H. P. Garg

Advances in Solar Energy Technology

Volume 2

Industrial Applications of Solar Energy

D. REIDEL PUBLISHING COMPANY

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Dedicated to my parents who believed in Honesty, Sincerety and Hardwork.

To my wife, Kusum, and my children, Meenu, Neelu, Naina and Darpan, for their support and perseverance.

CONTENTS

PREFACE

The purpose of writing this three volume 'Advances in Solar Energy Technology' is to provide all the relevant latest information available in the field of Solar Energy (Applied as well as Theoretical) to serve as the best source material at one place. Attempts are made to discuss topics in depth to assist both the students (i.e. undergraduate, postgraduate, research scholars etc.) and the professionals (i.e. Consultancy, design, and contracting firms).

Chapter 1 starts with a brief history of solar houses (active heating), one of the oldest and still the widely used application of Solar Energy. Various methods of building heating and other general aspects such as building form and functions are also described. Various components of active solar heating of building like solar collector, storage system, control unit, auxiliary heat source, etc. are discussed very briefly. Three types of solar active heating of buildings like Solar air systems, solar liquid systems, and solar assisted heat pump systems are discussed in detail in this chapter. Design details and performance of nine typical solar houses which are in use in different climatic conditions and using some newer concepts are also discussed in depth in this chapter.

Solar energy can play a significant role in providing process heat in industries thus saving conventional fuels like electricity, gas, oil, etc. This topic of recent interest is discussed in detail in chapter 2. All the three industrial process heat systems like hot water, hot air, and steam industrial process heat systems are described here. A few typical examples of solar process heat systems which are in use are discussed briefly and advantages derived are presented. Some of the problems of industrial process heat systems (IPHS) and a simple design method for solar IPHS are also discussed briefly.

Solar furnaces can provide very high temperatures and can be used for some specialised Research and Development work. The topic of solar furnace is discussed in detail in Chapter 3. Different types of solar furnaces and their components are also discussed in this chapter. A few typical solar furnace designs along with measuring intrumentation are also described. Various material properties both physical and chemical which can be studied and other applications of solar furnace are discussed in detail in this chapter.

Conversion of solar energy into mechanical power is the most important application of solar energy. Considerable effort in this direction has been made and the progress is reviewed in chapter 4. The principle of solar engines and their limitations along with different solar engines like steam engines, turbines, stirling engines, and Brayton engines are discussed in this chapter. A few typical solar power plants using linear parabolic concentrators, paraboloidal dish, central tower receiver, etc. of different capacities are briefly described here. For comparison a 6 MWe solar photovoltaic power plant is also discussed.

In developing countries, the use of solar energy for producing cold either for comfort or for preservation of food can go a long way. This topic of solar refrigeration and airconditioning is discussed in greater detail in chapter 5 of this volume. Various methods of producing airconditioning or refrigeration like absorption cooling, desiccant cooling, vapor compression cooling, and passive cooling are discussed in somewhat greater detail in this chapter. In each case a couple of typical examples are discussed. Various problems and successes achieved in each type of cooling are also presented.

The subject matter of chapter-6 is the passive solar house heating - a topic of the world interest and an application which does not need any moving part and used widely. In this chapter the building functions and forms typical passive buildings, design characteristics, and prediction models are discussed in depth. The various critical parameters affecting the performance of a passive building and which should be kept in mind by an engineer or an architect while planning the buildings are also briefly described.

Attempts are made to discuss important items in detail in each volume giving as many as possible graphs, illustrations, tables, equations to make the subject clear and useful. It is hoped that this volume also will be used as a reference book and as a text book for higher studies. In the end any comments / criticism which may help me in improving the other volumes in the series are also welcome.

New Delhi H P GARG

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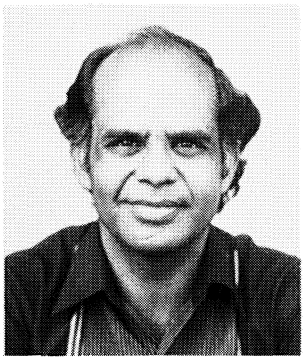
H.P. GARG



ABOUT THE AUTHOR

H.P.Garg is professor and coordinator of solar energy at Indian Institute of Technology, New Delhi, India. He is internationally recognised as one of the world's leading authorities in the field of solar thermal applications. Dr.Garg is involved in research and teaching of Solar Energy for the last 21 years and is the author of more than 250 research papers. He has arranged several national and international training programmes and conferences in the field of solar energy utilization and visited several countries of the world. He has made significant contribution to the field of Solar Energy Collectors and Solar Heating Systems. His designs on solar systems are quite popular both nationally and internationally and he has three Indian patents to his credit. Prof.Garg is the author of 'Treatise on Solar Energy' published by John Wiley & Sons (1982), England; co-author of 'Solar Thermal Energy Storage' published by D.Reidel Publishing Co. (1985), Holland; and edited a book, 'Solar Water Heating Systems' published by D.Reidel Publishing Co.(1985), Holland. His main fields of interest are Solar Energy Technology and Utilization, Energy conservation, Bio-gas systems, Wind power utilization, and Energy planning.

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CHAPTER 1

SOLAR HEATING OF BUILDINGS: ACTIVE SYSTEMS

1.1 INTRODUCTION

The function of a building or a house is to provide shelter to its occupants from weather. Since the weather conditions vary widely over the year and in different places, and the humans feel comfortable within certain range of air temperatures and humidities, the houses are made to provide everyday living comfort. The heating of houses in winter to provide comfort by the use of solar energy is an ancient concept and is in use since man started to build habitations. Basically the solar heating systems are divided into two categories: the passive heating and active heating. Passive systems do not need any mechanical system and are designed such that the glazed area, walls and roofs are made use for collecting, storing, and distributing the heat indoors by natural processess of convection, radiation, and conduction. Four basic concepts for passive solar heating are: direct gain, collector-storage wall, sunspace, and collector-storage roof. These concepts will be discussed in a separate chapter in volume 3. In active heating systems, separate solar collectors are used to heat a fluid; storage devices are used to store heat for use at night and on intermittent days; auxiliary heating systems to supply heat when required; and distribution systems alongwith controls to supply heat to required spaces.

Active heating of buildings can be done using either of the following three ways:

(i) Solar heating of water in liquid collectors and transferring the heat to living space from storage unit by means of liquid or air.

(ii) Solar heating of air in air collectors and transferring the heat to living space from storage unit by means of air or liquid, and

(iii) Using solar energy 'stored' in the environment in the form of heat (ambient heat) by means of heat pumps.

Any of the above active space heating systems if coupled with an appropriate auxiliary heating arrangement can provide the same comfortable conditions as by the conventional space heating system can be installed in existing buildings and thus will be termed as 'retrofit system' but such system generally turns out to be costly. The buildings which are specially designed for providing comfortable conditions using solar active heating systems are such that they get more solar energy in winter and less in summer and thus reduce the heating and cooling loads, are better known as 'Solar houses'. Such houses are adequately insulated; properly oriented; optimally glazed; adequately sealed against air leakages; and appropriate materials and combination of materials used to admit, absorb, store, release, and distribute solar energy to reduce the heating load and thereby the size of the heating system. In different countries the climate is different and varying, building codes are different, economic prosperity is different, different kind of building materials are available, and therefore the structure and thermal characteristics of the buildings are different. Therefore, there cannot be a common 'solar house' design for all the places.

Indeed the greatest need of the day is to develop a cost-effective active solar heating system. Therefore the problem is of two fold, the improved thermal perfromance at lower overall system cost. The initial cost and area required for a active solar heating system may be higher compared to the conventional heating system but the operational cost will be much lower. Therefore, a system analysis based on parametric studies should be carried out to predict the cost effectiveness and thermal performance of a system for different system configurations, climates, building types, and load. In this chapter some of the requirement of solar houses, solar active heating systems, a few typical solar houses, design methods, and ecnomics are discussed.

1.2 HISTORY OF SOLAR HOUSES

Solar active space heating is the oldest and matured technology and many thousands of solar heating systems are now operational in several countries and reliable design methods are now available. Although research on solar space heating has continued for the last forty years, most of the reliable and engineered installations based on early systems are made only during the last one decade. Perhaps the idea of solar space heating was first[1] given by Professor E.S. Morse of Salem, Massachustts (USA), in 1881, where a blackened slate properly glazed was fixed near the wall facing south and its function was almost similar to the modern 'Trombe wall'[2], presently practised all over the world.

Systematic studies on solar heating began in 1938 at the Massachusetts Institute of Technology (MIT), Cambridge, Massachusetts, USA, by Dr.H.C.Hottel and his group under a grant from Godfrey L.Cabot Foundation. The first classic paper[3] dealing with solar flat-plate collectors and liquid heating was published in 1942 by Dr.H.C.Hottel and his graduate student Mr.B.B.Woertz. Later analytical and experimental studies on solar collectors and fluid heating were carried out in USA[4,5] U.K.[6], Australia[7], Israel[8], South Africa[9], and India[10].

Perhaps the first 100 percent solar heated solar house known as MIT Solar House No. 1 was built at Lixington, Massachusets by Hottel[11] in 1939. The house was consisted of two rooms, one office and one Laboratory with a total floor area of 46.5 m^2. Liquid flat-plate collectors installed on the sloped roof with a total exposed area of 33.5 m^2 with triple glazing were used along with a large hot water storage tank of 65.86 m^3 capacity placed in the basement. Based on its experience, Solar House known as M.I.T. Houses NO.2, 3 and 4 were built in 1947,1949, and 1958 respectively. The M.I.T. Houses No.4 was built[12] in 1958 and the solar system provided 44 percent space heating load and 57 percent domestic hot water load. It was a two storey building having 135 m^2 of usable living area. The heating system consists of solar water having liquid double glazed flat plate collectors (59.5 m^2 installed on sloped roof, insulated hot water storage tank (5.7 m^3), oil fired auxillary furnace and hot-air supply arrangement to the rooms.

A solar house was designed in 1948 by Talkes and Raymond[13] under a grant from Miss Amelia Peabody and was constructed at Dover, Massachusetts, USA. The solar heating system consists of double glazed solar air heater (66.9 m^2) in a vertical position on the south wall of the building, storage bins (13.3 m^3) containing Glauber's salt (sodium sulphate decahydrate), and fans for distributing the heat from storage to the rooms.Only the ground floor area of 135.3 m^2 was heated. The entire heating load was to be met by the system as per design and the storage system is designed to provide heat for 5 days.

Seven houses were constructed by Thomason[14,15] in Washington D.C. with the first single storeyed built in 1959 with a floor area of 139 m^2. In this system, simple and inexpensive 'trickle' collectors[16] (78 m^2) were used. The hot water is stored in a water storage tank (6.1 m^3) surrounded by 50 Tons of small, 100 mm diameter rocks, which gets heated by conduction and convection heat transfer from the tank. This system had provided 95 percent of the heating load of the house. The seventh house uses the three principal innovations (i) use of 'Pancake' under the floor heat storage, (ii) use of shallow solar pond on the roof with a booster reflector, and (iii) draining hot water from shallow solar pond each night to underfloor 'Pancake' heat storage area where it warms the floor and living space. Since these Thomason houses are widely studied for many years, it has given a confidence and many new houses in USA are made on the principles of Thomason houses.

The pioneer work on Solar House Heating was done by George O.G.Lof at the Colorado State University, Colorado, USA. Lof designed a nine room resdence[17,18] in Denver, Colarado with a total Living area of 296 m^2, and heated with solar energy, in which he and his family have lived since 1959. The solar system consists of overlapped glass plate solar air heaters[19] (49.1 m^2); a rock bed storage system with 10651 Kg granite rock (13-25 mm diameter); an air to water heat exchanger; an auxiliary natural gas fired furnace; and associated fans, blowers and controls. The heating system is completely automatic with provisions for water heating in addition to space heating. The solar system is fully instrumented and very systematic performance data for the last 22 years is obtained and the decline in performance has been only 1.88 percent per year which has given a great confidence in the installation of solar active heating system.

R.W.Bliss[20] in 1955 constructed a solar heated house at Amado, Arizone having floor area 62.43 m^2, matrix air heaters (29.26 m^2) and a rock bed storage system (36.8 m^3). The solar collectors not only work as heat receiver but also as a heat radiator for cooling purposes. Auxiliary heat is supplied by a small heat pump. Although it was claimed at that time that this is the first 100 percent solar heated home in USA but the economics were adverse.

Perhaps the first single storeyed office building was heated by solar energy in 1956 at Albuquerque, New Mexico, USA, and the same was designed by Bridgers and Paxon[21]. Single glazed liquid flat plate collectors (70 m^2) were installed on the sloping south wall of the building having approximately 400 m^2 useful floor area. An underground hot water storage tank of 22.7 m^3 was used in this system. Space is heated by passing hot water from hot water storage tank to the tubes embeded in the ceiling and floors of the rooms. Auxiliary heat is supplied by a heat pump which performs the dual function of supplying part heat in winter if required and cooling in the summer season.

Yanagimachi[22,23] did pioneer work on solar house heating and cooling and designed Solar House I and II in 1958 and 1961 which used unglazed collectors for collecting heat during daytime and radiates heat during night time and this hot or cold is stored in water storage tanks to supply heating or cooling to the building and heat pumps to maintain adequate temperature difference between the two tanks. This priciple is in use in several buildings in Japan.

The Institute of Energy Conversion, University of Delaware, New york, Delware, USA[24] in 1973 designed a 'Solar One' which uses solar air heaters (70 m^2) for heating the storage unit and space; and photovoltaic panels made of Cadmium sulphide/copper sulphide (CdS/Cu_2S) which generate electricity which is stored in 120 volt d.c. storage batteries for use at night to run a heat pump which cools the building; and two types of heat of fusion materials for energy storage. One storage unit contains sodium thiosulphate pentahydrate ($Na_2SO_4.5H_2O$) with a change of phase of 49 °C and stores heat for use when building requires heating and the second storage contains a eutectic salt, mainly sodium sulphate decahydrate ($Na_2SO_4.10H_2O$) with a change of phase at 12.8 °C to store coolness in summer produced by a heat pump during the day when the solar cells are producing electricity, for use at night. More than a day's storage of heat and cold can be done in these systems.

Lof et al[25,26] have designed NSF/CSU house No.I and No.II in 1974 and 1976 respectively for heating and cooling of space. CSU I is a two-storeyed residence of 128 m^2 heated area with liquid flat plate collectors (67 m^2). The Colorado State University (CSU) Solar house II is of identical configuration except it uses a flat-plate sloar air heating collector (68.4 m^2), a 20 ton rock bed storage unit, an auxillary hot water heater, a day-night evaporative exchange cooler with outside air inlet and exhaust duct, blowers, fans, automatic dampers and controls etc.

Recently Schreitmuller[27] has described 'Solarhaus Freiburg' a 12-apartment house constructed in 1978 in the Rhine Valley 141 Km west of Freiburg, South West Germany and which is one of the biggest residential building equipped with a solar plant for hot water supply and partial space heating in West Germany. This house not only make use of active solar heating but also incorporates a number of so-called passive energy saving measures. The house is equipped with two arrays of evacuated tube collecters each of about 30 m^2 net solar absorbing area. The auxiliary energy is supplied by an oil fired heating system and is needed only during particularly cold weather and extended periods of poor sunshine. The house is fully instrumented and controlled and more than 40 different operational modes can be realized in the Freiburg Solar House. This is a joint Venture of Bundesministerium fur Furschung and Technologies, West Germany and U.S. Department of Energy, USA.

Kazuhiro Aiso[28] has designed in 1982 a Yazaki solar House II for Nagaoka, Niigata, Japan which is a 190 m^2 hybrid Solar House of two floors, the first made of reinforced concrete and the second of wood. The heating system consists of 48 high efficiency solar collectors (56.2 m^2), a 2-tons heat storage tank surrounded by 40 m^3 of soil storage media, a hot water fired absorption chiller, a gas fired auxillary boiler, 14 snow melting panels, 6 Ceiling mounted radiant heating panels, a floor heating system, and 7 fan coil units for room heating, and several passive design features. Inspite of poor insolation levels in winter, the system is able to provide comfortable, safe and convenient living environment.

Recently, in many countries of the world, demonstration Solar Houses with solar active space heating system have been built by Research Institutions, Industrial Companies and private Owners because of serious shortage and rapidly rising prices of petroleum products. A few impotant pioneer solar houses in Chronological order are listed in Table 1.1. Several review articles on solar active house heating have appeared in journals and books[29-46] which described the design features and performance characteristics of some of the Solar Houses.

Table 1.1 Typical Solar Houses around the world

1. MIT Solar House No.1 in Massachusetts, USA by H.C.Hottel (1939).
2. Boulder Solar House in Colorado, USA, by Lof (1945).
3. MIT Solar House No.2 in Massachusetts, USA by H.C.Hottel (1947).
4. Dover Solar House in Massachusetts, USA by M.Telkes, E.Raymond and A.Peaboy (1948).
5. MIT Solar House No.3 in Massachusetts, USA by H.C.Hottel (1949).
6. New Mexico State College Solar House, USA, by Gardenshire (1953).
7. Lefever Solar House in Pennsylvania, USA, by H.R.Lefever (1954).
8. Amado Solar House in Arizona, USA, by Denovan, E.Raymond, R.W.Bliss (1956).
9. Bridgers and Paxon Solar House, Albuquerque, New Mexico, USA (1956).
10. Solar House in Bristol, U.K. by L.Garden (1956).
11. Richmansworth Solar House, U.K. by E.Curtis (1956).
12. Solar House in Tokyo, Japan, by M.Yanagimachi (1956).
13. University of Toronto Solar House,Canada, by E.A.Allcut (1956).
14. MIT Solar House No.4 in Massachusetts, USA, by H.C.Hottel (1958).
15. Solar House in Casablanca, Marocco, by C.M.Shaw and Assosiates, (1958).
16. Solar House in Nagoya, Japan, by M.Yanagimachi (1956).
17. Denver Solar House in Colorado, USA, by G.O.G.Lof (1959).
18. Princeton University Solar House in New Jersey, USA, by A.Olgyay (1959).
19. Solar Office House in Tucson, Arizona, USA, by Bliss (1959).
20. Thomason Solar House No.1 in Washington D.C., USA, by A.Thomason (1959).

21. Thomason Solar House No.2 in Washington D.C., USA, by A.Thomason (1961).
22. George School Building, Wallasey, U.K., by A.E.Morgan (1962).
23. Henry Mathew Solar House, Coos Bay, Oregon, USA, by H.Mathew (1966).
24. Hoffman Solar House, Surrey, British Columbia, Canada, by Hoffman (1971).
25. Kimura Solar House,Tokorazawa, Japan, by K.Kimura (1972).
26. Solar One House at Newwark, Delaware, USA (1973).
27. Phoenix of Colorado Springs Solar House in Colorado Springs, Colorado, USA (1973).
28. NSF/CSU Solar House 1,Colorado State University, Colorado, USA, by D.S.Ward and G.O.G.Lof (1974).
29. OSE Project Solar House, Tokyo, Japan (1974).
30. Copper Development Assosiation Decade 80 House, Tucson, USA (1975).
31. Milton Keynes House at Milton Keynes, U.K., by S.V.Szokoley (1975).
32. Higher Bebington Houses,Liverpool, U.K. (1975).
33. Building Research Establishment Houses, Watford, U.K. (1975).
34. Ashi Kuzuha Solar House, Hirakata, Japan, by Yazaki Co. (1975).
35. Soka Solar House,Koka,Japan by S.Tanaka (1975).
36. Toshiba Solar House, Kawasaki, Japan, by Toshiba Co. (1975).
37. Pepper Solar House, Granton, Ontario, Canada by Pepper (1975).
38. Zero Energy House, Technical University of Denmark, Lyngby, Denmark (1975).
39. Philips Solar House, Philips Research Laboratory, Aachen, Germany (1975).
40. Solar House of Eindhaoven University of Technology, Eindhover, The Netherlands (1976).
41. Ishibashi Solar House, Kosai, Japan, by T.Ishibashi (1976).
42. LASL Mobile Solar House, Los Alamos Lab., New Mexico, USA (1976).
43. Granada House, Macclesfield, U.K. (1976).
44. NSF/CSU Solar House II,Colorado State University, Fort Collins, Colorado, USA, by D.S.Ward and G.O.G.Lof (1976).
45. Konishi Solar House,Kobubunyi, Japan, by Y.Nakajima (1976).
46. Solar House in New Zealand (1976).
47. Lorriman Solar House, Mississauge, Ontario, Canada, by Lorriman (1976).
48. La Macaza Solar House, La Makaza, Quebec, Canada, by Shelter Systems Group, McGill University, Montreal

(1976).
49. Provident Solar House,Ontario, Canada, by F.C.Hooper and J.Fix (1976).
50. IVES House, Hudson, Canada, by National Research Council of Canada (1977).
51. Solar House 'Helios I' in Trapeza Aigialeias, Greece, by N.Alexandros (1978).
52. Solarhaus Freiburg, Freiburg, West Germany, by K.R.Schreitmuller (1978).
53. Solar House in Studsyik, Sweden, by R.Roseen and B.Perers (1979).
54. Yazaki Solar House II, Japan, by Kazuhiro Aiso (1982).

1.3 BUILDING FORM AND FUNCTION

Everybody needs a comfortable house where activities like sitting, sleeping, dining, food preparation, storing, studying, recreation, bathing, hobbies, etc can be conducted. In the house the rooms should be located in such a way that the rooms which are more frequently required should be towards the south side of the house making efficient use of winter sun. The rooms required for morning activities should be located towards the east side. For late-afternoon use, the rooms should be towards the west side of the house. The rooms facing north receive little sun and therefore remain cooler and can be used for storage and sleeping. For living, required for major duration, the room facing South-West may be better. Energy conservation in building play a significant role which not only make a comfortable living but energy saver also. It is true that if there is a choice, then one should go to energy conservation measures and then to solar. A hybrid system is now preferred due to its economics which combines passive heating of building and an active solar heating system. In such a hybrid system[47] the indoor temperature can be controlled more precisely than with a purely passive or active design.

Building site and location is important not only from convenience point of view but from comfort also. The natural topography and micro-climate may significantly affect the solar performance. Building sites get affected by sun and prevailing wind, site slope, hills, vegetation, frost and drifting snow, soil types, and proximity of water ponds or water canals.

Building morphology can significantly reduce the heating load. Rectangular shaped houses with the length not more then 1.5 times the width and elongated in the east-west direction would minimize heat loss in winter and overheating in summer[48]. This is due to the fact that east and west faces receive more radiation in summer and should have less area while south face receive more radiation in winter and should have larger area. The large glazing area or windows should be in the south wall to permit the entry of winter sun. All other walls should contain minimum glass area. Such considerations has led Wright[49] to propose 'heliothermic' site planning. But the south facing windows design and area should be such that their energy balance at a place is positive i.e. the total solar heat gain during the winter seasons may exceed their total heat loss[50]. Therefore, it is necessary in winter to have double or triple glazed windows or even heat mirror coatings on glazing may be used. Movable insulation can also be used to reduce heat loss at night and on cloudy days.

Multistoreyed building should be preferred than the single storeyed building with the same volume, since it would have less exposed area and therefore comparatively less heat loss. Moreover, multistorey building is best suited for central hot air systems[51]. A part of the building if sunken in the ground will also be energy efficient [52], since the temperature in the underground at a 3 m depth remains constant and is generally equal to the annual average of air temperature.

The colour of roof and walls have tremendous effect on the indoor climate of absorption and reflection effect of solar radiation. In colder climates, the heating load can be reduced by using dark colours particularly on the southern wall which helps in the reduction of heat loss from building interior. In Israel it is a practise[53] to white wash the roof in the spring season. This reflects summer sun and allows more heat in winter because the rains in autumn removes this whitewash and leaves the roof dark in colour.

Vegetation near the house may also significantly change the indoor environment. Small and dense vegetation near the northern, western and eastern wall will change the wind pattern and thereby changing the external surface coefficient of wall resulting in increase in effective thermal resistance of the wall. This will also reduce to some extent the air infiltration through leaks in windows and walls. Deciduous trees if planted near the wall facing north will provide warmth from sun in winter and coolness due to shading in summer.

The three thermophysical properties, the thermal resistance, heat capacity and solar absorption of surface are of importance for energy conservation in buildings. A building material or a combination of building materials for walls, roofs, floors, and internal partitions should be so chosen that they should provide a high thermal resistance together with a high heat capacity. The thick and heavy structure of walls and roofs not only suppress the amplitude of the external temperature and thus smoothing out variation in inside temperature but also stores sufficient amount of heat which is useful for night time. In this respect 'Trombe wall' which is high mass wall on the south side of the house, blackened and glazed, will be very useful which not only serves as solar energy collecter but also as storage and built in radiant heating panel. It is recommended that walls and roof should be well insulated with double sided reflective aluminium foil.

Since the payback period of energy consevation features as discussed above is less compared to active solar heating system, it is recommended that these features should be included in the building design from the outset. There is no thumb rule or standard method to find out the optimum mix of solar and energy conservation features. This can only be done by using economic methodologies and performace prediction methods[54] using computer simulation, although this is an expensive design process.

1.4 CONVENTIONAL SPACE HEATING SYSTEMS

Heating of building space is required for providing comfort to the occupants. The heat for heating the space in the building should be supplied at a rate equal to the rate of heat loss by conduction through the building elements to outdoors and by convection due to air exchange from outside to inside. There is a widely varying standards[55] of heating in different countries, but these can be categorised in the following three forms:

(i) Complete house heating,
(ii) Partial house heating, and
(iii) Single room heating.

Depending on the above standard forms of heating, the heating methods may be different. Generally the space heating methods are radiant fires, convection stoves, hot water or steam radiator or convector system, and forced warm air heating. Heating fuels which are generally used are natural gas, fuel oil, electricity, coal, and wood.

For heating small rooms or where central heating facilities are not available, the unit type of radiant fires in individual rooms can be used. Radiant fires are generally: open fire place where wood is used as fuel, carefully designed grate where brown Coal is used as fuel, radiation ovens where gas is used as fuel, electric resistance type radiators where the element is exposed and electricity is used as fuel, and electrical radiator where the element is concealed. In such unit type of radiant fire systems, the heat is transferred by radiation, and therefore considerable temperature gradient exists.

Convection stoves in small rooms provide much uniform temperature compared to radiant fire systems. Here fuels like caol, electricity, oil, and gas can be used. In case of coal fired stoves burning rates can be controlled by adjusting the amount of input air. Heat transfer air generally passes through the ducts within the stove either by natural convection or by a small propeller fan. Better distribution of heat in the room can be achieved with fan but the fuel consumption in this case increases.

In some central heating systems, hot water or steam is used which passes through radiators or finned tubes installed in the space being heated. The space gets heated due to heat delivered from the fins by natural convection. Sometimes the long pipes or fins are embeded in the floor, walls, and ceiling of the room and the space gets heated due to natural convection and radiation heat loss from these panels. Generally hot water at a temperature of about 80 °C is supplied to these panels. If the surface area of these panels are large such as in case of floor, walls or ceiling embedded panels, hot water at a temperature between 40-50 °C can be supplied.

In most of the centrally heated buildings, hot air is used to heat the spaces. In such systems the cool room air is passed over a heat exchanger of a furnace by means of a blower fan. Sometimes steam or hot water boilers are used to heat the air which is then supplied through ducts to the rooms of a building. The heat exchanger is heated by burning of coal, oil, gas, or wood. The hot air generally supplied to the living space is at a temperatue of 50-60 °C which gets mixed with the cool room air which is later pumped back to the heat exchanger for further heating. Optimum arrangement of discharge points are as follows[56]:

(a) Hot air should be discharged in a room close to the floor and in a horizontal direction.
(b) Inlet height for the return air is not very important.
(c) The hot air in the room should be supplied at a rate that the supplied hot air temprature should not be more than 20 °C.

1.5 GENERAL ASPECTS OF SOLAR ACTIVE HEATING OF BUILDINGS

There is a large variety of solar active space heating system ranging from small size heaters with only a few m^2 of collector area to large and sophisticated systems used for heating community and industrial buildings where the collector area may be several thousands of m^2. Yet there is a large similarity in the basic layout of these systems. A typical active solar space heating system is shown in fig.1.1. All these active systems may be small or big consists of mainly the following five subsytems[59]:

1. The solar energy collectors generally the flat-plate liquid or air collectors, converting the

solar radiation into heat.

2. A suitable heat storage device, generally water or pebble bed or mix of these two or sometimes latent heat storage materials, synchronisation of heat supply and heat demand.
3. An auxiliary heat supply arrangement required in case of poor sunshine.
4. Control system and fluid flow devices, controlling the various operations.
5. Pumping and ductwork or the distribution systems supplying the heat at an appropriate place.

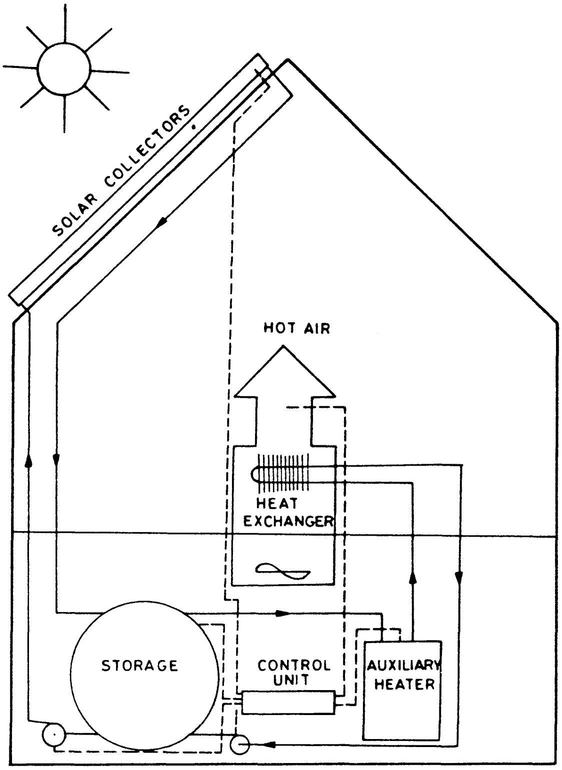


FIG.1.1 SIMPLE BASIC LAYOUT OF AN ACTIVE SOLAR HEATING SYSTEM

It has been seen earlier that in most of the solar space heating systems, the temperature requirement of the fluid is in the range of 50 to 80 ^{o}C, which makes the system much simpler in terms of fluid selection and the obvious choice is water or air, in terms of solar energy collectors and the choice is liquid or air heating collector, and in terms of storage device and the most obvious choice is water or rock bed storage system.

Apart from the above main components of a active solar heating system, there are many other additional components that are generally required and which depend on the climatic and operating conditions, materials used, purpose and size of installation, and preference of the individual designer. Some of these additional components are as follows:

(i) A heat exchanger in the collector and storage loop. This is required when the fluid in the collector loop and the storage tank is different such as water in the storage tank and antifreeze solution in the collectors or air in the collector loop and water in the storage tank.

(ii) Drain-down type of collectors where the water is drained in the evenings to protect it from damage due to freezing.

(iii) A heat exchanger in the delivery and storage loop which is also required when the fluids in the storage tank and the delivery pipe are different.

(iv) The distribution system can also be quite complex which may not only supply heat for space heating but also for hot water supply making the system more reliable.

(v) Multilayered and multiplex storage system, storing the heat at different temperatures for different applications and for different periods like diurnal or seasonal.

(vi) Use of heat pump along with the collectors operating at lower temperature difference and thereby increasing the efficiency.

(vii) A device for exhausting the surplus heat avoiding the boiling of water.

(viii) A suitable liquid to air heat exchanger or pipe work or panels in the load loop for dissipating heat to the space required for heating.

These eight additional components and many other components along the main five subcomponents discussed above makes a solar heating system quite complex and therefore an understanding of each of these components and their optimization is required to make a system cost effective. Various modes of operation of the heating system are :

(i) When solar energy is available, the energy collected by the collectors can be directly supplied to the buildings without storing it. If required the energy can be augmented by auxiliary heating arrangement. This mode may also be possible when the storage unit is fully charged and the solar collected heat is directly supplied to the building.

(ii) In this mode of operation, solar energy is available, but heat is not required in the building and therefore the solar collected heat is stored in the storage unit.[3]

(iii) In this mode of operation solar energy is not available, and the heat required in the building is supplied from the storage unit which was stored earlier.

(iv) In this special mode of operation, neither the solar energy is available nor the heat is stored in the storage unit, and hence the required heat in the building is supplied from the auxiliary heating source.

(v) In some special cases, neither the heat in the storage unit is required nor in the building is required and therfore the collector goes on collecting the solar heat. In such cases some energy dumping machanisms or pressure relief valves are used or otherwise a steady state condition reaches where the rate of absorption of heat equals the rate of heat loss.

(vi) In some special mode of operations, solar heat is not only used for heating the space but also for heating the water for domestic use.

As discussed earlier, solar space heating systems are of two types; the active heating where solar energy collector, storage system, and distribution system are separate units and pumps and fans etc. are used for circulating the fluid and distribution of heat etc.; and the passive heating systems where the solar energy collector, storage, and distribution are combined in the building envelope itself. Passive systems are much simpler and generally economical and can be incorporated only at the time of design of buildings. A photograph of a typical passive solar heated building[58] in U.K. is shown in fig.1.2. This is a St.Georges's School Building at Wallasey, U.K. completed in 1961 and is the oldest and probably the largest direct gain passive solar building in the U.K. The entire south wall of the building acts as solar wall and measures 70 x 8 m and is double glazed. The entire roof and north wall is well insulated. Observations have shown that solar radiation alone is able to heat the school building. In winters average temperatures of 16 ^{o}C and above are observed in the building and 24.5 ^{o}C in summmer.

Even the active heating systems are of three types: the one in which liquid flat-plate collectors are used, the second in which solar air heating collectors are used, and the third makes use of a heat pump system alongwith flat-plate collectors. These systems will be discussed later in this chapter.

In a liquid heating system, a liquid flat plate collector generally a tube in plate type of collector, a trickle

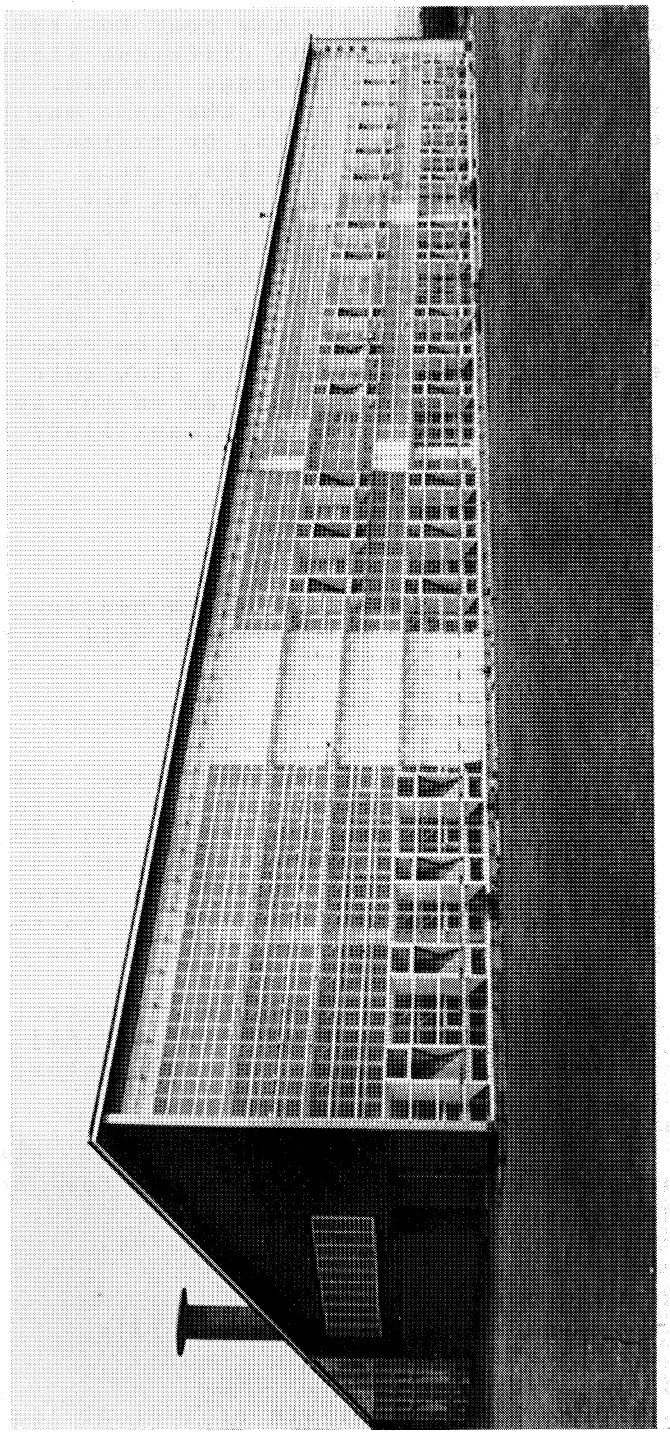


FIG.1.2 THE SOLAR WALL OF ST.GEORGE'S SCHOOL, WALLASAY, U.K.LATITUDE 53° 25'N
(Courtesy Dr.M.G.Davies, The University of Liverpool, U.K.)

collector or evacuated tube collector is used and a working fluid either water or antifreeze solution is used. The same liquid can also be used to supply the heat to the space which required heating. Alternatively different liquids can be used in the collector loop and storage system. If hot water is used as storage material then the same may be used to dissipate the heat through radiators, or radiant floor or ceiling panels, baseboard heating strips, etc. Sometimes liquid to air heat exchanger is used and hot air is distributed in the occupied space. If air is used as a working fluid in the collector than this hot air can directly be supplied to the space or through a rockbed storage system. Therefore, in such air heating systems, air can be the working fluid and this hot air can directly be supplied to the rooms. The fluid temperature and its flow rate in the room depends on the heating load as well as on the collector type and area, storage size and capacity, auxiliary heating arrangements, etc.

1.6 COMPONENTS OF SOLAR HEATING SYSTEM

The major components of an active solar heating systems are shown in fig.1.1. These five subsystems will be discussed here in brief.

1.6.1 The Solar Heat Collector

There is a large variety of solar energy collectors both of liquid and air heating type that are used for space heating purpose. A good review of both liquid and air collectors are given by Yellott[59], Kreith et al[60] Garg[61], Selcuk[62], and Garg et al[63]. The liquid collectors range from simple zigzag tube welded on a metal sheet to the more sophisticated evacuated tube collectors, and a few commonly used are listed below:

(i) Corrugated sheet type flat-plate collector[6,10]
(ii) Roll-bond type of flat-plate collector[64].
(iii) Tube-in-plate type of flat-plate collector[65,66].
(iv) Water trickle type collector[67].
(v) Evacuated tube type collector[68].

Similarily solar air heaters are also of different designs but are comparatively simpler and a few commonly used air heaters are as follows:

(i) Simple conventional air heater[69,70].
(ii) V-corrugated type air heater[71].
(iii) Finned type air heater[72].
(iv) Overlapped glass plate air heater[73].
(v) Matrix type air heater[74]
(vi) Porous bed air heater[75].

A systematic comparative analysis of typical liquid and air collectors is made by Kreith et al[60] and the results in respect of these two collectors for a typical space heating are shown in Table 1.2. From this Table it is seen that at high insolation values both the liquid and air collectors operate at the same efficiency while at low insolation values air collectors operate at much higher efficiency which is due to the low return air temperature in case of air collectors. Similarly air collectors supplies air at high temperatures reulting in lower cost in heat distribution.

TABLE 1.2 Comparison of typical solar heating systems employing liquid and air collectors (From Kreith et al[60]

DESIGN CHARACTERISTICS				
	liquid		air	
Heat removal efficiency factor, F_R	0.9		0.7	
Overall heat loss coefficient, U_L (w/m^2 °C)	4.26		4.26	
Cover transmittance, ζ	0.85		0.85	
Collector plate absorption, α	0.95		0.95	
$F_R(\zeta\alpha)$	0.73		0.57	
$F_R(U_L)(w/m^2$ °C$)$	3.86		3.01	
OPERATING CONDITIONS				
Atmospheric temperatures, Ta(°C)	-1.1	-1.1	-1.1	-1.1
Fluid inlet temperature, Ti(°C)	54.4	54.4	21.1	21.1
Solar insolation, $I_{Tt}(w/m^2)$	947	473	947	473
Fluid flow rate (l/m^2min)	0.814	0.814	0.61	0.61
$(Ti-Ta)/I_{Tt}(w/m^2C)^{-1}$	0.059	0.117	0.023	0.047
CALCULATED PERFORMANCE				
$F_RU_L(Ti-Ta)/I_{Tt}$	0.23	0.43	0.07	0.14
$F_R(\zeta\alpha)-F_RU_L\ (Ti-Ta)/I_{Tt}$	0.50	0.27	0.50	0.43
Collection efficiency (Percent)	50	27	50	43
Computed Outlet temperature(°C)	63	59	57	52

There are several other practical advantage of air collectors over liquid collectors such as freezing, corrosion problems and superheating are almost absent and durability is better and maintenance and repair is easier . For handling large amount of air, a large air duct work is required compared to a liquid heating system. Similarly the size of the rock bed storage system in case of air heating systems is 3 times more compared to a water storage system for storing the same amount of heat. But these two, the air duct system and rock bed storage can easily be integrated in the design of the house. However in large active heating system like in commercial and industrial buildings, the heat distribution can conveniently and economically be done by hot water and therefore liquid collectors are preferred. Moreover, if cooling and other multiple use like domestic hot water supply etc. are also required, then it is possible only with liquid collectors. Although liquid type collectors have drawn more attention and widely used and requires a compact storage size and flow conduits etc. the possible disadvantage of a liquid collectors are high initial cost, possibility of leaking because of many pipe joints, freezing of collector water, corrosion effects of water, and boiling under occasional conditions. All these problems can be satisfactorily solved but at an added cost. Evacuated tubular collectors either of liquid or air type are now manufactured commercially in many countries and are successfully used in many solar active heating systems. These collectors outperform in performance the conventional liquid and air collectors and with the setting up of large volume production lines, it is expected that very soon the cost will go below 100 $\$/m^2$ and therefore it is expected that in the coming decade these evacuated collectors will oust many flat-plate collectors.

The non-imaging concentrators which are low concentrating, stationary, and upto a concentration ratio of 3 and costing only 25 $\$/m^2$ can also be advantageously used for space heating systems.

1.6.2 The thermal storage system

Thermal storage is an essential part of an active solar space heating system due to anticyclic nature of heat demand and solar radiation, and also due to diurnal variation of solar radiation due to weather variability and also from season to season. The size and type of heat storage media depends on the weather conditions and the percentage of total heating load to be supplied. Therefore, the thermal storage can be a short term storage say from 1 day to 8 days and long term storage or generally termed as seasonal storage. Several storage media have been used in solar house heating systems. The most obvious choice with liquid collectors is water in tank and sometimes with some antifreeze liquid additive is used. The water has a high specific heat of 4.2 KJ/Kg K but a heat exchanger is required to transfer energy from stored water to the point of use[76,77]. Rock is another choice for use as sensible heat storage media[78,79] and has specific heat of 0.8 KJ/K. The rock bed system is used alongwith air collectors. Air can be used as a heat transfer media in rock bed system without heat exchanger. This combination has an additional advantage that it cannot freeze and a little leakage will not be a problem. Properties of some of the sensible heat storage materials suitable for space heating applications are shown in Table 1.3. Rock for a storage system should be carefully selected. It should be hard, of optimum size and the storage bin should be of proper voidage.

Table 1.3 Properties of sensible heat storage media

Storage media	Density	Specific heat	Heat capacity	
	(Kg/m^3)	(KJ/Kg K)	MJ/m^3	KWh/m^3
Water	1000	4.2	126	35
Rock	2240	0.8	54	15
Iron	7860	0.45	106	29.5
Mineral oil	900	1.8	48.5	13.5

In some buildings, storage space is limited, and in such case phase change storage materials [80,84] have merit over the sensible heat storage materials. Extensive research and development work for a suitable phase change storage material for building heating is carried out but no practical solution appears to be found out. Several problems and possible solutions with phase change materials are discussed by Garg[85]. Materials which have been tested for solar space heating are listed in Table 1.4. Although several materials have been tried, but the only phase change storage device which is commercially sold in a few countries for solar house heating is the eutectic salts commonly known as 'Glauber's Salt' and is sodium sulphate decahydrate. This material has a latent heat of about 251 KJ/Kg and melts and solidifies at approximately 32 °C. This material was used by Dr. Telkes in her first solar heated house and here only she realised several difficulties with this well known and

Table 1.4 Typical latent heat storage media suitable for solar space heating.

Storage media	Melting point (°C)	Density (Kg/m^3)	Heat of fusion (KJ/Kg)
$Na_2SO_4.10H_2O$	32	1458	251
$CaCl_2.6H_2O$	28	1634	174
$Na_2HPO_4.12H_2O$	36	1552	265
$Na_2CO_3.10H_2O$	32	1442	247
$Na_2S_2O_3.5H_2O$	48	1669	209
Capric acid	36	-	153
Lauric acid	49	-	177
n-Eicosane	37	-	251
Paraffin	56	-	209

Table 1.5 Thermal Storage of 1000 MJ with a 20 °C useful temperature rise

	Rock	Water	Phase change material
Specific heat (KJ/KG K)	0.8	4.2	2.1
Heat of fusion (KJ/Kg)	-	-	250
Density (Kg/m^3)	2240	1000	1500
For storage of 1000 MJ			
Weight (Kg)	62500	11900	3450
Volume (m^3)	28	12	2.3
Weight ratio	18.1	3.5	1
Volume ratio	12.2	5.2	1

inexpensive heat storage material. Many problems with sodium sulphate decahydrate and too with all the salt hydrates tested for solar space heating were encountered. A tendency for crystals to settle out and nonuniformity with numerous cycling of heating and cooling are the general problems. Several ways are suggested to overcome these difficulties.

* Use of nucleating agent, such as 3 to 5 percent borax/sodium tetraborate decahydrate ($Na_2B_4O_7.10H_2O$) to sodium sulphate decahydrate.
* Use of thickening agents having thixotropic gel-type structure like clay (Bentonite, Attapulgite) with sodium sulphate decahydrate to prevent setting.
* Encapsuling the latent heat material in multiple shallow containers.
* Occasionally agitating the phase change material container.
* Using extra water in the salt hydrate.

Tests using 3 per cent borax and 8 percent thickening agent with sodium sulphate decahydrate contained in 1.52cm O.D. plastic tubes of 0.15cm wall thickness were conducted in the laboratory[83]. No observable change or deterioration in the material is reported even after 1000 cycles.

So far as the energy storage capabilities are concerned, the phase change material is superior to rock and water storage systems as is evident from Table 1.5. It is seen from this table that so far as volume is concerned the water storage system and rock bed storage system (no voidage) occupies 5-times and 12-times spaces compared to phase change storage system. Based on these facts increased attention is diverted towards the use of latent heat storage system. but it is still an open question. Therefore, it appears that for the time being, the rock bed and water storage system will continue to be used for solar house heating applications.

1.6.3 Auxiliary heat supply system

Auxiliary heat supply arrangement is to be provided in all solar active house heating systems. This auxiliary heat can be supplied through a furnace which can be gas or oil fired, or electrically supplied or through a heat pump system in case of air heating systems. While in case of liquid heating systems auxiliary heat is supplied either directly through electric coils or through a boiler which is fired by oil, gas, electricity or any other fuel. Generally electricity is not preferred in such cases due to economical reasons.

As discussed earlier, in liquid systems, either hot water from the top of the solar storage system is directly supplied to the radiant panels in the rooms or it exchanges heat to the air in a liquid to air heat exchanger and the hot air is supplied to the rooms. In both the cases the cooled water or air from the rooms is collected and returned to the bottom of the solar storage tank. While in air heating systems hot air from the solar storage tank is directly supplied to the rooms and the cooled return air from rooms is supplied to the bottom of the storage tank.

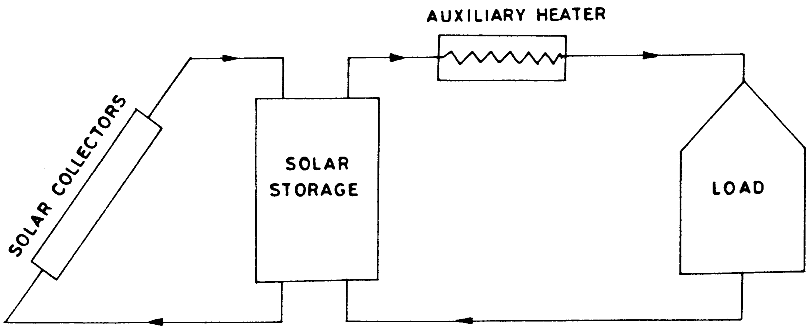
Whatever type of storage system or auxiliary system is used, it is essential that the solar collectors should be supplied with the coolest possible liquid or air so that the collectors operate at maximum efficiency. The hot liquid or air from the top of the solar storage tank should be supplied to the load loop and the cooled room liquid or air should be supplied to the bottom of the tank. Stratification in the storage tank helps in the performance of the solar collectors but it is difficult to maintain due to several operational parameters.

There are several modes of supplying auxiliary heat to the solar heating system as shown in fig.1.3. Auxiliary heat should be supplied to the load loop and not the collector loop. This will help in augmenting the solar heat or replace it when required. In collector loop this will unnecessarily raise the fluid inlet temperature to the collector.

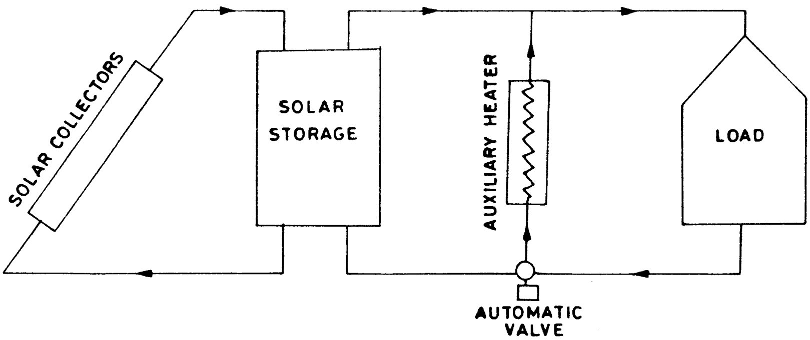
The conventional way of providing auxiliary heat in the load loop is shown in fig.1.3(a) where the auxiliary system is in series. In this system the auxiliary heater works as a booster, adds required amount of heat to the solar heated fluid, supplied to the load, and returned to the solar storage system. Here the possibility is there that the returned fluid temperature from the load is higher than the temperature in the bottom of the solar storage and thus adding heat to the storage and resulting in reduced collector efficiency and increase in the use of auxiliary energy.

Another auxiliary arrangement in a solar heating system which is generally preferred over as shown in fig.1.3(a) is shown in fig.1.3(b). Here the auxiliary arrangement is provided in parallel to the load loop and is used only when the desired temperature in the radiant panel or convector is not met by solar storage. If the solar storage temperature is low then with the help of special automatic valve only the auxiliary heat is supplied to the rooms to meet the requirement. In this parallel arrangement solar heat is not wasted, since it goes on adding up in the solar storage and can be used when desired.

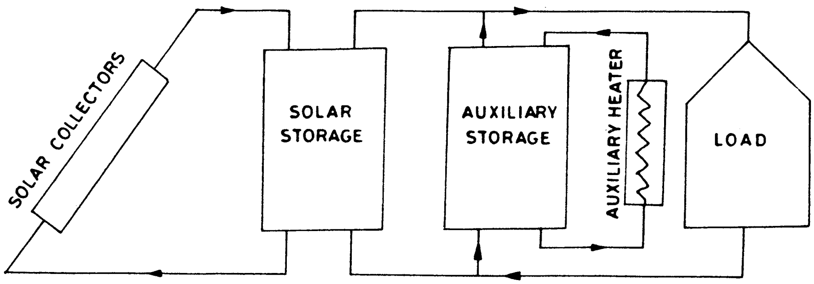
An auxiliary system with a separate auxiliary heat storage and solar storage system will perform even better since in this the off-peak electricity is used in charging a separate auxiliary storage system. The auxiliary storage will be charged whenever there is a generating capacity and will be used only at the end of the each off-peak period. This arrangement is shown in fig.1.3(c). Alternatively a



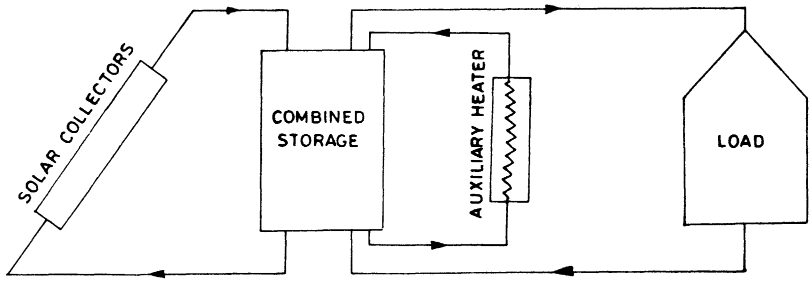
(a) SERIES AUXILIARY HEAT SUPPLY SYSTEM



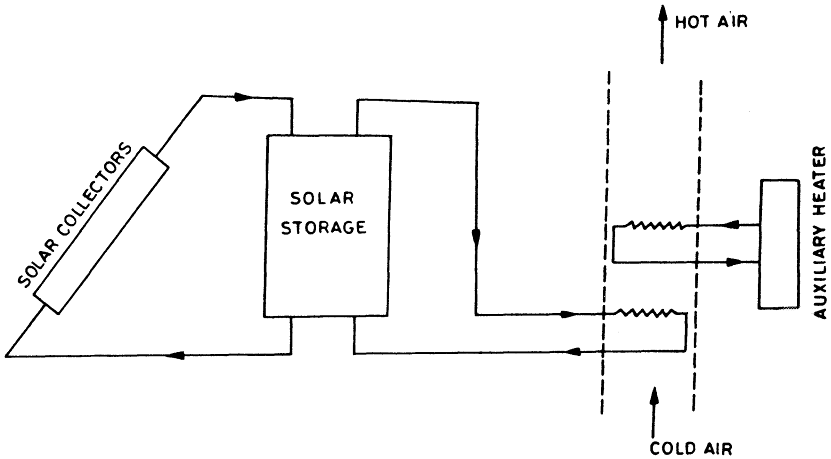
(b) PARALLEL AUXILIARY HEAT SUPPLY SYSTEM



(c) SEPARATE HEAT STORAGE WITH AUXILIARY HEAT



(d) COMBINED HEAT STORAGE WITH AUXILIARY HEAT



(e) SEPARATE AUXILIARY HEAT SUPPLY SYSTEM

FIG.1.3 POSSIBLE MODES OF AUXILIARY HEAT ARRANGEMENTS IN SOLAR ACTIVE SPACE HEATING SYSTEMS.

combined storage for storing solar and auxiliary heat can also be used as shown in Fig.1.3(d). Here during off-peak periods, the electric energy is used to charge the storage sytstem and is used for room heating. Whenever, solar energy is available it is used to charge the storage system. This combined storage system results in lowering the collector performance due to increase in inlet temperature. Mechanically single storage system is simpler compared to the two tank system, but suffers from performance penalty.

If air heating system is employed then still a better auxiliary arrangement is suggested by Hunn and Lof[35] where two separate heating coils, one heated with solar heated fluid and another heated by auxiliary arrangement is kept one over the other in the path of return air duct as shown in fig.1.3(e). Here the solar coil works as a preheater coil and the air is further heated to the desired level by auxiliary heating coil. This system appears to be simple, inexpensive, and efficient since solar energy is used even when it is very low and whenever it is available. An air furnace can also be used in place of auxiliary heating coil. The air furnace is generally designed to take the peak heating load.

The use of heat pump in series with the solar collectors can reduce the auxiliary electrical energy use in both liquid and air heating systems. With the heat pump combination a coefficient of performance (COP) of about 4 to 5 is generally obtained which means that it supplies 4 or 5 times of heat energy as is supplied by electric energy. Thus a heat pump offers a good solution in those areas where the ambient air temperatures are not too low but the system is expensive and it becomes complex.

Hughes et al[86] conducted a simulation study of three auxiliary heating options (Fig. 1.3(a), 1.3(c) and 1.3(d)) using conventional flat-plate collectors and evacuated collectors and concluded that the separate heat storage system, fig.1.3(c), is superior to combined storage, fig.1.3(d), irrespective of the collector types. It is also found that the combined storage unit can be small in case of evacuated tube collectors and the performance difference between separate and combined storage unit is relatively small at elevated temperatures of operation.

1.6.4 Control systems

A control system is a must for any active space heating system whether it is a liquid based or air based. A controller consists of three components, the sensors which sense the state of affair and communicate to the control unit, a control unit meant for taking decisions as per preprogrammed based on sensor inputs, and actuators which carry out the decisions based on the control unit. The requirement of the controller is that it should be low in cost, dependable, and provide energy at a maximum efficiency. Generally, controllers turn off and on pumps, blowers, dampers, valves, etc. Now-a-days various kinds of control units are available for solar active heating operations but one should be careful in

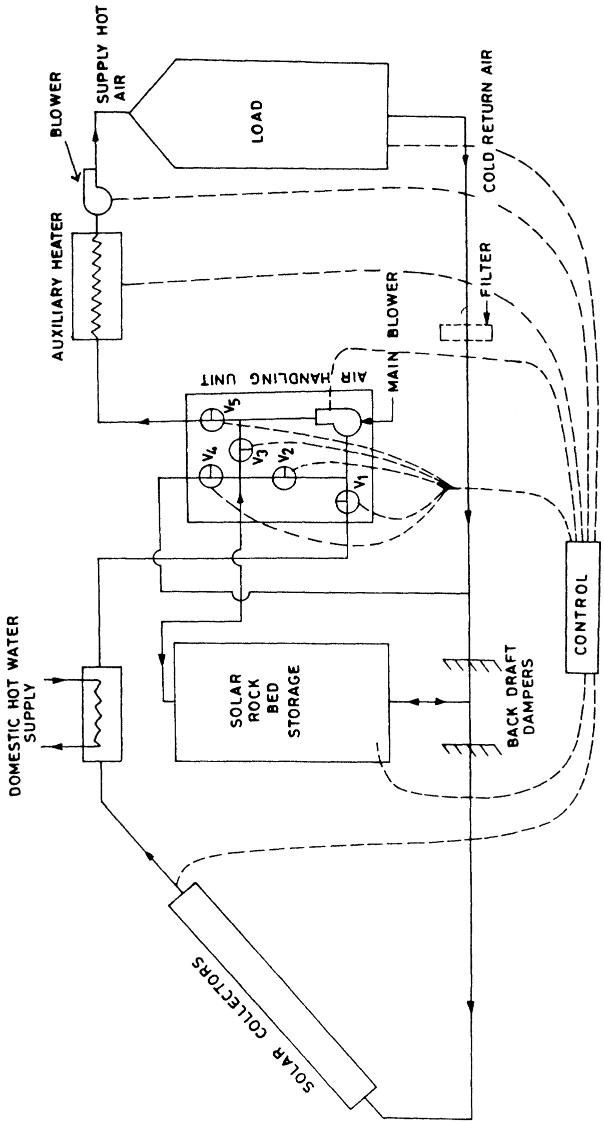


FIG.1.4 SCHEMATIC REPRESENTATION OF SOLAR AIR HEATING SYSTEM

deciding the type of control unit required for the system.

Although numerous operating and control procedures are possible, but generally in solar heating systems, following four modes of operation are required:

1. Transferring solar heated air directly from solar air collectors to the rock bed storage system from the top and supply the cold air from the bottom of the storage to the inlet of the solar air collectors. In this mode there is no heating load.
2. Transferring solar heated air directly from solar air collectors to the load and the return cold air from load directly to the inlet of the solar air collectors.
3. Transferring stored heat from the storage system to the load and cold air from the load directly to the bottom of the storage system. In this case the solar collectors are ineffective.
4. In this mode of operation both the solar storage and solar collectors are ineffective and the cold return room air is directly passed through the auxiliary heater and then to the room.

A typical solar air heating system[87,88,89] is schematically shown in fig.1.4 and for this system, various control modes are depicted in fig.1.5. In the simplest mode of operation, when the house does not require any heat and the sunshine is available, then the solar heat can be collected through the solar collectors and stored in the solar storage tank. For this mode of operation two types of controller, the differential controller[90] and proportional controller[91] are available. Differential controllers are more widely used in which there are two sensors, one sensor is fixed on the absorber plate at the exit of the collector and the second in the bottom of the storage unit. A temperature difference of about 6 °C between the collector outlet and storage bottom turn on the blower and opens the dampers which circulates the air at a flow rate of about 10 l/m^2s (2 cfm/ft^2) and collector and turn off the blower when this difference reaches 2 °C and repositions the directional dampers. In a proportional controller, the lower speed is variable and either the outlet temperature or temperature difference between collector outlet and inlet is maintained constant at a predetermined value. This type of controller performs slightly better compared to the differential controller but consumes more electric energy to circulate the fluid.

In the second simplest mode of operation when the building requires heating and solar collectors are able to supply the required heat, the room thermostat gives a signal to the control unit, which direct the flow of air directly from control through dampers, blower, auxiliary heater (Off) etc. to the building bypassing the storage unit and the cold air from the building returns to the solar collectors

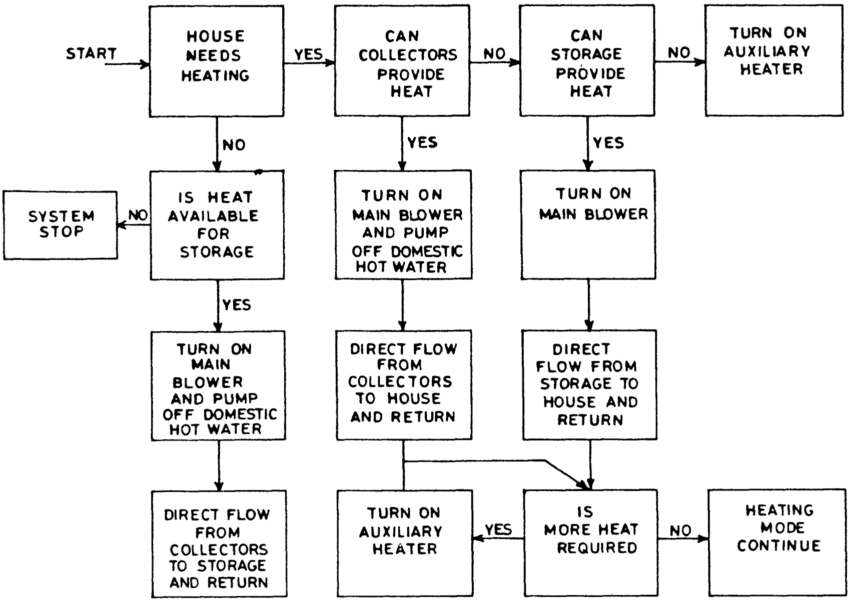


FIG.1.5 CONTROL MODES FOR A TYPICAL SOLAR AIR HEATING SYSTEM

through return duct. If the temperature in the building is achieved, the thermostat makes the circuit off and the first mode continues i.e. solar heat is stored in the storage tank. If solar heat is unable to maintain the building temperature, then the building temperature continues to fall until it reaches to the second lower set point in the thermostat which turns-on the auxiliary heater and the additional heat demand will be met by auxiliary heat.

In the third mode of operation, the stored solar heat is used for building heating. In this case when the building heating is required, the upper set point in the room thermostat gives a signal to the control unit which in turn operate the blower, the dampers, auxiliary heater (Off), etc. and the cold building air is sucked through the bottom of the rock bed storage systems, get heated and transferred from the top of storage unit to the building. If the required building temperature is met then the room thermostat through control unit will turn off the blower. However, if the hot air from the storage unit is not able to maintain the desired temperature in the building, then the building temperature will continue to fall, until the second lower set temperature of the room thermostat reached, which then turn on the auxiliary heater and maintain the building temperature. All these processes take place turn by turn e.g. the room thermostat will first try the solar heat directly from solar collectors to maintain the room temperature, if it is not sufficient then it will try the storage unit to supply necessary heat, and finally it will maintain the room temperature through auxiliary heat.

A schematic representation[92-94] of a typical liquid heating system with control positions is shown in fig.1.6. As is seen from this figure, there are five sensors T1, T2, T3, T4, and T5 located at the collector outlet, storage tank top, storage tank bottom, middle of preheat tank, and middle of room (load) respectively. The sensors T1 and T3 is a part of differential controller which turn-on and turn-off the pumps 1 and 2 at a preset temperature difference of 10 °C and 3 °C respectively. The ratio of turn-on and turn-off temperature difference should be 5:1 to 7:1.

The room thermostat T5 is a conventional two-stage thermostat of which the first stage (upper set temperature) controls the pump-3 and circulates liquid through storage tank and the radiant or convective panels inside the building. If the heat from the storage tank is not sufficient to maintain the room temperature then the second stage (lower set temperature) of the thermostat actuates the auxiliary heater which supplies the heat to the liquid circulating through the radiant or convective panels. The heated liquid from the solar collectors can also be directly supplied (not shown in fig.1.6) to the radiant or convective

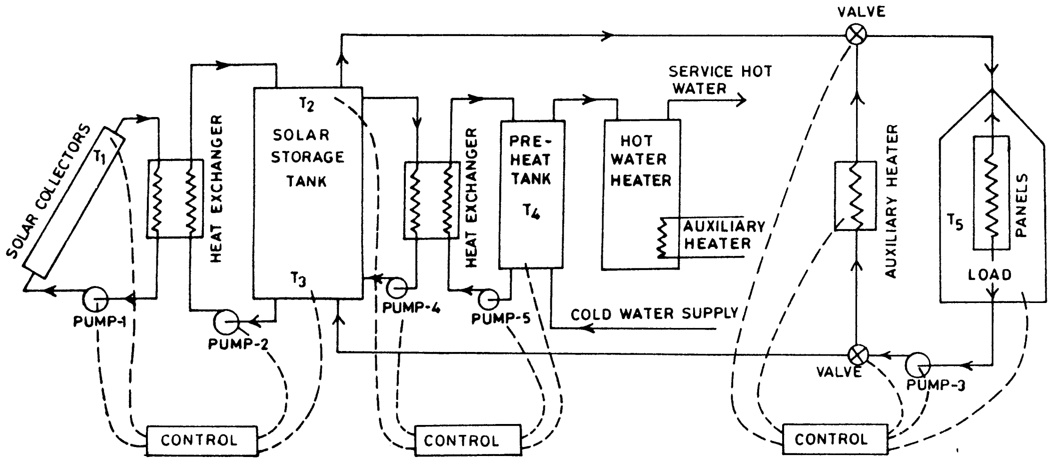


FIG.1.6 SCHEMATIC REPRESENTATION OF SOLAR LIQUID HEATING SYSTEM

panels in the building as is done in air heating system.

Finally the sensor T4 in the preheat tank controls the pump 4 and 5 which help in heating the water in the preheat tank. This preheat tank supplies hot water to the hot water heater where the water is further heated to the desired level by auxiliary heater. From hot water heater, hot water is supplied for household use. Thus there can be following five operational modes in any solar heating system.

1. Heating the space directly from the solar collectors.
2. Heating the space partly from solar collectors and additional heat supplied from auxiliary heater in case of low solar intensity.
3. Heating the space from storage.
4. Heating the space partly from storage and additional heat supplied by auxiliary heater.
5. Heating the space from auxiliary heater alone.

1.7 THREE WAYS OF SOLAR SPACE HEATING

As discussed earlier there are three distinct ways of solar heating (active) of space:

1. Solar air systems
2. Solar liquid systems
3. Solar heat pump systems

All the three systems have been widely tried and are in use for space heating. However, there are advantages and disadvantages of each system, and an appropriate system is to be selected based on the type of application, economics, geographical location, and the technology available. Salient features of each system will be discussed below.

1.7.1 Solar air systems

A typical solar air heating system is schematically shown in Fig. 1.4. As seen from this figure and discussed earlier also, a solar air heating system consists of solar air collectors, a storage system (generally rock bed system), auxiliary heater, automatic air dampers for directing air flow, air handing unit, blowers and pumps, control unit with necessary sensors, relays, etc., and heat distribution system in the rooms. If domestic hot water is also required then an air to water heat exchanger and hot water storage tank is also required. Details of air based solar systems for space heating are discussed by Lof[37] in depth and typical design parameters[35,46] for solar air heating system are listed in Table 1.6.

There are several advantages of air-based solar heating systems compared to liquid based systems. The main advantage in air system is that the same medium (air) is used for heat collection from solar air heaters (collectors) and space

Table 1.6 Solar air heating design parameters (From Reference 35 and 46)

Parameter	Value
Solar Collector	
1. Collector tilt from horizontal	Latitude $\pm$ 15°
2. Collector orientation	Due south $\pm$ 15° (In North hemisphere)
3. Number of glazings	1 or 2
4. Collector heat capacity	8 to 15 KJ/m^2 °C
5. Collector heat transfer coefficient from metal to air flowing inside	20 to 25 W/m^2 °C
6. Collector air flow rate	5 to 20 $liters/m^2s$
7. Collectors pressure drop	50 to 200 Pa
Thermal Storage	
1. Storage capacity	200 to 250 Kg $rocks/m^2$
2. Storage thermal capacity	180 to 250 KJ/m^2 °C
3. Rock size	2 to 5 cm
4. Storage length (flow direction)	1.5 to 2.5 m
5. Pressure drop in rock bed	60 Pa minimum
6. Maximum entry air velocity in rock bed	4 m/s
Air flow rate in heat distribution system	10 $liters/m^2$ s
Pressure drop in duct work	10 Pa

heating. Thus there is no need of a heat exchanger. The heat storage system consisting of small sized rocks acting as heat storage and heat exchanger and the high degree of stratification in the storage leads to lower inlet collector temperature resulting in higher collector efficiency. The combination of thermal and operating parameters of air like specific heat, flow rates, density, low inlet temperature results in higher collector outlet temperature. Apart from the above advantages, there are small-small other advantages also like, air systems are more durable, corrosion is not a problem, air leakage is not serious, air collectors can be made cheaply, control systems are readily available, most of the space heating systems use hot air and in air systems there is no freezing and boiling problems. There are disadvantages also with air based heating system such as: relatively high pumping cost since large volume of air is to be handled, relatively large volume of storage size and difficulty in combining with airconditioning systems.

1.7.2 Solar liquid systems

A typical solar liquid space heating system is schematically shown in fig.1.6. So far as the mode of operation is concerned both the liquid-based and air-based solar heating systems operate identically. Here in case of liquid-based system atleast three heat exchangers, one in the collector storage loop, second in the load and storage loop, and the third in the room in the load loop meant for dissipating heat into the room are used. Like air heating systems, the liquid systems also consist of liquid flat-plate collectors, storage system, auxiliary heater, radiant or convective panels, heat exchangers, pumps, valves, and automatic control unit. In liquid collectors, water or antifreeze liquid or some other heat transfer fluid is circulated and the building can be heated either by a hydronic system or air distribution system using liquid to air heat exchanger. The main advantage of a liquid heating system is that they are widely studied, used and can readily supply combined space heating and cooling and domestic hot water supply and therefore liquid systems are suitable for commercial buildings. For the same collector inlet temperatures, the liquid collectors operate at higher efficiencies and can provide domestic hot water efficiently compared to air collectors. The liquid systems occupy less space. The disadvantages in liquid systems are: freezing of collector water; corrosion at joints and collector tube and plate; boiling of water at some times; hazard due to leakage; air binding; high cost due to additional heat exchangers, leak proof joints, corrosion resistant metals, etc.; low overall efficiency due to several heat exchangers; and comparatively low durability.

The type and size of liquid collectors, storage system, heat distribution system, control etc. depend on the system heating load and geographical location. However typical design parameters[35,46] for a liquid-based solar heating system are listed in Table 1.7.

Table 1.7 Solar liquid heating design parameters (From Reference 35 and 46)

Parameter	Value
Solar Collector	
1. Collector tilt from horizontal	Latitude $\pm$ 15°
2. Collector tilt from horizontal	Due south $\pm$ 15° (In North hemisphere)
3. Number of glazings	1 or 2
4. Collector heat capacity	20 to 25 $KJ/m^2\ °C$
5. Collector heat transfer coefficient from liquid	150 to 250 $W/m^2\ °C$
6. Collector water flow rate	70 to 100 $liters/m^2 s$
7. Collectors heat exchanger	$F_R'/F_R > 0.9$
Thermal storage capacity	50 to 100 $litres/m^2$
Load heat exchanger	$1 < \varepsilon_L\ C_{min.}/(UA)_h < 5$
Design water temperature distribution	50 to 70 °C
Water preheat tank capacity	1.5 x capacity of water heater

1.7.3 Solar heat pump systems

Heat pump is a mechanical device which provides heating or cooling by using a reversible refrigeration cycle. In the heating mode, the heat pump extracts heat at low temperature from outside air and rejects this heat at higher temperature to the room air. In the cooling mode, a reversing valve reverses the roles of evaporator and condensor and therefore in this mode the heat is extracted from indoor air providing cooling and rejected to the outside air. If the source temperature increases, the coefficient of performance (COP) of the heat pump increases and therefore the solar assisted heat pumps may operate at higher efficiencies. The coefficient of performance (COP) of a heat pump in the heating mode can be as high as 4 which means that for every watt of electrical energy supplied to the heat pump in the heating mode, 4 watts of thermal energy will be transferred from outdoors to the indoors.

The idea of heat pump was given by Lord Kelvin in the 1850's who named this machine initially as 'heat multiplier' and he proposed this scheme for heating houses by extracting heat from the earth. Since then several hundred of papers and few books are written on heat pumps. These are generally known as refrigeration or cooling machines. Only during the last 30 years heat pumps are in use for heating the space also and only very recently[99-117] studies on solar assisted heat pumps are conducted.

Fig.1.7(a) shows a basic circuit of a simple heat pump which includes four main components: compressor, condenser, evaporator, working fluid generally one of the freons, and expander. In this figure a reversing valve is also shown by means of which the roles of the condenser and evaporator can be reversed. By using appropriate valves and controls the heating in winter and cooling in summer can be provided by the same unit.Most small commercial units use four-way solenoid valve as a reversing valve. Heat is continuously pumped from lower to higher temperature by periodic compression and decompression of the working fluid using compressor run by electricity. The working fluid in the gaseous state is compressed and sent to the condenser, where it gets cooled to liquify and releases large amount of heat to the surroundings. This liquified working fluid gets vapourised in passing through expansion valve. This evaporation in the evaporator requires large amount of heat which is drawn from the surroundings. This large amount of low-temperature heat is supplied to the evaporator by sourrounding air, underground water, earth, river water, waste water or solar heated air or water.

A simplified refrigeration/heat pump cycle can be best shown on Pressure-enthalpy diagram as is shown in fig. 1.7(b). Compressor at point B takes the saturated vapour compresses it to point C and in doing so because work is done increases its enthalpy, pressure and temperature. Now the high pressure vapour goes to condenser where the heat is rejected and the vapour comes to liquid form and this process is shown by line CD. This high pressure liquid in passing through the expansion valve at constant enthalpy converts into low pressuring liquid and its pressure and temperature gets reduced as shown by line DA. Now this low pressure liquid which contains about 75 percent liquid and 25 percent vapor enters to evaporator and gets evaporated for which heat is taken from outside (source) and this evaporation process is shown by line AB indicating the refrigeration effect. This working fluid takes heat, it get varourised, becomes low pressure vapour and then goes to the compressor and in this way cycle continues. When the evaporator temperature is in the range of 17-22 °C, a heat pump with a coefficient of performance of 3.5 can provide heat on condenser side at a temperature of 71-82 °C.

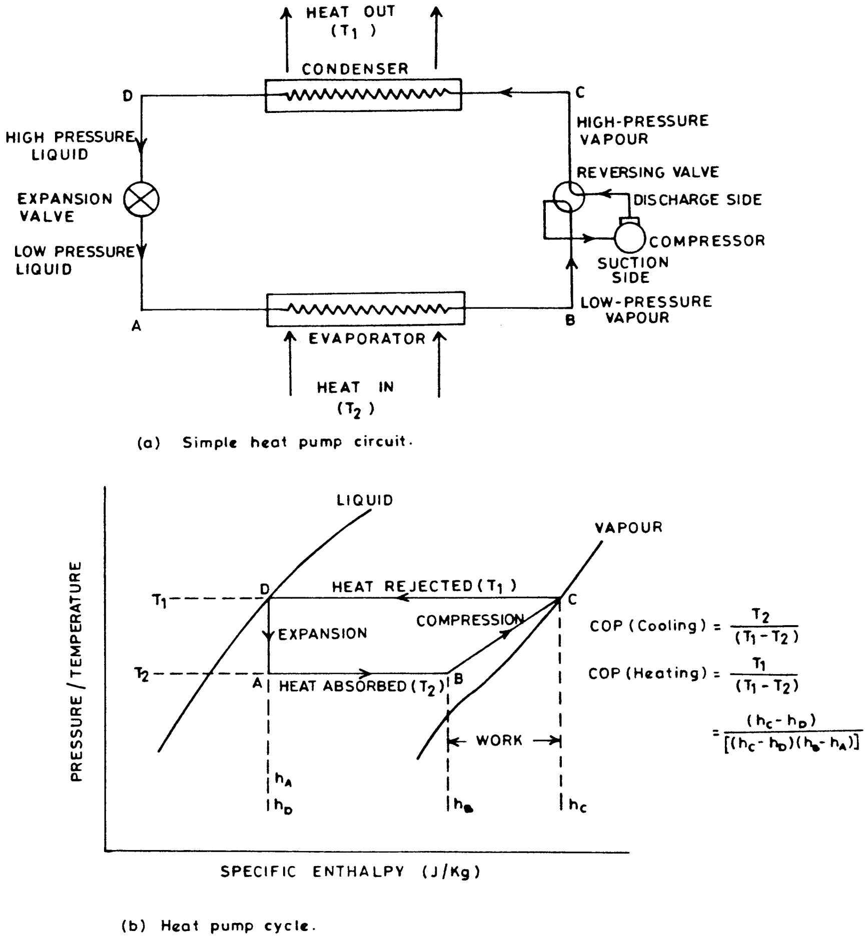


FIG.1.7 HEAT PUMP PRINCIPLES

A heat pump in combination with solar heated air or water is an economically attractive and viable system. In winter season, when the solar intensity is low, and the solar collected heat from solar flat-plate collectors is at a low temperature for space heating, and can be used as a source for the heat pump which will improve its performance considerably. At low collector inlet temperature, the collection efficiency will be higher, and at this higher collector outlet temperature the COP of heat pump will be higher, and therefore the combination of heat pump and solar supplied heat is a good combination. Even inexpensive solar air collectors can be used to supply heat and can improve the heat pump performance. This idea suggests the series combination of heat pump in which the evaporator of the heat pump gets heat from the solar heated air or water.

The heat pumps are often referred as air-to-air, air-to-water, air-to-earth, etc. according to the placement of evaporator and condenser respectively. But the most commonly heat pump system for domestic applications is air-to-air. Similarly solar assisted heat pump can be used either in parallel or in series and or dual source type, depending on the convenience, type of use and availability.

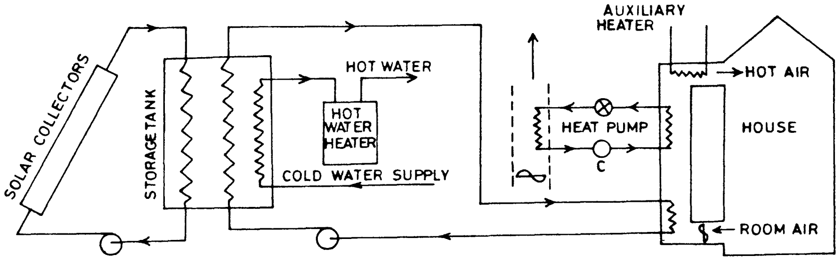


FIG.1.8 SCHEMATIC OF PARALLEL SOLAR HEAT PUMP SYSTEM

The simplest solar assisted air-to-air heat pump system as shown in fig.1.8 is the parallel system in which solar heating, heating hy heat pump, and auxiliary heating are independent and either of these three can be independently or unitedly used depending on the heating load of the house. Direct solar heating will be used whenever it is available and is able to supply required heat. Whenever the solar energy is not adequate, the heat pump, using ambient air as heat source, comes into operation and supplies necessary heat. Auxiliary heater can be pressed to service when neither of the source are adequate. In this parallel system, the heat pump does not take advantage of the solar heat and it works only as an auxiliary energy supplier.

Fig.1.9 shows a series system where the heat pump takes advantage of the solar heat and boosts the low temperature of the solar heat. Here the heat pump is placed between the solar system and the load. The evaporator of the heat pump can be placed directly in the solar storage tank or as shown in fig.1.9. Whenever the solar storage tank temperature reaches very high, the heat can directly be supplied to the load by bypassing the heat pump. The condenser or the heat pump supplies heat to the load. An auxiliary heater is also provided which can provide heat to the load if required. The series system has the advantage that it increases the COP of the heat pump by supplying solar heat to the evaporator and eliminates the need for coil defrosting and also the solar collector efficiency improves because the system operates at a lower temperature.

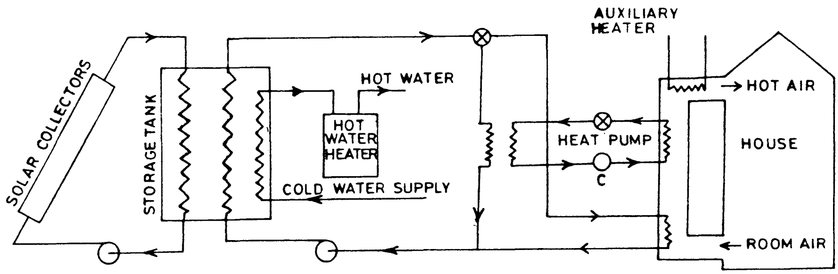


FIG.1.9 SCHEMATIC OF SERIES SOLAR HEAT PUMP SYSTEM.

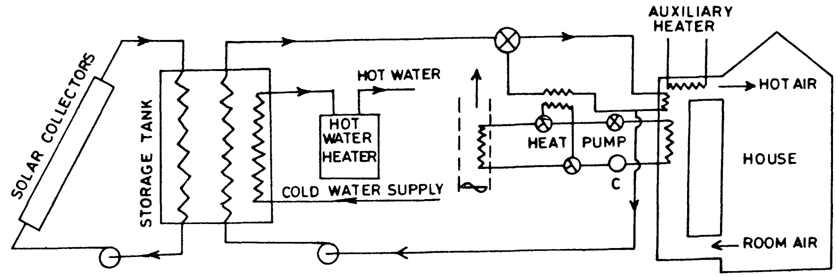


FIG.1.10 SCHEMATIC OF DUAL SOURCE SOLAR ASSITED HEAT PUMP SYSTEM

In a dual source system which is an improvement of the series and parallel system as shown in fig.1.10, the heat pump uses low evaporators one receiving heat from solar energy and another from ambient air. Here the heat pump uses either the collected solar energy or ambient air energy depending which gives higher COP. When the solar storage tank temperature is quite high (higher than predetermined control value) the building load is directly supplied by this stored heat, when the storage temperature drops from this control value, it supplies heat to the evaporator of the heat pump as is done in a series system. But if the storage temperature goes below the outside ambient air temperature the operation becomes like a parallel system. The auxiliary heater can be pressed into service if required. The service hot water can be supplied in all the three cases. It is seen that this dual source system takes all the benefits of series and parallel systems but requires complex controls and the system becomes expensive.

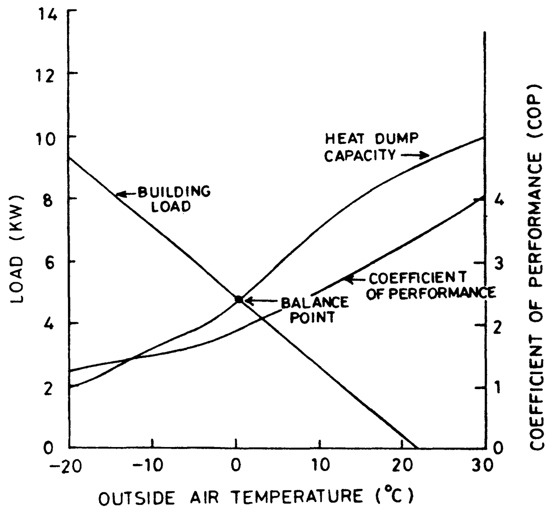


FIG. 1.11 AIR TO AIR HEAT PUMP DESIGN CURVE

A typical design curve for air-to-air heat pump is shown in fig.1.11. From this figure it is seen that as the outside air temperature increases, the building heating load decreases, but the coefficient of performance (COP) and the capacity of heat pump both increases. At the cross point known as balance point where the building load curve and heat pump capacity curve crosses, the building demand and heat pump supply matches. At lower temperatures below the balance point, the heat pump will not be able to supply the required heat and therefore supplimental heating is required. This is where the addition of solar energy can pay off. At higher temperatures above the balance point, the heat pump will be of higher capacity and therefore it is to be used only part of the time. Therefore, a air-to-air heat pump is so chosen that for a design outdoor temperature, the heat pump meets the requirement of the maximum heating load.

Several simulation studies[96,103,104,106,111,112,116] on the solar assisted heat pumps for house heating are carried out for series, parallel, and dual source heat pump systems for different climatic regions of the world. But the most comprehensive and widely used simulation program is the TRNSYS simulation program[118]. In a quasi steady state model developed to study the performance of solar assisted heat pump[103,104,116] a 3-ton air source unit is considered and a paramter F which is defined as the fraction of the total load that is met by the free energy is used. The energy balance of the whole system is:

$$Q_{load} = Q_{Solar} + Q_{Air} + E_{HP} + Q_{Aux}$$

where

Q_{load} – Total heating load of the building
Q_{Solar} = Heat supplied by solar energy
Q_{Air} = Heat extracted by heat pump from ambient air or water
E_{HP} = Electrical energy required by the heat compressor and fans, and
Q_{Aux} = Auxiliary energy required to meet the hot water and space heating loads.

The terms Q_{Solar} and Q_{Air} can be termed as free energy while E_{HP} and Q_{Aux} as purchased energy. Now the parameter F is given as:

$$F = \frac{\text{Free Energy}}{\text{Heating load}} = \frac{Q_{Solar} + Q_{Air}}{Q_{load}}$$

Now the values of F will be different for different combinations of heating systems and solar energy collectors. For a system where there is no heat pump and solar energy system, value of F will be zero. For a system where there is only a heat pump, F will be the ratio of Q_{Air} to Q_{load} only and in case F will be constant equal to 36 percent. The value of F for different heating systems and collector are determined[104] for a typical house in Madison, Wisconsin (USA) and the same are shown in fig.1.12.

