CONDENSER',12X,' QC =',F7.3,
   %' KW'/'  THE INPUT & OUTPUT TEMPERATURE OF EVAPORATOR',
   %'  TCI=',F7.3,' C.'/48X,'TCO=',F7.3,' C.'/
   %'  THE DIMENSIONS OF STORAGE TANK AS',13X,'AS =',F7.3,' M.'/
   %48X,'BS =',F7.3,' M.'/48X,'CS =',F7.3,' M.'/
   %'        VOLUME OF THE STORAGE TANK IS,',12X,'VOL=',F7.3,' M**3'/
   %/'  HEAT TRANS. COEFF. AND THICKNESS OF INSULATION FOR THE',
   %' STORAGE TANK'/48X,'K  =',F7.4,' KW/M*C'/48X,'X  =',F7.3,' M.'/
   %/'  THE COOLING CAPACITY OF HEAT EXCHANGER',8X,'QHE=',F7.3,' KW')
    WRITE(7,75) TFW,TFR,TMA,TMI
 75 FORMAT(/'  THE INPUT WATER TEMPERATURE TO THE STORAGE TANK ',
   %'TW=',F7.3,' C.'/
   %'  THE INITIAL ROOM TEMPERATURE',18X,'TFR=',F7.3,' C.'/
   %'  THE MAXIMUM AND MINIMUM CONTROL TEMPERATURE OF THE AIR-',
   %'CONDITIONED'/41X,'SPACE TMAX=',F7.3,' C.'/
   %47X,'TMIN=',F7.3,' C.'//)

143 UAFC=QF/(TFO-TFI)
    TD1=(TCO-TCI)/ALOG(TCO/TCI)
    UAC=QC7/TD1
    TD2=(TEO-TEI)/ALOG(TEO/TEI)
    UAE=QE/TD2
    TD3=1.0
    UAHE=QHE7/TD3
    U=CK/XX
    UAST=U*2.0*(A9*B9+A9*C9+B9*C9)/1000.0
    GAMA=1.04
    BETA=0.183
    FMAX=(TMA+273.15-TINF)/TINF
    FMIN=(TMI+273.15-TINF)/TINF
    FW=(TFW+273.15-TINF)/TINF
```

```
      FW=(TFW+273.15-TINF)/TINF
      FR=(TFR+273.15-TINF)/TINF
      FRP=FR
      FWP=FW
      X=1.
      CALL INPT1(WO,QF,QHE7,UAFC,TINF,AL,BL,CL,DET,QC7,TCI,TCO
     %,X,TWO,TR,TWI,QE,TEI,TEO,
     %UAE,UAC,A9,B9,C9,UAST,UAHE,W,S,DT,R,NC,P,RFCS,NHE,
     %NE,REFC,RHEFC,RCFC,QFO,NFC,DI,MW,MA,EPE,EFC,EHE,EC)
C
      WRITE(7,85) W,S,NC,P,RFCS,RHEFC,RCFC,FW,BETA,R,FR,GAMA,NHE,DT
   85 FORMAT(14X,'DIMENSIONLESS INPUTS FOR CALCULATIONS:'/
     *13X,40('-')/8X,'    W      =',F8.5,'   S     =',F8.5/
     *8X,'    NC     =',F8.5,'   P     =',F8.5,'   RFCS=',F8.5/
     *8X,'    RHEFC=',F8.5,'   RCFC=',F8.5,'   FW    =',F8.5/
     *8X,'    BETA =',F8.5,'   R     =',F8.5,'   FR    =',F8.5/
     *8X,'    GAMA =',F8.5,'   NHE =',F8.5,'   DT    =',F8.5/)
      IF(SD.EQ.1.) GO TO 1004
      REWIND 1
      REWIND 4
      REWIND 8
      IF(ITAC.EQ.1) GO TO 153
      IF(ITAC.EQ.2) GO TO 154
      WRITE(7,155)
  155 FORMAT(10X,'THE TYPE OF THE AC SYSTEM IS PARTIAL STORAGE')
      GO TO 158
  153 WRITE(7,156)
  156 FORMAT(10X,'THE TYPE OF THE AC SYSTEM IS NO STORAGE ')
      GO TO 158
  154 WRITE(7,157)
  157 FORMAT(10X,'THE TYPE OF THE AC SYSTEM IS FULL STORAGE')
  158 DO 451 K=1,24
      READ(1,401) QLM(K)
  401 FORMAT(25X,F8.6)
  451 CONTINUE
      READ(1,*) QAU1,QJN,QJL,QAU
      READ(4,15) (TA(I),I=1,8760)
   15 FORMAT(5X,12F5.2)
      DO 291 I4=1,12
      READ(8,290) NOFD(I4),AYADI(I4)
  290 FORMAT(I3,A6)
  291 CONTINUE
      K1=0
      DO 150 I2=1,12
      ND=NOFD(I2)
      DO 150 J=1,ND
      DO 160 K=1,24
      I=K1+K
      FA(I)=(TA(I)+273.15-TINF)/TINF
      QLD=0.0
      IF(I2.EQ.1) QLD=QJL
      IF(I2.EQ.2) QLD=QAU
      IF(I2.EQ.3.OR.I2.EQ.12) QLD=QJN
```

```
      IF(I2.EQ.3.OR.I2.EQ.12) QLD=QJN
      QL1(I)=(QLM(K)*QLD)/(QAU1*UAFC*TINF)
  160 CONTINUE
      K1=K1+24
  150 CONTINUE
      IF(NDY.EQ.10) GO TO 5000
      WRITE(7,1010)
 1010 FORMAT(/,10X,'SUMMER TIME ACTIVE A.C. SYSTEM WITH T.E.S.')
      A=GAMA*R+RCFC*NC
      RW=2.0*R*W+1.0
      WRB=W*(RCFC*NC-R*BETA)
      WWR=W*W*R*(GAMA BETA)
      DO 215 IY=1,NY
      TIPR1=TIPR+1.
      IF(IY.LT.NY) GO TO 283
      WRITE(7,282) IY,TIPR,TIPR1
  282 FORMAT(/2X,'/',66('-'),'\'/
     &2X,'|     RESULTS AT THE ',I2,'. YEAR ON LASTDAY OF EACH',20X,'|'/
     &2X,'|           MONTH AT TIME INT. ',F5.2,' TO ',F5.2,25X,'|'/
     &2X,'|',66('-'),'|'/2X,'| MONTH    TA    QL    QE     QC     QHE   TC'
     &,'     TE     TFI    TW     TR    |'/2X,'|',66('-'),'|')
  283 KJI=0
      WT=0.0
      IF(IY.NE.NY) GO TO 206
C     WRITE(7,203) IY
C 203 FORMAT(/ '    RESULT OF THE ',I2,'. YEAR')
      QHESM=0.0
      QCSM=0.0
      QLOSS=0.0
      QSTR=0.0
      QE51=0.0
  206 DO 210 MO=1,12
      WRITE(6,*) MO,'. MONTH'
      ND=NOFD(MO)
      IF(IY.NE.NY) GO TO 207
C     WRITE(7,282) ND,MO,AYADI(MO)
C 282 FORMAT(/'     RESULTS AT THE ',I2,' DAY OF ',I2,'. ',A6,' MONTH'/
C    &'      TIME      TA        QL      QE       QC      QHE     TC      TE     '
C    &,' TFI      TW      TR')
  207 T=0.0
      QE5=0.0
      W2=0.0
      IF(FR.GT.FMAX) GO TO 41
      IF(FR.LT.FMIN) GO TO 42
      GO TO 42
   41 KIM=4
      GO TO 43
   42 KIM=5
   43 DO 220 NOD=1,ND
      REWIND 1
      DO 230 NH=1,24
      I=NH
      IJK=KJI+NH
```

```
      IJK=KJI+NH
      IF(I.EQ.1.AND.NOD.EQ.1) GO TO 79
   95 D1=0.0
      QL=QL1(IJK)
      QCT=0.0
      QE2=0.0
   65 IF(KIM.EQ.4) GO TO 60
      S1=0.0
      QC=0.0
      QE=0.0
      GO TO 61
   60 S1=1.0
      IF(ITAC.EQ.1) FW=FA(IJK)
      IF(ITAC.EQ.3.AND.FW.GE.FA(IJK)) FW=FA(IJK)
      CALL QC1(FW,W,A,RW,WRB,WWR,GAMA,FR,BETA,AS,B,C)
      CALL QCON1(AS,B,C,QC)
      QE0=QC-W
      QCT=QCT+QC
      QE1=QE1+QE0
      QE2=QE2+QE0
   61 IF(ITAC.EQ.1) GO TO 151
      IF(FW.LE.FA(IJK)) GO TO 11
      S2=1.0
      GO TO 12
   11 S2=0.0
      QHE=0.0
      GO TO 13
   12 CALL QHE1(IJK,RHEFC,NHE,FW,FA,QHE)
   13 CALL FWP1(DT,P,S1,QC,S2,RFCS,QHE,FW,FWP)
  151 CALL FRP1(DT,S,S1,W,QC,QL,FR,FRP)
      FR=FRP
      FCON=FW+QC*RCFC*NC
      FW=FWP
      IF(FR.GE.FMAX) GO TO 97
      IF(FR.LE.FMIN) GO TO 98
      GO TO 97
   98 KIM=5
      GO TO 68
   97 KIM=4
   68 D1=D1+1.0
      IF(QC.GT.0.0) GO TO 76
      GO TO 77
   76 QE4=QC-W
      QE5=QE5+QE4
      QE51=QE51+QE4
      QCSM=QCSM+QC
      W2=W2+W
      GO TO 78
   77 QE4=0.0
   78 QHESM=QHESM+QHE
      QLOSS=QLOSS+RFCS*FW
      QSTR=QCSM-QE5-QLOSS
   79 IF(D1.LT.DI) GO TO 65
```

```
 79 IF(D1.LT.DI) GO TO 65
    FAU=(1.0+FA(IJK))*TINF 273.15
    FWPU=(1.0+FWP)*TINF-273.15
    FRPU=(1.0+FRP)*TINF-273.15
    FCONU=(1.0+FCON)*TINF-273.15
    IF(QC.LE.0.0) GO TO 47
    FEV=((FCON+1)/(W*BETA/(QC-W)+GAMA))-1.0
    FFC=FR-NFC*(QC-W)
    GO TO 48
 47 FEV=0.0
    FFC=0.0
 48 FEVU=(1.0+FEV)*TINF-273.15
    FFCU=(1.0+FFC)*TINF-273.15
    QLU=QL1(IJK)*UAFC*TINF
    QCU=QC*UAFC*TINF
    QEU=QE4*UAFC*TINF
    QHEU=QHE*UAFC*TINF
    READ(1,166) TIME, ZAMAN
166 FORMAT(2F5.1)
    IF(NOD.EQ.ND.AND.IY.EQ.NY.AND.TIME.EQ.TIPR) GO TO 102
    GO TO 230
102 QL1U=QL1(IJK)*UAFC*TINF
    WRITE(7,66) AYADI(MO),TA(IJK),QL1U,QEU,QCU,QHEU,FCONU,FEVU,
   &FFCU,FWPU,FRPU
 66 FORMAT(2X,'| ',A6,2F5.1,8F6.2,' |')
230 CONTINUE
    T=T+24
    KJI=KJI+24
220 CONTINUE
    IF(W2.EQ.0.0) GO TO 211
    COP=QE5/W2
    WT=WT+W2
    CP(MO)=COP
    GO TO 218
211 CP(MO)=0.0
218 Q20(I)=QHESM
210 CONTINUE
    IF(IY.LT.NY) GO TO 215
    WRITE(7,797)
797 FORMAT(2X,'|',66('-'),'|',/
   &2X,'|  UNITS OF HEAT & TEMPERATURE ARE KW & CELSIUS',20X,'|'
   &/2X,'\',65('-')
   &,'/'//'     YEARLY DIMENSIONLESS ENERGY BALANCE OF THE SYSTEM')
    WRITE(7,776) QHESM,QCSM,QE51,QLOSS,WT,QSTR
776 FORMAT(3X,'    QHESM=',F8.3,'    QCSM=',F8.3,'    QESM=',F8.3/
   &3X,'    QLOSS=',F8.3,'    WT   =',F8.3,'    QSTR=',F8.3)
    WRITE(7,1113) IY
1113 FORMAT(/'  THE RESULTANT COP AT THE END OF EACH MONTH FOR ',I2,
   &'. YEAR')
    WRITE(7,1112) (I,CP(I),I=1,MO-1)
1112 FORMAT(3('    COP(',I2,')=',F7.4))
215 CONTINUE
    MT=MT+1
```