These simulation studies are carried out for a residential house with 150 m^2 floor area, and an annual space heating load of 18000 KWhr and an annual water heating load

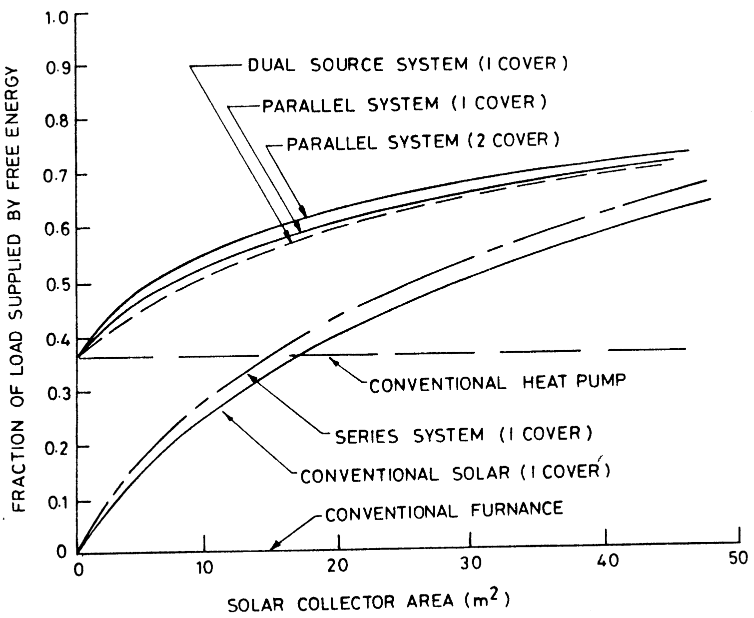


FIG.1.12 FRACTION OF HEATING LOAD FROM FREE ENERGY SOURCE FROM VARIOUS HEATING SYSTEMS(From Mitchell et al [104])

of 5800 KWhr in a climate like Madison, Wisconsin (USA). The solar collector properties are assumed to be independent of temperature and time and a solar storage tank with a capacity of 0.075 m^3 /(collector area)^2. From this figure it is seen as obvious that F is independent of collector area in case of conventional heat pump and conventional furnace. It is also seen that the minimum area required in case of conventional solar system to consume less auxiliary energy than a conventional heat pump is about 17 m^2.The performance of a series system is better than the conventional solar system at all collector areas and is zero in both cases for zero collector area. For smaller collector areas say below 30m^2 , the solar energy will not be able to supply heat to the heat pump and it will be working at its lowest COP. In this case, the storage temperature will not be at a usable temperature and more heat will be supplied by heat pump only. As the collector area is increased, the difference between the series and conventional solar system becomes constant which is about 10 percent. It is seen that with a series heat pump about 60 to 70 percent of the heat load can be met by these free energies.

The performance of parallel system is better than the dual system, the series system, and also the conventional solar system. In a parallel system, if the area is increased from 0 to 20 m^2, the fraction of the free energy supplied increases from 40 to 60 percent. The performance further increases with collector area but not at a faster rate as in case of series system or conventional solar system and this fraction becomes 80 percent at a collector area of 60 m^2. When the collector area is large, most of the load is met by the solar and less by heat pump and therefore the dual system will perform just like a conventional solar system and therefore in such cases the cost of heat pump is not justified. Inspite of lower collector efficiency and lower COP of heat pump, the parallel system performs better than the series and dual source systems. This is due to the fact the heat pumps in series and dual source systems must operate to deliver all stored solar energy below 20 °C and therefore the extra electerical energy required to deliver this energy more than compensates than the advantages obtained due to higher collector efficiency and higher heat pump COP. In case of a series system, there is hardly any heat which can usefully be added to the space for heating.

It is seen above that solar, heat pump and solar assisted heat pumps all save energy compared to conventional furnaces. A combined solar heat pump system should be designed in such a way that the use of auxiliary energy is minimum. The combined heat pump system will be more effective and economical at those places where either the electricity is very expensive, cooling is required or it can improve the collector efficiency. However, if cheap but efficient collectors are produced then the series heat pump design can be used economically.

From the above, it appears that from economic considerations, there is hardly any justification in using a combined solar and heat pump system compared to pure solar and conventional heat pump systems separately.

1.8 SOLAR HEATING PRACTICAL SYSTEMS

As discussed earlier several thousand practical solar heating systems are installed all over the world for demonstration, simulation, experimentation, and for actual use. All these houses use some kind of auxiliary heating arrangement for making the heating system fool proof and completely dependable. These houses use either liquid collectors, or air collectors, or evacuated tube collectors with various kinds of auxiliary heating and cooling combinations. As an example, design and performance of few typical solar houses which are architecturally or design wise are different are described here:

1.8.1 The MIT Solar House

Systematic studies on Solar Heating began in 1938 at the Massachussetts Institute of Technology (MIT), Cambridge, Massachussetts under the leadership of Prof. H.C. Hottel. Prof. Hottel and his associates designed[11,12,119,120] several Solar Houses known as MIT House No. I,II,III, and IV. These houses were built in the year 1939, 1947, 1949, and 1958 respectively. The MIT Solar House No. I built in 1939 was of two-rooms and approximate living area of 46.5m^2. Flat Plate collectors of about 33.45 m^2 area consisting of blackened copper sheet to which copper pipes are welded were installed on the roof slopped about 30 southward. A large hot water storage tank of 65.8 m^3 capacity was kept in the basement of the house. This was the first 100 percent solar heated house, but the storage size was bigger than required and was enough to accumulate summer heat for winter use and therefore this system was highly uneconomical and was used only for test purposes. Extensive instrumentation provided accurate determination of solar collector performance and of the house.

The second house (MIT House No. II) constructed in 1947 was bungalow-style laboratory building divided into seven cubicles with south windows, each provided with a different type of collector and storage system directly associated with the window. Each collector was having an area of about 10m^2. The dimensions of the house were about 4.28 x 13.40 x 2.44 m. With this type of combinations various type of storage systems, several type of shading devices, etc. were compared. Later during the year 1947-49 this house No. II was converted into house No. III.

The solar house No. III was the same structure, enlarged and remodelled for occupancy as a dwelling. The solar collectors used have the same specifications as in House No. I but with an area of 37.2 m^2, double glazed, and installed at the roof with a tilt of 57° to the horizontal. Under the roof, a 4.5 m^3 hot water cylindrical storage tank was provided. Auxiliary Energy was provided using electric resistance heating. More than 80 percent of the heating load was supplied by solar energy. This house was destroyed in fire during the year 1955.

One of the most modern solar house, making fullest use of solar energy availability and to conserve as much of energy as possible to provide comfort, was built[12] during the year 1958-59. This house known as MIT house IV is a two storey building with a living space of about 135 m^2. The south side of the house is completely occupied by 60 solar collector penels each of 1.21 m high and 0.81 m wide with a total area of 59 m^2 at an angle of 60° to the horrizontal. The house had a functional structure and pleasant look. The heating system as schematically shown[12] in fig.1.13, consists of an array of flat plate liquid collectors mounted on large slopping roof, a large storage tank (main tank), oil fired auxiliary furnace, auxiliary tank, and hot air supply system to the rooms.

The liquid flat plate collectors with a total area of about 59 m^2 consist of black painted aluminium absorbing surface on which copper tubes are mechanically clipped, double glazed with low iron content glass, and fibre glass insulation on the back. The overall measured absorptivity of the absorbing plate being 0.97, overall heat loss coefficient being 3.97 $w/m^2\,°C$, and heat removal efficiency factor being 0.86. The water is pumped through the collectors and storage tank and hot water is stored in a hot water storage tank of capacity 5670 litres.. An expansion tank of 760 litres capacity is also used which helps in draining the collectors when these are not in use. An auxiliary water heater storage tank of 1041 litres capacity is also used.

Domestic hot water is also provided with this system by passing the city main water through the coils passing in series through a coil in the main storage tank and also through a coil in the auxiliary tank. If the temperature in the main tank is quite high then the water will pass through the coil in it and then through the coil in the auxiliary tank. If the temperature of the water in the large tank is not enough then the water bypasses it and goes straight to the auxiliary tank for heating and then mixed with the main water to obtain the desired temperature (60 °C).

A differencial controller is used to circulate the water through the collectors and the main storage tank. One sensing element of the differential controller is fitted at the collector plate and the second sensing element in the storage tank. A pre-set temperature difference between collector outlet and storage tank which is generally 6 °C will 'turn on' the collector pump and 'turn off' when the temperature difference drops to 2 °C. To control the space heating, a two-set point room thermostat is used which calls the energy either from the main storage tank if the heat is sufficient to meet the heating load of the house or from the auxiliary storage tank if the main storage tank is not able to meet the heating load.

Various parameters like temperature at various points in the house as well as at the collector array, flow rates in the ducts, electric energy consumption, solar radiation, etc. were measured to determine the energy balance on all the components as well as on the system as a function of time. Thorough analysis of system performance for two winter seasons i.e. 1959-60 and 1960-61 was carried out and the integrated energy balance for these two heating seasons is

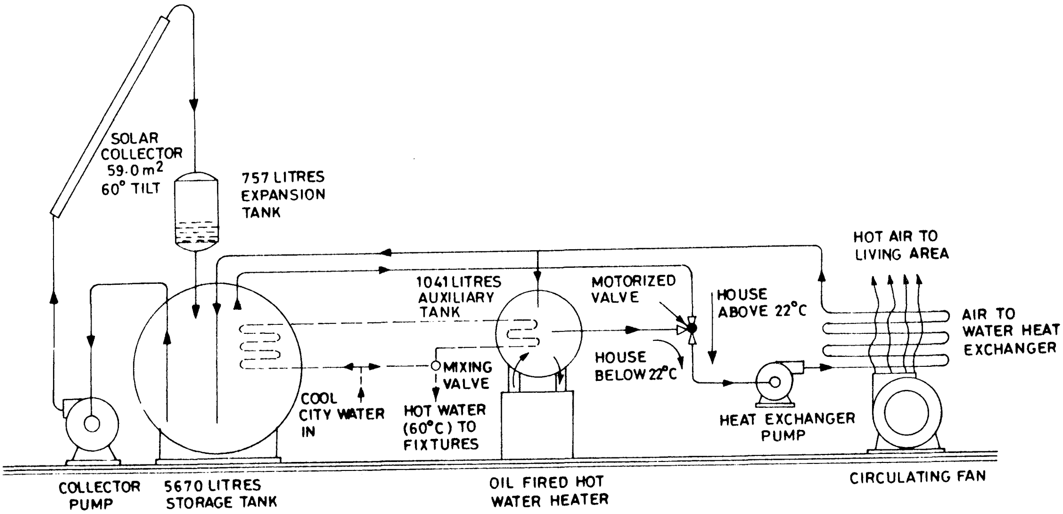


FIG.1.13 SCHEMATIC OF SOLAR SYSTEM FOR MIT HOUSE IV (From Enebretson[12])

Table 1.8 Energy Balance on MIT House IV during the two heating seasons 1959-60 and 1960-61

1.	Total solar radiation available on 60 deg	238	GJ
2.	Useful heat collected	102	GJ
3.	Net collector efficiency	43	%
4.	Space heating demand	143.2	GJ
5.	Water heating demand	32.2	GJ
6.	Total Heating demand	175.5	GJ
7.	Heat supplied for space heating by solar	73.8	GJ
8.	Heat supplied for space	18.1	GJ
9.	Total heat supplied by solar energy	91.9	GJ
10.	Percent of heat load supplied by solar (included for water heating).	52.5	%

shown in table 1.8. From this table it is seen that the net collector efficiency is of the order of 43 percent. During these two heating seasons 56 percent of the domestic hot water load and 52 percent of the space heating load was met by the solar energy system. About 52.5 percent of the total heating load including for domestic hot water supply is met by solar energy which appears to be quite low. This may be due to severe winter conditions experienced during these two years and more particularly during the year 1959-60. This system was to be abandoned after two years due to large maintenance problems.

1.8.2 The Colorado Solar House

A nine room residence near Denver, Colorado, and its associated Solar Heating System was completed during 1958 to 1959. This house was earlier designed to be used without any active solar heating systems but the solar system was later incorporated into its final designs. The house is of contemporary architectural[17,18,121] style flat roof having about 296 m^2 of living area; 194 m^2 on the main level and 102 m in the basement. Large window areas are employed to admit the direct sun in winters. In the house, two banks of solar collectors (air type), two vertical heat storage cylinders, hot water pre-heater, natural gas furnace, blower, control equipment, air ducts, etc. are used. At an outdoor design temperature of -18 °C and wind speed of 3.9 m/sec., the heating load for the house was computed as about 31.8 KW.

Figure 1.14 shows[121] the complete heating system for this house. The solar collector is an air heating type with the overlapped glass plate construction. Two collector banks each consisting of 10 cold penels and 10 hot panels located alternatively are used. The total area of the two collector banks containing 40 collectors is 55.74 m^2 of which about 49.24 m^2 is effective heat collection area. These collectors are installed at a slope of 45° from the roof. The cold panels have a single cover glass, 0.76 m x 1.83 m, whereas double cover plate are used on the hot panels. Each panel has 6 black plates 3.05 x 6.75 m. and seven clear glass plates, 5 of which are 6.1 x 6.75 m and two are 3.05 x 6.75 m. They are made out of ordinary window glass of 2.5 mm thick and spaced 6 mm apart.

The heat storage unit consist of two fibreboard cylinders each of 0.91 m in diameter and 5.5 m high extending from the basement floor to the roof. In the centre of the cylinder, a 28 cm diameter duct extends from the top to the bottom. These cylinders are filled with about 10650 kg of mixture of crushed and uncrushed rock, primarily granite with a specific heat of 0.75KJ/kg °C, closely sized to 25 to 40 mm equivelent diameter. The storage unit is charged by allowing solar heated air to the bottom of the cylinders. For heating purpose the direction is reversed and heated air is withdrawn from the bottom plenum chambers.

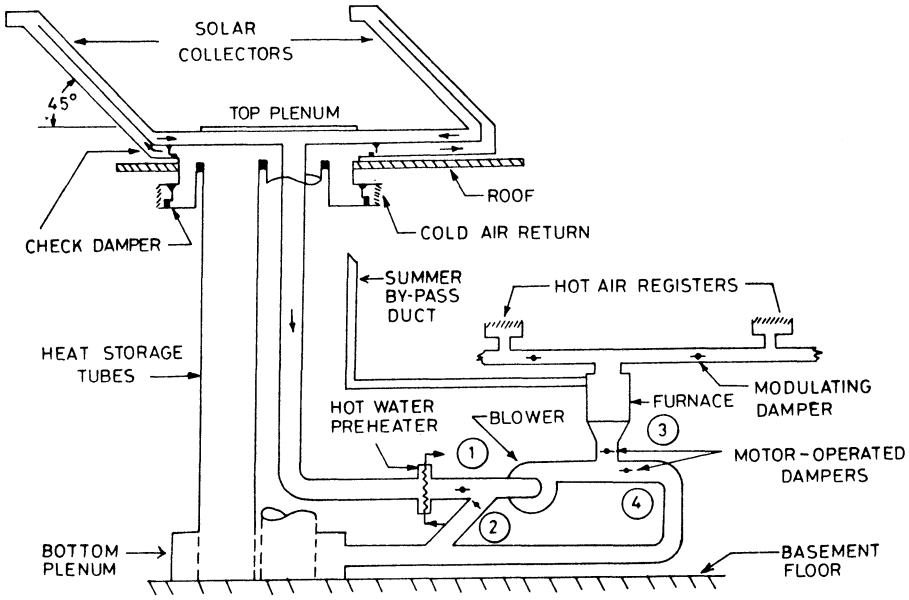


FIG.1.14 SCHEMATIC OF COLORADO SOLAR SPACE HEATING SYSTEM (From Lof et al[121])

For the supply of service hot water for domestic purposes an air to water heat exchanger is used which is connected in the hot air duct entering the blower, and is connected to 300 litre water storage tank. This tank serves as pre-heater and the balance heat required is met by a conventional fuel-fired heater.

A centrifugal blower with a 23 cm inpeller is driven with a 2-speed motor of 1120 watts. The blower can supply air at low and high speed. A natural gas furnace rated at 168800 KJ output was used which can supply the gas at two rates i.e. 5.2 m^3/hr. and 2.65 m^3/hr. The blower is controlled based on the temperature difference between the collector plate and the heat storage unit which is senced by two resistance thermometers, one placed on the metal plate of the air collector and the other at the top of the heat storage unit.

There are two ducts with variable flow dampers which supply air to the rooms in the two different zones. A double contact thermostat in each zone controls the blower, the main dampers, and one of the modulating dampers. The first contact of the thermostat closes if the temperature in one of the zones decreases below the set-point and the appropriate modulation damper opens slowly. At this stage the two main dampers allow the air to flow from the collector or the storage unit and the blower motor is run at the low speed if the air is supplied through the collector or a high speed if it is through the storage. If the heating demand of the rooms is not met for 10 to 15 minutes then the hot air is supplied from the furnace first at the lower rate. Even if at this rate the room temperature is not met, the second thermostat contact closes, and the gas to the furnace is supplied at a high rate and meets the heating load. Four swing type dampers are used to check the reverse flow of air and to stop the air leakage.

The heating system can operate in any of the following modes;

1. House heating with the collectors alone. In this mode of operation the air route will be: from collectors to vertical duct to water preheater to blower to furnace to hot air registers to rooms to cold air return grillers to top plenum chamber and then back to collectors.
2. When house heating is not required then the solar heat can be stored in the storage cylinders. Here the air route will be: from collectors to vertical duct to water preheater to blower to bottom plenum chamber to storage unit to top plenum chamber and then back to collectors.
3. If the solar collectors are not able to supply heat to the house then the stored heat can be used for house heating. In this case the air route will be: from rooms to cold air return grilles to top plenum chamber to storage unit to bottom plenum chamber to blower to furnace to hot air registers and then back to rooms.

In summer the solar energy is used for water heating only. In this case the air route will be: ambient air to top plenum chamber to collectors to vertical duct to water preheater to blower to furnace to summer bypass duct to top plenum chamber.

This house was completely monitored from September 1959 to 1960 for its thermal performance. Parameters like solar radiation; ambient air temperature; wind speed; air and room temperatures at various positions of duct, storage cylinder, rooms etc.; air flow rates in the ducts and air pressure; and gas and electricity consumption were continuously recorded for the above period. The results of these measurements for the period from 18 September 1959 to 10 June 1960 are shown in table 1.9. From this table it is seen that the solar heating system is undersized and meets only 25.7

percent of the house heating load including water preheating and heating.

The monthly variation of useful solar heat collected [121] for the Colorado Solar House is shown in fig.1.15. It is seen from this figure that in winter months, the heating load requirement of the house was even greater than the total solar radiation available on the solar collectors and it is much more than the collected heat. In spring and early falls only the solar radiation on the collector was more then the heating load. These figures very clearly indicate that small collector area was used compared to the heating demand of the house.

The Colorado Solar House was tested for its performance during the year 1959-1960 and also again during the heating season of 1974-75 with no maintenance and the results are reported in a paper by Ward and Lof[18]. It is reported that even after 15 years of continuous operation and without much maintenance the system is not deteorated from stability and performance point of view and after 15 years the solar output remained 72 percent of its initial value. From this study it can be concluded that the house should be properly designed and the solar system should be appropriately sized to meet the heating load and the solar air heaters are more durable and can work without maintenance for longer ruration.

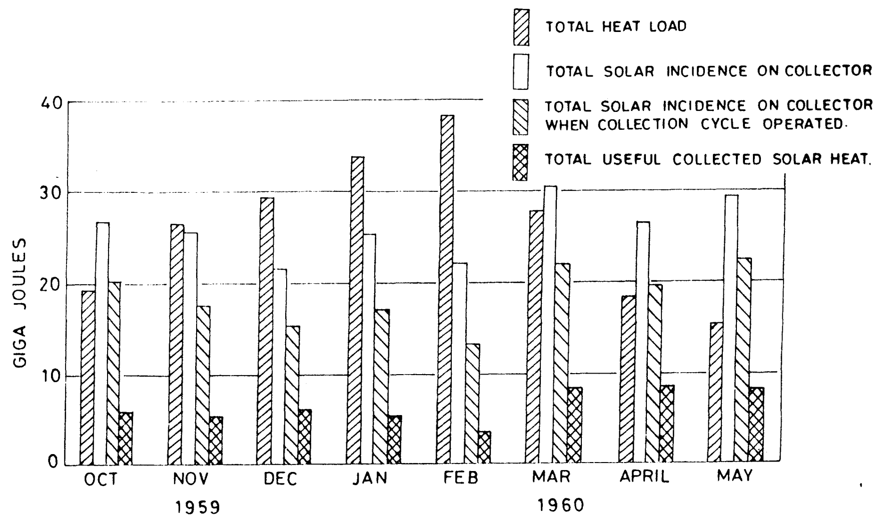


FIG.1.15 HEAT UTILIZATION IN COLORADO HOUSE (From Lof et al[211])

Table 1.9 Energy balance in Colorado Solar House During Winter 1959-1960 (From Lof et al[121])

		Energy GJ
1.	Total solar incidence on 45 deg collector area (55.74 m^2)	239
2.	Total solar incidence available on 45 deg. collector area when collection cycle operated (55.74 m^2)	170
3.	Useful heat collected	59
4.	Net Collector efficiency (percent)	35
5.	Solar heat absorbed by storage	27
6.	Solar heat absorbed by water preheater	4
7.	Heat delivered by natural gas for house heating	150
8.	Heat delivered by natural gas for water heating	22
9.	Total heat load	230
10.	Percent of useful collected heat absored by water preheater	7.09
11.	Percent of total water heating load supplied by solar energy	16.25
12.	Percent of house heat load supplied by solar energy (including water preheating water heating).	28.20
13.	Percent of house heat load supplied by solar energy (including both water preheating and heating)	25.7

1.8.3. The Thomason House

Thomason designed and built several solar heated houses [14, 15, 122, 123] in USA. The first house was built during the year 1959 in Washington, D.C. with a floor area of

139 m^2 and solar collectors of area 78 m^2 installed on the roof and sloping south wall. The basic system in the heating mode is shown in fig.1.16. Special 'Trickle type' collectors consisting of blackened corrugated aluminium sheet placed over an 7.5 cm thick rock wool and glazed with one or two layers of plastic film and glass are used. The corrugations in the aluminium sheet were spaced 3.2 cm. apart and were running vertically from top to bottom. The main hot water storage tank consists of galvanised steel with a capacity of 6.1 m^3 and is surrounded by 50 tons of small 10 cm diameter rocks. Whenever the sun is shining, the water from the main hot water storage tank is pumped to the top of collectors and flows along the corrugations in thin film. During its passage it gets heated due to solar radiation and this hot water is collected at the bottom in channel from where it flows to a 1040 litres domestic water preheater tank and then to the main water storage tank. When the house needs heating, the thermostat automatically

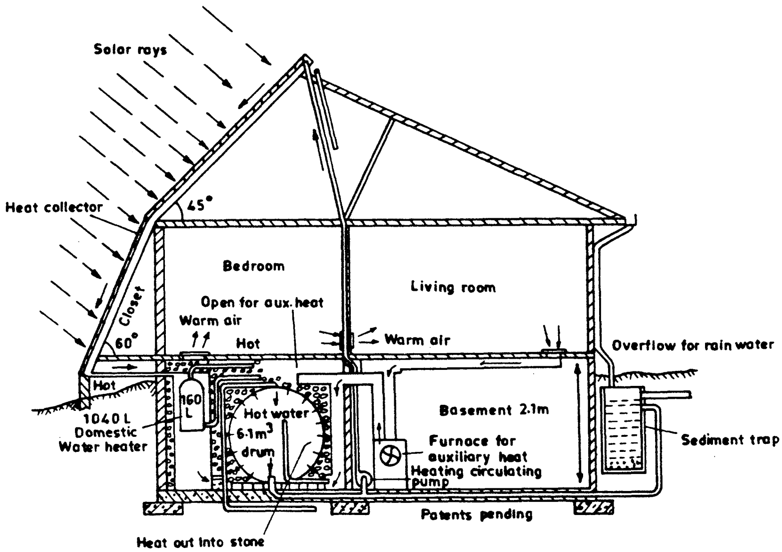


FIG.1.16 THOMASON SOLAR HOUSE NO.1 IN THE HEATING MODE. (From Thomason and Thomason[15])

starts a 200 watt blower which blows air through the warmed stones and around warm water tank and thereby heating the air. This heated air is directly supplied to the living rooms. When the heating is not required, the thermostat stops the blower and the solar heat is stored in the hot water and stored for future use. If the heat supply is not adequate, the storage system is bypassed, and an auxiliary oil furnace supplies heat to the air stream.

The day-night temperature difference is made use for providing cooling in summer. In summer nights, water is directed to flow over the unglazed north facing roof channels, which gets cooled due to evaporation, convection, and radiation, and is stored in the water storage tank from where the stones get cooled. The living rooms get cooled in summer by passing the room hot air through the stones (storage bin) and then to the living rooms. This is automatically done by a reverse acting thermostat controlling the blower. This system is shown[15] in fig.1.17.

Performance data on this system has indicated that 95 percent of the heating load is met by solar energy alone and the rest 5 percent is met by 190 litres of fuel oil.

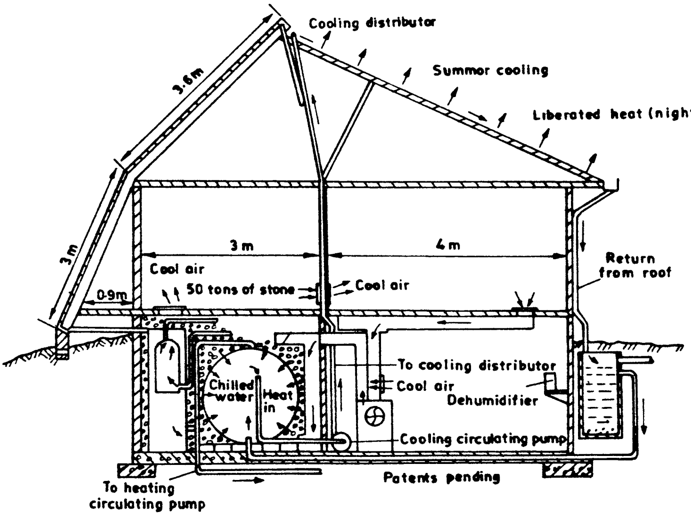


FIG 1.17 THOMASON SOLAR HOUSE NO.1 IN THE COOLING MODE (From Thomason and Thomason[15])

The basic features of solar house No.2 built in 1960 and 1961 are the same as of solar house No.1 except that it had collector area of $52m^2$ and living area required heating of $63m^2$. However, in this design, ordinary aluminium relectors of $31m^3$ area extending from the base of the south facing collector is used which increased the solar heat collection by about 15 percent. Initially the solar reflectivity of this reflector was 70 percent and after 10 years of use the solar reflectivity remained 30 percent. For summer cooling, additional cooling is produced by using a 560 watt refrigeration compressor which chills the tank of water and stores at night. This is done by putting a large length of copper tube in the tank and used as the evaporator coil for the cooling system.

In the solar house No.3, all the solar collectors were installed on the roof of the house and therefore the winter sun shines directly into living room and swimming pool windows on the south side. In this design instead of a galvanised steel water storage tank, a tank made out of concrete blocks is used making it inexpensive and more durable. For cooling operations, a standard 2240 watts compressor unit is used and the cooling coil is placed in the duct work ahead of the heat (cold) storage bin. Here the air gets dehumidified and chilled and then cold is stored in the storage bin. This cold dry air chills and dry the stones while the cold air temperature is moderated and becomes only cool. This cooled and dehumidified air is then supplied to the space for cooling. Like this, several other houses were built with some minor modifications in the solar part like collector material was different in few cases, collector itself served as roof in some other cases, storage tank has coil for providing auxiliary heat directly into the storage tank, etc.

1.8.4 Colorado State University Solar House, CSU Solar House I

On receiving a grant from National Science Foundation in September 1973, the Solar Energy Applications Laboratory of Colorado State University started a comprehensive programme on the design, construction, and testing of a Residential Solar Heating and Cooling System. As a result four Solar Houses known as CSU-I, CSU-II, CSU-III, and CSU-IV were built[25,26,124-126] in the campus of Colorado State University in Fort Collins, Colorado, USA. Three of the houses are of identical construction having same floor area but each is heated and cooled by different type of solar system. The CSU solar house-IV is smaller than other three houses and is a dwelling and greenhouse combination.

The solar heating and cooling system in CSU Solar House I became operational[124] on July 1, 1974. This house was designed with many new idea of energy savings such as very few and small north and west windows and south window provided with shading devices such that the windows remain exposed in winter while shaded in summer. The house is of modern construction with three bed rooms, with a total floor area of 140 m^2 each for the main level as well as the heated basement. The slope of the south roof is 45° from horizontal and is having an area of 93 m^2. The area of the roof facing north is about 120 m^2. The walls and ceiling are insulated with fibre glass insulation of about 9.0 cm and 14.0 cm thickness respectively. The design heating load of the house is 16.1 KW at -23 °C and the design cooling load is about 10.5 KW. Both floor of the house (living and basement) are heated and cooled using solar energy. The south wall of the basement is above grade while the north wall is below grade.

The solar heating and cooling system used in the house is schematically shown in[125] fig.1.18. It uses an array of liquid flat-plate collectors, main water storage tank, lithium bromide absorption cooling unit, gas-fired auxiliary water boiler, and associated piping, ducts, pumps, blower, etc.

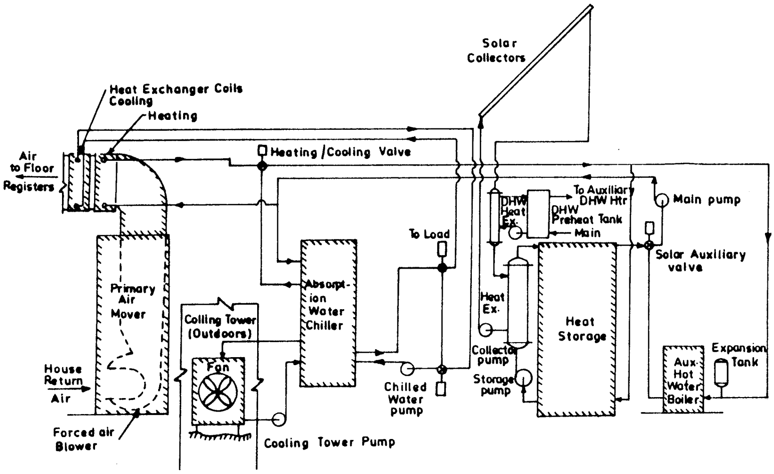


FIG 1.18 SCHEMATIC DIAGRAM OF SOLAR HEATING AND COOLING SYSTEM IN CSU SOLAR HOUSE I (From Lof[125])

Sixteen liquid flat plate collectors, each 0.9 m wide and 4.9 m long with a total area of 71.0 m^2 and absorber area of 67 m^2 are used and installed on the roof facing south at an angle of 45°. The solar collector consists of aluminium absorber plates with built in expanded tubes known as 'Roll-Bond panels', with a non selective paint, double glazing of glass, and fibre glass insulation on the rear side. A flow rate of 3632 litre per second through the sixteen collectors is maintained. The performance data on these site built collectors show $F_R U_L$ of 3.3 W/m^2 °C and F $(\zeta\alpha)_e$ of 0.58 when the wind speed is about 0.3 m/s. During November and December, 1976 another collector array of evacuated tube collector (made by Corning Glass Works) was used adjacent to the solar house. In this collector array 36 modules in which each module is having 6 evacuated tubes with a total absorber area of 1.11 m^2 are used. When mounted on the test bed, the gross area required for collectors and manifolds is 75.2 m^2 although the total absorber area is 39.9 m^2. A separate storage tank of 4275 litres capacity is used for evacuated tube collectors. The pipings etc. in the heating system are arranged in such a fashion that either of the two collector array i.e. the liquid flat plate collector or evacuated tube collector can be used and can deliver heat to the same space of the house. In October and November 1976 the liquid flat plate collectors are used while in subsequent months the evacuated tube collectors are put to service. The evacuated tube collectors had $F_R U_L$ of 2.03W/m^2 °C and $F_R (\zeta\alpha)_n$ of 0.97 based on the absorber area. A 60-40 mixture (by weight) of Elthylene Glycol (with Corrosion inhibitors) and water is used as working fluid which flows through the collectors and is separated from the storage tank fluid with a counterflow heat exchanger.

The energy is stored in water in storage tank made of 16 gauge, vertical galvanized steel cyliner, 1.83 m high and 1.68 m in diameter. The capacity of the storage tank is about 4275 litres. The tank is heavily insulated by 15 cm of fibre glass on the sides and 5 cm of fibre glass and 15 cm of concrete block tank supports. The service hot water is pre-heated in a 300 litre hot water tank and transferred on demand to a standard 150 litre gas-fired hot water heater. Cold water from the main enters the pre-heater tank and is heated through a heat exchanger by the main tank.

Single pass, counterflow, shell and tube heat exchanger are used to transfer the heat from the collector to the main storage tank and to the preheater tank for domestic use[126]. The collector heat exchanger is two units in series mounted horizontally in front of the main tank while the single service hot water heat exchanger is positioned vertically between the main storatge and pre-heat tanks.

An Arkla Solaire-3 ton nominal capacity LiBr-H_2O absorption air conditioner is used for providing cooling in the house. Hot water either from the main storage tank or auxiliary is used as a source of heat to the generator and a cooling tower outside the house provides the source of cold water for the absorber and condensor. The chiller has a capacity of 50 MJ/hr and a coefficient of performance of about 0.8 and it can be operated at generator temperature as low as 66 °C with a corresponding capacity of 20 MJ/hr.

The auxiliary gas fired hot water boiler is sized such that it can alone meet the total heating demand. Several centrifugal pumps which run at a constant speed of 1750 rpm are used in the system. Only one pump is in the ethylene glycol loop while others are in the water loops. The air handling unit is located in the enclosure of absorption airconditioner and is rated at 1200 cfm.

There are only two modes of operation for heating the house. If the room temperature drops by 1 °C below a desired level, mode 1 comes into operation, in which the entire heat is supplied through the main storage tank heated by solar energy. If the room temperature continues to fall further, the mode II comes into operation in which case the entire heating load is supplied by auxiliary source. A few limited experiments are also conducted for mode II operation in which even the low temperature solar heat stored in the main storage tank is to preheat the air and the remaining heat is boosted by auxiliary source.

The absorption air-conditoner also works in two modes. In mode I, if the room air temperature rises 1 °C above the desired level and storage tank temperature is above 83 °C, the necessary heat to the generator is supplied by solar energy.If the room temperature continues to rise and storage temperature drops below 83 °C, the necessary energy to the generator is supplied by the auxiliary system. Flow of hot water in the heat exchanger of preheat water tank starts only when its temperature is below 41 °C and the main tank temperature is 11 °C warmer than the preheat tank.

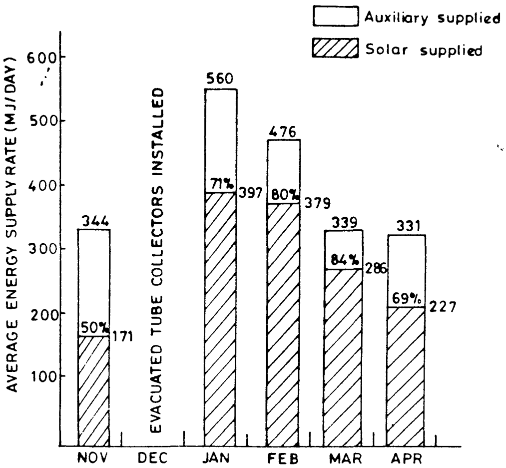


FIG.1.19 SOLAR AND AUXILIARY CONTRIBUTION TO TOTAL SPACE HEATING AND DHW HEATING, SOLAR HOUSE I, 1976-77 HEATING SEASON. (From Lof[125])

The results of monthly means of energy supplied by solar and auxiliary source for space heating and domestic hot water heating during the heating season of 1976-77 are shown[125] in fig.1.19. In the months of October and November 1976, flat plate collectors were used while evacuated tube collectors were used in all other months. In November 1976, the performance of the heating system was quite low i.e. only 50 percent of the heating load was met by solar energy while in earlier years in November only more than 80 percent load was met by solar energy. This exceptional behaviour is due to very bad weather during November 1976. The evacuated tube collectors have improved the solar contribution considerably. It is seen that out of 51127 MJ required for space heating and domestic hot water supply for January-April period, about 38595 MJ is provided by solar, making the solar contribution as 75.5 percent. The results of performance of cooling system are summarised[125] in fig.1.20.

The data on solar cooling is not extensive. It is seen that although the cooling load varies considerably from month to month the cooling load supplied by solar energy is always more than 40 percent. This poor performance and high cooling load of the house is due to the heavy heat losses from the storage tank, high electric consumption by equipments, and high heat generation in rooms due to several

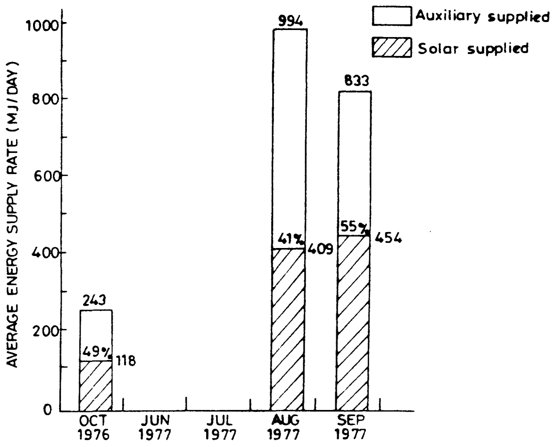
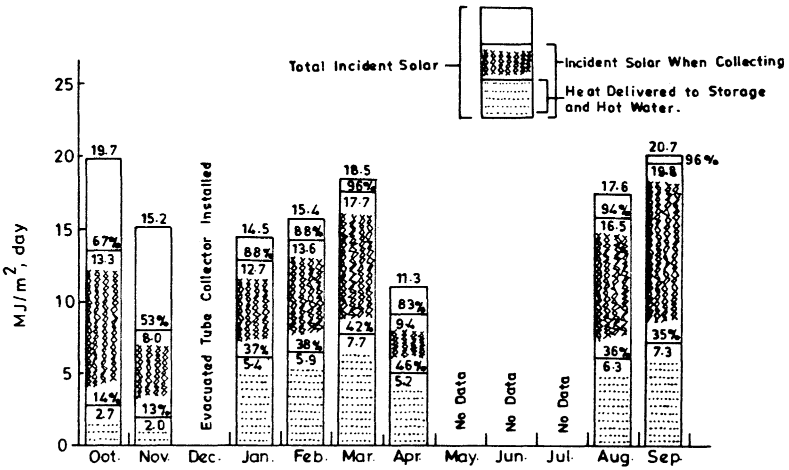


FIG.1.20 SOLAR AND AUXILIARY CONTRIBUTION TO SPACE COOLING AND HOT WATER HEATING, SOLAR HOUSE I, 1976-77 COOLING SEASON (From Lof[125])



Based on 71·3m^2 flat plate collector area and 75·2 m^2 evacuated tube collector area both equal to total occupied area including space for mani folds.

FIG.1.21 COLLECTOR PERFORMANCE LIQUID SYSTEM, SOLAR HOUSE 1 (From Lof[125])

reasons. The performance of two types of collectors: flat-plate liquid collectors, and evacuated tube collectors is compared[125] in fig.1.21. From this figure it is seen that evacuated tube collectors show high collection efficiency and supply more heating and cooling load to the house compared to conventional liquid flat-plate collectors.

1.8.5 Lorriman Solar House

A single family, two-storey, three bed room solar assisted house was designed by Doug Lorriman[127-129] and the same was built at Mississauga, Ontario (Canada) in November 1975. The heating system mainly consists of flat-plate liquid collectors, two storage tanks, solar water preheater for domestic hot water supply, a solar assisted heat pump system, electric auxiliary heating system, automatic controls, ducts, pumps, blowers, necessary heat exchangers,etc. (fig.1.22).

The house is a 2-storey, 3-bed room, 2-bath rooms, and single car attached garage with a total heated floor area of 184 m2. The complete house was well insulated and was having double glazed windows with area of 20.9 m2, 70 percent of which is facing south. The heating load was estimated as 35000 KW hr/year.

Thirty three liquid flat-plate collectors (each 2.2 x 0.9 m) supplied by Sunworks, Inc., Connecticut, made of copper pipes on copper sheet with selective black chrome coating were used. Single glazing of tempered glass was used over the collectors and the system is drained when not in use so that freezing does not occur in the pipes.

Two steel-reinforced concrete tanks each of $9.1 m^3$ capacity are used which are kept in the basement of the home. One tank is for high temperature storage and is used for direct heating of house through water to air heat exchanger put in the duct. The other tank is for low temperature storage and is used to supply water to the collectors. The hot water from the solar collectors can be supplied to either of the storage tanks or to both depending on the temperature of hot water. When the temperature of water in the storage tank is lower than 50 °C, the hot water is supplied to the heat exchanger of evaporator of heat pump and thus this low temperature heat is utilized by heat pumps which augment this heat and supply the same to the house through duct.

Domestic hot water is supplied only when the system has excess capacity. The auxiliary electric heating arrangment is also provided to supply the balance heat required for house heating. Cooling in summer is also provided by reversing the operation of heat pump. All the controls are automatic and an electronic differential controller with a set temperature difference of 6 °C between collector plate and cool tank is used in the heat collection loop. A double set house thermostat is used to control the circulating pump, fan, heat pump, and auxiliary heater.

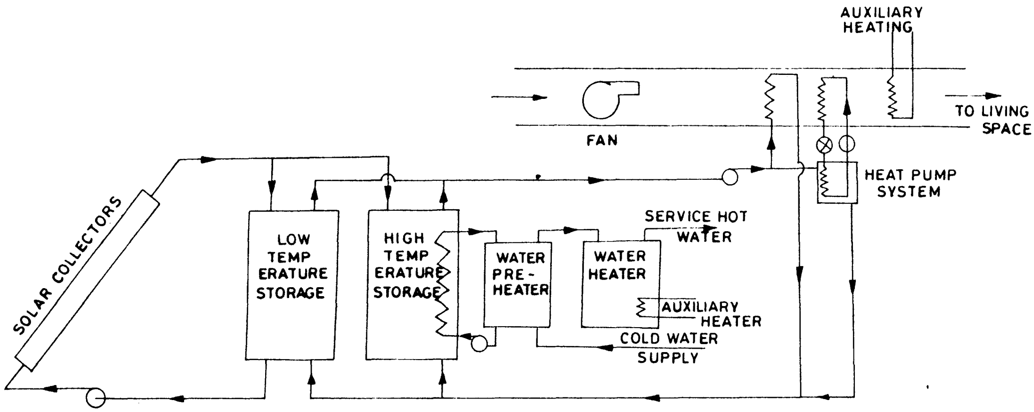


FIG.1.22 SCHEMATIC OF LORRIMAN SOLAR HOUSE INCLUDING HEAT PUMP SYSTEM AT ONTARIO(CANADA)

Table 1.10 Details of the Lorriman Solar House

1.	Floor area (excluding basement) of house	135	m^2
2.	Heated floor area	184	m^2
3.	Total solar collector	65	m^2
4.	Storage tank capacity	2x9.1	m^3
5.	Solar Energy added to the storage (From Oct. 1976 to April 1977)	40.16	GJ
6.	Thermal Energy extracted from Storage	23.25	GJ
7.	Energy supplied by heat pumps	13.69	GJ
8.	Energy supplied by auxiliary energy source	9.64	GJ
9.	Total Energy supplied to the heating system	46.59	GJ
10.	Percent of house heat load supplied by solar energy (Excluding heat supplied to water parameter)	50	

Hot water from the high temperature storage circulates through the heat exchanger located in the hot air plenum of the central heating system as well as for preheating the water for domestic use. Hot water is circulated only when its temperature is above 45 °C, below this temperature the heat pump gets activated till the water temperature reaches to 14 °C. When the water temperature in the low temperature storage drops below 14 °C, the auxiliary heater gets activated. After the collector pump stops, water drains from the collector back to the storage tank.

This house was kept under complete observation during the heating season of 1976-77 i.e. from October 1976 to April 1977. All the necessary climatic parameters, solar contribution, electrical energy consumption, etc. were measured for this period and summary results for this period are given in table 1.10. On seasonal basis the solar contribution was about 50 percent while it has ranged from 34 to 76 percent in different months. These results were not to the expectations of 60 to 70 percent which is attributed due to high heat loss from the storage tanks.

1.8.6 Colorado State University Solar House, CSU Solar House II

The CSU Solar House II is similar in physical size, orientation and thermal characteristics to Solar House I except that it uses site built Solar air heaters and the system became operational from February 1, 1976. The schematic diagram of solar heating system for CSU II is shown [130] in fig.1.23. The main features[26] of the solar system here are as follows:

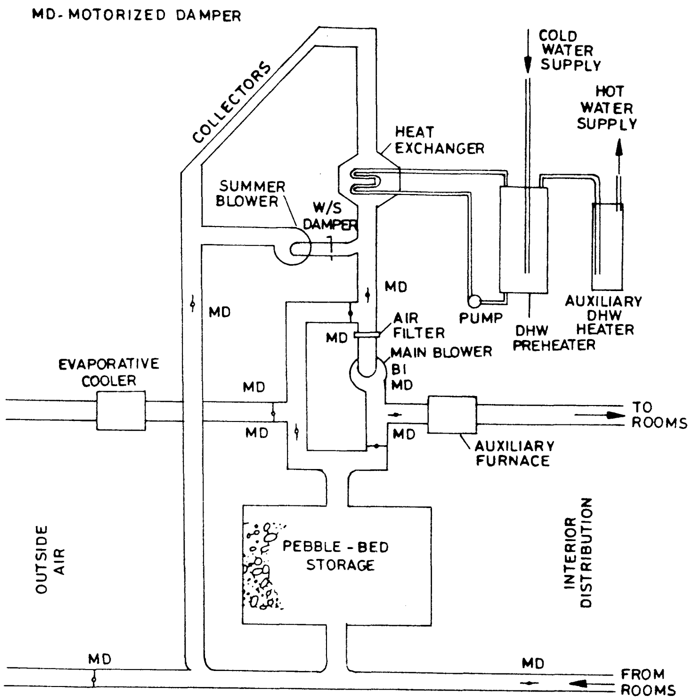


FIG.1.23 SCHEMATIC DIAGRAM OF SOLAR HEATING SYSTEM IN CSU SOLAR HOUSE II (From Karaki[130])

1. Flat plate solar air heating collectors which are site built and occupy an overall roof area of 68.4 m^2 with 68.4 m^2 net collector area are used. The air passes beneath a black painted steel plate, internally manifold and insulation and glazed with two glass sheets.
2. The storage unit consists of a tank of 10.2 m^3 capacity filled with about 18200 kg of roundish and crushed pebbles with 1.9 to 3.8 cm equivalent diameter. This

storage unit has supplied solar heated air in winter and evaporatively cooled night air in summer.

3. The system is also used to supply solar preheated water to the conventional domestic hot water tank (150 litres) fitted with auxiliary heater. The heat to the water in the preheater water tank (300 litres) is supplied through an air-to-water heat exchanger.
4. A day-night evaporative exchanger cooler with outdoor air inlet and exhaust air.
5. The auxiliary heat is supplied by a gas-fired duct furnace which is sized to supply the total heating load of the house.
6. The air handling module consists of automatic dampers, necessary ducts, filters, and only one blower for the heating operation.
7. The system shown in fig.1.23 was replaced by a new system in June 1978 as shown[130] in fig.1.24 where the gas-fired furnace is replaced by a heat pump. The site built solar air heaters were also changed to the factory built solar air heaters and were used during the 1978-79 heating season. Here instead of one blower, two air blowers are used one in the air handler and the other in the heat pump air handler.
8. In the new system provision is also made to store cold and hot.

There are following five operational modes for heating with solar energy:

1. Heating the rooms directly from solar collected heat from solar collectors directly.
2. Charging the storage with solar energy when heat is not required by rooms.
3. Heating the rooms partially from heat directly coming from solar collectors and further boosted by auxiliary energy. This system is applicable when solar intensity is low or heating load is high.
4. When sunshine is not available, heating the rooms from the heat stored in the storage unit.
5. When the heating load is high or less heat is stored in storage unit, heating rooms from heat stored in storage tank and also from auxiliary energy.

Three possible modes have been tried in the experimental heat pump system shown in fig.1.24 in the cooling mode:

1. Cooling the rooms directly with the heat pump.
2. Charging the storage unit with cold during off-peak hours.
3. Cooling the rooms from storage unit during the daytime only using only the blower in the heat pump air handler.

Table 1.11 Thermal performance data (MJ/day) for CSU Solar House II (From Karaki[130])

Months	0 Year	1 Days of Data	2 Total Incident Solar	3 Incident Solar while Operating	4 Heat from Collector	5 Heat to storage	6 Storage to load	7 Storage Heat Loss to ground	8 Storage Heat Loss to space	9 Solar to space Heating	10 Auxiliary to space heating
October	77	24	1248	1033	416.1	127.6	38.9	20.6	46.6	288.9	2.8
November	77	25	797	658	292.1	112.1	104.8	9.3	19.9	249.8	74.0
December	77	31	878	779	334.3	156.5	120.0	5.6	27.6	277.6	120.2
January	77	31	827	679	303.5	128.9	116.7	4.6	4.2	250.3	177.2
February	78	28	850	705	289.2	148.0	126.0	8.2	9.8	277.0	129.2
March	78	31	1288	1093	416.5	191.0	112.6	18.8	31.5	369.6	32.2
April	78	30	1219	988	353.2	154.0	93.6	23.3	42.8	335.6	0.0
May	78	11	1208	965	360.8	160.6	66.9	19.2	47.7	314.8	8.9
Season	77-78	212	1026	856	344.3	147.8	101.5	13.1	26.9	295.3	74.8

	11	12	13	14	15	16	17	18
Months	Solar to DHW	Auxiliary to DHW	Percent Solar of Space heat	Percent Solar of DHW	Electricity to collect Solar	Electricity to Distribute Heat	COP for Solar Collectors	COP for Solar space and DHW Heating
October	85.1	6.6	99	93	20.2	3.1	20.8	16.2
November	54.9	21.3	77	72	18.3	14.0	16	9.4
December	47.8	29.1	70	62	20.6	23.8	16.2	7.3
January	45.2	25.8	59	64	16.2	26.0	18.7	7.0
February	0.0	9.4	68	0	15.8	22.5	18.3	7.2
March	0.0	7.3	92	0	19.2	10.8	21.7	12.3
April	0.0	7.3	100	0	18.6	5.9	19.0	13.7
May	0.0	9.2	97	0	18.4	9.8	19.6	11.2
Season	29.7	15.2	80	72	18.4	15.2	18.7	10.5

Performance data for several heating and cooling seasons have been collected through a data acquisition system which is a Doric Scientific Model Digitrand 220 and a Kennedy incremental tape deck. The desk top computer is a Wang Model 2200 with 32 K-byte memory. The solar heating results[130] are summarised in table 1.11. It is seen from this table that during the entire season, more than 80 percent of the space heating load is borne by solar energy and more than 72 percent DHW load is met by solar energy. It is also seen that the electricity consumption for the operation of blowers and pumps for collecting solar heat and distributing it to rooms heating and for hot water supply is about 13 percent of the total solar heat supplied to rooms and to hot water. Therefore, the coefficient of performance (COP) for solar space heating and domestic hot water heating defined as ratio of solar energy delivered to electrical used to operate the system is 10.5.

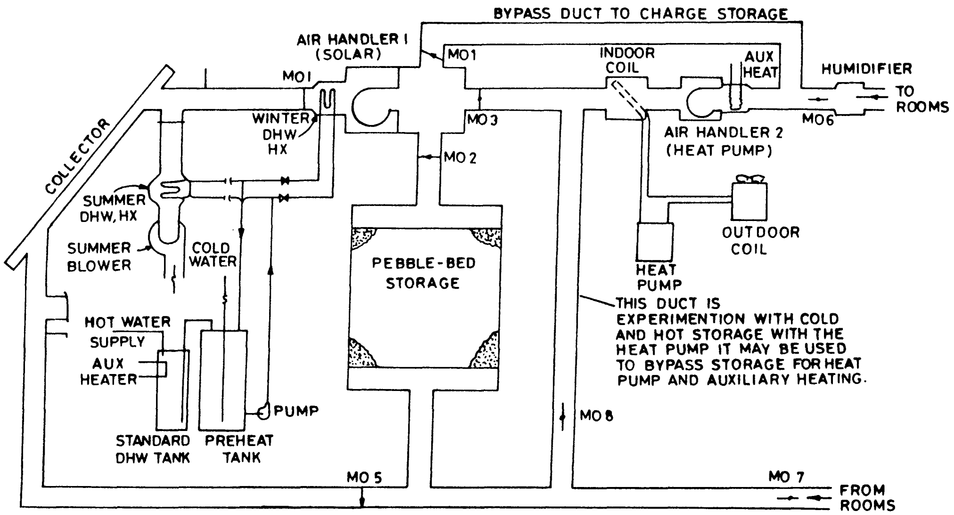


FIG 1.24 SCHEMATIC DIAGRAM OF SOLAR HEATING SYSTEM WITH AUXILIARY HEAT PUMP CSU SOLAR HOUSE II (From Karaki[130])

1.8.7 Mobile Modular Solar House

The Los Alamos Scientific Laboratory (LASL) of USA has developed a solar heated mobile/modular home which is suitable for assembly line production of factory-built mobile/

modular housing. This mobile/modular system is claimed to be the country's first air-type solar energy system suitable for such units and can have far reaching effects for the housing industry. The house alongwith the solar heating system was completed in the summer of 1976 and since then it is in operation. This mobile/modular[131-132] home is a factory built residential unit 13.41 m lomg and 7.31 m wide. The house has a conventional wood frame construction with a specially designed truss structure to accomadate the solar collector at an angle of 60° from horizontal. The house was heavily insulated with 10 cm of fibre glass insulation in the wall, about 17.5 cm in the ceiling, and 15.0 cm in the floor. All doors and windows were provided with double panes of glass to minimize heat loss. A photograph of the house is shown in fig.1.25.

Seventeen solar hot air collectors each of 0.60 m wide and 3.05 m long with a total collector area of 31.11 m^2 are used. The solar collectors are fastened directly to the sloping south wall of the home and attached together by a steel cap strip to form a weather tight roof/wall structure. They are insulated from the house by a 5.0 cm fibre glass matte, the plywood deck fastened to the trusses, and a 15 cm layer of fibre glass within the truss space.

A special storage[131] system consisting of 870 liters glass jars filled with ordinary tap water and sealed is used. These jars are spaced 1.6 cm apart and fitted in a space of 0.9 m high, 1.2 m wide, and 2.4 m long which is a furnace room and is insulated. Heat from the solar heated air is transferred to the storage unit by passing it around the jars. It is expected that this storage unit will store heat which is sufficient for 6 to 8 hours on a cold night. The auxiliary heat is provided by an electric furnace.

The solar heat is also used for heating water for domestic purposes by using an air to water heat exchanger fitted in the collector hot air return duct. This hot water is stored in a 196 litres solar preheated tank from where it goes to a 113 litres tank which is a hot water storage tank and supplies the necessary hot water for domestic use. The necessary temperature rise is supplied by the electric furnace. The hot water system is designed with a capacity of 83 litres/person/day at 50 °C for meeting 85 percent of hot water requirement for an average family size of 3.7 persons.

The heating system can be primarily[131] operated in four basic modes: heating the house directly from solar collectors, heating (charging) the storage bin only, heating the house from storage, and in summer heating the water for domestic use only. There are three primary independent control logic circuits controlling the dampers, the fan, and the furnace heating coils. The first circuit employs an electronic differential controller used to charge the storage unit and at a set temperature difference, opens the

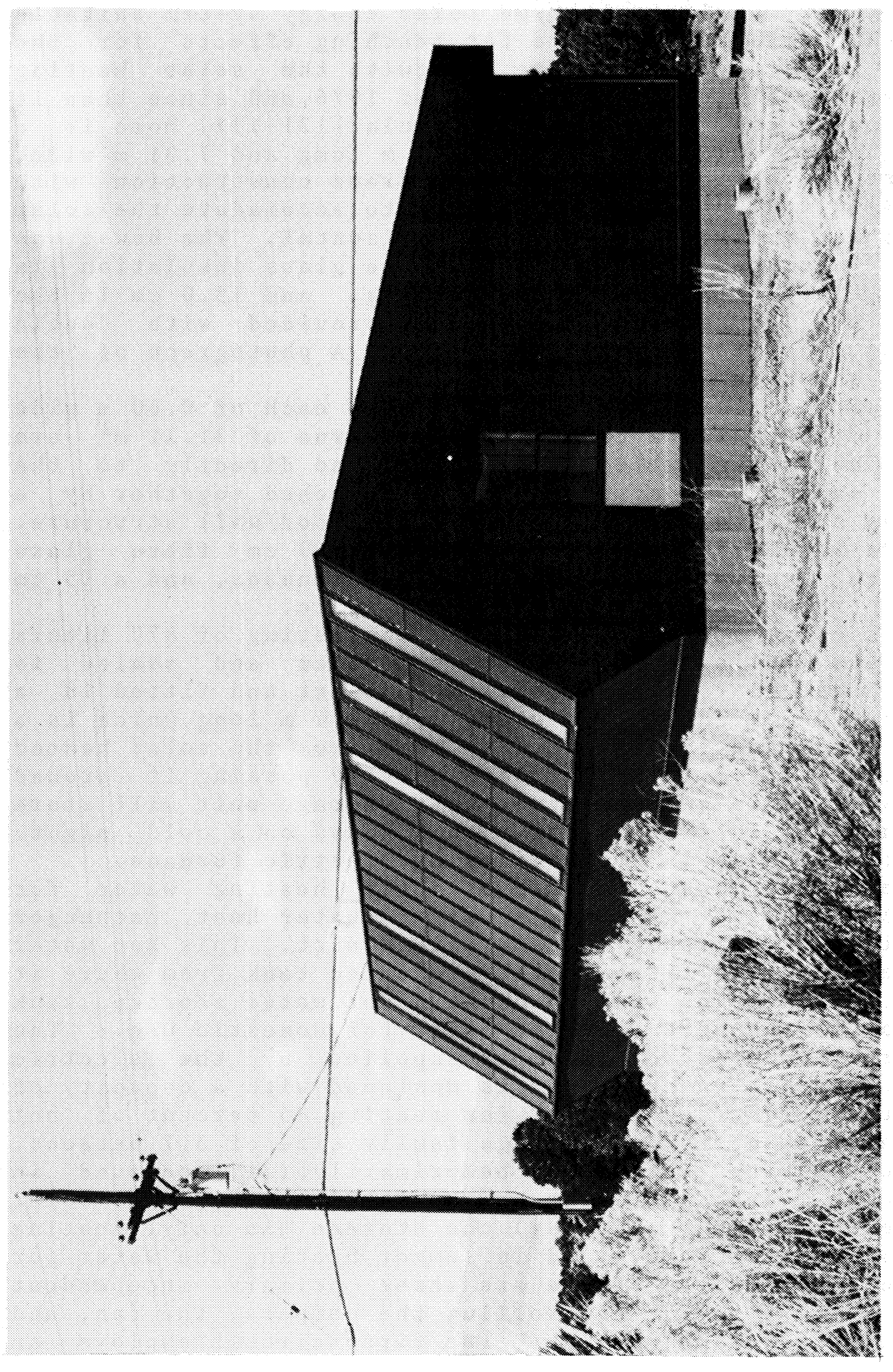


FIG.1.25 PHOTOGRAPH OF MOBILE MODULAR SOLAR HOME SYSTEM AT LOS ALAMOS LABORATORY (Courtesey S.K.Reisfeld, LASL, New Mexico)

collector damper, closes the storage return damper and turns on the fan. The second circuit employs the room dual set point thermostat which depending on the heating load requirement and availability calls the heat either from the solar collectors directly or from the storage unit. The third circuit controls the auxiliary heating coil and employs a room thermostat.

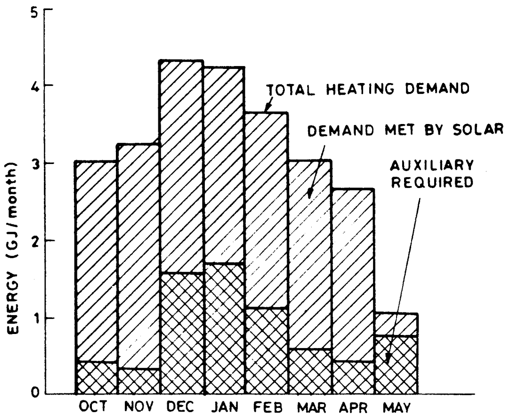


FIG.1.26 PERFORMANCE OF MOBILE MODULAR HOUSE SOLAR SYSTEM DURING THE HEATING SEASON OF 1977-78 (From Reference[132])

The heating load for the house for the outside and inside design temperatures of -18 °C and 21 °C respectively is 582 MJ/day and the heating system was designed to meet about 80 percent of this load. Extensive performance data of this house is available since October 1976. The house has a total of 63 sensors of various types. The summary of measured performance[132] is shown in fig.1.26. From this figure it is seen that about 74 percent of the heating load is supplied by solar energy. This departure from the design heating load is attributed to several reasons. This is due to small storage mass, heating demand of the house is lower than the predicted, large unexplained losses in the attic, etc. Therefore, the house was further redesigned based on these experiences to get more solar contribution.

1.8.8 National Security and Resources Study Centre (NSRSC)

The National Security and Resources Study Centre (NSRSC) in Los Alamos, New Mexico is a multizone building with three floors of 5574 m^2 and is both solar heated and cooled. The solar system provides about 94 percent of buildings heat and 70 percent of its airconditioning. The heating degree days for the place calculated on long term basis is 3500 C-day per year. The first and the second floor house the library and the third floor houses the meeting rooms, offices, and conference areas. The entire facility is designed to enhance the efficiency of the solar-energy system by incorporating several energy conserving design features.

* The walls and roof are heavily insulated using 10 cm fibre glass insulation.
* Very few windows are used and that too are tinted and double glazed,
* For the speedy and efficient transfer of heat from one location to another, a heat pipe with a heat recovery unit is used. This system preheat outside air in the heating mode and precool outside air in the cooling mode.
* A controlled, balanced air circulation is achieved for providing heating, cooling, and ventilation using signals from 53 building thermostats.
* The heat generated by light fixtures is returned to the system by allowing the return air to circulate around them.
* The air conditioning is provided by evaporative cooling using air washers.

The energy system is shown[133] schematically in fig.1.27. The photograph of the NSRSC building is shown in fig.1.28. The energy system mainly consists of a flat-plate collector array, two storage tanks, and two water chillers - a lithium bromide absorption chiller and an experimental solar Rankine Cycle Chiller. Either of these chillers can be used in series with the cold storage tank, an 85 ton absorption unit, or a 77 ton Rankine Cycle unit.

Four hundred liquid flat-plate collectors each of 0.6 m wide and 3.0 m long with a total area of 720 m are used in the system. The collector array is architecturally and structurally integrated into the building, forming the roof of the mechanical equipment room. The collector steel panels are electroplated with chrome black coating and using single glazing of tempered plate glass (3 mm thick). The collector array faces south at a 35^{o} tilt from horizontal. The heat transfer fluid used is light paraffin oil, Shell Thermia 33. This heat transfer fluid flows through the collectors and transfers its heat through a heat exchanger in the water in a 3785 litres storage tank located in the equipment room.

Hot water is stored in a low pressure 3785 litres tank for heating in winter. Hot water is stored in a 1892 litres pressurized tank which is used in summer for producing cold by operating either of the chillers. The cold water for summer use is stored in 3785 litres tank which is chilled to about 15 ℃ by circulating it through the cooling tower at night or by using one of the chillers. This chilled water is then circulated through the cooling coils in series with the chillers during the daytime. The two cooling systems

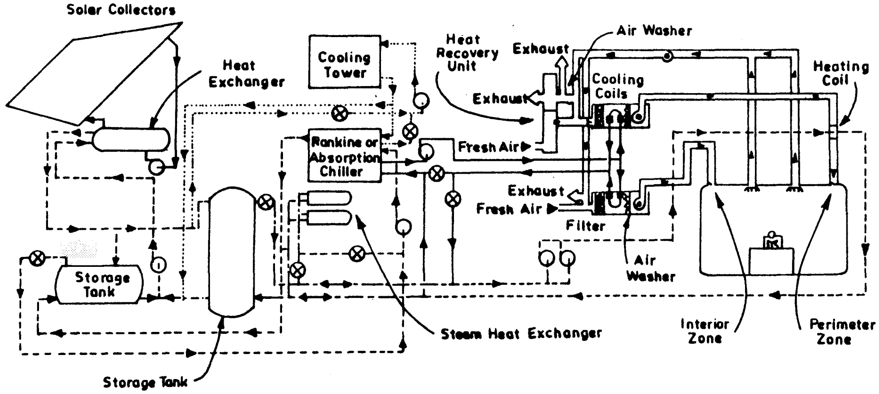


FIG.1.27 SCHEMATIC OF HVAC SYSTEM FOR NSRSC, LOS ALAMOS, NEW MEXICO (From Hedstrom et al[133])

are used to compare their effectiveness and reliability. In a Rankine Cycle Chiller, the solar heated water at 70-80 °C boils the organic fluid which drives a turbine connected to a conventional airconditioning compressor. The lithium bromide absorption chiller works similar to a cooling mechanism in a gas refrigerator. The two systems share a common fluid, condenser, and cooling tower. The auxiliary heat for heating water either for heating the space or to power the chillers is provided by steam heat exchanger system. Hot water for domestic use is also provided through a solar water preheater tank of 378 litres capacity connected to the solar hot water storage tank. There are two basic modes of operation: the winter mode (for heating), and summer mode (for cooling). In winter mode there can be either solar heating mode or auxiliary heating mode. In summer mode there can be four modes of operation: solar cooling, auxiliary cooling, cooling from cold storage, and night evaporative cold storage. If the outside air temperature is -18 °C then the hot water temperature required is 60 °C and if the outside air temperature is 21 °C then the hot water temperature required is 21 °C for building heating purposes and controls will work accordingly. The auxiliary system for heating will automatically be turned on if the hot water temperature drops below this required temperature. The cooling control system for providing the cooling will

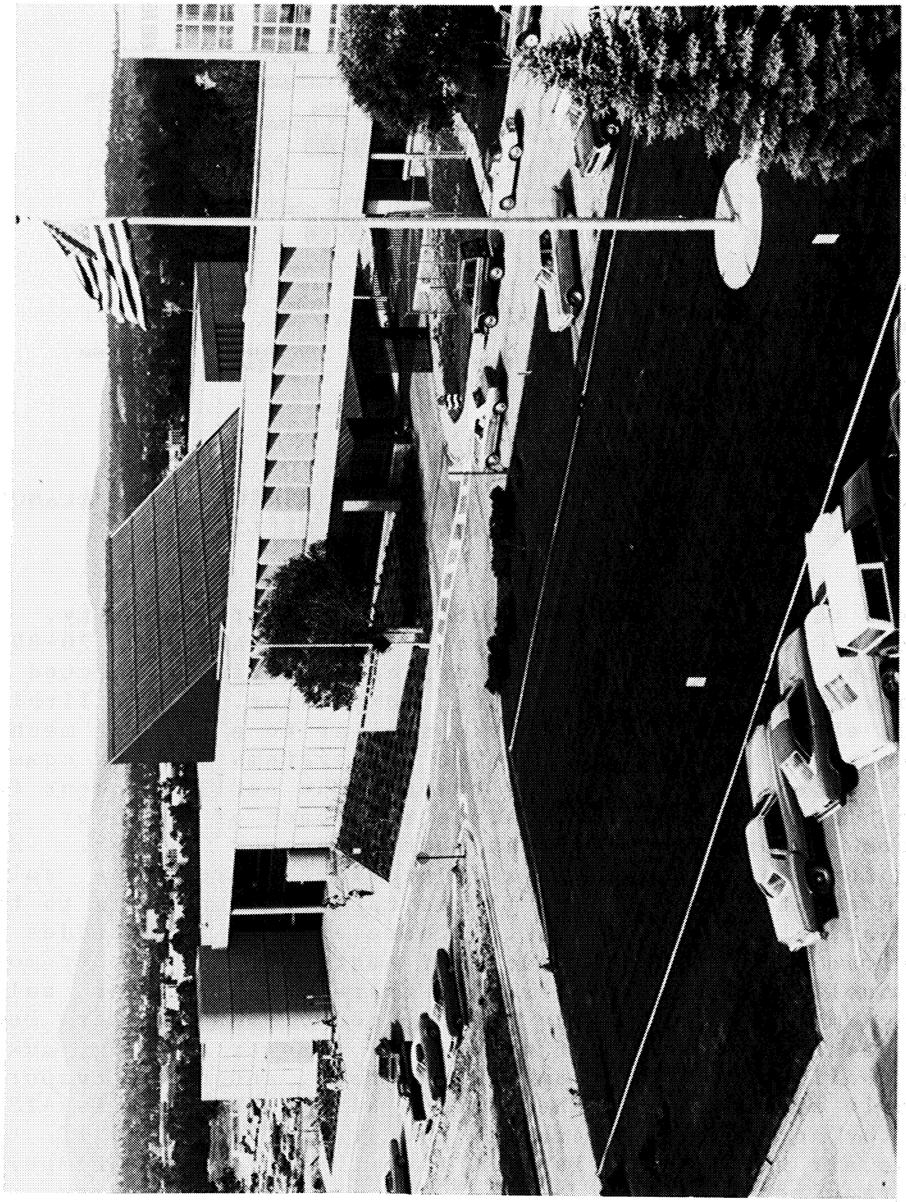


FIG.1.28 PHOTOGRAPH OF NSRSC BUILDING (Courtesey S.K.Reisfeld LASL, New Mexico)

operate in the sequence: (i) Full fresh air (ii) Air washers start (iii) Exhaust spray (iv) Chilled water from cold storage (v) Water chiller starts.

For evaluating the system performance about 160 channels of instrumentation are used in the building. All the parameters like flow rates at various points, temperatures at various junctions, energy flow, electrical power consumption by various fans and pumps, and solar radiation etc. are recorded. The heating and cooling system is in operation since October 1976 to yield background data to LASL/DOE on solar heating and cooling of large buildings and since then extensive data is being recorded. Performance summary of the solar heat supplied to the building for the year 1978 and 1979 is compared[134] in fig.1.29.

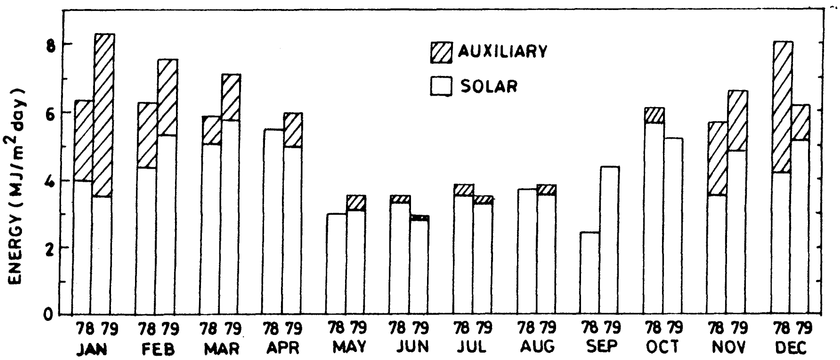


FIG.1.29 PERFORMANCE SUMMARY OF SOLAR HEATING SYSTEM FOR NSRSC, LOS ALAMOS, NEW MEXICO (From Reference [134])

It is seen that solar contribution varies in different months due to variation in solar intensity and building heating demand. The average collector efficiency was worked out to be about 34 percent. The heating system has delivered about 680 GJ of energy in a heating season which comes out to be about 80 percent of the heating demand.

1.8.9 Solarhaus Freiburg

Solarhaus Freiburg[27] is a 12-apartment house located in a Rhine Valley 14 Km west of the city of Freiburg (South Western Germany) and was constructed in 1978 and is completely monitored since March 1979. This is one of the biggest housing complex using solar energy for space heating and domestic hot water supply. Apart from using active solar

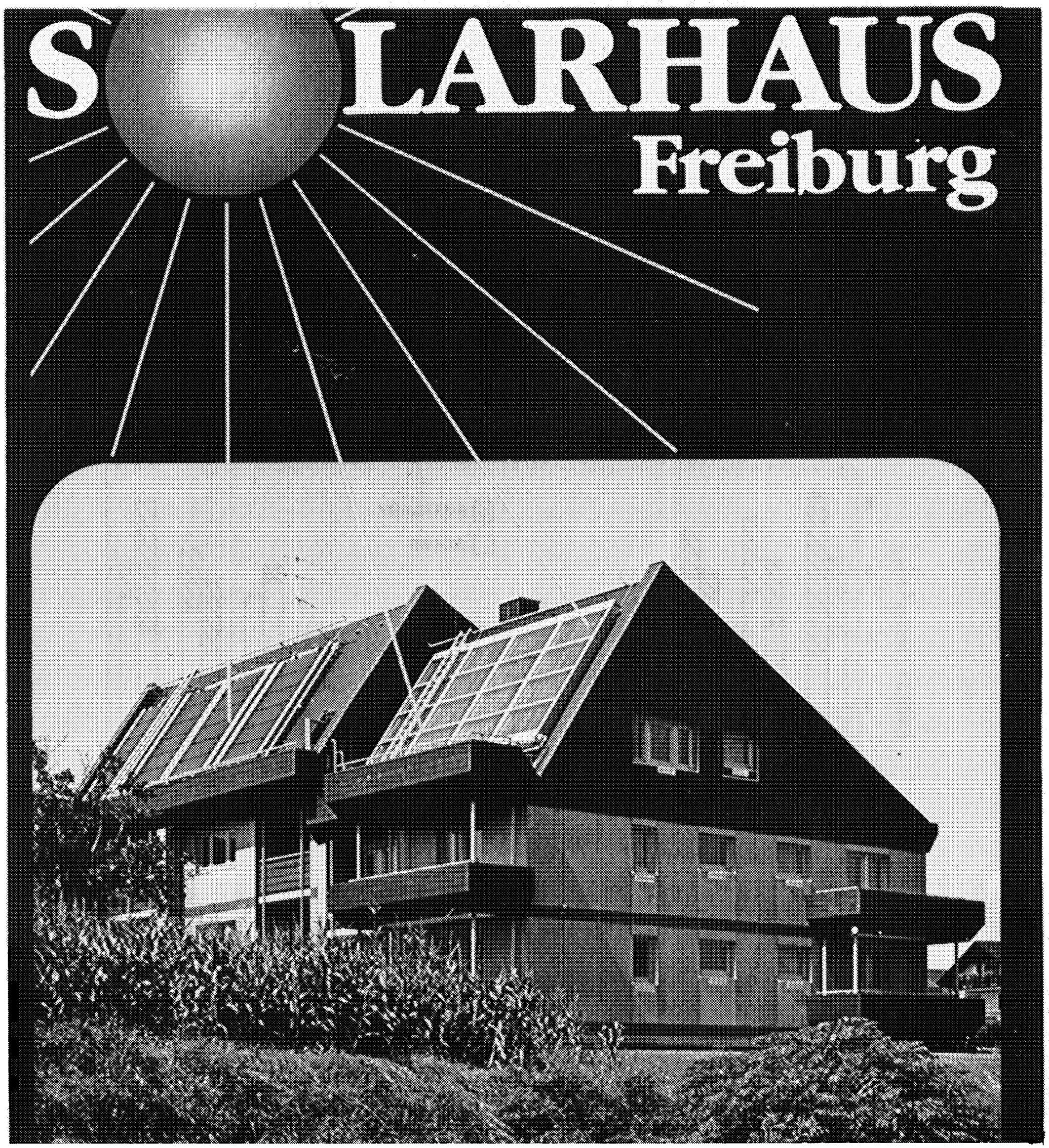


FIG 1.30 PHOTOGRAPH OF SOLARHAUS FREIBURG

heating system several passive concepts are employed to keep the house requirement as low as possible. The Freiburg solar house experiment is a joint cooperative venture between the U.S. and German governments.

The 12-apartment house , 'Solarhaus Freiburg' is occupied by 12 families or 24-27 inhabitants. The structure is made with a high thermal capacity. The specifications of the building are given in table 1.12.

Table 1.12 Specifications of Solarhaus Freiburg Building

1.	Apartments-rooms	6x1, 4x2, 2x3	
2.	Total floor area	641	m^2
3.	Volume of interior space	4284	m^3
4.	Slope of south facing roof	55	deg.
5.	Slope of north facing roof	28	deg.
6.	Thermal conduction coefficients of		
	i) Outer wall	0.34	w/m^2K
	ii) Partitions and roofs of apartments	0.43	w/m^2K
	iii) Insulation of the loft	0.24	w/m^2K
	iv) Roof of the attics	0.36	w/m^2K
	v) Windows	1.86	w/m^2K
7.	Window construction	Three glass panes	
8.	Total heating load (outdoor design temperature -12 °C)	47	KW

A photograph of the house is shown in fig.1.30. In the present system as is schematically shown[27] in fig.1.31, two completely separated load systems i.e. the DHW system and space heating system have been installed which can either be connected with two different make of evacuated tube collectors i.e. Philips evacuated collectors or Corning glass works evacuated collectors. The design details of these collectors are shown in table 1.13.

The domestic hot water heating system consists of two tanks, one preheater tank (Warm water) of 1500 litre capacity and one hot water storage tank of 1000 litre capacity. Built-in-tubular heat exchanger is used to supply the heat in the preheater tank and auxiliary energy to the hot water storage tank is provided by a electric resistance heater of

12 KW capacity controlled by thermostat to supply hot water at a temperature of 46-48 °C. For space heating also two tanks, a preheat tank and hot water storage tank each of 15000 and 5000 litres capacity are used. It is hoped that

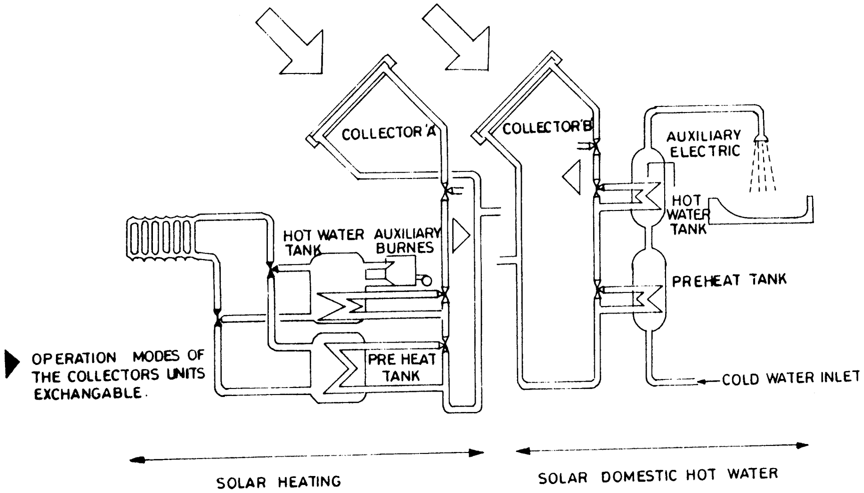


FIG.1.31 SCHEMATIC OF SOLAR HEATING SYSTEM IN SOLAR HOUSE FREIBURG (From Schreitmuller and Vonoli[27])

Table 1.13 Specifications of evaluated tube collectors used in the Solarhaus Freiburg.

		Corning	Philips
1.	Total aperture area of collectors(m^2)	33.1	29.5
2.	Total absorber area of collectors(m2)	26.8	27.1
3.	Effective transmittance-absorptance product.	0.81	0.74
4.	Overall heat loss coefficient(w/m2K)	1.7	1.9
5.	Tilt from horizontal (deg.)	55	55
6.	Heat transfer fluid	Water - glycol solution	

this two tank system in the space heating mode and DHW supply mode consumes less auxiliary energy. The auxiliary energy for space heating is provided by oil fired furnace with a maximum heating capacity of 72 KW. The heat to the rooms is supplied through radiators. An inlet temperature of about 54 ℃ to these radiators is sufficient to provide comfortable conditions in rooms even when the outdoor temperature is -12 ℃. Each radiator is fitted with its own thermostat to operate at its maximum efficiency. The rooms can be heated through storage during the transition periods between warm and cold weather and on cool summer days.

A sophisticated data acquisition system alongwith about 180 sensors are used inside and outside the house to get the information on the house operation, collector behaviour, energy flows, etc. More than 50000 observations per day are collected which are further analysed by computer to arrive at a concise description of the house energy system. The data is still being recorded and concluded for making final recommendations.

1.9 PREDICTION OF HEATING LOADS

The size of the solar heating system depends on many parameters of which the most important ones are: local climatic conditions, solar intensity, building type and size, thermophysical properties of materials used, degree of building use, hot water requirement and its frequency of use, capacity of auxiliary heating system, etc. Since many of these parameters are time dependent and hence the heating load, it therefore becomes difficult to give a simple but accurate method to determine the building heating load. For accurate computation of heating loads, complex computer programs which will calculate thermal losses through various components of a building (wall, roof and floor structural details and their areas; window and door material specifications and their areas; air leakages through doors and windows; orientation of the building and tilts of roof; occupancy; surface colour and finish of outer surfaces, etc.) using detailed hourly weather data (like temperature, windspeed and direction, humidity, solar insolation, etc.) are availabale[136,137,138] but these are time consuming and require expertise services. For active heating systems and for places where the amplitude of the outside diurnal variation of temperatures are not large, simple steady state methods are adequate for heating load calculations.Standard heat load calculation procedures are given in details in ASHRAE Handbook of Fundamentals[139] and the same is described here in brief.

Heat from the building takes place by conduction, convection, radiation, and infiltration. Typical heat losses are: 15 percent through the roof, 30 percent through the walls, 30 percent doors and windows, and 25 percent by infiltration. The heat loss is directly proportional to the temperature difference between the indoor temperature and outside ambient air temperature. Analytical expressions used for the computation of various heat losses are given in table 1.14. The overall heat transfer coefficient U is the reciprocal of the total resistance offered by a wall or roof structure:

$$U = \frac{1}{R_T} \tag{1.1}$$

Since a wall or roof does not consist of single material but of layers of different materials with different thicknesses and the loss is from air to air, the total thermal resistance R_T is given as:

$$R_T = \frac{1}{f_o} + \frac{1}{f_i} + R_1 + R_2 + R_3 + \text{———} \tag{1.2}$$

where f_o and f_i are the thermal conductance for outside and inside surfaces due to air, and R_1, R_2, R_3....., are the thermal resistances of various layers. For a single layer the resistance R_1 is given as:

$$R_1 = \frac{L_1}{K_1} \tag{1.3}$$

where L_1 is the thickness of a single layer and K_1 is the thermal conductivity of the material in the layer. Values of thermal conductivities of various building matereials are given by Garg[61]. Values of surface conductance[140], f_o and f_i are given in table 1.17.

The indoor air temperature is maintained at constant value and is known as design indoor temperature and is independent of building type. This design indoor temperature for United States is 18.3 °C (65 °F) and for UK 15.6 °C (60 °F). The outdoor design temperature is not the lowest minimum temperature which will unnecessarily increase the heating load but is taken as the hourly minimum temperature which are exceeded 95 percent of the time. Since here for simplicity, the outdoor temperature is not assumed to be varying, hence a constant outdoor design temperature, generally the lowest minimum temperature is considered for design purposes.

Table 1.14 Analytical expressions for heat load calculations for a building

Type of heat loss	Expression	Symbols used		
1. Conduction heat loss	$UA\ \Delta T$	U	=	overall heat transfer coefficient from indoor air to outside air through walls, roof, window, door, etc ($w/m^2\ °C$)
		A	=	Area of walls, roof, window, door etc. (m^2)
		ΔT	=	Temperature difference between indoor air and outside ambient air ($°C$)
2. Air charge Ventilation heat loss	$\rho_a \dot{\nu} C_p \Delta T$	ρ_a	=	density of air (Kg/m^3)
		$\dot{\nu}$	=	volumetric exchange rate ($m^3/hr.$)
		C_p	=	specific heat of air ($KJ/Kg\ °C$)
3. Infiltration heat loss	$\rho_a n V C_p \Delta T$	n	=	Air change per hour
		V	=	Volume of building (m^3)
4. Basement floor heat loss	$U_f A_f \Delta_{Tg}$	U_f	=	Heat transfer coeffieient from floor (burried) (w/m^2) which depends only on the ground water temperature and not the air temperature. Values are given[139] in table 1.15.
		A_f	=	Area of floor (m^2).
5. Exposed floor heat loss	$F_e P_e \Delta T$	F_e	=	Heat loss coefficient ($w/m^2\ °C$). Typical values are given in Table 1.16.
		P_e	=	Edge perimeter (m)

Table 1.15 Conduction heat loss rates from basement floors and floors and walls (from ASHRAE[139])

Ground water temp (°C)	Basement floors (w/m^2)	Basement walls (w/m^2)
5	9.5	18.9
10	6.3	12.6
15	3.2	6.3

Table 1.16 Values of F_e at floor for various values of inculation (From ASHRAE[140])

U-value of insulation (w/m^2 °C)	F_e (w/m °C)
0.85	0.43
1.13	0.57
1.42	0.73
1.70	0.87
1.98	1.02
2.27	1.16

Since the total heat loss from a building is proportional to the indoor-to-outdoor temperature difference, the losses are approximately calculated using 'degree-day' method. If the outside temperature is 3 °C and indoor air temperature is 18.3 °C, then the temperature difference is 15.3 °C. If this summation of daily average temperature differences over a month is done, then this total degrees difference is called the degree-days for the month. Since both heating and degree-days are proportional to the indoor-outdoor temperature difference, the number of degree days is

Table 1.17 Values of surface conductances (From Harkness and Mehta[140])

Surface	Conductance ($w/m^2\ ^\circ C$)
Internal Surface (F)	
Walls	8.4
Roof, heat flow up	9.4
Roof, heat flow down	6.8
External surface (f)	
Sheltered	8.0
Normal	10.0
Sever exposure	13.2
Walls facing north	
Sheltered	13.6
Normal	19.9
Severe exposure	32.0
Walls (all other)	
Sheltered	10.0
Normal	13.6
Severe exposure	19.9

a measure of the energy required or fuel consumption for heating for that month. Thus the number of heating degree days[141] for a given day is

$$d = \begin{cases} 0 & \text{if } T > T_b \\ T_b - T & \text{if } T < T_b \end{cases} \tag{1.4}$$

where T_b is the design indoor temperature taken as 18.3 °C and T is the mean of outdoor air temperature

$$T = 0.5\ (T_{max} + T_{min}) \tag{1.5}$$

Here T_{max} and T_{min} are the maximum and minimum temperature of the outdoor air respectively. For a month or a season of N days, the number of degree days, DD, is given as

$$DD = \sum_{1}^{N} d \tag{1.6}$$

Thus the monthly space heating load (Q_s) is now given as:

$$Q_s = UA(DD) \tag{1.7}$$

where UA is the loss coefficient and area product of the building.