```
      MT=MT+1
      GO TO 136
C......CALCULATION OF THE COMPRESSOR WORKS DURING WINTER SEASON
 5000 WRITE(7,3011)
 3011 FORMAT(/,10X,'WINTER TIME ACTIVE A.C. SYSTEM WITH T.E.S.')
      FWC=FR-QF/(NFC*UAFC*TINF)
      AA=GAMA*R+RCFC*NC
      RW=2.0*R*W+1.0
      WRB=W*(RCFC*NC-R*BETA)
      WWR=W*W*R*(GAMA BETA)
      WRITE*,GAMA,REFC,NE,RCFC,NC
      A=GAMA*REFC*NE+RCFC*NC
      BC=NFC*(1.0-X*EFC)
      B1=GAMA*(2.0*REFC*NE*W+1.0)
      B2=W*(RCFC*NC-BETA*REFC*NE)
      C1=W*W*REFC*NE*(GAMA-BETA)
      DO 5215 IY=1,NY
      WTP=0.0
      QE51=0.0
      QE61=0.0
      QCSM=0.0
      QLOSS=0.0
      QSTR=0.0
      TIPR1=TIPR+1.
      KJI=0
      IF(IY.NE.NY) GO TO 5206
      WRITE(7,1282) IY,TIPR,TIPR1
 1282 FORMAT(/2X,'/',60('-'),'\'/
     &2X,'|    RESULTS AT THE ',I2,'. YEAR ON LASTDAY OF EACH',14X,'|'/
     &2X,'|             MONTH AT TIME INT. ',F5.2,' TO ',F5.2,19X,'|'/
     &2X,'|',60('-'),'|'/2X,'| MONTH    TA    QL     QE     QC     QFC  TC'
     &,'     TE     TW     TR    |'/2X,'|',60(' '),'|')
C     WRITE(7,5203) IY
C5203 FORMAT(// '   RESULT OF THE ',I2,'. YEAR')
 5206 DO 5210 MD=1,12
      WRITE(6,*) MD,'. MONTH'
      ND=NOFD(MD)
      IF(IY.NE.NY) GO TO 5207
C     WRITE(7,5282) ND,MD,AYADI(MD)
C5282 FORMAT(/'    RESULTS AT THE ',I2,' DAY OF ',I2,'. ',A6,' MONTH'/
C    &'     TIME       TA      QL      QHE     QE      QF      QC     TC',
C    &'     TE      TW      TR')
 5207 T=0.0
      QE5=0.0
      QE6=0.0
      W2=0.0
      DO 5220 NOD=1,ND
      REWIND 1
      DO 5230 NH=1,24
      I=NH
      IJK=KJI+NH
  595 D2=1.0
      QL=QL1(IJK)
```

```
      QL=QL1(IJK)
      QCT=0.0
      QE2=0.0
  565 IF(FRP.LT.FR) GO TO 5505
      IF(FRP.GE.FR) GO TO 5510
 5505 IF(FRP.LT.FR.AND.FRP.LT.FMIN) GO TO 5520
       GO TO 5515
 5510 IF(FRP.GT.FR.AND.FRP.GT.FMAX) GO TO 5515
      GO TO 5520
 5515 KIM=4
      GO TO 568
 5520 KIM=5
  568 FR=FRP
      IF(I.EQ.1.AND.NOD.EQ.1.AND.MO.EQ.1.AND.IY.EQ.1) GO TO 579
      NT=5
      IF(ITAC.EQ.3.AND.FW.GE.FWC) GO TO 1560
      QHE=0.0
      QC=0.0
      IF(FWP.LT.FA(IJK)) GO TO 665
  665 IF(FWP.LT.FW) GO TO 3505
      IF(FWP.GE.FW) GO TO 3510
 3505 IF(FWP.LT.FW.AND.FWP.LT.-0.1215) GO TO 3520
       GO TO 3515
 3510 IF(FWP.GT.FW.AND.FWP.GT.-0.0867) GO TO 3515
      GO TO 3520
 3515 MIM=4
      GO TO 3568
 3520 MIM=5
 3568 FW=FWP
      IF(IJK.GT.6000.AND.IJK.LT.8040.AND.MIM.EQ.4) GO TO 560
      GO TO 1559
 1560 NT=10
      IF(KIM.EQ.4) GO TO 560
 1559 S2=0.0
      QC=0.0
      QE=0.0
      GO TO 561
  560 S2=1.0
      IF(ITAC.EQ.1) FW=FA(IJK)
      CALL QC3(IJK,NT,AA,RW,WRB,WWR,FA,FW,W,A,BC,B1,B2,C1,QFC,
     *GAMA,FR,BETA,AS,B,C)
      CALL QCON2(AS,B,C,QC)
      QE0=QC-W
      QE=QC-W
      QCT=QCT+QC
      QE1=QE1+QE0
      QE2=QE2+QE0
  561 IF(ITAC.EQ.1) GO TO 5151
      IF(FR.LT.FMIN) KIM=5
      IF(KIM.NE.4) GO TO 5111
      S1=1.0
      GO TO 512
 5111 S1=0.0
```