The heating degree-days for various places are given in text books[42,43,46,88,89]. The above calculation procedure in steps is given below:

(i) Select a design outdoor and indoor air temperature from the local weather data. The design indoor air temperature can be taken as 18.3 °C.

(ii) Calculate areas of walls, roof, floor, windows, doors etc.

(iii) Calculate the thermal resistance values, R'S, for each component of the building.

(iv) Calculate total conduction, infiltration, and ventilation loss and sum it to get the total heating load of the building.

(v) Express this as a design heat loss rate per degree day.

1.10 SIZING OF ACTIVE SOLAR SPACE HEATING SYSTEMS

In recent past, several simulation and design procedures are developed which can be used to optimize the collector array size for most economic conditions. These methods range from simple rules of-thumb aimed at builders and designers to the sophisticated simulation methods used by researchers. These design procedures are reviewed by Hunn [142], Garg[143], and in a publication from SERI, USA[144]. The most widely used and accepted design procedure which is based on detailed simulation studies is the f-chart method [145-148] and the same will be described here in short.

The f-chart method is used to determine the solar load fraction, f, i.e. the total heating load supplied by solar energy for a solar heating system. The main design variables are: collector type and its area, storage capacity, heating load, various heat exchangers type and size, and fluid flow rates.The space heating systems used in this study using liquid and air as the medium are schematically shown[46] in fig.1.32 and 1.33 respectively. The monthly solar load fraction, f, is correlated with two dimensionless paramteres X and Y which are ratios of collector losses and heating load, and collector gain and heating load respectively. These two parameters are given as:

$$X = \frac{\text{Collector losses}}{\text{heating load}} = \frac{F_R' U_L (T_{ref} - T_a)\Delta\zeta A_c}{L} \quad (1.8)$$

$$Y = \frac{\text{Collector gain}}{\text{heating load}} = \frac{F_R' H_T N(\zeta\alpha) A_C}{L} \quad (1.9)$$

$$F_R' = F_R \left[\frac{1}{1 + \left[\dfrac{F_R U_L A}{(mC_p)_c}\right] \left[\dfrac{(mC)_C}{\varepsilon_c (mC_p)_{min}} - 1\right]} \right] \quad (1.10)$$

F_R' = A modified collector heat exchanger efficiency factor that takes into account the penalty due to the use of heat exchanger in the collector-storage loop.

F_R = Heat removal efficiency factor of collector.

A_C = collector area (m^2)

U_L = collector overall heat loss coefficient ($w/m^2\ ^\circ C$)

T_{ref} = reference temperature = 100 °C

T_a = monthly average ambient temperature (°C)

$\Delta\zeta$ = number of seconds in a month

L = monthly total heating load including service hot water

H_T = monthly average solar insolation on collector (J/m^2)

N = number of days in a month

$(\zeta\alpha)$ = average monthly product of transmittance and absorptance of collector

$(mC_p)_c$ = collector fluid capacitance rate (J/hr °C)

$(mC_p)_{min}$ = minimum fluid capacitance rate in the heat exchanger (J/hr °C)

ε_c = effectiveness of the collector-storage tank heat exchanger

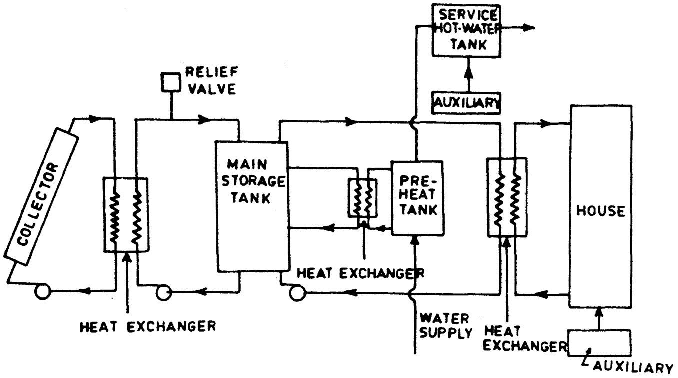


FIG.1.32 SCHEMATIC DIAGRAM OF A LIQUID BASED SOLAR HEATING SYSTEM. (From Duffie and Beckman[46]).

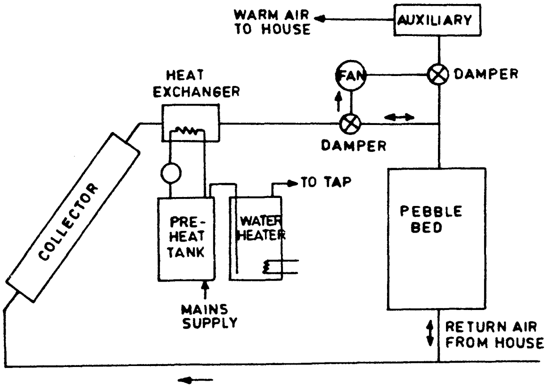


FIG.1.33 SCHEMATIC DIAGRAM OF A SOLAR AIR HEATING SYSTEM. (From Duffie and Beckman[46])

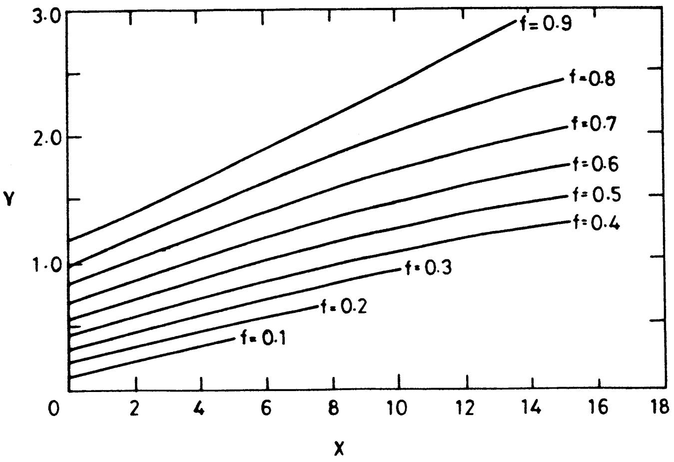


FIG.1.34 DESIGN CURVES FOR LIQUID BASED SOLAR HEATING SYSTEMS (From Beckman et al[147])

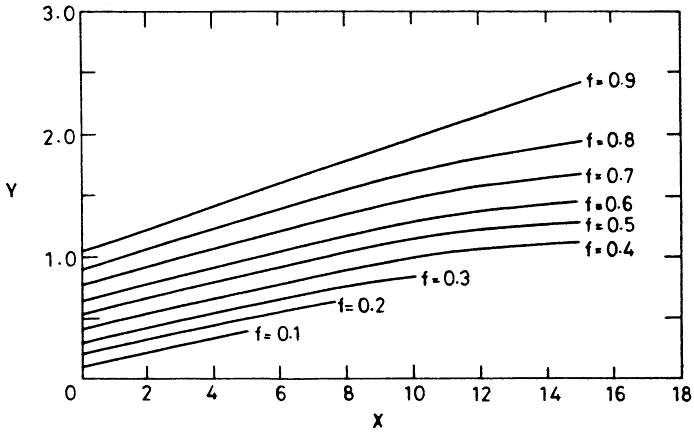


FIG.1.35 DESIGN CURVES FOR AIR HEATING SYSTEMS (From Beckman et al[147])

These correlations or design curves for liquid heating system and air heating systems are graphically shown[147] in fig.1.34 and fig.1.35 respectively. The f-chart for liquid systems is developed for the following conditions [148]

$$0.6 \leq (\zeta\alpha)_n \leq 0.9$$

$$5 \leq F_R' A_C \leq 120 \quad m^2$$

$$2.1 \leq U_L \leq 8.3 \quad w/m^2\,{}^\circ C$$

$$30 \leq \beta \leq 80 \quad deg$$

$$83 \leq (UA)_h \leq 667 \quad w/{}^\circ C$$

$$37.5 \leq \text{Storage Capacity} \leq 300 \quad l/m^2$$

$$0.5 \leq \frac{L(mC_p)_{min}}{(UA)_h} \leq 50$$

where

β = tilt of collector from horizontal (deg.), and

$(UA)_h$ = product of heat loss coefficient and area of bldg

Similarly the f-chart for air heating systems is developed for the following conditions[148] :

$$0.6 \leq (\zeta\alpha)_n \leq 0.9$$

$$5 \leq F_R' A_C \leq 120 \quad m^2$$

$$2.1 \leq U_L \leq 8.3 \quad w/m^2\,{}^\circ C$$

$$30 \leq \beta \leq 90 \quad deg$$

$$83 \leq (UA)_h \leq 667 \quad w/{}^\circ C$$

$$125 \leq \text{Storage Volume} \leq 1.0 \quad m^3/m^2$$

$$5 \leq \text{Volumetric flow} \leq 20 \quad l/m^2 s$$

The terms $(F_R U_L)$ and $F_R(\zeta\alpha)_n$ can be obtained from the slope and intercept of the standard test curve of the collector at a particular flow rate. The term $(\zeta\alpha)/(\zeta\alpha)_n$ can also be determined from standard correlations or can be taken as 0.96 for single glazed system and 0.94 for double glazed system. For liquid loop the collector fluid capacitance rate can be taken as 210 $KJ/m^2\,hr\,{}^\circ C$ which corresponds to a flow rate of 50 $l/m^2 hr$ of collector area. For air systems, the collector fluid capacitance rate is 45

KJ/m^2 hr °C which corresponds to a flow rate of 0.6 m^3/min. m^2 of collector area. If an antifreez solution is used as heat transfer fluid in the collector, then ε_c can be taken as 0.7 for the collector-storage loop heat exchanger. In case of air-rock and water drain down systems, ε_c wil be unity and hence F_R'/F_R and $(mC_p)_c/\varepsilon_c(mC_p)_{min}$ will also be unity.

Thus in the f - chart method, the parameters F_R'/F_R, $F_R'U_L$ and $F_R'(\zeta\alpha)_e$ are first determined and then by estimating the values of X and Y for a particular system the average monthly solar load fraction f is detrmined using either fig.1.34 or 1.35 for each month. The annual solar load fraction F is then determined from the following relation.

$$F = \frac{\Sigma \text{ Monthly solar load fraction}}{\Sigma \text{ monthly loads}} \qquad (1.11)$$

Recently Duffie and Mitchell[146] have used the above method and did detailed calculations and determined annual solar load fraction for 10 liquid heating systems, 10 air heating systems and 13 water heating systems. The places of installations are distributed in various parts of USA. It is found that in most of the cases the agreement between the measured and predicted annual solar load fraction is within 15 percent. The agreement is pretty close in those systems where the heating system is similar to those considered in the formulation of f - chart.

1.11 SIMULATION OF SOLAR HEATING SYSTEMS

Simulation program is a powerful tool in designing a most economic heating system and several simulation programs are developed in the recent past including those developed by Butz et al[149], Howels and Marshall[150], Sillman[151], Chang and Minardi[151] Brandemuehl and Beckman[153], Klein et al[154], Hunn et al[155], Buchberg and Roulet[156], Winn and Johnson[157], and solar scientists at the Winsconsin University[118] which is based on modular approach in which each component of the heating system is modelled independently and the governing equations are solved in a serial fashion and therefore make use of iterative technique. Recently Howells and Marshall[150] have developed a generalized simulation program for solar heating systems which shows a significant improvement over the fixed step iterative simulation program like TRNSYS. Here the main features are that all the differential equations[150] appearing in the closed loop form are simultaneously solved, and the technique for advancing the simulation form 'control function' to Control function'. Using this improved simulation program, the performance of two solar systems, one the domestic solar water heating system, and the second a combined space heating and service hot water, is predicted.

Table 1.18 Simulation data for solar heating system including service hot water supply (from Howells and Marshall[150])

Parameter	Value	
Collector Area	50	m^2
Overall collector heat loss coefficient	6.0	w/m^2K
Mass flow rate per unit area in collector loop	0.017	Kg/sec m^2
Collector plate efficiency factor	0.9	
Collector heat capacity	10.0	KJ/m^2K
Transmittance for diffuse radiation	0.76	
Transmittance for direct radiation	0.84	
Specific heat capacity	4.2	KJ/Kg K
Absorptance for solar radiation	0.9	
Heat exchanger effectiveness in the collector loop	1.0	
Volume of preheat tank	200	liters
Volume of thermal store	75	liters/m^2
Heat loss coefficient of preheat tank	0.4	W/m^2K
Mass flow rate in the domestic hot water loop	0.068	Kg/sec.
Domestic hot water demand temperature	60	°C
Heat loss coefficient of thermal store	0.4	W/m^2K
Space heating demand temperature	20	°C
Room temperature	20	°C
Building heat loss coefficient and its area product	400	W/K
Heat exchanger effectiveness of space heating loop	0.8	
Mass flow rate in the space heating loop	0.017	Kg/sec m^2
Specific heat of air	1.007	KJ/Kg K
Mass flow rate of air per unit area	0.008	Kg/m^2sec.
Integration scheme	RIKM	
Weather data	kew (England)	

The combined space heating system and service hot water used for simulation studies is the same as shown in fig, 1.32. The simulation data is given[150] in table 1.18. Performance of this solar heating system is predicted using hourly values of meteorological data for one complete year for differnt values of the user-specified error tolerance. For one complete year's simulation, the computer (ICL-470) took only 9 seconds and yield the yearly energy balance correct to within 1 percent. The predicted results are shown[150] in table 1.19. It is seen from this table that about 27 percent of the space heating load and about 58 percent of the service hot water requirement is met by solar energy.

Table 1.19 Performance date for solar heating system including service hot water supply (from Howells and Marshall[150])

Collector input	177.3	GJ
Collector output	32.87	GJ
Thermal store input	32.87	GJ
Thermal store output	28.6	GJ
Thermal store loss	4.3	GJ
Preheat tank input	8.4	GJ
Preheat tank output	8.0	GJ
Preheat tank loss	0.4	GJ
Auxiliary energy required by hot water system	5.4	GJ
Hot water load	13.7	GJ
Space heating heat exchanger input	20.2	GJ
Auxiliary energy required by space heating sytem	53.6	GJ
Space heating load	73.9	GJ
Hot water load supplied by solar energy	58	percent
Space heating load supplied by solar energy	27	percent

1.12 ECONOMICS OF SOLAR SPACE HEATING SYSTEM

The combined system for active space heating and domestic hot water supply has been proved as technically feasible. Its widespread use depends now on its economic feasibility. The economics of a solar heating system depends on many parameters and on individual circumstances of the user. In this direction attempts have been made by Butz et al[149], Brandemuehl and Beckman[153], Hunn et al [155], Lof and Taybout[158,159], Kreith and Kreider[160], Silman [161,162], Drew and Selvage[163], Esbansel and Korsgoard [164], Baylin et al[165], Barley[166], Lunde[167], and Auh[168]. Here the results as presented by Auh[168] are discussed.

The f-Chart method[147] is used to predict the solar load fraction which in turn depends on meteorological conditions, building location, collector type, collector area, annual load, and the annual load distribution. For economic analysis the parameters used are given in Table 1.20. The annual heating cost of any system is expressed as:

$$\begin{aligned}\text{Annual heating cost} &= \text{Mortage payment + Fuel cost}\\ &\quad \text{+Maintenance/Insurance expance}\\ &\quad \text{+Property tax - Tax saving} \qquad (1.12)\end{aligned}$$

Table 1.20. Values of Parameters used in Economic Analysis (From Auh[168])

Parameters	Value
Term of mortgage (yrs.)	20
Term of Economic Analysis (yrs.)	20
Term of Depreciation (yrs.)	20
Salvage value as Fraction of Investment	0
Effective Income Tax Bracket	0.46
Property Tax Rate as fraction of Investment	0.013
Extra Maintenance/Insurance costs as fraction of investment	0.01
Down payment as Fraction of Investment	0.1
Market discount rate	0.08
Annual Mortgage interest rate	0.09
General inflation rate per year	0.06
Fuel inflation rate per year	0.1
Collector area Dependent costs ($/m)	200
System fixed costs ($)	1000
Annual Total Heating load (GJ)	73.55
Cost of delivered energy ($/GJ)	16.67 (.06$/kWhr)

The most appropriate parameter for economic comparisons is the life cycle costs of solar heating system and life cycle cost of conventional heating system. The difference of first with the later will be the life cycle solar cost savings and is expressed as:

Life cycle solar cost savings = Fuel cost savings - Extra mortgage payment - Extra maintenance/insurance expence - Extra property tax + Tax savings (1.13)

The negative value of life cycle solar cost savings will show that the solar heating system is not cost effective compared to the conventional heating system. The results of this economic analysis for a house with annual domestic hot water load of 24 GJ and UA (Building heat loss coefficient x area) of 216 W/ °C which is applicable to the well insulated building for different values of delivered energy or fuel cost (FC), collector cost (CC) capitalization period in years (CP), and annual mortgage interest rate (MIR) for various collector areas are shown in fig.1.36, 1.37, 1.38 and 1.39 respectively.

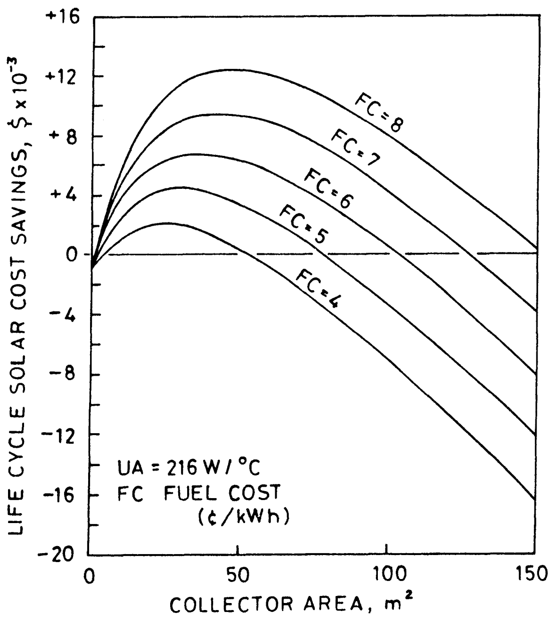


FIG.1.36 LIFE CYCLE SOLAR COST SAVINGS VS COLLECTOR AREA WITH COST OF DELIVERED ENERGY AS PARAMETER (UA = 216 W/ °C) (From Auh[168])

From fig.1.36 it is observed that as the fuel cost increases the advantage of solar heating system increases. As the fuel cost increases, the optimum collector area also increases and the solar load fraction supplied also increases. The cost of the solar heating system increases with the collector area and the collector cost per unit area. If the collector cost per unit area (CC) decreases the life cycle cost savings increases resulting the solar system more cost effective as shown in fig.1.37. If the capatalization period is increased then the life cycle cost savings also increases and the heating system becomes more

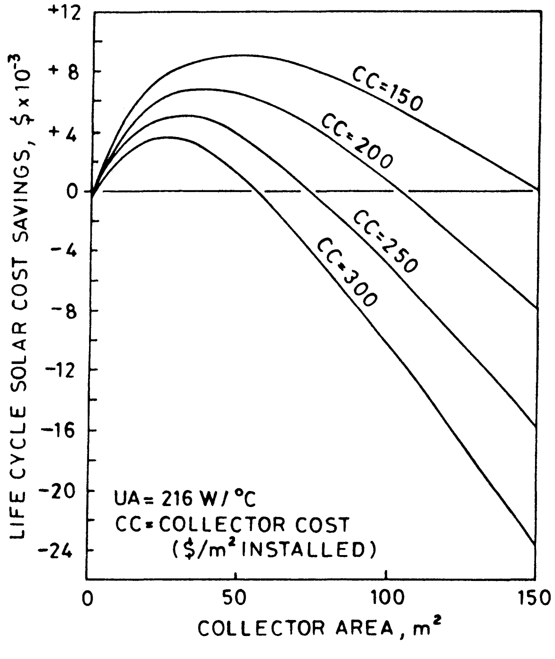


FIG.1.37 LIFE CYCLE SOLAR COST SAVINGS VS COLLECTOR AREA WITH COLLECTOR AREA DEPENDENT COST AS PARAMETER (UA=216 W/ °C) (From Auh[168])

and more reliable (fig.1.38). As the mortgage interest rate decreases, the life cycle cost savings increases as is seen in fig.1.39. The annual solar load fraction increases nonlinearly with the increase in collector area, hence the solar system will be cost effective only at an optimum collector area.

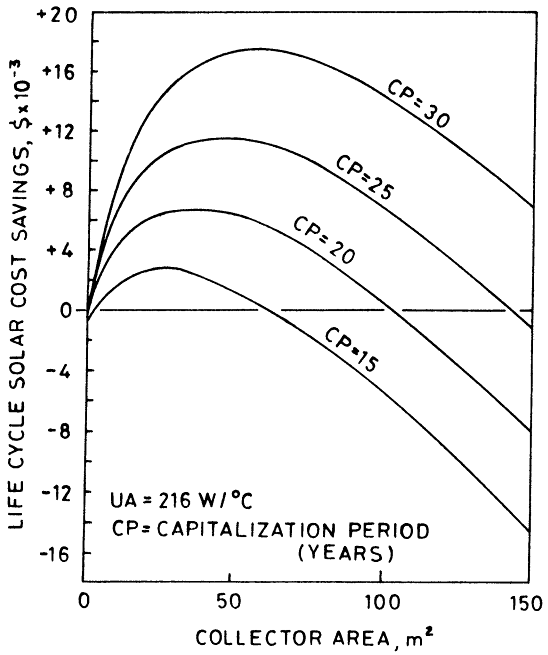


FIG 1.38 LIFE CYCLE SOLAR COST SAVINGS VS COLLECTOR AREA WITH CAPTILIZATION PERIOD AS PARAMETER (UA=216 W/ °C) (From Auh[168])

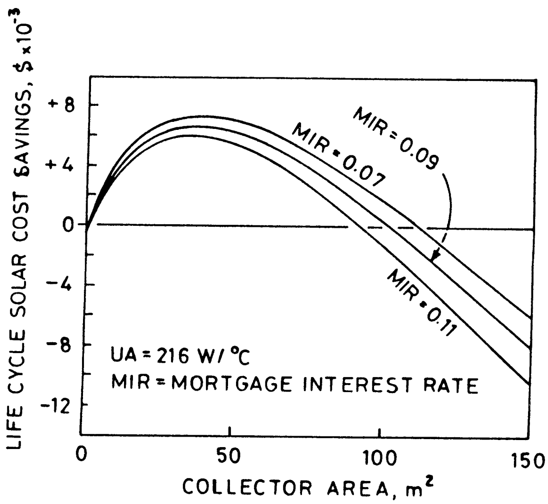


FIG 1.39 LIFE CYCLE SOLAR COST SAVINGS VS COLLECTOR AREA WITH MORTGAGE INTEREST RATE AS PARAMETER (UA=216 W/ °C) (FROM Auh[168])

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CHAPTER-2

SOLAR ENERGY FOR INDUSTRIAL PROCESS HEAT

2.1 INTRODUCTION

Industrial process heat is the thermal energy used directly in the preparation or treatment of materials and items manufactured by an industry. Presently, the heat required by these industries is met by oil, natural gas, coal or electricity, but a large portion of industrial process heat is at sufficiently low temperatures which can easily be supplied by solar energy. The year round need for energy in industries allows a maximum utilization of solar equipment. Some advantage and disadvantage of using solar energy for providing process heat are listed in table 2.1.

Recent industrial surveys[1 - 5] show that upto 24 percent of all industrial heat is directly used in processes at temperatures below 180 °C. In the remaining 76 percent, where higher temperatures are required, considerable amount of heat can be supplied to preheat it upto 180 °C. Thus about 40 percent of all process heat is found to be needed in the

Table 2.1 Characteristics of Solar Industrial Process Heat
(From Kreith et al[4])

Advantages

* It can replace scarce fossil fuels like oil and gas.
* Many processes are in temperature range well suited for solar technologies.
* Year-round loads give good utilization of solar equipment.
* This is 3-times as effective as heating applications.
* Many end uses are possible without thermal storage requirements.
* Larges installations may give economics of scale.
* Experts field maintenance staff are available in industry.
* Industry is accustomed to life cycle costing and long-term financing.
* No heat engine required and therefore higher efficiencies are obtained.

Disadvantages

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* Industry gets favourable rates from utilities.
* Process requirements, such as temperature control, may make integration with solar system difficult.
* Variable nature of insolation required solar systems to have full auxiliary backup or long term storage.
* Lack of familiarity with soalr equipment and solar system operation.
* Size of system may be limited by land availability and maintenance.
* Solar industrial process heat is a new and still a unproven technology.
* Industry expects rapid pay back of investments(3 to 5 years).
* Lower temperature operation gives lower efficiency.
* Industrial environment affects the life of the system due to surface contamination.
* Unfavourable economic criteria taxes, investments.

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temperature range from ambient to 180 °C. In several industries 100 percent process heat is required in the temperature range from ambient to 180 °C which can easily and economically be supplied by flat-plate collectors, solar ponds, evacuated collectors, and linear concentrators where the technology is sufficiently advanced. Thus this low temperature process heat requirement in industries makes the solar system quite attractive.

2.2 INDUSTRIAL ENERGY USE

The demand of the industrial energy in industrialized countries is about 35 to 40 percent of the overall energy consumption, while in developing or less developing countries this value is even higher, in some cases this is upto 50 percent. Figures 2.1 shows[3] the share of the 1980 industrial energy consumption: total industiral energy use was 30.6 quad (1 quad = 10^{15} Btu), which counted for 39 percent of the 1980 total U.S. energy consumption. It was found that 39 percent of this industrial energy was consumed in process steam, 26 percent in direct heat, 19 percent in electric drive, and 16 percent in feedstock. It is seen from Figure 2.2(a) that in U.S.A. this industrial energy is supplied by electricity, natural gas, oiland coal with the major contribution by electricity, natural gas and oil. The pattern of industrial energy consumption in developing countries with an example of India is shown in figure 2.2(b). From this figure also it is seen that even in a

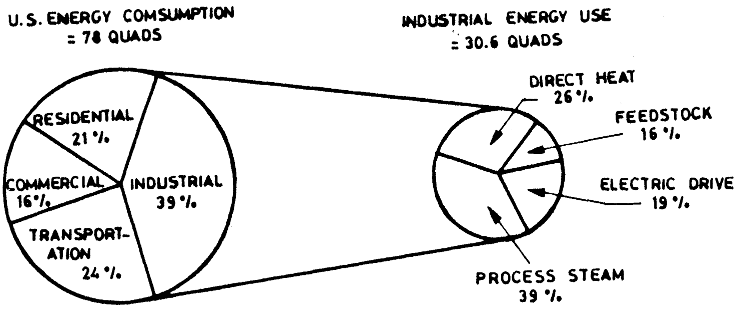
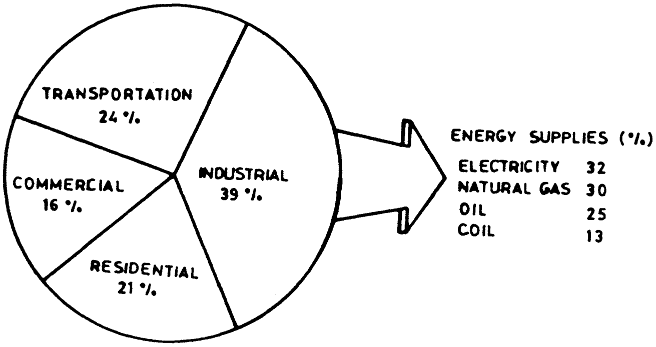
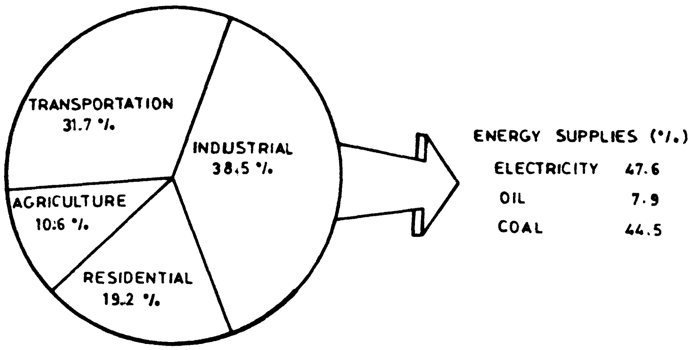


FIG.2.1 INDUSTRIAL ENERGY CONSUMPTION IN USA (1980)
(From David Hu[3])



(a)



(b)

(a) IN USA (1980) (b) IN INDIA (1979)

FIG.2.2 PATTERN OF INDUSTRIAL ENERGY CONSUMPTION developing country like India the energy consumption in the industrial sector is also about 39 percent of the total energy consumption but the total amount consumed is significantly low compared to U.S. One interesting observation is that the major share of energy supply in India in the industrial sector is met by electricity, and coal.

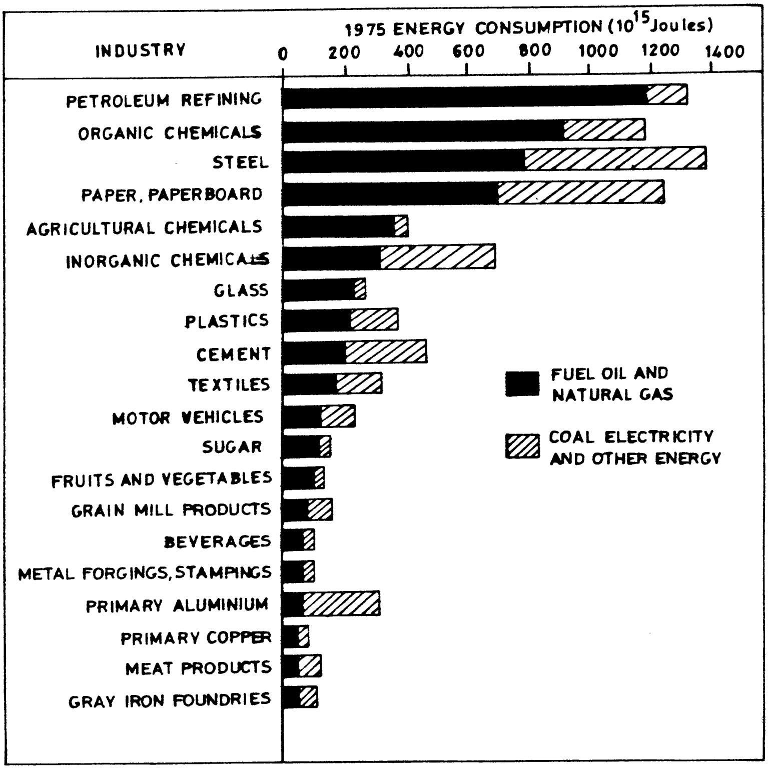


FIG.2.3 ENERGY CONSUMPTION IN FEW INDUSTRIES IN USA (From David Hu[3])

The energy consumption by some of the industries in U.S. is shown[3] in figure 2.3. From this figure it is seen that the most basic industries such as petroleum refining, chemical, steels, paper, glass, plastics, cements and textiles, consume most energy and are heavily dependent on fuels like oil and natural gas. In particular the following four major groups consume nearly 60 percent of all industrially used oil and natural gas[3]:

- Four key chemical industries: 23 percent.
- Petroleum refining: 15 percent.
- Steel Industry: 10 percent.
- Paper Industry: 9 percent.

Figure 2.4 shows the industrial energy consumption percentage by the 6 major energy consuming industries. Figure 2.4(b) shows the process heat consumption by these 6 largest energy consuming industries.

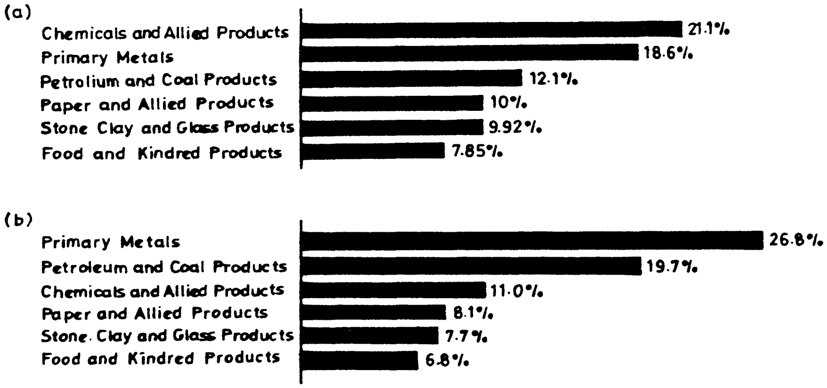


FIG.2.4 APPROXIMATE DITRIBUTION OF (A) INDUSTRIAL ENERGY CONSUMPTION AND (B) PROCESS HEAT CONSUMPTION AMONG THE SIX LARGEST ENERGY CONSUMING INDUSTRIES (From NA-58-31293 contract report)

The energy required by industries vary considerably. Apart from energy requirement, the thermodynamcis quality or temperature level is also important since it determines the type of solar collector which can meet the requirement. Figure 2.5 shows[1] the fraction of energy required below various temperatures for a group of industries that use some 48 percent of the total energy consumed by U.S. industry. It is seen from this figure that slightly less than 5 percent energy is used at temperatures below 100 °C. But if preheating is also included then about 30 percent of the total consumption occurs below 100 °C. The preheating curve from a initial temperature of 16 °C is also included in the same figure. Thus even a simple solar collector like flat-plate collector can play a significant role for preheating in industries. For some selected industries, table 2.2 shows[6] the range of process temperature and the total demand of energy in that temperature range. It is estimated that in USA about 10^{18} J of process heat is consumed annually below 100 °C and 3.7×10^{18} J is used annually between 1000 °C and 175 °C. This gives a good estimate of industrial solar appli-

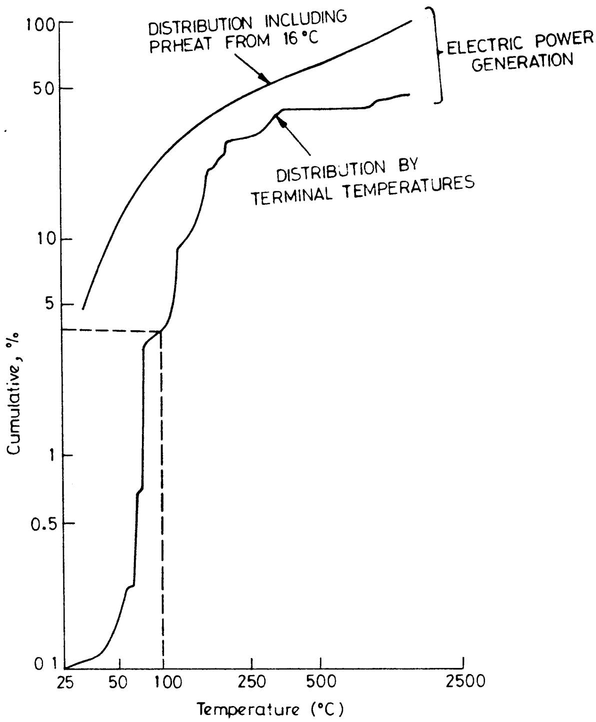


FIG.2.5 DISTRIBUTION OF PROCESS HEAT USE AT A TEMPERATURE LEVEL IN USA (From M.D.Fraser[1])

cation potential. Table 2.3 shows [7] some typical examples of thermal processes that seem especially interesting for the application of solar process heat generation. It is certainly unrealistic to assume that solar energy can meet all the process heat demand in large or small industries due to general limitations e.g. by the unrealiability of sunshine, insolation profile, and available land. The coverage rate of solar industrial process heat (SIPH) system will be limited by a maximum of 40 to 50 percent by the above factors and the user's profile, in most cases it will be less than 40 percent due to practical reasons. Therefore, most SIPH systems will require complete backup, so the solar equipment will get credit only for saving fuel, not for displacing boiler capacity. It can also be said that solar energy is a poor match for a capital intesive energy source with cheap fuel like electricity from nuclear energy or heat from coal. But solar energy as IPH has a better potential compared to oil and gas.

Table 2.2 Process temperatures in major industries (percent of total demand in each temperature range used in each industry class) (From Kelly and Gawell[6])

Process Temperature (°C)	Food	Paper	Chemicals	Petroleum	Stone/glass	Primary metals	others
upto 100	16	22	8	0	2	16	36
100 - 180	11	33	29	5	1	1	20
180 - 290	5	18	33	16	0	11	17
290 - 590	2	0	8	72	0	0	18
590 - 1100	0	0	0	46	54	0	0
1000 +	0	0	0	0	34	66	0

Table 2.3 Examples of medium temperature industrial process (From Feustel[7])

Industry/Process	Medium	Temperature range (°C)
Food Industry		
- Cooking	Steam	120 - 185
- Drying	air, steam	120 - 230
- Canning	water, steam	80 - 130
Textile Industry		
- mercerizing	water, steam	upto 100
- drying	steam	60 - 135
- finishing	steam	60 - 150
Chemical Industry		
- drying	air	60 - 125
- dissolving, distillation, thickening, leaching, etc	steam	85 - 170
Pulp and paper Industry		
- kraft pulping	steam	185
- kraft bleaching	steam	140
- drying	steam	180
Stone-glass-clay-Industry		
- brick curing	steam	75 - 180
- gypsum calcining	air	160
- gypsum curing	steam	300
- glass fibre drying	air	100 - 180

Solar Industrial process heat (SIPH) systems are simple and consist of a suitable solar collector field, storage system, heat exchanger, fluid flow pipes, pumps, and controls. The size of any SIPH system depends on the Solar insolation, requirement of process heat, its demand pattern and temperatures and the size is generally decided based on cost-benefit analysis. The most important and most expensive single component in any SIPH system is the solar collector field and a general procedure for deciding the type of collector is as follows[8]:

1. Determine the solar insolation, ambient temperature, and duration of sunshine for the location[9].
2. Determine the heating load, load pattern, and temperature of the process under consideration.
3. Devise the flow schematic and conceptualize the solar system.
4. Determine the collector efficiency of a given collector from the standard curves of efficiency versus $(T_i - T_a)/I_{Tt}$ as described by Garg[9].
5. Determine the system performance using standard computer simulation models or simpler design models described in Vol 1 of the book[10].
6. Determine the annual solar load fraction for different collector array sizes.
7. Determine the most cost effective collector system using the economic optimizing procedure on several collector systems.

The procedure described above is more systematic, accurate, and elaborate but time consuming and hence simpler procedure based on simplified assumptions and thumb rules is used[8] in designing a solar industrial process heat (SIPH) system.

2.3 SOLAR COLLECTOR TECHNOLOGY

Solar energy collector is a heart of any solar energy utilzation device. The solar collector technology is summarized[4,5] in table 2.4. Solar energy collectors as discussed by Garg[9,10] are flat plate type and concentrator type collectors.Flat plate collectors are characterized by durability, dependability, simplicity, and high solar collector efficiency. At low temperatures, the flat-plate collectors operate at high optical and thermal efficiency compared to concetrators. However, as the collection temperature goes on increasing, the efficiency of a concentrator decreases very slowly while the flat-palte collector efficiency decreases very fast. Therefore, at a slightly higher temperature, a concentrator will outperform a typical flat-plate collector. This cross over temperature depends on many design, operating, and climatic conditions. In practice, the choice between a concentrating and a flat-plate collector is based on cost difference (both initial and operating) and local insolation characteristics as well as the load temperature requirements and the performance of the particular collector. Therfore, the most obvious choice for low temperature applications (< 100°C) is the flat-plate collectors and solar ponds. Flat-plate collectors are extensively studied [9,10] and used with water or air as

Table 2.4 Solar Collector Technology

Collector Type 1	Maximum operating temperature (°C) 2	Remarks 3
Flat Plate (Single glazed and flat black absorber coating)	40 - 80	Best known, widely used and studied collector .Need freez protection when water is a working fluid. Requires less maintenance compared to other collectors.
Flat Plate (Single glazed and selective coating)	60 - 100	Develpoed and used in many countries. Problems of surface degradation on over-heating.
Shallow Solar Pond	40 - 60	Plastic cover needs periodic replacement. Suitable for sunny climates only.
Non Convective solar pond (salt gradient type)	40 - 90	Considerably low efficiency (10 to 20 percent) and requires maintenance. Both collector and storage are combined and even seasonal storage is possible with a depth of 3 meters.

Table 2.4 cont.

1	2	3
Non evaculated CPC, stationary or summer to winter tilt adjustment	80 - 120	Requires large reflecting surface per unit of aperture area.
Evacuated tubular collector(with reflector enhancement, including CPC)	120 - 200	Cost can be reduced on mass production. Breakage problems.
V-trough	150	Does not require continous tracking but require weekly tilt adjustments.
Inflated cylindrical reflector	150	Does not require continous tracking but require weekly tilt adjustments.
Parabolic trough	300	Accepts only beam radiation and require accurate tracking. Usable only in predominantly clear sky areas and sensitive to dirt.
Segmented parabolic trough	300	- do -

Table 2.4 cont.

1	2	3
Fresnel concentrator	250	It can be made very light. The plastic material deteriorate with time. Requires accurate tracking and accepts only beam radiation.
Parabolic trough with tracking receiver	250	Dirt gets accumulated on reflector.
Heliostats	1000	Accepts only beam radiation. Problems with heat transport to point of use.
Spherical bowl with tracking receiver	600	Accepts only beam radiation. Problems with heat transport to point of use.
Parabolic dish	1500	Accepts only beam radiation. Problems with heat transport to point of use.

working fluid. But water or liquid flat-plate collectors are preferred since they are more efficient compared to flat-plate collectors using air as the working fluid.

Solar ponds are of two types: convective type or shallow solar pond and non-convective type or deep solar pond. The shallow solar ponds are extensively studied [11,12] for supplying industrial process heat upto a temperature of 60°C and generally consist of a plastic horizontal water bag, insulated by one or more plastic films and air layers. There is no storage provision and hot water is drained for use in the evening. Because of ultravoilet degradation of plastic material, the covers etc. require periodic replacement. The deep non convective type or popularly known as salt gradient solar pond provides built-in long-term storage and can provide heat upto 90°C in sunny locations and therefore can be used for providing large industrial process heat. An inverse salt gradient layer with more concetration at the bottom prevents the heat loss by convection. Such salt gradient solar ponds have been built and tested extensively [10,13-15] in many countries including Israel, Canada, USA, Australia, India, and New Mexico. So far no quantative data or cost analysis of salt-gradient solar pond for the supply of industrial process heat is available.

Evacuated tubular collectors are suitable for operation upto a temperature of 200°C but recommended for use in the temperature range of 100 - 150°C. They do not need any tracking arrangement but use some reflective arrangement. They accept diffuse radiation also and can outperform tracking collectors particularly in cloudy regions. These evacuated tube collectors are pioneered by Owens-Illinois and General Electric of USA but are now produced commercially in many countries[16,17,18]. Evacuated tubular collectors are currently not economically competitive at low temperature, but their high effieciency at high temperature coupled with a potential for substantial cost reduction makes them candidates for future use. For getting higher temperatures both reflective type or refractive type concentrators are employed. A variety of geometric shapes have been proposed [19,20] and prototype or production units built in each category. The concetration ratio(CR) is considered to be the most pertinent parameter in deciding the type of concentrator for a particular application, since higher the CR higher will be the temperature. The CR may range from 2 - 10 for compound parabolic concentrator (CPC) or V-trough to 2000 for a parabolic dish. The affect of CR on absorber (receiver) temperature for various concentrator type is shown in fig.2.6. The same figure also indicates the relative merits of concentration ratio anhancement vis-a-vis improving the selective coating performance for a contrast 25 percent energy loss. Increasing the absorptivity /emissivity ratio (α/ε) from 4 to 10 is comparable to changing the concentration ratio from 100 to 250 at $(\alpha/\varepsilon) = 4$ interms of achieving the same rate of absorber temperature.

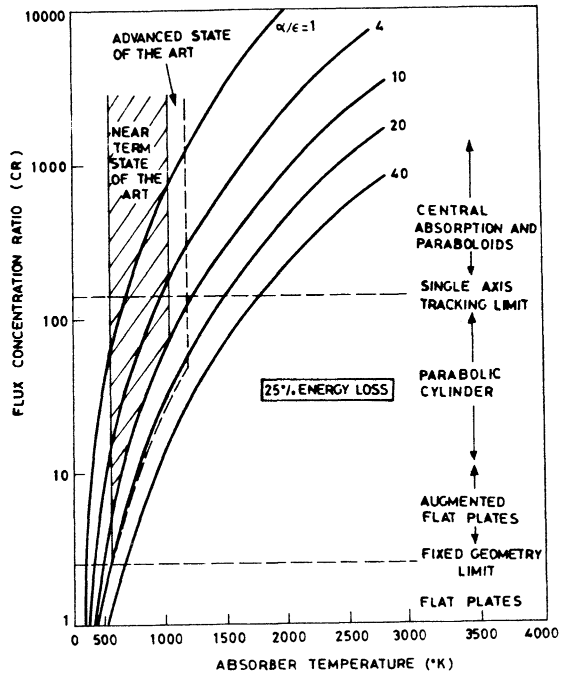


FIG.2.6 EFFECT OF CONCENTRATION RATIO AND ABSORPTIVITY EMISSIVITY RATIO ON ABSORBER TEMPERATURE FOR SOLAR CONCENTRATING COLLECTORS (From Gervais and Box[21])

Compound parabolic concentrator (CPC) collectors[22,23] with low CR can operate easily upto a temperature of 200°C. From a practical view point this so called 'full' CPC deisgn requires an excessive reflector area per unit of aperture area. For example, a CR of 9.4 can be attained with a total acceptance angle of 12°, but the reflector area is approximately six times larger than the aperture area. The reflector area can be reduced by truncating the CPC without significantly degrading performance. The parabolic trough concentrator (PTC) collectors are commercially available, quite versatile, and can be used both for medium and high temperature applications[24]. PTC collectors are operated in the temperature range of 120 to 300 °C. PTC collectors usually track the sun with one degree of freedom using one of the three orientations: east-west, north-south, or polar.

The east-west and north-south configurations are the simplest to assemble into large arrays but have higher incidence angle and cosine losses than the polar mount. However, the polar mount intercepts more solar radiation per unit area. PTC collectors are manufactured by several companies in USA and are used for providing industrial process heat in series with flat-plate collectors. These collectors require continous and accurate tracking and recommended for temperatures upto 200°C. Regular cleaning, perhaps every week may be necessary in dirty environments. An alternative to the parabolic trough is a segmented mirror system, which replaces the large curved surface of the trough with a series of narrow segments that are tilted to approximately the surface angles of a trough. The advantages of the segmened mirror reflector are that the receiver is fixed and the mirror stats are small and presumably relatively inexpensive and the concentrator can be made easily. The mirror segements can be made flat so that the concentration is simply the number of stats. Losses in segmented mirror collectors are caused by a variety of contributing effects like shading and screening by mirror stats, end losses, cosine losses and losses due to imperfect reflectivities. Another potential candidate collector for process steam is the linear Fresnel reflector collector in which individual tracking mirror stats focus solar radiation onto a common linear receiver. The Fresnel reflector collector can have either tracking mirror (TM) with fixed receiver or fixed mirror (FM) segemnts with tracking receiver. The FM design is a fixed aperture device and hence is subjected to cosine losses at off-normal incidence. The TM collector can be considered as a series of small rim-angle tracking collectors with a common absorber. Linear Fresnel lens[25] collector made of plastic is another candidate collector for industrial process heat. The grooves in a Fresnel lens may either face toward or away from the sun, but the later is preferred because the facet edges are protected and it is easier to keep the lens clean. These collectors can be used in a non-tracking mode aligned east-west, so long as north-south tilt adjustments are made. However, off-axis aberrations experienced early and late in the day cause severe defocussing. Therefore, a north-south alignment with east-west axis tracking, possibly with seasonal tilt adjustments, is preferred. Further improvements in the linear concentrating collectors can be made by using high technology such as evacuated receiver, improved coatings, etc. At the present time, only parabolic trough, evacuated tube and linear Fresnel lens have achieved a large degree of market penetration and are used upto a temperature of 300°C.

Very high temperatures (>500°C) can be obtained by using two axis tracking system by parabolic dish or spherical dish or Fresnel lens or large heliostat system. Dish type concentrators have the advantage of higher concetration and much greater utilization of solar intensity at off-noon hours. There are mainly two disadvantages in using dish concentrators. One is that two axis tracking is required which is more complicated, expensive and unwieldy than the single axis tracking used for linear systems. Second is that when used as a distributed field of modular dishes, the heat losses in the field can be great at high operating temperatures. Moreover, it is very difficult to collect high temperature heat from a field of parabolic dishes or Fresnel lenses. The alternative could be to use a single very large dish in a fixed configuration with a tracking receiver. An interesting concentrator known as FMDF(fixed mirror-diffuse focus) has been proposed by Reichert[26] which is similar to SRTA (stationary reflector-tracking absorber) concentrator[27], in which a large dish of about 61 m diameter is built into the ground to resemble a football stadium. Unlike the smaller dishes, which are paraboloids of revolution, the large dish has been proposed as a spherical mirror. These dishes are very expensive and are again only capable of concentrating the direct components of solar radiation. Recently Nix[28] has studied the feasibility of using a parabolic dish solar collectors using various energy transport processes. The system delivers 1.0 MW to a single user at 4.14 MPa, 400 $^{\circ}$C superheated steam. He also compared thermochemical methods of transporting energy with direct sensible methods of transporting energy for a small Industrial Process Heat system using parabolic dish receivers.

The popularized central receiver collector or power tower provides concentation ratios of several thousand and is an ideal system for providing high temperature heating from perhaps 500 $^{\circ}$C to 1000 $^{\circ}$C. The Central receiver system is a distributed set of fixed mirrors (with adjustable orientation) that can concentrate their reflected rays on to a receiver located atop a nearby tower. Each of the fixed mirror is called a heliostat which is either of metallized plastic reflector or silvered glass mirror with each heliostat having an area of about 37-74 m^2. Such heliostat-power tower systems are intended to be large thermal power plants, and are not easily scaled down for modest energy demands (e.g., a single factory). These systems also do not utilize the diffuse component of solar radiation. The temperature that can be obtained by these collectors are shown[4] in fig. 2.7.

It is concluded by Grimmner and Herr [29] that looking to the complexity and operational and maintenance problems, the needs of industrial and commercial sectors for process heat can be met most economically by the installation of concentrating flat-plat collectors. These concentrating flat-plate collectors include reflective side panels (boosters), which produces a concentration ratio of 2 to 4; and the winston collector or CPC collector which can produce concentration ratios of 4 to 10. The chief advantages of these collectors are that minimal or no tracking is required, direct and diffuse radiations are utilized, and that temperatures upto 250°C are achievable. Even higher temperatures can be obtained by using selective coating and evacuated receiver tube alongwith the above collectors.

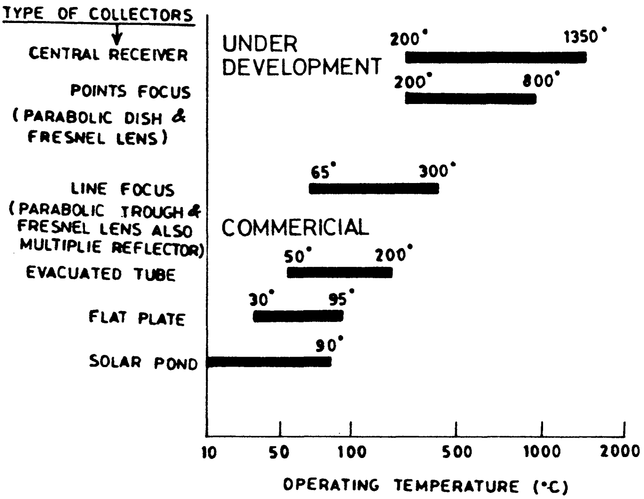


FIG.2.7 OPERATIONAL TEMPERATURE FOR VARIOUS COLLECTOR TYPES (From Kreith et al[4])

2.4. SOLAR THERMAL ENERGY STORAGE TECHNOLOGY

Thermal energy can be stored in the form of sensible heat, latent heat, chemical reaction or in combinations of these. The thermal storage is required, if solar industrial process heat (SIPH) system do not simply operate as fuel saver being directly coupled to a conventional boiler system and if the consumer requires a certain load profile. The major problem in thermal storage is the selection of materials having suitable thermophysical characteristics. Several articles [30-34] both of research and review type and recently a book by Garg et al[35] are written recently describing the thermal storage technology. Table 2.5 lists briefly the various processes and materials for thermal energy storage both at low temperatures and high temperatures. Selection of a particular solar thermal energy

Table 2.5 Heat storage capacities of some common materials

Process	Materials	Heat Capacity (KJ/KG K)	Transition Temp. (°C)	Heat of fusion (KJ/Kg)
A. Sensible heating				
	Water	4.190		
	Calorie HT 43	2.300		
	Rock	0.820		
	Brick	0.840		
	Concrete	1.130		
	Cast iron	0.837		
B. Phase Change				
i) Solid-Solid				
	F_e S	-	138	27
	V_2 O_5	-	72	52
	KNO_3	-	128	51
	L_iSO_4	-	575	125
	Cross linked high density polyethylene	-	-	125
ii) Solid-Liquid				
a) Pure				
	Water (Ice)	-	0	335
	Aluminum bromide	-	97	42
b) Inorganic				
	$LiNO_3.3H_2O$	-	30	296
	$Na_2SO_4.10H_2O$	-	32	241

$CaCl_2.6H_2O$	-	30	170
$Mg(NO_3)_2.6H_2O$	-	90	167
$MgCl_2.6H_2O$	-	117	165
NaCl	-	804	486
$MgCl_2$	-	714	452
Al	-	659	401
c) Organic			
Steric acid	-	70	203
Capric acid	-	36	152
Bees wax	-	62	177
Paraffin	-	74	230
Napthalene	-	80	149
d) Eutectics			
$CaCl_2$-$MgCl_2$-H_2O (41-10-49%)	-	25	95
$Ca(NO_3)_2.4\ H_2O$ + $Mg(NO_3)_2.6H_2O$ (46-33%)	-	30	136
iii) Liquid vapour			
a) Aquous solutions			
H_2SO_4 + H_2O	-	40-60	653
NaOH + H_2O	-	40-60	904
LiBr + H_2O	-	40-60	750
b) Solid hydrates			
Silcagel + H_2O	-	-	990
$NiCl_2.6NH_3$	-	175	250

C. Heats of solutions			
Ammonium nitrate in water	-	80-100	113
D. Chemical reactions			
$N_2O_4 = 2NO_2$	-	21	830
$NO = NO + \frac{1}{2} O_2$	-	>60	632
$Ca(OH)_2 = CaO + H_2O$	-	580	837
$CH_4 + H_2O = CO + 3H_2$	-	-	-

storage mode and basic technology for a given application, and the resultant system performance and economics, will depend on detailed engineering effort. In some cases, the techniques are in arrested state of development, in others, they are developing rapidly, and basic design and operating experience is inadequate. In general, development of a broad spectrum of thermal storage technology appears to be warranted.

With air collectors, solid sensible heat storage materials like rocks, sand, bricks, cast iron, magnesium oxide, etc. are the best materials but require large volume and they cause additional problems like heat exchange and dynamic behaviour. Water is the best heat storage medium for liquid collectors below its boiling point, but above 100 °C tanks require pressurization so that they become quite expensive. Organic oils, molten salts, and liquid matals circumcent the vapour pressure problem but their storage capacities are less. Moreover, the cost of these fluids are far higher than that of water. In some cases a combination[35,36] out of liquid and solid storage media may be appropriate such as thermal oil combined with a rock storage system. In this case cost may be positively affected by reducing the mass of thermal oil, on the other hand the technical complexity increases.

Considerable work has been done in identifying suitable phase change materials (PCM's) for low and intermediate-temperature solar energy storage. Although significant quantities of energy are usually involved in any change of phase, since a high volumetric energy storage density is essential, only solid-liquid or possibly solid-solid tranformations are of practical interest. Material requirements include low cost, high heat of transition, high density, approapriate transition temperature, low toxicity, and longterm performance. However, development of materials for high temperature applications has only recently begun. The solid-solid phase transition gives an additional advantage of efficient heat exchange. Lithium sulphate has a solid-solid phase transition with a very high heat release, three times larger than the heat of melting and comparable with the latent heat of melting of many materials. This high value is related to the onset of rotational disorder of the sulphate groups, demonstrated by neutron diffraction and fast ionic motion of the lithium ions. The high transition temperature of Li_2SO_4 can be significantly reduced by allowing with other sulphates. In this type of transition where there is no melt or volume change, the containment and corrosion problems are much reduced.

Thermal energy can also be stored in chemical bonds by means of reversible thermochemical reactions. For example, $Ca(OH)_2$ will endothermically decompose to CaO and water vapour if it is heated to 520°C at one atmosphere. The water vapour is condensed for storage. When heat is to be supplied from storage, the water is simply mixed with the CaO, and the exothermic reverse reaction of CaO with H_2O produces energy. These reversible chemical reactions are conveniently categorized in terms of applications: Thermochemical energy storage (primarily long-duration storage of thermal decomposition products), thermochemical energy transport (closed loop' chemical heat pipes), and 'chemical heat pump' storage. Current research on thermochemical reactions is on 'Solchem process' proposed by Chubb[37] which uses the reaction $2SO_3 = 2SO_2 + O_2$ to transmit thermal energy from a field of high concentration parabolic collectors to a molten salt heat of fusion energy storage system. Here SO_3 decomposes into SO_2 and O_2 at temperatures in the range 800-1000 C range with absorption of energy; in turn O_2 and SO_2 can be catalytically recombined at 500-600°C to produce SO_3 plus 23 Kcal/mol of chemical reaction energy. The ammonia dissociation reaction has also been proposed for this application by Carden[38]. Some studies[39] have also been conducted on the use of reforming/methanation reaction $CH_4 + H_2O = CO + 3H_2$ to transport solar generated energy for industrial process heat. In this process known as Adam/Eve concept, the reaction is driven to the right hand side by heat (950°C)in a high temperature reactor, and the products frozen and piped at room temperature over long distances to the point of use where the action of a suitable catalyst will release the stored energy. The methane is piped back to the heat source for water addition and regeneration. Use of ammoniated salts are also proposed [40] to perform the dual function of thermal energy storage and heat pumping. Here two thermochemical reactions which have a common vapour species are needed. in the example, $NH_3(g)$ is the transferring gas species, and the two reactions are:

$$Heat + 4MgCl_2.2NH_3 \qquad 4MgCl_2.NH + 4NH_3$$

$$CaCl_2.4NH_3 + 4NH_3 \qquad CaCl_2.8NH_3 + Heat$$

Dischrging mode:

$$4MgCl_2.NH_3 + 4NH_3 \qquad 4MgCl_2.2NH_3 + Heat$$

$$Heat + CaCl_2.8NH_3 \qquad CaCl_2.4NH_3 + 4NH_3$$

In the charging mode solar heat is supplied to the high temperature salt bed ($MgCl_2$) from where the NH_3 is driven to the low temperature salt bed ($CaCl_2$) where it combines with the salt. Heating this low temperature bed slightly will dissociate the NH_3 which returns to the high temperature bed, reacts with the salt exothermally, freeing almost the same amount of heat and at the same temperature absorbed by the high temperature salt bed. The size of a storage system can finally be determined by the characteristics of the delivery profile of the solar collector field and the required consumer profile .

2.5 INDUSTRIAL PROCESS HEAT (IPH) SYSTEM

The detailed knowledge of the way heat is supplied and used in industrial processes is required for economic use of Solar Energy. It is essential to know that how the heat is supplied to each process, the quantity of heat required daily, the temperature at which the heat required, the fuel used and the heat transfer fluid required for transferring the heat. Generally this type of information required for the design of process heat systems is not available. The economic and technical feasibility of any solar industrial process heat system depends on four factors, namely: (i) heat must be supplied in sufficient quantity, (ii) heat must be of adequate quality i.e. at an appropriate temperature, (iii) heat must be transferred directly from the solar collector to the process where it is to be used, and (iv) Solar Energy must be used profitably.

Each industrial plant has unique requirement and hence the SIPH system is to be carefully designed. Because of the specific intermittent nature of solar radiation, SIPH must be backed up with alternate fossil-fuel systems so that the industry gets uniterrupted supply of process heat. Generally SIPH has one of the following three possible modes :

1. Solar Augmentation without energy storage.
2. Solar Augmentation with energy storage.
3. Solar Pre-Heating with and without storage.

The first system appears to be quite attractive because the cost of storage is eliminated but can work only during the day time and can save only upto 30 percent of the total process heat load. If more fraction of the total process of the heat load by solar energy is required then the second alternative is preferred but it will be a costly system. The third alternative can universally and profitably be used in many industries to supply pre-heat boiler hot water or hot air. The pre-heat possibilities exist practically for all process applications.

The process heat in various industries is supplied generally in the following three modes :

(i) Process hot water
(ii) Hot air
(iii) Process steam

In hot water process systems both the direct solar water heater system (Once through type) where the heated water from the solar collector is directly supplied as process heat and indirect solar hot water system where a heat exchanger is used between the collector loop and delivery loop are used. In cold climates, an indirect water system is used with some antifreez mixtures in the collector and storage loop. Direct systems although work at higher efficiency are preferred only in hot climates or during the day time or in special process industries or with some precautionary measures for protecting it against damage due to freezing.

Hot air systems are employed for drying or dehydration processes in industries and such systems are safe from damage due to freezing. The hot air if sufficiently heated by Solar Energy can be directly supplied for drying/dehydration or can be further heated by an auxiliary heater before it goes to the process load. An additional advantage of hot air system is the use of rock bed storage system which can simultaneously be used for charging and discharging of heat. An alternative to the direct hot air system is the use of liquid collectors (since they are better than air collectors) and a liquid-to-air heat exchanger (which reduce the efficiency) and finally heated air can be supplied to the process load.

In process industries, the steam is most common commodity particularly the low pressure steam. The steam can easily be produced using solar energy collectors and which could directly be fed into the industrial steam distribution system without any change in the existing processing practices. All the three options, the flash-steam system, Unfired-boiler-system, and direct steam generation system can be employed to meet the requirement of the process heat in industries. Some of the systems using the above modes will be discussed here in brief.

2.5.1 Hot Water Industrial Process Heat System

The Department of Energy, USA, has funded the design and construction of several IPH systems for various industrial applications ranging from drying of fruits to curing of building materials. In some projects hot water is produced while in others hot air is produced while in few others steam is produced using solar energy for different industrial applications. The aim of building these systems were to demonstrate the potential for solar IPHS to the industrial sector and to encourage their widespread acceptance.

In industries large amounts of hot water in the temperature range of 50-100 °C is required for applications like cooking, washing, bleaching, anodizing, etc. The solar preheated water can also be used as feedwater to boilers. For cleaning or curing purposes, large amounts of hot water is required in food processing and building material industries, and recycling of hot water is not preferred because of contaminations picked up by the water during the process. Hence for such applications, Once-through industrial water heating systems are used. This once through system which is simple in operation can be used with or without any intermediate storage system.

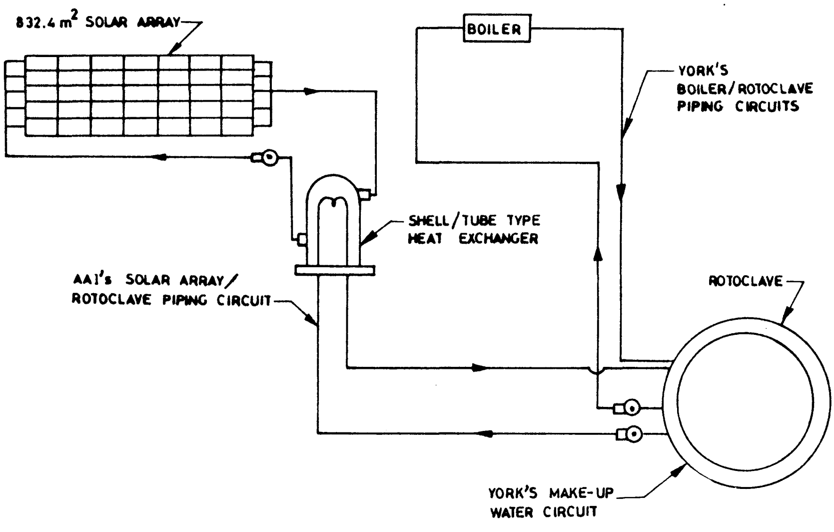


FIG.2.8 SCHEMATIC OF EXPERIMENTAL YORK BUILDING PRODUCTS SOLAR HOT WATER SYSTEM (From Wilkening[41])

An example of the York Building Products; Middetown, USA, solar industrial process hot water system for curing of concrete blocks (41) is shown in fig.2.8. This particular system supplied over 1514 litres/min of hot water in the range of 55 to 85 $^{\circ}C$ to the York Building Products concrete block curing plant. A shell-and-tube heat exchanger is used to transfer heat from the water/ethylene glycol (50:50) collector loop to the process water. The system utilizes 35 linear slat concentrating collectors having concentration ratio 24 and with an effective area of 832.4 m^2 with single axis tracking. The concentrator is composed of individual reflectors which are 2.2 m long and 0.3 m wide. The receiver is an unglazed steel tube fixed at its focus. A unique feature of this application is that the large underground concrete curing area or 'rotoclave' contains about 189250 litres of water and serves as built-in storage. Over thirty percent of the curing energy is supplied by the solar system.

Another example of once through industrial water heating system is at Compbell Soup plant in Scramento, California[42], where solar heated water is used to wash empty and full soup cans on one of the production lines. The system is schematically shown in fig.2.9. Solar energy is collected

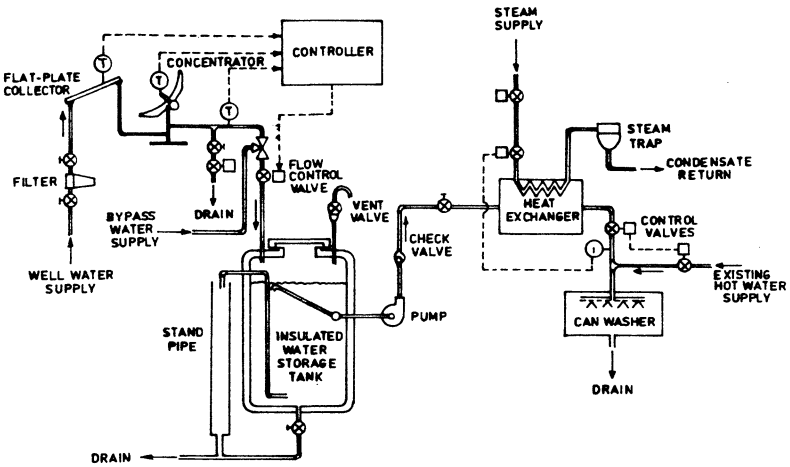


FIG.2.9 SCHEMATIC OF EXPERIMENTAL CAMBELL SOUP PLANT SOLAR HOT WATER SYSTEM (From Vindum and Bonds[42])

using an optimum mixture of flat-plate and concentrating solar collectors. The flat-plate collectors with an area of 413.9 m^2 are single glazed with ordinary blackened absorber plate. These flat-plate collectors preheat the water to

$60\,^{\circ}C$. Final heating of water to $88\,^{\circ}C$ takes place in Acurex (An American firm) model 3001 trough-shaped parabolic concentrating collectors. The trough axis is east-west and the concentrator array area is 267.6 m^2. This installation supplies hot water at a rate of 45420 litres per day during peak season. At other times during the year, the same amount of water is supplied at a lower temperature and brought upto the required $88\,^{\circ}C$ by a steam heat exchanger. A storage tank of 75700 litres capacity is used to ensure a continuous supply of water for two 8-hour shifts, energy working day-demonstrating that solar energy can provide this important industrial need. This system is designed to provide 74 percent of the energy required for one canline of 20 in the plant and therefore the requirement met by the solar system is very small compared to the energy demand by the whole plant.

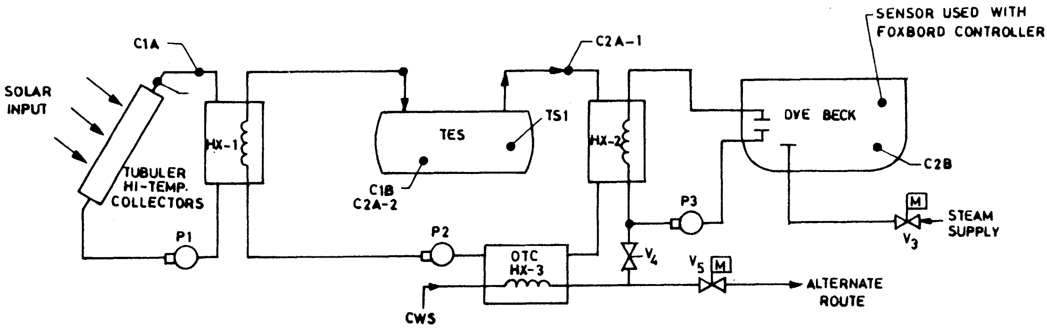


FIG.2.10 SCHEMATIC OF THE SOLAR ENERGY SYSTEM AT LA.FRANCE TEXTILE MILL (From Trice and Cohen[43])

A schematic diagram of a solar hot water system for dye back application in a textile industry in LaFrance, South Carolina is shown[43] in fig.2.10. The system is comprised of three independent circulating loops, connected thermally by means of heat exchangers:

* The solar collector loop (extreme left in fig.2.10) transfers the energy collected by the solar array to the collectorloop heat exchanger (HX-1). The freeze protection is done by using ethylene-glycol/water mixture as heat transfer fluid.
* The thermal-energy storage loop (middle in fig. 2.10), containing 30280 litres sorage tank (TES), transfers the energy either to storage or to the

plant process heat exchanger (HX-2). Energy is apportioned between the two as appropriate, by automatically switching the flow.

* The stainless steel dye-systems loop (extreme right in fig. 2.10) thermally couples the solar system to the dye back water.

The solar system consists of 396 General Electric TC-100 evacuated tube collectors which provide a net area of 544.5 m^2. Solar energy is utilized in a 132 °C maximum temperature water/glycol loop which produces 88 °C water used in a dye beck. A dye beck is a special vat used in the batch process dying operations of a textile plant. According to the estimate this system can provide 80 percent of the energy required by the dye beck during summer and spring months. Even in mid winter, about half of the energy can be supplied by the solar system.

2.5.2 Hot air industrial process heat system

Large amounts of hot air is used in industries particularly in food industries. The two most common ways to supply heated air are: to heat air directly in the colectors or to heat a liquid in collectors and use a liquid-to-air heat exchanger. An example of the first type is schematically shown[44,45] in fig.2.11 which is installed by Lamanuzzi

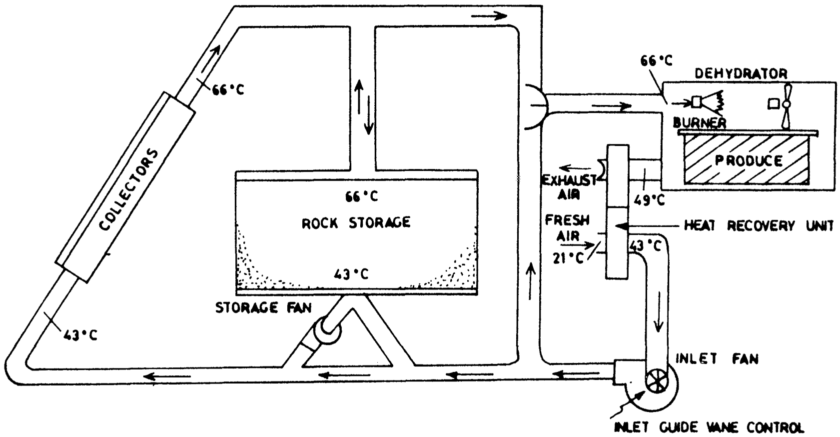


FIG.2.11 SCHEMATIC OF THE SOLAR CROP DEHYDRATION SYSTEM AT LAMANUZZI AND PANTALEO DEHYDRATIONS, FRESNO, CALIFORNIA. (From Carnegie et al[44])

and Pantaleo Dehydrators at Energy Eresno California for drying grapes. This solar dehydrator principally consists of the following four elements:

* A large array of air conventional flat-plate, single glazed and unglazed collectors of 1950 m^2 area.
* A crushed rock bed storage bin of about 396 m^3 volume used to supply heat during the second work shift at the plant.
* A heat recovery wheel of about 3.65 m in diameter and 0.3 m thick fitted with wedge sections of knitted and corrugated aluminium wire, rotates at about 3 rpm. This wheel transfers heat from the tunnel exhaust to the collector array inlet.
* Air ducting to connect the dehydrator and solar collector array, the heat recovery unit and the heat storage facility, complete with fans and an automatic air movement control system.

The unique feature of this dehydrator is the use of heat recovery system in which a portion of the warm moist air leaving the dryer is mixed with the cooler ambient air and recirculated through the collector air while the rest of the warm air is wasted to the atmosphere. Although, this arrangement, raises the collector aray temperature and thus lowers collector array efficiency, the heat recovered more than makes up for the lowered collector efficiency. Experiments conducted on the dehydration of primes have shown that about 84 percent of the energy i.e. 44.7 Giga Joules of natural gas per day can be saved using the solar dehydrator.

An example of heating water in order to supply hot air is shown[46] in fig.2.12, which illustrates the system at the Gilroy Foods Onion and garlic dehydration plant at Gilroy, California. The solar field consists of 432 modules of Owens-Illinois evacuated tube collectors with a total area of 562 m^2. Water is passed through the collector modules in parallel and supplied to one of the two continuous-operation belt dehydrators. A liquid-to-air heat exchanger delivers solar energy to the drying air stream. An inline natural gas burner then heats the air to operating temperature. The cool outlet water from the heat exchanger is then pumped back to the collector array. No storage is provided in the system. When heat is not required by the primary process, the solar energy is used to preheat the plant's steam boiler feed water. Out of the eight continuous drying unit in the factory, solar system was used in one of the units and as such contribute very little towards overall energy saving in the factory.

One of the most cost effective but time consuming process for drying agricultural produce and timber etc. is the use of direct drying houses like green house. In such systems sun light directly impinges on to the product to be dried. This arrangement results in relatively low temperatures and long drying time and larger areas to spread the product.

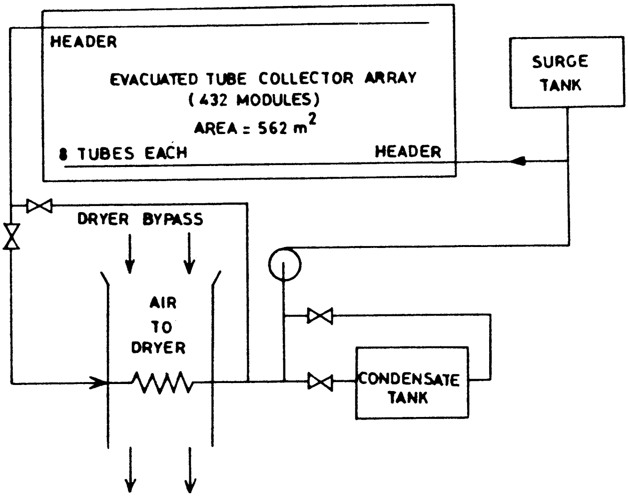


FIG.2.12 SCHEMATIC OF THE WATER HEATING SYSTEM FOR GILROY FOODS ONION DRYING FACILITY (From Graham et al[46])

2.5.3 Steam industrial process heat system

In industries the largest share of process heat (two thirds of all industrial process heat) is met by steam. Significantly different approach is used for producing steam using solar energy then that for air or water process heating. Following three possible ways to supply steam with solar collectors are tried:

* Circulation of pressurized water in the collectors with subsequent flashing to steam in a flash tank.
* Use of high temperature fluid in the collectors with heat transferred to an unfired boiler.
* Boiling of water in collectors.

In a flash steam system, water under pressure, to prevent boiling, is circulated through collectors and flashed to steam across a throttling valve into a seperator. Flashing is a constant enthalpy process that converts the sensible heat of the water into a two-phase mixture of saturated water and saturated steam at conditions prevailing in the flash tank. Steam separated in the flash tank is recirculated through the collector field. To maintain the necessary liquid level in the flash tank, boiler feed water is injected into the pump suction. A schematic diagram[47] of a flash-steam system designed for Johnson and Johnson Company, Sherman, Texas for providing process steam for the

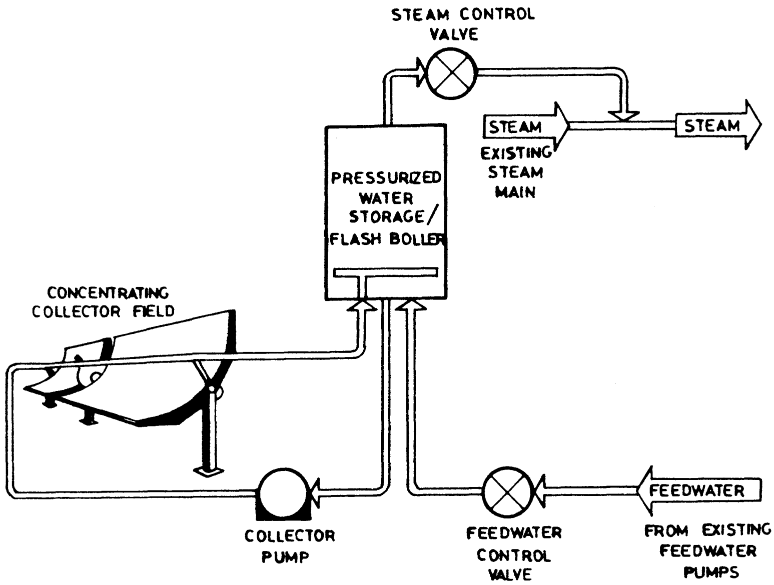


FIG.2.13 SCHEMATIC OF THE SOLAR PROCESS STEAM SYSTEM USING A FLASH TANK AT JOHNSON AND JOHNSON COMPANY TAXAS. (From Youngblood[47])

manufacture of gauge and baby products is shown in fig. 2.13. Acurex model 3001 parabolic relfectors with orientation north east-south west to align with the building orientation with a total aperture area of 1070.2 m^2 are used to generate steam. Pressurized water is circulated with a pump of 18.5 KW through collectors and into a flash boiler. High temperature and pressure water is stored in 18920 litre flash boiler. The heated, pressurized water is flashed to steam and supplied to the plant steam main through a pressure regulating valve. Makeup water to the system is supplied from the existing plant boiler feedwater system. Freeze protection is accomplished by circulating boiler water at a reduced rate through the collectors; the throttling valve is bypassed in this mode. With this demonstration the technical and economic feasibility of generating low-pressure steam, 446 K, 862 KPa for an industrial process is demonstrated. It is estimated that the system will be

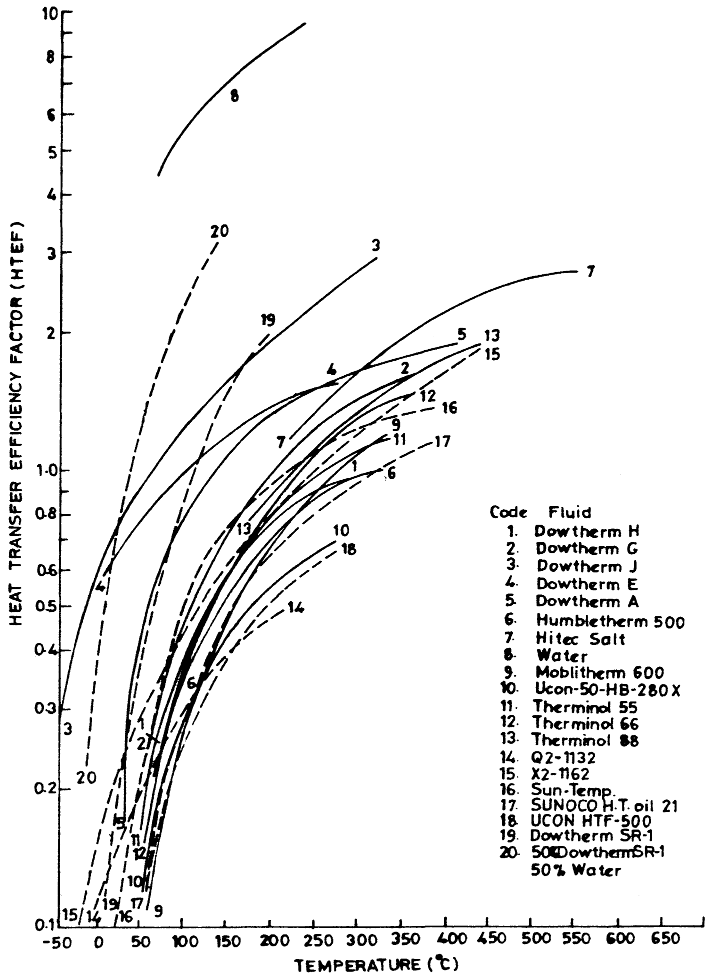


FIG.2.14 HEAT TRANSFER EFFICIENCY FACTORS FOR VARIOUS HEAT TRANSFER FLUIDS USED IN SOLAR COLLECTORS (From Kutscher et al[48])

able to save 51.67 m^3 of oil and 56640 m^3 of natural gas per year. The advantage of the flash system is that relatively simple controls are used and boiler water can be preheated. The disadvantage being that relatively little steam is produced efficiently by this process.

In the unfired-boiler steam system, some organic fluid is pumped through the collector field and then to the unfired boiler. This hot organic fluid vaporizes the water in the unfired boiler. This saturated steam is fed to the existing steam header, which delivers the energy to the industrial process. As the steam is generated, additional condensate is supplied to the boiler. One should be very careful in deciding the heat teansfer fluid. Fig. 2.14 shows the heat transfer efficiency factor[48] for some 20 heat transfer fluids generally used in solar energy collectors. From this figure it is seen that water is the best heat transfer fluid but is not recommended in the unfired

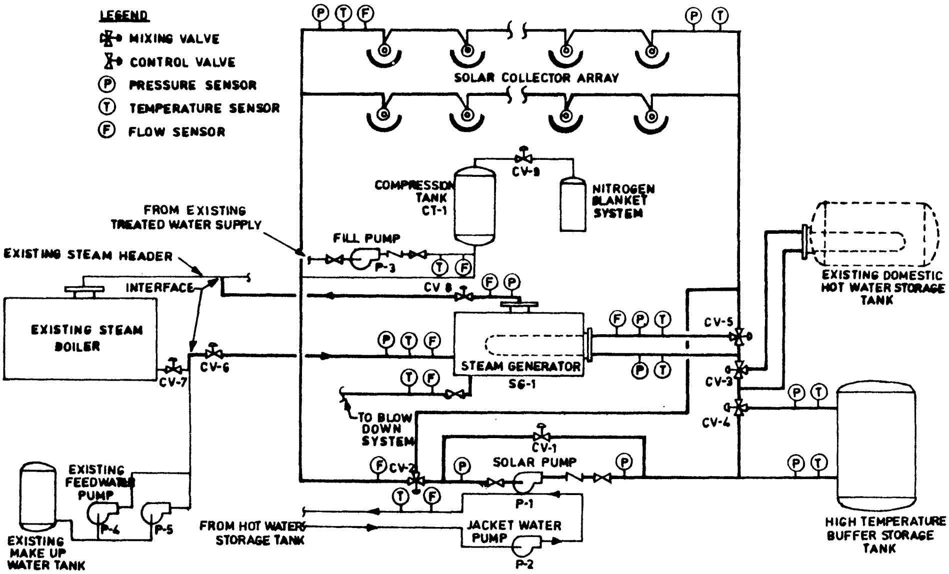


FIG.2.15 SCHEMATIC OF SOLAR PROCESS STEAM SYSTEM USING AN UNIFIED BOILER AT HOME LAUNDRY CALIFORNIA (From Eldridge et al[49])

steam system because of its high vapour pressure and freezing. Therefore, a low-vapour-pressure and non-freezing hydrocarbon or silicone oil is preferred. But these fluids may create other problems such as leaking, poor heat transfer properties, becomes viscous at low temperatures, expensive, etc. A schematic diagram[49] of a unfired steam system located at Home Laundry, Pasadena, California, for the supply of water and steam for commercial laundry is shown in fig.2.15. This system is designed for The Home Laundry to meet 25 percent of the Laundry's annual steam and 21 percent of its annual, combined steam and hot water requirements. The final design[50] consists of 951 m^2 of Del, single axis tracking, parabolic trough concentrating collectors. The solar energy collection loop is a closed-circuit piping system, pressurized with nitrogen. Water is used as a heat transfer fluid which is circulated by a pump through the collectors to a steam generator, then back to the collectors. The system is designed to produce hot water at a temperature of 250 ^{o}C and at a pressure of 2895 KPa. When solar energy is strong, it is used to generate steam which is not stored and directly used. When there is insufficient solar energy for the generation of steam, it is used to heat water in a 90840 litre process hot water tank for further use in the laundry. It is estimated that this system will be able to save about 95.4 m^3 of oil every year.

There are several controls in this system and each control has to play its own role. When the fluid temperature at the outlet of the solar array is 215 ^{o}C, the steam generator control valve CV6 and CV8 modulate to produce steam at 170 ^{o}C. If the collector fluid temperature drops to 182 ^{o}C, the back up boiler provides the required load. Another control inverts the collector array for overnight storage or for a loss of coolant episode or for power or pump failure. Storage fluid is circulated through the collectors when the ambient temperature is less than 1 ^{o}C to protect them from damage due to freezing.

Another simple and more dependable solar steam system named as solar boiler-auxiliary boiler system is described in reference 51 and the same is schematically shown in Fig.2.16. This system is quite flexible and solar heat is independently and continuously collected unconcerning with what is taking place at the end. The controls are relatively simpler and maximum solar energy is collected since because of the advantage of boiler auxiliary preheat. In this design minimum energy is used by the auxiliary boiler. Generally the flow rate from the solar storage to the boiler is maximum except when the flow controller FC2 shows that the steam produced by the auxiliary boiler is below the minimum required value, then the valve V4 gets activated by FC2 and supplies the reduced amount of energy from the solar storage. In this case of reduced flow rate in the solar boiler, the water level in the boiler rises. The increased water level is sensed by level controller LC which opens valve V2 increasing the flow rate through auxiliary system as required.