```
5111 S1=0.0
     QFC=0.0
     GO TO 513
 512 IF(ITAC.EQ.3.AND.FW.GE.FWC) GO TO 5802
     CALL QFC3(S1,NFC,FR,FW,QFC)
     GO TO 513
5802 QFC=QC-W
 513 CALL FWP3(DT,P,S1,QFC,QHE,S2,RFCS,QEO,FW,FWP)
5151 CALL FRP3(DT,S,S1,W,QFC,QL,FR,FRP)
     FCON=FA(IJK)+QC*RCFC*NC
     D2=D2+1.0
     IF(QC.GT.0.0) GO TO 576
     GO TO 577
 576 QE4=QC-W
     QE5=QE5+QE4
     QE51=QE51+QE4
     QCSM=QCSM+QC
     W2=W2+W
     GO TO 578
 577 QE4=0.0
 578 QE6=QE6+QFC
     QE61=QE61+QFC
     QLOSS=QLOSS+RFCS*FW
     QSTR=QE6-QE5-QLOSS
     IF(D2.LE.DI) GO TO 565
 579 FAU=(1.0+FA(IJK))*TINF-273.15
     FWPJ=(1.0+FW)*TINF-273.15
     FRPJ=(1.0+FR)*TINF-273.15
     FCONU=(1.0+FCON)*TINF-273.15
     IF(QC.LE.0.0) GO TO 547
     FEV=((FCON+1)/(W*BETA/(QC-W)+GAMA))-1.0
     GO TO 548
 547 FEV=0.0
     FFC=0.0
     QE4=0.0
 548 FEVJ=(1.0+FEV)*TINF-273.15
     FFCU=(1.0+FFC)*TINF-273.15
     QCU=-1.0*QC*UAFC*TINF
     QFCJ=QFC*UAFC*TINF
     QEU=QE4*UAFC*TINF
     QHEU=QHE*UAFC*TINF
     READ(1,5166) TIME,ZAMAN
5166 FORMAT(2F5.1)
     IF(NOD.EQ.ND.AND.IY.EQ.NY.AND.TIME.EQ.TIPR) GO TO 5102
     GO TO 5230
5102 QL1J=QL1(IJK)*UAFC*TINF
     WRITE(7,566) AYADI(MO),TA(IJK),QL1U,QEU,QCU,QFCU,FCONU,FEVU
    &,FWPU,FRPU
 566 FORMAT(2X,'| ',A6,2F5.1,7F6.2,' |')
5230 CONTINUE
     T=T+24
     KJI=KJI+24
5220 CONTINUE
```

```
5220 CONTINUE
     IF(W2.EQ.0.0) GO TO 5211
     COP=QE5/W2
     WTP=WTP+W2
     COP3=QE6/W2
     CP(MO)=COP
     CP1(MO)=COP3
     GO TO 5210
5211 CP(MO)=0.0
     GO TO 5210
5210 CONTINUE
     IF(IY.LT.NY) GO TO 5215
     WRITE(7,1797)
1797 FORMAT(2X,'|',60('-'),'|',/
    &2X,'|  UNITS OF HEAT & TEMPERATURE ARE KW & CELSIUS',14X,'|'
    &/2X,'\',60('-')
    &,'/'//'     YEARLY DIMENSIONLESS ENERGY BALANCE OF THE SYSTEM')
5112 WRITE(7,777) QCSM,QE5,QE6,WTP,QLOSS,QSTR
 777 FORMAT('  QCSM=',F8.3,'  QESM =',F8.3,'  QFCSM=',F8.3/
    &'  WT   =',F8.3,'  QLOSS=',F8.3,'  QSTR =',F8.3)
     WRITE(7,7113)
7113 FORMAT(/'  THE RESULTANT COP AT THE END OF EACH MONTH FOR',I2,
    %'. YEAR')
     WRITE(7,7112) (I,CP(I),I=1,MO-1)
7112 FORMAT(3('    COP(',I2,')=',F7.4))
5215 CONTINUE
     MT=MT+1
     GO TO 136
C
C    DAILY STORAGE PART
C
     MT=1
1001 WRITE(6,352)
 352 FORMAT(/'    IF THE COMPRESSOR WORKS AT DAY TIME',
    %5X,'ENTER -->  5'/'    IF THE COMPRESSOR WORKS AT NIGTH TIME',
    %3X,'ENTER --> 10'//
    %' WHAT IS THE TYPE OF AIR-CONDITIONING SYSTEM AS'/
    %'    FOR AIR-CONDITIONING SYSTEM WITH NO STORAGE ',5X,
    %'ENTER --> 1 WITH 5'/'    FOR AIR-CONDITIONING SYSTEM WITH STORAGE'
    %,9X,'ENTER --> 2'/'    FOR AIR-CONDITIONING SYSTEM WITH PARTIAL',
    %' STORAGE ENTER - > 3')
     READ(5,*) NDY,ITAC
     GO TO 1003
1004 IF(MT.GT.1.OR.ITAC.GT.1) GO TO 2119
     WRITE(7,720)
 720 FORMAT(/5X,' DISTRIBUTION OF DAILY OUTSIDE TEMPERATURE',
    ?' AND COOLING LOAD:'/6X,22('-'),' ON AUGUST, 2 ',22('-')/)
     REWIND 1
     DO 96 L=1,24
     READ(1,101) TIME,ZAMAN,TA(L),QML
 101 FORMAT(2F5.1,2F8.6)
     WRITE(7,69) TIME,ZAMAN,TA(L),QML
  69 FORMAT(10X,'TIME:',F5.2,' ',F5.2,'     TA=',F7.3,
```