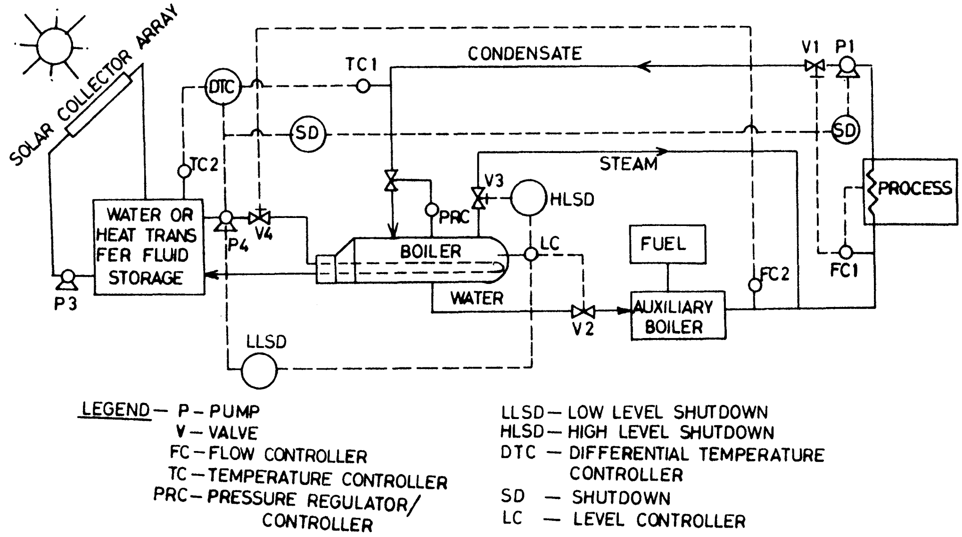


FIG.2.16 SCHEMATIC OF SOLAR PROCESS STEAM SYSTEM INTERFACED WITH A CONVENTIONAL AUXILIARY BOILER (From Reference[51])

During sunny days under normal conditions, the supply of solar heat may exceed the process demands. Hence solar boiler could boil dry which is prevented by means of low level shut down (LLSD) operating delivery pump P2. As the temperature in the storage tank reduces, the boiling ceases and the water level in the boiler rises. The liquid level is sensed and high level shutdown (HLSD) switch closes valve V3 preventing the further rise of water level. At this point the auxiliary boiler supplies the required steam (100 percent) and the solar heater works only as a preheater until TC1 > TC2, shutting off the solar pump P2. With the help of flow controller FC1 and valve V1, the process flow requirement can be adjusted..

Direct steam generation through solar collector array appears to be attractive but has not been seriously attem-

Table 2.6 Solar industrial process heat projects in the United States (From Kutscher et al[48]

Company 1	Location 2	Process application 3	Process temperature (°C) 4	Size of collector array(m^2) 5
HOT WATER SYSTEMS				
1. Sweet Sue Kitchens Inc.	Athens, Alatama	Preheat boiler feedwater	55	1538
2. American Linen Supply Co.	El Centro, California	Preheat boiler feedwater, wash water	94	1129
3. Aratex Service Inc.	Fresno, California	Heat process water	50-70	624
4. Iris Images	Mill Valley, California	Film processing	24-38	3530
5. Stauffer Chemical Co.	Oxnard, California	Chemical processing	52	3530
6. Thirmack Enterprises Inc.	Redding, California	Preheat boiler feedwater.	70-95	627

Table 2.6 cont.

	1	2	3	4	5
7.	Campbell Soup Co.	Sacramento, California	Can washing	80-90	681
8.	Caterpillar Tractor Co.	San Leandro, California	Heat wash water	110	4682
9.	Salz Leathers Inc.	Santa Cruz, California	Tanning and Finishing	30-70	3270
10.	Barkley Meat CO.	S.Lake Tahoe, California	Sanitation	80	232
11.	Anheuser-Busch Inc.	Jacksonville, Florida	Beer pasteur-ization	60	427
12.	Dana Corp. Spicer Clutch Div.	Auburn, Indsana	Parts washing	55	87
13.	Oscar Meyer Corp.	Perry, Iowa	Meat processing	85	3746
14.	Sohio Petroleum Co.	Grants, New Mexico	Uranium-ore processing	60	630
15.	General Extrusion Inc.	Youngstown, Ohio	Solution heating	70-80	409

Table 2.6 cont.

	1	2	3	4	5
16.	York Building Products Inc.	Harrisburg, Pa.	Concrete-block curing	57	856
17.	Nestle Enterprises Inc.	Santa Isabel, Pr	Juice Pasteurization	99	4645
18.	Riegel Textile Corp.	LaFrance, South Carodina	Heat dye-beck water	88	620
19.	Coca-cola Bottling Co.	Jackson, Tennessee	Bottle washing	-	880
20.	Tyson Foods Inc.	Shelbyville, Tennessee	Poultry processing	54-60	4963
21.	Mary Kay Cosmetics Inc.	Dallas, Texas	Sanitizing	60	93
22.	M & M Mars Corp.	Waco, Texas	Heat water for Cafeteria	-	-
23.	Easco Photo	Richmond, Virgina	Film processing	46	-

Table 2.6 cont.

1	2	3	4	5
HOT AIR DRYING SYSTEM				
24. Gold Kist Inc.	Decantur,Alapama	Preheat dryer air	80	1220
25. Lamanuzzi & Pant-aleo Food Inc.	Fresno, California	Drying	62	1951
26. Gilroy Foods Inc.	Gilroy, California	Preheat dryer air boiler feedwater	90	553
27. Western Alfalfa Corp.	Lawrence, Kansas	Preheat dryer air	204	-
28. LaCour Kiln Services	Canton, Miss. Inc.	Lumber drying	82	234
29. U.S. Gypsum Co.	Sweetwater,Texas.	Board drying	482	20628
THERMAL-LIQUID HEATING SYSTEMS/DIRECT FIRED PROCESS HEATING				
30. Ergon Inc.	Mobile, Alabama	Heat thermal liquid	55-88	1873
31. Atlantic Rich-Field Oil & Gas Co.	Bakersfield, California	Heat thermal liquid	293	16826
32. Valley Nitrogen Producers Inc	El Centro, California	Direct process	870	58839

Table 2.6 cont.

1	2	3	4	5
STEAM SYSTEMS				
33. West Point Pepp-erell Inc.	Fairfax, Alabama	Fabric drying	160	698
34. Provident Energy Co.	Mobile, Arizona	Produce refinery Steam	370	66177
35. Exxon Corp.	Bakersfield California	Oil recovery	260	23616
36. Exxon Corp.	- do -	- do -	300	40133
37. Petro Lewis Corp.	- do -	- do -	260	31660
38. Home Loundry	Pasadena, California	Produce steam, preheat wash water	182	603
39. NL Industries Inc.	Newberry Springs, California	Hectosite drying	189	962
40. Tropicana Products Inc.	Bradenton, Florida	Ice-block thawing	155	929
41. Dow Chemical Co.	Dalton, Georgia	Latex manufacturing	185	922

Table 2.6 cont.

	1	2	3	4	5
42.	Bleyle of America Inc.	Shenandoah, Georgia	Produce steam, heat water	400	4587
43.	Hilo Coast Processing Co.	Pepeeko,	Sugar-cane	204	4682
44.	Stauffer Chemical Co.	Henderson, Nanada	Chemical Stripping	186	984
45.	Southern Union Co.	Hobbs, New Mexico	Produce refinery main steam	190	936
46.	Gulf Mineral Resources Co.	San Mateo, New Mexico	Uranium-ore	185	21593
47.	U.S. Steel Chemical Co.	Haverhill, Ohio	Polystyrene processing	190	4645
48.	One-Ida Foods Co.	Ontario, Oregon	Produce steam	214	884
49.	Bates Container Inc.	Ft.Worth, Texas	Produce steam	188	3225
50.	Lone Star Brewing Co.	San Antonio, Texas	Produce steam	178	878
51.	Johnson & Johnson	Sherman, Texas	Produce process steam	174	1070

pted. Only some recent investigations by Murphy and May [52], and Pederson and May[53] have proven its significance for the supply of heat to process industries. Presently there is no working system on this concept. The system schematic would be similar to that of flash-steam system but without a flash valve.

The idea of using heat pump along with flat-plate collectors to boost hot water upto 90 ^{0}C temperatures for providing process heat for a cleaning tank in the aluminium anodizing line of General Extrusions, Inc. Youngstown, Ohio, USA[54, 55] has already been tried. The system installed at General Extrusions consists of a solar collector array (304 m^2) of their own model LTC-367 with a concentration ratio of 3.67 and a high temperature heat pump called a Templifier made by Westinghouse[55]. The output of this Templifier is 70560 Kcal/hr, and the source water from the solar collector storage is at a minimum temperature of 40 ^{0}C. All the five following possible modes can be switched from the main control board to provide heat to the cleaning tank of the anodizing line:

* Heating directly from solar collectors.
* Heating incorporating the solar collectors, heat pump and storage.
* Heating from solar collectors and heat pump.
* Heating from storage.
* Heating storage from solar collectors.

It has been estimated by performing computer studies that for a given fraction supplied by solar, the solar-only system requires about twice as much collector area as does the Solar-Assisted-Templifier (S-A-T) systems. In some cases it has also been shown that S-A-T system can save over 50 percent in the initial investment.

The U.S.Department of Energy (DOE) has given several contracts to advance the state of the art the solar industrial process heat ranging from small collector area to large collector area and for different solar applications like hot water, hot air, low-temperature steam and intermediate temperature steam. A list of SIPH projects in USA which are either funded by DOE or due to private efforts is given[48] in table 2.6. Some of these projects are fully operational while others are only at the design or near operational stage. Many experiences are gathered and problems encountered in these projects. These will be discussed later in this Chapter.

2.6. EXAMPLES OF SOLAR PROCESS HEAT SYSTEMS

Although application of solar energy to industrial process heat systems is one of the most logical solar thermal applications in the near-term and in many countries including Germany [56], Isreal[57], Australia[58], U.K.[59], Japan[60], India[61] and U.S.A.[62],work in this direction is in progress but very little information on the actual system, performance, data and experiences available. Here examples of three solar industrial process heat systems which are fully operational are described.

2.6.1. Shallow solar pond water heaters

The shallow solar pond (SSP) is a solar energy collector and storage system that is intended to supply large amount of heat for industrial applications at a cost that is competitive with fossil fuel. Its use for the conversion of solar energy into low grade thermal energy has been a subject of intensive investigation for a number of years in USA[63,64], Israel[65], and India[66].

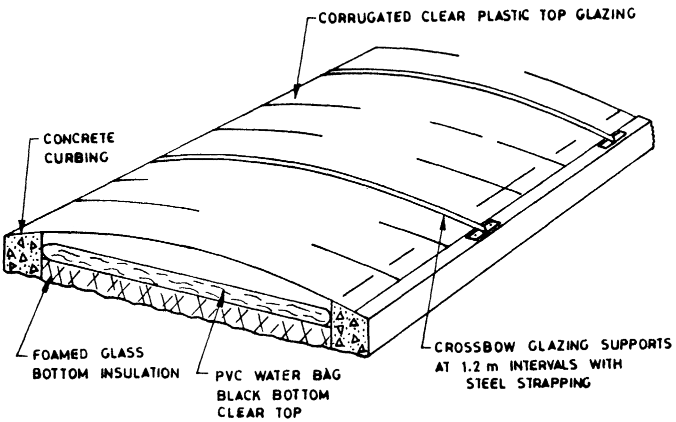


FIG.2.17 SCHEMATIC DIAGRAM OF A SHALLOW SOLAR POND WATER HEATER (From Dickinson et al[63])

A shallow solar pond is schematically shown in fig.2.17. It essentially consists of a large (as large as 3.5 m x 60 m) plastic bag or pillow constructed from a clear upper plastic film and a black lower film, placed in an enclosure with a clear plastic upper glazing and foamed glass insulation on the rear side. The depth of the water within the bag is normally in the range of 4-15 cm. Typical peak temperatures for a shallow solar pond range from 60 °C in the summer down to 40 °C in winter. With a nominal 7.5 - 10 cm depth, the annual efficiency of the shallow solar pond system is about 50 percent. With smaller depths, higher temperatures can be obtained resulting in lower collection efficiency. The water should be withdrawn from SSP before sunset (or more precisely when the collection efficiency approaches zero) for utilization or storage.

The Lawrence Livermore Laboratory has built and operated a shallow solar pond facility at the Sohio-Uranium milling plant near Grants, New Mexico. Uranium is dissolved from the ore in a chemical leaching process requiring dilute hot sulfuric acid. This process requires about 30 litres/s of water at 60 °C for 24 hours a day. The factory is using the oil for the supply for hot water and had plans to supplement with solar heating. The Sohio facility consisted of three full size, 3.5 m x60 m, SSP modules of different designs alongwith cold and hot water storage facility and instrumentation for monitoring the performance. Each module heats about 25000 litres of water per day between 54-60 °C in summer afternoons and 29-32 °C in the winter afternoons. The system is used in the batch process in which the pond is filled in early morning with cold water and drained in an insulated storage reservoir in the afternoon. The annual collection efficiency was calculated to be 48 percent. With this annual collection efficiency, and to provide about 80 to 90 percent of the required process heat in the mill, the total SSP area required is about 2.1 hectares, which corresponds to 100 SSP modules with dimensions 3.5 m x 60 m. The whole facility will require about 4 hactares of land including service roads, curbings, and water storage reservoirs.

2.6.2. Solar beer pasteurizer plant at Adelaide, Australia.

Several solar demonstration systems for the supply of process heat in industries are being used in Australia[67] to obtain operating experience and performance data with real systems, which can then be used as a data base for the design and economic assessment of future systems. One of the demonstration is made at the South Work Brewery in Adelaide to supply heat to a beer pasteurizer. This system is schematically[68] shown in fig.2.18. The solar array consists of 178 m^2 flat-plate collector having copper absorbing plates with Chrome-black selective coating. The collectors are arranged in three seperate arrays, of which two are made up of double glazed units with 114.5 m^2 area and the third 63.5 m^2 are single glazed. Solar heated water is stored into two well insulated tanks with a total capacity of 52000 litres, one made from bolted mild steel and other welded stainless steel. These tanks were available at the Brewery. All the pipes connecting the collector and storage tank and of the process circuit are of copper material and are heavily insulated. The pump in the solar collector and storage tank is controlled by a differential controller which senses a 2 °C temperature rise. The process pump draws water from the pasteurizer at a rate of 10 litres/s, pumping the water into the bottom of the storage tank and causing the same flow from the top of the tank through the supply line to the pasteurizer. This pump is controlled simultaneously by both on 'on-off' and also a differential temperature controller. The steam supply is controlled by means of a pneumatic proportional controller which operates over the range of $66.5\,^{\circ}C$ (fully closed) to $63.5\,^{\circ}C$ (fully open). The controls are able to control the sump water temperature within $64.5 \pm 0.5\,^{\circ}C$. As is seen in the figure, in this process, the sealed bottles of bear are conveyed through the pasteurizing machine, passing through a series of zones where they are progressively heated from $2.8\,^{\circ}C$ to the pasteurizing temperature of about $80\,^{\circ}C$ and then cooled by water sprayed from seperate sumps located along the bottom of the pasteurizer. Here the solar system supplies hot water to the first section which is maintained at $65 \pm 0.5\,^{\circ}C$.

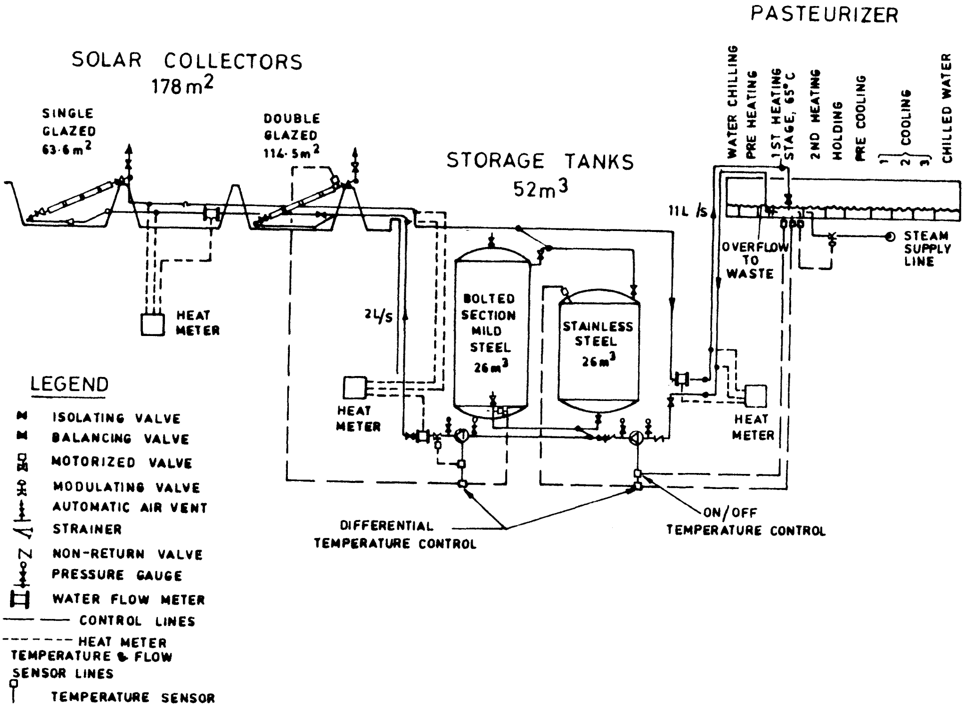


FIG.2.18 SCHEMATIC DIAGRAM OF DEMONSTRATION SOLAR INDUSTRIAL PROCESS HEAT SYSTEM AT SOUTHWORK BREWERY, AUSTRALIA (From Read[68])

Table 2.7 Estimate of load and solar performance of Southwark Brewery, Australia (From Read[68])

	Jan.	Feb.	March	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
Estimate of Process Load Plus 15% Heat Loss-Monthly - GJ	35.4	100.8	125.1	91.4	17.5	73.7	152.7	141.9	97.7	122.9	189.0	131.5	1279
Process Temp. 65°C Collector Heat output-MJm^{-2} day	9.2	8.84	7.3	4.9	3.1	2.0	2.2	3.4	5.1	7.0	8.0	9.1	
Area Needed for Monthly Load-m^2 plus 20% system loss	148	488	661	641	216	1473	2661	1614	742	677	944	560	
Total Energy Collected per month GJ using 178 m^2	50.76	44.06	40.33	26.43	17.27	10.68	12.25	18.76	27.18	38.79	42.72	50.05	379
Estimated Radiation Adelaide 20° Inclination MJm^{-2} day^{-1}	25	23.9	20.6	15.2	12.2	10.2	10.9	13.8	18.3	21.9	23.7	24.3	

Some of the performance data[68] of this system is shown in table 2.7. Because of the confidentiality of some aspects of the Brewery production, only the estimated monthly loads for the particular heating process are given. From this table it is seen that the anticipated annual energy collected is 379 GJ. If 20 percent heat loss is allowed then it is seen that the solar contribution to the process load is 24 percent.

2.6.3. Solar laundry plant, Meitetsu Cleaning Company, Japan.

A solar retrofit system is described by Nanya et al[60] to provide process steam at 175 $^{\circ}$C in the laundry plant of Meitetsu Cleaning Company of Japan. The complete system is schematically shown in fig.2.19. The present laundary system consists of a boiler (steam generation unit of 4.8 tons per hour) a hot water tank (21 m^3), a feed water tank (4.7 m^3), a drain heat exchanger (heat transfer area of 4.7 m^2), a drain heat exchanger and loads. In the laundary system, the process steam demand is about 30 tons per day at 0.883 MPa at three temperature levels namely, 125, 145 and 175 $^{\circ}$C. There are three sections in the plant and in the first section, 10.8 percent of total steam consumption is required for laundary process at 0.883 MPa and 25 percent on drying process at 0.441 and 0.24 MPa. Waste steam from these processes at 100 $^{\circ}$C goes to the drain heat exchanger where hot water for the washing machine is heated to 80 $^{\circ}$C.

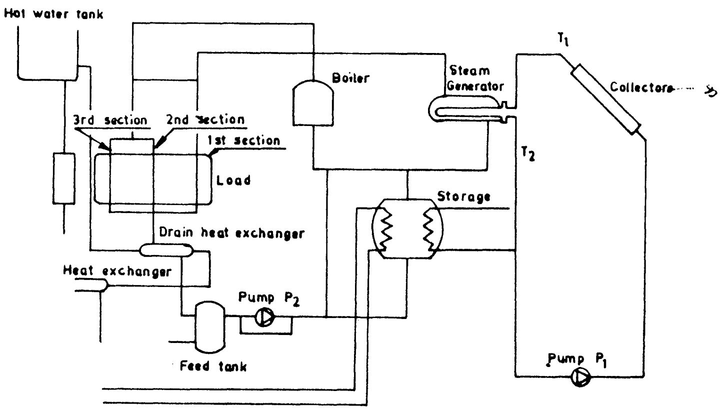


FIG.2.19 SCHEMATIC DIAGRAM OF SOLAR STEAM SYSTEM FOR LAUNDRY PLANT MEITETSU CLEANING COMPANY, JAPAN (From Nanya et al[60])

In this present system, solar system consisting of solar collectors, a solar steam generator, and a storage are added. The solar collectors proposed to be used are of evacuated tube with internal reflector with heat pipe system. The effective area is 540 m^2. Water under pressure to prevent boiling at working temperature is used as working fluid. The flow rate is being maintained at 1.15 Kg/s. Water from the collectors is sent to the steam generator only when it is at 200 ^oC to generate steam at 0.883 MPa, otherwise it is recirculated in the collectors. If the outlet temperature from the steam generator remains above 170 ^oC, water at 170 ^oC is circulated through heat exchanger coil of the storage tank and heated water stored in the hot water tank. Steam from the solar steam generator is supplied directly in the first section of the laundary. Heat from the storage tank is transfered to heat the water in the hot water tank. Stored energy in the hot water tank or storage tank can be used on the next morning.

From the system performance and economic analysis it is concluded that the steam generator for the solar system has its optimum size and that the system with the hot water storage tank is advantageous for the safety and economy of the system.

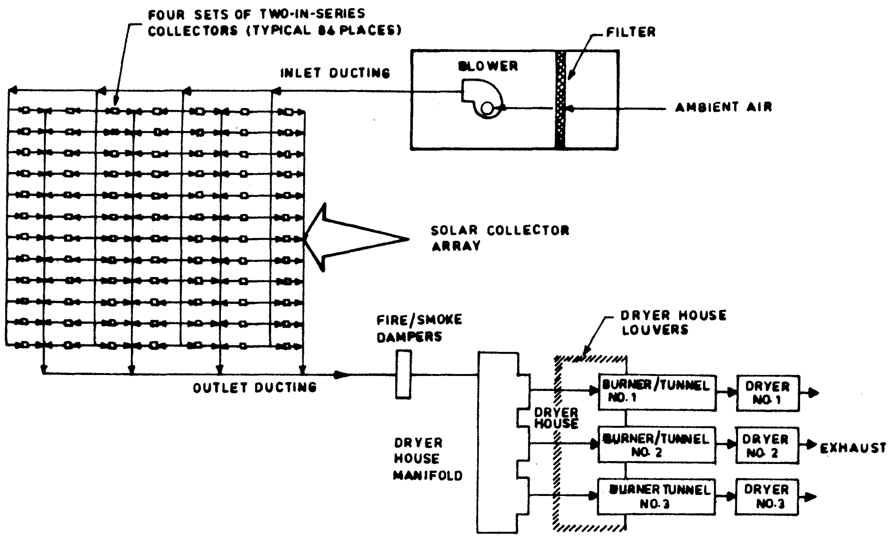


FIG.2.20 SCHEMATIC DIAGRAM OF SOLAR DRYING SYSTEM AT GOLD KIST, INC., DECATUR, ALABAMA (From Guinn[69])

2.6.4. Solar Drying Plant at Gold Kist, Inc. at Decatur, Alabama.

The Gold Kist, Inc., extraction plant at Decatur was constructed in 1973 to extract oil from soyabeans and provide as a product the oil and soyabean meal made from the meal and bulk after the oil is extracted. This plant employs three large, continuous-flow dryers that are capable of processing 3000 bushels of grain per hour each. Each dryer requires approximately 4248 m^3/min. of combustion air. During the year 1975 the energy consumption for drying was about 184×10^{12} Joules. As a part of series of demonstrations of solar Industrial Process Heating of DOE a system was designed to supply part energy for the drying operations. The system is schematically shown[69] in fig.2.20.

The solar system consists of 672 modules of solaron collectors (1200 m^2) arrayed in 84 sets of 8 modules encompassing 4 two-in-series pairs of collectors for heating air which is supplied to the drier. This air is mixed with additional ambient air, heated by oil to the desired temperature and used for drying operation. Since the collectable energy was a small fraction of the energy requirements for one dryer, a storage system was not considered justifiable. The principal components of the subsystem are: the air filter, blower, inlet ducting, dryer house, manifold, and fire dampers. The relative poisitions of these items are shown in fig.2.20. The dryer house manifold provides the interface between the solar drying system and the conventional drying processes. The collectors are kept under positive pressure to avoid ingestion of dirty air into the system by providing the blower and filter on the upstream side of the collectors.

Preliminary performance results indicated that efficiency gets degraded due to accumulation of plant affluents. Fossil fuel savings was also reported to be low due to low utilization and small size. One of the significant operational problems of this installation was the deposition of industrial pollutants into the collectors. Some oil also gets deposited on to the panels which helps in the adhesion of very fine particulates which can only be cleaned by using detergents with some mechanical action. Bigger system is planned in this plant.

2.7. PROBLEMS WITH SIPH SYSTEMS.

As is seen above, several SIPH systems are built and installed, but practically in all of them several problems have been observed which are to be avoided to make SIPH systems successful. The feasibility of SIPH system depends on cost, performance, and reliability of the solar system

vis-a-vis the conventional system[70]. No systematic study of these factors have been carried out and Cassamajor and Wood[71] have pointed out the need for such an investigation. One of the most crucial parameter for the future of SIPH system is the energy payback of the system. Generally energy paybacks[72] for most active solar water heaters and solar house heating systems ranges from 5 to 20 years. But industries want SIPH system with a payback of less than 7 years. This is considered as an absolute requirement. If the payback is not fast, the installation is not made. The installed cost of the system should be reduced and maximize the collector output. Generally in a SIPH system the collector arrays cost from 30 to 50 percent of the total cost. Hence the efforts should be made to reduce the collector cost. But the collector must be durable and capable of withstanding some abuse during handling, transportation and operation. Collectors have proved to be a major problem in the field because of degradation of absorbers, leaking
vis-a-vis the conventional system[70]. No systematic study of these factors have been carried out and Cassamajor and Wood[71] have pointed out the need for such an investigation. One of the most crucial parameter for the future of SIPH system is the energy payback of the system. Generally energy paybacks[72] for most active solar water heaters and solar house heating systems ranges from 5 to 20 years. But industries want SIPH system with a payback of less than 7 years. This is considered as an absolute requirement. If the payback is not fast, the installation is not made. The installed cost of the system should be reduced and maximize the collector output. Generally in a SIPH system the collector arrays cost from 30 to 50 percent of the total cost. Hence the efforts should be made to reduce the collector cost. But the collector must be durable and capable of withstanding some abuse during handling, transportation and operation. Collectors have proved to be a major problem in the field because of degradation of absorbers, leaking
through them and therefore spoiling the insulation and breaking of glazings. Another big constraining factor for the future of SIPH system is the limited land availabilities in existing factories. Due to the high cost of piping it is preferred that the collector array should be located close to the process plant. In Urban Industries the collector arrays can be put on to the roof by making proper arrangements if the roof can bear extra load. In some process industries like that of mining and some chemical and food processing located remotely, there can be sufficient of surplus land available and the collector array can be mounted on to the ground.

Apart from the above two major constraints for SIPH system there are several operational, and system design problems.

Some of the problems with SIPH systems are listed by Read[67], Kreider[19], Rabl[5] and Dickinson and Casamajor [73]. There is a range of collectors available operating at different range of temperatures and it is necessary to decide the collector which gives the maximum cost effectiveness. The actual energy requirement for a particular process and load pattern are generally unknown and therefore a continuous monitoring equipment is required to get the actual values. The mounting of the collector should be as simple and low cost as possible with optimum tilt and orientation. All SIPH systems require sofisticated controls which make or break a circuit and therefore must be analyzed for function in all possible modes and should be able to operate under all dusty, rainy, windy, and industrial environments. Similarly trackers and its control should also be able to withstand the adverse conditions. It is observed in many SIPH systems that the joints are not properly made and they generally leak which spoils the insulation and the working space. Another major problem is the deposition of fine industrial pollutant on to the collector glazings which becomes difficult to clean. Therefore, the collectors should be mounted to minimize particulate pollutant fall out.During transport, the reflectors get misaligned and therefore alignment and levelling of absorber relative to the mirror is required at the sight.

If a storage is needed then its size should be optimum consistant with the operating requirements of the system and the cost of storage. Maintenance can be a nuinsence in a SIPH system and therefore the most suitable materials with proven and reliable construction techniques should be used to keep the maintenance cost to the absolute minimum.

Although there are considerable improvement in the designing of SIPH systems and many sophisticated computer simulation techniques and other simple techniques are available, however there is a scope of developing a simple design method for SIPH system to make the system optimally cost effective.

2.8 A DESIGN METHOD FOR SIPH SYSTEMS

Several methods of sizing solar energy active systems are developed recently to varying degree of sophistication. Probably the most accurate, widely accepted, and the most sophisticated computer simulation program is TRNSYS (acronym for'A transient system simulation program') developed by the University of Wisconsin (USA) and first released in March 1975 and since then updated several times[74]. This program does not optimize the system either on a performance or on an economic basis and requires large computer with a fair amount of expertise. Realizing the need for a sizing procedure, several simple design methods are developed, which require a simple computer or a hand-held programmable calculator or extensive hand calculations or a handbook of tables or a simple step by step calculation with memory only in a reasonable amount of time. The most widely used design method for sizing solar active heating systems is the f-chart method[75,76], which is a correlation of hundreds of simulation performed for a system using TRNSYS. Several assumptions involved in its development limit the applicability of f-chart method to common type of water heating systems. The f-chart method is applicable:(1) for flat-plate collectors where Hottel-Whillier-Bliss equations[77] are applicable, (2) for separate preheat and auxiliary tanks, (3) for a preheat storage tank with heat loss coefficient of 0.42 $W/m^2\,^{\circ}C$, (4) for mains temperature between 5 to 20 °C, (5) for water set temperature between 50 and 70 °C, and (6) with no auxiliary tank losses. These collector type and temperature limitations do not permit the use of f-chart method to design most SIPH systems.

Earlier to the f-chart design method, a monthly average hourly utilizability method for flat-plate collectors known as $\bar{\Phi}$-method was developed by Whillier[78], and Hottel and Whillier[79] and later detailed design curves were given by Liu and Jorden[80]. Klein[81] and Collares-Pereira and Rabl [82] later used the idea of daily utilizability and given $\bar{\bar{\Phi}}$-method which reduces considerably the computational time. The drawback of f-chart and $\bar{\bar{\Phi}}$-method were removed by Klein and Beckman[83] and given a more comprehensive $\bar{\Phi}$, f-chart method suitable for closed-loop liquid based solar heating systems. This original $\bar{\Phi}$, f-chart method assumes a closed loop system, with a set outlet temperature to the load and a minimum return temperature from the load. Recently[84] the $\bar{\Phi}$, f-chart method is modified which is suitable for closed loop as well as open loop solar heating systems and does not put any limitations on the outlet temperature, the mains water temperature, the preheat tank loss coefficient or the auxiliary tank loss coefficient and is appliable to one-tank or two-tank systems. This modified method can be used to analyse a wider range of systems than f-chart and the collectors may be flat-plate, compound parabolic concentrator or imaging concentrator. Thus for certain SIPH systems, the modified $\bar{\Phi}$, f-chart method can be used for design purposes.

A simple step by step method for designing and optimizing of SIPH systems which have constant daytime load without storage, including a quick procedure for selecting the most cost effective collector and calculation of pumping energy is discussed by Gordon and Rabl[57]. This procedure makes use of commonly available annual average climatic, load and collector characteristics. A detailed computer simulation program SOLIPH was developed by Kutscher[85] specially for SIPH systems. Based on hourly weather and solar radiation data for 26 stations, thousands of SOLIPH runs were made to provide a large data base for developing empirical correlations for various system components. The agreement between the empirical correlations and the SOLIPH runs was found to be better than 4 percent (rms error). Based on these empirical correlations and SOLIPH runs, a simple design method to size the system components and to help make the trade-offs specially for SIPH systems is discussed by Kutscher et al[48] for all type of collectors and SIPH systems. Gee[86] has discribed this method with an example of special case of SIPH system using parabolic trough steam generation system. A few simulation and design methods for solar heating systems are recently reviewed by Garg[87].

Here a simple step procedure as described by Kutscher et al[48] is discussed in brief. The solar energy contribution depends on many parameters including collector type, properties of heat transfer fluids heat exchanger effectivness, storage size, the temperature at which process heat is supplied, process heat load and its duration. Stepwise procedure[48,86] is given below:

Step 1: Collect the following basic information.

* Collect the climatic data like longterm yearly average daytime temperature $\bar{T}_a$, longterm yearly average night time temperature $\bar{T}_{a,n}$, and longterm yearly irradiance data $\bar{I}$ for a particular place under consideration.
* For the solar collector (flat-plate or parabolic) which is to be used find out values of $F_R U_L$, $F_R \eta_0$, and its incident angle modifier $k_{\zeta\alpha}$. These can be obtained from the collector as per ASHRAE[88] standard 93-77 procedure. Here F_R is the collector heat removal efficiency factor, given as:

$$F_R = \frac{\dot{m}_c C_p}{U_L}\left[1 - \exp\left(\frac{-F_P U_L}{\dot{m}_c C_p}\right)\right] \qquad (2.1)$$

where

$\dot{m}_c$ = fluid flow rate in collector per unit area,
C_p = specific heat of fluid,
U_L = Collector overall heat loss coefficient,
F_P = Collector plate efficiency factor, and
η_0 = optical efficiency of collector.

* Find out the process load temperature T_1, the process load return temperature $T_{1,r}$, and load mass flow rate $\dot{m}_{load}$ for hot water heating systems and process steam temperature T_S and feed water temperatures T_f for steam systems.
* Calculate the system dependent heat exchange factor, F_S as follows:

For hot water and hot air systems

$$F_S = F_X = \left[1 - \frac{F_R U_L A_C}{\dot{M}_c C_p}\left(\frac{1}{\varepsilon} - 1\right)\right]^{-1} \qquad (2.2)$$

where
A_c = collector area,
$\dot{M}_c$ = mass flow rate in the collector, and
ε = heat exchanger effectiveness[89].
For steam flash systems

$$F_S = F_F = \left[1 - \frac{F_R U_L / \dot{m}_c}{\Delta h_{fg} + C_p (T_S - T_f)} (T_S - T_f)\right]^{-1} \qquad (2.3)$$

where Δh_{fg} is the heat of vaporization of steam at its saturation temperature.
For unfired boiler systems

$$F_S = F_B = \left[1 + \frac{F_R U_L}{\dot{m}_c C_p \left[e^{U_b A_b / \dot{M}_c C_p} - 1\right]}\right]^{-1} \qquad (2.4)$$

where $U_b A_b$ is the product of unfired-boiler heat transfer coefficient and boiler surface area.

Step2: Apply the correction factors.

* Correct the collector heat loss coefficient U_L and collector optical efficiency η_0 due to steady state pipe losses as follows:
  Calculate $U_o A_o$ and $U_i A_i$ which are the overall heat loss coefficient of piping and other components located on outlet (hot) side of field and on inlet (cold) side of field respectively.
  Now the modified values η'_0 and U_L' are as follows:

$$\frac{F_R \eta'_0}{F_R \eta_0} = e^{-U_o A_o / \dot{M}_c C_p} \qquad (2.5)$$

and

$$\frac{F_R U_L'}{F_R U_L} = e^{-U_o A_o / \dot{M}_c C_p} \left[e^{-U_i A_i / \dot{M}_c C_p} + \right.$$

$$\left. + \frac{\dot{M}_c C_p}{A_C F_R U_L} \left(e^{-U_o A_o / \dot{M}_c C_p} - e^{-U_i A_i / \dot{M}_c C_p} \right) \right] \quad (2.6)$$

* Calculate the longterm average optical efficiency $\overline{F_R \eta 0}$, by multiplying the normal optical efficiency, $F_R \eta 0$, with the incident-angle modifier annual correction, $\overline{K\zeta\alpha}$, and dirt and dust optical loss modifier.
The incident angle modifier for a flat-plate collector is given as:

$$K\zeta\alpha = 1 - b_o \left[(\cos\Theta)^{-1} - 1 \right] \quad (2.7)$$

where
b_o = a constant dependent on the optical properties of collector
Θ = angle of incidence of direct solar radiation.
For the evacuated tube collector, the incidence angle modifier annual correction, $\overline{K}\zeta\alpha$ is given as:

$$\overline{K}\zeta\alpha = 0.24K_{7.5} + 0.23K_{22.5} + 0.22K_{37.5} + 0.20K_{52.5} + 0.15K_{67.5} \quad (2.8)$$

where K is the value of the incident angle modifier for the collector at an angle shown by subscript.

For a East-West oriented parabolic trough, the incident angle modifier annual correction $\overline{K}\zeta\alpha$ is given as:

$$\overline{K}\zeta\alpha = 0.33K_{7.5} + 0.30K_{22.5} + 0.22K_{37.5} + 0.14K_{52.5} + 0.03K_{67.5} \quad (2.9)$$

Here also the values of K's are obtained from the experimental curve of $\overline{K}\zeta\alpha$ for angles shown by subscript.

The annual corrections for dirt and dust is difficult to find out because it is highly sight specific and is based on material coupon degradation [90].

Step 3: Calculate annual average collection rate, $\bar{\dot{q}}_c$

* For hot air and hot water systems with no storage calculate the maximum collector area $A_{c,max}$ based on the process load energy use rate $\dot{Q}_{load}$, as follows:

$$A_{c,max} = \dot{Q}_{load} \Big/ \left[F_x F_R (\eta_0 I_{max} - U_L (T_{in} - T_{a,max})\right] \quad (2.10)$$

$$\dot{Q}_{load} = \dot{m}_{load} C_p (T_l - T_{l,r}) \quad (2.11)$$

I_{max} = peak irradiance available to collector and can be taken as 1000 w/m^2 for parabolic troughs and 1100 w/m^2 for flat-plate and evacuated tube collectors.

$T_{a,max}$ = highest expected ambient temperature.

For steam systems, the maximum collector area $A_{c,max}$ is given as:

$$A_{c,max} = \dot{Q}_{load} / \left[F_B F_R (\eta_0 I_{max} - U_L (T_S - T_f)\right] \quad (2.12)$$

$$\dot{Q}_{load} = \dot{m}_{load}\left[h_{fg} + C_p (T_s - T_f)\right] \quad (2.13)$$

$A_{c,max}$ is the upper limit of the collector area, but less area can be used depending on the land availability, cost considerations etc.

* Depending on the $A_{c,max}$ and collector mass flow rate $\dot{m}_c$, calculate F_R.

* Now calculate the ratio $F_R U_L (T_{in} - \bar{T}_a)/(F_R \bar{\eta}_0 \bar{I})$ for a particular collector and by locating its value on the x-axis of fig.2.21[48], find out $\bar{q}_c/(F_S F_R \bar{\eta}_0 \bar{I})$ on y-axis for the required latitude. Instead following equations may be used for its determination:

For flat-plate or evaculated tube collectors:

$$\frac{\bar{q}_c}{F_S F_R \bar{\eta}_0 \bar{I}} = 0.8813 - 1.095X + 0.3905X^2 + 0.003655L + 0.006785LX - 0.004602LX^2 \quad (2.14)$$

For East-West oriented parabolic trough collectors:

$$\frac{\bar{q}_c}{F_S F_R \eta_0 (I_b + 50)} = 0.6688 - 0.6745X + 0.3166X^2 \quad (2.15)$$

and for North-South oriented parabolic trough collectors:

$$\frac{\bar{\dot{q}}_c}{F_S F_R \eta_0 (I_b + 50)} = 0.8810 - 0.8117X + 0.3130X^2 - 0.003919L + 0.003864LX - 0.001484LX^2 \qquad (2.16)$$

where

L = latitude of place in degrees,

X = $F_R U_L (T_{in} - \bar{T}_a)/(F_R \bar{\eta}_0 \bar{I})$

T_{in} = $T_{l,r}$ (load return temperature) for hot air or hot water systems.

= T_S for both infired - boiler and flash steam systems.

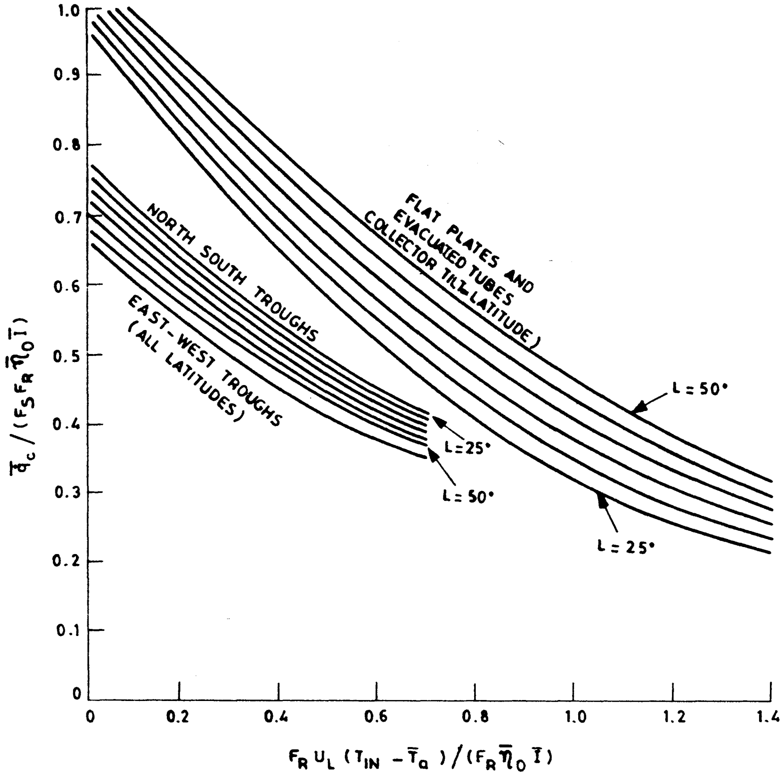


FIG.2.21 ANNUAL AVERAGE ENERGY COLLECTION RATE FOR SOLAR COLLECTORS (From Kutscher et al[48])

* Multiply $\bar{\dot{q}}_c/(F_S F_R \bar{\eta}_0 \bar{I})$ by F_S, $F_R \bar{\eta}_0$ and $\bar{I}$ to get $\bar{\dot{q}}_c$.

Step 4: Calculate the annual energy collection.

* Calculate the annual shading loss factor and annual end loss factor for parabolic troughs which depends on the collector field size, position, land availability and

design of collector and can be seen from graphs given in reference 48. Correct $\bar{\dot{q}}_c$ for shading losses and end losses by multiplying it with their correction factors.

* Calculate the annual average energy collection rate of the collector field $\bar{\dot{Q}}_c$ by multiplying $\bar{\dot{q}}_c$ with the collector area A_c.
* Now the annual energy collection (watt-hours) is obtained by multiplying $\bar{\dot{Q}}_c$ by 4380 (the number of daylight hours in a year).

Step 5: Calculate collector and storage annual night losses.

* The annual overnight losses Q_o from a heating system can be calculaed from the following expression:

$$Q_o = \frac{\Sigma\,(MC_p)_{pipe}(T_{1,r}-\bar{T}_{a,n})N_{oper}}{2} + \Sigma\,(MC_p)_{coll}(T_{1,r}-\bar{T}_{a,n})N_{oper} + U_{Stor}A_{Stor}\,(T_{Stor}-\bar{T}_{a,n})\,nN_d \qquad (2.17)$$

where

$(M\ C_p)_{coll}$ = thermal capacitance of collector including fluid in it,
$(M\ C_p)_{pipe}$ = thermal capacitance of piping etc.
N_{oper} = Number of days of operation per year.
n = Number of nonoperational hours per day when the storage tank losses heat.
U_{Stor} = storage tank heat loss coefficient.
T_{Stor} = storage temperature.
A_{Stor} = surface area of storage tank.
N_d = Number of days per year when the storage tank losses heat.

Step 6: Calculate solar system use factor F_{use}.

The solar system use factor F_{use} can be computed from the following equation:

$$F_{use} = 1 - \frac{H_S - H_p}{4380} \qquad (2.18)$$

where

H_S = annual number of daylight hours when the solar system is non-operational.
H_P = annual number of daylight hours when the industrial process will be non operational.
4380 = annual number of daytime hours.

Step 7: Calculate annual energy delivery and solar fraction.

* Estimate the annual energy delivery Q_d as follows:

$$Q_d = (\dot{\bar{Q}}_c - Q_o)\ F_{use} \tag{2.19}$$

* The solar fraction f i.e. fraction of the process load delivered by solar energy (neglecting electrical parasitics) will be the ratio of the annual energy delivered Q_d and the annual process load Q_{load}.

$$\text{Solar fraction } f = \frac{Q_d}{Q_{load}} \tag{2.20}$$

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CHAPTER - 3

SOLAR FURNACES

3.1 INTRODUCTION

Solar Furnace is an optical system in which solar radiations are concentrated in to a small area (generally a cavity) where very high temperatures are obtained. Solar furnaces may also be called as solar energy concentrators where area may vary from as small as 1 m^2 paraboloid to several thousands of m^2 of reflectors known as heliostats. Solar furnaces are ideal tools to study the chemical, optical, electrical, and thermodynamic properties of materials at high temperatures such as phase studies, vaporization studies, melting behaviours, purification and stabilization of ceramics and refractory materials, crystal growth, specific heat, thermal conductance, etc. Recently many alternative high temperature devices (beyond 2000 $^{\circ}$C) like induction and arc furnaces, electron beam bombardment apparatus, plasma torch, etc. are used with and without electromagnetic fields in varied atmospheres for the treatment and studies on different materials. However, contamination is inevitable in practically all these heating devices. Thus the main advantages of solar furnaces are: comparatively simpler device, high heat flux (corresponding black body temperature of 3500 $^{\circ}$C are obtainable), heating without contamination and container, controlled atmosphere, continuous observations, and observations in the absence of any electromagnetic fields. Large solar furnaces with the objective to study the material properties at elevated temperatures are installed in countries like France, USA, USSR and Japan. There is also considerable amount of proprietary work going on in many countries which will be listed in the next section while dealing with the history of solar furnaces.

Although more serious efforts on the use of solar furnaces in industries were started in 1935, and inspite of many advantages of solar furnaces, these are not becoming popular in industries. Some of the reasons are as follows:

* The initial capital cost of a solar furnace is quite high. Although its simplicity, advantage of getting very high temperatures in any desired place low maintenance cost, no fuel costs etc. may overcome its drawback of high capital cost.

* Solar furnace is suitable only for intermittent operations due to intermittent nature of sunshine.
* In a solar furnace, very high temperatures are obtained on a very small area (few millimeter area) and the temperature gradient in the focal spot is also quite high.

The use of solar furnace is likely to become more lucrative in materials industries in the near future due to high conventional fuel costs and advanced level research in the design and studies on solar furnaces. Several review articles[1-9] are recently published on the principles, designs, comparative performance data, and applications on solar furnaces.

3.2 HISTORY OF SOLAR FURNACES

One of the oldest use of sun by men is the production of high temperature for various purposes. It is said that Archimedes in 212 B.C. set fire to the Roman fleet by concentrating solar radiations onto ships at the distance of a bowshot using large numbers of small plane mirrors. Similarly during the siege of Constantinople in 6th century A.D., Proclus presumably set fire to the fleet of Vitellius with almost a similar arrangement. Averoni and Targioni of Florence in 1695 decomposed a diamond[3] using a single large glass lens. In the years 1702 - 1709, Techirnhouss in Germany, Parker in England, and Geoffroy and Homberg in France used a glass lens of 0.84 m diameter for melting many metals and materials. With the successful experiments of De Buffon[10] and his colleagues in 1747 using 10 mirrors, and burning wood at a distance of 61 m, melting lead at 39.6 m, and silver at 18.3 m, the solar furnace as a high temperature tool became a reality. Later during the seventeenth and eighteenth centuries various kinds and sizes of lenses and reflectors were used, but the practical working furnace was first built by Lavoisier[11] in 1772 who used a lens of about 1.52 m diameter and reached to a temperature of about 1773 °C (melting point of platinum) and demonstrated that diamond is a form of carbon. Solar furnace was almost a forgotten thing for over century and interest reborn in the beginning of this century with the work of Stock and Heyneman in 1909 who made a two-lens adjustable mounting type of furnace.

The use of large glass lenses for the concentration of solar radiations is not preferred because of its limitations to attain high temperature due to large losses in the lens material and by spherical aberrations etc. The first reflector type of solar furnace using a glass paraboloidal reflector of about 1.83 m in aperture and of 0.61 m in focal length which has given a temperature of the order of 3000 °C was made by a German scientist Straubel[12] in 1921 at the Zeiss Company at Jena in Germany. During the year 1932 G.E. Hale and his colleagues built a furnace[13] using a large glass lens with the specific objective of obtaining high temperatures for spectroscopic studies at the California Institute of Technology, Pasadena, California (USA). Later Prof. Trombe[14] of CNRS France in 1946 built a small experimental solar furnace at MontLouis in the Pyrenees in the south France and Guillemonar and Betier[15] at Bouzarcah near Algiers.

The design and development work on the World's largest solar furnace was started under the leadership of Prof. Trombe[16] in France in 1948 and the same was completed in 1952 located in Montlouis in the French Pyrenees mountains. This furnace had provided 50 KW of thermal energy and used a single large heliostat (13 m wide and 10.5 m tall consisting of 540 flat mirrors each of 50 cm x 50 cm) which conveys the sunlight on to a paraboloidal reflector (10 m in aperture and 6 m in focal length and consisting of 3500 mirrors each of 16 cm x 16 cm). The total parabolic mirror area was about 90 m^2. This development has resulted in several such solar furnaces built in many parts of the world.

After the successful completion of a solar furnace by Trombe in France in 1947 using a military searchlight mirror, a number of such furnaces were constructed as reviewed by Conn[17] in 1954 and by Cohen and Hiester[18] in 1957. Generally copper on glass paraboloidal reflectors of various aperture areas coated with rhodium or aluminium were used which have given temperatures in excess of 3000°C suitable for investigation of some refractory materials ideal for use in jet, rocket, and for other high temperature applications.

In 1958, 'The US Army Quartermaster Research and Development Laboratories[19], built a solar furnace at Natick, Massachussetts which was similar in construction and size to that of the French Montlouis furnace except it uses a spherical reflector concentrator instead of a paraboloidal reflector. This furnace had given considerably less thermal power compared to the Montlouis furnace mainly due to the radiation loss by an attenuator of the venetian blind tyre placed in front of the concentrator. In this furnace, 180 spherical mirrors, 10.9 m in focal length are arranged on a spherical surface with a center at the focus, in order to satisfy the sign condition hence eliminate the spherical aberration. It is an excellent characteristic of this design. In 1974, this furnace was moved to Nuclear Weapon Effects Laboratory, White Sands Missile Range, New Mexico, where it became operational in 1974.

In 1964, Sakurai et al[20] at the Tohoku University, Japan built a solar furnace with a single largest heliostat (238 mirrors each of 100x90 cms) with an area of 234.2 m^2

and a paraboloidal mirror concentrator, 10 m in aperture and 3.2 m focal length, consists of a mosaic composed of 181 segments. The concentration ratio at the focal point was of about 36000 with an optical efficiency of about 70 percent. With this system 4 gm of tungston specimen melted (3400 °C) in 30 seconds.

The world's largest solar furnace[21] (1000 Kw) was built by the National Center for Scientific Research at Odeillo, Font-Romeu near Montlouis in the Pyrenees in France. A paraboloidal concentrator 40 m X 54m is mounted on the north side of building which consists of 9500 back-silvered plane mirrors, 45 cm X 45 cm in dimension, curved mechanically, giving the effective area of 1900 m^2. The solar radiation is conveyed into a concentrator by 63 heliostats each of 7.5 m wide and 6.0m high having 180 single flat mirrors (50 cm X 50 cm). The total area of a heliostats is 2835 m^2 which is about half the playing area of the football field. The flux concentration obtained with this furnace is equivalent to the black-body radiation around 3800 °C.

At Odeillo, there is another furnace constructed by the French army. A concentrator is a spherical mirror 10 m X 10 m with a focal length of 10.75 m consisting of 384 curved mirrors, 50 cm X 50 cm in dimension, with a total area of 96 m^2. A heliostat 17.4 m wide and 13.2 m high has 638 single flat mirrors (60 cm X 60 cm) with the total area of 229.7 m2. This furnace seems to be used to simulate the thermal radiation environment produced by a nuclear explosion and others.

Three solar furnaces[22], one with direct type built in 1955, second heliostat type of which optical axis was horizontal was constructed in 1963 and the third solar furnace also of heliostat type of which optical axis was vertical was built in 1977 under the guidance of Prof. Noguchi at the Solar Energy Laboratory, Government Industrial Research Institute, Nagoya, Japan. The paraboloidal mirror consists of back aluminium glass having diameter of 1.5 m, focal length of 0.65 m, aperture ratio of 2.4 giving a sun's image of 6.0 mm. The heliostat consists of 25 aluminium evaporated front mirrors (50 cm X 50 cm). The highest temperature attained is 3500 °C.

The characteristics[6] of the four major, single heliostat solar furnace are shown in Table 3.1. Looking to the importance of solar and image furnaces for high temperature applications Laszlo[23] conducted a survey and the data for solar furnaces is presented in tabular form in table 3.2. The solar furnaces which are built later are also included by the author of this book.

Table 3.1 Properties and performance of a few large solar furnaces (From Ref.6)

Parameter 1	CNRS Montlouis France 2	Tohoku Univ. Sendai, Japan 3	French Army Odeillo Font-Romeu France 4	US Army White Sands Missile range New Mexico 5
Construction date	1952	1962	1972	1974
HELIOSTAT				
Heliostat Size(m^2)	10.5x13	14x15.5	13.2x17.5	11x12.2
Number of mirrors	540	238	638	356
Mirror size (cm)	50x50	90x100	50x50	62x62
REFLECTOR				
Configuration	Parabolic	Parabolic	Spherical	Spherical
Reflector size (m)	9x11	10(dia)	10x10	8.5x8.5
Focal lenght (m)	6	3.2	10.7	10.9
Number of Mirrors	3500	181	384	180
Mirror size (cm)	16x16	80x75	50x50	62x62
Total mirror area (m^2)	89.6	78.5	96	72.6

Table 3.1 cont.

1	2	3	4	5
THERMAL PERFORMANCE				
(Insolation 900-950 w/m^2)				
Total Thermal Power (KW)	45	35	42.5	32
Thermal efficiency (%)	55	50	48	50
Max. heat flux (w/cm^2)	1200	-	580	400

Table 3.2 Survey of Solar furnaces (Hitherto) constructed (A part taken from Ref.23)

Country 1	Investigator and address 2	Optical system 3	Experiments 4
Algeria	M.Perrot[15], Institut L'Energie Solaire De L'Universite D'Alger, Algeria.	8.40 m diameter paraboloidal mirror, 1.5 m diameter silvered glass mirror, 0.5 m diameter metal mirror.	Photochemical research, Thermoinic generator, preparation of boron carbide.
Australia	J.H.Weymouth, Chemical Research Laboratories, Melbourne.	1.50 m diameter paraboloidal mirror with heliostat	Investigations of high lime and of system as $CaO - TiO_2$, $CaO - CeO_2$ $CaO - ThO_2$.
	C.J.Milner[24,25], J.E Giutronich K.G. O'Brien The University of New South Wales, Kensington N.S.W.	Vertical type 3.7 m diameter paraboloidal mirror with heliostat and another small furnace with 0.9 m diameter paraboloidal mirror.	High temperature measurements with refractory and non metals, crystallization behaveriour and emissivity measurements.
Brasil	S.N.Vannucci[26], Instituto Technologico de Aeronautica Sao Jose dos Campos, Sao Paula, Brasil.	1.0 diameter paraboloidal mirror without heliostat	Experimental determination of optimal area of heat exchanger for use in a solar furnace.

Table 3.2 cont.

1	2	3	4
Canada	G.C.Drew[27], Defence Chemical, Biological and Radiation Laboratories, Ottawa, Ontario, Canada	Cassegrain type with 1.50 m dia parabolic primary mirror 0.38 m aluminized hyperboloid second mirror	Study of materials used to protect men and equipment from thermal radiation of atomic weapons.
France	F.Trombe[16,21], M. Foex, La Blanchetais, La Phat Vinh, Solar Energy Laboratory Montlouis, France	10 m aperture paraboloidal mirror with 136.5 m^2 heliostat.	Research and Industrial applications.
		10 m diameter spherical mirror with 231 m^2 heliostat.	Research and Industiral applications.
Germany	G.Brauer, University of Freiburg, Freiburg, Germany.	Paraboloidal mirror	Thermal dissociation of metal oxides (TiO_2) measurement of O_2 equilibrium pressure.
India	P.K.Rohatgi[28,29], D. Suresh and S.Seshan, Indian Institute of Science, Banglore, India.	0.35 m spherical mirror without heliostat.	Experiments and study the use of solar furnace in foundaries.

Table 3.2 cont.

1	2	3	4
		3 m diameter paraboloidal mirror without heliostat.	Experiments and study the use of solar furnace in foundaries.
Iraq	A.K.Kaddou[30] and A. Abdul-Latif, Univer- of Baghdad, Iraq.	0.35 m diameter plastic Fresnel lens concentrator	Investigations on the use of solar furnace for joining metals.
Japan	T.Noguchi[22], Government Industrial Research Institute, Nagoya, Japan.	2.0 m diameter paraboloidal mirror without any heliostat showing temperature of 2300 °C	High temperature material properties, Emissivity and phase studies.
		1.5 m diameter paraboloidal mirror with heliostat of 16 mirrors each of 60 x 60 cms.	High temperature material properties; Emissivity and phase studies.
		1.5 m diameter paraboloidal mirror with heliostat of 25 mirrors each of 50 x 50 cms.	High temperature material properties; Emissivity and phase studies.
	T.Sakurai[20], Tahoku University, Sendri Japan.	10 m diameter paraboloidal mirror with 14 m x 15.5 m heliostat.	Temperature measurement and high melting ceramic semiconductor.

Table 3.2 cont.

1	2	3	4
U.S.A	P.Duwez[13], T.E.Tretz, E.Loh & N.K.Hiester, California Institute of Technology, California, USA.	19 primary lenses each each of 0.60 m diameter with a total collecting area of 5.3 m^2 and 19 secondary lenses and 18 oblique mirrors.	Research on high temperature properties of materials.
	W.Conn[7], University of Missouri, Kansas city Missouri, USA.	Several furnaces with 1.5 - 3.0 mm aperture paraboloid reflector. Eighteen search light reflectors or specia-lly prepared aluminium sheets were used.	Development of solar furnaces for different high temperature applications.
	T.S.Laszlo[31,32], P.J. Sheenhan and R.E.Cannon, Research and Advanced Development Division, Aveo Corporation, Wilm-entgton, Mass (USA).	1.5 m diameter parab-oloidal mirror.	Calorimetric and radio-metric flux measurement; measurements of elec-trical resistivity and emittance; study of refractory materials.

Table 3.2 cont.

1	2	3	4
	J.W.Mc Donald, Air Research Manufacturing of Arizona, Phoenix, Arizona.	1.5 m diameter paraboloidal mirror.	Evaluation of solar turboelectric power plant.
	P.D.Jose, [33,34], Air Force Office of Scientific Research.	1.5 m diameter paraboloidal mirror with heliostat.	Investigation of Experimental Techniques
	F.A.Blake, General Electric Company, Solar Test facility Phoenix, Arizona	1.5 m diameter paraboloidal mirror.	Evaluation of Solar Power and test system components and performance endurance testing of solar thermonic generator.
	G.Benveniste[35], General Electric Co., Nela Park, Cleveland.	1.5 m diameter paraboloidal mirror.	High temperature research.
	W.M.Tuddenhem[36], Kennecott copper Corp., Western Minning Div., Salt lake City, Utah.	1.5 m diameter paraboloidal mirror.	Synthesis of Olivine type material; zone refining and vacuum fusion studies of copper

Table 3.2 cont.

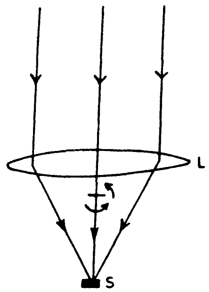
1	2	3	4
	R.Gardon[37], Massachusstes Institute of Technology, Massachussetts, USA	400 flat mirrors used which concentrate sunlight on a single target.	Design studies and thermal radiation studies.
	J.A.Duffie,University of Winconsin, Madison, Winconsin	1.5 m diameter paraboloidal mirror.	Investigation of evaporation and sublimation processes.
	R.J.Marcus[38] and H.C. Wohlers, Stanford Research Institute, Menlo Park, California.	0.6 m diameter paraboloidal mirror.	Studies on Photolysis of Nitrosyl Chloride.
	E.A.Farber[39], University of Florida, Florida	1.5 m diamater paraboloidal mirror.	Studies on growing of Crystals.
	J.E.Giutronich[19,40-42], S.Cotton, M.Daivies,etc US Army Natick Lab., Natick, Masachusetts.	8.5 m x 8.5 m size spherical reflector heliostat of 11 m x 12.2 m	To study the effect of thermal radiation on the fabric system.

Table 3.2 cont.

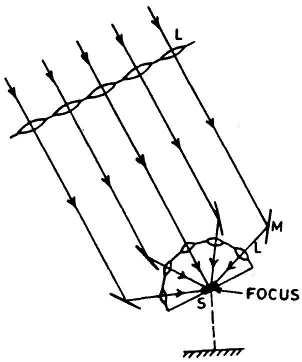
1	2	3	4
U.S.S.R	M.Yu.Borukhov[43-45],Yu. Z.Mavahev, A.Ya.Bashnyak, A.A.Annaev, and V.A.Baum, High Temp.Lab.Electronics Inst.Academy of Sciences, Uzbek SSR.	2.0 m diameter paraboloidal mirror with heliostat.	To investigate the Physics, radiation, and other properties of high melting materials at high temperatures.
Yogoslavia	J.Muster[46], Metallurgical Institute, Ljubljana, Yogoslavia.	1.5 m diameter paraboloidal mirror	Synthesis of minerals.

3.3 TYPE OF SOLAR FURNACES

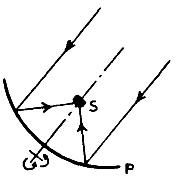
Depending upon the requirements of a solar furnace, it can be designed in several ways. A few geometrical arrangements are shown in fig.3.1. Generally there are two types of solar furnaces: direct type and heliostat type. The direct type solar furnace can also be of two types: the one using a



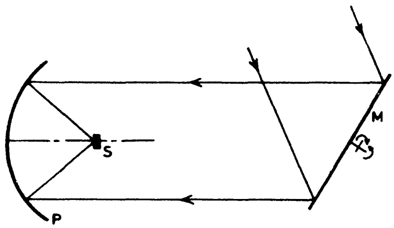
(a) SINGLE LENS DIRECT TYPE.



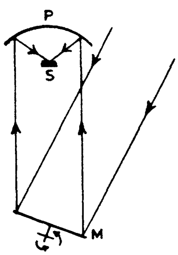
(b) MULTIPLE LENS TYPE.



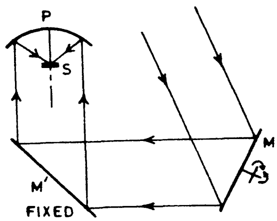
(c) SINGLE PARABOLOID DIRECT TYPE.



(d) HELIOSTAT TYPE (OPTICAL AXIS HORIZONTAL)



(e) HELIOSTAT TYPE (OPTICAL AXIS VERTICAL)



(f) HELIOSTAT TYPE (OPTICAL AXIS VERTICAL)

FIG.3.1 SCHEMATICS OF SOME SOLAR FURNACES (L-LENS; M-MIRROR; P-PARABOLOIDAL MIRROR; S-SAMPLE) single lens or multiple lens system as shown in fig.3.1 (a and b) and the second using a paraboloid reflector (fig.3.1(c)). Glass lens type solar furnaces are now not preferred because the lenses are generally very heavy, very expensive, and unmanegable. Paraboloid or spherical reflectors can easily be made in very large sizes using glass (generally mosaic construction), or spun metal or plastic materials. In the direct type solar furnace, the heat flux density obtained is highest but in this case both the lens or reflector and the target moves with the sun, making the system impractical and most inconvenient. Instead, a single heliostat or multiple heliostat type furnace, heliostat consisting of small flat-mirrors, which follows the sun and reflect the solar radiations onto the stationary spherical or paraboloidal reflector is preferred. In this manner the target (object) remains stationary making the observations easy. The heliostat type furnace apart from several advantages has some disadvantages also such as: it is an expensive system, loss of radiation due to multiple reflection, and extra cleaning of the mirrors. Heliostat type furnaces are also of three types as shown in fig.3.1 (d,e,& f). The heliostat type solar furnace with optical axis horizontal as shown in fig.3.1(d) is most convenient and practical and several large furnaces on this principle are made. This appears to be the most economical configuration for large solar furnaces. There are two possible configurations of solar furnace where the optical axis is vertical as shown in fig.3.1 (e & f). In this vertical axis arrangement the unmelted portion of specimen forms a crucible to hold the melted portion and is suitable for fusion studies. In the vertical axis solar furnace as shown in fig.3.1 (e), a large distance between the heliostat and concentrator is required in case of a large size solar furnace and for low latitude stations. Therefore, an alternative for this system is shown in fig.3.1 (f). But such a system suffers from additional reflection losses.

3.4 THEORETICAL CONSIDERATIONS IN A SOLAR FURNACE

As discussed earlier, in all solar furnaces either direct type or heliostat type, a small or large paraboloidal reflector is used. It is possible with a paraboloidal reflector to get a heat flux of 857 w/cm^2 which is equivalent to a black body temperature of 3500 $^\circ$C. To evaluate the heat flux in the focal plane of a paraboloidal reflector and temperature at the target, two parameters i.e. the concentration ratio and the concentration efficiency are to be determined. Hiester et al[47], Farber and Davis[48], Cobble[49], Fukuo and Mii[50], and Kamada[51] have done considerable theoretical analysis of a paraboloidal mirror type solar furnace by considering various configurations of the target and assuming both a uniform and nonuniform brightness distribution of the solar disc.

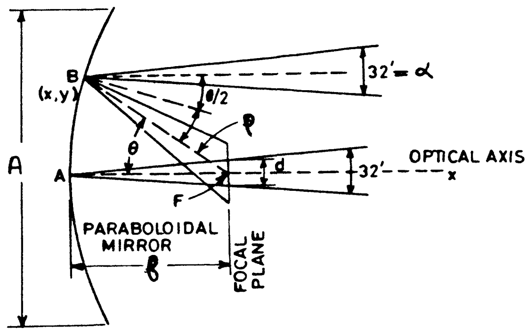


FIG.3.2 PARABOLOIDAL MIRROR SHOWING THE INCIDENT CONE OF SUN RAYS REFLECTED TO FOCAL PLANE.

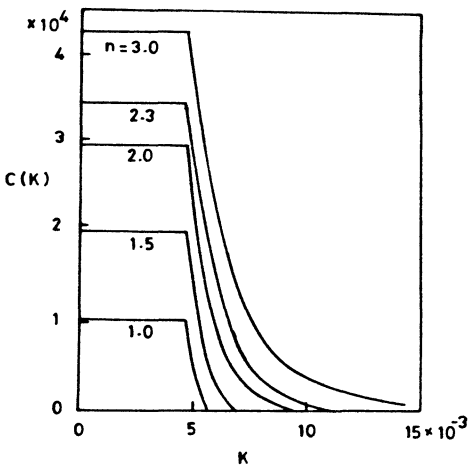


FIG.3.3 CALCULATED CONCETRATION RATIO FOR A FLAT TARGET ASSUMING UNIFORM BRIGHTNESS IN SOLAR DISC(From Fukuo and Mii[50])

The rays coming from the sun are not parallel but subtends a solid angle of 32 minute and therefore the reflected beam is also of a cone with an angle 32 minute as shown in fig.3.2. It is therefore clear that only the central beam forms a circular image of the sun on the focal plane of the paraboloid and the other reflected cones form ellipses. The size or the diameter of sun image formed due to the reflection of the cone from vertex A is α f, where f is the focal length of the paraboloid. The size of the major and minor axis of the ellipse formed due to reflected cone from a point B (x,y) is $\alpha\rho/\cos\Theta$ and $\alpha\rho$ respectively, where Θ is the azimuthal angle, ρ is the distance of B from F and given as, $\rho = f + x$. It is therefore clear that the circular image of the sun formed due to vertex is completely covered by the ellipses formed due to various other points. The intensity also remains uniform in the sun's images and decreases in the outside. The concentration ratio of a concentrator, C, is defined as the ratio of concentrated radiation to the incident radiation. From simple analysis, the expression for maximum concentration ratio, Cm, can be written[52] as:

$$C_m = \frac{4r}{\alpha^2}\left[1 - \frac{(16 - n^2)}{(16 + n^2)}\right] \qquad (3.1)$$

where r is the reflectivity of the mirror material, and n is the aperture ratio defined as the ratio of aperture D and the focal length f, i.e. $n = D/f$. By assuming a uniform brightness of the solar disc, and for a flat target, the concentration ratios for different values of n as a function of distance k from the focal point by taking $r = 1$ are calculated and the same are shown[50] in fig.3.3. From this figure it is seen that to attain a higher concentration ratio and higher temperature, the aperture ratio must be increased and not the local length. For high temperature investigations, the highest attainable temperature is required usually, hence a large aperture ratio upto 4 is desirable. In this case, however, it must be remembered that radiation loss takes place in the marginal rays by that a considerable amount of reflected cones spills outside the target. For solar energy utilization, on the other hand, it is desirable that the whole radiation incident the aperture of given D is concentrated into the possible smallest area. This condition is satisfied by minimizing the diameter of the sun image built by reflected cones from the rim, $\alpha\rho_m/\cos\Theta_m = \alpha D/(2\sin\Theta_m \cos\Theta_m)$, which leads to that $\Theta_m = 45^o$ or $n = 4(\sqrt{2}-1) \approx 1.66$. Thus the figure of paraboloid for a given D is to be changed by what purpose the furnace is used for.

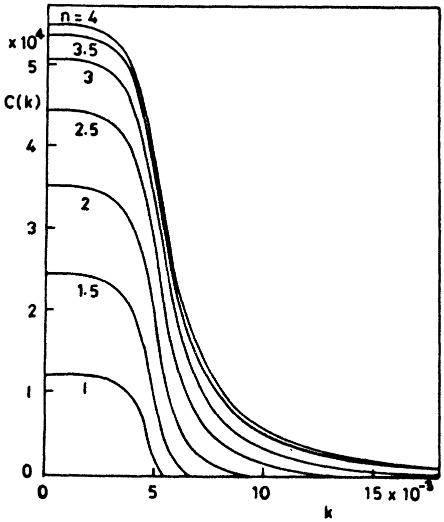


FIG.3.4 CALCULATED CONCETRATION RATIO FOR A FLAT TARGET ASSUMING BRIGHTNESS DISTRIBUTION IN SOLAR DISC (From Kamada[51]).

Considering the brightness distribution in the solar disc Kamada[51] calculated the concentration ratio for various values of K and n for a flat-target, and the results are shown in fig.3.4. It is seen from this figure that about 18 percent higher maximum concentration ratio can be achieved by assuming a brightness distribution compared to obtained from uniform brightness of the solar disc.

The radiant energy concentrated on a target surface is absorbed to give rise to the increase of temperature, the amount of which is governed by the optical property of the target. On the other hand, the absorbed energy is lost by thermal radiation and conduction, which will differ case by case. Therefore, it is very difficult to give the exact temperature by theoretical treatment. However, by assuming (1) the absorbed energy is lost only by thermal radiation and (2) the emission and absorption follow the Lambert's cosine law, it is possible to obtain an attainable temperature by theoretical calculation. When the uniform brightness of the solar disk is assumed, the temperature for a flat target is uniform and highest within the sun image d in diameter, and decreases towards outside. The maximum attainable temperature Tm is expressed as:

$$T_m^4 - T_o^4 = \frac{4 r I_s \alpha_n}{\sigma \alpha^2 \varepsilon_n}\left[1 - \left\{\frac{16 - n^2}{16 + n^2}\right\}^3\right] \qquad (3.2)$$

where T_o is the ambient temperature, σ is the Stefan-Boltzman constant, I_s is the intensity of incident solar radiation, and ε_n and α_n are the normal emissivity and absorptivity. In most cases, it may be allowed to put $T_o = 0$ and $\varepsilon_n = \alpha_n$. Here also it is seen that the maximum temperature attained is a function of aperture ratio and not of D and f alone. The distributions of attainable temperature for a flat target are also calculated by Kamada[51] considering the brightness distribution in the solar disc. for various values of n, and the results are shown[51] in fig.3.5. In the calculation it is assumed that r = 1 and I=0.91 KW/m2. The maximum values of these curves are about 4.3 percent larger than T_m given by the above equation. It will be remarked here that the temperatures really attained are considerably lower due to reflection and other losses.

The concentration ratio and the attainable temperatures obtained with a paraboloidal reflector with a spherical target and cyclinderical target are also calculated by Kamada[51] considering the brightness distribution in the

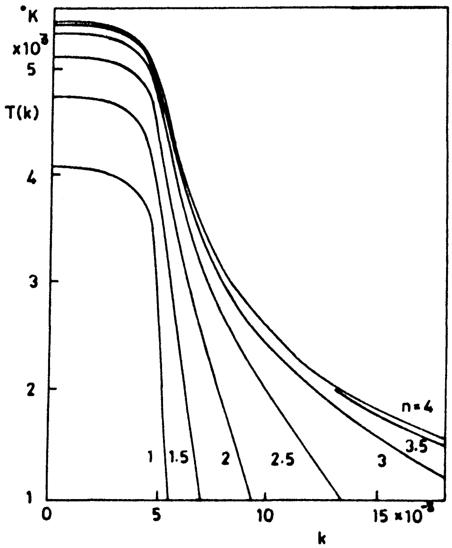


FIG.3.5 CALCULATED ATTAINABLE TEMPERATURE FOR A FLAT-TARGET ASSUMING BRIGHTNESS DISTRIBUTION IN SOLAR DISC(From Kamada[51])

solar disc. The concentration ratio and attainable temperature in case of spherical target are shown[51] in fig.3.6. and 3.7 respectively, where b is the angle between the normal to the surface element and the optical axis. In case of a very small cylindrical target with two different radii placed along the optical axis, the concentration ratio and attainable temperature are shown[51] in fig.3.8 and 3.9 respectively. Here K' is the distance from the focal point along the optical axis in unit of f.

As seen earlier the concentration ratio and the attainable temperature is a strong function of n which is the ratio of aperture D and the focal length f. If the aperture ratio, n, is to be made larger, then for a fixed value of aperture D, the paraboloidal mirror is to be made deeper. For aperature ratio of four, the circumference of the paraboloid mirror coincides with the focal plane. Generally for a flat target, a paraboloid mirror with n from 2 to 3 is preferred from operational point of view. For a spherical or cylindrical targets, n can be greater than 4.

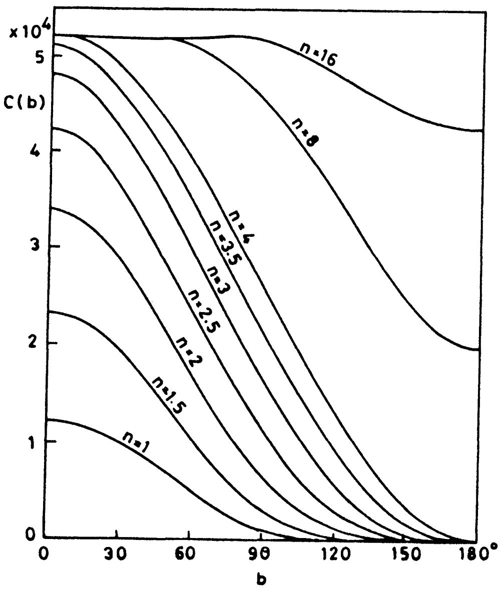


FIG.3.6 CALCULATED CONCENTRATION RATIO FOR A SPHERICAL TARGET WITH RADII EQUAL TO f $\alpha/2$ ASSUMING BRIGHTNESS DISTRIBUTION IN SOLAR DISC (From Kamada[51])

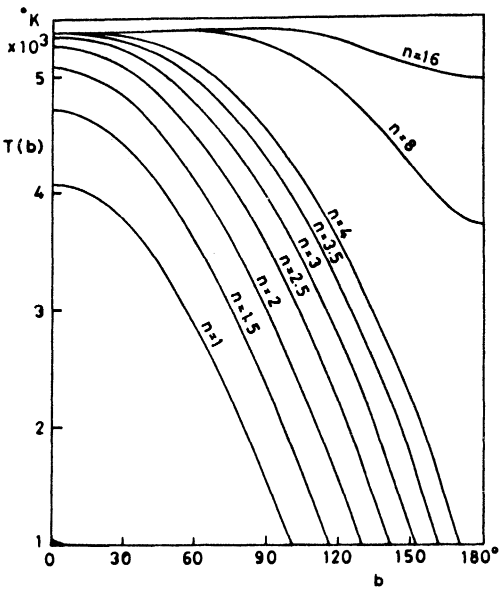


FIG.3.7 CALCULATED ATTAINABLE TEMPERATURE FOR A SPHERICAL TARGET WITH RADII EQUAL TO f $\alpha/2$ ASSUMING BRIGHTNESS DISTRIBUTION IN SOLAR DISC.(From Kamada[51])

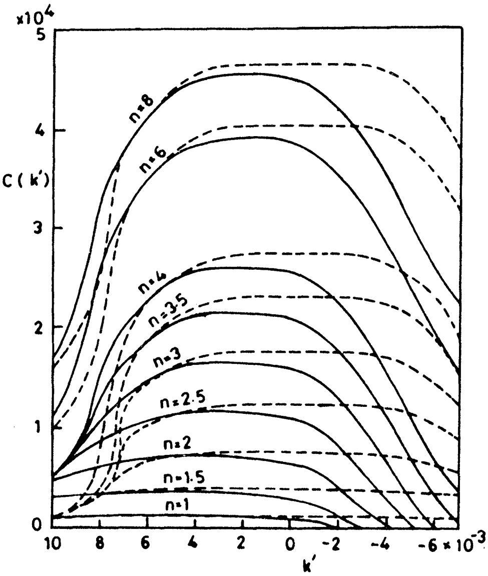


FIG.3.8 CALCULATED CONCENTRATION RATIO WITH CYLINDRICAL TARGET; DOTTED CURVES ARE FOR SMALL RADII AND SOLID CURVES FOR RADII EQUAL TO f $\alpha/2$. THE MIRROR IS ON THE POSITIVE SIDE OF K' (From Kamada[51]).

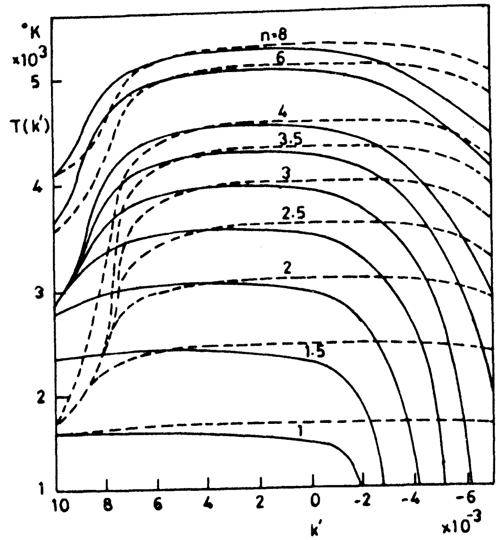


FIG.3.9 CALCULATED ATTAINABLE TEMPERATURE WITH CYLINDRICAL TARGET; DOTTED CURVES ARE FOR SMALL RADII AND SOLID CURVES FOR RADII EQUAL TO $f\alpha/2$. THE MIRROR ON THE POSITIVE SIDE OF K'(From Kamada[51])

In the above analysis, it has been assumed that the paraboloid reflector is optically and geometrically perfect. While in real practice, it is impossible to get the theoretically predicted concentration ratio and attainable temperature due to the above two reasons. The optical efficiency of a paraboloid mirror, η_0 defined as the energy absorbed by the target to the energy incident on the reflector's aperture, is given as:

$$\eta_o = \rho_m \alpha_r F_1 \quad \delta(\Psi_1, \Psi_2) \quad F(\Psi_3) \qquad (3.3)$$

where ρ_m is the reflectance of the mirror, α_r the absorptance of target, F_1 the fraction of aperture not shaded by target and suports, $\delta(\Psi_1,\Psi_2)$ the intercept factor depending on mirror slope errors Ψ_1 and solar beam spread Ψ_2 and $F(\Psi_3)$ the tracking error depending on the angle Ψ_3 between the direct sun ray and aperture normal. The optical intercept factor $\delta(\Psi_1,\Psi_2)$ is given as:

$$\delta(\Psi_1, \Psi_2) = 1 - \exp\,[\Pi r^2/\sigma_y^2] \tag{3.4}$$

where σ_y^2 is the beam spread variance at the target (receiver) of radius r. For a spherical target it is given as:

$$\sigma y^2 = \frac{2\ A_a (4\ \sigma_{\Psi_1}^2 + \sigma_{\Psi_2}^2\ )\ (2 + \cos\Phi)}{3\,\Phi\ \mathrm{Sin}\,\Phi} \tag{3.5}$$

where A_a is the aperture area and Φ is the rim half angle of paraboloid.

In case of a flat target, the beam spread variance at the target, $\sigma^2{}_y$ is given as:

$$\sigma y^2 = \frac{2\ A_a(4\,\sigma_{\Psi_1}^2 + \sigma_{\Psi_2}^2\ )}{\mathrm{Sin}^2\,\Phi}$$

The tracking error $F(\Psi_3)$ can be included in the intercept factor $\delta\,(\Psi_1, \Psi_2)$ by defining an appropriate $\sigma_{\Psi_4}^2$ and adding it to the slope and solar image variance $\sigma_{\Psi_1}^2$ and $\sigma_{\Psi_2}^2$. Thus for designing and deciding a solar furnace for a particular application following three factors should be considered:

1) The aperture ratio, n, plays a significant role in designing a solar furnace. Once this is fixed, then the image diameter, concentration ratio, temperature attained, and the ideal maximum image flux is automatically get fixed.
2) The second important factor relates to the inefficiency in construction and location. Some of these parameters are the imperfectness in the design of the paraboloid, local slope errors in the mirror, tracking inaccuracies, shading coefficients, reflection coefficients, etc. and the atmospheric transmission coefficient
3) The third parameter is regarding the inefficiencies in the target including its absoptivity and emissivity.

Sometimes the target is kept not exactly in the focal plane purposely to obtain decreased flux and hence decreased temperature. The effect is known as defocussing, and its effect is studied in detail by de la Rue et al[54] and Simon[55]. In the defocussing case the flux is not uniformly distributed on the target and it can be calculated only approximately. The relative flux values on a plane parallel to the focal plane for a paraboloid mirror with a rim angle of 60° and ratio of distances between the plane and the focal plane to the diameter of the solar image are plotted [54], and shown in fig.3.10. It is seen from this figure, that as the distance between the target and the focal plane increases, the flux distribution ceases to be uniform and

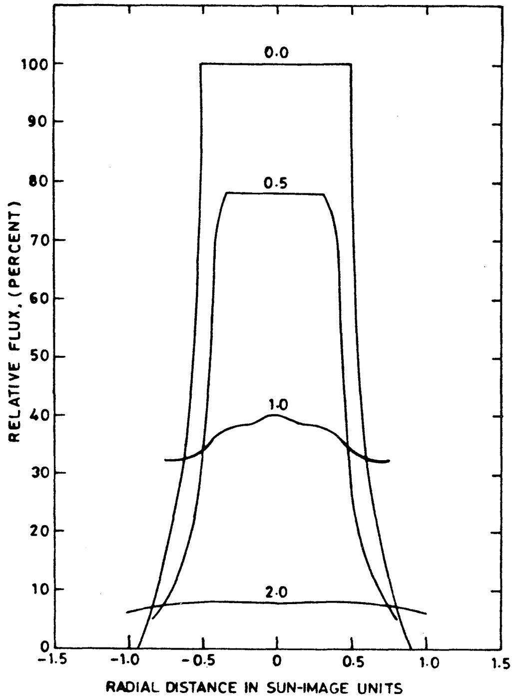


FIG.3.10 EFFECT OF DEFOCUSSING IN CASE OF A PARABOLOIDAL MIRROR (From Rue de la et al[54])

becomes uniform again at large distances. It is also seen that the maximum flux along the axis decreases very rapidly as the distance between the target and the focus increases.

3.5 COMPONENTS OF SOLAR FURNACE

A solar furnace consists of several components and the few ones will be briefly described here:

3.5.1 Concentrator

In both the direct and indirect type of solar furnaces either a paraboloidal or spherical reflector concentrator is employed. Paraboloidal reflector is generally preferred due to large spherical aberreation in a spherical reflector. The paraboloidal reflector have should aperture ratio less than 4, and should have optically smooth surface with accurate geometry and high reflectivity. Metal reflectors, or plastic metallized reflectors or back or front coated glass reflectors can be employed as a reflector material for paraboloidal reflector. Sometimes metal reflectors particularly aluminium reflector is choosen to avoid accident which may take place in case the melted material falls on the reflector material. Commercially available aluminium reflectors, chrome and nickel plated reflectors, showed a 50 percent decrease in reflectivity after a few months of outdoor expoxure. An electrolytically polished aluminium sheet with an anodization can prove to be a better reflecting surface. Rodium coated copper sheets which are polished can also be used as a good metal reflecting surface.

Metallized plastics can also be used as mirror material for paraboloidal reflectors. These are of low cost, easy to fabricate,low density, and ressistance to corrosion. The disadvantages of plastic reflectors are: low strength, very poor weatherability, and dimensional and thermal instability.

Glass is therefore the most preferred reflecting material due to its dimensional stability and an optically smooth surface can easily be obtained with it. Some kind of reflective coating generally aluminium or silver is deposited on to the glass either chemically or in vacuum. Both silver and aluminium coated glass reflectors either on the back side or front side of the glass sheet are employed and both the systems have advantages and disadvantages [56,57]. Vacuum or sputter deposited silver adheres to the glass sheet better than the chemical deposited silver. A glass sheet optically polished with a vacuum deposition of aluminium on front side will show a solar reflectance value more than 90 percent. But this front coated glass is to be provided a protective overcoating generally by a thin film of dielectric material, such as SiO, Al_2O_3, or MgF,to avoid the reflecting surface from environmental corrosion. Although front coated glass gives higher solar reflectance but is not liked because of additional maintenance, and therefore back side coated glass mirrors are used inspite of large radiant losses due to absorption and surface reflection in this case. The most disadvantage of a glass reflector is that it is fragile and of higher density compared to other metal or polymeric materials. Moreover, difficulties are experienced in making a large size, thin, and preciely curved glass reflectors. Recently some firms have started manufacturing [58] extra thin glass, 0.6 - 0.8 mm thick, in 1200 x 1000 mm size with silver (0.8 g/m^2) or copper (0.3 g/m^2) coating with a protective coating of organic paint. These extra-thin glasses made specially as solar mirrors have shown an average solar reflectivity of 0.93 to 0.94.

Instead of one continuous paraboloidial mirror which is difficult to make particularly in large size solar furnaces, small mechanically bent mirrors are used in a paraboloidal reflector. These small glass mirrors generally of 50x50 cm are heated to softening point and pressed against a parabolic or spherical mirror or they may be forced with pressure and then screwed to give a curved surface. By using a large number of mirror segments, the imperfectness in the mirror surface can also be minimized. If the size of the mirror segment is smaller than $\alpha\rho$, then even a plane mirror element can give a correct sun image.

A simple method of making a plastic paraboloid is described by Archibald[59] which is based on the principle that a liquid in a revolving horizontal pan takes the shape of a paraboloid. A systematic and accurate method of fabricating mirror segments of a large paraboloid mirror is described by Sakurai and Shishido[60] in which the window glass sheets were cut into segments, curved on a mold in an electric furnace, annealed, given an accurate shape on a special paraboloidal surface grinder, and then aluminium coating was done by vacuum evaporation. Trombe[61] used a novel method of fabricating the concentrator segments in a 1000 KW solar furnace in which the plane mirror is mechanically bent which is based on the principle that when the four corners of a square plate are fixed and the centre is pulled, one gets almost a spherical surface. Similarily if the four corners of a rectangular plate are fixed and the centre is pulled, the bent surface obtained is toroidal.

3.5.2 Heliostat

The purpose of a heliostat or heliostats in a solar furnace is to direct the solar radiation parallel to the optical axis of the concentrator. The solar furnace may consists of one concentrator and one large heliostat or one concentrator with several heliostats. The size and shape of the heliostat[62,63] depends on the aperture of the concentrator since the heliostat is to convey the solar radiation over the whole aperture of the concentrator, the latitude of the place of installion of solar furnace, the solar declination, and the angular width of the rays reflected from the heliostat. Generally the dimensions of the heliostat is taken as 1.4D x 1.4D, where D is the aperture of the concentrator. The heliostat represents a fresnel mirror, which may consists of several hundreds of facets and hence plane mirrors of small size may work. The concentration can be further improved by using a little concavity in the mirror facets. The quality of mirrors employed in the heliostat should be the same as that of mirrors in concentrator. Ordinary commercially available window glass sheets with low iron content, and less than1/2000 radian in portional displacements, with silver coating on the back side can form segments of a heliostat. Generally a heliostat with a sandwich type structure which consist of a second surface mirror (a transparant protective layer, a layer of reflective material, and a protective layer on the back side) mounted onto a structural support material with an adhesive is used. Several designs of a heliostats are available. Five designs[64] which are tested and used in solar tower systems are given by the following firms of USA:

(i) Boeing Engineering and construction, Seattle, Washington.
(ii) McDonnell Douglas Astronautics Company, Hintington Beach, California.
(iii) Martin Marietta Corporation, Denver, Colorado.
(iv) Northrup Incorporated, Hutchins, Texas.
(v) Westinghouse Electric Corporation, Pittsburgh, Pennsylvania.

The Boeing heliostat has 12 reflector facets, made by laminating fusion glass skins on a cellular glass core, with a total reflecting area of 44 m^2 and a reflectivity of 0.94. The McDonnell-Douglas heliostat consists of a glass, front surface, silvered, octaganol mirror with acrylic coating for protection with a total reflecting area of 56.9 m^2. The Martin Mariette heliostat uses 11 flat or focussed and individually canted mirror assemblies mounted on a rigid structure with a total reflector area of 57.4 m^2. The Northrup's heliostat is a dual axis unit with a central support pedestal drive mount with twelve 1.2 m by 3.6 m mirror modules. The Westinghouse heliostat uses thirteen 1.5 m x 3.6 m and two 1.5 X 3.0 m mirrors which are front coated with silver and then with a protective layer of titanium dioxide. Whatever may be the type and size of the heliostat, it will have four subassamblies: the heliostat (reflector panel); the drive unit (including the pedestal); the foundation; and the heliostat electronics (including controllers and control sensors).

3.5.3. Sun tracking

In a solar furnace, the concentrator in a direct type, and the heliostat in a indirect type is to follow the apparent movement of the sun. This tracking can be done either by manual operation, or by astronomical method or by servosystem. Manual tracking results in jerky rotation and is generally not accurate and therefore unsatisfactory. An astronomical method is used in a solar furnace near Bouzereah, Algeria, where the concentrator is mounted on an equatorial axis and its motion is controlled by an astronomical clock. Here the declination of the mirror is daily adjusted by hand but the axis of the paraboloid remains in the direction of sun even if the solar radiation is intercepted by the clouds. Such tracking systems are not accurate and are expensive for large solar systems. In a servosystem which is generally employed, a little deviation of solar radiation incident on the concentrator is sensed by photocells, which operates the azimuth and elevation driving system and make the radiation parallel to the optical axis of the concentrator. A highly sensitive servosystem, developed by Army Medical Corps, USA, uses a selsyn system alongwith four photocells mounted on the mirror. If all the four cells are not uniformly illuminated (out of focus), the system generate an a-c-signal which, through the selsyns, makes two motors turn the mirror back to focus. This system differentiate between elevational and azimuthal movement, and permits rotation in two direction around both axis. In case the sun is momentarily covered by cloud, the system may immediately turn the concentrator as much as 90^{0} off the sun, only to swing back again when the sun is again uncovered.

The above system is made less sensitive[65] by using a 37 watts, 400 rpm, 110 Volt DC shunt gear motors in place of 186 watts, 1725 rpm motors used earlier, in order to reduce starting torque and angular momentum. The deviation from the direction of sun is detected by an optical tube which is fixed parallel to the axis of the concentrator. The direct solar radiation enters through one end of the optical tube and is incident on four photo-electric tubes on the other end of the optical tube. The phototubes are optically isolated and those on opposite quadrants are paired. One pair is aligned parallel to the horizontal axis of rotation of the concentrator and the other pair parallel to the vertical axis. The electrical circuit remains balanced if all the four phototubes are equally illuminated. As soon as one pair of phototubes become less illuminated, the electrical circuit becomes unbalanced, resulting an error signal directing the mechanism to rotate the concentrator until all the phototubes are equally illuminated i.e. the electrical balance is restored. It is doubted that this system does not provide smooth movement to the mirror towards the sun.

A simple guidance system specially for solar furnaces is developed by Schweiger and Laszlo[66], which provides a smooth and accurate movement to the mirror. In this system all the components used are commercially available. The same type of sensor is used as described above. The control unit consists of two identical modules, one to monitor and control the altitude sensor and drive motor, and the other to monitor and control the azimuth sensor and drive motor. Each module uses an electronic circuit as shown[66] in fig.3.11. The error signal from the paired phototubes changes the intensity and polarity of the current output of the balance units. The modulated current is fed to two small motors that drive the furnace rotating gears. This system is found to be most satisfactory, does not need any manual adjustment, and without oscillations.

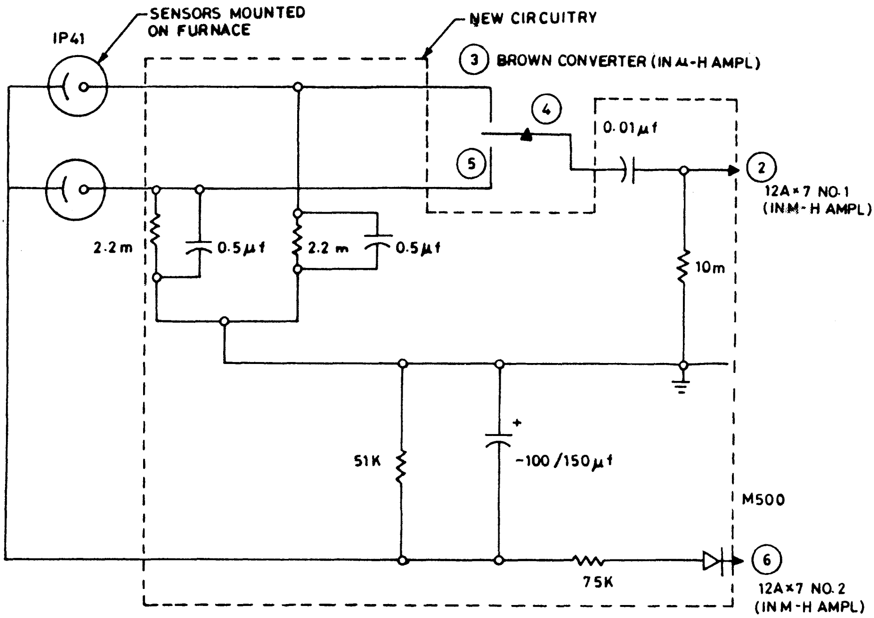


FIG.3.11 ELECTRONIC CIRCUIT FOR GUIDING SOLAR FURNACE (From Schweiger and Laszlo[66])

Another system for guiding the solar furnace which uses a servo system and time system is used by Sakurai et al[20] in their large solar furnace. Here the deviation is detected using four photo transister around a sun image obtained by focussing the solar radiation from a heliostat with a lens with about 6.5 m in focal length. In case cloud appears, the servo system ceases to operate, but the time system continues the heliostat rotation. This system helps in rapid and perfect recapture of the sun image when the cloud passes.

3.6 TYPICAL SOLAR FURNACE DESIGNS

As discussed earlier the solar furnace can have single concentrator or single concentrator and single heliostat or single concentrator with several heliostats. Here an example of each of the three types are discussed in brief.

3.6.1 Single concentrator furnace

Most of the solar furnaces of single concentrator type make use of search light 1.5 m diameter paraboloid reflector in equatorial mounting. A large solar furnace using a large single paraboloid of 8.4 m aperture, 3.14 m focal length, and 2.68 aperture ratio was designed by Messers Betier and Guillemonar[15] at Bouzareah, near Alger for use in nitrogen fixation processes. The paraboloid consists of 144 aluminium mirror segments, specially electropolished to high reflectivity. The total reflector area is about 52.58 m^2 with theoretical concentration ratio of 40000. The image diameter is calculated as 2.92 cm with a flux density of about 3200 watts/cm^2. In practice, however, the concentration and flux density are considerably lowered by the lack of optical smoothness of aluminium mirrors. The paraboloid is mounted on a equatorial axis and its motion is controlled by an astronomical clock. Here the declination of the mirror is daily adjusted by hand, but the axis of the paraboloid remains in the direction of the sun even the solar radiation is intercepted by the clouds.

3.6.2 Single heliostat solar furnace

Although several solar furnaces with single concentrator and single heliostat are built after the first solar furnace made at Montlouis, France[16,21], but the single largest heliostat solar furnace was made at Tohoku University, Sendai, Japan[20]. This furnace was designed by Sakurai et al[20] in the Research Institute for Scientific Measurements, Tohoku University, Sendai, Japan in 1964 for doing basic and applied research in high temperature physics. The aim was for getting high temperature for flat target and therefore a paraboloid of high aperture ratio, high reflectivity, correct geometry, smooth tracking system, and good optical design was chosen. The solar furnace consists of a paraboloidal concentrator with horizontal optical axis and a heliostat to convey the solar radiation into the concentrator as shown in fig.3.12.

The concentrator used is a paraboloid with aperture 10 m and focal length 3.2 m giving the aperture ratio of 3.1. The paraboloid consists of 181 mirror segments each of 80 cm X 75 cm with a total mirror area of 78.5 m^2. Each glass blanks were appropriately bent by heating them and then grinded on a specially designed surface grinder to give them the exact shape of the paraboloid. Each of the blanks were smoothed and polished to an accuracy of 1/2000 radian. The finished product is aluminized by vacuum evaporation on the front side and then a protective coating of vinyl chloride is provided. The paraboloidal mirror is supported on a steel skelton of 11 m wide, 7 m deep, and 13 m high. The temperature at the target is controlled by using two V-shaped diaphragms, by moving them toward the optical axis.

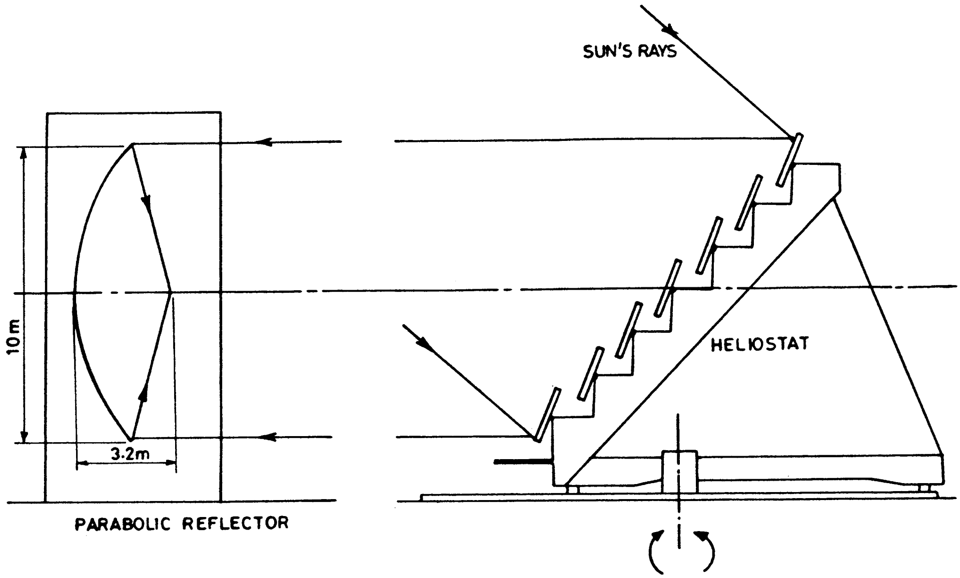


FIG.3.12 OPTICAL SYSTEM OF 70 KW SOLAR FURNACE AT SENDAI, JAPAN

The heliostat consists of 238 mirrors each of 90 cm X 100 cm and 10 mm thick arranged in seven rows, each row being 2 m wide and 15.5 m long with 34 segments. The steel skelton of the heliostat is of the form of a tumbled pyramid, and can be rotated on a circular track to change the azimuth. Here also the mirror segments are selected from best quality window glass sheets to meet the accuracy of 1/2000 radians. These mirrors are aluminized on front side by vacuum evaporation and coated with resin film to prevent the degradation of reflectivity. The sun tracking arrangement and focussing is done using time system, servo-system, and with little manual operation. When the heliostat is not in use, it is kept facing north and protected with a cover which moves along rails and is driven by a motor.

The performance of the solar furnace was quite satisfactory and in initial experimentation alumina and other refractory materials were melted, and tantalum (melting point 3030 °C) and tungsten (melting point 3400 °C) plates were melted in a helium atmosphere. In very fine weather, the flux density equivalent to the blackbody at the temperature of 3800 °C can be obtained at the centre of the sun image. The furnace is used to grow crystals of various oxides including, $MgAl_2O_4$, NiO, CaO, Y_2O_3, ZrO_2, and UO_2, and other optical properties are studied. The thermal expansion of materials like CaO, and SnO_2 were also measured at very high temperatures using a specially built high-temperature X-ray diffractometer and solar furnace.

3.6.3 Multiple heliostats solar furnace

One of the world's largest solar furnace, 1000 KW, was completed in 1970 and is located at Odeillo, Font-Romeu at an elevation of 1800 m, about 40 Km east of Andorra and 8 Km west of Montlouis in France[21]. The furnace was completed on October 1,1970 after 10 years of design, construction, and alignment. The installation site has more than 180 days of bright sunshine in one year i.e. about 1200 hr of sunshine per year and solar intensity of about 1000 W/m^2 is also common. The solar furnace consists of a paraboloidal concentrator with its optical axis horizontal and 63 heliostats directing sun rays to the concentrator parallel to its optical axis as shown in fig.3.13.

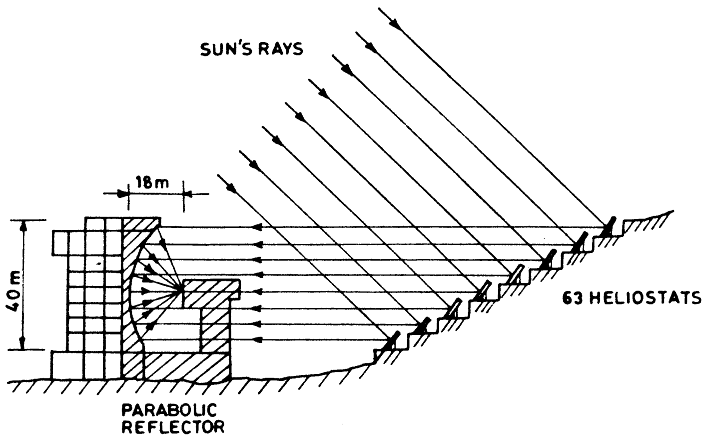


FIG.3.13 OPTICAL SYSTEM OF 1000 KW SOLAR FURNACE AT ODEILLO, FRANCE.

The paraboloidal concentrator with a focal length of 18 m, 40 m high, 54 m wide, and 13 m above ground level uses 9500 back-silvered plane mirrors with 45 cm X 45 cm in dimension. These small mirrors are mechanically curved and given an effective mirror area of 1920 m^2. The effective aperture ratio is about 2.8 and provides the input solar energy of 1800 KW and after considering reflection loss the power is estimated as 1000 KW. The flux concentration ratio on a 0.10 m^2 absorber is calculated as 20000.

The solar radiation is directed towards the concentrator by 63 heliostats located on hills on eight levels corresponding to the eight floors of the paraboloidal concentrator support structure. Each heliostat is 7.5 m wide, 6 m high and consists of 180 single flat-plate mirror elements 50 cm X 50 cm. The total heliostat area is about 2835 m^2.

Each heliostat illuminates a particular area of the concentrator and their orientation is controlled by a dual optical control system maintaining the orientation of heliostat using a dual hydraulic system. This dual system keep the heliostat either in 'search'or track mode. In each case, the optical guidance system uses an optical tube 100 cm long and 1.2 cm in diameter containing four photocells, which control the heliostat motion in east-west and up-down direction. The tracking accuracy is 1 minute of arc.

The experimental data like useful heat flux, and temperatures in the Odeillo solar furnace for different receiving diameters, 2-40 cms, is given[67] in table 3.3. From this

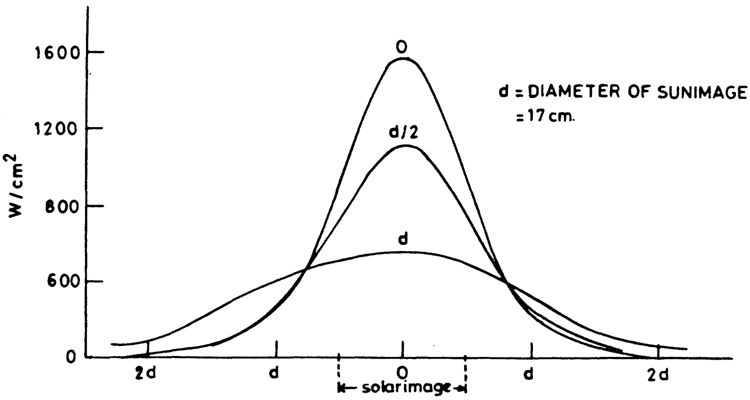


FIG.3.14 FLUX DENSITY VERSUS DISTANCE FROM FOCAL POINT (From Trombe et al[21]).

Table 3.3 Thermal performance of Odeillo solar furnace (From Trombe at al[67])

Diameter of radiation (cm)	2	6	12	16.8	20	30	40
Percent of total energy in area of radiation	0.5	4.50	15.5	27	35	58	75
Energy in area of radiation (KW)	5	45	155	270	350	580	750
Minimum heat flux (w/cm^2)	1600	1472	1200	912	800	400	192
Average heat flux (w/cm^2)	1600	1595	1370	1215	1115	820	595
Minimum temp. of radiation (℃)	3825	3740	3540	3285	3170	2625	2140
Average temp. of radiation (℃)	3825	3805	3665	3585	3465	3185	2950

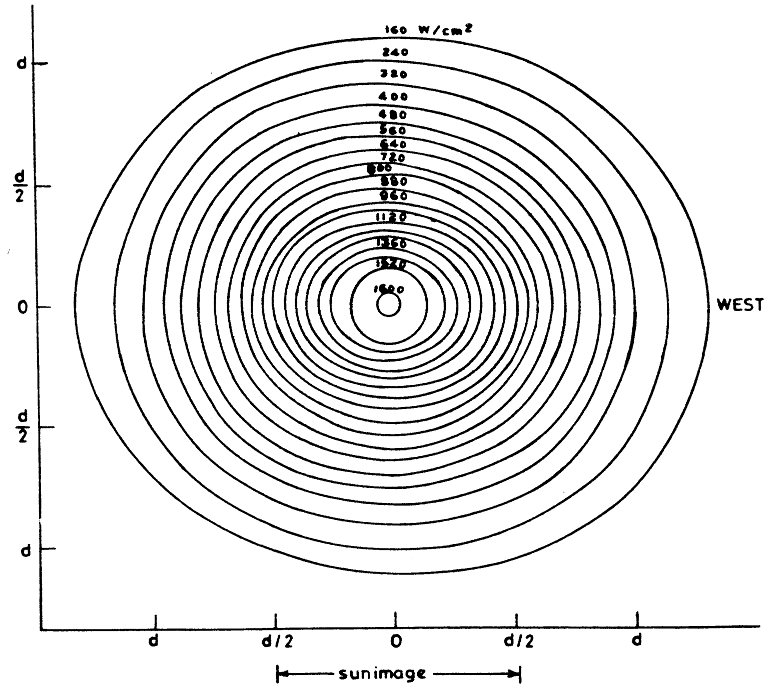


FIG.3.15 DISTRIBUTION OF HEAT FLUX DENSITY IN 25 DEG. TILTED PLANE (from Trombe et al[21])

table it is seen that as the required flux density increases, the useful area decreases. For instance, if we want to obtain the average heat flux of 1370 w/cm^2, then the area to be used will be of about 12 cm in diameter in the focal spot The energy received in this area will be 155 KW, while the equivalent temperature of blackbody amounts to 3665 ^{o}C. For many processes, lower temperatures are required and in these conditions the useful power will be much higher. In the area 40 cm in diameter, the received power is 750 KW, the average heat flux 595 w/cm^2 and the equivalent black body temperature 2950 ^{o}C. The diameter of the sun image corresponds to a 17 cm diameter zone, and the average heat flux in this area is 1215 w/cm^2 and the temperature corresponds to 3585 ^{o}C. The heat flux data in the focal plane of the furnace is shown[21] in fig.3.14. Curves 0, d/2, and d represent respectively the heat flux distribution on the focal plane, the distribution on this plane at a distance one half the diameter of solar image (8.5 cm) behind the focal plane, and that on the plane at a distance on one diameter (17 cm) behind the focal plane. The flux density in w/cm^2 on a plane tilted up from the vertical position to a 25° inclined position is shown in fig.3.15.

3.7 MEASURING INSTRUMENTS IN SOLAR FURNACE

Temperatures in the solar furnace may reach as high as $3500\,^\circ C$, hence conventional instruments used for the measurement of heat flux, temperature, thermal conductivity, thermal diffusivity, thermal expansion, specific heat, spectral emissivity, electrical properties, crystal structure, etc. generally cannot be used. In many cases, completely new methods and instruments are designed to accomodate the radiation conditions and existing unusual space. If accurate instrumentation is done then only the solar furnace may prove to be a unique and versatile tool in high temperature research.

3.7.1 Measurement of heat flux

In the focal area of the solar furnace, heat fluxes corresponding to a blackbody temperature of about $3500\,^\circ C$ can be obtained. The absolute value of incident heat flux at various points of the focal spot of a solar furnace can be determined using a black body cavity which receives the heat flux and the resulting thermal effect can be measured using standard calorimetric measurements. These cavities behave like black bodies, and internal walls heats up due to internal multireflections. Calorimetric methods for flux measurement in solar furnace are described by Trome et al [68], Laszlo[69], Glaser[70], Cotton et al[71], and Laszlo[72]. Laszlo[72] had developed a special blackbody cavity and an improved calorimetric method for its use even to the upper flux limits of the solar flux. The block diagram of the experimental set up is shown in fig.3.16. The concentrated radiation flux through the orifice enters the black body placed in the focal area. The absorbed heat by the walls of the cavity is transferred to the distilled water flowing through it at constant temperature and at constant flow rate. The temperature rise is measured with differential thermopiles. The direct radiation is measured by standard pyrheliometer mounted on the concentrator parallel to its optical axis. Both the time and length of calorimetric run is recorded on recorder. From both these measurements the radiant energy flux per unit normal incident solar radiation can be calculated.

Another simple, quick, and direct method is the radio-

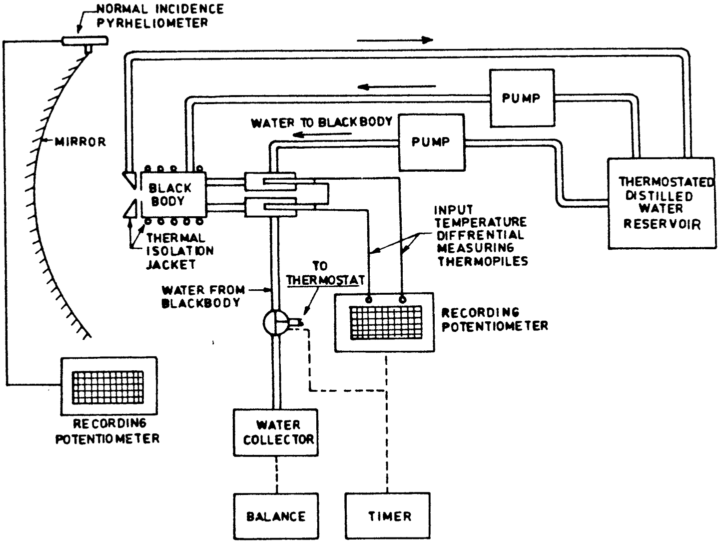


FIG.3.16 BLOCK DIAGRAM FOR THE THERMAL FLUX MEASUREMENT BY CALORIMETRIC METHOD (From Laszlo[72].

metric method as suggested by Gardon[73]. This instrument shows fast response (60 per cent of signal reached in 0.02 sec.) and very fine resolution (sensing disc. diameter is 0.899 mm). However, the camphor soot coating and M_gO coating at elevated temperature shows deterioration and their properties also changes at such temperatures. These coatings can be redone and therefore instrument is to be calibrated for precise measurements.

3.7.2 Heat flux regulation

The heat flux reaching the target can be regulated either by using venetian blinds, or some kinds of screens in the path of the solar radiation reflected from the heliostat or in the path of radiation reflected from concentrator. Special attenuators are described by Penniman et al[74] which are used in the USA Natick Solar furnace and gives much uniform energy flux. In the 70 kw solar furnace at Sendai, Japan, two V -shaped diaphragms, above and below, are used which can be brought towards the optical axis, to control the flux reaching the target.

3.7.3 Measurement of temperature

Temperature at the focal spot of a solar furnace is quite high and hence its direct measurement is not possible. Indirect methods like optical pyrometers can be used for such measurements. But the pyrometer apart from receiving the thermal radiation emitted from the heated target also receives the radiations after reflection or scattering from the target. Many investigators have suggested the accurate measurement of temperatures by cutting the reflected radiations. Conn and Braught[75] have suggested the use of a rotating cylindrical sector, which rotates about an axis parallel to the optical axis of the concentrator and interrupts the incident solar radiation momentarily and allowing only the emitted radiation to the pyrometer. Special pyrometric method for avoiding the reflected radiation reaching the pyrometer are developed by Kamada[76], Mann[77], Noguchi et al[78], Diamond and Schneider[79], and Noguchi and Kozuka [80]. Kamada[76] and Mann[77] have selected a wevelength of 1.38 μ m, which is absent in the solar radiation due to its absorption in water vapor, as a measure of temperature with sufficient accuracy while Noguchi et al[78] have selected a wavelength of 2.72 μ m which is also absent. The specular radiation surface of a molten material has been used to separate the two components of radiation by Diamond and Schneider[79], and Noguchi and Kozuka[80].

Looking to the problems in the method proposed by Kamada[76] and selecting a wavelength 1.38 μ m (wavenumber of 7250 cm^{-1}) at which most of the metal oxides are transparent, molten material may have low emissivity compared to

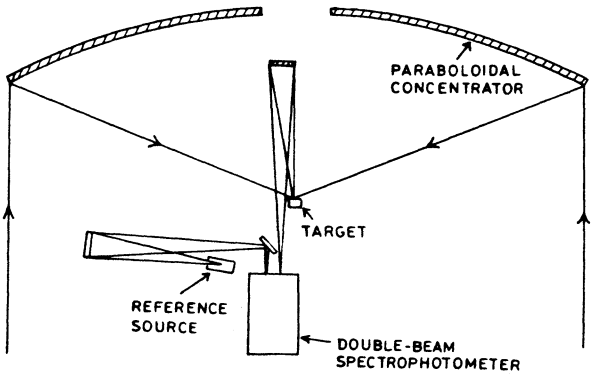


FIG.3.17 DOUBLE BEAM OPTICAL SYSTEM FOR TEMPERATURE MEASUREMENT

the solids one, and in case of transition metal oxides the molten material may show high emissivity due to absorption of transition metal ions, Arashi and Sakurai[81] have suggested a new infrared pyrometer working in higher wavelength region. The optical arrangement suggested is schematically shown in fig.3.17. The thermal radiation from the target irradiated at the focus of the concentrator of the solar furnace and from the reference sample kept at $1000 \pm 1\ ^{o}C$ in an electric furnace are introduced into a double beam infrared spectrophotometer as sample and reference beams respectively. The optical path lengths of two beams are same. The spectrophotometer records the ratio of the intensity of target-radiation to that of reference-radiation with time at a given wavelength by an optical null method. The temperature of the target is obtained from the intensity ratio of reference to target radiation referring to the black-body radiation formulae. Accuracy in temperature measurement obtained is about 0.5 percent.

3.7.4 Measurement of emissivity

Measurement of emissivity at high temperatures is required to estimate the radiation exchange and to interpret the temperatures. Data of emissivity of materials at elevated temperatures is scarce. Emissivity is a function of temperature but no definite correlation is developed[82], since this is a intrinsic property of the material and very much depends on phase changes, sintering, surface texture, stoichiometry, etc.

Glaser[83,84] was the first to suggest a method to measure the spectral emissivity of metals, refractory materials, and other construction materials. Laszlo et al [85,86] have also suggested a method for emittance measurement and thermal emissivities of alumina, magnesia and zironia are measured. This method was found inadequate in large solar furnace and hence Guitronich[87] used a special equipment in which a rotating chopper system is used in conjunction with a monochromator and optical pyrometer, for the measurement of emittance in the large solar furnace at the US Army Natick Laboratories. Shcherbina et al[88,89], Yanulis and Mayauskas[90], and Mavashev et al[91] described methods for the measurements of spectral reflectances of various materials. From the spectral reflectance measurements, they calculated the spectral emittance by applying Kirchhoff's law. Recently Noguchi et al[80,92,93] have developed an improved technique for the measurement of spectral emittance and reflectance of materials at elevated temperatures in solar furnaces.

The instrument described by Noguchi et al[93] is suitable for spectral reflectance and emittance measurements of materials at temperatures beyond $2000\ ^{o}C$ in the spectral range of 0.38-25 μm. Radiation from a heated specimen is directly compared to that from a tungsten iodine lamp (0.38-2.5 μm) or a blackbody furnace (2.5-2.5 μm).The spectrometer used is schematically shown[93] in fig.3.18.

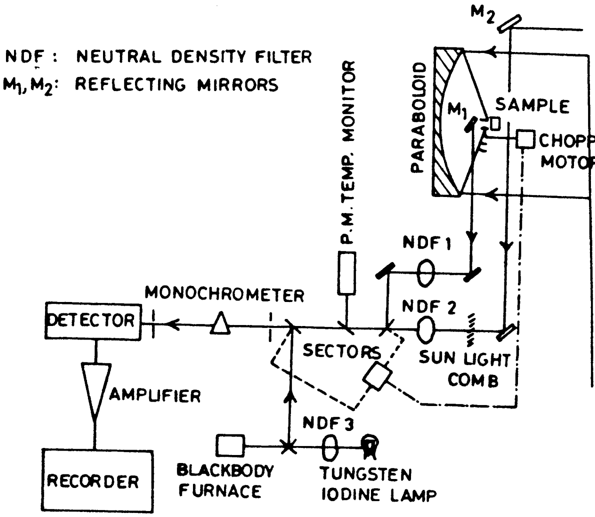


FIG.3.18 SCHEMATIC DIAGRAM OF SPECIAL EMITTANCE AND REFLECTANCE MEASUREMENT. (From Noguchi et al[93])

The apparatus consists of the following components:

* The solar furnace consists of a paraboloid mirror of 1.5 m in aperture, 64 cm in focal length and one heliostat, 240 X 240 cm with 16 segments.
* A chopper is used to seperate the emittance and reflectance from the specimen surface. Chopper consists of 4 aluminium cylinders with 5.8 cm in diameter, 10 cm in length and rotates at 300 rpm. The schematics of chopper mechanism is shown[93] in fig.3.19. By the help of phototransistors phase seperation is made in the respective signals.
* Three detectors in their respective spectral ranges i.e.photomultiplier for 0.38 - 1.0 μm, PbS for 1.0 - 2.5 μm, and Cu - Ge detector cooled in the liquid helium for 2.5-25 μm are used.
* Double monochromator of JASCO model IRS type with 60° KBr prism equipped with collimating mirror of focal length 30 cm and variable slit width 0.01-3 mm is used.

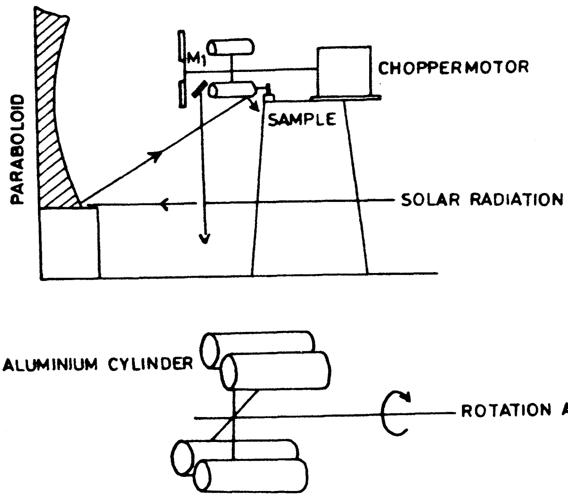


FIG.3.19 SCHEMATIC OF CHOPPER MECHANISM (From Noguchi et al[93]

* The energy level of the input signals is monitored with the combination of circular neutral variable density filters, model NVDF 270 and light wedges for the input signal from the sample.
* Two types of reference standards, a tungsten iodine lamp (0.38-2.5 µm) of model JC-24-150, and blackbody furnace (2.5-25 µm) of JASCO model RA-10 K are used.
* The spectrometer used has four scanning speeds, 10, 20, 30, and 60 minutes for the full range with two speed synchronous motor. Five signals along with the wavelength are recorded using electromagnetic oscillograph.

The spectral emittance ε_λ, and spectral reflectance r_λ can be calculated using following relations:

$$\varepsilon_\lambda = 1 - r_\lambda \tag{3.7}$$

and

$$r_\lambda = \frac{V_{2\lambda}\,\tau_{1\lambda}}{V_{1\lambda}\,\tau_{2\lambda}} \tag{3.8}$$

where

$V_{1\lambda}$ = output of spectrometer by solar radiation.

$V_{2\lambda}$ = output of spectrometer by reflected radiations from specimen surface.

$\tau_{1\lambda}$ = spectrol transmittance of neutral density filter NDF1.

and $\tau_{2\lambda}$ = spectral transmittance of neutral density filter NDF 2.

3.7.5 Measurement of electrical conductivity

Several workers[94-98] have reported measurement of electrical conductivity of magnesium oxide crystals and many other materials. Sakurai et al[98] have described a microwave ellipsometric method for the measurement of electrical conductivity of oxides in an oxidising atmosphere at elevated temperatures without the use of electrodes, which is not possible with any other method. The instrument is based on the principle that when a polarized microwave is reflected from a plane of conductive medium, amplitude and phase change take place which are different between the two components parallel and perpendicular to the plane of incidence; hence, the reflected microwave is elliptically polarised. The optical constants n and k of the medium can be deduced by measuring the amplitude ratio $\tan\psi$ and the relative phase difference Δ between the two polarised components using the following Fresnel equations:

$$n^2-K^2 = \mathrm{Sin}^2\Theta\left\{1+\frac{\tan^2\Theta(\mathrm{Cos}^2 2\psi-\mathrm{Sin}^2 2\psi\ \mathrm{Sin}^2\Delta)}{(1+\mathrm{Sin}2\psi\ \mathrm{Cos}\Delta)^2}\right\} \qquad (3.9)$$

$$2nk = \mathrm{Sin}^2\Theta\ \tan^2\Theta\left\{\frac{\mathrm{Sin}4\psi\ \mathrm{Sin}\,\Delta}{(1+\mathrm{Sin}2\psi\ \mathrm{Cos}\Delta)^2}\right\} \qquad (3.10)$$

where Θ is the angle of incidence.

Now the electrical conductivity σ of the medium is given by nKC/λ and therefore can be determined at any wavelength λ without the use of electrodes.

The apparatus used is schematically shown[98] in fig.3.20 and the microwave ellipsometer showing the details is shown[98] in fig.3.21. Microwave of 50 GH (λ = 6 mm) polarised 45° to the plane of incidence come from the transmitter to a target surface at the incident angle of 45°. The reflected wave then enters the receiver to be analysed. The detailed components of the receiver and transmitter are as follows:

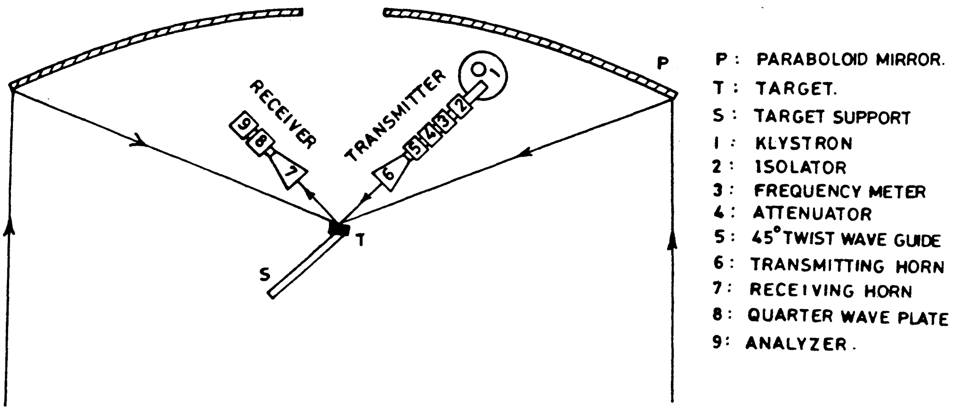


FIG.3.20 MICROWAVE ELLIPSOMETER USED FOR ELECTRICAL CONDUCTIVITY MEASUREMENT (From Sakurai et al[98])

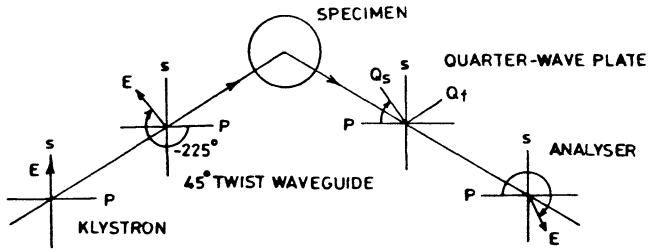


FIG.3.21 DETAILS OF MICROWAVE ELLIPSOMETER (From Sakurai et al[98])

Receiver : Specimen → receiving horn → quarter wave plate → analyser → detector.

Transmitter: Klystron → isolator → frequency meter → attenuator → 45° twist waveguide → transmitting horn → specimen.

With this apparatus the parameters ψ and Δ can be measured to an accuracy of 0.1°.

3.7.6 Measurement of thermal expansion

The knowledge of thermal expansion of materials at elevated temperatures is required to know the fatigue failures in anisotropic and two-phase systems, to estimate the thermal-spalling resistance, to get information on the disorder state of solid solutions, etc. This information will also be useful in selecting proper refractory materials, structural materials and material for special apparatus design. One simple method of measuring the thermal expansion of materials is to take the material in the rod form and heating it in a blackbody cavity and measuring its length and temperature optically. A method similar to this was used by Chalmin[99] to measure the thermal expansion of zirconium silicate at a temperature of 1900 $^{\circ}$C. His apparatus using a cavity type dilatometer with twin microscope comparator is shown[100] in fig.3.22. A similar dilatometer was

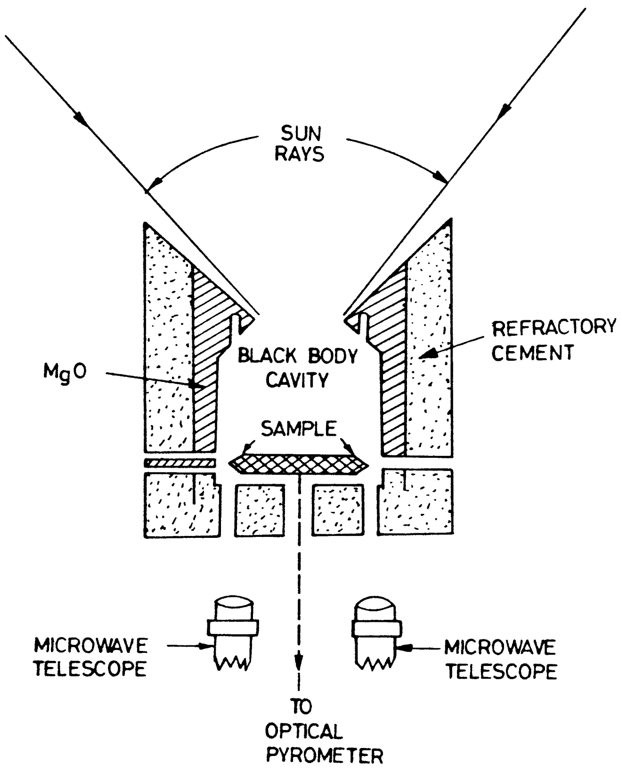


FIG.3.22 CAVITY TYPE DILATOMETER IN SOLAR FURNACE

suggested and used by Glaser[83]. This type of dilatometer gives several difficulties and sometimes the results are inaccurate.

Some of the properties like thermal expansion, phase transition, and other thermophysical properties can easily and more precisely be determined using high temperature x-ray technique. The X-ray method will also be useful for certain oxides for which the single crystals are difficult to prepare. Noguchi[101] and Kamada et al[102] separately studied the thermal expansion of certain oxides beyond 2000°C in solar furnaces equipped with specially designed goniometer used for the high temperature X-ray diffractometry. The goniometer as designed by Kamada et al[102] with its position in the solar furnace is shown in fig.3.23 and the details of the goniometer are given in fig.3.24. The specimen remains fixed, with its surface normal to the optical axis of the concentrator, and the source and detector are rotated about the axis of goniometer at the same angle in opposite directions with the help of A-B-C link system. The source S is rotated around the goniometer axis with the help of worm W and a gear G. The X-ray source used is Toshiba A-15 tube with a Cu-target. The X-rays after diffraction from specimen enters through an antiscatter slit, a Sollar slit, and a variable receiving slit and finally to the Geiger-Muller's counter Amperex 153C fitted with a preamplifier.

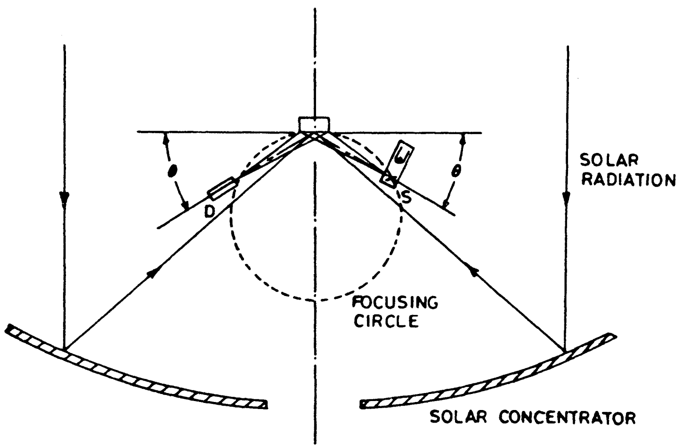


FIG.3.23 SCHEMATIC DIAGRAM OF GONIOMETER IN A SOLAR FURNACE (From Kamada et al[102])

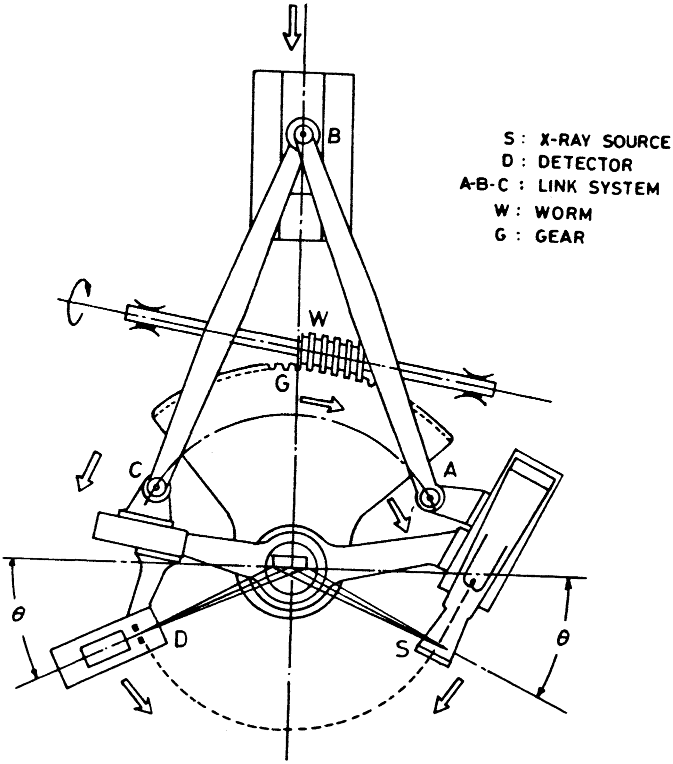


FIG.3.24 SCHEMATIC DIAGRAM OF GONIMETER (From Kamada et al[102])

3.7.7 Ablation studies

Ablation studies are useful in the heat-shield design. For a reentry vehicle at an escape velocity in the earth's atmosphere both the radiant energy and convective energy are equally significant. The combined effect of radiant and convective heating forces cannot be studied together, and hence the effects are studied separately. The effects of convecting heating can be studied using electric arcs and windtunnel while the effects of radiant heating can be studied in a solar furnace.

The first attempt to study the ablation properties of some plastic materials was made by Gruntfest and Shenker

[103], but his work was of exploratory nature. An apparatus used for the measurement of ablation properties of materials was described by Sheehan et al[104], in which a solar furnace of 1.5 m paraboloid reflector is used as a source of high intensity radiant energy. The apparatus is schematically shown[104] in fig.3.25. During the heating processes,

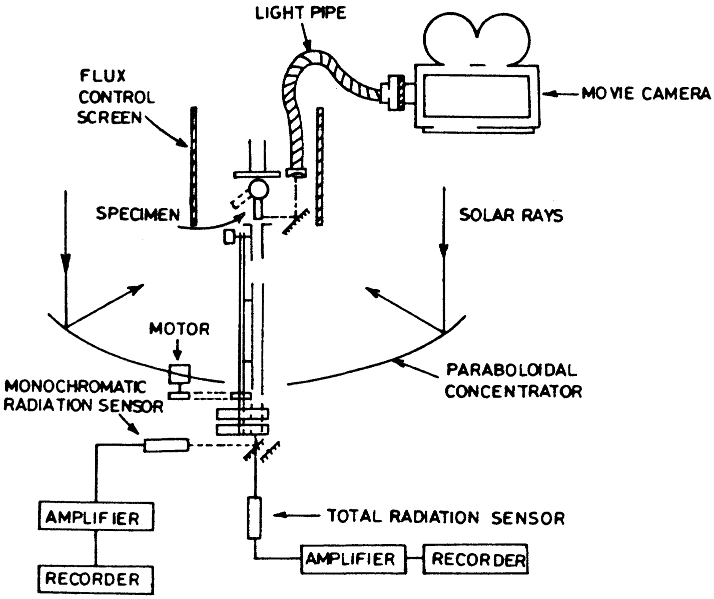


FIG.3.25 SCHEMATIC OF ABLATION TESTS FACILITY (From Sheehan et al[104])

the recession rate of the surface of the sample is observed and measured using a movie camera taking pictures at the rate of one photo per second. To avoid shading of camera, a 3.65 m long flexible fibre optics light pipe is used to transfer the image. The difficulties as pointed out by the authors are the lack of accuracy in the measurement of reradiated flux, and the surface temperature of the sample. Laszlo et al[106] used this apparatus and measured ablation properties of three refractory materials and three polymeric heat shield materials as a function of incident flux and time. This study has clearly indicated that the optical properties of materials should also be considered while selecting ablators.

3.7.8 Thermal diffusivity measurement

The thermal diffusivity ($\alpha = K/\rho C$), where K is the thermal conductivity, ρ the density, and C the specific heat of the material, is a important property of the material and is responsible for the temperature distribution in the material under unsteady state conditions. The thermal diffusivity of both metals and nonmetals at elevated temperatures in vacuum or desired gaseous atmosphere can be determined in a solar furnace using the thermal wave method. This method is used by Mavashev[106], and Borukhov et al [107,108] for the measurement of thermal diffusivity of metals in the form of rod in temperature range of 400-1600°C and nonmetals in the form of disc in the temperature range of 1000-3500°C using a 3 m diameter solar furnace. The apparatus used is schematically shown in fig.3.26. When the

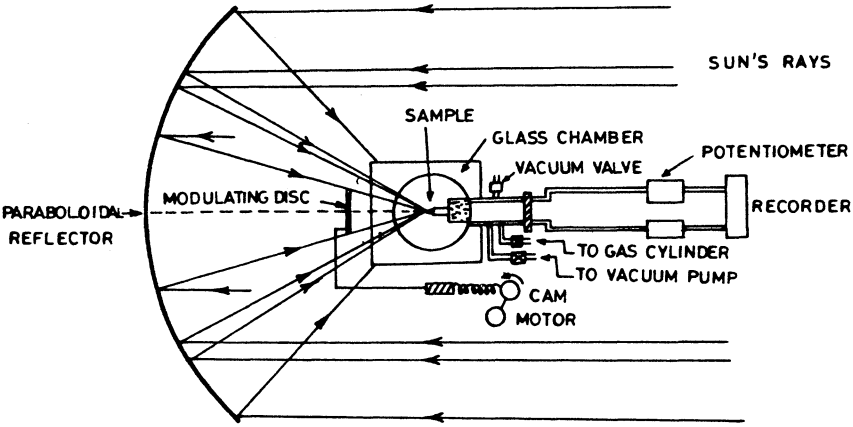


FIG.3.26 SCHEMATIC OF SET FOR MEASURING THERMAL DIFFUSIVITY (From Borukhov et al[107])

specimen is the form of a rod, one end is exposed to periodically varying thermal flux placed in the focal zone of the solar furnace. At two points along the axis of the rod, the temperature wave amplitudes is measured and from the amplitude ratio, the thermal diffusivity of the rod material is determined.

When the specimen is in the form of the disc, it is placed in the focal zone of the furnace, and the thermal flux is made to vary periodically, giving rise to temperature oscillations on the exposed side of the specimen. These oscillations on the other side of the specimen are recorded by radiation detector. By knowing the phase shift in the temperature oscillations on both the sides, the temperature diffusivity of the disc material can be determined. This method is also suggested to be used for measuring thermal conductivity and specific heat by knowing absolute values of temperature amplitude and thermal flux input. The method is used for measuring thermal diffusivity of materials like Armco iron, zirconium dioxide, aluminium dioxide, boron carbonitrate, nickel oxide, titanium carbide, etc.

3.7.9 Measurement of specific heat

Specific heat data shows the heat content value and is therefore extremely useful parameter for selecting a material suitable for heat shields of reentry vehicles or in combustion chambers. Data on heat content of materials at high temperatures is hardly available. Apparatus for the measurement of specific heats at high temperatures (upto 2500 ^{o}C) are described by Glaser[83] and Bulter[109] in which

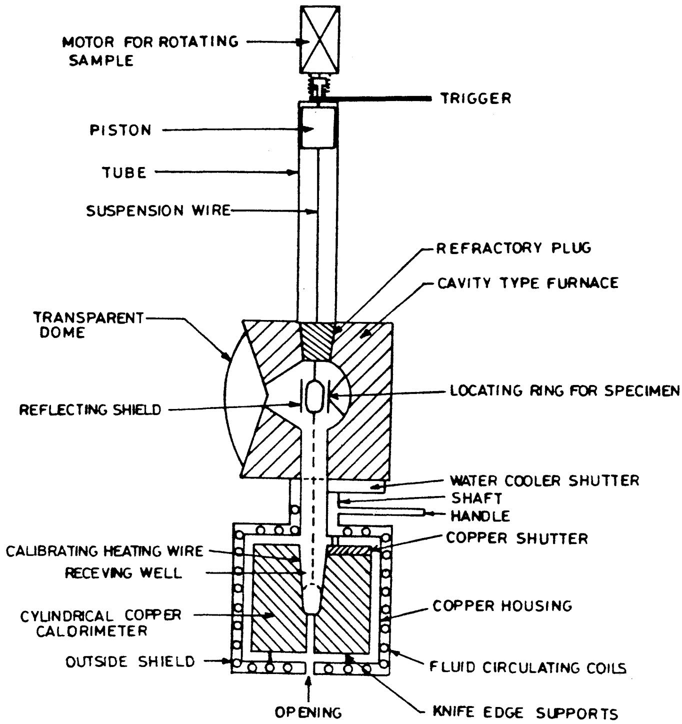


FIG.3.27 HIGH TEMPERATURE SPECIFIC HEAT APPARATUS (From Glaser[83])

the use of solar furnace is made. The apparatus as suggested by Glaser[83] is schematically shown in fig.3.27. The apparatus consist of two parts: a cavity type furnace and an aneroid drop calorimeter. The specimen is kept in a suitable capsule and suspended in the cavity by a pair of thin wires. The temperature of the specimen can be measured by thermocouples at low temperatures and by optical pyrometer at high temperatures. The sample can be dropped into the calorimeter below the cavity. Experimental data is not reported in either of these two papers.

3.7.10 Measurements in controlled atmosphere

In a solar furnace, the testing can be done either in an air or in a controlled atmosphere, and instrumentation can easily be done by bringing them near to the sample without any damage to them. Moreover, the sample is avoided from contamination from unfavourable environmental factors, since the heat source remains quite far from the sample. Under such controlled atmosphere single crystals of highest purity could be grown which otherwise decompose or vary in stoichiometry under normal heating conditions. The controlled chamber should essentially have a pyrex or fused silica container such as a tube, flask, or hemi-spherical cover, a sample holder, tubing for evacuation and gas flow etc. The sample holder should be provided with some kind of rotating mechanism, so that the sample can be located at the focal plane. The temperature of the sample can be controlled by using a attenuator, door, or curtain in front of the paraboloid reflector.

3.8 STUDIES ON MATERIAL PROPERTIES USING SOLAR FURNACE

Since solar furnace provides high intensity heat flux and high temperature in a controlled atmosphere, it has been and is being used for the following applications[6]:

* High temperature chemistry involving the formation of pure materials.
* High temperature processes involving purification and stabilisation of materials.
* Property studies of materials of high temperatures under controlled atmosphere.
* Material behaviour studies and thermal shock resistances under high temperature and high radiant energy environment.
* Solar thermal conversion systems operating at high temperatures.

Some of the applications of solar furnace are recently reviewed by Suresh et al[109]. This part of the chapter will be based on this review paper[109].

3.8.1 Crystal growth

Solar furnace can be conveniently employed for growing single crystals of highest purity because of controlled atmosphere. Many crystals which is difficult to grow by the conventional methods like metal flux method or RF-heated floating zone method, can also be grown here. Attempts are made to grow crystals of ZrO_2, Al_2O_3, CaO, MgO, HfO_2, TiC, TiB_2, ThO_2, UO_2, NiO, Ni^{2+} doped CaO, and Y_2O_3, yet several difficulties are encountered and these are yet to be solved. The results on crystal growth as summarized by Suresh et al [109] are given in Table 3.4.

Laszlo[110] was the first to grow ZrO_2 single crystal in a solar furnace. E.A. Farber[39] has successfully grown crystals of aluminium oxide, calcium oxide, magnesium oxide, hafnium oxide, titanium carbide, and titanium diboride and determined their physical properties in a solar furnace. He tried both the methods of vapor deposition and puddle melt for growing crystals and found the second as better one. Even impure materials in the form of powder, rock, or crystal form, are employed to grow crystals. Crystals as large as 15 mm long and 5 mm thick have been grown using above methods. Single crystals of thoria using vapor deposition method were grown by Laszlo et al[111] using solar furnace. The size of crystals obtained were of 3 mm long and 1 mm thick and the growth rate at a flux of 21 MW/m^2 was $0.8x10^{-2}$ mm/sec. Sakurai and Ishigame[112] have grown Nickel oxide crystal using fusion technique in a solar furnace of 10 m aperture. Optically flat crystal surfaces of about 40 mm^2 in size are obtained. From microscopic and X-ray studies, 3-layered specimen was observed. A surface layer, 0.5-2 mm thick, is a single crystal. In the middle-layer, 1-5 mm thick, Ni metal crystals are found embeded by NiO crystal. The bottom layer is polycrystalline NiO and has a thickness of 3-5 mm. Sakurai et al[113] have grown uranium dioxide crystals in a helium atmosphere. Pyramidal shaped crystals with optically flat surface in 0.15 mm size were obtained. Ishigame et al[114] have grown crystals of CaO and Ni^{2+} doped CaO which are few mm in grain size using solar furnace. Yttrium oxide crystals which have hexagonal structure and few mm in size are grown by Nigara et al[115] using a solar furnace. Lack of suitable instrumentation and control mechanism may be the possible reasons for not getting purest and good size crystals, by solar furnace.

3.8.2 Phase-change studies

Solar furnaces have been successfully employed to study the phase diagram of ceramic, refractory and other materials melting at elevated temperatures, since a material can easily be fused in a solar furnace and quenched in air at a

Table 3.4 Crystal growth studies in a solar furnace (from Suresh et al[109])

Investigator	Crystal grown	Nature of Crystal
T.S.Laszlo et al, USA, (1956)	ZrO_2	Single crystal
E.A.Farber, USA (1964)	Al_2O_3, CaO, MgO, HfO_2, Tic, TiB_2	Single crystal
T.S.Laszlo et al, USA (1967)	ThO_2	Single crystal (3 x 1 x 1 mm)
T.Sakurai et al, Japan (1968)	UO_2	Crystals with pyramidal form of size 0.15 mm flat faces.
T.Sakurai et al, Japan (1968)	NiO	Polycrystalline, optically flat crystals of 40 mm^2 area.
M.Ishigame et al, Japan (1968)	Ni^{2+} doped CaO	Single crystals - colourless - a few mm in grain size.
Y.Nigara et al, Japan (1971)	Y_2O_3	Single crystal - a few mm size.

cooling rate of about 2300-1800 °C/sec. One can estimate the high temperature phase diagram of a polycomponent system using a solar furnace both as a static and a dynamic system. Good reviews of phase change studies of some materials using solar furnace are done by Noguchi[5] and Suresh et al[109]. These studies can broadly be classified into three systems: (i) zirconia based systems, (ii) alumina based systems, and (iii) lanthanide based systems. Some of the results are summarized[109] in table 3.5. Phase diagrams of all systems cannot be described here and therefore only the typical ones will be discussed.

One of the most zirconia refractory is the ZrO_2-CaO system, and the phase relationship of this system is studied by several workers including Duwez et al[137] and Noguchi et al[118]. Some inconsistant results are observed by many investigators. It is now confirmed that the monoclinic and tetragonal modifications exist at low and elevated temperatures respectively, and the cubic modification is stabilized

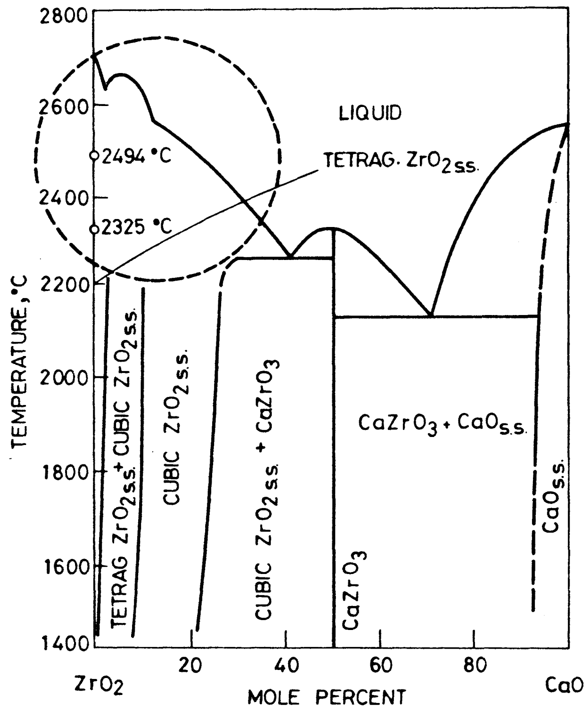


FIG.3.28 PHASE DIAGRAM OF ZrO_2-CaO SYSTEM (From Noguchi et al[118])

Table 3.5 Work on phase studies on some materials using solar furnace from Suresh et al[109]

P.Duwez et al[116] (1967)	ZrO_2-ThO_2	Tentative phase diagram given.
T.Sakurai et al[117] (1975)	ZrO_2-ThO_2	Complete phase diagram given.
T.Noguchi et al[118] (1967)	ZrO_2-CaO	Tentative phase diagram given.
T.Noguchi et al[119] (1968)	ZrO_2-MgO	Incomplete phase diagram given.
A.Rouanet[120] (1971)	ZrO_2-LnO_3	Complete phase diagram given.
A.Rouanet[120] (1971)	ZrO_2-Sm_2O_3	Complete phase diagram given.
A.Rouanet[120] (1971)	ZrO_2-Gd_2O_3	Complete phase diagram given.
T.Noguchi et al[121] (1967)	ZrO_2-TiO_2	Tentative phase diagram given.
P.E.Evans[122] (1960)	ZrO_2-UO_2	Tentative phase diagram given.
T.Noguchi et al[5], (1969)	ZrO_2-SrO	Tentative phase diagram given.
T.Noguchi et al[123] (1970)	ZrO_2-Y_2O_3	Liquidus curve given.
W.M.Conn[124] (1954)	Al_2O_3-SiO_2	Liquidus curve given.
T.Noguchi et al[5] (1967)	Al_2O_3-Y_2O_3	Tentative phase diagram given.

Table 3.5 cont.

M.Mizuno et al[125] (1975)	Al_2O_3-La_2O_3	Tentative phase diagram given.
M.Mizuno et al[125] (1975)	Al_2O_3-CeO_2	Tentative phase diagram given.
M.Mizuno et al[126] (1975)	Al_2O_3-Ga_2O_3	Tentative phase diagram given.
J.P.Coutures et al[127] (1976)	Al_2O_3-Nd_2O_3	Tentative phase diagram given.
M.Mizuno et al[128] (1977)	Al_2O_3-Pr_2O_3	Tentative phase diagram given.
M.Mizuno et al[129] (1977)	Al_2O_3-Sm_2O_3	Tentative phase diagram given.
M.Mizuno et al[130] (1977)	Al_2O_3-Eu_2O_3	Tentative phase diagram given.
M.Mizuno et al[130] (1977)	Al_2O_3-Gd_2O_3	Tentative phase diagram given.
M.Mizuno et al[131] (1976)	La_2O_3-Y_2O_3	Tentative phase diagram given.
F.Sibieude et al[132] (1975)	ThO_2-Ln_2O_3	Tentative phase diagram given.
J.P.Coutures[133] (1977)	Ln_2O_3-A_2O_3	Complete phase diagram given.
M.Yoshimura[134] (1977)	La O_3-WO_3	Complete phase diagram given.
M.Yoshimura[135] (1976)	Ce_2O_3-WO_3	Tentative phase diagram given.
G.Benezech et al[136] (1971)	Yb_2O_3-Cr_2O_3	Complete phase diagram given.

by a number of additives. In the experiments conducted by Noguchi et al[118], ZrO_2 of 99.8 percent purity and CaO of 99 percent purity was used. The liquidus curve data for ZrO_2-CaO system as obtained by them is shown[118] in fig.3.28. Anomalies in the liquidus curve were observed for 3.5, 7.0, and 10.0 mole percent CaO at 2603°, 2622°, and 2578° $\pm$20°C respectively. The maximum point was observed at 7.0 mole percent CaO. The crystal structure of the single phase of composition $Ca_{0.07}Zr_{0.93}O_{1.93}$ was suggested to be rhombic with optic angle 2V = -76° and birefringence as $(r-\alpha)=0.013$ from microscopic observation, while no clearly separated, extra reflection pattern was obtained. The quenched speciman got decomposed into monoclinic and tetragonal forms when reheated at 1300 °C, and remained stable at very high temperatures. The interpretation of data in the regions shown by broken lines is difficult because the behaviour of cubic ZrO_2 and cubic Zerconia solid solution and stability of $Ca_{0.07}Zr_{0.93}O_{1.93}$ in this region is not clear. The subliquidus phase in this region should be investigated with a dynamic method above 2500 °C in an oxidizing atmosphere.

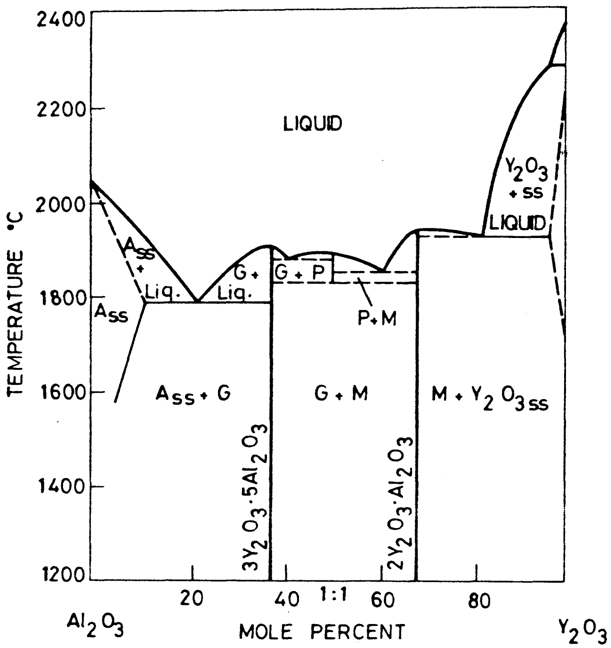


FIG.3.29 PHASE DIAGRAM OF AL_2O_3-Y_2O_3 SYSTEM (From Noguchi)
A, Al_2O_3 . G, $3Y_2O_3$ $5A_2O_3$. P, Y_2O_3 Al_2O_3 . M, $2Y_2O_3$ Al_2O_3

The phase diagram of Al_2O_3-Y_2O_3 is well studied and the garnet structure 5 Al_2O_3. $3Y_2O_3$ and monoclinic Al_2O_3. $2Y_2O_3$ is also established. But the stability of perovskite structure and the liquidus curve data near stoichiometric compound $Al_2O_3.Y_2O_3$ is not well reported. Noguchi[5] has studied the $Al_2O_3.Y_2O_3$ system using a solar furnace and the liquidus curve is shown in fig.3.29. The liquidus curve between 40 and 60 mole percent Y_2O_3 showed a gentle peak which might signify the metastable compound $Al_2O_3.H_2O_3$, and got decomposed at 1600 ^{o}C into a mixture of garnet, perovskite, and monoclinic Al_2O_3 $2Y_2O_3$.

Mizuno et al[131] have studied the phase diagram of $La_2O_3.Y_2O_3$ with the investigation on crystal modification of quenched specimen from the melt using a solar furnace. The liquidus curve for the system is shown in fig.3.30. With this study the presence of monoclinic and orthorhombic forms of La YO_3 is well established.

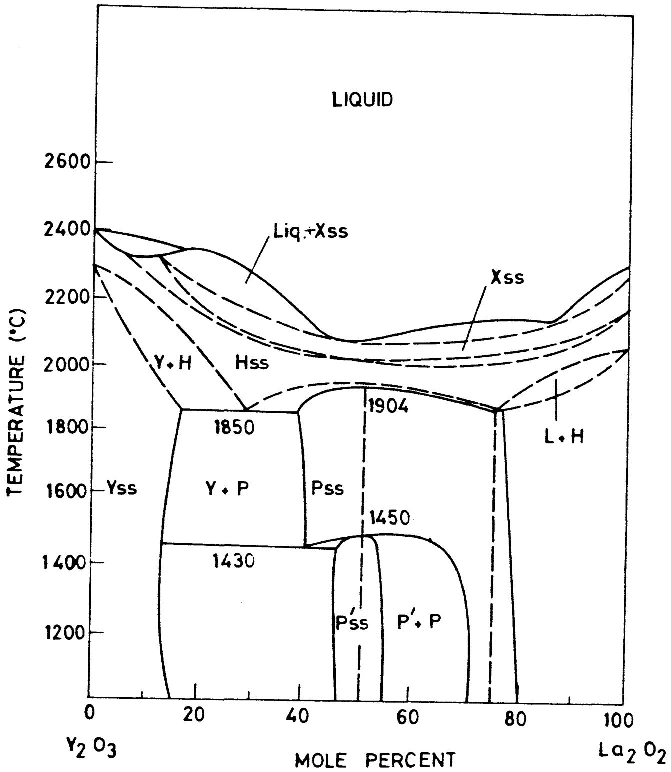


FIG.3.30 PHASE DIAGRAM OF La_2O_3·Y_2O_3 system (From Mizuno et al[131])

3.8.3 Other Applications

Since solar furnace provides high intensity heat flux, it can well duplicate the heat flux generated by a nuclear explosion and therefore can be used for ignition studies. By using accurately controlled and measured heat flux, one can easily know the maximum heat flux under which the materials will not ignite. Military furnaces both in France and USA are used to test materials in a simulated nuclear explosion environment.

Solar furnace offers a unique facility to purify the materials particularly at very high temperatures. In a solar furnace because the thermal cycling can be done easily, many of the volatile impurities will either vapourise or migrate to coolar regions of the speciman. The process of extraction of base materials from natural ores, clay and sands using solar furnace has reached the stage of commercialization. Purification of materials by zone melting using solar furnace has been successfully tried for several materials, and extremely pure samples of oxides of zirconium, thorium and uranium can also be obtained. Trombe and Foex[138] were able to get 98 percent pure amorphous ZrO_2; purify ThO_2, La_2O_3, and alumina. Pure synthetic minerals can also be produced in a solar furnace.

Materials suitable for high temperature applications can well be tested with a solar furnace. Studies are conducted on suitable high temperature-resistant materials to withstand the extreme conditions in atomic reactors, aircraft engines, and guided missiles. Otts et al[139] have reported that concrete can withstand 500 - 1000 suns and alumina fire bricks can withstand upto 2500 suns for 2 minutes. Properties of materials like refractories suitable as electrodes for MHD processes can also be studied. These materials should have good mechanical and electrical properties, good refractarity and must resist vaporization and corrosion at high temperatures. Noguchi et al[5] and Coutures et al[133] have identified materials using solar furnace for use in MHD processes.

Several chemical studies like decomposition reactions, vaporisation, oxidation, nitridation, etc. on refractory and other materials can also be ideally conducted in a solar furnace in controlled atmosphere. Wang et al[140] studied the properties of chromic oxide at high temperatures. The behaviour of TiO_2 was studied by Brover et al[141] in a 1.5 m solar furnace.

There are a variety of other applications[142] which have been tried like:

(i) Chemical vapour deposition of $Nb\,B_2$ and T_aB_2.

(ii) Production of sintered refractories such as zirconium oxide, calcium zirconate and alumina.

(iii) Studying the ice - nucleating properties of meteoritic material.

(iv) Welding and brazing in vacuum and preparation of hot junction bead of thermocouple.

(v) Studying nitrogen fixation process and other photochemical reactions.

(vi) Many applications in solar chemical engineering.

(vii) Synthesis of nitric acid.

Several futuristic applications of solar furnace are suggested as follows:

(i) New materials can be produced and properties of the materials can be studied in the outer space using solar furnace, since the outer space provides the unique advantage of utilizing ultra high vacuum (10^{-13} Torr), intense radiation, and weightlessness. Moreover, better control of heat and mass transfer would be possible due to the absence of gravity induced convection and buoyancy in molten metals.

(ii) Solar furnace can also be used for lunar water production.

(iii) In the outer space, solar furnace can also be used for the production of single crystal growth of silicon for semiconductor chips in microcircuits and solar cells, deposition of materials on surfaces, preparation of eutectic alloys, etc.

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CHAPTER - 4

SOLAR THERMO-MECHANICAL POWER

4.1 INTRODUCTION

Availability of cheap power is an index of technological advancement and standard of living of a country. Conversion of solar energy into mechanical power or electrical power has been a subject of research for nearly last three centuries. Most of the early research conducted on solar mechanical power generation was for small power generation and was abandoned not due to technological reasons but more due to economically more viable and cost-effective power options. It is hoped that solar generated power will play a significant role by the end of this century. There are several options of converting solar energy into electrical energy but the main ones are shown in Figure 4.1 The first option is the direct conversion of solar energy into electricity and the direct conversion can be done mainly by the following four methods:

1. Photovoltaic
2. Photogalvanic
3. Photoemissive
4. Photomagnetic

The direct conversion is not practically adopted for commercial purposes for large power demands due to high cost of cells. Thermal energy obtained from solar energy can also be directly converted into electricity by the following methods:

1. Thermoelectric
2. Thermionic
3. Ferroelectricity
4. Magnetohydrodynamics
5. Electrogasdynamics

The conversion efficiency of all the above methods is generally very low and therefore the system is not cost effective.

There is yet another very important way of converting solar energy into electricity which is known as thermodynamic way in which solar energy is converted into thermal energy; thermal energy into shaft work through heat engines based on the principle of either Rankine cycle, Stirling cycle, or Brayton cycle; and shaft work (mechanical energy) into electricity using alternator. In this chapter

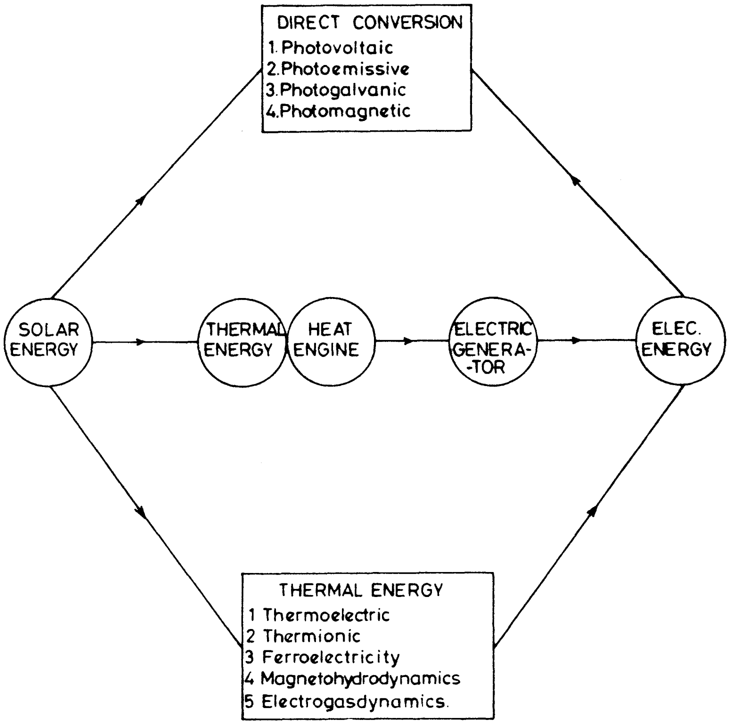


FIG.4.1 FEW SCHEMES OF CONVERTING SOLAR ENERGY INTO ELECTRICITY

only the thermodynamic way of converting solar energy into mechanical energy or electrical energy will be discussed.

A solar thermal power system mainly consists of a solar energy collector field, some kind of fossil fuel combustor (auxiliary system), a fluid flow distribution system, some kind of suitable thermal energy storage device, a heat engine, electric generator, and a control system as shown in figure 4.2 Amongst the many available systems, the two generic type of systems, the central receiver thermal electric power system and the distributed solar thermal electric power system are considered suitable because of their comparatively high efficiency and cost effectiveness. In the central receiver concept large arrays of sun-tracking mirrors known as heliostats reflect the solar flux on to the central receiver boiler at the top of the tower. Here concentration ratios of the order of 1000 are used and turbine (steam type) operates at about 600 ^{o}C. In the

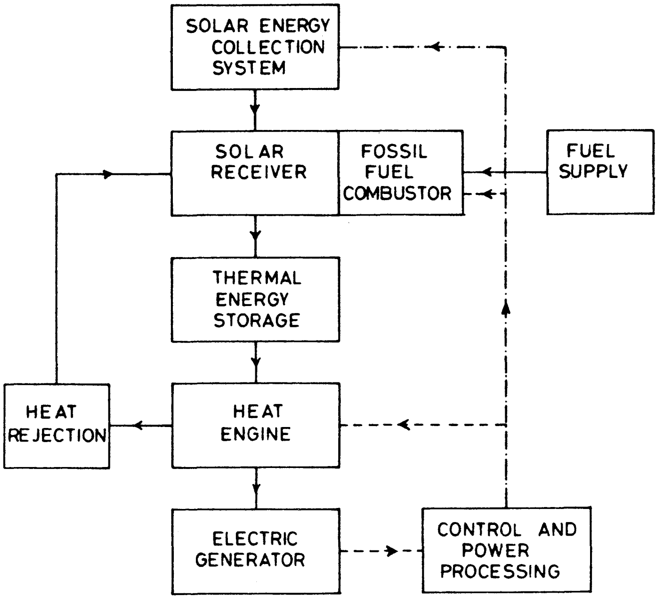


FIG.4.2 SIMPLIFIED BLOCK DIAGRAM OF SOLAR THERMAL POWER SYSTEM

dispersed or distributed power system, a large collector array consisting of line focus collectors such as parabolic troughs, linear fresnel reflectors, linear fresnel lenses or parabolic dish, spread over a large area is used. Energy is collected through pipes interconnecting the dispersed collector units and then supplied to the heat engines directly or through a heat exchanger where two different fluids are used.

The type of heat engine and the associated thermodynamic cycle depends on many parameters including temperature of operation, cycle efficiency, working fluid, cost, reliability, conversion efficiency, transport fluid, size of load, collector type, storage size, etc. Depending on the power produced, the solar thermal power systems are divided into the following three classes:

(i) Small units in the watts range. These units operate at very low thermodynamics efficiencies and are costly due to low efficiency, and are costly due to high heat and mechanical losses. If the power requirements are only in the watts range then photovoltaic systems will be more feasible.

(ii) Medium range units or KW range units. Here generally dispersed or distributed solar system is used and can be feasible. Photovoltaic systems may compete under certain situations.

(iii) Large range units of MW capacity where the central receiver solar system is preferred.

In the recent past, successful demonstrations of solar thermal power systems in the above three ranges i.e. watts, KW and MW have been made and many technical and operational problems solved, however, a suitable storage device which still remains an unsolved technical problem, particularly for large units.

In this chapter we briefly describe the history of solar power units, power cycles and different technical limitations in power production, types of heat engines and turbines, examples of dispersed and central power systems, etc. The components like solar collectors, storage systems, etc. will not be discussed in this chapter as these are discussed in other chapters.

4.2 HISTORY OF SOLAR THERMAL POWER

Several surveys have recently been made on the use of solar energy for generating power[1,2] or for lift irrigation[3]. This historical review is due to Pytlinski[3]. Although solar energy has been used for variety of applications since time immemorial but the first successful application of solar energy for raising water was made by a French engineer[4] Solomon de Caux (1576-1625) who through the expansion of air using solar energy was able to pump water. Over 200 years later (1854-1873), G.Guntner of Austria[5] used 18.6 m^2 of narrow mirror strips to generate steam for producing 746 watts. August Mouchot[6], a Frenchman, during the years (1860-1878) built several solar engines, the first solar engine was made in 1866, the second in 1875 and the third in 1878.

In the second solar engine, Mouchot used a 4 m^2 truncated reflecting cone which produced steam at 507 KPa at 154°C and used alongwith it a rotary engine driving a water pump. An American, John Ericsson[7] (1803 - 1889) worked for 20 years on solar engines and developed several solar steam power engines and solar hot air eninges. In his experiments, he used parabolic troughs of various dimensions, and in one of the experiment at New York, he used a parabolic trough 3.3 m long and 4.9 m wide which produced steam at a pressure of 240 KPa driving a reciproca-ting engine and generated 1.2 KW. In 1876 W. Adams[8] in Bombay, India operated the first solar steam engine in India where the solar energy was collected using a spherical mirror of 12.2 m diameter and the system was able to produce steam at a pressure of 207 KPa producing a power of about 1.9 KW. Abel Pifre[9] of France in 1880 used a parabolic reflector of 9.3 m^2 area to power his rotary pump raising about 100 litres of water through 3 m head in 14 minutes.

C.L.A.Tellier[10] of France in 1885 used 20 m^2 flat plate collectors (metal roof top collectors) and produced less than one horse power using a small vertical engine with ammonia as a working fluid. According to a report published in Scientific American[11] an American, A.G. Eneas (1901 to 1905) built several solar power plants using truncated cone shaped reflectors and water as a working fluid and a compound condensing engine connected to a centrifugal pump. In one of the demonstrations in California in 1901, he used a turncated cone of 60 m^2 of collecting surface and produced steam at a pressure of 1000 KPa and pumped water at a rate of 5.3 m^3/min through a 3.6 m head developing about 7.6 KW.

During the years 1902-1908, H.R.Willsie[12] and John Boyle, Jr., built several solar engines. Probably they were the first to use two-fluid system in solar engines. They used flat plate collectors with double glazing and low temperature working fluid such as ammonia, ether or sulphur dioxide. In 1904, they developed a 4.5 KW sun powered plant at St.Louis in which they have used a double glazed flat plate collector of 55.7 m^2 area using an ammonia engine. In 1905, they built a solar engine of 15 KW capacity using sulphur dioxide as working fluid near Needles, California. Another solar engine of 1.5 KW capacity was built by them in 1908 at Needles which used 9.3 m^2 flat plate collectors. An American Frank Shuman[13] in 1907 developed a solar pump in which 1200 m^2 flat plate collectors were used to evaporate ether from water and the vapours were used to drive a vertical vapour engine of 2.6 KW at Tacony, Pennsylvania. In 1911 Frank Shuman developed[14] a 24 KW solar engine in which double glazed flat plate collectors with plane booster reflectors with a total collecting area of 956.5 m^2 were used. This system pumped 11.3 m^3 of water per minute to a height of 10 m. One of the most successful and largest solar pump was operated at Meadi, Egypt by F. Shuman and C. Boys[15] in 1913. In this system parabolic troughs with a total area of 1263 m^2 were used, which generated steam and inturn developed 37.3 KW power. This pump was used for pumping irrigation water from the Nile river but was later dismantled. During the years 1919 - 1934, R.H.Goddard published several papers related to solar power production. In 1919 Goddard described[16] one of the largest solar dish electric power plant. J.J.Harrington in New Mexico[17] in 1920 developed a solar steam engine and pumped 8.9 m^3 of water through a height of 6 m.

C.G.Abbot[18] in 1936 contributed greatly towards solar power generation and used equatorially mounted parabolic trough reflectors alongwith an ingenious single tube solar flash boiler which produced steam within five minutes of sun's exposure. The system had produced saturated steam at 647 K and ran a steam engine of 370 watts. In 1940, F.Molero[19] of the Heliotechnical Laboratory at Tashkent, USSR used a 10 m. diameter parabolic dish for generating steam at a pressure of 203 KPa which was used for pumping water for irrigation and livestock watering. M.L.Ghai and M.L.Khanna[20] of New Delhi, India during 1951-1955 used a small reciprocating-piston, hot air engine and operated with small paraboloidal solar reflector at about 800 K and produced power between 100 to 125 watts. This was used for demonstration purposes and pumped water from a depth of 5.0 m.

In 1961, Tabor and Bronicki[21] in Israel described a 4 KW solar turbine using a binary Rankine cycle with monochlorobenzine as a working fluid. These turbines were later modified and known as ORMAT Rankine power units[22] and are now available in the power range of 100 watts to 15 KW. These turbines are now in use in many parts of the world.

In Dakar, Senegal, Masson and Girardier[23] during 1962 to 1966 developed two solar motors. The solar motor operating since 1962 lifted 8 to 10 litres water per minute from a depth of 13 metres which is equivalent to 21 watts. In another experiment in 1966 flat plate collectors of 300 m^2 were used and the solar engine pumped water at the rate of 40 m^3/hr from a depth of 8 to 10 m.

The first photovaltaic power plant of 0.25 KW using concentrated solar radiation on photovoltaic cells was installed in September 1964 near Gelendzhik, USSR[24]. The total area of the silicon solar cells was 3.6 m^2 and the panels could rotate equatorially. Plane mirror boosters were used to concentrate solar radiation on solar cells.

During the years 1974 to 1980, a French company, Societe Francaise d'Etudes Thermiques et d' Energie Solaire, SOFRETES[25,26] installed several solar irrigation pumps in many countries of the world. The first solar pump[1975] of 25 KW capacity using 1500 m^2 flat plate collectors was installed in San Luis de la Paz, Mexico which had given an output of 2600 m^3/day of water from a head of 10 m. Later a 50 KW solar pump was also developed by this firm. Here water is heated through flat plate collectors which in turn boils the organic liquid through heat exchanger such as butane or Freon operating a Rankine cycle reciprocating engine or turbine.

A solar irrigation pump of 37.3 KW capacity was installed[27] in April 1977 at Gila Bend Ranch southwest of Phoenix, Arizona which was designed and built by Columbus laboratories of Battelle Memorial Institute, USA and Northwest Mutual Life Insurance Company of Milwaukee. The system consists of parabolic tracking solar collectors (564 m^2), a Rankine cycle power unit comprised of a turbine, boiler, condenser,regenerator, and preheater; and a low-lift, high volume flow propeller pump. The working fluid used is Freon 113 and the pump was able to pump about 37 m^3

of water per minute at peak operation. The concentrating collectors were manufactured by Hexcel corporation, Casa Grande, Arizona and the Freon turbine by Barber-Nichols Engineering Company, Arvada, Colorado.