```
   69 FORMAT(10X,'TIME:',F5.2,' ',F5.2,'      T'=',F7.3,
     *' C     QL=',F7.3,' KW')
   96 CONTINUE
 2119 IF(MT.GT.1) GO TO 158
      IF(ITAC.EQ.1) GO TO 1153
      IF(ITAC.EQ.2) GO TO 1154
      WRITE(7,1155)
 1155 FORMAT(/10X,'-> THE TYPE OF THE AC SYSTEM IS PARTIAL STORAGE')
      GO TO 1158
 1153 WRITE(7,1156)
 1156 FORMAT(/10X,'-> THE TYPE OF THE AC SYSTEM IS NO STORAGE ')
      GO TO 1158
 1154 WRITE(7,1157)
 1157 FORMAT(/10X,'-> THE TYPE OF THE AC SYSTEM IS FULL STORAGE')
 1158 IF(NDY.EQ.10) GO TO 1119
      WRITE(7,3010)
 3010 FORMAT(10X,'--> DAY TIME ACTIVE A.C. SYSTEM WITH T.E.S.')
      A=GAMA*R+RCFC*NC
      RW=2.0*R*W+1.0
      WRB=W*(RCFC*NC-R*BETA)
      WWR=W*W*R*(GAMA BETA)
      IF(FR.GT.FMAX) GO TO 411
      IF(FR.LT.FMIN) GO TO 421
      GO TO 421
  411 KIM=4
      GO TO 431
  421 KIM=5
  431 KJ=1
  501 QE1=0.0
      QE5=0.0
      W2=0.0
      QCSM=0.0
      QHESM=0.0
      QLOSS=0.0
      QSTR=0.0
      REWIND 1
      IF(KJ.EQ.ND) GO TO 103
      GO TO 104
  103 WRITE(7,52) KJ
   52 FORMAT(//'/',69(' '),'\'/'|    *** RESULTS OF '
     &,I2,'. DAY ***   ',38X,'|'/'|',69('-'),'|'/
     &'|      TIME        QE       QC       QHE      TC       TE       TFL '
     &,'    TW       TR    |'/'|',69(' '),'|')
  104 I=1
   10 READ(1,2521) TIME,ZAMAN,TA(I),QML
 2521 FORMAT(2F5.1,2F8.6)
      FA(I)=(TA(I)+273.15-TINF)/TINF
      QL1(I)=QML/(UAFC*TINF)
  951 D=0.0
      QL=QL1(I)
      QCT=0.0
      QE2=0.0
  651 IF(KIM.EQ.4) GO TO 601
```

```
651 IF(KIM.EQ.4) GO TO 601
    S1=0.0
    QC=0.0
    QE=0.0
    QHE=0.0
    GO TO 611
601 S1=1.0
    IF(ITAC.EQ.1) FW=FA(I)
    IF(ITAC.EQ.3.AND.FW.GE.FA(I)) FW=FA(I)
    CALL QC1(FW,W,A,RW,WRB,WWR,GAMA,FR,BETA,AS,B,C)
    CALL QCON1(AS,B,C,QC)
    QEO=QC-W
    QCT=QCT+QC
    QE1=QE1+QEO
    QE2=QE2+QEO
    FA1=FA(I)
611 IF(ITAC.EQ.1) GO TO 1511
    IF(FW.LE.FA(I)) GO TO 111
    S2=1.0
    GO TO 121
111 S2=0.0
    GO TO 131
121 CALL QHE11(RHEFC,NHE,FW,FA1,QHE)
131 CALL FWP11(DT,P,S1,QC,S2,RFCS,QHE,FA1,FW,FWP)
    FW=FWP
1511 CALL FRP1(DT,S,S1,W,QC,QL,FR,FRP)
    FR=FRP
    FCON=FW+QC*RCFC*NC
    IF(FR.GE.FMAX) GO TO 971
    IF(FR.LE.FMIN) GO TO 981
    GO TO 971
981 KIM=5
    GO TO 681
971 KIM=4
681 D=D+1.0
    IF(QC.GT.0.0) GO TO 761
    GO TO 771
761 QE4=QC-W
    QE5=QE5+QE4
    QCSM=QCSM+QC
    W2=W2+W
    GO TO 781
771 QE4=0.0
781 QHESM=QHESM+QHE
    QLOSS=QLOSS+RFCS*(FW-FA(I))
    QSTR=QCSM-QHESM QLOSS
791 IF(D.LT.DI) GO TO 651
    FAU=(1.0+FA(I))*TINF-273.15
    FWPU=(1.0+FWP)*TINF-273.15
    FRPU=(1.0+FRP)*TINF-273.15
    FCONU=(1.0+FCON)*TINF-273.15
    IF(QC.LE.0.0) GO TO 471
    FEV=((FCON+1)/(W*BETA/(QC-W)+GAMA))-1.0
```

```
      FEV=((FCON+1)/(W*BETA/(QC-W)+GAMA))-1.0
      FFC=FR-NFC*(QC-W)
      GO TO 481
  471 FEV=0.0
      FFC=0.0
  481 FEVU=(1.0+FEV)*TINF-273.15
      FFCU=(1.0+FFC)*TINF-273.15
      QLU=QL1(I)*UAFC*TINF
      QCU=QC*UAFC*TINF
      QEU=QE4*UAFC*TINF
      QHEU=QHE*UAFC*TINF
      IF(KJ.EQ.ND) GO TO 2021
      GO TO 4611
 2021 WRITE(7,661) TIME,ZAMAN,QEU,QCU,QHEU,FCONU,FEVU,FFCU,FWPU,FRPU
  661 FORMAT('| ',F5.2,'-',F5.2,8F7.2,' |')
 4611 I=I+1
      IF(I.GE.25) GO TO 201
      GO TO 101
  201 KJ=KJ+1
      COP=QE5/W2
      IKJ=KJ-1
      IF(IKJ.EQ.ND) GO TO 1021
      GO TO 461
 1021 WRITE(7,7761) QHESM,QCSM,QE5,QLOSS,W2,QSTR
 7761 FORMAT('|',69(' '),'|'/'|     THE UNITS OF HEAT & TEMPERATURE',
     &' ARE KW & CELSIUS RESPECTIVELY',3X,'|'/'\',69('-'),'/'
     &/'   DAILY DIMENSIONLESS ENERGY BALANCE OF THE SYSTEM'/
     &'   QHESM=',F10.4,'   QCSM=',F10.4,'   QESM=',F10.4/
     &'   QLOSS=',F10.4,'   WT   =',F10.4,'   QSTR=',F10.4)
  461 CP(KJ)=COP
      IF(KJ.GE.(ND+1)) GO TO 511
      GO TO 501
  511 WRITE(7,2213)
 2213 FORMAT(/'  THE RESULTANT COP AT THE END OF EACH DAY')
      WRITE(7,2212) (I-1,CP(I),I=2,ND+1)
 2212 FORMAT(4(2X,'COP(',I2,')=',F7.4))
      MT=MT+1
      GO TO 136
C.......NIGTH TIME CALCULATION PART.........
 1119 IF(MT.GT.1) GO TO 4119
      WRITE(7,1011)
 1011 FORMAT(10X,'--> NIGTH TIME ACTIVE A.C. SYSTEM WITH T.E.S.')
 4119 FWC=FR-QF/(NFC*UAFC*TINF)
      FWC=0.00347
      A=GAMA*REFC*NE+RCFC*NC
      BC=NFC*(1.0-X*EFC)
      B1=GAMA*(2.0*REFC*NE*W+1.0)
      B2=W*(RCFC*NC-BETA*REFC*NE)
      C1=W*W*REFC*NE*(GAMA-BETA)
      AA=GAMA*R+RCFC*NC
      RW=2.0*R*W+1.0
      WRB=W*(RCFC*NC-R*BETA)
      WWR=W*W*R*(GAMA BETA)
```

```
      WWR=W*W*R*(GAMA BETA)
      KJ=1
1501  QE1=0.0
      QE5=0.0
      QE6=0.0
      W2=0.0
      QCSM=0.0
      QLOSS=0.0
      QSTR=0.0
      REWIND 1
      IF(KJ.EQ.ND) GO TO 2103
      GO TO 2104
2103  WRITE(7,1521) KJ
1521  FORMAT(//'/',62(' '),'\'/'|',10X,'*** RESULTS OF '
     &,I2,'. DAY ***',26X,'|'/'|',62('-'),'|'
     &/'|     TIME        QE        QF       QC     TC      TE'
     &,'      TW      TR   |'/'|',62('-'),'|')
2104  I=1
1101  READ(1,1151) TIME,ZAMAN,TA(I),QML
1151  FORMAT(2F5.1,2F8.6)
      FA(I)=(TA(I)+273.15-TINF)/TINF
      QL=QML/(UAFC*TINF)
1951  D=1.0
      QCT=0.0
      QE2=0.0
1651  IF(FRP.LT.FR) GO TO 6505
      IF(FRP.GE.FR) GO TO 6510
6505  IF(FRP.LT.FR.AND.FRP.LT.FMIN) GO TO 6520
      GO TO 6515
6510  IF(FRP.GT.FR.AND.FRP.GT.FMAX) GO TO 6515
      GO TO 6520
6515  KIM=4
      GO TO 658
6520  KIM=5
 658  FR=FRP
      BIB=7.0
      FA1=FA(I)
      IF(TIME.GE.23.0.OR.TIME.LE.6.0) GO TO 1601
      BIB=8.0
      IF(ITAC.EQ.3.AND.FW.GE.FWC) GO TO 2160
      GO TO 2159
2160  NT=10
      IF(KIM.EQ.4) GO TO 1601
2159  S2=0.0
      QC=0.0
      QE=0.0
      GO TO 1511
1601  S2=1.0
      CALL QC31(NT,AA,RW,WRB,WWR,FA1,FW,W,A,BC,B1,B2,C1,QFC,GAMA,FR,
     *BETA,AS,B,C)
      CALL QCON2(AS,B,C,QC)
      QE0=QC-W
      QE=QC-W
```

```
      QE=QC-W
      QCT=QCT+QC
      QE1=QE1+QEO
      QE2=QE2+QEO
 1611 IF(ITAC.EQ.1) GO TO 2151
      IF(KIM.NE.4.0) GO TO 1111
      S1=1.0
      GO TO 1121
 1111 S1=0.0
      QFC=0.0
      GO TO 1131
 1121 IF(ITAC.EQ.3.AND.FW.GT.FWC) GO TO 4811
      CALL QFC3(S1,NFC,FR,FW,QFC)
      GO TO 1131
 4811 QFC=0.0
      IF(TIME.LT.23.0.AND.TIME.GT.6.0) QEO=.0
 1131 CALL FWP31(DT,P,S1,QFC,S2,RFCS,QEO,FA1,FW,FWP)
      FW=FWP
      IF(ITAC.EQ.3.AND.FW.GT.FWC.AND.QC.GT.0.0) QFC=QC-W
      IF(BIB.EQ.7.0) QFC=0.0
 2151 CALL FRP3(DT,S,S1,W,QFC,QL,FR,FRP)
      FCON=FA(I)+QC*RCFC*NC
      D=D+1.0
      IF(QC.GT.0.0) GO TO 1761
      GO TO 1771
 1761 QE4=QC-W
      QE5=QE5+QE4
      QCSM=QCSM+QC
      W2=W2+W
      GO TO 1781
 1771 QE4=0.0
 1781 QE6=QE6+QFC
      QLOSS=QLOSS+RFCS*(FW-FA(I))
      QSTR=QE6-QE5-QLOSS
 1791 IF(D.LE.DI) GO TO 1651
      FAU=(1.0+FA(I))*TINF-273.15
      FWPJ=(1.0+FWP)*TINF-273.15
      FRPJ=(1.0+FRP)*TINF-273.15
      FCONU=(1.0+FCON)*TINF-273.15
      IF(QC.LE.0.0) GO TO 1471
      FEV=((FCON+1)/(W*BETA/(QC-W)+GAMA))-1.0
      GO TO 1481
 1471 FEV=0.0
      FFC=0.0
 1481 FEVU=(1.0+FEV)*TINF-273.15
      FFCJ=(1.0+FFC)*TINF-273.15
      QLU=QL1(I)*UAFC*TINF
      QCU=-1.0*QC*UAFC*TINF
      QFCJ=QFC*UAFC*TINF
      QEU=QE4*UAFC*TINF
      IF(KJ.EQ.ND) GO TO 2102
      GO TO 2461
 2102 WRITE(7,1661) TIME,ZAMAN,QEU,QFCU,QCU,FCONU,FEVU,FWPU,FRPU
```