In April 1977, a 18.7 KW solar pump was designed and tested by Barber-Nichols Engineering Company, of Colorado [28,29] with the design specifications supplied by Sandia Laboratories and the solar pump delivered 3.3 m^3/min of water from a well 23 m deep at Willard, New Mexico. The system consists of field of tracking parabolic trough collectors (625 m^2) supplied by Acurex Solar Corporation, a thermocline thermal storage system (22.7 m^3) and a Rankine cycle freon turbine. The Freon turbine runs at 36300 rpm and developed 1760 rpm at the output shaft of the gearbox.

A 7.5 KW solar photovoltaic solar pump was put into operation in August 1977 near Mead, Nebraska[30] which was the joint effort of MIT/Lincoln Laboratory and the University of Nebraska and sponsored by ERDA. The photovoltaic panel consists of 120000 individual silicon solar cells with a peak power rating of 25 KW and driving a 7.5 KW pump. Power from the photovoltaic panels is stored in large lead acid batteries capable of storing 85 KW. This pump can pump 3.8 m^3/min. water from a reservoir for 12 hours a day.

Under a joint programme of Government of India and Government of West Germany a 10 KW solar thermal power station was built[31] and operated at Madras, India in 1978. This experimental power station uses flat plate collectors(756 m^2) with mirror boosters(756 m^2) storage unit of 35 m^3, vapour generator, a freon-114 operated screw expander, and an alternator generating 440 volts, 3 phase, 50 Hz AC power. The system generated 10 KW net at peak load and the designed output per day is 35 KWH on a standard day with 1000 watts/m^2 peak insolation at noon.

A solar power generation[32-34] unit of 150 KWe with a technical support from Sandia Laboratory, Arizona Solar Energy Commission and University of Arizona in Oct. 1979 on the Dalton Cole farm south-west of Coolidge in Central Arizona. The plant was completely designed by Acurex Corporation, California and used 2140.5 m^2 line-focus collector subsystem, a storage of 114 m^3 tank of hot oil, and an organic Rankine cycle turbine engine built by Sundstrand Corporation. This plant was operated for three years by the University of Arizona to characterize energy performance, identify needed equipment improvements and quantify operating and maintenance requirements. This plant was later deeded to the owner of the farm on which it is sited, Dalton Cole, Jr.

Under the National Sunshine Project of Japan, a 1000 KW solar thermal electric power plant[35] was installed in March 1981 at Nio-cho in Kagawa prefecture of Shikoku. In this system a combination of plane mirrors and cyclindrical parabolic mirrors (hybrid mirror system) with a total mirror area of 11160 m^2, a steam accumulator molten salt heat storage (KCl-LiCl), and a Rankine cycle-steam turbine is used.

A 100 KW solar thermal electric power station is in operation in Sulaibyh, Kuwait[36] since June 1981 which is a joint Venture of Kuwait Institute for Scientific Research (KSIR), Kuwait, and Messers-chmitt-Bolkow-Blohm (MBB) of Germany. The solar power station consists of 56 point focussing parabolic dishes, each of 5 m diameter with total collector area of 1025 m^2, a thermal storage tank and an organic Rankine cycle engine using toluene as the working fluid and provides both electric and thermal energy for heating and cooling of greenhouses, desalination, irrigation, and pumping of brackish water.

On July 31, 1981, a 500 KWe solar power plant under the auspices of the International Energy Agency (IEA) by eight countries (Germany, USA, Spain, Greece, Switzerland, Sweden, Belgium, Austria) was made operational in the remote village of Tabernas, Almeria, Spain[37,38]. The operating agent for this project was DFVLR (space Agency) of Germany and the design, construction and commissioning was done by a consortium consisting of Tecnicas Reunidas, S.A. of Madrid, Spain; Acurex Solar Corporation of California, USA; and MAN-Neue Technology of Munich, Germany. The solar-thermal electric generation plant consists of two sets of linear parabolic collectors, one set single axis tracking type supplied by Acurex Solar Corporation of 2674 m^2 area and second set two-axis tracking type supplied bb MAN of 2688 m^2, a thermal storage of 0.8 MWhe capacity with Santotherm 55-a synthetic heat transfer oil; an oil-to-water Baeltz steam generator, and a Stal-Laval condensing steam turbine-generator.

In Shenandoah[39], Georgia, 40 of south of Atlanta, a 400 KWe parabolic dish power plant started in April 1982 to supply electricity to the Georgia Power plant started in April 1982 to supply electricity to the Georgia Power Company grid and at the same time cogenerate electricity and process steam to a nearby garment factory operated by Bleyle of America, Inc. It employs 114 parabolic dishes each of 7m diameter with a total collecting area of 4330 m^2, a heat exchanger producing superheated steam, a thermocline heat storage tank filled with silicon heat transfer fluid, a steam turbine, and an alternator. The overall efficiency, solar to electrical, is about 10 per cent.

Australian National University, Canberra in March 1982 designed and built a 25 KWe and 140 KW low quality heat solar power station at White Cliffs, New South Wales, Australia[40]. The power station comprises 14 modular semi-autonomous parabolodial tracking collectors, each of 5 m diameter with a total collecting area of 277 m^2, generating steam upto 550 °C of 7 MPa; and a reciprocating steam engine which powers an alternator to produce 25 KW of 240 volt electricity and the exhaust steam used for desalination, with minor amounts for hot water and for space heating and cooling.

A 350 KW photovoltaic power system[41] was installed near the villages of Al Jubay-Lah, Al Uyayanah, and Al Hijra, which are about 50 Km northwest of Riyadh, Saudi Arabia in 1982 sponsored by the Saudi Arabian-United States Program for cooperation. The design and fabrication of the system was largely carried out by Martin Marietta Corporation, Denver, Colorado. The photovoltaic system consists of 160 single pedestal concentrator arrays, each array containing 260 circular solar cells of 5.71 cm diameter and providing 2.2 KW of power under optical concentration of 33 suns. This has automatic 2-axis tracking arrangement. The battery subsystem consists of four batteries and each battery has 116 commercial lead acid cells of 1600 Ah each in series. The inverter converts 215/300 VDC power to 277/480 V AC, 3-phase 60 Hz AC power.

A hybrid solar/diesel power station of 100 KWe using parabolic troughs with a total aperture area of 920 m^2 was built at Meekatharra, Australia[42] in April 1982. Thermal oil is used to collect and store the heat and through a heat exchanger generate steam which is fed into the two stage screw expansion engine which drives an electric generator.

Photovoltaic power plant[43] of 1-MWe capacity was installed in December 1982, in California's high desert 160 KM northwest of Los Angles which contains 900000 single crystal solar cells mounted on computer controlled two-axis trackers. One-megawatt inverter system converts the $\pm$300 VDC from the field to 12 KV AC, for delivery to the Edison grid near the site. The photovoltaic power station is commercial, unmanned and fully automated. ARCO Solar Industries of USA has also announced a 6-KW photovoltaic plant to be installed on the Carrisa Plain east of Bakersfiled, California and will be operational by March 1984. Simple reflector units on the trackers and high efficiency solar cells are supposed to be used. The Sacramento Municipal Utility District (SMUC), USA has announced a 100-MW photovoltaic plant to be installed near Ranch Seco Nuclear power plant to be completed in 10 stages.

Non-convective solar ponds can be used as an effective means for collecting and storage of solar heat and can be used for power production. Israel[44] is the first to develop a solar pond power station. A 6 KWe solar pond power facility was created in 1978 at Yavne, Israel[1] in which 1500 m^2 density gradient solar pond alongwith a low temperature turbogenerator was used. Another solar pond power plant was made in December 1979 in Ein Bokek near Dead

Sea, Israel, in which 7500 m^2 solar pond with a depth of 2.5 m was used where the bottom temperature reached 93 ℃. With the success of Ein Bokek power plant, Israel has made a ambitious plant as follows:

- o A 5-MW power plant to be operational by 1982 using 0.25 Km^2 solar pond.
- o By 1983 the above pond will be expanded to 1 Km^2 providing 20 MW power and additional construction of 100000 m^2 solar pond.
- o An additional 20 MW power plant and 1-Km^2 solar lake (peaking/intermediate) to be completed by 1985.
- o Construction of 50 MW unit and 4 Km^2 solar pond to be completed by 1986.

One of the most promising methods of collecting solar energy and converting into electricity is the central receiver system, in which a field of large individual controlled mirrors (heliostats) reflect solar radiation on a receiver at the top of a tall tower, heating a fluid at very high temperature, producing steam and driving the conventional steam engine or turbine. The concept of central receiver system was first given (1957) by V.A.Baum, R.R.Aparsi and B.A.Garf in USSR[45] in which heliostats are moving on circular railroad cars and reflecting solar radiation on an elevated cavity receiver boiler. The cavity was to be rotated to face the heliostats throughout the day to achieve better performance. Later Francia[46] in 1967 built a pilot model of central receiver system at the University of Genoa, Italy in which a clock driven field of 271 heliostats were used reflecting the solar radiation on a receiver at the top of the tower producing useful steam at temperatures upto 650 °C at a rate equivalent to 150 KWt.

An advanced components test facility (ACTF) of 400 KW capacity is operated by the Georgia Institute of Technology [47], Engineering Experiment Station for the US Department of Energy since September 1977. The ACTF consists of a 550 tracking heliostat field each of 111 cm in diameter with total area of 532 m^2, a tower 21.3 m tall located in the centre of the mirror field, instrument and control building, a computerized data collection system, and a heat rejection system. The heliostats are electrically driven by their mechanical supports without feed back which is a unique facility.

A 5-MW solar thermal test facility[48][STTF] for testing receivers of solar tower system is constructed in October 1978 by US Department of Energy about 32.2 Km southeast of downtown Alburquerque, New Mexico, and operated by Sandia Laboratories. The STTF uses 222 heliostats with a total reflector area of 8250 m^2. The tower provides test platforms at 36.6, 42.7, 48.8 m on north face and 61 m at the top. The operating conditions for the receiver are 516 ℃, 10.4 MPa, and 1.5 Kg/s with feed water being supplied to the receiver at 288 °C.

A 2.5 MWe central receiver system of the THEMIS project is in operation at Targasonne in Southern France since March 1981. Solar radiation is focussed on a cavity type receiver on a 100 m tall concrete tower by 200 heliostats each with a mirrored surface area of 54 m^2. Energy absorbed in the receiver heats molten salts (sodium and potassium nitrates and nitrites) to 450 °C from where it is piped to ground level where it generates steam which drives the conventional steam turbine. This work was undertaken by the French National Centre of Solar Tests, which is funded by commissariat a l' 'Energie Solaire (a government organisation) and Electricite de France (the French national utility).

Eurelios is a central receiver plant located in Sicily, Italy[49], sponsored by the commission of European Communities and producing 1 MW since May 1981. A European Consortium of Italian, French, and German Companies designed and built the plant. Eurelios uses two type of heliostats with a total area of 6216 m^2 with total 182 heliostats. 70 heliostats each of 52 m^2 are made by Cethel and 112 heliostats each of 23 m^2 are made by MBB (Messerschmitt Boelkhow-Blohm). These heliostats are arranged in subfields beneath the cavity type receiver, which is mounted on a 55 m high tower. Steam exiting the receiver, at 512 °C enters the steam turbine without going through an intermediate heat exchanger. Hitec salt is used as a heat storage material which can provide 30 minutes of energy to smooth out cloud transients.

A central receiver electrical generating plant of 1.0 MWe capacity is located on the island of Shikoku[50], [Nio] Japan and is generating electricity since August, 1981. The plant uses 807 heliostats each with a reflecting area of 16 m^2, surrounding a conical-cavity receiver and steam drum on top of a 69 m high tower. The receiver produces steam at roughly 250 °C and 4 MPa and enters the turbine generator at a temperature of 187 °C and at flow rate of 7940 Kg/hr. The energy is stored in tanks containing pressured water which is equivalent of 1 MWe for 3 hrs.

A 500 KWe central receiver system is in operation at Almeria[51], Spain since September, 1981 (SSPS) which is sponsored by nine member countries of the international Energy Agency (Germany, Spain, Italy, Austria, Sweden, Belgium, Switzerland, Greece and USA). The system employs a field of 93 heliostats having a total reflecting area of 3660 m^2, a cavity receiver using sodium as the heat transfer fluid at an operating temperature of 530 °C, a steam driven piston engine coupled to a three-phase-current generator and a hot tank/cold tank sodium storage system.

The solar-one is the world's largest solar powered central receiver plant[52] which is of 10 MWe capacity and is in operation since April 1982 near Barstow, California, USA. This pilot solar facility is sponsored by US Department of Energy and cooperating agencies are: Southern California Edison Company, the Los Angles Department of Water and Power, and the California Energy Commission. The collector field consists of 1818 Martin Marietta sun tracking heliostats with a total reflecting area of 72538 m^2. An external type receiver is used at the top of 90.8 m tall steel tower and water-steam as the heat transport fluid. Energy is stored in one tank of 13.7 m high and 19.8 m diameter containing heat transfer oil and 6798 tons of rocks. Stored hot oil is used to generate steam at 274 °C and 2.7 MPa which drives the conventional steam turbine.

A 1.2 MWe central receiver plant (CESA-1) funded by United States and Spanish Joint Committee for Scientific Technological Corporation and built by Centro de Estudios de la Energia (CEE) of Spain and working since June 1983. It uses 300 heliostats each of 38 m^2 and a cavity receiver on a 60 m tall concrete tower. The energy is stored in 300 tons of moten salt. The receiver uses water as the working fluid and steam at 520 °C drives a Rankine Cycle turbine engine.

A central receiver plant known as 20 MWe Gasgekuhltes Sonnenturn-Kraftwerk GAST' is built[53] in 1983 in Germany by a German industrial Consotium and sponsored by the German Government (BMFT). The GAST pilot plant employs 3000 heliostats each of reflecting area 40 m^2 concentrating solar radiation on to two receiver modules mounted on top of tower at a height of 200 m. The heat transfer medium is air which is heated to 800 °C. A combined gas/steam thermal energy conversion process using two open-cycle gas turbine generator sets and one steam turbine generator set is employed. The plant does not employ any storage system but equiped with an auxiliary fossil fuel-fired system.

A central receiver electric plant of 30 MWe is planned [54] by a team drawn from Rockwell International's Energy System Industries,the Pacific Gas and Electric Company, and ARCO Solar Industries and the same will be operational by the end of 1986 at the Carrizo plain in Central California, USA. The collector field will consist of 1904 advanced third generation ARCO heliostats each of 95 m^2 reflective area. These heliostats reflect the solar radiation on to the receiver at the top of 122 m tall tower. Sodium heated to more than 600 °C would be piped to ground level, where it would generate steam to drive a turbine-generator. The warm storage tank will contain about 1419420 litres of sodium.

It is reported that USSR is also planning to build a 5 MWe central receiver plant in the Crimea but its details are not available.

4.3 PRINCIPLES OF SOLAR ENGINES

The heat energy can be converted into mechanical power through any of the energy conversion cycles: the Ranking cycle, the Stirling cycle, and the Brayton cycle. The heat engine based on any of the above cycle is a thermodynamic device operating between a high temperature heat source and a low temperature heat sink, extracting some of the thermodynamic heat energy from the working fluid and converting it into shaft power. Only a small portion of heat energy added to the cycle can be converted into mechanical power. If Q_H heat is added at temperature T_H and after performing the work the fluid rejects Q_C heat at a low temperature T_C then the Carnot efficiency, η_{carnot}, is obtained by dividing the difference between heat input and rejected heat by the heat input i.e.

$$\eta_{carnot} = \frac{Q_H - Q_C}{Q_H} \tag{4.1}$$

This concept was developed by a Frenchman, Sadi Carnot and is named as Carnot efficiency. The Carnot efficiency η_{carnot} is also defined as the ratio of temperature (absolute) difference between the source T_H and sink temperature T_C divided by the source temperature T_H i.e.

$$\eta_{carnot} = \frac{T_H - T_C}{T_H} \tag{4.2}$$

From this expression it is seen that: (1) the engine efficiency increases as the source temperature increases and the rate of increase is greater at low temperature than at high temperature; (2) the engine efficiency increases with the decrease in sink temperature at a faster rate than if the source temperature is increased by the same amount; (3) if the sink temperature reaches to 0 °K the carnot cycle efficiency reaches to 100 per cent which is an impossible situation according to the third law of thermodynamics. For a given temperature range the Carnot efficiency is the maximum possible cycle efficiency and is limited by the second law of thermodynamics.

The schematics of heat engine cycles and their corresponding P-V diagrams are shown in figure 4.3. In the Carnot cycle as shown in figure 4.3 (a), there are two reversible isothermal processes at temperatures T_H and T_C respectively, connected by two reversible adiabatic processes. When the working fluid is a condensable vapour, the two isothermal processes are easily obtained by heating and cooling at constant pressure while the fluid is a wet vapour. Saturated water in State 1 is evaporated in the boiler at constant pressure to form steam in State 2. The steam is expanded adiabatically to State 3 while doing work

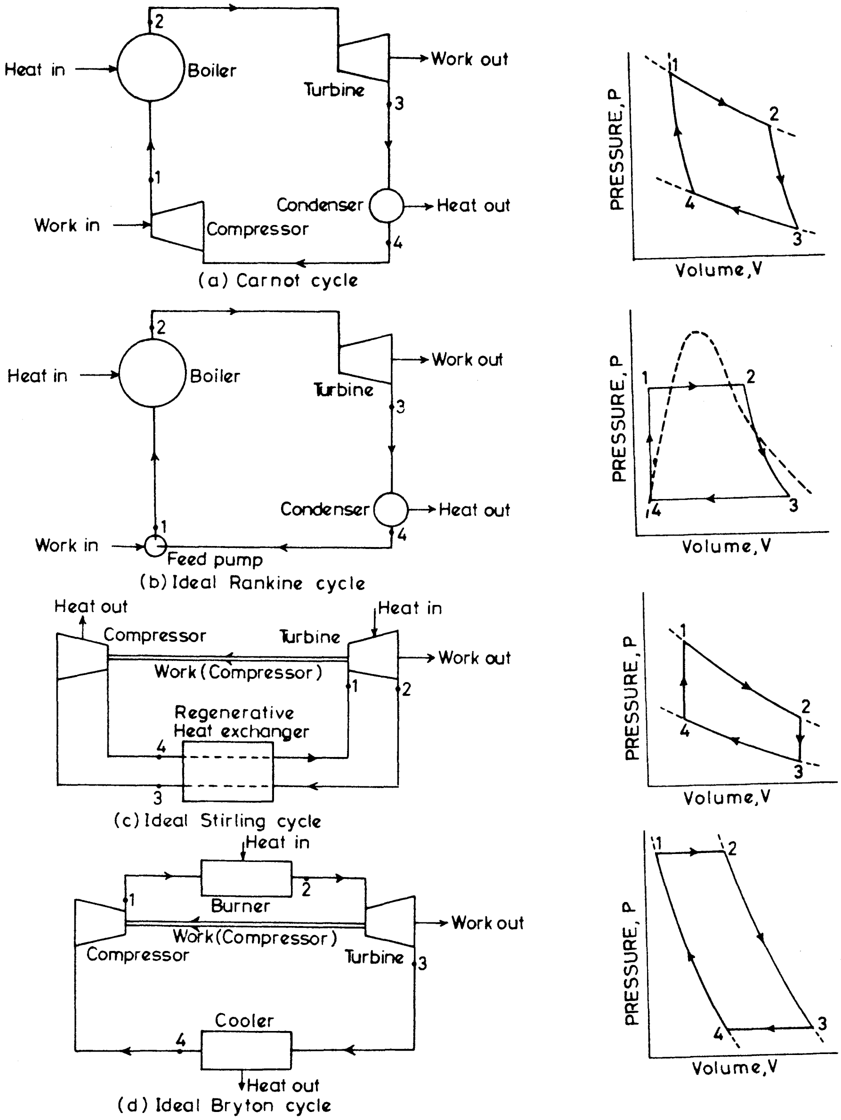


FIG.4.3 SCHEMATICS AND P-V DIAGRAMS OF HEAT ENGINE CYCLES

in a turbine or reciprocating engine. After expansion the steam is partially condensed at constant pressure and the heat is rejected. The condensation is stopped at Stage 4. Finally the steam is compressed adiabatically in a rotary or reciprocating compressor to State 1. The efficiency of a practical heat engine is always lower than the Carnot efficiency due to many reasons such as heat losses through insulation, friction between moving parts, compressor not working at 100 per cent efficiency, the expansion device not working at 100 per cent efficiency, problems with the actual working fluids, etc. If the engines are carefully designed then efficiency can be about 50 to 80 per cent of the Carnot efficiency depending on the temperature difference. The actual cycle efficiency obtained with flat plate collectors working at a temperature difference of about 90 °C is about 8 to 10 per cent while with concentrating collectors operating at a temperature different of 400 °C the cycle efficiency is about 20 to 25 per cent.

The ideal Rankine cycle is shown in figure 4.3 (b). It differs from the Carnot cycle in that the heat-addition process does not occur at constant temperature. In an ideal Rankine cycle the working fluid is heated in the boiler, and the vapour so produced is expanded in the turbine to do mechanical work. The exhaust from the turbine consists of mixture of vapour and liquid droplets at a much lower temperature and pressure than at the inlet of the turbine. Exhaust vapour is liquified in the condenser rejecting the heat. This liquid residue is pumped up by a feed pump to high pressure and fed to the boiler to complete the cycle. It can easily be shown that the efficiency of Rankine cycle is less than the Carnot cycle operating between the same temperatures, because all the heat supplied is not transferred at the upper temperature. Inspite of the low efficiency of a Rankine cycle it has high work ratio and the steam consumption is less compared to Carnot cycle. Thus the size of the Rankine cycle can be increased by superheating the vapour or by partially expanding it and then reheating it several times. The efficiency can also be increased by using a part of rejected heat in heating the liquid by using a regenerator before the liquid enters the boiler. Some improvements can also be made by using more than one working fluid. In this case the heat rejected by a high temperature cycle is used as input to a cycle using a low boiling point fluid.

The Stirling cycle is similar to a Carnot cycle except that two adiabatic steps are replaced by two constant volume steps[(figure 4.3 (c)]. Here some suitable gas or air is used as working fluid and the turbine, compressor and heat exchanger are very closely coupled in a single housing and not as shown in figure 4.3 (c). Here the heat addition and rejection takes place at constant temperatures. The heat supplied during the process 4-1 is equal in quantity to the heat rejected during the process 2-3. Moreover, the temperature of the working fluid varies between the same limits during these two processes. It is therefore theoretically possible that the heat rejected is returned to the working fluid. This heat transfer is accomplished reversibly in a regenerator consisting of a matrix of wire gauze or small tubes. During the passage of the fluid through the regenerator, the volume of the working fluid does not change in either of the direction. The Stirling cycle has higher efficiency than the Rankine cycle between the same two temperatures because in the Stirling cycle heat is delivered at high temperature and rejected at the low temperature as in the Carnot cycle. The main difficulty with the Stirling cycle lies in making efficient regenerator of reasonable size which can operate at temperatures comparable to temperature used in internal combustion engines.

The Brayton cycle as shown in figure 4.3 (d) uses a gas or air as the working fluid and works at temperature well in excess of 500 °C. In the Brayton cycle the expansion and compresson processes are reversible and adiabatic and heat addition and rejection takes place at constant pressure. In the Brayton cycle the hot compressed gas is allowed to expand through a turbine producing work. The exhaust gas from the turbine is fed to the heat exchanger where the heat is rejected and then compressed by the compressor to complete the cycle. The Brayton cycle is less efficient compared to the Stirling cycle but larger power plants can be made using Brayton cycle, than those using a Stirling cycle. The performance of the Brayton cycle can be improved by inserting a regenerator between the turbine exhaust and the cooler for preheating the compressed gas prior to the heater. These power cycles are described in details in text books on thermodynamics[55] and in a paper by Howe[56].

The efficiency of different thermodynamic cycles can be compared on a temperature-entropy (T-S) diagram. The temperature entropy diagram of these cycles are shown in figure 4.4. It is seen from this figure that the Carnot cycle can be represented by a rectangle and its area will give the cycle efficiency. The areas within solid lines give the efficiency of corresponding cycles. It is seen from this figure that the Stirling cycle is better than the ideal Rankine cycle and also the ideal Brayton cycle. The main limitation as mentioned earlier with the Stirling cycle is the design of a suitable efficient regenerator. Moreover, in the larger power plants, Stirling cycle is not preferred due to many practical problems. In solar energy applications the Rankine cycle is generally preferred and widely used due to its superior overall cycle efficiency and component sizes. Net engine efficiency of various peak cycle temperatures for all the four power cycle is compared[57] in figure 4.5 which also indicate the superiority of the Stirling cycle.

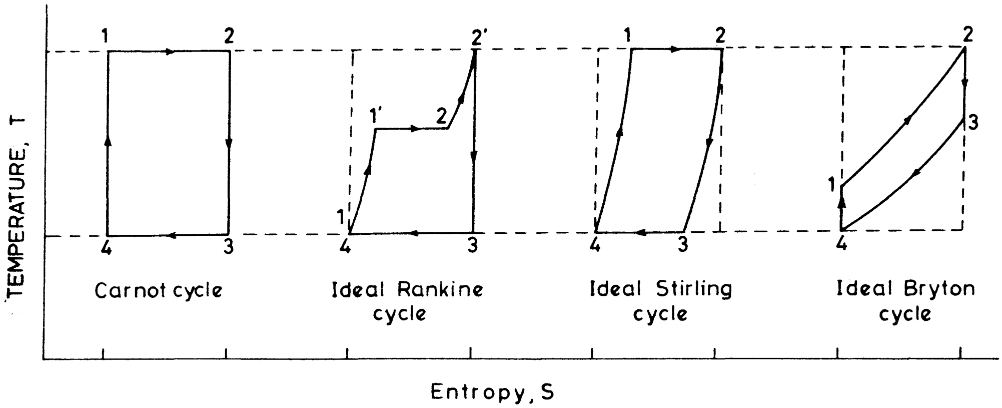


FIG.4.4 COMPARISON OF HEAT ENGINE CYCLES

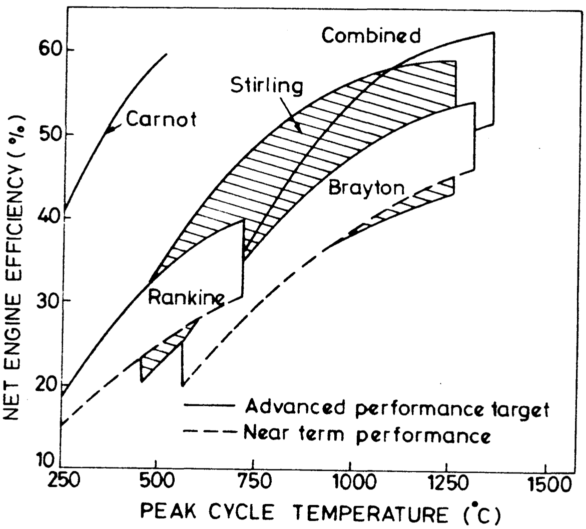


FIG.4.5 ADVANCED ENGINE PERFORMANCE POTENTIAL FOR RANKINE, STIRLING, BRAYTON AND COMBINED BRAYTON/RANKINE CYCLES (From Stearns et al[57])

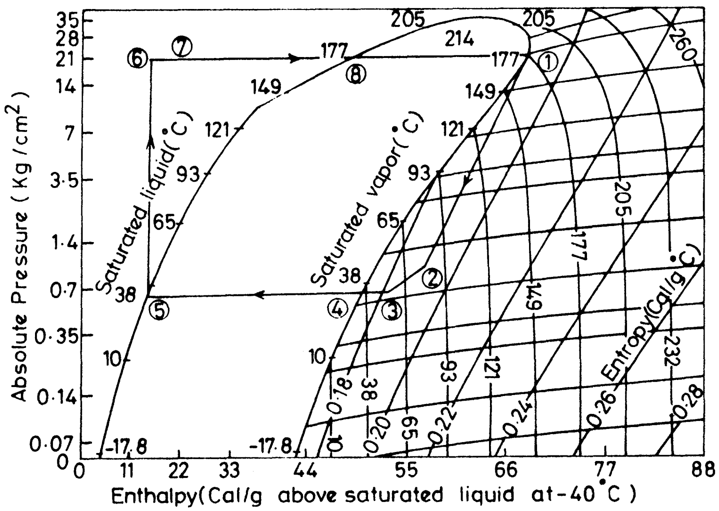


FIG.4.6 MOLLIER DIAGRAM FOR FREON-113

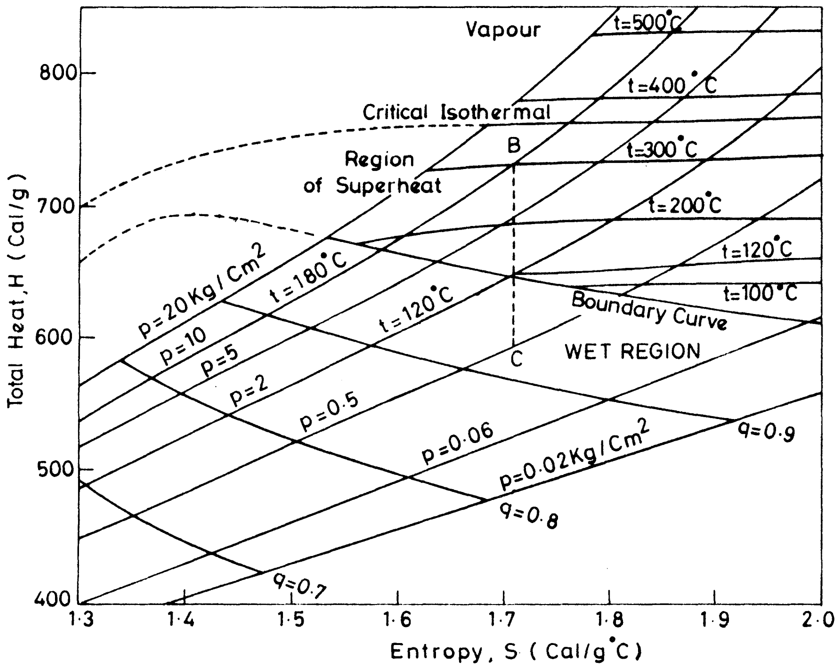


FIG.4.7 MOLLIER DIAGRAM FOR STEAM

A diagram between the enthalpy (H) and entropy (S) generally known as Mollier diagram for a working substance is used for analysing the performance of a cycle. The Mollier diagrams for a typical working fluid (refrigerant 113) and steam used in the Rankine cycle power systems are shown in figure 4.6 and 4.7 respectively.

4.4 IDEAL WORKING FLUID

The maximum output of any energy conversion device is limited by the second law of thermodynamics and depends on heat sink and heat source temperatures. The actual efficiency of any practical system is governed by the properties of the working fluid. The choice of working fluid depends on the operating temperatures in the boiler and condenser and the type of engine. Steam is the most widely used working fluid in heat engines greater than 1000 KW due to its low cost, high chemical stability, universal availability and better cycle efficiency. However, for smaller engines, due to some technical and other operational reasons, generally organic fluids are used as working fluids and are selected based on their physical and thermodynamic properties. The characteristics of an ideal working fluid are plotted as T-S diagram in figure 4.8 and its features are as follows:

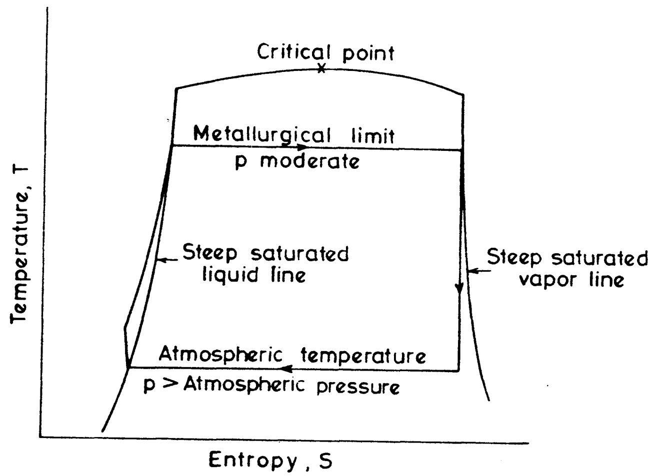


FIG.4.8 THE IDEAL FLUID FOR A VAPOUR POWER CYCLE

a) The saturated vapour line of the working fluid is steep so that the saturated vapours after expansion remain saturated. The dryness fraction after the expansion is maintained at about 0.9 without recourse to superheating.
b) The critical temperature is above the metallurgical limit. Superheating is therefore superfluous and most of the heat can be added at the upper temperature. The saturation pressure is also moderate at the metallurgical limit resulting in low maintenance and capital cost of the plant.
c) The heat of vapourization (latent heat) is high resulting in less mass of working fluid to be circulated. Thus for a given work output, the plant size and weight is reduced.
d) The specific heat of liquid is small and therefore the saturation line is steep. Therefore the heat required to bring the liquid to boiling point is small.
e) The saturation pressure at the atmospheric temperature is slightly higher than the atmospheric pressure. This avoids the use of vacuum in the condenser.

Apart from the above desired theremodynamic properties of the working fluid, it should be cheap, thermally and chemically stable, nonflammable, non-corrosive, non-toxic, small number of atoms in the molecule, and of high molecular weight. Properties of some of the working fluids are listed [58] in table 4.1

4.5 LIMITATIONS OF SOLAR MECHANICAL POWER CONVERSION

There are several limitations in the effective conversion of solar energy into mechanical power. Some of them are:

1. The main problem is that the efficiency of the collection system decreases as the collection (operating) temperature increases while the efficiency of the engine increases as the working fluid temperature increases.
2. The theoretical efficiency that can be attained by any heat engine operating between two temperatures is well understood and provide fixed fundamental barriers.
3. The heat engine used is not reversible and therefore efficiency obtained would be less than the maximum limiting theoretical efficiency.
4. A part of the heat is lost from the working fluid during its passage from collector or boiler.

Table 4.1 Properties of some working fluids used in solar heat engines (From Curran[58])

Fluid	Formula	Molecular weight	Critical point (°C/KPa)	Freezing point (°C)	Approx. thermal stability (°C)	Vapor pressure at 40°C (KPa)
1	2	3	4	5	6	7
Acetone	C_3H_6O	58	235	-95	-	52
Ammonia	NH_3	17	133/11429	-	-	1554
Butane	C_4H_{10}	58	152.6	-	-	410
Ether	$C_4H_{10}O$	74	194	-116	-	102
Ethyl alcohol	C_2H_6O	46	243	-117	-	20
Ethyl chloride	C_2H_5Cl	64	187	-	-	277
Freon - 11	CCl_3F	137	198/4413	-111	120	159
Freon - 12	CCl_2F_2	121	112/4114	-	-	958
Freon - 21	$CHCl_2F$	103	178/5102	-135	-	276
Freon - 22	$CHClF_2$	86	96/4978	-160	200	1455

Table 4.1 cont.

1	2	3	4	5	6	7
Freon -113	$C_2Cl_3F_3$	187	214/3441	-35	175	76
Freon -114	$C_2Cl_2F_4$	171	146/3261	-94	175	317
Freon -133	$C_2H_2ClF_3$	118	152/4068	-106	200	310
Monoisopropyl biphenyl	$C_5\ H_{16}$	196	522/2448	-55	370	-
Toluene	C_7H_8	92	321/4254	-95	480	7
Monochloro-benzene	C_6H_5Cl	113	359/4523	-55	320	7
Pyridine	C_5H_5N	79	347/5633	-42	370	7
Azeotrope of pyridine and water	$0.23C_5H_5N + 0.77H_2O$	33	366/8964	-18	400	14
Thiophene	C_4H_4S	84	307/5461	-40	290	21
Perfluoro-2-butyltetrahy-drofuran	$C_8F_{16}O$	416	227/1607	-62	320	7

Table 4.1 cont.

1	2	3	4	5	6	7
Perfluoro-pentene	C_5F_{12}	288	150/2131	-115	200	140
Fluorinol (trifluoro-ethanol/water mixture)	$0.85CF_3CH_2OH + 0.15\ H_2O$	88	240/6412	-	290	21
Dowtherm	$0.265\ (C_6H_5)_2 + 0.735\ (C_6H_5)_2O$	166	499/3241	-48	370	-
Biphenyl	$(C_6H_5)_2$	154	498/3289	69	370	-
P - 1 D	$C_{10}F_{22}O_2$	570	243/1186	-85	370	2
Methyl alcohol	CH_4O	32	240	-94	-	26
Steam	H_2O	18	374/22109	-9	-	8
Sulphur dioxide	SO_2	64	157/7873	-	-	740
Air	-	-	-141/3769	-	-	-

5. Due to the intermittant nature of the solar radiation some kind of thermal storage device is required to operate the heat engine continuously. Generally the heat storage materials degrade with time.
6. Like in many other fields the materials of construction of heat engine and suitable working fluid and their interaction cause a problem. The construction materials should withstand the high temperature and pressure.
7. In solar mechanical power generation both solar collectors and engines cause problems. Solar collectors are generally more expensive than the engines. Moreover, they require large areas for installation.

4.6 RANKINE CYCLE CHARACTERISTICS

As mentioned earlier the Rankine cycle solar power pumps are preferred because their efficiencies are little smaller to carnot cycle while the steam consumption is less and the work ratio is high. Moreover, in the temperature range of solar systems, 50-300 ℃, the Rankine cycle is superior to other cycles in terms of cycle efficiency. Following guidelines should be followed to convert efficiently the heat into the shaft power :

(i) Collect as much of heat as possible from the solar collector.
(ii) Supply heat to the engine at a temperature as high as possible.
(iii) Reject heat at a temperature as low as possible.
(iv) All the components like boiler, preheater, turbine, regenerator, condenser, feed pump, etc. should be designed for maximum efficiency.
(v) Minimize the pressure drop in all components and pipes.

A typical Rankine cycle power plant is schematically shown in figure 4.9 and its operation shown on pressure-enthalpy diagram of figure 4.6 for Refrigerant - 113. Two small electric pumps are shown in figure 4.9. One pump circulates hot water through the solar concentrators, boiler, preheater and back to solar concentrators. The second pump circulates the working fluid (Refrigerant-113) through various heat exchangers. As is seen from figure 4.9, the working fluid in the liquid condition, is pumped at high pressure by the feed pump through regenerator and preheater where it is partially heated and then to the boiler. In the boiler, the working fluid gets vapourized by hot water coming from solar concentrators. The high pressure vapour gets expanded through the turbine producing shaft power which can be used for generating electricity using an alter-

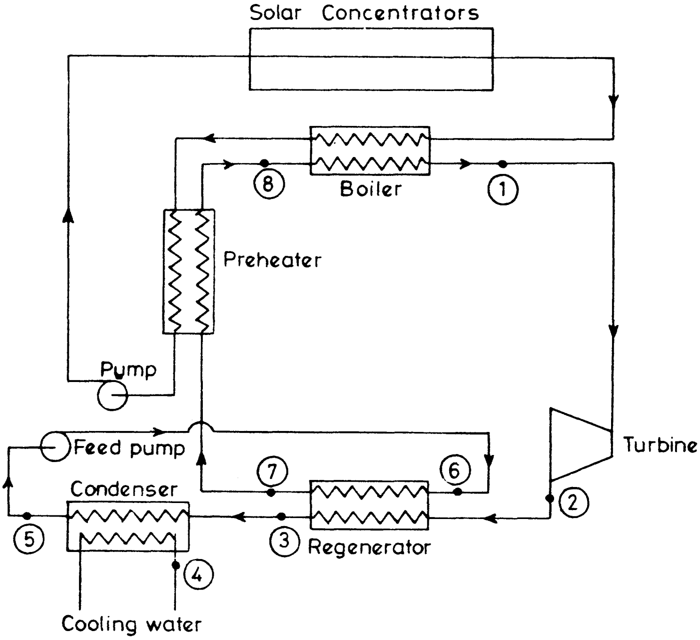


FIG.4.9 SCHEMATIC OF SOLAR POWERED RANKINE CYCLE

nator or for pumping water using a pump. The low pressure and low temperature vapour passes through the regenerator transfering some of its heat to the high pressure working fluid and then liquified through the condenser. The cycle points in 1 to 8 shown in figure 4.6 relate to the cycle points shown in figure 4.9. The cycle points from 1 to 2 on figure 4.6 show the expansion of fluid in the turbine. Points 2 to 3 show that some heat is transferred from the low pressure and low temperature vapour by the regenerator to the liquid fluid from point 6 to 7. The vapour gets condensed in the condenser by cooling water from point 3 to 5. Points 5 to 6 show that the feed pump raises the pressure of the liquid. The preheater raises the temperature of the liquid from point 7 to 8. The boiler converts the hot liquid into a vapour from point 8 to 1. The performance of a Rankine cycle can be determined if the thermodynamic fluid properties at various points are accurately known; efficiencies of pump, turbine, and other mechanical components and effectiveness of regenerator and heat exchanger are known:

and component pressure drops are known. In the literature several definitions of cycle efficiency are given. Here we define the cycle efficiency as the ratio of useful output to the heat added to the cycle from the source. The thermodynamic cycle efficiency is defined[59] as:

$$\eta_{Tc} = \frac{\text{work done by expander - work done by pump}}{\text{Heat added}}$$

$$= \frac{(H_1 - H_2) - (H_6 - H_5)}{(H_1 - H_7)} \tag{4.3}$$

where H is the enthalpy at a particular cycle point.

The cycle efficiency η_c is defined as :

$$\eta_c = \frac{\text{useful shaft power}}{\text{Heat added}}$$

$$= \frac{\text{(work done by expander - work done by pump - mechanical losses)}}{\text{Heat added}} \tag{4.4}$$

The solar collector area, A_c (m^2) for the maximum power output, P, can be calculated from the following equation:

$$Ac = \frac{P}{I \quad \eta\eta_{rc}} \tag{4.5}$$

where

P = power of the engine

I = incident solar radiation on collector aperture at solar noon on the design day (KW/m^2)

η = collector efficiency (instantaneous)

η_{rc} = thermal efficiency of Rankine cycle system

$$= (1 - (T_c / T_H)\eta_e \; \eta_r \tag{4.6}$$

η_e = engine efficiency, and

η_r = ratio of Rankine to carnot cycle efficiency

The cycle efficiencies for various flow rates, fluid properties, heat transfer requirements, etc. can be accurately obtained using computer program as described by Albin et al[60] and Badr et al[61].

4.7 SOLAR HEAT ENGINES

For converting solar energy into shaft power or mechanical energy any heat engine like single or multistage reciprocating steam engines, steam engines of rotary type (not successful), turbomachines (using steam or organic vapours as working fluid), Stirling hot air engines, Brayton engines can be employed [62,63]. The selection of a particular solar (heat) engine depends on many parameters including power requirement (size), type of working fluid, temperature of operation and solar energy collection device. For low power requirements (< 50 KW), reciprocating engines, rotary displacement engines and Stirling hot air engines potentially offer high efficiency. However, turbines are generally used because of lower cost and higher reliability. For larger power capacities, turbines (rotodynamic) are favoured due to several advantages such as:

1. Wear and tear and maintenance cost in case of turbine is low and therefore their reliability is high compared to reciprocating engines.
2. High thermal efficiency particularly in large sizes can be obtained. Efficiencies of around 80 percent has already been demonstrated while in the case of steam engines the thermal efficiency is of the order of 20 to 25 percent.
3. Turbines are most apropriate for use with generators.
4. High speed turbines upto 4000 rpm can be obtained thereby increasing the power output per unit volume of the working fluid, while speed more than 250 rpm in steam piston engines is not available.
5. Since no internal lubrication is required in case of turbine except on main bearings, there is no chance of mixing lubricant oil with the working fluid.
6. Very large capacity turbines upto 5 MW capacity have already been built while steam piston engines of this capacity are impossible to build and operate.
7. In turbines the motion in the form of rotating shaft is directly obtained and hence perfect balancing of the whole system is theoretically possible.
8. In case of turbines the condenser pressure can be at low vacuum, converting the energy of working fluid into useful work almost upto maximum.
9. The operation and regulation of turbine is easy.
10. There is a possibility of large expansion to condenser pressure.

There are some problems also with the turbines like small turbines are less efficient, costly, and the moisture content in the expanded vapour can result in the erosion of turbine blades.

As mentioned earlier both steam and organic vapours can be used to drive the turbine. The minimum vapour pressure of a working fluid required to drive a turbine is about 700 KPa, and the preferred pressure is around 2000 KPa for useful work. These pressures are possible with water (steam) when it is heated to a temperature of 150 to 200 °C. Sometimes such high temperatures are not possible with solar energy such as by flat plate collectors or simple low cost cocentrating collectors where temperatures around 100 °C can be obtained. In such cases, turbines, alongwith a suitable organic working fluid can be employed and therefore water becomes completly inappropriate. However, there are disadvantages of using organic fluids such as some are expensive, poisonous, flammable, and leakage can be a great problem. Moreover, large parasitic pumping energy is required due to high amount of organic fluid is required for pumping to produce the same power output compared to a steam cycle. The energy requirement (parasitic energy) in case of organic fluid turbine can be 2 to 10 times compared to steam operated turbines.

4.7.1 STEAM ENGINES

In the past, several attempts[7,16,18] have been made to use reciprocating type steam engines for producing power using solar energy. Steam engines were earlier considered to be of great significance but their use in modern industries is declining with the advent of turbines due to their large capacity, better efficiency and superior economy. The 'reciprocating' type steam engine where the piston moves backward and forward in a stationary cylinder can be horizontal or vertical type depending upon whether the axis of reciprocation is horizontal or vertical. A single cylinder vertical steam engine showing various important components is schematically shown in figure 4.10. These engines can be compounded to provide for expansion in two or more stages. A theoretical steam engine cycle, 01234, is shown on P-V coordinate in figure 4.11. The steam enters the cylinder at constant pressure and the piston moves from position 0 to 1 (figure 4.11). Reversible adiabatic expansion takes place in process 1 to 2. At point 2 called the point of release, the pressure is instantaneously brought down to that of exhaust along line 2 to 3. On the return stroke 3 to 4, the expanded steam is discharged at constant pressure from the cylinder. The theoretical engine discussed is assumed to function with zero clearance i.e. V_0 and V_4 are equal.

The theoretical indicator diagram shown in figure 4.11

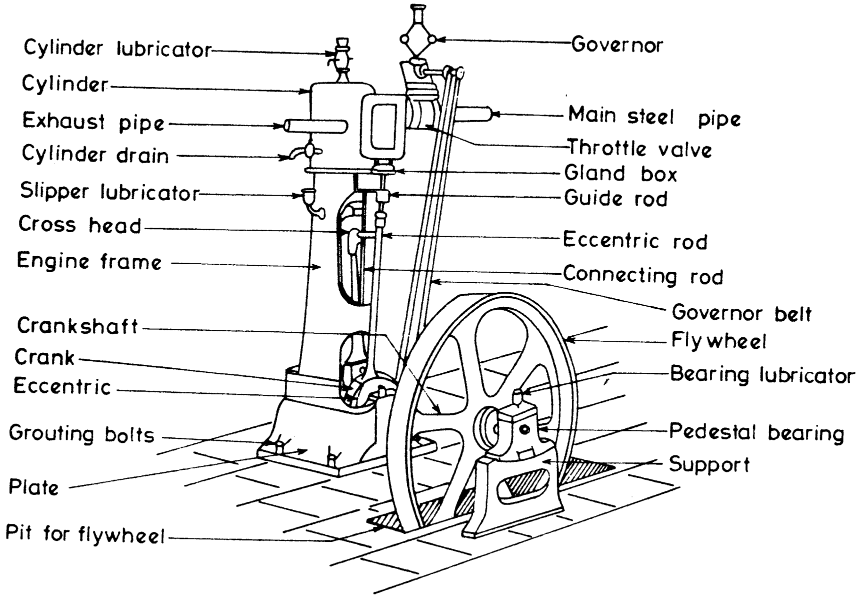


FIG.4.10 STEAM ENGINE (VERTICAL)

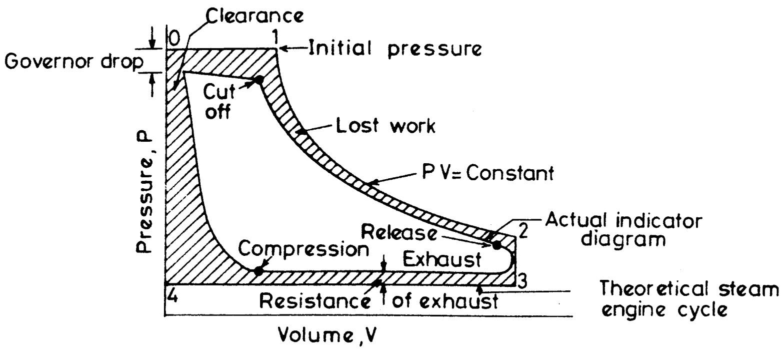


FIG.4.11 OPERATION (THEORETICAL AND ACTUAL) OF A SIMPLE STEAM ENGINE

assumes no loss. However, if actual P-V values are measured for a steam engine then we get an actual indicator diagrams as shown in figure 4.11 (unhatched). The difference between the theoretical and actual indicator diagram is attributed to a number of factors including heat loss, internal irreversibility, presence of clearance steam at the end of the exhaust stroke, and throttling through valve passage. The ratio of areas of the two diagrams is known as 'diagram factor' which is generally in the range of 0.6 to 0.9. This diagram reveals the following facts:-

1. Due to condensation caused by heat loss in pipes and wire-drawing through the admission port, the steam pressure drops considerably between the boiler and the engine cylinder.
2. Since the cylinder walls are at a lower temperature than the incoming steam, there is a gradual drop in steam pressure before the point of cut-off is reached.
3. Since the admission port does not close instantaneously there is a rounding-off of the diagram at cut-off.
4. The exhaust pressure is slightly above the condensor pressure as the steam has to be forced out of cylinder.
5. Release takes place before the end of the expansion stroke.
6. There is a rounding-off of the toe of the diagram because the exhaust port does not open instantaneously.
7. Admission occurs just before the end of compression of entrained steam.
8. Due to varying intercharge of heat between the steam and the cylinder walls, the expansion curve is not a true hyperbola.

The work done by the steam engine can thus be determined by evaluating the area of theoretical indicator diagram and then multiplying it with the diagram factor. The actual work done (W) by a steam engine can be calculated from the following expression :

Work done per minute = $f p_m LAN$ (Nm) (4.7)

where

f = diagram factor

L = stroke of piston, m

A = area of piston, m^2

N = number of strokes per minute (n for single acting and 2 n for double acting engine)

n = rpm

P_m = mean effective pressure on piston (N/cm^2)

= $(P_1/r)(1 + \log_e r) - P_3$ (4.8)

P_1 = absolute pressure of steam at entry to the steam engine (N/cm^2)

P_3 = absolute exhaust pressure of steam (N/cm^2)
r = expansion ratio, V_2/V_1

Small solar steam engines are not promising because they are expensive and inefficient while small internal combustion engines running on gasoline are widely used for pumping of water, lawn mowers, motor boats, automobiles and diesel power sets. The internal combustion engines are inexpensive, efficient and operated through out the world without much difficulties and technical expertise. Attempts have been made to convert the internal combustion engines

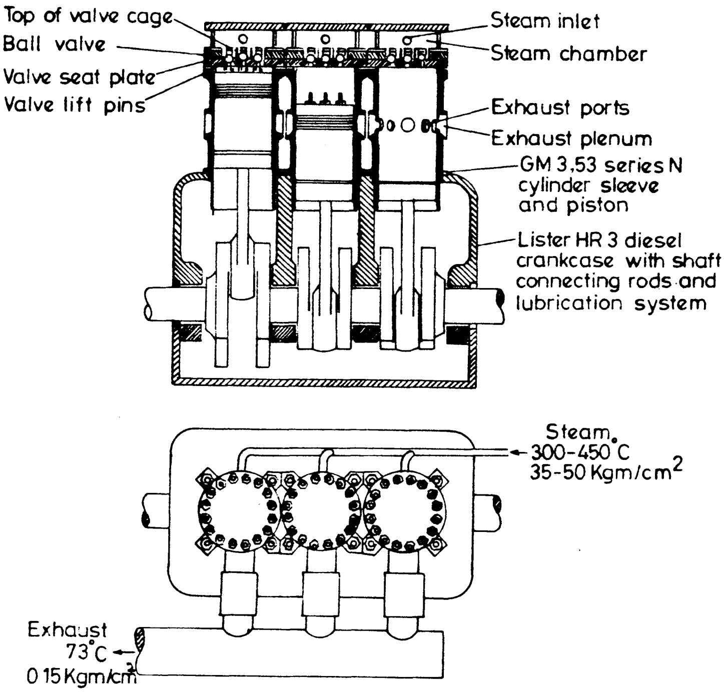


FIG.4.12 CULINDER PISTON AND VALVE-ARRANGEMENT IN THE STEAM CONVERTED DIESEL ENGINE(From Inall[65])

into steam engine with high efficiency[17]. Farber[64] converted a lawn-mower gasoline operated engine into a solar steam engine. Recently Inall[65] converted a diesel engine into a 25 KW solar steam engine which is used at the White

Cliffs Solar Power Station in Australia. The general forms of the engine is shown[65] in figure 4.12. This engine is made from parts of two diesel engines[65]. The engine starts by a standard electric motor and solar generated steam is supplied to a chamber in the head of each cylinder. As the piston reaches 15° before top dead centre the pins in its crown lift the three ball valves from their seats and steam enters the cylinder until the valves seat again at 15° past dead centre. The steam expands while pushing the piston until it reaches to the exhaust parts in the cylinder. The parts like crankcase, crankshaft, flywheels, connecting rods, starter, and sump are taken from a Lister Diesel Model HR3. The cylinder liners and piston are taken from GM diesel Model 53 and other parts like cylinders, cylinder heads, valve seats, and steam chambers are locally made. The specifications of the engine are as follows :

Bore	98.4 mm
Stroke	114.3 mm
No.of cylinders	3
Maximum steam pressure	70 Kg/cm^2 (abs)
Condenser pressure	0.25 Kg/cm^2 (abs)
Maximum stead temperature	450 °C
Expansion ratio (Adjustable)	≃ 1.25 (used)
Lubrication	as in Lister engine
Lubricant	Mobil oil XRN 1301C
Measured efficiency (steam pressure 4.2 Kg/cm^2, temp. 415 °C)	21.9 percent

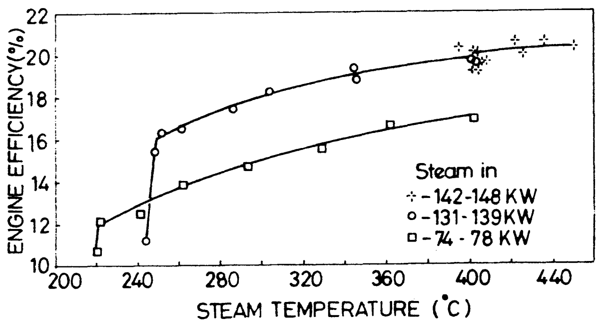


FIG.4.13 PERFORMANCE OF STEAM ENGINE AT THREE POWER LEVEL (From Inall[65])

The efficiency of this engine was measured[65] and the same is shown in figure 4.13 at different steam temperatures for three different power levels. A sharp fall in efficiency as is obvious is observed when the steam is wet. In the present system the solar concentrators were able to produce 170 KW of steam at more than 350 °C at solar insolation of 1KW/m².

Two conceptual design contracts of reciprocating steam engines suitable with solar energy were awarded by the NASA/Lewis Research Centre to Jay Carter Enterprises and Foster-Miller Associates of USA[66]. Jay Carter Enterpries analysed a reciprocating Rankine steam engine concept and developed a 3600 rpm reciprocation expander with maximum input thermal power of 80 KW with 677 °C and 17.2 MPa as maximum inlet temperature and pressure. This conceptual design dealt with two engine configurations :(1) a single cylinder design for simple cycle operation, and (2) cylinder design for reheat cycle operation, where the reheat cylinder contains a high pressure cylinder and a low pressure cylinder with steam being reheated to the initial inlet temperature after expansion in the high pressure cylinder. The maximum engine efficiency was estimated to be about 33 percent.

The Foster-Miller Associates analysed a reciprocating reheat steam engine for steam conditions of 700 °C as maximum steam inlet temperature, 12.1 MPa as maximum steam pressure. The conceptual design uses a two-cylinder/opposed engine operating at 1800 rpm directly coupled to a induction generator. In this design the use of carbon/graphite piston rings are proposed which eliminate the use of oil as an upper cylinder lubricant. The engine efficiency was estimated to be about 33 percent.

4.7.2 Turbines

Turbine is a machine which converts enthalpy to kinetic energy and then to mechanical work. Generally steam[67] or organic[68] vapour turbine both based on Rankine cycle can be used to produce mechanical work using solar energy. The basic principles of construction for these two types of turbine are generally similar, but owing to the difference in characteristics between steam and organic vapours, several design deviations exist. A turbine in its simple form consists of the following four parts:

1. A set of nozzles, where the working fluid (steam or organic vapours) expands to increase the kinetic energy.
2. A rotor assembly consisting of some form of blades (or buckets) on a rotating shaft.
3. The blades or vanes or buckets which can either be moving or fixed type. The fixed type vanes are fitted with the casing and meant to direct the working fluid towards the moving vanes. The moving

vanes are mounted on the rim of the rotor.

4. The casing which encases the rotor, vanes, shaft, bearings, glands, etc. and prevents the leakage of working fluid.

Turbines are basically of two types : impulse turbines and reaction turbines. But turbines are classified in several ways as follows:

1. By shaft or casing arrangement : It can be of a single casing, cross compound (two or more shafts not in line), tandem compound (two or more casings with the shaft coupled together in line).
2. By exhaust stages : It can be double flow or triple flow.
3. By class design : It can be on impulse or reaction principle.
4. By steam supply and exhaust conditions : It can be condensing or non-condensing type, automatic extraction type, or reheat type.
5. By direction of steam flow : It can be axial flow, radial flow, or tangential flow type.
6. Multi stage or single stage.

In the impulse type turbines, the entire pressure drop takes place across the nozzles or stationary elements, the flow through rotor blades then being substantially at constant static pressure. In the reaction turbines the entire pressure drop is divided between the stationary blades and the rotor blades. Multistage turbines often incorporate both impulse and reaction stages. The efficiency of a steam impulse turbine can be improved by using more than one set of nozzles, blades, rotors in series. This process is known as compounding of impulse turbines which can be achieved by three ways :

1. Compounding for velocity
2. Compounding for pressure
3. Compounding for velocity and pressure

More than three-fourth of the electric power produced in the world is generated using some kind of steam turbines. Steam turbines as large as 1000 MW capacity are presently feasible. For solar power generation also; steam turbines have been used. Hsu and Leo[69] developed a simple and inexpensive reaction turbine in which the steam is ejected from a nozzle to rotate a wheel at high velocity. The steam pressure used is 450 KPa producing 200 watts at an efficiency of 1.2 percent only. Higher efficiencies were obtained when exhaust steam from nozzle was condensed to water in a condensor rather being discharged into the air.

Practically in all the central receiver solar power plants, steam turbines which are commercially available are used. In the 1-MWe experimental solar thermal electric power plant[67] installed at Sicily (italy),7-stage impulse turbine working at a steam inlet pressure of 6000 KPa at temperature 510 °C with a flow rate of 5200 Kg/hr and hexhaust pressure 6 KPa at 36 °C is employed. The capacity of the turbo-generator unit is about 1 MWe.

Another steam turbine-generator unit with a capacity of 1.5 MWe was selected for the pilot plant[70] of central receiver solar thermal power plant in California. The turbine selected for the pilot plant is a tandem - compound, single flow, single automatic admission condensing industrial turbine with a rating of 1.5 MWe (gross) when exhausting at 17 KPa absolute. The steam conditions are 10000 KPa at 477 °C, steam flow 14.2 Kg/sec.

Organic Rankine cycle turbines are preferred for solar power generation in the low and medium power levels i.e. upto 1000 KW due to their high cycle efficiency. Due to high molecular weight of the organic working fluid, the vapours can give the same kinetic energy ($1/2\ mv^2$) at lower turbine nozzle velocities compared to that when steam is used as working fluid. Because many organic fluids become superheated as they expand through the turbine, all regenerative heating can be accomplished with a re-generator located between the turbine exhaust and the condensor. Moreover, the vapour generator is to produce saturated vapour only and thereby avoiding the complexity of the vapour superheater. But due to low enthalpy change during expansion, large flow rate of vapour through the turbine will be required for the same power output relative to a steam cycle. This is advantageous in small turbines, since the blades of the turbine will be made larger bringing the nozzels nearer to full admission. Due to the large dimention and mass flow rate, the efficiency of the small turbine, will be high. Moreover, due to positive slope of saturation curve for organic vapours, there will not be erosion of blades while expansion. As discussed earlier many organic working fluids have been used in solar thermal power plants.

A survey of organic Rankine Engines was conducted by Curran[58]. He collected data on 2150 Rankine engines which use 16 different organic fluids providing power in the range of 0.1 to 1120 KW and were operational for different durations upto September 1979. This data is correlated by Badr Et al [61] in the form of the power output with rotational speed (rpm) and observed that the design data collected or available is not sufficient to draw any specific coclusion except that low speed engines ($\leq$5000 rpm) are predominately positive displacement engines providing power output upto 10KW and turbines with high rotational speed (>5000 rpm) are used for meeting greater demands (>10 KW). The data as complied by Curran[58] and in the format as presented by Badr et al[61] is shown in Table 4.2. It is observed from the table that power output for positive displacement expanders rises with increasing speed where as

Table 4.2 SURVEY OF RANKINE CYCLE ENGINES (FROM BADR et al[61])

Expander			Energy source	Equipment driven	Working fluid	Maximum working fluid temperature (°C)	Number of engines operating	Total operational period
Type	Power output (KW)	Speed (rpm)						
1	2	3	4	5	6	7	8	9
Single-stage radial inflow turbine	1.7	35000	Solar insolation	Vapour compressor	R113	80	2	-
Single-stage radial inflow turbine	1.7	60000	Solar insolation	Vapour compressor	R11	95	4	-
Two-stage multi-vane	2.3	1625	Solar insolation, Electricity[a]	Vapour compressor; Dynamometer[a]	FC-88	175	1	1000h[a]
Multi-Vane	2.3	1200-1800	Electricity[a]	Vapour compressor; Dynamometer[a]	R11	105	None at present	1200h
Reciprocating	3.0	1800	Gas	Electricity generator	CP-34	290	1	>100h

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Reciprocating	3.0	1800	-	Electricity generator; sweeper Vehicle	F-85	290	3	>100h
Reciprocating	3.0	3600	-	None	R-22	230	1	>100h
Single-stage turbine	3.0	70000	-	Electricity generator; vehicle	F-85	290	1	>100h
Single-stage reaction turbines	4.0	1200	Solar insolation, Geothermal heat	Water pump; Electricity generator	Tetrachlo-roethylene	80	1 4	1000h
Screw	5-100	1500-1800	Solar insolation	Water pump; Electricity generator	R11	95	70	>4000h
Multi-vane	7.5	1800	Gasoline	Electricity generator; Dunamometer	Alcohol/ water	340	-	18000h

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Multi-vane	7.9	1625	Solar insolation, Electricity [a]	Vapour compressor; Dynamometer [a]	FC-88	175	2	1000 [a]
Single-stage radial inflow turbine	12	30000	Solar insolation	Vapour compressor	R11	95	2	-
Single-stage radial inflow turbine	15	24400	Solar insolation [b]	Vapour compressor	R113	80	7	750h
Turbine	15	42000	Solar insolation	Electricity generator	R113	110	3	500h
Single-stage radial inflow turbine	16	40000	Solar insolation [b]	Vapour compressor	R11	150	1	25h
Single-stage radial inflow turbine	19	36300	Solar insolation	Irrigation pump; Electricity generator	R113	165	1	600h
Helical screw	20	7500	Solar insolation; oil	Electricity generator	R114	200	5	1000h

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Single-stage radial inflow turbine	20	20100	Solar insolation	Vapour compressor	R113	150	1	-
Single-stage impulse turbine	32	19400	Solar insolation; gas	Electricity generator	CP-25	300	1	2400h
Single-stage radial inflow turbine	34	11950	Solar insolation	Vapour compressor	R11	86	1	2000h
Single-stage radial inflow turbine	35	5500	Solar insolation	Irrigation pump	R11	86	1	>100h
Single-stage radial inflow turbine	37	18000	Solar insolation	Vapour compressor	R11	95	2	-
Single-stage radial inflow turbine	37	30700	Solar Insolation	Irrigation pump	R113	135	1	500h
Single-stage turbine	38	35000	Diesel engine exhaust gases	Automobile	F-50	315	3	>100h
Three-stage turbine	38	60000	Diesel engine exhaust gases	Automobile	F-50	315	1	>100h

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Turbine	40	6700	Exhaust gases	Electricity generator	Tetra-chloro-ethylene	115	1	300h
Turbine	45	6700	Solar insolation	Electricity generator	Flutec PP3	280	1	-
Turbine	50	6600	Geothermal	Electricity generator	Trichlo-roethylene	70	1	-
Turbine	60	-	Steam produced by diesel engine exhaust	Electricity generator	R113	70	1	-
Single-stage radial inflow turbine	63.4	22400	Solar insolation	Vapour compressor	R113	135	2	100h
Reciprocating	112	1800	-	Automobile	F-85	315	1	>100h
Turbine	150	20000	Solar insolation; oil	Electricity generator	CP-25	450	1	

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Radial inflow turbine	335	9500	Steam produced by diesel engine exhaust	Electricity generator	R11	88	1	
Single-stage impulse turbine	375	1800	R114 vapour from process	Compressor	R114	120	1	6 years
Single-stage impulse turbine	375	1800	R114 vapour from process	Compressor	R114	120[c]	1	10 years
Six-stage turbine	450	12500	-	Electricity generator	F-85	290	1	>100h
Single-stage turbine	450	18000	-	Electricity generator	F-85	290	1	>100h
Single-stage impulse turbine	600	9300	Waste gas	Electricity generator	CP-25	240	5	-
Turbine	670	11100	Furnace Exhaust	Electricity generator	F-85	290	1	-
Turbine	1000	-	Geothermal heat	Electricity generator	-	-	-	-

Table 4.2 cont.

1	2	3	4	5	6	7	8	9
Single-stage reaction turbine	1050	4800	R114 vapour from process	Compressor	R114	120	1	2.5years
Six-stage impulse turbine	1120	1200	R114 vapour	Compressor	R114	120	1	1.75years

a Laboratory test

b Experimental unit with simulated solar input

c Value estimated by Curran[58].

in case of turbine the behaviour is reversed. This data was collected from more than 20 leading manufacturers and observed that the major manufacturer of organic Rankine turbine is Ormat turbines of Isreal, who have produced few thousands of such engines and the recommended working fluid is trichlorobenzene.

The concept of similarity can be employed to compare the performance of different types of expander. It is shown by Balje[71] that four parameters like specific speed, specific diameter, Mach number and Reynolds number can be employed to represent the maximum obtainable efficiencies and the optimum design geometry of turbines. The specific sped N_N is a measure of the rotational speed of the expander for a given volume flow rate and a given enthalpy change through the expander. The specific diameter D_S can be a measure of the size of the machine. Reynolds number Re represents the physical properties of the working fluid. The Mach number M is the ratio of the velocity of the fluid to the acoustic velocity in the fluid. It was observed by Barber and Prigmore[59], that only the two parameters i.e. the specific diameter D_S and the specific Speed N_S can be used to determine the performance of the expander and the other two parameter i.e Raynolds number and Mach Number have only secondary effects on the performance of expander. These two similarity parameters i.e. specific speed N_S and specific diameter D_S are given as :

$$N_S = \frac{NV^{\frac{1}{2}}}{(\Delta h)_{is}^{3/4}} \tag{4.9}$$

$$D_S = \frac{D\ (\Delta h)_{is}^{\frac{1}{4}}}{V^{\frac{1}{2}}} \tag{4.10}$$

where

N = expander rotational speed (rpm)
V = expander exit flow rate (m^3/sec)
$(\Delta h)_{is}$ = adiabatic enthalpy drop across the expander (J/kg)
D = diameter of the expander (m)

The available performance data on expanders was used by Balje[71] who used the similarity concepts and computed the optimal geometries and maximum obtainable efficiencies for different types of expander. This information was plotted by Balje[71] in the form of N_S - D_S diagrams for all

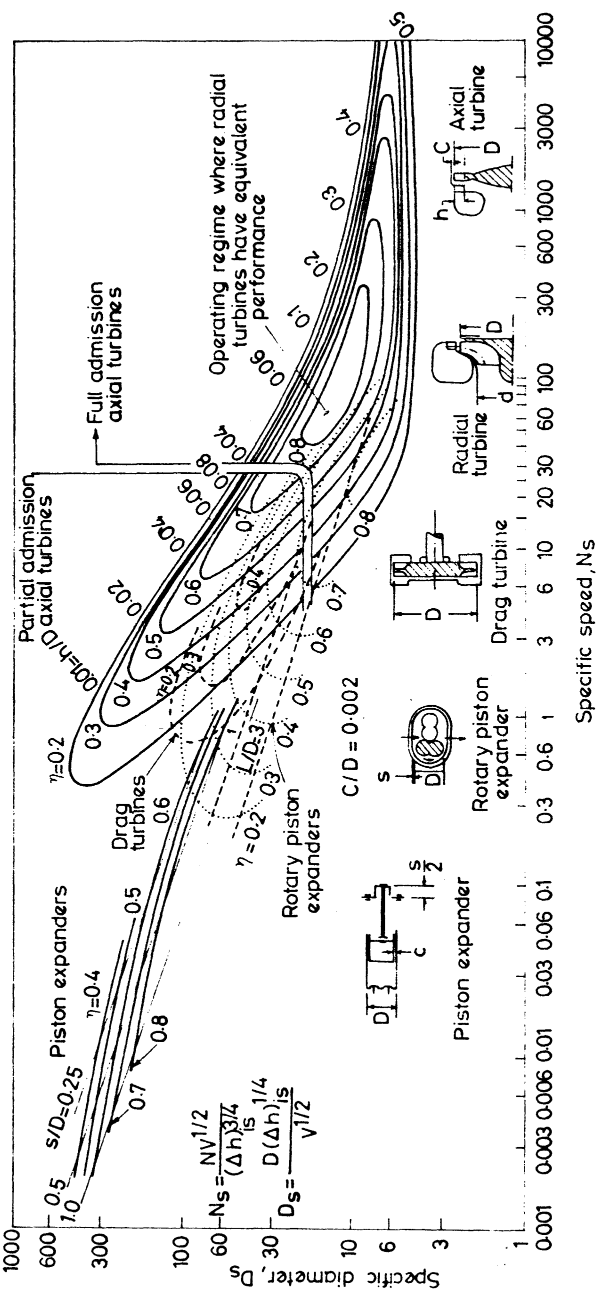


FIG.4.14 - PERFORMANCE CURVES OF VARIOUS TYPE OF EXPANDERS (Courtsey O.E.Belje)

expander types and the same is shown in figure 4.14. From this figure it is seen that in different specific speed ranges there can be a particular type of expander which can given better performance. It is observed that in the low specific speed range of 30 - 100, the performance of radial turbines is similar to that of full admission axial turbines. The optimized performance chart for axial flow turbines showing efficiency as a function of N and D is shown[59,71] in figure 4.15.

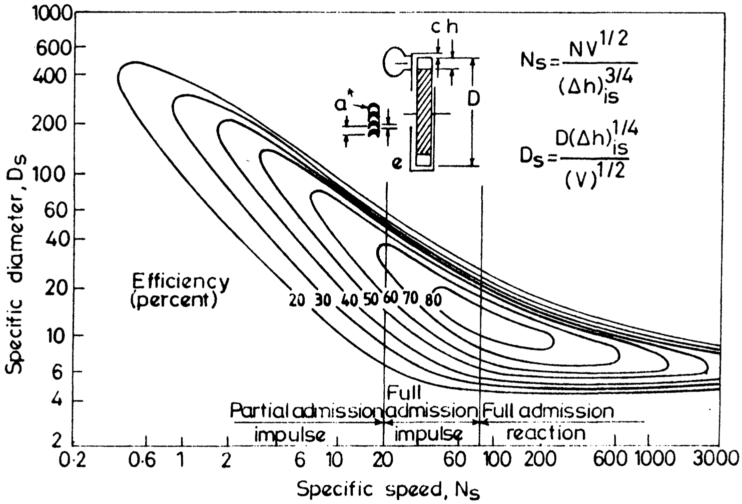


FIG.4.15 PERFORMANCE OF AXIAL FLOW TURBINES (Courtesy O.E.Beije)

Tabor and Bronicki[21] examined 16 different organic fluids for Rankine cycle organic vapour turbine and developed a 2 KW monochlorobenzene turbine operating at 150 °C and at 18000 rpm using a 6 to 1 reduction gear. The overall efficiency of converting heat to shaft power was 10 to 15 percent. Tabor also built a 3.8 KW solar operated organic vapour turbine and demonstrated the same for pumping of water in Rome in 1961. Efficiency calculations are also made for a 10 KW turbines with 10 nozzles and 100 percent admission using monochlorobenzene as the working fluid. In the temperature range of 160-180 °C the efficiencies were found to be 15-20 percent,.

The Barber-Nichols Engineering Company of Colorado, USA has developed[72] an organic Rankine cycle power conversion subsystem in 1980 under a subcontract from Ford Aerospace and Communications Corporation (FACC). The working fluid of organic vapour turbine is toluene which is heated to about 371 °C in the receiver of a parabolic disc and is expanded through a single stage axial flow turbine. The basic concept is shown in figure 4.16, alongwith the engine. The exhaust vapour from the turbine is passed through an integral recuperator or regenerator and into a forced air cooled condenser which forms the outer annuals of the converter assembly. This regnerator is a heat exchanger which transfers a part of exhaust waste heat to the low

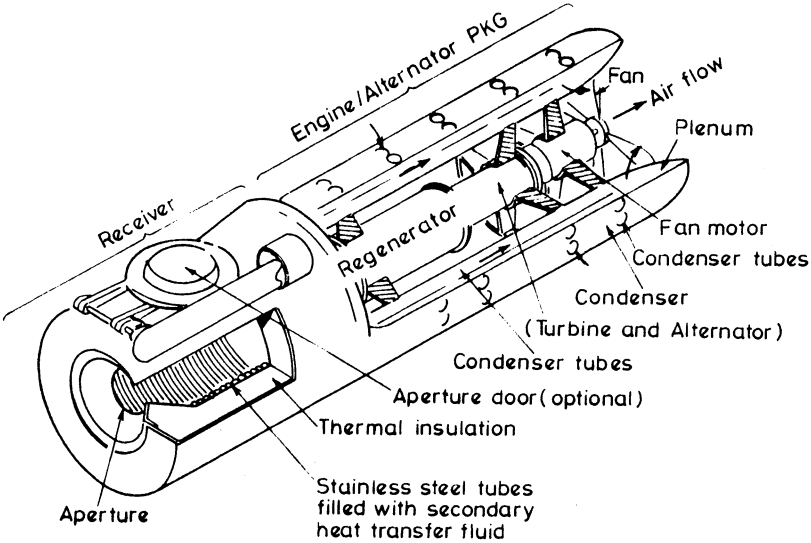
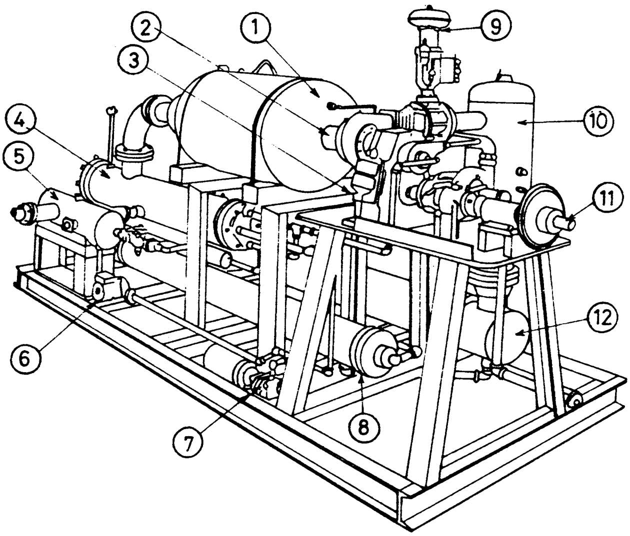


FIG.4.16 ORGANIC RANKINE CYCLE VAPOUR TURBINE AND ALTERNATOR

temperature incoming fluid. An alternator of permanent magnet type which is of high speed is directly coupled to the turbine. Several simulation tests are conducted on this power converter and found that when the engine input is 56 KWt the power output is about 16.5 KWe. The engine is capable to operate upto 83 KWt and the net efficiency (heat to electric engine) was observed to be about 25 per cent. Barber and Prigmore[59] described the working of a 19 KW Rankine cycle engine developed by Barber-Michols Engineering Company which was installed at Willard, New Mexico in April 1977. The Rankine power system is sketched in figure 4.17. The working fluid is Refrigerant-113 which is vapourized at a temperature of 160 °C and supplied to the turbine. The collector working fluid is Caloria HT-43 which transfers its heat through a heat exchanger in the boiler to R-113. The turbine shaft speed is 36300 rpm and the output shaft speed is 1730 rpm. The predicted turbine efficiency is 75 per cent and the pump efficiency is 40 percent. The overall cycle efficiency is calculated to be 15.3 percent.



(1) Regenerator
(2) Turbine
(3) Gear box
(4) Condensor
(5) Float tank
(6) Boost pump
(7) Startup pump
(8) Preheater
(9) Speed control valve
(10) Demister
(11) Output shaft
(12) Boiler (heat exchanger)

FIG.4.17 ORGANIC RANKINE CYCLE ENGINE (19 KW)

Badr et al[61] in their recent review pointed out the limitations of the positive displacement type engines and turbines particularly in the low power range. It is concluded that for low power requirements the organic Rankine cycle engines like the reciprocating piston, the rotary screw (i.e. the helical system) and the rotary multivane expander (MVE) will show high efficiency compared to turbines.

Like that of the expander, the performance of the feed pump required in the Rankine cycle is also dependent on the four parameters : namely, the specific speed, specific

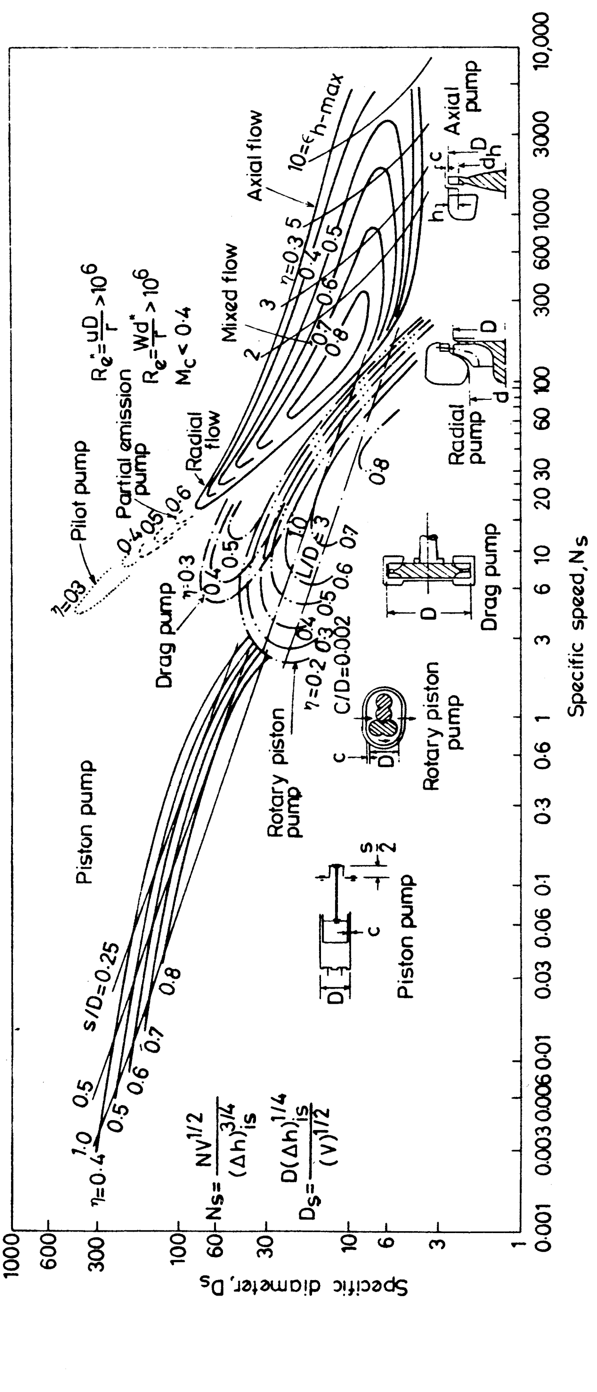


FIG.4.18 PERFORMANCE CURVES FOR PUMPS AND COMPRESSORS (Courtsey O.E.Belje)

diameter, Raynolds number and suction specific speed[59,73]. The data given by Balje[73] for different pumps and compressors is shown in the form of N_S-D_S diagram in figure 4.18. Like expanders, here also for a certain range of specific speed, a particular type of pump will perform better.

4.7.3 Stirling Engines

Stirling engine as discussed earlier has the highest theoretical efficiency reaching to the Carnot engine due to isothermal heat addition and removal during expansion and compression, and through isothermal regenerative heat addition and removal at constant volume[80]. But unfortunately practical stirling engines suffer from many defects and are therefore not a good approximation to the theoretical engines.

The stirling engine was first invented by Robert Stirling of Scotland in 1816 and used by Erricson[74] in 1870. These engines were earlier called air engines upto 1950's. From 1860 to 1920 several thousand air engines with a low efficiency (upto 3 percent) were made in sizes upto 4 KW. The N.V.Philips Company of Netherlands in the year 1937 revived the interest in air engines and used modern engineering and concepts and obtained an efficiency of 30 per cent. Dr.R.J.Meijer used helium or hydrogen as a working fluid and reached to an efficiency of 38 per cent and called these air engines as Stirling engines. Today several firms like General Motors and Ford Motors of USA, United Stirling Engines of Sweden, Sun Power Inc. of USA and a firm in Germany are making Stirling engines of various sizes[75].

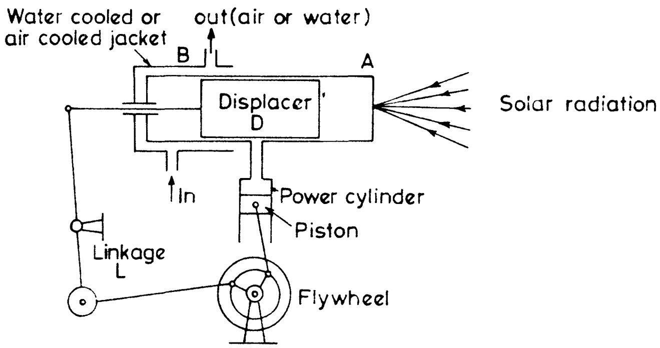


FIG.4.19 PRICIPLE OF STIRLING ENGINE

Farber and Prescott[76] studied a Stirling hot air engine and obtained an efficiency of 9 percent at 100 rpm with a brake horse power of about 150 Watt. The principle of a hot air engine is shown in figure 4.19. The focussed sunlight heats the air contained in the right side of the cylinder, which then expands and forced down the piston P turning the flywheel clock wise. When the flywheel turns it moves the displacer D towards left in the cylinder through linkage L. During the up stroke of the piston, the displacer moves to the right leaving the hot air in the left section of the cylinder B from where the heat is taken away by water cooled jacket or air cooled jacket. The simplified version of the four stages of hot air engine is shown in figure 4.20. In process one, the air is first compressed in

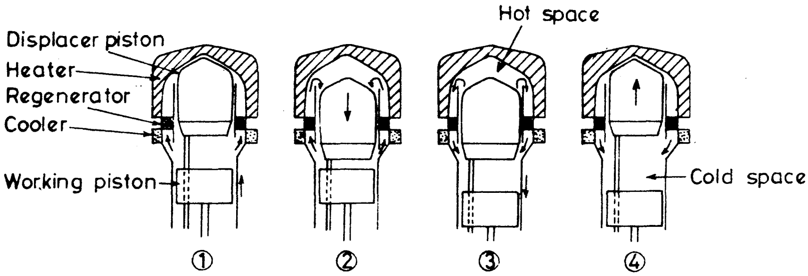


FIG.4.20 STIRLING ENGINE CYCLE

the cold space of the engine and then by means of a displacer piston forced to flow through a regenerator into a hot space surrounded by the heat source. In the second process the air gets heated and at a high pressure passes through the regenerator to the cold space where the high pressure air expands and activate the piston. The actual P-V diagram of this engine is shown in figure 4.21 which is quite different from the ideal Stirling cycle shown in figure 4.3(c).

The main limitation with the Stirling hot air engine is the poor heat transfer across the head of the cylinder, due to its small area and of metallic construction. An improvement was made by Finkelstein[77] and also by Trayser and Eibliing[78] who used transparent quartz windows for the cylinder head to focus the solar radiation directly inside the engine. Thus the solar radiation is absorbed directly by the inside air without any loss. An efficiency of about 32 percent was reported in this improved stirling engine.

A good review of solar stirling engine is made by Martini[79] and who in his recent paper[75] classified the Stirling engine into three main engine types as shown in figure 4.22. The alpha type stirling engine are of high efficiency and are generally used. In the Beta type engines the power piston and displacer are in the same cylinder and so their strokes can overlap. The Gamma type engines are mechanically more convenient but are less thermally efficient.