```
2102 WRITE(7,1661) TIME,ZAMAN,QEU,QFCU,QCU,FCONU,FEVU,FWPU,FRPU
1661 FORMAT('| ',F5.2,'-',F5.2,7F7.2,' |')
2461 I=I+1
     IF(I.GE.25) GO TO 1201
     GO TO 1101
1201 KJ=KJ+1
     IKJ=KJ-1
     COP=QE5/W2
     COP3=QE5/W2
     IF(IKJ.EQ.ND) GO TO 9102
     GO TO 1461
9102 WRITE(7,7771) QCSM,QE5,QE6,W2,QLOSS,QSTR
7771 FORMAT('|',62(' '),'|'/'|      THE UNITS OF HEAT & TEMPERATURE',
    &' ARE KW & CELSIUS RESP.',3X,'|'/'\',62(' '),'/'
    &/'    DAILY DIMENSIONLESS ENERGY BALANCE OF THE SYSTEM'/
    &'    QCSM=',F10.4,'    QESM =',F10.4,'    QFCSM=',F10.4/
    &'    WT   =',F10.4,'    QLOSS=',F10.4,'    QSTR =',F10.4)
1461 CP(KJ)=COP
     CP1(KJ)=COP3
     IF(KJ.GE.(ND+1)) GO TO 2511
     GO TO 1501
2511 WRITE(7,2113)
2113 FORMAT(/'        THE RESULTANT COP AT THE END OF EACH DAY'/
    &16X,'WRT QE              WRT QFC')
     WRITE(7,2112) (I-1,CP(I),CP1(I),I=2,ND+1)
2112 FORMAT(2(3X,I2,'. DAY COP=',F7.4,'  COP=',F7.4))
     MT=MT+1
     GO TO 136
 134 STOP
     END
C
C     CALCULATION OF THE ENERGY GIVEN FROM HEAT EXCHANGER TO STORAGE
C
     SUBROUTINE QHE1(IJK,RHEFC,NHE,FW,FA,QHE)
     DIMENSION FA(8760)
     REAL NHE
     QHE=(1.0/(RHEFC*NHE))*(FW-FA(IJK))
     RETURN
     END
C
C     CALCULATION OF THE DIMENSIONLESS WATER TEMPERATURE IN STR. TANK
C
     SUBROUTINE FWP1(DT,P,S1,QC,S2,RFCS,QHE,FA,FW,FWP)
     FWP=(DT/P)*(S1*QC-S2*QHE-RFCS*FW)+FW
     RETURN
     END
C
C     CALCULATION OF THE HEAT DISSIPATTED FROM THE CONDENSER
C
     SUBROUTINE QC3(IJK,NT,AA,RW,WRB,WWR,FA,FW,W,A,BC,B1,B2,C1,QFC
    *,GAMA,FR,BETA,AS,B,C)
     DIMENSION FA(8760)
     AS=A
```

```
      AS=A
      IF(NT.EQ.10) GO TO 800
      B=0.5*(FA(IJK)+1.0-GAMA*(FR-QFC*BC)-B1-B2)
      C= W*(FA(IJK)+1.0)+W*(GAMA-BETA)*(1.0+FR)+W*BC*(BETA-GAMA)*QFC+C1
      GO TO 801
  800 AS=AA
      FW=FA(IJK)
      B=0.5*(FW+1.0-GAMA*(FR+FW)-WRB)
      C=WWR-W*(FW+1.0+(BETA-GAMA)*(FR+1.0))
  801 RETURN
      END
C
C     CALCULATION OF THE DIMENSIONLESS WATER TEMPERATURE IN STR. TANK
C
      SUBROUTINE FWP3(DT,P,S1,QFC,S2,RFCS,QEO,FA,FW,FWP)
      FWP=(DT/P)*(S1*QFC-S2*QEO-RFCS*FW)+FW
      RETURN
      END
C
C     CALCULATION OF THE CONDENSER ENERGY COEFFICIENTS A, B, C
C
      SUBROUTINE QC1(FW,W,A,RW,WRB,WWR,GAMA,FR,BETA,AS,B,C)
      AS=A
      B=0.5*(FW+1.0-GAMA*(FR+RW)-WRB)
      C=WWR-W*(FW+1+(BETA-GAMA)*(FR+1.0))
      END
C
C     CALCULATION OF THE ENERGY GIVEN FROM HEAT EXCHANGER TO STORAGE
C
      SUBROUTINE QHE11(RHEFC,NHE,FW,FA,QHE)
      REAL NHE
      QHE=(1.0/(RHEFC*NHE))*(FW-FA)
      RETURN
      END
C
C     CALCULATION OF THE DIMENSIONLESS WATER TEMPERATURE IN STR. TANK
C
      SUBROUTINE FWP11(DT,P,S1,QC,S2,RFCS,QHE,FA,FW,FWP)
      FWP=(DT/P)*(S1*QC-S2*QHE-RFCS*(FW-FA))+FW
      RETURN
      END
C
C     CALCULATION OF THE CHANGE IN ROOM TEMPERATURE
C
      SUBROUTINE FRP1(DT,S,S1,W,QC,QL,FR,FRP)
      FRP=(DT/S)*(S1*(W-QC)+QL)+FR
      RETURN
      END
C
C
C     CALCULATION OF THE HEAT DISSIPATTED FROM THE CONDENSER
C
      SUBROUTINE QC31(NT,AA,RW,WRB,WWR,FA,FW,W,A,BC,B1,B2,C1,QFC,
```