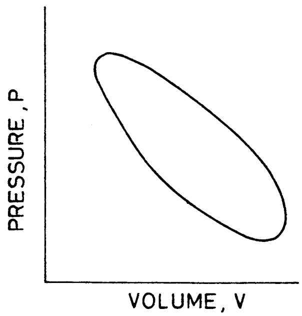


FIG.4.21 ACTUAL STIRLING CYCLE

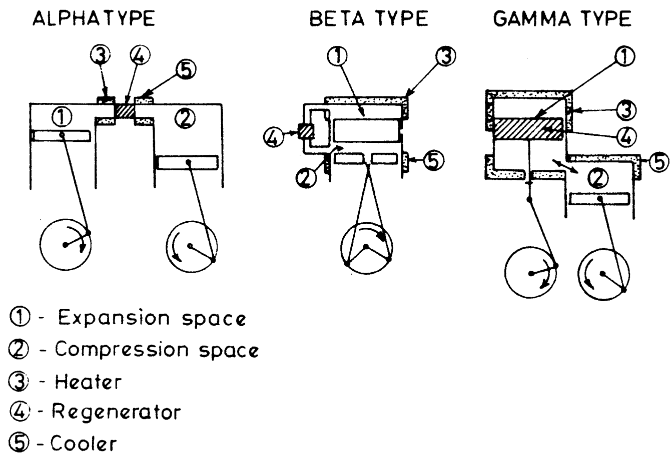


FIG.4.22 TYPES OF STIRLING ENGINE

Basically solar operated Stirling engines are of two types[80,81] e.g., free piston Stirling engine (FPSE) and the kinetic Stirling engine (KSE). The free piston Stirling engine was first invented by William Beale[82] and is basically a thermally driven mechanical oscillator and with the linear alternator hermetically sealed. In this arrangement the piston remains stationary and the displacer moves under the influence of pressure differential between the working space and the bounce space in the cylinder. The movement of the cylinder due to pressure differential performs work. Since the entire pressure enclosure is to move as a unit, the free piston Stirling engine eliminates the need for high pressure shaft seals or lubricants. The free piston Stirling engines have high mechanical efficiency but suffers from many problems[81] such as reproducible and tuning performance, starting, and power take-off. Because of the required masses and resulting inertial effects, the free piston Stirling engines are not supposed to be practical for large power demands (>100 KWe). The Kinematic Stirling engines require a mechanical drive external to the gas-cycle, with a working seal interface[80]. These engines are bulky, and therefore, not suitable for mounting at the focus of solar concentrating collector. Moreover, due to their poor mechanical efficiency, the overall performance gets reduced.

The Stirling engine as discussed above differs from gasoline engine in two ways: firstly the working fluid(gas) is recycled and secondly the heating and cooling takes place by heat transfer successively. The Stirling cycle differs from Rankine cycle in the sense that the working fluid in the Stirling cycle remains only in the gaseous phase which is generally air, hydrogen or helium. The only disadvantage is that it operates at high temperatures ($\simeq 600\ ^{\circ}C$).

Mechanically[83], there are two classes of Stirling engines: single acting (displacer) and double acting. In the single acting unit two pistons are used for each power unit. In the double acting type one piston per cylinder is used which acts as compressor, expander and displacer by interaction between neighbouring cylinders. There is no limit for the number of cylinders but generally for a single power unit 3 to 7 cylinders are used. Most commonly, four cylinders are used which are arranged either in line or axially in a square. The Stirling engine has two parts one external and another internal. The external system supplies heat either from sun or by combustion of fuel to the engine heater. The internal system is filled with a working gas like air or helium or hydrogen at an elevated pressure and consisted of two variable volumes and three heat exchanger called the heater, regenerator and cooler.

The sun Power Inc., Ohio (USA) which was founded by William Beale who is the inventor of Free Piston Stirling

Engine started manufacturing it since 1971. As mentioned earlier these free piston Stirling engines have several advantages over the kinetic Stirling engines. The firm has developed[84,85] three models: Model-10 (10 watt capacity), Model SD-100 (100 watt capacity), and model RE-1000 (1150 watt capacity). The working of the free piston engine as described by Taylor[85] is shown in figure 4.23. It is shown

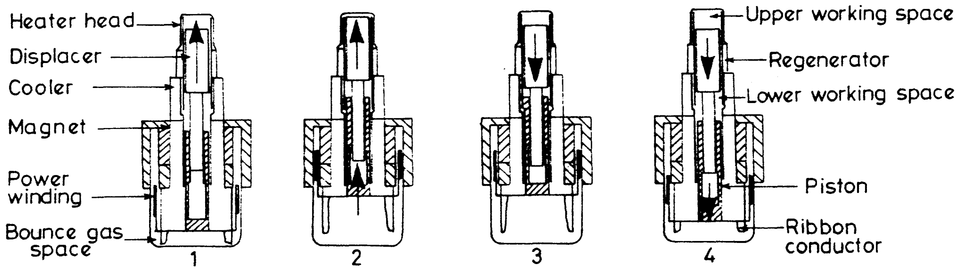


FIG.4.23 WORKING PRINCIPLE OF FREE PISTON STIRLING ENGINE (From Taylor[85])

in Part I of the figure that when the gas pressure in the lower space pushes the displacer up, the gas in the upper hot end space is forced through the heater head and regenerator into the cold lower space. The displacer continues to move up till the pressure in the bounce space equals the pressure of the cool working gas thereby reducing the total working gas volume and compressing the gas as shown in Part two. Now the head of the displacer gets heated, the pressure force difference forces the displacer down as shown in part three of the figure. It pushes the working gas through the regenerator and the heater head thereby heating the gas. This hot working gas shown in part four pushes down the displacer -via the remaining cold end gas-the power piston. Now the gas pressure in the upper portion drops and the gas in the lower portion pushes the displacer up again and thereby completing the cycle. Extensive data have been collected on Stirling engine Model RE-1000 and its output power is plotted verses piston stroke for different heater head temperatures as shown in figure 4.24. Details of a free piston Stirling engine is shown in figure 4.25.

Mechanical Technology, Inc., of USA analyzed several Free Piston Stirling Engines. Their design calculations for a 50-KWe Stirling engine with linear alternator have shown an efficiency of about 38 per cent.

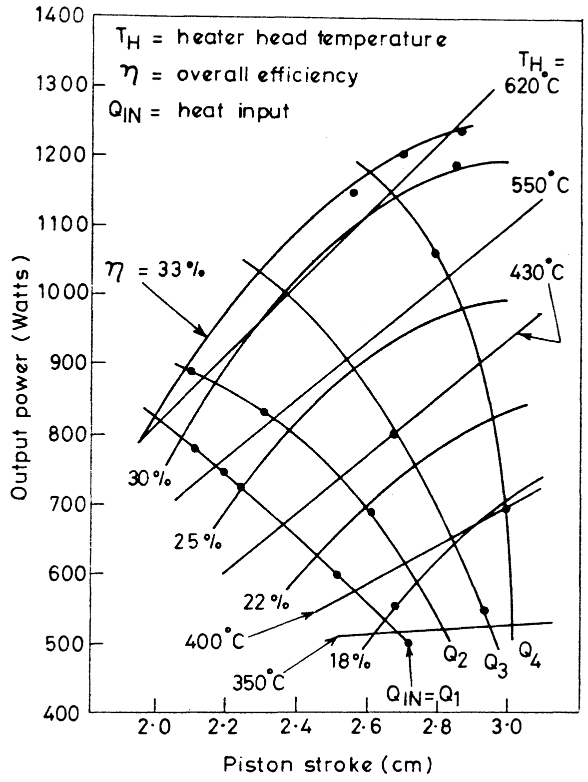


FIG.2.24 PERFORMANCE OF FREE PISTON STIRLING ENGINE RE-1000 (From Taylor[85])

The firm United Stirling located in Malma, Sweden (USS) is one of the largest [86,87] and most experienced one in the field of Stirling engines. The first commercial stirling engine of 150 KW capacity, with 4 cylinder in-line engine with a rhombic drive mechanism was made in 1971 which was a single acting displacer type engine having two piston per cylinder. Later in 1972 the United Stirling decided to work only on double acting systems. Since then, this firm has made hundreds of Stirling engines of different capacities and for different applications. With the interest on solar parabolic dish for power production, the interest on Stirling engines was revived and United Stirling started concentrating on compact and efficient engines for parabolic disc. The charactersitics[87] of some of the Stirling engines developed by USS are shown in table 4.3. A

Table 4.3 United Stirling Engine Family Characteristics For Modular Solar Thermal Electric Power Systems (From Wells et al[87])

Engine Model	RPM	Pressure (MPA)	Shaft Power (KW)	Solar engine efficiency (per cent)	Thermal input (KW)	Collector area (m^2)	Collector diameter (M)	Module electric output (KW)*	Number of modules for 1 MW
V-160	1800	12	9.4	33	28.5	39.1	7.1	8.74	114.4
4-95	1800	12	24.5	40	61.2	84.0	10.3	22.78	43.9
4-275	1800	12	62.0	42.5	145.9	200.1	16.0	57.66	17.3
4-3456	600	12	310.0	46.7	633.8	910.6	34.0	288.30	3.47

Conditions: Outlet water temp = 50 °C, Tube temp = 720 °C, Net Heat Input to Heater Heads, Solar: 9 KW/m^2, 90% Reflective Dish, 90% Receiver Efficiency, Hydrogen as working gas.

* based on 93% electric generator efficiency for all modules.

survey of Stirling engine for solar power generation is recently made by Wells et al[87].

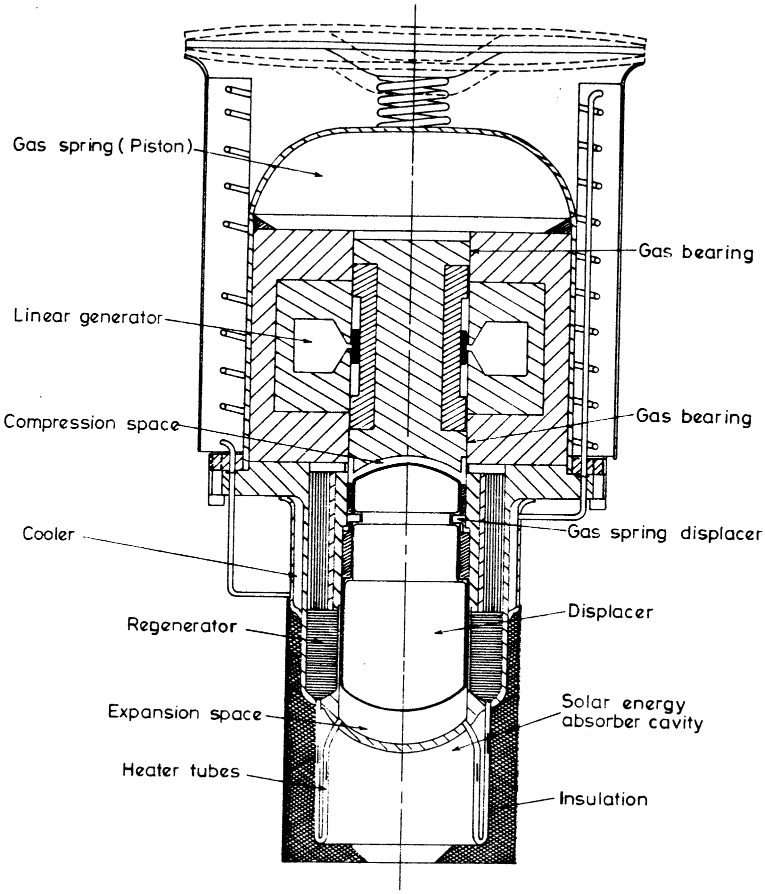


FIG.4.25 SUNPOWER 1 KILLOWATT ENGINE SPIKE (Courtsey Sunpower Inc.Ohio)

On the request of Jet Propulsion laboratory, USA the USS tested the performance of their Stirling engine P-40 in the inverted position so that it can be kept at the focus of parabolic dish concentrator. This engine ran successfully[72] for 3 hours at 720 °C, 11 MPa at 1500 rpm producing about 16 KWe. This engine had later given an efficiency of 38.1 per cent. This P-40 stirling engine known[87,88] as 4-95 engine is a four-cylinder, 95 cc displacement, double acting, twin crank drive machine. Its detail are described by Christer in an early publication[83]. In this engine the four cylinder are arranged in a square. Eight re-generators and coolers are placed in a ring outside the cylinders which minimise the dead volumes in the hot and manifolds as well as the dead volumes in the ducts connecting the cooler with the compression space. The heater is suitable for direct solar energy absorption and is a two pass cross flow tubular type with rotational symmetry. The engine is shown in figure 4.26.

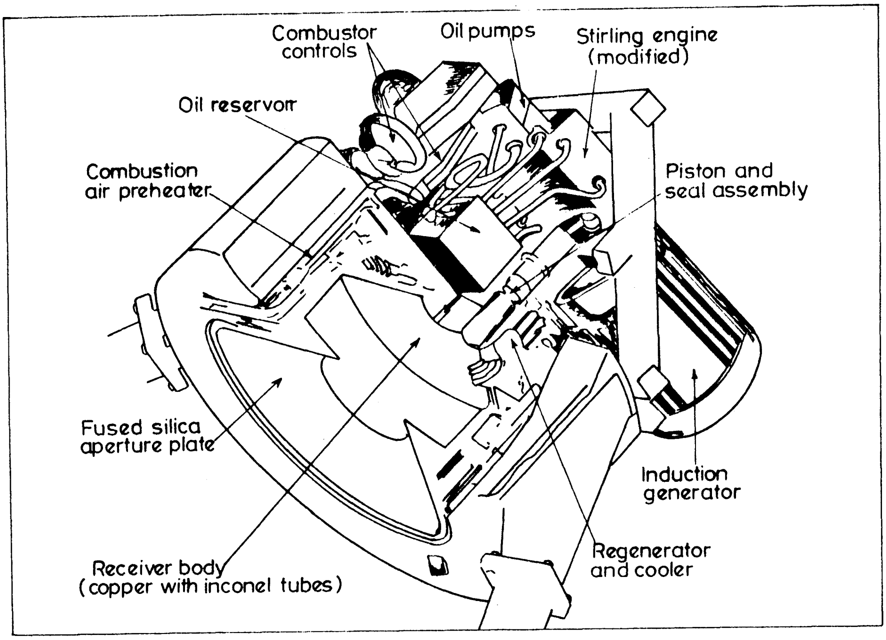


FIG.4.26 STIRLING ENGINE (P-40) WITH GENERATOR

Another 85KW capacity Stirling engine 4-275 called as P-75, has a maximum speed of 1800 rpm. The displacement per cylinder in this case is 275 cc. Studies are also in progress on designing of 400 KW and 500 KW stirling engines suitable for solar power generating based on the concepts of 4-95 and 4-275 engines.

The general equation to know approximately the power output (W in watts) as given by Martini[75] is as follows:

$$W = 0.035\ f p_m \left(\frac{T_e - T_c}{T_e + T_c}\right) \left(\frac{\pi}{2} \frac{V_c V_e}{V_m}\right) \text{Sin } F \qquad (4.11)$$

where

W = power output (watts)
f = operating frequency (Hz)
p_m = mean cycle pressure (bar)
T_e = expansion space(heater) temperature (K)
T_c = Compression space (cooler) temperature (k)
V_c = volume swept by power piston (cm^3)
V_e = volume swept by displacer (cm^3)
V_m = gas volume of mid-stroke of power piston or of both power pistons (cm^3)
F = phase angle between displacer and power piston motion.

4.7.4 Brayton engines

As discussed earlier the engines based on Brayton cycle can be of large capacity, more efficient (>30 per cent) compared to Rankine engines, and require high temperature (>600 °C) for operation. The cycle consists of adiabatic compression, constant pressure heating, and adiabatic expansion. Work is done by the hot gas while expansion which is more then the work of compression. By using a regenerator in the exhaust of the gas turbine, intercooler in the compressor, and reheating the working fluid during expansion the performance of the turbine can be improved.

Although the thermal efficiency of a Brayton cycle mainly depends on compressor ratio ($R = P_2/P_1$), the turbine inlet temperature, and the parasitic losses (like efficiency of turbine and compressor), but for an ideal cycle, the thermal efficiency depends only on pressure ratio.

$$\text{Theramal efficiency} = 1 - (1/R)^{(r-1/r)} \qquad (4.12)$$

where r is the ratio of specific heat of air at constant pressure to constant volume. Figure 4.27 shows the effect of compressor pressure ratio on the thermal efficiency of Brayton cycle at different turbine inlet temperatures. The actual Brayton cycle differs from the above ideal cycle because of:

1. the air properties (K, C_p) are not constant over the range of operating temperatures.
2. compression and expansion processes are not

frictionless and take place with increase in entropy.
3. the internal losses are difficult to control
4. the mass of gas flowing through the turbine is more than the mass of air flowing through the compressor
5. the specific heat of combustion gas is higher than air.

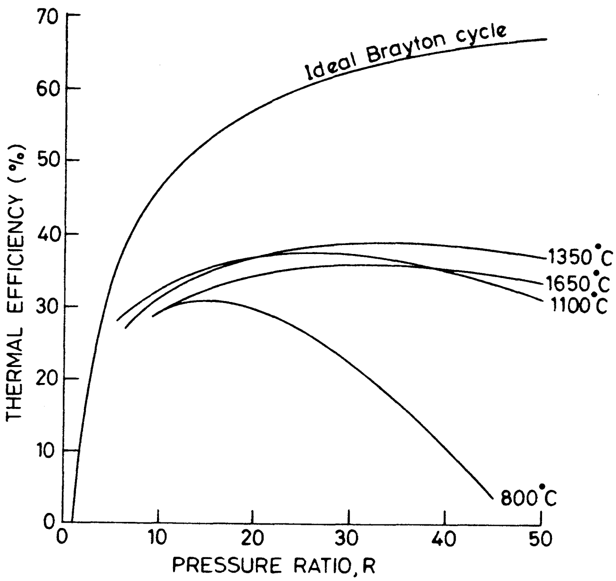
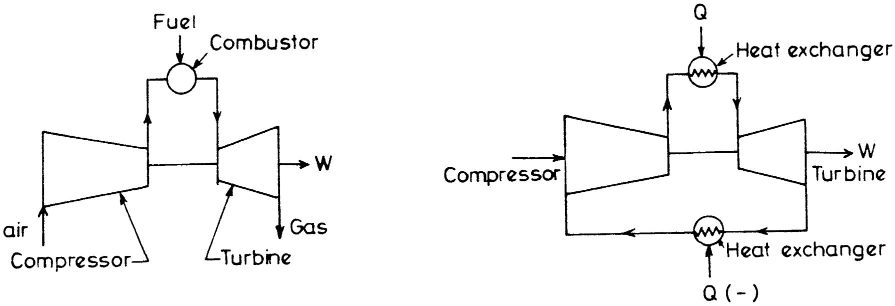


FIG.4.27 BRAYTON CYCLE EFFIIENCY FOR DIFFERENT PRESSURE RATIO

If the Brayton engine operates at a compression ratio less than the optimum value as shown in figure 4.27., then the temperature of the turbine exhaust gas becomes more than the compressor exhaust. In such a case the performance can be improved by using a regenerator transfering some exhaust heat to the compressed airstream. There are two general types of gas-turbine power plants as shown in figure 4.28. In the open cycle plant (figure 4.28(a)), the fuel is injected into the combustion air subsequent to compression and the resulting gas expands in the turbine and finally exhausted into the atmosphere. The open cycle plant is generally preferred due to its simplicity, no need of cooling unit, simple controls, and possible design for high power-weight ratio. In the closed cycle plant (figure 4.28(b)), the working gas is continuously recycled. Fuel is burned externally to the system, and the energy liberated in the combustion reaction is transferred as heat to the circulating gas. The advantages of a closed cycle plant are: low grades of fuels may be burned, clean working gas,ability to control the density of working gas. The density control helps in changing the power output without changing the compression ratio and turbine inlet temperature. The closed cycle power plant is suitable with nuclear reactors since fluids other than air like helium can be used at high temperatures. The disadvantage of closed cycle being the large size and additional cost of heat exchangers.



(a) Open-cycle gas turbine plant. (b) Closed cycle gas turbine plant.

FIG.4.28 SIMPLE GAS TURBINE CYCLES

The simple gas turbine cycle with intercooling, reheat and exhaust heat exchange described by Gupta and Prakash[89] is shown in figure 4.29 alongwith the T-S diagram showing various positions of the cycle. The thermal efficiency of this cycle is given as :

$$\text{Thermal efficiency} = [(T_6 - T_7)+(T_8 - T_9)-(T_2 - T_1)-(T_4 - T_3)-(T_4 - T_3)]/[(T_6 - T_5) + (T_8 - T_7)] \qquad (4.13)$$

The effect of various modifications in the simple Brayton cycle on the work output and thermal efficiency are also computed by them for a turbine with pressure ratio of 4, T_{max} = 684 K, T_{min} = 288 K and P_1 = 1 atmosphere. The comparison is given[89] in table 4.4 :

Table 4.4 Effect of various modifications in the simple Brayton cycle turbine(From Gupta & Prakash[89])

Modification	Percentage effect on efficiency	Percentage effect on out put
Heat exchanger	+50.0	nill
Intercooling	-6.5	+10.2
Reheat	+10.4	+24.5
Reheat + heat exchanger	+66.7	+24.5
Intercooling + heat excahnger	+68.0	+10.2
Reheat + intercooling	-18.2	+34.7
Reheat + intercooling + heat exchanger	+80.0	+34.7

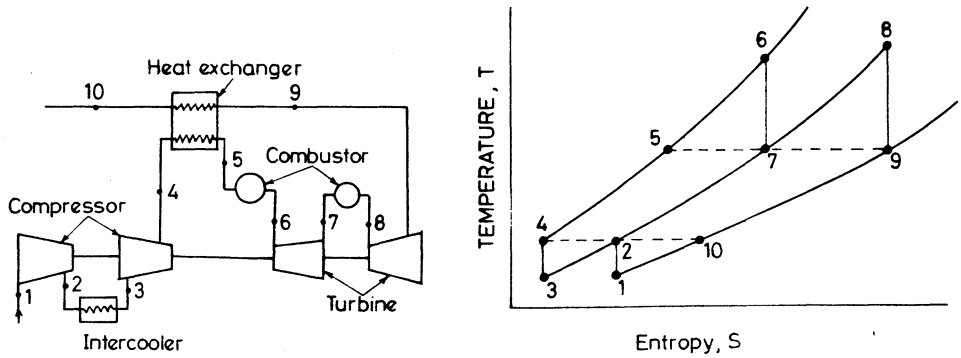


FIG.4.29 BRAYTON CYCLE WITH DIFFERENT MODIFICATIONS (From Gupta and Prkash[89])

The main components of the Brayton power plant are as follows:

1. Turbine : All gas turbines are axial flow type except in small sized installations where radial inward flow type turbines are used. Although pressure in the turbine is quite low but due to very high gas temperature special kind of cooling arrangements are to be made and special materials for turbine blades and other hot parts are to be employed. High temperature alloys like nickel and cobalt are used. Research is in progress for special kind of high temperature coatings and use of ceramic materials for turbine blades etc.
2. Regenerator : The purpose of the regenerator is to transfer a part of heat from the turbine exhaust to the air stream from the compressor resulting in increase in efficiency, increase in volume and weight of the plant, and increase in cost of the system. Generally Rotory regenerators which show high performance and of low weight are preferred. The regenerator should be able to handle large amount of working fluid with low pressure drop and with rapid large temperature change. The intercoolers have also the similar problems.
3. Combustor : The main purpose of the combustor is to bring the gas at constant uniform temperature with little loss of pressure. Combustors of very high rates are available (2×10^{11} J/hr. m^3 atm.).
4. Compressor : Generally axial-flow compressors are employed because of their high efficiency and capacity. Compressors which can handle air capacity of 380 m^3/sec are available.

The open cycle Brayton engine with parabolic dish shows higher efficiency and is cost effective compared to the organic Rankine cycle engines. Under a contract from Lewis Research Centre. USA, the Garrett Ai Research of Torrance, California, USA is building several advanced gas turbines suitable for parabolic dish. The turbine is made of ceramic materials[90] to operate in the temperature range of 1100-1400°C and all metallic components to operate in the range of 700-900°C. The first generation engine developed by Garrett is shown in figure 4.30 which consists of two parts, a receiver and a engine/generator. The engine is a recuperated[72,90], open cycle of 20 KW capacity with overall efficiency (heat to electricity) of 30 per cent. In the second generation engines where ceramic materials are used and which will operate at higher temperatures, higher efficiencies are expected. Under a contract from JPL (jet Propulsion Laboratory) the Garrett also built a all metal receiver suitable for the above machine. The receiver is designed for an outlet temperature of 815 °C and inlet temperature of 565 °C. Design and performance data of Brayton engines with maximum power less than 97 KW is compiled by Fujita et al[91] from the publications made by Rackely[92] and Helins[93]. The data is reproduced in Table 4.5.

Table 4.5 Component efficiency of Brayton engines

Component	Current Technology (all metal) CCPS-40	Current Technology (all metal) Solar	Advanced Technology (ceramic) Garrett	Advanced Technology (ceramic) Allison
Turbine inlet temp.(°C)	816	816	1371	1178
Pressure ratio	1.9	2.7	5.0	4.5
Recuperator effectiveness	0.90	0.94	0.93	-
Compressor efficiency	0.76	0.76	0.80	0.80
Turbine efficiency	0.86	0.86	0.87	0.88
Loss factor	0.94	0.90	0.91	-

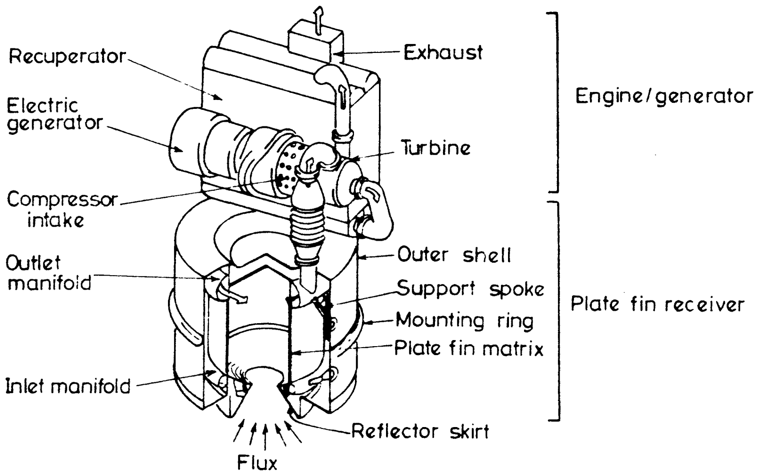


FIG.4.30 AIR-BRAYTON ENGINE, GENERATOR AND RECEIVER

4.8 SOLAR POWER PLANTS

Several solar Power plants ranging from few KWe to 10 MWe are built in many countries for field studies. A few of them are very successful and are competitive to conventional power plants. A contract was given to Solar Energy Research Institute, Colorado USA by US Department of Energy to conduct a comparative analysis and ranking of eight generic solar thermal systems in the 0.1 to 10 MWe capacity range[94]. A simple optimization procedure for solar thermal electric power plant design is discussed by Bohon and Levy[95]. This optimization technique is based on suboptimization in which the performance (energy output) to each power plant selected for the study is maximized and then the best design via differential cost estimates (minimum cost of electricity) is selected. It is not possible to discuss all the solar thermal power plant and the optimization procedure, but a few typical power plants which differ in concepts or size are discussed briefly in this section.

4.8.1 COOLIDGE 150 KWe POWER PLANT (Ref.33)

The coolidge 150 KWe solar thermal power plant[32-34] which is operational since November,1979 is a joint effort of University of Arizona, Sandia Laboratory, U.S. Department of Energy, Acurex Corporation, Sandstrand Corporation and Sullivan and Masson consulting Engineers. This plant was constructed to study the feasibility of using solar energy for driving irrigation pumps. The plant was installed at the Dalton Cole farm, south of coolidge, Arizona.

The coolidge solar thermal electric plant as shown schematically[33] in figure 4.31 consists of an array of solar collectors, thermal energy storage unit, and a power conversion subsystem. The details of the plant are listed in table 4.6[33]. The collector field with a total area of 2140.5 m^2 of line-focussing parabolic trough collectors, manufactured by Accurex Corporation, California, is arranged in 8 north-south oriented loops, each containing 48 collectors. Each collector trough is of about 1.8 m wide and 3 m long and originally had aluminium reflective surface which was later (spring 1981) laminated with aluminium acrylic film (FEK-244) to improve their performance. The collector receiver tubes, located at the solar collector focus, are coated with black chrome selective coating and surrounded by a pyrex glass tube. The concentration ratio of the collector receiver system is about 36. A heat transfer oil, Caloria HT-43, is pumped through the receiver tube by a pump at a controlled flow rate such that the outlet oil temperature reaches to 288 °C. The collector field is shown in figure 4.32 (photo).

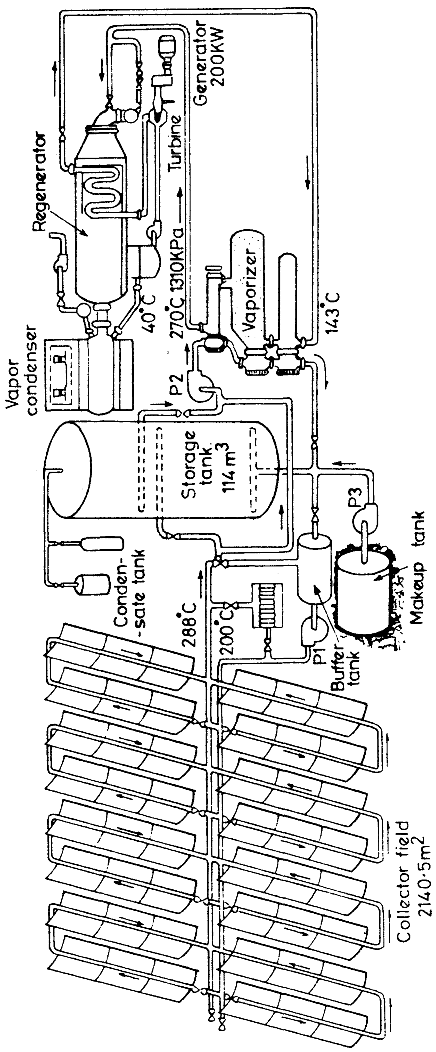


FIG 4.31 FLOW DIAGRAM OF COOLIDGE 150 KWe SOLAR POWER PLANT (From Larson[33])

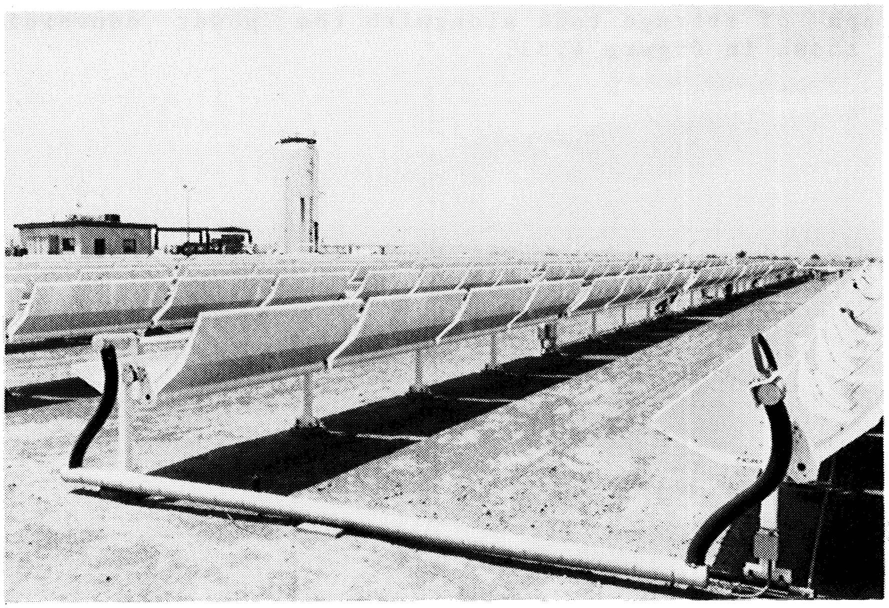


FIG.4.32 COLLECTOR FIELD ARRAY OF COOLIDGE SOLAR POWER PLANT (Courtsey D.L.Larson)

Heated caloria at 288 °C is returned to the top of the 114 m^3 insulated storage tank which is 4.16 m in diameter and 14.93 m high and provides sufficient energy to operate the power conversion subsystem for more than 5 hrs. A thermocline separates the heated caloria at the top of the tank from the cooler caloria at the bottom of the tank.

The power conversion unit consists of a heat exchanger transferring heat from heated caloria to the working fluid toluene, single stage impulse turbine made by sandstrand Corporation, synchronous generator for generating electric power, and evaporative cooling tower for condensing the toluene.

Thus basically there are three closed heat transfer loops. In the first loop warm caloria from the bottom of the storage tank is extracted, circulated through the receiver tubes of the collector field and the heated caloria is returned at the top of the storage tank. In the second loop hot caloria from the top of the tank is extracted, circulated through the vapourizer heat exchanger and returned to the bottom of the storage tank. In the third heat transfer loop, the high pressure vapourized toluene from the heat exchanger is sent to the turbine to produce shaft work, the vapour expands which is condensed in an evaporative cooling tower and then sent to the inlet of the vapourizer heat exchanger for further vapourization. The photograph of storage tank alongwith the power conversion unit is shown in figure 4.33.

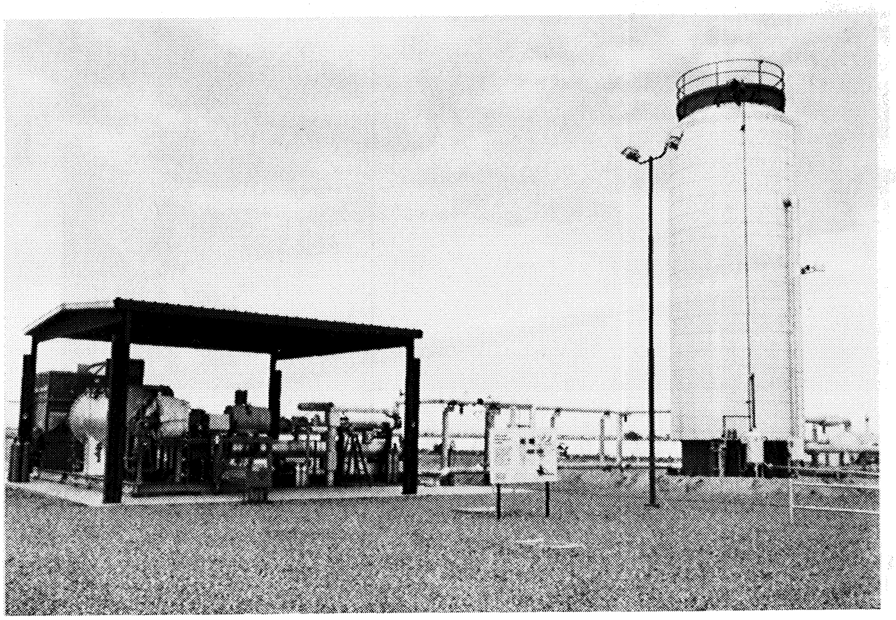


FIG.4.33 POWER CONVERSION UNIT AND STORAGE TANK
(Courtsey D.L.Larson.)

A photograph of the full coolidge solar thermal electric plant is shown in figure 4.34. An automatic control subsystem monitors and controls the tracking and collection of solar energy, flow rates of caloria and storage of heat, flow rates of caloria and heat exchanger and flow rates of toluene in the power conversion unit, and the generation and supply of electric power. The control unit also protects the solar thermal electric power plant arising due to system related anomalies or due to natural means. A natural gas fired auxiliary heater is also provided for testing purposes in case of low insolation values.

Table 4.6 Summary of the 150 KW coolidge solar thermal power plant (From Larson[33])

1. Collector	
Type	: Parabolic trough collectors (1.8 m x 3 m)
Reflector material	: Polished aluminium, aluminzed mylar (FEK-244)
Number and area of collectors	: 384 collectors in 48 groups, 2140.5 m^2
Orientatiln of collectors	: N-S axis
Receiver coating	: Black chrome
Concentration ratio	: 36
Collector fluid	: Caloria HT-43
Temperatures	: Inlet temperature = 200 °C Outlet temperature = 288 °C
Design conditions	: ϕ_i = 600 W/m^2 $\dot{m}$ = 7167 Kg/hr system efficiency = 38.6 percent
2. Storage	
Type	: Stratified liquid (thermocline) (sufficient for 5 hrs operation)
Size	: 4.16 m diameter and 14.93 m high tank (114 m^3 usable storage)
Storage medium	: Caloria HT-43
Storage temperature	: 200 °C to 288 °C
Insulation thickness	: 30 cm (Fibre glass)
3. Cooling system	
Type	: vapour condenser
Water (makeup)	: 2270 litres/hr
Condensing temp.	: 40 °C

4. Power generation

Type	:	Organic Rankine cycle
Working fluid	:	Toluene
Gross efficiency	:	20%

FIG.4.34 FULL VIEW OF COOLIDGE SOLAR THERMAL POWER PLANT (Courtsey D.L.Larson)

This plant was operated daily except during periods of collector testing, equipment modification activities and other breakdowns and repairs. During the years 1980, 1981, and 1982 the collector sub-system operated 89,93, and 98 per cent of the possible operating hours respectively. The power conversion subsystem operated about 90,97, and 97 per cent of the possible operating hours during the year 1980, 1981 and 1982 respectively. Daily data on available solar energy (compiled only that direct radiation which is more than 300 w/m^2), collected thermal energy, and electrical energy generated have been compiled for 1980, 81 and 82.

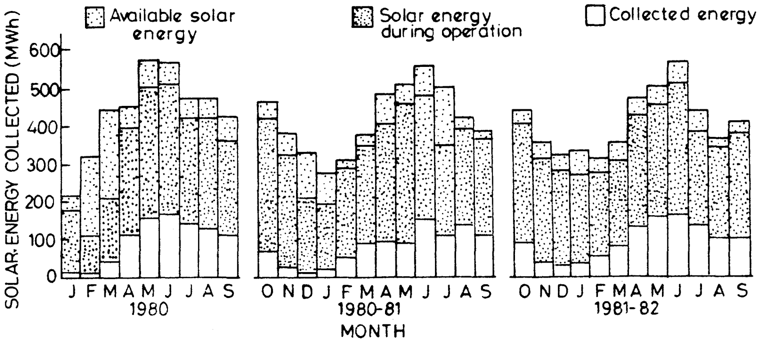


FIG.4.35 TOTAL AVAILABLE SOLAR ENERGY,RECEIVED DURING COLLECTOR SUBSYSTEM OPERATION AND COLLECTED THERMAL ENERGY. JANUARY 1980-SEPTEMBER 1982 (From Larson[33])

Monthly values of avialable solar energy and collected energy for 1981-82 is plotted in figure 4.35[33]. It is seen that maximum solar energy is available and collected in June which is the clearest month of the year. It is calculated that average monthly collection efficiency ranges from 7 per cent in winter to 27 per cent in spring and fall to 35 per cent in summer. Some collector efficiency tests are conducted on clear days near winter and summer solastice; and spring and autumnal equinox. During the year 1980, the peak collector subsystem efficiency on clear days was 14 per cent in winter, 34 per cent in fall spring, and 42 per cent in summer which is lower than the earlier estimated. This low efficiency is attributed to the low reflectivity (<60 percent) of the polished aluminium reflectors. Therefore in spring 1981, aluminized acrylic film, FEK-244, was laminated to the reflective panels of all the collectors. This reflector lining has improved the collection efficiency considerably reaching to 47 per cent in summer and 39 per cent in spring.

The monthly average electric power (KWh) produced by the plant for the years 1980,81 and 82 is shown in figure 4.36[33]. The total energy produced during the years 1980, 81 and 82 is 114930, 163410, and 178030 KWh respectively. This increase in electric power production is due to some equipment and collector improvements and operating experience. It is seen from figure 4.36 that maximum electrical power was produced in June 1982, which was 27350 KWh, and 17000 KWh in September and only 3000 KWh in

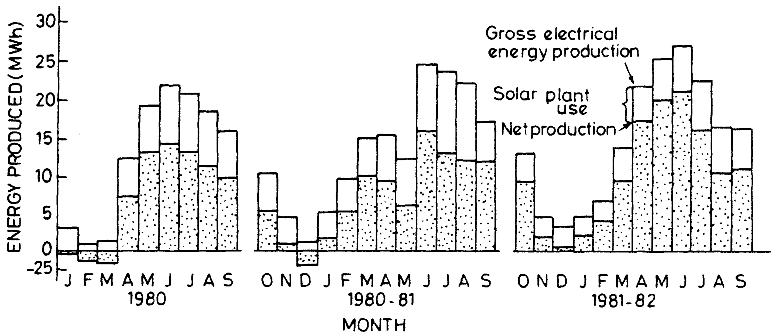


FIG.4.36 GROSS AND NET ELECTRICAL ENERGY PRODUCTION BY THE SOLAR POWER PLANT, JANUARY 1980-SEPTEMBER 1982 (From Larson[33])

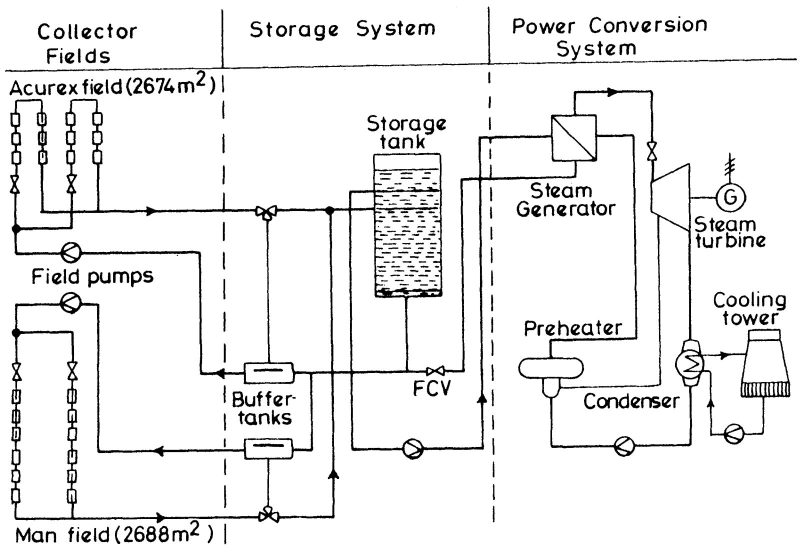


FIG.4.37 FLOW DIAGRAM OF 500 KWe SOALR THERMAL POWER STATION AT ALMERIA, SPAIN(From Kalt et al[96])

December. On a single clear June day in 1982, the system produced about 1300 KWh electricity. The average thermal to electrical energy conversion efficiency ranges from 9 to 12 per cent in winter months and 12 to 18 per cent in summer months. During January 1980 tests, the thermal to electrical conversion efficiency at 200 KW design point was 19.7 per cent. The parasitic power requirement (for running various pumps, auxiliary equipment and cooling tower operation etc.) was about 24 KW thereby showing a net cycle efficiency of 17.3 per cent for power conversion subsystem.

Several changes are made during the operation of the plant like reflector panels were lined with aluminized acrylic film (FEK-244), a new automatic tracking unit was used, etc. Several small equipment problems in the toluene pump seal,in the generator relay, in the vapourizer gasket, in the Caloria pump controller etc. were also experienced. Some incidents causing the equipment failure like toluene contamination, Caloria fire, reflector film delamination, collector no-flow overheating, etc. were also experienced.

4.8.2 Solar thermal electric generation plant of 500 KWe at Almeria, Spain[96]

A 500 KWe capacity solar thermal electric plant was[37,38,96,97] commissioned on July 31, 1981 which is built by an International Consortium and sponsored by nine countries and operated by Spanish Utility, Sevillana Electricidad y gas, under the direction of DFVLR (Space Agency) of Germany. The plant was installed on the Spanish plataforma solar in Almeria, Southern Spain with the specific objective of generating electric power using solar energy for supply to established electric grid system or in communities where electricity is not available and difficult to supply by conventional means.

The simplified flow diagram of the solar thermal plant is shown[96] in figure 4.37 and its details are listed in table 4.7. The plant utilizes two sets of line focus collectors; 2674 m^2 Accurex (USA) model 3001 type E-W oriented single axis tracking collector and 2688 m MAN (Germany) model 3/32 type two axis tracking type collector, a stratified liquid storage tank with a working volume of 114 m^3 and Santotherm 55 heat transfer oil as the storage media; oil to water Baeltz steam generator; and a stal-Laval condensing steam turbine generator. Apart from the above main subsystems there are many other systems[37] like: an automatic control system, nitrogen ullage system to protect hot oil from fire, water treatment plant, solar collector cleaning system, a computer for master control functions and data collection, a constant power supply for protection, etc. The whole system is designed around three heat transfer loops. In one loop, cold heat transfer oil is

Table 4.7 Summary of 500 KWe distributed collector solar thermal electric plant at Almeria, Spain.

1. Collector Accurax Model 3001

Type	: Parabolic trough collectors (1.83 mx3.05m)
Reflector material	: Glaverbel thin glass reflector (0.6 mm)
Number and area of Collectors	: 480 collectors in 10 parallel loops with each loop having 4 groups of 12 module collectors. (2674 m^2)
Orientation of collectors	: East-west oriented with single axis tracking
Land use factor	: 0.27
Receiver coating	: Black chrome
Concentration ratio	: 35.5
Collector fluid	: Synthetic oil-santotherm 55
Temperatures	: Inlet temperature = 225 °C Outlet temperature= 295 °C

Man Collector Helioman Model 3/32

Type	: Parabolic trough collectors (5.16mx7.96m)
Reflector material	: Glass reflector (3 mm)
Number and area of collectors	: 84 Collectors in 14 rows, 2688 m^2
Orientation of collectors	: Azimuth orientation with two axis tracking
Land use factor	: 0.32
Receiver coating	: Solartex selective coating

Table 4.7 cont.

2. Storage	
Type	: Stratified liquid (thermocline) 4.2 m diameter and 15 m high tank (114 m^3 working fluid)
Storage media	: Santotherm 55 heat transfer oil
Capacity	: 0.8 MWhe
Storage temperature	: 225 °C - 295 °C
Working fluid	: Water (steam)
3. Steam Generator	
Maximum steam output	: 4230 Kg/hr.
Steam pressure	: 2870 KPa
Steam outlet temperature at the oil inlet temperature of 295 °C	: 285 °C
4. Power Generator	
Type	: Multistage condensing axial flow steam turbine generator (Stal-Laval steam turbine)
Working fluid	: Water
Inlet steam pressure	: 2500 KPa
Inlet steam temperature	: 283 °C
Inlet steam flow	: 3767 Kg/hr
Exhaust steam Pressure	: 7 KPa
Exhaust steam enthalpy	: 2277 KJ/Kg
Exhaust steam flow	: 3215 Kg/hr
Generator rating	: 713 KVA

Table 4.7 cont.

5.Power at design point	
Solar insolation	: 4933 KW
Thermal collection	: 2580 KW
Gross electric	: 577 KW
Net electric	: 500 KW
Thermal/gross electric efficiency	: 22.4%
Thermal/Net electric efficiency	: 19.4%
Isolation/net electric efficiency	: 10.1%
6.Cooling conditions;	: Wet cooling 81 m^3 per day cooling water
7.Data acquision system and weather station	: Pyrehliometer for insolation Temperature Wind speed and direction 15 seconds intervals Averaging to 5 minutes intervals

extracted from the bottom of the storage tank circulated through the collector field and returned to the top of the storage tank. In the second loop, hot oil is extracted from the top of the tank, circulated through the oil to water heat exchanger unit and returned to the bottom of the storage tank. In the third loop high pressure steam is supplied to the steam turbine where it is expanded and cooled in a cooling tower and supplied to the inlet of the steam generator unit. A photograph of the system showing the Accurex collector field, storage tank, and the turbine housing alongwith the tower of the central receiver plant is shown in figure 4.38.

As described earlier the collector field consists of two subfields and each field can be operated seperately. One field consists of line focus parabolic trough type

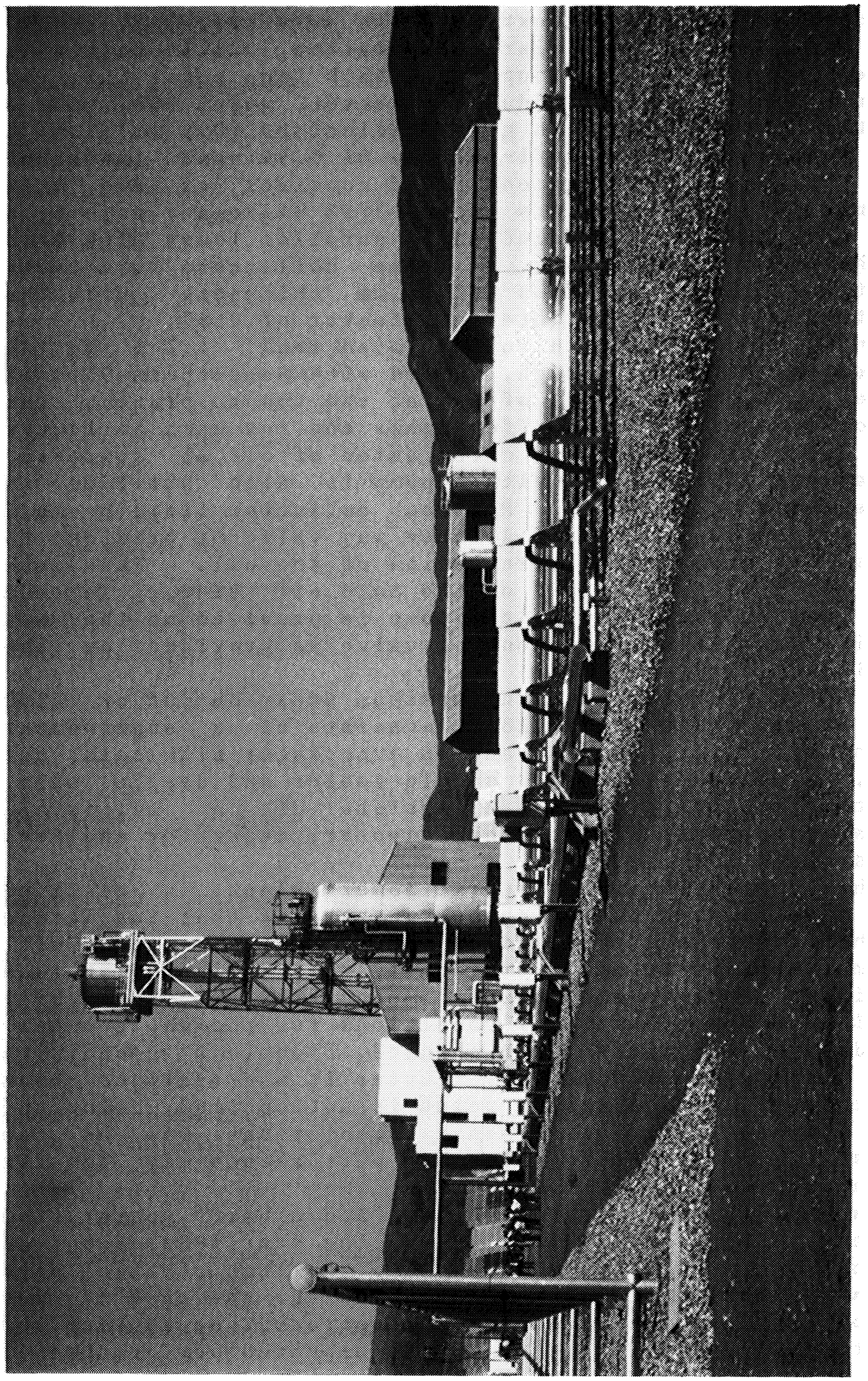


FIG.4.38 PHOTOGRAPH OF A 550 KWe SOLAR THERMAL POWER STATION AT ALMERIA, SPAIN (Courtesy Acurex Solar Corporation, California)

collectors oriented east-west with one axis tracking (Model 3001) made by Acurex Solar Corporation, California(USA) arranged in 10 parallel loops with each loop consisting of four groups of Acurex 12-module collector, model 3001. These collectors use extra thin glass reflectors (0.6 mm) on a metal support with a reflectivity of 0.94 and intercept factor of 0.95. The second field consists of two axis tracking type of collectors (model 3/32 Helioman) made by a German firm MAN and arranged in 14 parallel loops with each loop having 6 collectors. These collectors use self supporting glass troughs of 3 mm thickness with a reflectivity of 0.86 and intercept factor of 0.97.

Energy is stored in an insulated tank, 4.2 m inside diameter and about 15 m high, filled with Santotherm-55 heat transfer oil with a layer of N_2 at the top to protect the oil from oxidation. Two buffer tanks one for each collector field are provided for the circulation of oil at start-up. Oil make-up tank along with a pump is also provided to compensate any oil loss. For each collector field a main piping system is provided. A three way valve is provided to by-pass the storage for circulation of the oil. For each field, a pump is provided on the cold oil side. In the steam generator oil circuit a pump is provided on the hot oil side and steam flow control valve is provided on the cold side.

A steam generator having maximum steam output of 4230 Kg/hr at 2870 KPa and at 285 °C consists of a superheater with hot oil inlet, a drum and an evaporator with cold oil outlet, a blow down tank, sample cooler and design unit. The turbine-generator set consists of a multistage condensing turbine, a generator, ejector, deaerater and feed water storage tank.

The solar thermal electric power plant is designed using a 'design point' which means that for that insolation and other conditions expected at that moment the plant will perform the year round. Originally this design point was the winter solastice noon which means that the plant will work with better operational conditions for the whole year. This design point was later changed to equinox noon with solar insolation of 920 W/m^2. Later it was realised that the method of using a design point is not optimum. Using the design point, the performance predictions may be too optimistic and the actual average performance may be far from these predictions.

The plant was tested for one and a half years from September 1981 to May 1983. During the year 1982, about 28 per cent of the days were clear, 45 percent of the days were hazy with intermittent clouds and for the rest 27 per cent of the days the plant was not operated due to very low insolation. In table 4.8[97], the expected performance of the plant is compared with the actually measured plant

Table 4.8 Actual and expected performance of 500 KWe Solar thermal electric plant (from Kalt[97]

	Actual performance 1	Expected performance 2
Power output		
Net electric output (KWe)	456	507
Range of net electric output(KWe)	0-600	50-557
Collector's field thermal output to the storage device (KWt)	2685	2602
Acurex field thermal output(KWt)	1486	1362
MAN field thermal output (KWt)	1199	1240
Thermal power to steam generator for 500 KWe electric net output (KWt)	2953	2599
Efficiency		
Overall plant efficiency for 500 KWe net electric output(%)	9.1	10.1
Collector field's efficiency(%)	53.1	52.7
Acurex Collector field efficiency (percent)	59.1	55.4
MAN collector field efficiency (%)	48.3	50.1
Storage efficiency for 24 hrs (%)	-	92.2
Steam generator/power conversion system efficiency for 500 KWe (%)	19.1	22.2
Storage capacity		
Useful thermal energy content after charging (KWh_t)	5316	-

Table 4.8 cont.

	1	2
Equivalent electric capacity (KWh_e)	900	-
Useful thermal energy content after 24 hrs. (KWh)	4900	4160
Equivalent electric capacity (KWh_e)	830	923
Parasitic Consumptions		
Total plant consumption (KWe)	60	70
Other performance		
Minimum effective insolation for a collector field operation (W/m^2)	400	300
Cooling water consumption (m^3/day)	81	81
Land use factor	0.30	0.30

performance adjusted analytically to design conditions. It is seen from this table that actual efficiency of the Acurex collector field is 59.1 percent which is 3.7 per cent higher than the predicted efficiency. The MAN collector field has given a lower efficiency than the predicted (48.3 per cent instead of 50.1 percent) The actual overall plant efficiency was only 9.1 percent compared to the expected 10.1 percent. This lower efficiency is due to the low power conversion efficiency (19.1 percent instead of 22.2 percent).

During the commisioning and operation of the plant several management, site specific, technical, and material problems were encountered. Some of the problems which required special attention included: water removal from thermal oil, soiling of collector field, over heating and damage of some collectors, water and power failures, low insolation, turbine condenser water level control problems, breaking of reflector mirrors and receiver covers, leakage through different valves, changes and often failure of control system, steam generator leakage, defects in tracking control system, etc. All these problems were overcome during one a half years of operation and the system worked well to satisfaction later.

4.8.3. The white Cliffs (Australia) solar power station (Ref.40,98)

The white cliffs solar power station was built and put to use in March 1982 with objective to study the feasibility of paraboloidal dish system to provide electric power (25 KWe) and thermal energy (140 KW_t) to an isolated small community at White Cliffs (1100 Km west of Sydney) of 40-50 people who have no existing power supply and have an extreme and hostile climate. The main electrical load at White Cliffs consists of some 8 houses, community hall, street lights, post office, school and hospital all within 1 km from the power station. Electrical power is supplied continuously on a stand alone basis with diesel back-up.

The main functional diagram of the power station is shown[98] in figure 4.39. The power station mainly comprises 14 modular semi-autonomous parabolodial tracking collectors, a reciprocating uniflow steam engine with an alternator and a central controller giving instructions to modular units for starting, offsteering, stopping and parking. The details of the plant are shown[98] in table 4.9.

The collector field consists of two north-south parrallel rows of total 14 modular semi-autonomous parabolodial tracking collectors, each of 50.2 m diameter, 70° rim angle-rim supported on fibreglass substrate of 6 mm thick-pasted about 2300 mirrors of 2.5mm thick back silvered glass with dimensions 100 mm x 100 mm. Each dish is mounted on a frame pivoting on a horizontal axis (absorber is mounted on this axis) which in turn can rotate about a vertical axis carried on the pedestal pipe. Each collector carries its own battery supply, charged from the central plant, and each collector is tracking the sun by printed circuit motors through actuators and controlled by a Table dish-mounted sun sensor. When there is a 'Start' signal from the central control room clock, the dish starts tracking the sun all day until it receives a 'Park' signal in the late afternoon, when it parks facing horizontally south. In case of cloudy weather, the dish control system generate time pulses keeping the dish approximately facing the sun and when the sun reappears the sensor signals override these timed pulses.

In case of Windspeed exceeding 80 Km/hr, the dish automatically gets parked facing vertically. An Array of collectors is shown in figure 4.40. Each dish carries a semi-cavity tube type absorber with 160 mm diameter and 160 mm long and connected to inlet water and outlet steam tube.

4.9 Details of 25 KWe and 140 KWt solar plant of White Cliffs, Australia.

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1. Collectors

Type	: Parabolic dish collector of 5.02 m diameter.
Reflector material	: Fibreglass substrate paraboloidal shell 6 mm thick on which about 2300 back silvered glass reflectors of 2.5 mm thick and 100 mm x 100 mm are pasted. Reflectivity 0.86.
Number and area of collectors	: 14 dishes each of 19.8 m^2 operature area.
Focal length	: 1.808 m
Rim angle	: 70 deg.
Intercept factor	: 0.95
Geometric concentration ratio	: 1000
Tracking mode	: Semi-automatic, altitude/ azimuth
Absorber type	: Coil type, semi-cavity, 160 mm. diameter and 160 mm long.
Working fluid	: Steam
System pressure	: 7 MPa
Absorber pressure drop	: 7 KPa
Collector fluid inlet temperature	: 50 °C
Collector fluid outlet temperature	: 550 °C
Rated mass flow	: 50 ml/s total

Table 4.9 cont.

Design output power	: 14.0 KW_t at 1000 W/m^2
Collector efficiency	: 72.5 percent at 840 W/m^2 insolation and steam at 415 °C
Typical heat transfer coefficient from absorber	: 0.2 W/cm^2 K.
2.Storage:	
Thermal energy	: nil
Electrical energy	: Lead acid battery
3.Power conversion:	
Type	: Rankine reciprocating uniflow steam engine.
Maximum steam pressure	: 7 MPa
Maximum steam temperature	: 450 °C
Condenser pressure	: 24.5 KPa
Measured heat-to-mechanical work conversion efficiency (at 4.1 MPa and 415 °C)	: 21.9 percent
Thermal to net electric efficiency	: 9-10 percent
Mode of operation	: Stand alone - continuous + Battery, diesel

The water and steam pipes are connected through horizontal and vertical axis rotary joints to the main ducts and then to the engine feed water pump and steam is conveyed to the engine room through insulated ducts. Maximum steam pressure allowed in the absorbers is 550 °C at a pressure of 7 MPa and total water flow of 50 ml/s at an insolation of 1000 W/m . The quality of the steam is controlled by the speed of the positive displacement 3 cylinder feed water pump which is driven by a thyristor controlled servomotor acting in response to control signals from an optimizing unit which

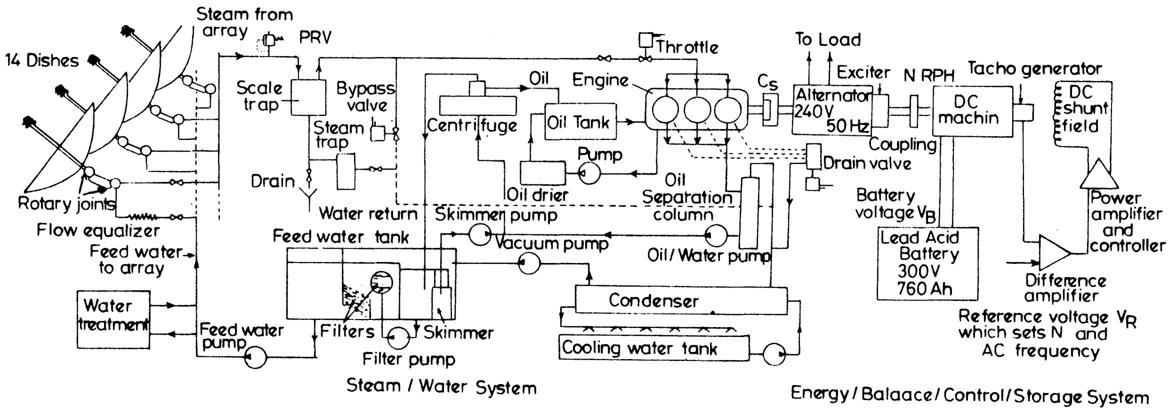

FIG.4.39 FUNCTIONAL DIAGRAM FOR THE WHITE CLIFFS SOLAR THERMAL POWER(From Kaneff[98])

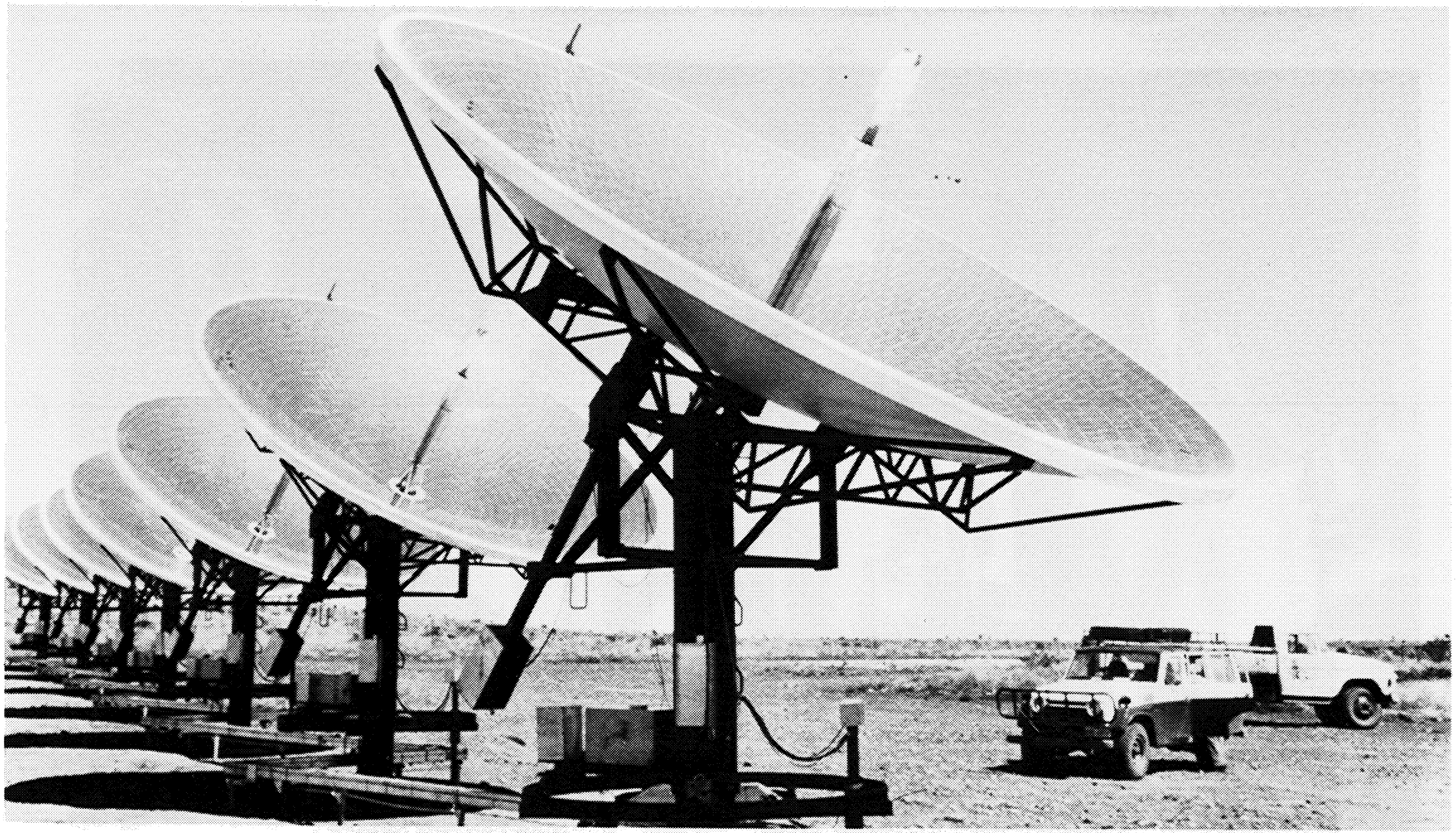

FIG.4.40 AN ARRAY OF PARABOLOIDAL DISH COLLECTORS AT THE WHITE CLIFFS(Courtesy S Kaneff, Australian National University)

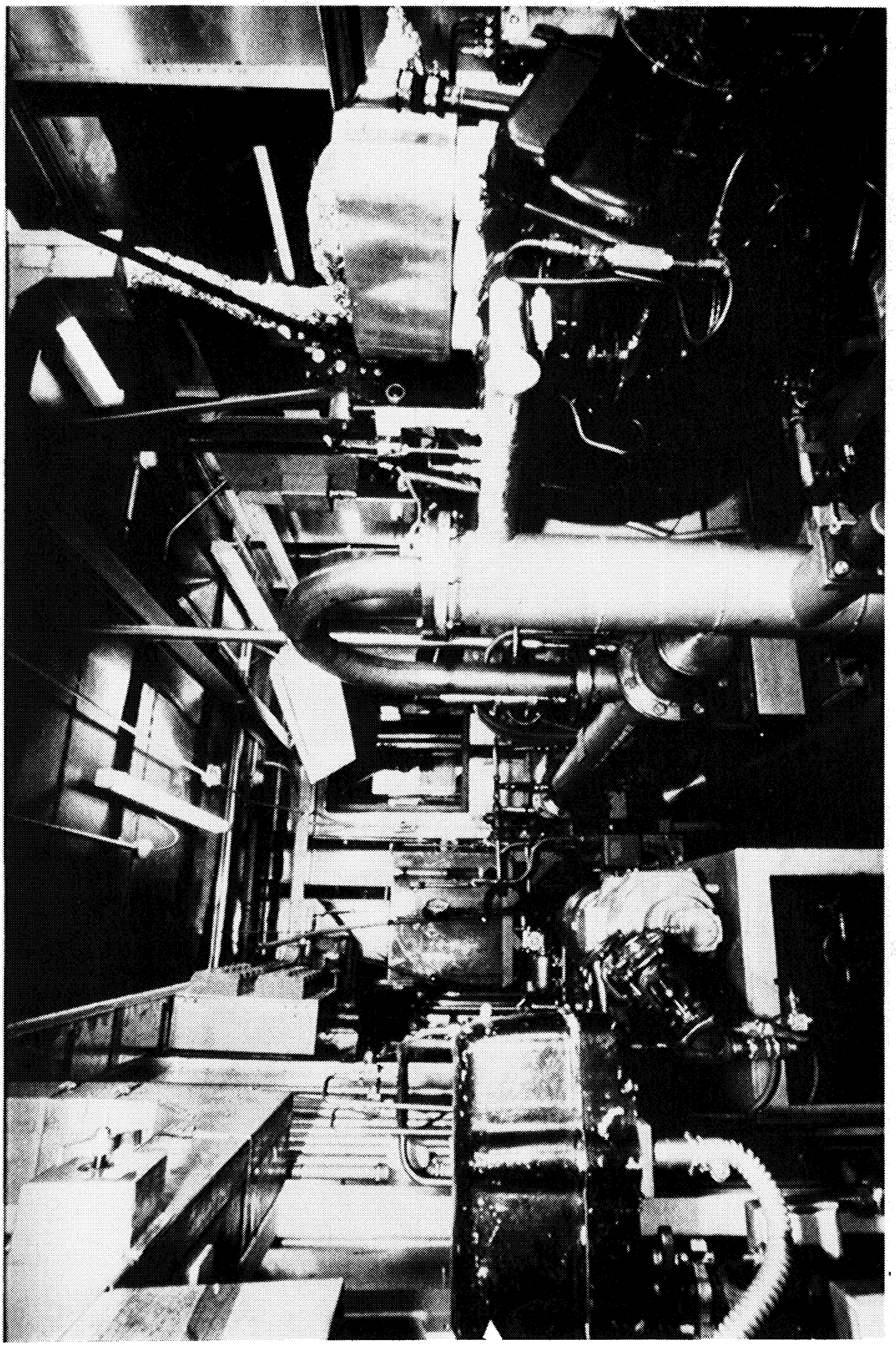

FIG.4.41 PHOTOGRAPH OF ENGINE ROOM HOUSING VARIOUS UNITS (Courtsey S.Kaneff, Australian National University)

takes account of insolation and other variables.

The steam from the solar collectors array is used to drive a high performance uniflow reciprocating steam engine. The steam engine accepts all the energy from the array till the steam quality is adequate to run the engine and till the power produce by the engine is greater than to run the auxiliaries. Since steam engine[65] of proper specifications and high efficiency was not available, a 3-cylinder lister diesel engine Model HR3 was converted for steam operation - retaining engine block, crankshaft, oil pump and filter, starter, flywheel and connecting rods, but adding General Motors (diesel model 53) pistons, rings and cylinder liners. Three new cylinder shells were made out of 420 stainless steel, each with valve seats, guides and steam chamber and 3 impulse pins were fitted to the top of each piston. The engine configurations are discussed earlier in the section 4.7.1.

After 10-25 minutes of sunrise, the cycle starts automatically, the collectors take the direction of the sun, feeding the water to the collector array, generating the steam. The bypass valve closes as soon as the steam temperature reaches to 180 °C. As the steam pressure reaches to 2.5 MPa, the throttle valve opens and the engine starts and delivers useful energy. Engine starts delivering useful power within 45 minutes from the start signal. About 30-40 minutes before the sunset the clock signal causes the array to part horizontally facing south. During cloudy or intermittent weather conditions, depending on the quality of the steam, the engine stops and starts automatically. During days of high insolation or if the power demand is less, the excess power, via dc machine is stored in lead acid heavy traction type battery. During periods of low insolation, the battery/dc machine drive the alternator and meeting the requirement. In case battery gets discharged, a back up diesel unit starts automatically to supply the load. The engine room along with various units are shown in figure 4.41. It was observed that unless the solar insolation reaches 400 W/m^2, there will be no net useful electric power.

The power plant was tested with dummy load for one complete year i.e. from june 1982 to June 1983 to assess the reliability, operation and maintenance procedures. The power plant was connected to the town load in November 1983 on a continuous, stand alone basis with diesel backup. The collectors are tested at different insolation levels and steam temperature. At a steam temperature of 400 °C and insolation levels of 1000, 800, 600, and 400 W/m^2, the energy delivered to the engine room is 165, 107.5, 57.5 and 7.5 KW respectively. It is also observed that significant amount of heat output from collector array is not used by the steam engine particularly during the morning and evening hours since the insolation is not sufficient to produce the useful net electric power. This heat and other losses are being employed for water desalination. The electric power consumed by auxiliaries is of the order of 3 KWe. The gross electric power produced at different insolation levels is also measured. The gross electric power is 29, 18, 10 and 3 KW at the insolation levels of 100, 800, 600 and 400 W/m^2 respectively thereby showing solar to net electric power conversion efficiency of 9-10 per cent. The overall efficiency can be made double fold by using better quality and precisely made glass reflectors, reducing the steam duct losses, improving the steam engine efficiency, improving the generator efficiency, and optimizing the dish dimensions.

It has been reported[98] that except of cleaning of dishes and various other components due to dusty conditions of white Cliffs which otherwise may reduce the thermal performance upto 20 per cent, there has been no maintenance and operational problems for two years. The system has proved to be of a level of sophistication well able to be handled by local personnel with largely automotive and agricultural machinery type skills.

4.8.4 Central receiver electric plant of 1 MWe capacity (EURELIOS)[49,100])

A project for constructing 1.0 MWe helioelectric power plant was sponsored by the commission of the European communities (CEC) in the year 1975. The site selected for the erection of the plant is some 40 Km northwest of Catania, Sicily at the village of Adrano (latitude 37.64 N) longitude 14.80 °E). The plant has been built by an industrial consortium consisting of:

- Messerchmitt - Bolkow - Blohm (MBB),F.R. Germany
- ANSALDO SpA and Ente Nazionale per 1' Energia Electtrica (ENEL),Itlay
- CETHEL (Combining Renault, Five-Cail-Babcok, Saint-Gobain Point-a-Mousson and Heurtry S.A.), France.

These firms alongwith CES completed the design specifications of all the subsystems of the plant including the testing of the prototype models by November 1978 and the construction completed in December, 1980. In April 1981, the plant was fully tested and started delivering power to the Italian Electricity Generating Board grid. The plant is designed to supply 1.0 MWe power at an insolation of 1 KW/m^2 at equinox noon at the site of Adrano/Sicily.

The power plant as is shown schematically in figure 4.42 and photo in figure 4.43 is based on the principle of central receiver and consists of large field of heliostats concentrating direct solar radiation on to a receiver mounted on the top of tower converting water into high pressure steam which is used to run a turbine coupled to an alternator. The electrical energy produced is fed to the

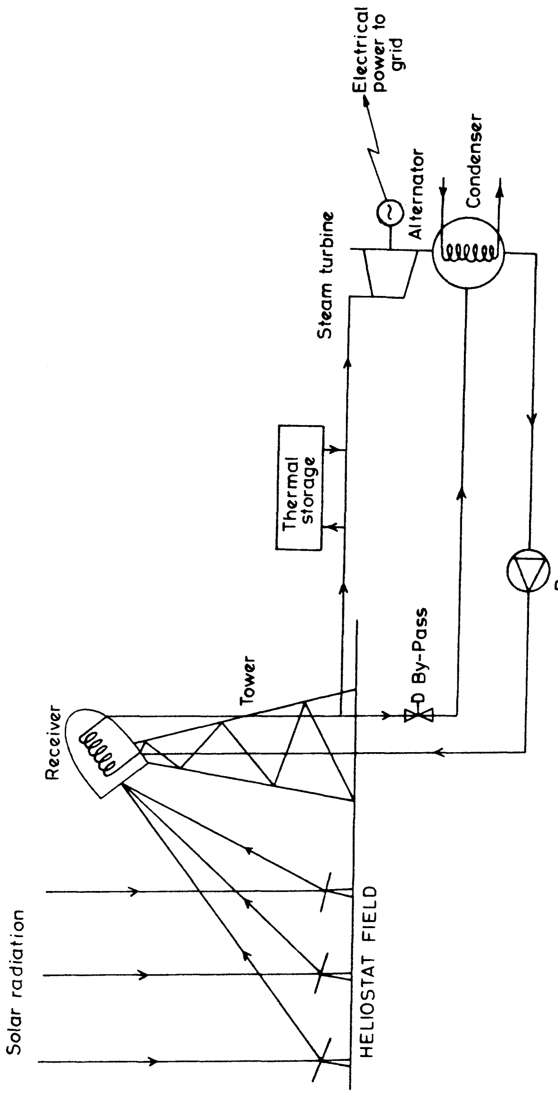

FIG.4.42 SCHEMATIC OF EURELIOS CETRAL RECEIVER SYSTEM AT ADRANO, ITALY

existing electric grid. A thermal buffer storage is used so that the plant can continue to operate for a maximum period of 30 minutes in case of cloud cover and a bypass is used for starting and shutdown operations. Table 4.10 shows the details of the EURELIOS.

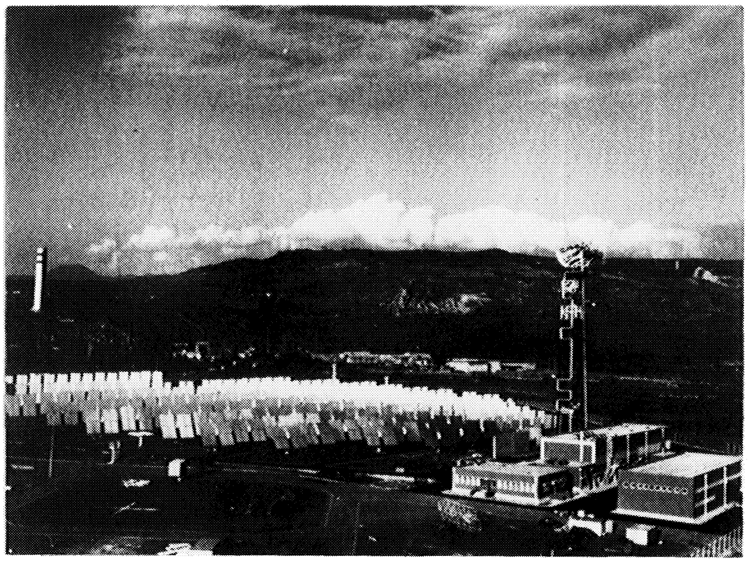

FIG.4.43 PHOTOGRAPH OF CENTRAL RECEIVER POWER PLANT OF 1MWe CAPACITY AT ADRANO, ITALY (Courtesy J.Gretz)

Two different types of heliostats each having roughly equal 'subfields' supplied by CETEL, France and MBB, Germany which are considerably different in size, are placed within two subfields divided by a line going north-south : 70 CETEL heliostats each of area 51.8 m^2 located in the western part, and the 112 MBB heliostats each area 23 m^2 in the eastern part. Each heliostat has its own microprocessor-based local electronics unit which is controlled by a central unit. The central unit can control a single heliostat, a group of heliostats or a subfield of heliostats. The heliostat field is designed in such a way so as to avoid the shadowing and blocking effects at equinox noon. The MBB heliostat uses 16 mirrors each of 1.2 m x 1.2 m x 3 mm back silvered float glass (average reflectivity 85 per cent) mounted on a frame with a swiveling pedestal. The CETHEL heliostat consists of 8 modules of 6 mirror strips. Each mirror strip is 1.8 m x 0.6 m with a thickness of 6 mm and is silvered from backside showing on average reflectivity of 80 percent. The heliostats can be driven at two speeds:

Table 4.10 Main characteristics of 1 MWe solar power plant (EURELIOS) of Andrano, Sicily, Italy

1. Heliostats			
Type	:	Two types (CETHEL and MBB) with a total area of 6202 m^2, two axis controlled, focussing	
Configuration	:	CETHEL	MBB
		8 Modules of 6 mirrors of 1.8 x 0.6 m with an area of 51.8 m^2	16 mirrors of 1.2 x 1.2 m with an area 23 m^2
Number of heliostats	:	70	112
Total mirror area	:	3626 m^2	2576 m^2
Mirror type	:	6 mm backsilvered float glass	3 mm backsilvered float glass sandwiched
Reflectivity	:	0.80	0.85
Overall inaccuracy	:	4 mrad	4 mrad
Panel dimensions	:	8.84 m x 7.34 m	5.63 m x 5.01 m

Table 4.10 cont.

	Panel rotational speeds (azimuth)	: 12° per min. fast, 0.6° per min. slow	4.79° per min. fast, 0.48° per min. slow
	Panel rotation speeds (elevation)	: 12° per min. fast, 0.6° per min. slow	6.28° per min. fast; 0.63° per min. slow
	Range of rotation	: $\pm 12^\circ$ Az+75°-$\pm 120^\circ$ Az, $\pm 90^\circ$ El	$\pm 90^\circ$ El
	Position of heliostats	: West field	East field
	Spacing of helostats	: 11.7 m	7.8 m
2.	Receiver/Tower		
	Type	: Once through cavity type receiver, 4.5 m aperature at 55 m height, 110° inclination	
	Coil length	: Two identical parallel and independent branches each of 482 m long and 55 mm diameter	
	Volume	: 3.4 m^3	
	Outlet steam conditions	: 512 °C, 6.48 MPa, 4860 Kg/hr	

Table 4.10 cont.

3.	Steam cycle	
	Feed water temperature at receiver inlet	: 36 °C
	Cooling water temperature	: 25 °C (maximum)
	Nominal power	: 1.2 MW (Mechanical) with steam at 510 °C, 6.8 MPa
	Connection	: Direct connection of turbine with the receiver (No heat exchanger)
4.	Thermal Storage	
	Capacity	: Steam 300 KWh, Hitec 60 KWh capable to operate plant for 30 minutes at reduced power
	Water storage	: Pressurized water storage for 4300 Kg vapor from 1.9 MPA to 0.97 MPa
	Molten salt storage	: Two tanks, each of 1 m^3 volume containing 1250 Kg HITEC salt (an eutectic mixture: 53% KNO; 40% $NaNO_2$; 7% $NaNO_3$) and two steam (salt) heat exchanger for 1.9 MPa, 480 °C and 410 °C steam temperature

Table 4.10 cont.

5.	Turbine	
	Type	: Rankine cycle 7-stage steam turbine
	Steam conditions at inlet	: 6.8 MPa, 510 °C, 5200 Kg/h
	Steam conditions at exhaust	: 6 KPa, 36 °C
	Turbine speed	: 8195 rpm
	Turbine output at turbine/generator joint	: 1.2 MW
6.	Alternator	
	Type	: GSN 500 Y 4 Alternator (3 phase) made by ANSALDO with 1500 KVA at 0.8 phase
	Speed	: 1500 rpm
	Efficiency	: 94.9 per cent at full load

fast speed to reach a desired direction and slow speed to track the sun.

The solar receiver is placed on the top of 55 m high steel tower, the centre line being inclined downwards 22^{o} from horizontal towards the heliostat field and is constructed by ANSALDO which is based on the results and experience gained by Prof.Francia with the 100 KW plant. The receiver is a once-through circular cavity type steam generator, conical in shape. It consists of two parallel tubes, through which pressurised water flows, rolled up in a coil to form the conical internal wall of the cavity into which the solar radiation is focussed by the heliostats. Each tube is of about 482 m long and the pipe work is divided into four sections:

1. An economiser, an open bottom less basket like structure in the centre of the cavity which is designed to heat the incoming feed water to about 200-280 °C.
2. A truncated conical evaporator section, which makes the wall of the receiver and is designed to boil the feed water without superheating.
3. A truncated conical primary syperheater, where the steam is superheated between 350 to 400 °C and is mounted above the evaporator.
4. A cylindrical secondary superheater, consisting of circular and straight pipes forming a honeycomb like structure brining the steam to its final state generally at 512 °C and a pressure of 6.48 MPa.

The tubes of the receiver are blackened and finned so as to absorb maximum solar radiation and behaves like a black body. The heat loss from the receiver is reduced by using an anti radiating structure made of pyrex sheets (transparent only to solar radiation) located from the inside over the tubes.

The outlet steam temperature is maintained constant inspite of the solar flux variation by using four automatically controlled direct contact attemperators spray valves, two in each branch.

The steam cycle is single superheating Rankine cycle consisting of a receiver boiler, a steam turbine, a condensor and other appropriate pumps, valves safety system, controls, etc. A thermal storage is used as an auxiliary system designed to operate the plant upto 30 minutes without solar insolation in order to protect the turbine against thermal shocks. Under clear sky conditions, this system takes about 45 minutes to provide steam at a temperature of 350 °C which is the temperature required to operate the turbine from the feed water temperature of 25 °C. The steam produced during this period is taken through a bypass valve and attemperator/desuperheater into a Flash tank before sending it to the condensor and pumped through the system again. The upsteam pressure is maintained by adjusting the bypass valve which is done automatically. The water level and pressure in the Flashtank is also automatically adjusted. The condensor is an evacuated four-pass water cooled unit. As soon as the steam temperature reaches to the design level it is automatically admitted to the turbine. The exhaust of the turbine is passed directly to the condenser.

A 7-stage impulse steam turbine runs at a speed of 8195 rpm and through the reduction gear is connected to the alternator with a flexible joint providing the alternator a speed of 1500 rpm. The turbine receives steam at a temperature of 510 ^{o}C and at a pressure of 6.8 MPa.
The thermal storage system consists of two parts and provide steam at a temperature of 510 ^{o}C. The storage system consists of a tank providing saturated steam at a pressure decreasing from 1.9 MPa to 0.7 MPa and a molten salt storage subsystem containing HITEC with two steam/salt heat exchangers and two salt storage tanks superheating the saturated steam from the hot water tank upto 410 ^{o}C.

The control is an essencial part of the whole system. There is a master control centre whose main function is to control and supervise the whole plant including receiver, steam cycle, heliostat field, and storage. Basically the plant control system consists of two parts : the heliostat control system and the turbine and the steam cycle control system. Each heliostat is automatically controlled and can take any of the three positions - storage, standby and tracking mode. The turbine and steam cycle control system control: steam temperature at the outlet of the receiver, steam pressure at the outlet of the receiver and the steam-water transition point. The bypass circuit in case of start up and shutdown of the plant is controlled manually.

The power plant supplies electricity to the local grid. In the begining, to start the plant, electricity from the grid is used to start the auxiliaries of the plant through transformers and used to operate the heliostat field, the feed water system, receiver, turbine, and alternator. After the plant comes to its full operation, the auxiliaries consumes a small fraction of the generated power and the balance is fed to the 20 KW grid through breakers.

4.8.5 Carrisa Plain Solar Photovoltaic power plant(Ref.101)

Perhaps the world's first largest, commercial and impressive example of the photovoltaic plant of 1.0 MWe capacity was built in December, 1982, by ARCO Solar, Inc. on Southern California, Edison (SCE) property adjacent to the utility's Lugo substation at Hesperia, California. On November 14,1983 a 6 MWe plant located on the Carrisa Plain in San Luis Obispo County, California, was interconnected to the Pacific Gas and Electric Company (PG&E) grid. The main objective of this commercial plant is to show the feasibility of the photovoltically generated electricity to PG&E.

The 6 MWe photovoltaic power plant is located at 35° N latitude on an elevation of 615 m on PG&E property adjacent to a 115 KV transmission line. The main components of the power plants are : photovoltaic modules to produce electricity,, two-axis trackers to track the sun, investers to convert DC into AC,, switch gear to direct flow of current, and distributed data acquision components. The facility utilizes 756 computer controlled two axis trackers, each tracker with 128 photovoltaic modules augmented by reflective glass panels. The cells used are single crystal silicon solar cells. The 65 hactare array field is divided into nine independent segments, each with 84 two axis trackers. It has been estimated that by tracking the photovoltaic module, upto 40 per cent more kilowatt hours on the average over a year's time can be produced against a fixed mounted module. By using laminated glass reflectors, an increase in power output upto 50 per cent on an average yearly basis is expected. A close up view of a photovoltaic module alongwith plane glass reflectors is shown in figure 4.44. A view of tracker and photovoltaic modules is shown in figure 4.45. Figure 4.46 shows the view of array of photovoltaic modules and trackers.