```
      SUBROUTINE QC31(NT,AA,RW,WRB,WWR,FA,FW,W,A,BC,B1,B2,C1,QFC,
     *GAMA,FR,BETA,AS,B,C)
      REAL NC,NFC
      IF(NT.EQ.10) GO TO 800
      AS=A
      B=0.5*(FA+1.0-GAMA*(FR-QFC*BC)-B1-B2)
      C=-W*(FA+1.0)+W*(GAMA-BETA)*(1.0+FR)+W*BC*(BETA-GAMA)*QFC+C1
      GO TO 801
  800 AS=AA
      FW=FA
      B=0.5*(FW+1.0-GAMA*(FR+RW)-WRB)
      C=WWR-W*(FW+1+(BETA-GAMA)*(FR+1.0))
  801 RETURN
      END
C
C     CALCULATION OF THE ENERGY GIVEN FROM HEAT EXCHANGER TO STORAGE
C
      SUBROUTINE QFC3(S1,NFC,FR,FW,QFC)
      REAL NFC
      QFC=(1.0/NFC)*(FR-FW)
      IF(S1.EQ.0.0) QFC=0.0
      RETURN
      END
C
C     CALCULATION OF THE DIMENSIONLESS WATER TEMPERATURE IN STR. TANK
C
      SUBROUTINE FWP31(DT,P,S1,QFC,S2,RFCS,QEO,FA,FW,FWP)
      FWP=(DT/P)*(S1*QFC-S2*QEO-RFCS*(FW-FA))+FW
      RETURN
      END
C
C     CALCULATION OF THE CHANGE IN ROOM TEMPERATURE
C
      SUBROUTINE FRP3(DT,S,S1,W,QFC,QL,FR,FRP)
      FRP=(DT/S)*(QL-S1*QFC)+FR
      RETURN
      END
```

```
C
C     SUBPROGRAM ' INPT1' FOR CALCULATION OF DIMENSIONLESS INPUTS
C
      SUBROUTINE INPT1(WO,QF,QHE7,UAFC,TINF,AL,BL,CL,DET,QC7,TCI,TCO
     %,X,TWO,TR,TW1,QE,
     %TEI,TEO,UAE,UAC,A9,B9,C9,UAST,UAHE,W,S,DT,R,
     %NC,P,RFCS,NHE,NE,REFC,RHEFC,RCFC,QFD,NFC,DI,MW,MA,EPE,EFC,EHE,EC)
      REAL NC,NHE,NE,NFC,MW,MA,MCPR,MCPA,MCPW
      CPAIR=1.0
      ROAIR=1.2
      CPW=3.333
      MW=2.3
      ANHE=QHE7/0.9
      ROW=1175.0
      VAIR=12000.
      CMHE=VAIR*CPAIR*ROAIR/3600.
      HE=UAHE/CMHE
      EHE=1.-EXP(-HE)
      DT=DET/(3600.0*24)
      DI=3600.0/DET
      W=WO/(UAFC*TINF)
      QFD=QF/(UAFC*TINF)
      MA=ROAIR*AL*BL*CL
      MCPR=CPAIR*MA
      S=MCPR/(UAFC*24.0*3600.0)
      MCPW=MW*CPW
      MCPA=MA*CPAIR
      EV=UAE/MCPW
      EPE=1.-EXP(-EV)
      NE=UAE/(EPE*MCPW)
      FCO=UAFC/MCPW
      EFC=1.-EXP(-FCO)
      NFC=UAFC/(EFC*MCPW)
      REFC=UAFC/UAE
      R=REFC*NE+(1.0-EFC*X)*NFC
      RCFC=UAFC/UAC
      DTIO=5.
      MCPWC=QC7/DTIO
      CON=UAC/MCPW
      EC=1.-EXP(-CON)
      NC=UAC/(EC*MCPW)
      P=ROW*A9*B9*C9*CPW/(UAFC*24.0*3600.0)
      RFCS=UAST/UAFC
      RHEFC=UAFC/UAHE
      NHE=UAHE/(EHE*CMHE*ANHE)
      WRITE(7,71) EPE,EFC,EHE,EC
   71 FORMAT(5X,'EPE=',F7.3,'  EFC=',F7.3,'  EHE=',F7.3,'  EC =',F7.3/)
      RETURN
      END
```

```
C
C      SUBROUTINE 'QCON1'
C      CALCULATION OF THE HEAT DISSIPATTED FORM THE CONDENSER
C
       SUBROUTINE QCON1(A,B,C,QC)
       DEL=(B*B)/(A*A) (C/A)
       IF(DEL.LT.0.0) DEL=0.0
       QCP=(-1.0*B/A)+SQRT(DEL)
       QCN=(-1.0*B/A)-SQRT(DEL)
       IF(QCP.GT.0.0.AND.QCN.GT.0.0) GO TO 29
       GO TO 27
    29 IF(ABS(QCP).GT.ABS(QCN)) GO TO 28
       QC=QCN
       GO TO 25
    28 QC=QCP
    27 IF(QCP.LT.0.0) GO TO 24
       QC=QCP
       GO TO 25
    24 IF(QCN.LE.0.0) GO TO 26
       QC=QCN
       GO TO 25
    26 WRITE(6,*) '    * * * * * * * * * * *'
    25 RETURN
       END
```