The facility uses the third generation dual axis tracking with central pedestal unit of 95 m^2 with independent azimuth and elevation drive unit. The power requirement is about 0.4 KWh per tracker. Each tracker contains eight 1.22 m x 4.88 m panels of unframed 41 W square cell laminates, and 4.88 x 4.88 panels of laminated reflective glass mounted at a 60° angle. Solar tracking is done using microcomputer based systems using a special clock/calender program without the requirement of sun sensors. This tracking accuracy is found to be adequate using flat plate photovoltaic panels. During periods of night or high wind velocity, trackers are driven to a vertical stow position or horizontal stow position respectively.

Fig. 4.44 Close up view of solar photvoltaic module at Carrisa Plains California. (Courtesy of M.C.Recchuite, Arco Solar Inc.)

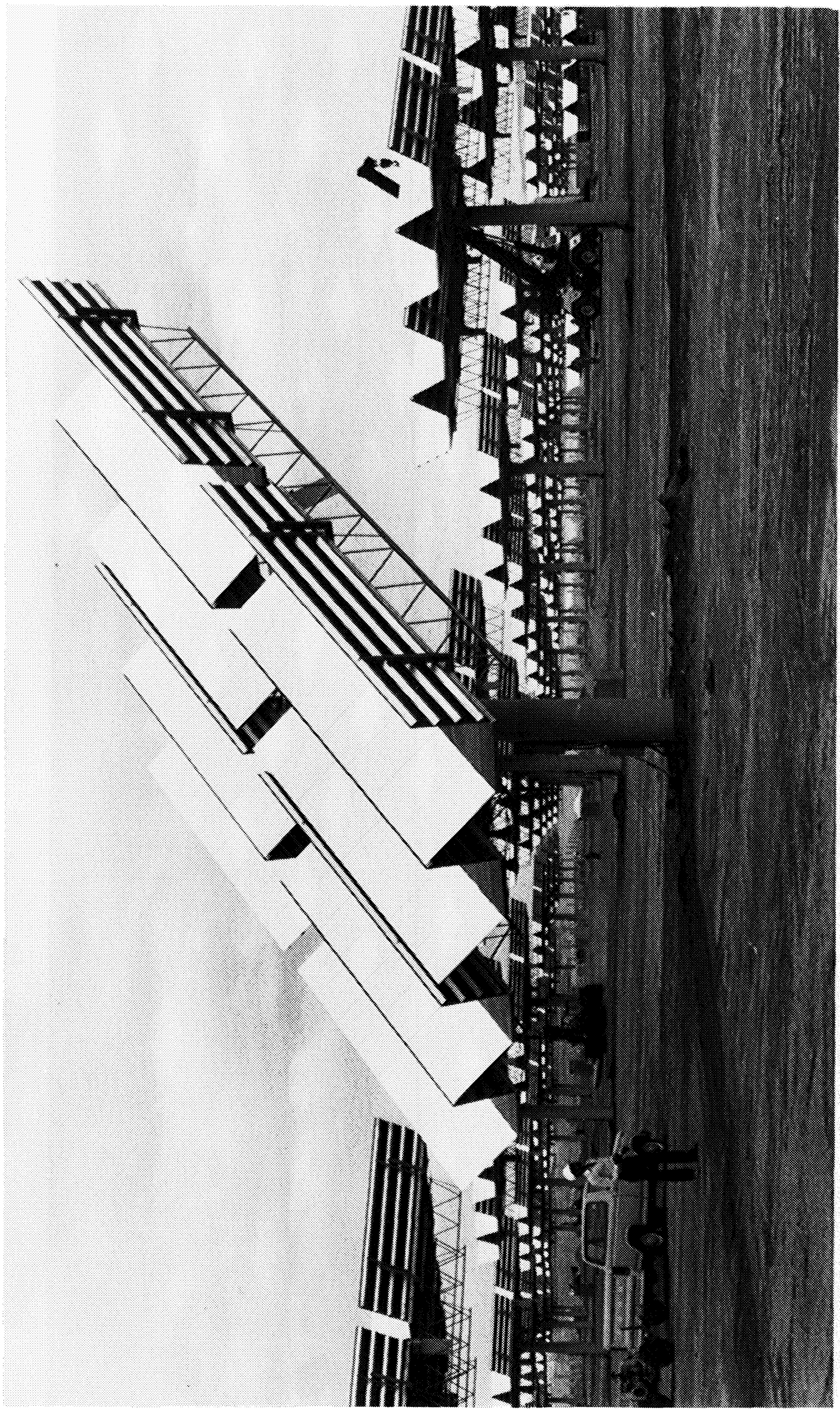

FIG.4.45 CLOSE-UP VIEW OF A TWO-AXIS TRACKER WITH PHOTOVOLTAIC MODULES AT CARRISA PLAINS, CALIFORNIA (Courtsey of M.C.Recchuite, Acro)

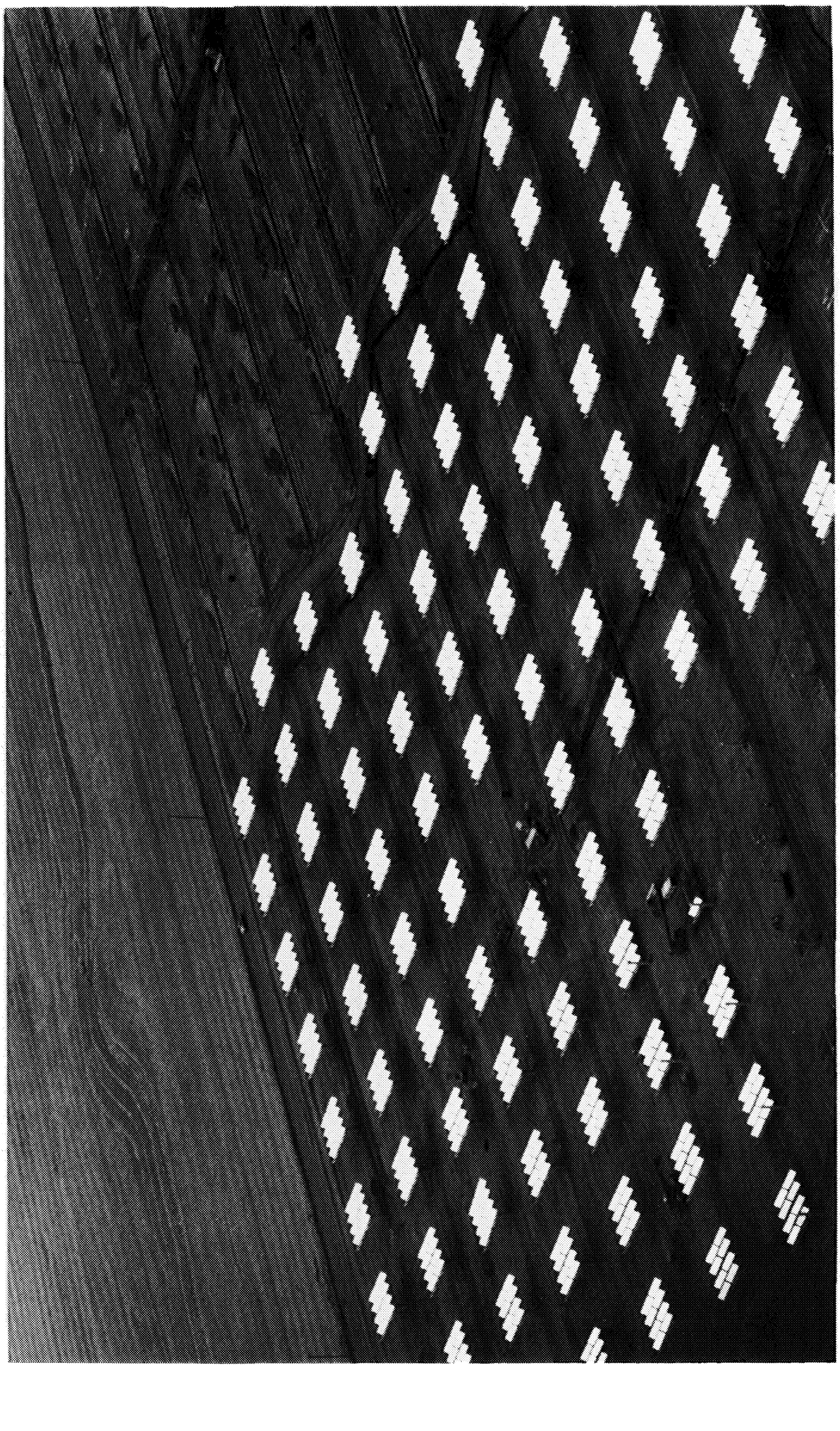

Fig. 4.46 Photovoltaic array field at Carrisa plain, California.
(Courtesy of M.C.Recchuite, Arco Solarr Inc.)

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CHAPTER - 5

SOLAR REFRIGERATION AND AIR-CONDITIONING

5.1 INTRODUCTION

Solar Energy can be used for producing cold either for cooling of buildings (generally known as air-conditioning) or for refrigeration required for preserving food. Solar cooling appears to be an attractive proposition due to the fact that when the cooling demand is more the sunshine is strongest. This, alongwith the necessity for providing thermal comfort for people in hot areas of the world, and for providing food preservation, may be the motivating factors in continuing research and development in the field of solar cooling systems. Although considerable work on solar cooling system has been done for the last three decades, due to its complexity, both in concept and in construction, the utilization and commercialisation of solar cooling is not as widespread as other solar energy applications like solar water heating and solar space heating. However, if solar cooling of buildings is combined with the solar heating then the combined solar cooling and heating systems can become economical. Similarly solar refrigerators or cooled space (cold storage) can be provided economically for preserving essential drugs and food in isolated localities.

In recent past several review articles[1-18] have been published on solar cooling methods and devices listing of the successes and failures of the systems. There are several ways of providing cold using solar energy such as:

* Using the absorption cycle with liquid absorbents such as LiBr - H_2O, H_2O, -NH_3, LiCl - H_2O, NH_3 - $LiNO_3$,R22 - DMF, NH_3 - NaSCN.
* Using the absorption cycle with solid absorbents such as: $CaCl_2$ - NH_3
* Using adsorption cycle with solid absorbents such as: Silicagel - H O, Zeolites - H_2O
* Using the vapor compression cycle employing a solar powered Rankine engine.
* Using the vapor compression cycle with the compressor driven by electricity from photovoltaic panels
* Nocturnal passive cooling

Several prototype systems based on some of the above principles have already been made and demonstrated but these are still under development to be dependable and commercial. The choice of a particular system not only depends on its economics but will depend on local factors such as climate, availability of cooling water, auxiliary energy source, and the type of collector available. The temperature limitation of solar energy collectors alongwith the need of a suitable heat storage device makes the solar cooling system more costly and bulky.

A solar air conditioning system is complicated and will consist of many components, the major ones being the field of solar collectors, a heat storage device, a solar cooling device (based on absorption or Rankine cycle), a cold storage device, a heat rejection device, air handling system, etc. as shown schematically in figure 5.1. A simple flat plate collector or evacuated tube collector or concentrating collector, depending on the temperature requirement, can be employed to heat the heat transfer fluid which is used to operate the cooling device. A part of the heat can be stored in the storage unit. The heat collected from the building is rejected to the atmosphere using a cooling tower or any other suitable heat rejecting device. If air is cooled by the cooling device then it is directly supplied to the building to be cooled or if chilled water is produced then it is circulated through fan coil units and a part of a chilled water is stored for use when the cooling device is not in operation.

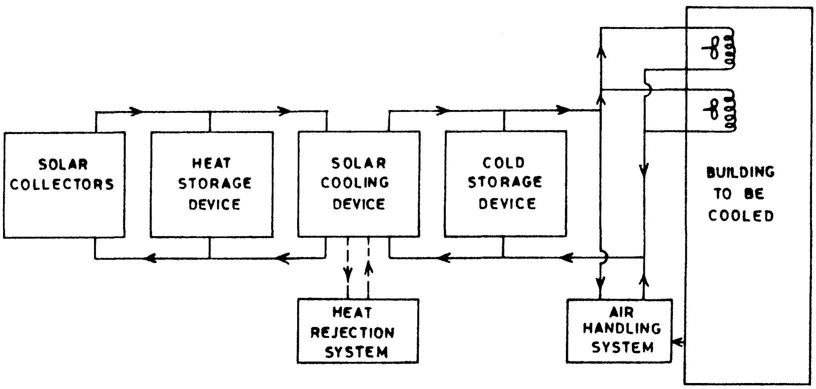


FIG.5.1 SCHEMATIC OF SOLAR COOLING SYSTEM

The performance of a cooling process is judged from its COP (coefficient of performance), which is the ratio of the amount of cooling to the energy input. The overall COP for a Rankine cycle operated solar cooling system is of about 0.3 to 0.4 which very much depends on the solar collector efficiency. The main advantage of the solar Rankine vapor compression cooling process is that it can be used in the heat pump mode also, and for electricity generation as well when cooling is not required. Moreover, the system may be designed for any operating range of temperatures with minimum pumping operations. The main problems with the system is with the controls during variable solar insolation. Many solar Rankine vapour compression cooling systems have been designed and made with different capacities, and tested for performance, but more research and development is required to improve performance and reliability.

The vapour compression cooling process operated by photovoltaic panels gives a COP in the range of 0.25 to 0.35 due to lower solar cell efficiency. This system can also be used in the heat pump mode, and the electricity can be used for other applications when cooling is not required. Here no auxiliary pumps are required. The main problem with the system is of its very high cost due to low cell efficiency and high cost of photovoltaic panels and also due to need of a costly electrical storage system . The cooling system is quiet, reliable and technically feasible.

The cooling system based on absorption closed cycle gives a COP of about 0.10 to 0.20 depending on the collector efficiency. The advantage of this system is that it can be used with low grade heat (even waste heat can be used) and is very quiet in operation. In the absorption cooling system some auxiliary power is required to drive fans and pumps. The system is uneconomical for places where heating is not required. Commercial systems based on the absorption principle are available, but more research and development is required to improve their performance, reliability and economics.

Cooling systems based on adsorption cycle are simple, quiet in operation, and operate at a COP of about 0.2. Here also auxiliary power is required to drive fans and pumps. Some experimental systems are made based on adsorption cycle but considerable research and development is required to improve performance and reliability. Commercial systems based on the cycle are not available.

Solar passive cooling concept is not new and is quite reliable and economical. No auxiliary power is required if a little discomfort is tolerated. Solar passive cooling concepts can be applied to existing buildings (retrofit systems) or in new buildings where there is a scope for change. These concepts are slowly gathering momentum due to the shortage of conventional energy.

In this chapter the thermodynamic principle of solar cooling, types of solar cooling systems, design and simulation studies, and some commercial solar cooling systems will be described.

5.2 CARNOT REFRIGERATION CYCLE

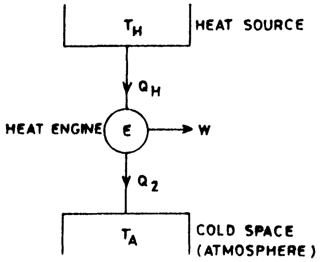
The Carnot refrigeration cycle shows the highest coefficient of performance (COP) between the two temperature limits and is the reverse effect of the heat engine. The thermodynamic principle[19,20] of a refrigeration cycle should be understood before selecting a particular cycle for use with solar energy. In a Carnot heat engine as shown in figure 5.2 (a), Q_H amount of heat at temperature T_H from solar collector is supplied to the heat engine E. The heat engine converts parts of heat into work W and rejects the rest Q_2 to atmosphere at temperature T_A. Therefore, the thermal efficiency of the engine is

$$\eta_c = \frac{\text{work done}}{\text{heat supplied}} = \frac{W}{Q_H} = \frac{Q_H - Q_2}{Q_H} = 1 - \frac{Q_2}{Q_H}$$

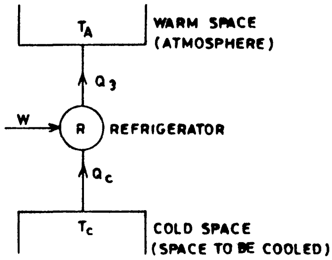
$$= 1 - \frac{T_A}{T_H} \qquad (5.1)$$

In an ideal refrigerator which is a reverse of heat engine as shown in figure 5.2(b), the work W is used by refrigerator (machine) R to withdraw Q_c amount of heat at temperature T_c from the space to be cooled and rejects Q_3 amount of heat to the atmosphere at temperature T_A. The refrigeration cycle is shown on the temperature entropy diagram in figure 5.2(c). The upper portion i.e. rectangle 1-2-3-4 gives the net work done and shows : 1-2 isothermal addition of heat, 2-3 adiabatic compression, 3-4 isothermal rejection of heat, and 4-1 adiabatic expansion. The process 1-2 is the refrigeration step where heat is withdrawn from the cold space. The other processes are meant to reject heat from cold space to a high temperature heat sink (atmosphere). Thus the area below line 1-2 shows the useful refrigeration, and the area below the line 3-4 represents the heat rejected to the atmosphere. Therefore, the coefficient of performance of refrigeration cycle is given as:

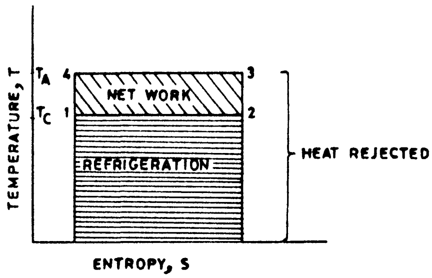
$$\eta_R = \frac{\text{useful refrigeration}}{\text{Net Work}} = \frac{Q_c}{W} = \frac{T_c}{T_A - T_c} \qquad (5.2)$$



(a) Representation of heat engine



(b) Representation of ideal refrigerator



(c) Temperature-entropy diagram of ideal refrigerator.

(c) Temperature-entropy diagram of ideal refrigerator

FIG.5.2 IDEAL REFRIGERATOR

Therefore the coefficient of performance of combined ideal heat engine/refrigerator is:

$$COP = Q_C/Q_H = (T_c/T_H).(T_H - T_A)/(T_A - T_c) \tag{5.3}$$

The above expression is true for ideal systems only but shows the limitations for producing cold using solar energy and sets the upper limit for COP. The engine efficiency will be low if solar energy is used as heat source because of low value of source temperature T_H. This lower engine efficiency will further lower the COP of the combined engine/refrigerator system. Therefore, to have higher COP, the source(solar collector) temperature (T_H) should be as high as possible and the heat rejection temperature T_A should be as low as possible. Therefore, concentrating collectors with high T_H as the heat source and cooling water as the heat sink are preferred.

Cooling systems are broadly divided into two categories: the closed cycle and the open cycle. In the closed cycle cooling systems there are two separate loops : one for the refrigeration process and second for transferring heat from load. Both the loops are closed and use some kind of heat exchanger. Each loop is optimized to operate at maximum efficiency at low cost. While in the open cycle cooling process, the cooling is done by controlled dehydration and evaporation which is simple, and eliminates the interfacing heat exchanger. Although there are several ways of producing cold using solar energy as discussed earlier but the most reliable and preferred ways are two : Rankine cycle and absorption cycle. There are some definite advantages for the absorption cycle such as:

(i) The main advantage of an absorption refrigeration cycle is that in this method the amount of mechanical work required is small. The operation is quiet and maintenance cost low.

(ii) Although the heat input required in absorption refrigeration cycle is many times more than the mechanical work required in a vapor compression cycle, but if the heat is produced at a cheap rate (such as solar heat), then the absorption cycle will be economically attractive.

(iii) The absorption unit can be made of very large capacity (>1000 tons)

(iv) At reduced loads, the absorption system is almost as efficient as at full load.

(v) The exhaust steam from the turbine can be used for heating in winter and cooling in summer using the absorption cycle.

(vi) The presence of liquid in vapour leaving the evaporator will not have any deterimental effect in case of vapour absorption machine while in the vapour compression system the vapour has to be superheated.

(vii) As the desired evaporator temperature drops, the space requirement for absorption machines becomes favourable.

(viii) The absorption machine will work even at reduced evaporator pressure without any change in its COP.

5.3 ABSORPTION REFRIGERATION

5.3.1. History

Solar energy operated absorption airconditioners are now most widely used today, and are of two types : intermittent and continuous. There are four major components in an absorption cooling device : generator, condenser, evaporator and absorber. In an absorption cycle two working fluids - a refrigerant and an absorbent is used.

In the intermittent type cooling, there are two separate operations: regeneration and cooling taking place at different times. Here the generator and absorber are combined and condenser and the evaporator are combined. In the generator/absorber unit, the refrigerant rich absorbent solution is heated, and the refrigerant vapour is condensed and stored in the condenser/evaporator unit. In the cooling process the liquid refrigerant evaporates from the evaporator/condenser unit producing a cooling effect, and is re-absorbed by the absorbent in the generator/absorbent unit. Here the cooling effect is produced discontinuously and is suitable for intermittent sources of energy like solar energy. For most applications continuous cooling is required such as for cooling of buildings where the energy source would be a solar collector-storage-auxiliary system. The continuous absorption cycle is more reliable where refrigeration and regeneration take place simultaneously producing a continuous effect.

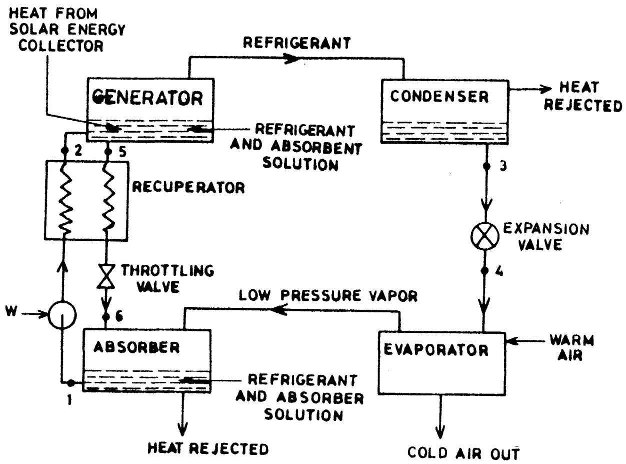
The absorption cooling is based on the principle that some refrigerants combine chemically with an absorbent to release heat during absorption, while they absorb heat during evaporation. The absorption cycle which is one of the oldest cycles for producing a refrigeration effect, differs fundamentally from the vapour compression cycle in the method for compressing the refrigerant. In the absorption system, the compressor is replaced by a generator, as absorber and a small pump. The principle of absorption refrigeration was first demonstrated by Faraday early in

1825. Later Ferdenand Carre, a Frenchman took a patent for an absorption system in 1860 in United States. In the year 1862, Mignon and Rouart built the first continuous ammonia-water absorption cooling machine and demonstrated it at the London exhibition. Later Carre[21] and Buffington[22] suggested the use of solid absorbents for absorption cooling. Earlier absorption cooling machines were operated with LPG, natural gas or kerosine as the heat source in places where electricity was not available. Later with the abundant availability of electricity, and the invention of vapour compression cooling machine, absorption cooling machines were not used. Recently, with the renewed interest in solar energy and the use of other low grade heat for producing cooling, absorption cooling machines have again come in to the market.

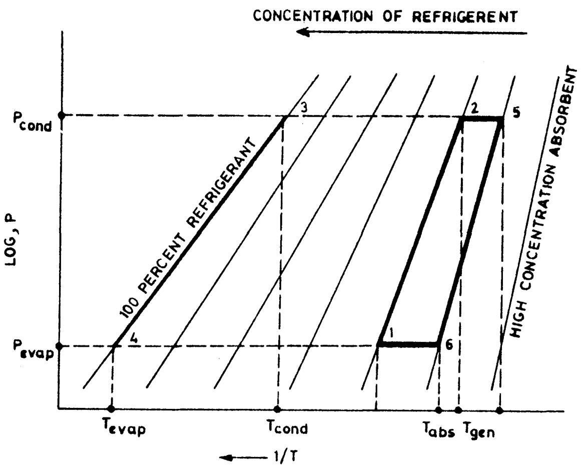
Some basic studies on absorption coolers are conducted by Buffinton[23], Trombe and Foex[24], Eisinstadt et al[25], Chinnappa[26], Duffie and Sheridan[27], Swartman and Swaminathan[28] and Sargent and Beckman[29] who have clearly shown that flat plate collectors can be used for producing cold. The choice of a suitable working fluid (combinations of refrigerant and absorbent) effects the COP of the absorption system considerably. Several papers on suitable refrigerant-absorbent combinations are published by Mansoori and Patel[30], Buffington[23], Hainsworth[31], Agarwal et al[32], Iyer et al[33], Ferreira[34], Tyagi and Shankar [35], and Macriss[36]. Extensive experiments on solar heating and cooling of buildings using absorption cycle have been conducted by many workers and the pioneers in this field are ward and Lof[37], Ward et al[38, 39], Namkong[43], diamond et al[41], Jacobsen[42], Van Hatten and Actis Dato[43], and Sheridan[44]. Based on the experiences and successes of experiments, several firms started manufacturing solar airconditioning and refrigeration machines. A few typical examples are described by Anderson[45], Simmons and Wahlig[46], Dao et al[47], Ishibashi[48] and Conway and Lof[49]. A few firms which require mentioning are Yazaki corporation[56] of Japan, York[51] and Arkla Industries[52-53] of USA. Several workers including Ward[54], Krieth and Kreider[55], Bartlett[56], Johannsen[57], and Ayyash[58] have shown that the absorption cooling is technically feasible, but is not cost effective as also shown by Lof and Tybout[59], and Roberts and Sheridan[60]. Several review articles exclusively on solar absorption cooling have been recently written by Ellington et al[61], Allen et al[62], Auh[63], Paul[64], Alizadeh et al[65], Shawarts and Shitzer[66], Kaushik[67], and Krusi[68] giving the state of art, concepts, analysis and commercialization.

5.3.2 Principle of absorption cooling

The basic absorption cooling system is schematically shown in figure 5.3(a) showing the major components. The system cannot be illustrated on a simple T-S diagram because



(a) Schematic of absorption cooling system



(b) Simple absorption cycle

FIG.5.3 WORKING OF ABSORPTION CYCLE

two working fluids i.e. refrigerant and absorbent are involved. The absorbent-refrigerant combination is so selected that the absorbent has a high affinity for the refrigerant. It is assumed that the absorbent-refrigerent solution is liquid. The strong solution (rich in refrigerent) contained in the generator is heated by solar energy using collectors, such that the vapour pressure equals the saturation pressure in the condenser. The refrigerant in vapour form goes to the condenser, while the weak solution returns to the absorber through a recuperator and a throttling valve. In the condenser the refrigerant gets condensed rejecting heat, and comes in the liquid form at high pressure. The refrigerant now passes through the expansion valve and evaporates in the evaporator, thereby extracting heat from the surroundings. Now the low vapour pressure refrigerant from the evaporator goes to the absorber where it is reabsorbed with the liberation of heat. The low pressure, rich absorbent refrigerant solution is now pumped to the generator at high pressure to complete the cycle. A recuperator heat exchanger is used to transfer heat between the solutions passing between the absorber and the generator. The evaporator and absorber of the system are in the low pressure side and the generator and condenser are in the high pressure side of the system.

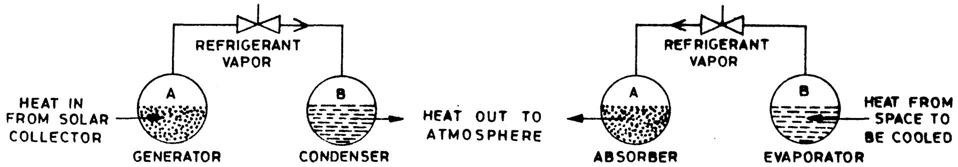
The ideal thermodynamic vapour absorption cycle can be represented on a 'pressure-temperature-concentration' diagram as shown in figure 5.3(b). On the abscissa scale one can read the values for evaporator temperature, condenser temperature, absorber temperature, generator temperature and the temperature ranges during absorption and generation. The pressure in the condenser, generator, evaporator, and absorber; and the concentration of the refrigerant can be read on the ordinate scale. On this diagram the number in circle corresponds to those in figure 5.3(a) showing the states of various points in the system.

In the absorption cycle (figure 5.3(b)) heat is usually rejected to cooling water or the atmosphere. So the temperature in the condenser (Tcond) and in the absorber (Tabs) is about the same. For a particular evaporator temperature (Tevap) and a given Tcond, there is a minimum value for the concentration of strong solution which will just enable boiling to commence in the generator. If the strong solution concentration is greater than this critical value, then boiling will occur while the temperature increases from T_2 to T_5, and the concentration is reduced to the critical value.

The greater the boiling range in the generator, the greater the concentration change. This means that less solution has to be circulated to produce the same refrigeration effect (i.e. the circulation ratio, the ratio of mass flow rate of strong solution to the mass flow rate of refrigerant is reduced). T_5 is dependent on collector outlet temperature.

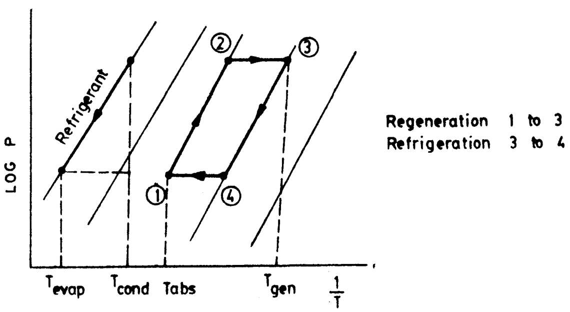
5.3.3 Basics of Absorption Cooling

Cooling using absorption cycle can be produced either continuously in a cycle or discontinuously. The cooling produced discontinuously is known as intermittent cooling, and the intermittent refrigerator is simple in operation and portable. Such intermittent cooling units are most suitable to operate with solar energy which itself is an intermittent heat source and they can be used in rural areas for producing ice etc. Studies on intermittent coolers have been conducted by Keith[69], Williams et al[70,71], Chinnappa[26, 72] and Trombe[24]. The advantages of an intermittent refrigeration system are: compactness, built-in refrigeration storage, and the possibility of using solid absorbents. The working of a intermittent absorption cooling system is clear from figure 5.4.



(I) Regenerator mode (II) Regeneration mode

(a) Intermittent-absorption cooling process



(b) Equilibrium diagram of intermittent absorption cooling

FIG.5.4 WORKING OF INTERMITTENT-ABSORPTION COOLING

In the intermittent system as shown in figure 5.4(a) there are two interconnected vessels A and B, acting first as generator and condenser, and subsequently as absorber and evaporator. The first mode is known as the regeneration mode and the second the refrigeration mode. VesselA contains suitable refrigerant - absorbent solution while vessel B contains only pure refrigerant. In the begining when the valve is closed and both the vessels are at the same temperature, the vapour pressure in tank B, will be higher then the tank A since pure refrigerant has higher vapour pressure than refrigerant - absorbent solution at the same temperature. In the regeneration mode, the valve is opened and solar heat is supplied in vessel A acting as generator through heat exchanger thereby raising the vapour pressure of refrigerant which goes in vesssel B and gets condensed in it which acts as condenser where low temperature is maintained generally by circulating cold water through the heat exchanger. The system can now be kept in this charged form after closing the valve till the time cooling is required. In the refrigeration mode as shown in figure 5.4 b(II), the valve is first opened, the refrigerant vapour from vessel B now acting as an evaporator rushes to vessel A working as an absorber, and is absorbed producing heat. The evaporation of refrigerant from evaporator (vessel B) causes cooling in this vessel. The vapour pressure temperature equilibrium diagram of this intermittent cycle is shown in figure 5.4 (b).

The COP and the reversibility of the absorption cooling cycle very much depends on the properties of the refrigerant - absorbent combination, and the concentration of refrigerant in it. Some of the desirable properties of a absorbent - refrigerant solution are as follows

1. In the absorbent - refrigerant solution the absorbent should be nontoxic, and non corrosive, and should have high boiling point, low specific heat, high chemical stability, and high solubility with the refrigerant. The absorbent should remain in the liquid form in the operating range of conditions and the heat liberated during absorption of refrigerant should be minimum. The deciding parameters of absorbents would be the boiling point and melting point of the absorbent.
2. The refrigerant in the absorbent - refrigerant solution should be nontoxic, noncorrosive and should have high critical temperature, large latent heat of vapourization, low specific heat and should be stable throughout the cycle.
3. The absorbent - refrigerant solution should have high solubility at the required temperature and pressure in the absorber but should have low solubility at the temperature and pressure in the

generator.

4. The freezing point of the absorbent-refrigerant solution should be lower than the lowest operating temperature in the cycle.
5. The absorbent-refrigerant solution should have low viscosity in the operating range of temperature, and be non flammable, non toxic, non corrosive, chemically stable, and low in cost. The amount of heat required to vapourize the refrigerant in the generator should be as close as possible to the heat of vapourization of the refrigerant in the evaporator.

Although more than 200 refrigerant-absorbent combinations have been listed as possible candidates for absorption cooling process only 3 refrigerant-absorbent combination systems as shown in Table 5.1 have tried out in practical systems.

The two commercially tried absorbent-refrigerant combinations are lithium bromide-water and ammonia-water combination. The most studied absorption cooling machine utilizes lithium bromide-water combination. The combination has several advantages such as : (i) the lithium bromide which acts as an absorbent is non volatile and therefore avoids the need for rectifying equipment; (ii) in this combination the water which acts as a refrigerant which has high latent heat of vapourization; (iii) the water - lithium bromide system is comparatively simpler, operates at low pressure requiring less pumping power, and operates at higher COP; and (iv) the water - lithium bromide solution is non-toxic, and non-flammable. The water lithium bromide system has some disadvantages also such as: (i) the system can be used for airconditioning applications only; (ii) the solution is corrosive and the system works under high vacuum conditions; and (iii)the system requires a water cooled condenser.

In the ammonia-water (aqua-ammonia) system, ammonia acts as the refrigerant and water as the absorbent. this combination is one of the oldest combinations used in industries for airconditioning and refrigeration. The ammonia-water absorption system has many advantages such as: (i) It is widely available and has large negative deviation from Raoult's law; (ii) it has a low molecular weight and therefore a high heat of vapourization of refrigerant; (iii) the water which acts as an absorbent has very high affinity for ammonia and is non-toxic and inexpensive.

The disadvantages of aqua - ammonia solution are : (i) since absorbent (water) is volatile, a part of it gets vapourized and goes with the refrigerant vapour, and therefore a rectifying system is required; (ii) in the ammonia-water system pumping of working fluid from the absorber pressure to the generator pressure is required, and

Table 5.1 Candidate for absorption cooling machines (From Kaushik[67])

Refrigerant	Absorbent	Reference
Water	LiBr	Duffie and Sheridan[27]
	LiI,LiSCN,CsF,RbF	Buffington[23]
	or their multiple	Ainbinder[73]
	salt solutions	Macriss and Rush[74]
		Mansoori and Patel[30]
Ammonia	H_2O	Eisentadt et al[75]
	NaSCN	Blytas and Daniels[76]
	$CaCl_2$	Sargent and Beckman[29]
	$LiNO_2$ or others	Nielsen et al[77]
		Chinnappa[78]
		Roberson et al[79]
Halogenated	ethers	Zellhoeffer[80]
organic	esters	Buffington[23]
compounds	amides	Eiseman[81]
(Freons R-21	amines & others	Aker et al[82]
or R-22)		ASHRAE Handbook[83]
		Albright[84-86]
		Hessilberth and Albright[87]

(iii) since ammonia is flammable and toxic, it cannot be employed in a direct expansion evaporator coil and requires a separate chilled water loop.

During the phase transitions (i.e. in the processes of evaporation and condensation) the temperature and the pressure remains constant while entropy and volume changes. According to Clapeyron's equation if a curve is plotted between Logarithm of vapour pressure and reciprocal of the absolute temperature then the lines of constant concentration will be practically straight lines. Such diagrams are available for many refrigerant - absorbent solutions and are of great value for the comprehension of an absorption machine. The equilibrium chart for lithium bromide-water solution is studied and discussed by many workers including Lower[88] and McNeely[89] and the chart is shown in figure 5.5. Here water vapour pressure is plotted versus solution temperature for different solution concentrations. Empirical equations are also derived by McNeely relating vapour pressure with the enthalpy for lithium bromide - water solution. This diagram applied to saturated conditions only where the solution is in equilibrium with the water vapour.

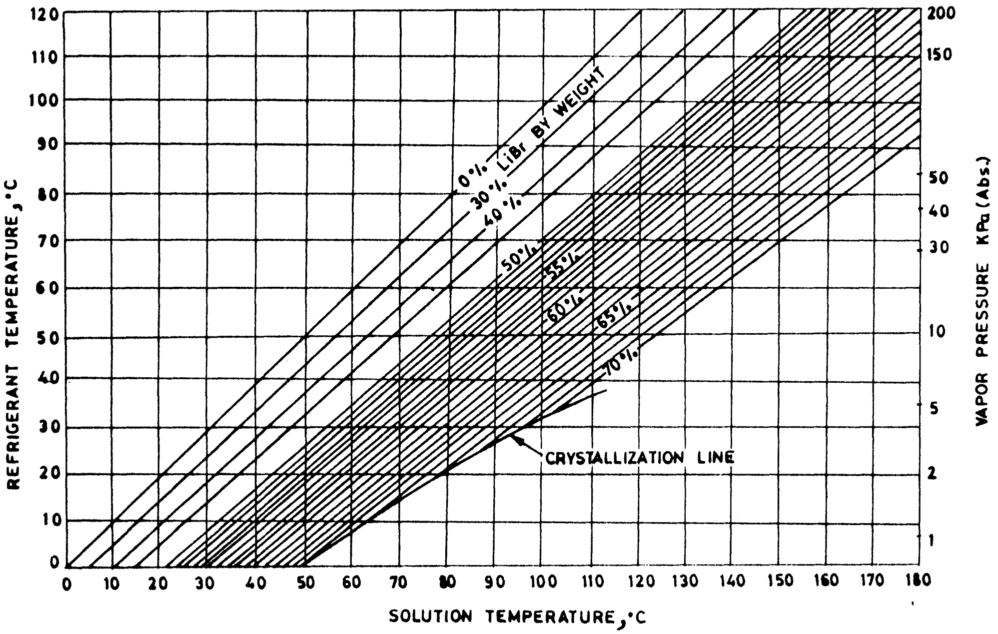


FIG.5.5 EQUILIBRIUM CHART FOR LiBr-H_2O SOLUTIONS

From the chart it is seen that if the water temperature is 70 ^{o}C, the vapour pressure of the liquid will be 32 KPa. The lithium bromide-water solution having a concentration of 50 percent at a temperature of 100 ^{o}C will also develop a vapour pressure of 32 KPa and the same vapour pressure can be obtained with a concentration of 60 percent at a temperature of 120 ^{o}C. The enthalpy of lithium bromide water solution at different concentrations and solution temperatures can be seen from figure 5.6. This diagram will be useful for designing the lithium bromide-water cooling system.

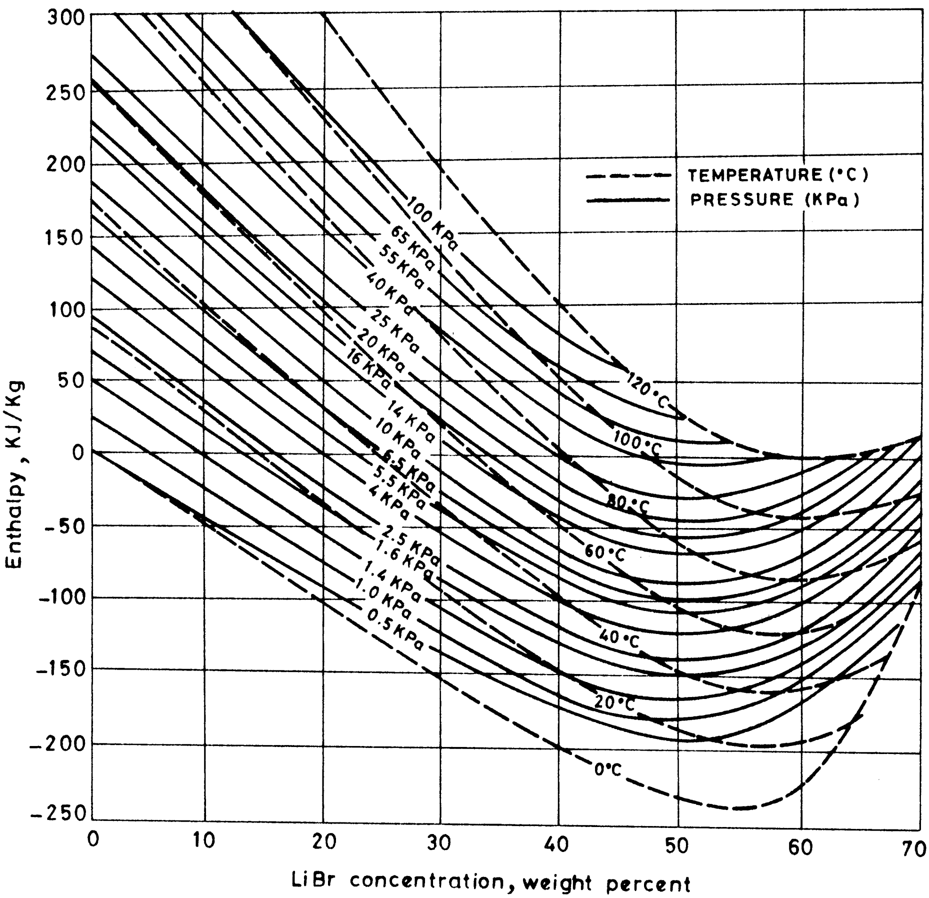


FIG.5.6 ENTHALPY-CONCENTRATION CHART FOR H_2O-LiBr SOLUTION

The equilibrium diagram for aqua-ammonia solution is shown in figure 5.7, which is drawn from the chart supplied by Institute of Gas Technology,USA. Studies are conducted by

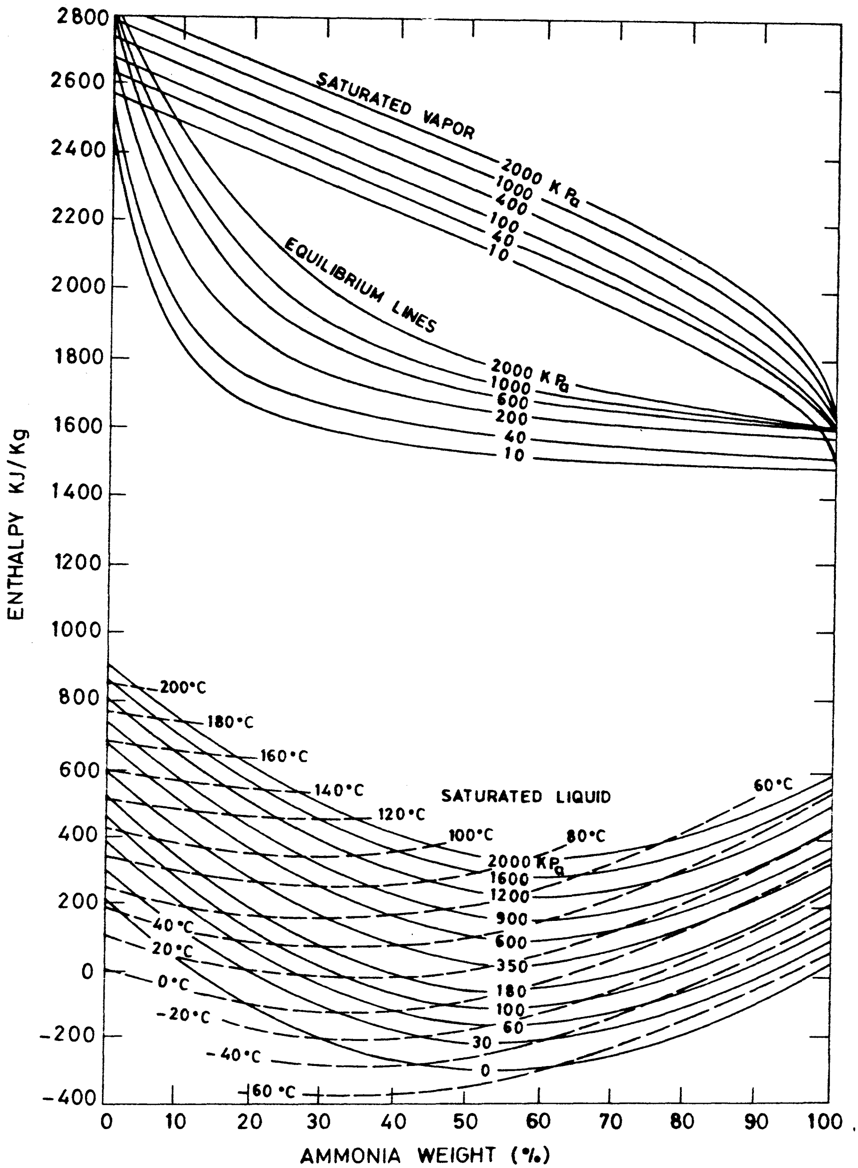


FIG.5.7 PRESSURE-ENTHALPY CHART FOR AQUA-AMMONIA SOLUTION

Scatchard et al[90], Kohloss and Scott[91] and Jain and Gable[92] on ammonia-water solution to get pressure-enthalpy-concentration data. As discussed earlier, in a water-ammonia solution, water vapour goes with the ammonia vapour and therefore a rectifier is required. The pressure enthalpy chart for pure ammonia vapour is shown in fig.5.8.

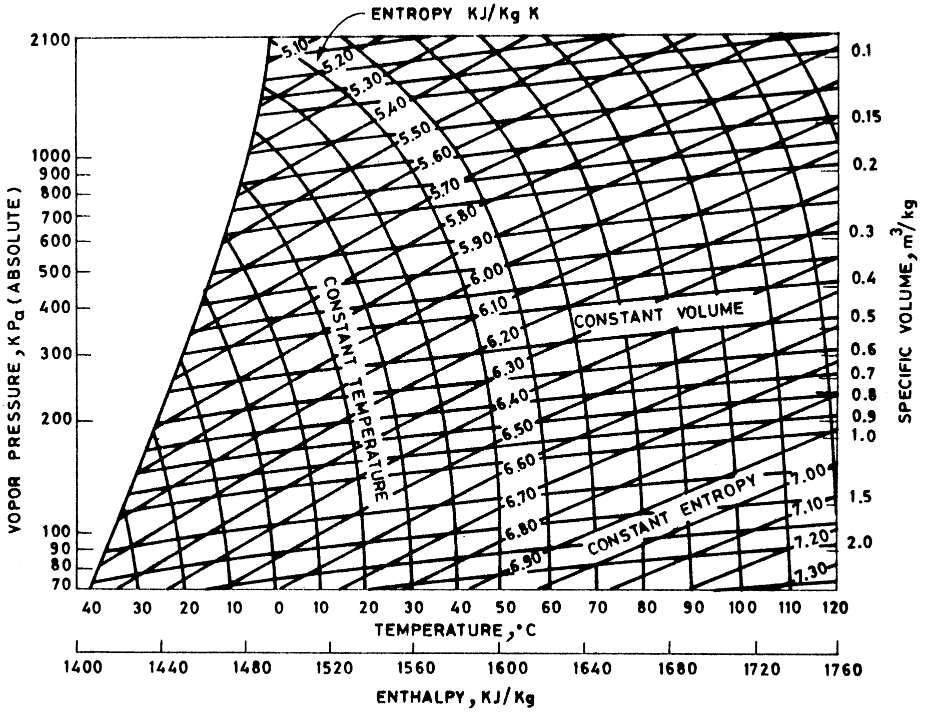


FIG.5.8 PRESSURE-ENTHALPY CHART FOR AMMONIA VAPOUR

Attempts are being made to identify new pairs of fluids to improve the performance of the absorption cooling system[93]. Alefeld[94] conducted absorption cooling experiments with three material combination. He used lithium bromide-water-ammonia solution, where lithium bromide reduces the equilibrium vapour pressure and the ammonia evaporates and acts as a refrigerant. In this system the lithium bromide holds the water in the solution and therefore eliminates the vapourization of the water with the ammonia. The use of a tertiary mixture avoids the use of high pressure in the system simplifying the overall design.

5.3.4 Water-Lithium Bromide Absorption System

The most popular absorption cooling system is the water-lithium bromide airconditioner which can provide evaporation temperature around 4 °C, and operates in the pressure range of 4 KPa to 0.8 KPa. As discussed earlier in the water-lithium bromide system, water acts as a refrigerant and lithium bromide in water acts as an absorbent. In this cycle water gets evaporated from the hot weak solution and finally gets absorbed in the cold strong solution. Cooling is produced during evaporation of water which takes place in the reabsorption process. The only problem in the lithium bromide - water system is the risk of crystallization of lithium bromide which may happen due to either low condenser temperature or high generation temperature.

The solar operated water-lithium bromide absorption system can be operated either in the closed cycle or in an open cycle. The open cycle absorption system is simple in operation and cheap collectors can be used but the auxiliary power required to operate the pumps is high and there is a risk of contaminants entering the system while in a closed cycle less auxiliary power is required and system can work continuously without any contamination.

As discussed earlier the pioneer work on water-lithium bromide cycle was conducted at the Wisconsin Solar Energy Laboratory, USA[8,17] and at the University of Queensland, Australia[44,95]. After this, considerable work was conducted on thermodynamics, plant design, and full scale performance testing of water-lithium bromide cooling systems. Systematic analysis of the thermodynamic cycle was conducted by Allen at al[62,69]. The effect of cooling water and chilled water temperatures on the performance of system was computed by Miller[97]. Computer studies on multistage lithium bromide system are conducted by Allen et al[62] and Wilbur and Mitchell[98]. Butz et al[99] have given a simple model of water lithium bromide cooler. Later this model was modified by Oonk et al[100] and Ward and Lof[37]. Several design improvements in the water-lithium bromide absorption coolers are suggested by Grossman et al[101]. A transient model of the absorption cooler is introduced by Blinn[102].

The water-lithium bromide coolers are now commercially available in United States[51-53] and Japan[50]. A solar operated LiBr-water system is shown in figure 5.9. Flat plate collectors are used to supply heat to the generator, where the water is boiled off at a temperature of 93 °C. The water vapour goes to the condenser where it is cooled to about 38 °C by the water supplied from the cooling tower. In the condenser the water vapour gets condensed and revapourized on passing through the expansion valve and produces cooling in the evaporator at about 5 °C. Room air or water gets cooled when circulated through coils in the evaporator. The water vapour from evaporator goes to the absorber where it is absorbed in the lithium bromide solution at about 38 °C. In the absorber the liberated heat is removed by the cooling water and the cycle is repeated. The generator temperature is maintained between 75 and 100 °C to avoid crystallization of lithium bromide.

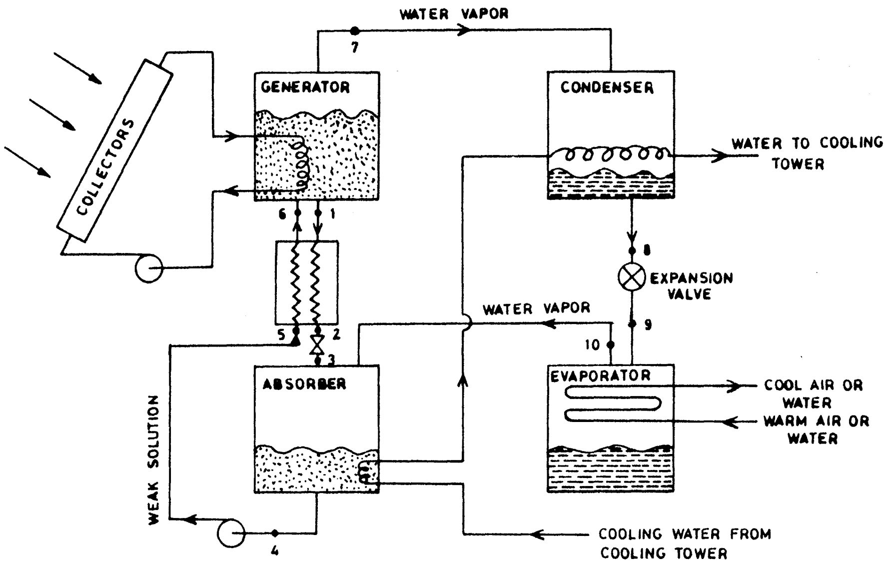


FIG.5.9 SCHEMATIC REPRESENTATION OF SOLAR OPERATED LiBr-H_2O ABSORPTION COOLING SYSTEM

This temperature is maintained by using an auxiliary heat source if required.

For a closed absorption system as shown in fig. 5.9, the energy balance equation[103] is

$$Q_o = Q_A + Q_C = Q_G + Q_E + Q_P \qquad (5.4)$$

Where Q_G = heat supplied by the generator at temperature T_T

Q_E = heat supplied to the system at the evaporator by the medium to be cooled at temperature T_E

Q_P = pump work

Q_A = heat rejected by absorber at temperature T_A

Q_C = heat rejected by condenser at temperature T_C

Q_o = total heat rejected to the environment at temperature T_o

The energy required to pump the liquid $Q_P = -\int v\,dp$ is very small and can be neglected. The entropy changes for the generator heating medium (ΔS_G), environment (ΔS_o), and refrigerated substance (ΔS_E), are respectively given as:

$$\begin{aligned} \Delta S_G &= -\,Q_G/T_G \\ \Delta S_o &= Q_o/T_o \\ \Delta S_E &= -\,Q_E/T_E \end{aligned} \tag{5.5}$$

The total entropy changes ΔS is greater than or equal to zero according to the second low of thermodynamics

$$\Delta S = \Delta S_E + \Delta S_G + \Delta S_O \geq 0 \tag{5.6}$$

or

$$\Delta S = Q_O/T_O - Q_G/T_G - Q_E/T_E \geq 0 \tag{5.7}$$

From equations (5.7) and (5.4), one gets

$$Q_G\left(\frac{T_G - T_O}{T_G}\right) \geq Q_E\left(\frac{T_O - T_E}{T_E}\right) \tag{5.8}$$

Now coefficient of performance COP, for cooling is given as:

$$COP = \frac{Q_E}{Q_G} \leq \frac{T_E\,(T_G - T_O)}{T_G\,(T_O - T_E)}$$

For a completely reversible system, it is given as:

$$(COP)_{max} = \left(\frac{T}{T_O - T_E}\right)\left(\frac{T - T}{T_G}\right) \tag{5.9}$$

which is a product of COP of Carnot refrigerating cycle and Carnot engine efficiency. The thermodynamic analysis of an absorption cooling system can be carried out by applying the following three equations to any part of the system :

(i) Mass balance : (5.10)

$$\varepsilon\dot{m} = 0$$

(ii) Material balance :

$$\varepsilon \dot{m}\, x = 0 \tag{5.11}$$

(iii) Energy balance :

$$\varepsilon Q + \varepsilon \dot{m}\, h = 0 \tag{5.12}$$

Some of the simplifying assumptions[14] made are as follows:

1. Absorbent does not evaporate from the generator,
2. The temperatures in generator T_G, evaporator T_E, absorber T_A, condenser T_C, and environment T_0 are constant,
3. Pressure drop along the lines due to friction is negligible,
4. Thermodynamic properties of the working fluid are known,
5. Heat exchange to the surroundings from components except those considered are negligible, and
6. At a given temperature the refrigerant and absorbent phases are in equilibrium.

Referring to Fig. 5.9, the mass, material and energy balance equations on the different components can be written as :

Generator :

$$\dot{m}_6 = \dot{m}_1 + \dot{m}_7 \quad \text{mass balance}$$

$$\dot{m}_6\, x_6 = \dot{m}_1\, x_1 \quad \text{material balance} \tag{5.13}$$

$$Q_G = \dot{m}_7\, h_7 + \dot{m}_1\, h_1 - \dot{m}_6\, h_6 \quad \text{energy balance}$$

Absorber :

$$\dot{m}_4 = \dot{m}_{10} + \dot{m}_3$$

$$\dot{m}_4 x_4 = \dot{m}_{10}\, x_{10} + \dot{m}_3\, x_3 \tag{5.14}$$

$$Q_A = \dot{m}_{10} h_{10} + \dot{m}_3 h_3 - \dot{m}_4\, x_4$$

Condenser :

$$\dot{m}_7 = \dot{m}_8$$

$$\dot{m}_7 x_7 = \dot{m}_8 x_8 \tag{5.15}$$

$$Q_C = \dot{m}_7 h_7 - \dot{m}_8 h_8$$

Evaporator :

$$\dot{m}_9 = \dot{m}_{10}$$

$$\dot{m}_9 x_9 = \dot{m}_{10} x_{10} \tag{5.16}$$

$$Q_E = \dot{m}_9 h_9 - \dot{m}_{10} h_{10}$$

Heat exchanger

$$\dot{m}_1 = \dot{m}_2$$

$$\dot{m}_5 = \dot{m}_6$$

$$\dot{m}_1 x_1 = \dot{m}_2 x_2 \tag{5.17}$$

$$\dot{m}_5 x_5 = \dot{m}_6 x_6$$

$$\dot{m}_5 h_5 + \dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_6 h_6$$

where points 1,2,........refers to the positions shown in fig. 5.9, h is specific enthalpy in KJ/Kg, x is concentration in Kg of LiBr per Kg of solution, and $\dot{m}$ is the mass flow rate in Kg of solution per unit of time. It is seen from the figure 5.9 that $h_7 = h_r$(enthalpy of refrigerant), $\dot{m}_7 = \dot{m}_r$ (flow rate of refrigerant), $x_6 = x_r$ (cconcentration of refrigerant in LiBr), $x_1 = x_{ab}$ (concentration of LiBr in absorbent). Using above equations, the mass flow rates, the amount of heat required by generator, rejected by condenser and absorber, and cooling produced in evaporator can be calculated. The COP of the cooling system will be

$$\mathrm{COP} = \frac{Q_E}{Q_G} = \frac{\dot{m}_9 h_9 - \dot{m}_{10} h_{10}}{\dot{m}_7 h_7 + \dot{m}_1 h_1 - \dot{m}_6 h_6} \tag{5.18}$$

For the temperatures at various points specified earlier and using the steam tables, pressure-enthalpy diagram shown in fig.5.6, and using above equations, Threlkeld[103] has calculated pressure, concentrations, enthalpy and flow rates at various points shown in fig.5.9 for producing 1 ton of refrigeration. This data is shown in table 5.2.

As discussed earlier while dealing with the basic absorption cycle that the COP of a single effect absorption system as shown in figure 5.9 is theoretically always less than 1 and in most practical cases ranges from 0.6 to 0.8. Moreover, as shown by Wilbur and Mitchell[98] for a given evaporator and cooling water temperature, there is a fixed minimum generator temperature to operate the system as shown in figure 5.10.It is experienced that most of the $LiBr-H_2O$ machines are operating nearly at constant COP, since the effect of generator temperature on COP above a minimum value will not effect COP. Some improvements are possible by using higher generator temperatures and lower condenser and

Table 5.2 Thermodynamic properties at various points in a H_2O-LiBr Cooler

Point	Temp. (°C)	Pressure KPa	Concentration (percent)	Enthalpy KJ/Kg	Flow rate Kg/min ton
1	93	6.54	0.65	-63	1.08
2	-	6.54	0.65	-	1.08
3	-	0.84	0.65	-	1.08
4	38	0.84	0.60	-	1.17
5	38	6.54	0.60	-	1.17
6	82	6.54	0.60	-81	1.17
7	93	6.54	0.00	2677	0.09
8	38	6.54	0.00	158	0.09
9	5	0.84	0.00	158	0.09
10	5	0.84	0.00	2509	0.09

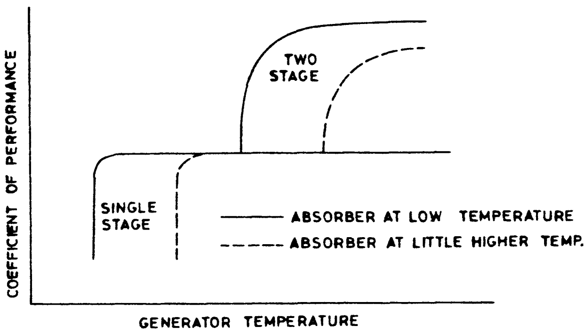


FIG.5.10 PERFORMANCE OF SINGLE EFFECT AND DOUBLE EFFECT ABSORPTION MACHINE.

absorber temperatures. Exploiting the adaptibility of the absorption cooling machine, attempts are made by Richter [104], Grossman et al[101], Grossman & Johansen[2], Auh[105] and Wilbur and Mitchell[98] to improve the performance of absorption cooling machine with special reference to LiBr-H_2O machine. Some of the suggestions made are as follows :

1. Heat exchangers have been modified to minimise temperature drop, and submergence in the generator is reduced to minimise in solution boiling point due to static heads of solution.
2. Both the generator and condenser can be combined into a single unit because they operate at high but at a equal pressure. Similarly absorber and evaporator can also be combined because both of them operate at low, and equal pressure. This is possible only in a small residential cooling unit such as is done in Arkla 3-ton LiBr-H_2O Chiller[106].
3. A thermosyphon pump can be employed to transfer solution due to temperature gradient and which works automatically without electric power. This arrangement is employed and is suitable only in small residential, cooling units such as in Arkla 3-ton LiBr-H_2O Chiller. In larger units by using electric operated mechanical pump, the performance can be improved and the same offers greater flexibility in operation and provide necessary pressure required by spray nozzles, purging system and at other parts of the system.
4. Since temperature of absorber has a greater effect on the COP of the unit, the cooling water entry is made first through the absorber and then to the condenser. This combination which is widely used in absorption chiller is called series combination of absorber and condenser. If the size of the cooling tower is increased, then both the absorber and the condenser can be connected in parallel receiving cold water at the lowest possible temperature. This system shows higher COP. In small residential cooling units the absorption device can be combined with the cooling tower in a single unit as is done in Arkla 3-ton LiBr-H_2O Chiller reducing the cost of cooling water, parasitic power, space and maintenance.
5. Since the solar energy flat plate collectors can provide heat in the temperature range of 70-95 °C, the generator of LiBr-H_2O chiller is designed to operate in this temperature range only. In a solar operated generator hot liquid flows in horizontal tubes in generator kept above the solution and on which solution from absorber is sprayed.

This arrangement improves the performance. In some small compact solar operated units, the solar collector is combined with the generator. In this case the performance gets improved but involve the risk of solution leaking and use of large amount of solution[107,108].

6. The performance of the absorber can be improved by properly distributing, sizing, and spacing the tubes in the absorber on which the solution is sprayed by spray nozzles and where refrigerant vapour gets absorbed. A proper purging arrangement in the absorber will improve the performance. By adding some suitable additives called surfactants in the absorbent solution will lower the crystallization temperature and improve heat and mass transfer and therefore the performance.
7. By employing a two-stage (double effect) system, the COP of the absorption machine can be improved [98]. The double effect absorption system employes single evaporator, absorber and condenser as in the simple single stage absorption system but uses two generators one operating at high temperatures and second operating at comparatively low temperature and two recouperator one working at high

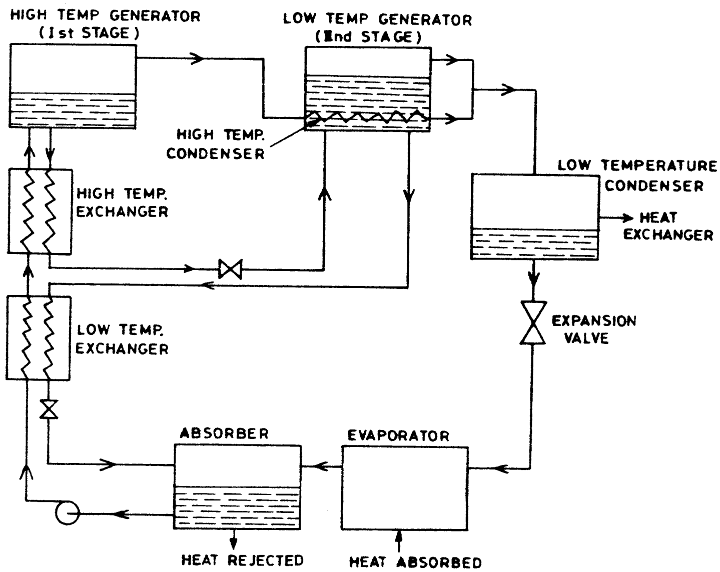


FIG.5.11 SCHEMATIC OF TWO-STAGE ABSORPTION COOLING SYSTEM

temperature and another working at low temperature. The double effect system is schematically shown in fig.5.11. As is seen from the figure, double effect system is based on the principle that the refrigerant vapour from the high temperature generator is used to drive further refrigerant vapour from low temperature generator while refrigerant from the first generator condenses. The rest of the system functions as a single-stage system. Theoretically the COP of this double-stage unit will be 2 but in practice COP in the range 1.3 to 1.5 is possible. COP of 2 can be obtained if all the refrigerant vapour generated from high temperature generator condenses in high temperature condenser. This double stage unit also requires less amount of cooling water. The only disadvantage of the double effect system is the requirement of high temperature i.e. of the order of 120 ^{o}C which is beyond the reach of simple flat plate collectors and therefore the use of evacuated tube collector or linear-parabolic collector is recommended. The double effect LiBr-H_2O airconditioning system is studied by Vliet and Saiidi[109].

8. If water is not available for cooling purposes then air cooling can be employed in a dual-series-connected absorption system[98] as is schematically shown in figure 5.12. The two single stage

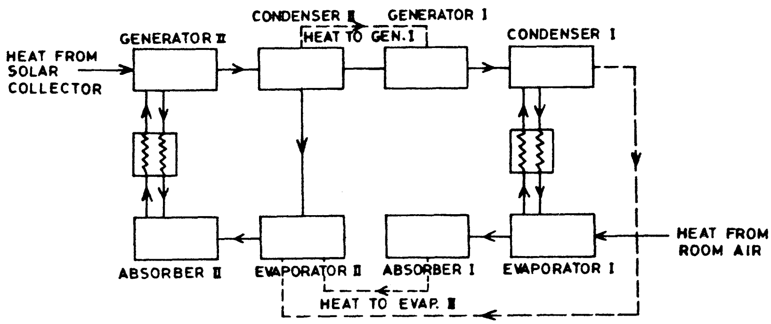


FIG.5.12 SCHEMATIC OF SERIES(DUAL) CONNECTED ABSORPTION COOLING SYSTEM

absorption subsystems are connected in such a way that the heat from the absorber and condenser of sub-system 1 is rejected directly into evaporator 2. The generator of the low temperature sub-system is heated by the heat rejected by the condenser of the high temperature sub-system. In this dual series system evaporator 1 can be maintained at a temperature of about 4 ^{o}C (depending on cooling requirement) while the temperature of absorber 2 can work at more than 50 ^{o}C and therefore the aircooling will be effective. The overall COP of dual series connected system is low ≃0.36 while each of the sub-system may have COP of 0.8.

9. As suggested by Grassie and Sheridan[110], the COP of the LiBr-H_2O system can be further improved by the use of refrigerant storage. The principle is schematically shown in fig.5.13. The idea is to

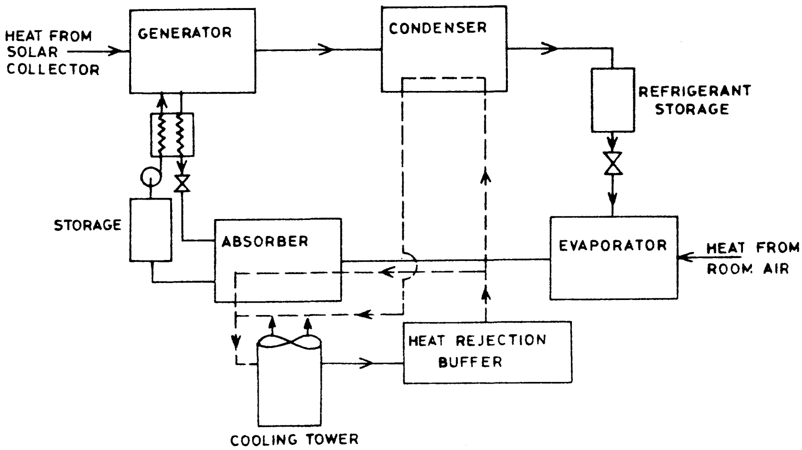


FIG.5.13 SCHEMATIC OF REFRIGERANT STORAGE SYSTEM

provide in association with the condenser, a storage volume where refrigerant can be accumulated during the peak hours of sunshine. At times when cooling is required, the stored refrigerant is released by expansion into the evaporator. Here hot water from solar collectors drives the refrigerant vapour from the generator which goes on condensing in the condenser and stored till the cooling is required. The use of heat rejection buffer tank as shown in fig.5.13, will help in reducing the size of the cooling tower and operating the system at lower generator temperature.

5.3.5 The Aqua-ammonia absorption system

The water ammonia absorption system known as aqua-ammonia absorption system is one of the oldest methods of refrigeration and airconditioning due to its wide range of operating conditions (upto 0 °C) and the elimination of a water cooling tower thereby making the system more useful even for residential requirements. The pioneer work on solar powered continuous ammonia-water absorption cooling was done by Farber et al[111,112] at the University of Florida and some computational analysis on the system was done by Chinnappa[113]. The main limitation in commercializing the solar operated aqua-ammonia airconditioning unit is due to the requirement of generator temperatures in excess of 150 °C which is beyond the supply limit of flat plate collectors therefore the generator requires to be completely redesigned so that the same can be made operational by the hot water at temperatures around 100 °C. Simmons et al[46] at the University of California, Berkeley, USA are modifying and designing a new generator to be heated by hot water.

The aqua-ammonia combination can be employed in open cycles[114], continuous closed cycle[111], and intermittant closed cycle[70-72] both for refrigeration and airconditioning applications. The aqua-ammonia system can also be employed in the heat pump mode also since there is no problem of crystallization with it. Dao et al[115] have recently reported the results of longterm research and development on aqua-ammonia absorption chillers.

The aqua-ammonia cooling system as shown schematically in fig.5.14 consists of all the previously described components-generator, absorber, evaporator, condenser and solution heat exchanger-plus a rectifier and analyser. The purpose of a rectifier and analyser is to stop the water vapour which gets evaporated alongwith the ammonia otherwise if it reaches to evaporator, it will elevates its temperature. The water vapour is first stopped in the rectifier by passing the mixture of vapour of ammonia and water countercurrently to the incoming solution in the rectifier. The remaining water vapour is checked in the analyser which is water cooled heat exchanger by condensing the water rich vapour and draining it to rectifier.

The solar heat is provided through a heat exchanger to the generator containing ammonia water mixture with a strong concentration of ammonia which acts as refrigerant. To stop the water vapour going to the condenser sometimes a single or double rectifier is used. The refrigerant directly from the rectifier and analyser goes to the condenser from where it is either stored or goes to the evaporator through the expansion valve where the liquid is evaporated at a constant temperature by heat transfer by the air stream from a room or building which is to be cooled. The ammonia vapour from evaporator now passes to the absorber where it recombines with the weak solution coming from the generator. The heat released is dissipated to the outside through cooling water and the temperature of the absorber is maintained. This high concentration solution can now be stored and passed through a sensible heat exchanger provided between the generator and absorber.

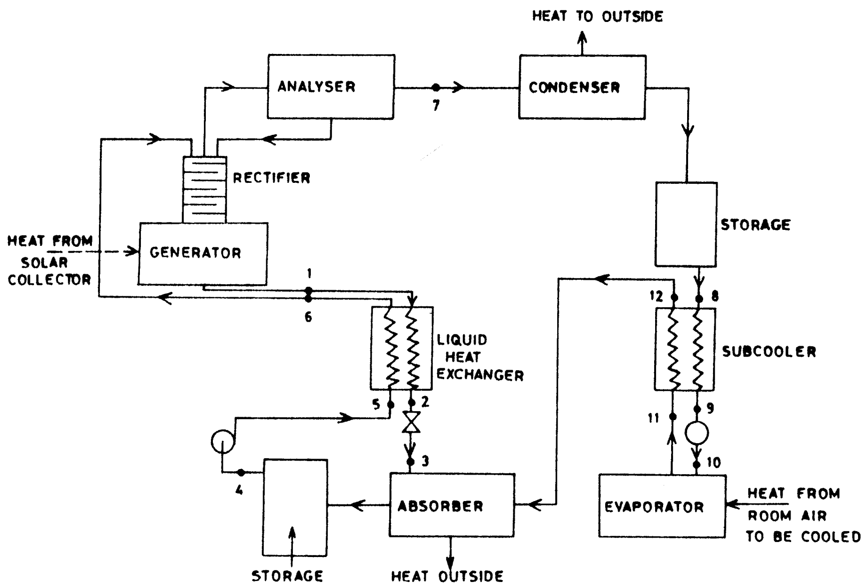


FIG.5.14 SCHEMATIC REPRESENTATION OF AQUA-AMMONIA ABSORPTION COOLING SYSTEM

Pressure, temperature, concentration, enthalpy chart for aqua-ammonia solution is shown in fig.5.7. Pressure, temperature, volume, enthalpy, entropy chart for pure ammonia vapour is shown in fig.5.8. Pressure-temperature-concentration chart for ammonia is replotted in fig.5.15 to show the effect of the air cooling and water cooling. If the condenser and absorber are air-cooled, then the generator may operate in the temperature range of 120-180°C. If condenser and absorber are water-cooled, the generator may operate in the temperature range of 95-120 °C. If air-cooling of absorber and condenser is done then generator temperature required is too high to be heated by flat-plate collectors. In such cases, an improved cycle[111,115] using a high concentration of NH_3 and an improved generator designed to operate at lower temperature may be used. Higher operating generator temperatures may be obtained by using advanced flat-plate collectors, evacuated tube collectors, CPC collectors or other concentrating collectors.

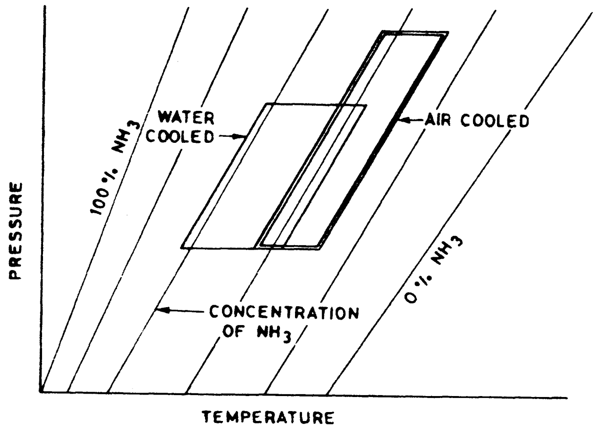


FIG.5.15 SCHEMATIC REPRESENTATION OF WATER AND AIR COOLING CYCLES ON PRESSURE-TEMPERATURE-CONCENTRATION DIAGRAM FOR NH_3-H_2O SYSTEM.

As discussed earlier in the case of LiBr-H_2O absorption system, the thermodynamic analysis of NH_3-H_2O cycle can also be performed using the mass balance, material balance, and energy balance equations for each component of the system.

Various state points shown in fig.5.14 are schematically shown in the enthalpy-concentration in fig.5.16. Chinnappa[26,73], Whitlow[116], Lauk et al[117]. Swartman and Swaminathan[28], Dhar et al[118]. Allen et al[62], Johnston[119] have conducted extensive theoretical design studies on aqua-ammonia cooling system. Stoecker and Reed [120], studied the effect of operating temperature on the COP of Carnot cycle, simple aqua-ammonia cycle and the modified aqua-ammonia cycle. They showed that by refining the heat exchangers and rectifier shift the COP of the cycle can be increased considerably. The effect of water vapour entering the evaporator is studied by Kouremenos[121]. Brouse et al[122] have given a simulation model for performance prediction of solar operated water-ammonia absorption heat pump for air-conditioning. Neal[123], Wilbur and Mitchell[98], and Alizadeh et al[65] have compared the H_2O-LiBr and H_2O-NH_3 cooling systems.

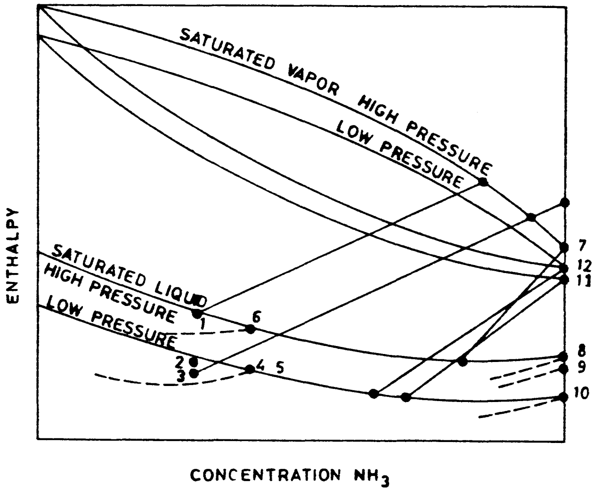


FIG.5.16 REPRESENTATION OF POINTS OF FIG.5.14 ON ENTHALPY-CONCENTRATION DIAGRAM.

5.3.6 Intermittent absorption refrigeration system

Intermittent type of solar absorption refrigeration is studied by many investigators because of its suitability in areas where there is no electricity, and because of the intermittent nature of sunlight. The solar intermittent refrigerator may be used for making ice, or as cold storage for food or vaccine in remote areas or small islands. The overall performance of solar intermittent refrigerators so far produced is low, operating at low efficiency i.e. about 8 to 10 percent. The principle of intermittent cooling is already described in section 5.3.3. The thermodynamic processes of intermittent operation are not reversible and its operation depends on the absorbent-refrigerant combination and concentration of refrigerant.

Williams et al[70] and Chinnappa[72] discussed the reasons for getting the low performance in case of intermittent operation compared to continuous operation. Williams et al[70] conducted experiments on ammonia-water and Freon

21-ethylene glycol combinations using a paraboloidal concentrator of 0.77 m^2 and obtained COP of 0.15 and 0.06 respectively. Trombe and Foex[12] used 1.5 m^2 cylindro-parabolic reflector and produced 4 Kg/m^2 day ice using water - ammonia combination. Chung and Duffie[124] worked on advanced NH_3 - H_2O system using parabolic cylinder reflector and produced 9.5 Kg of ice per m^2 of collector per day. The pioneer work on intermittent absorption refrigeration both experimental and theoretical was done by Chinnappa [26,72] who used flat plate collector and NH_3 - H_2O as the refrigerant-absorbent pair and obtained COP of 0.06 with generator temperature of 99 °C and evaporator temperature below 0 °C. Sargent and Beckman[29] suggested the use of NH_3-NaSCN combination instead of NH_3 H_2O due to its better thermodynamic properties and not requiring a rectifier. Swartman and Swaminathan[125], Swartman and Alward[126] and Swartman et al[127] conducted experiments on NH_3 - H_2O and NH_3-NaSCN combination using flat plate collector and showed that NH_3-NaSCN refrigerant-absorbent has better performance. Perry[128] proposed the use of H_2O-LiBr combination for the intermittent absorption refrigeration cycle. Exell et al [129-131] conducted detailed experimental and theoretical studies on a NH_3 - H_2O intermittent absorption refrigerator and used 5 m^2 flat plate collector with plane mirror booster at Thailand and produced 3.2 Kg ice/m^2 day. Aggarwal et al [132] conducted experimental studies on a solar powered R22-DMF (Freon 22 - Dimethyl Formamide) intermittent refrigeration system using a flat plate collector.

Some of the most promising refrigerant-absorbent pairs are given in table 5.3. The COP of these five combinations with solar collector are computed by Stubkier and Nielsen

Table 5.3 Refrigerant-absorbent pairs for intermittent cycle

Refrigerant	Absorbents
Ammonia (NH_3)	Water (H_2O)
	Sodium Thiocyanide (NaSCN)
	Lithium nitrate ($LiNO_3$)
	Calcium chloride ($CaCl_2$)
	Stroncium chloride ($SrCl_2$)
	Dimethyl Formamide (DMF)

[133]. Agarwal et al[132] studied experimentally and compared the four combinations NH_3-H_2O, NH_3-CaCl_2, $NH_3-NaSCN$, and NH_3-LiNO_3 for various combinations of condenser and evaporator temperatures. Both these studies have indicated that the best overall performance is obtained with solid absorbents, $CaCl_2$ and $SrCl_2$. Out of the three liquid absorbent, it is observed[133] that NH_3-LiNO_3 shows significantly better results compared to the other two systems. The poor performance of $NH_3 - H_2O$ system may be due to the requirement of a rectifier.

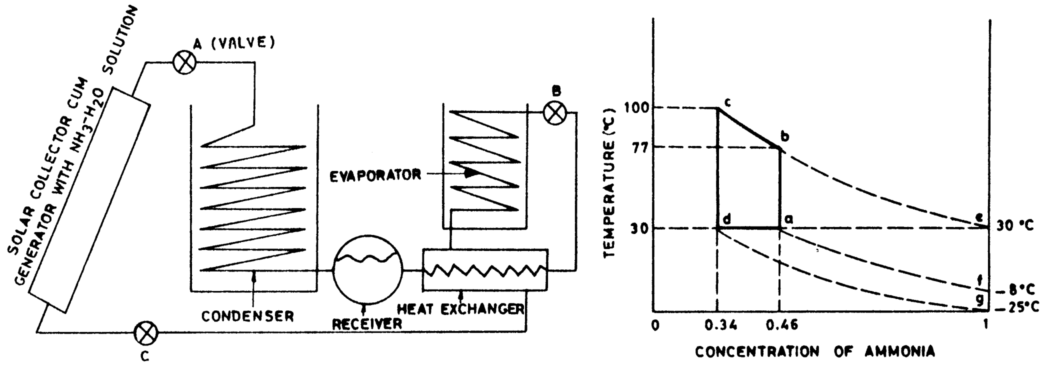


FIG.5.17 SCHEMATIC DIAGRAM OF INTERMITTENT SOLAR REFRIGERATOR AND THERMODYNAMIC CYCLE

The solar intermittent refrigerator as described by Exell and Kornsakoo[131] is described here. The flow diagram and the related thermodynamic cycle is shown in fig. 5.17. Flat plate collectors of 5.0 m^2 area with plane mirror boosters containing about 67 Kg of NH_3-H_2O solution of 0.46 ammonia concentration are used as generators from where ammonia gets vapourized on solar heating during the day with control valve A open and valve B and C closed. The ammonia is condensed and stored in a water cooled receiver. At night the collector is allowed to cool by opening the glass panes and valve A is closed and Valve B and C are opened to produce refrigeration in the evaporator. The ammonia vapour goes through the bottom of the collector and gets reabsorbed into the solution and heat of absorption escapes through the collector. In the thermodynamic cycle a → b represents the heating of the solution in the morning, b → c the generation of ammonia at midday. Point e shows the state of ammonia in the receiver. The curve c → d shows the cooling of solution in the evening when valve A is opened. The curve g→ f shows vapourization of ammonia in the evaporator and d → a reabsorption of ammonia in the solulion releasing the heat to the outside. On average bright days about 14 Kg of ammonia was distilled and about 25 Kg of ice from water at 28 °C was produced on the following night.

5.3.7 Arkla 3-ton absorption chiller

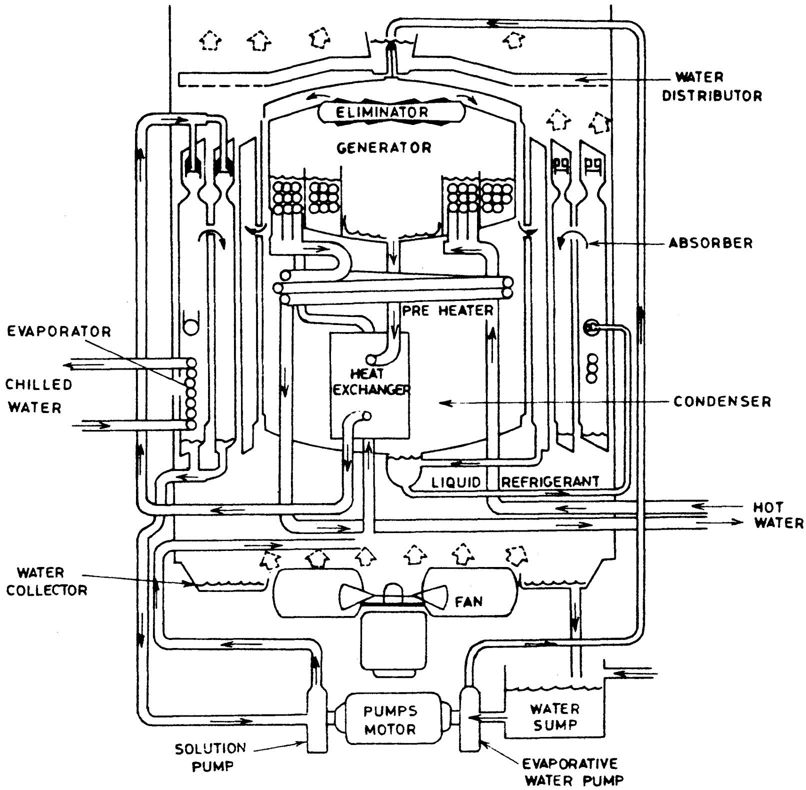


FIG.5.18 SCHEMATIC OF ARKLA 3-TON $LiBr-H_2O$ CHILLER WITH EVAPORATIVE COOLING (From Merrick[134])

Several companies like Arkla Industries (formerly Servel Corporation) USA; York Division of Borg-Warner corporation, USA; Yazaki Corporation, Japan have recently started marketing solar operated LiBr-H_2O absorption chillers in various cooling capacities ranging from 1 ton to 100 ton.The firms had experience of manufacturing gas operated absorption chillers and modified the absorption units for operation with solar heated water. The firm Arkla Industries earlier used to sell gas fired domestic refrigerators and now produces a 3-ton absorption chiller for residential applications[52, 134] and 25 ton chiller for commercial use. The Arkla 3-ton absorption chiller is schematically shown in fig. 5.18 which is a compact unit, and where heat is rejected by evaporative cooling avoiding a separate cooling tower reducing capital cost, installation cost and parasitic power. Table 5.4 gives the details of Arkla-3 ton unit[52,134]

Table 5.4 Design data for Arkla 3-ton absorption chiller.

1.	Design Delivered Capacity	10.5KW(3.0 tons)
2.	Energy Requirement(Design)	
	Hot water input	14.6 KW
	Hot water inlet temperature	90.4 °C
	Hot water outlet temperature	85.5 °C
	Hot water flow rate	40 l/m
	Pressure drop at 40 l/m	29 KPa
	Maximum working pressure	690 KPa
	Unit water volume	11.3 litres
3.	Chilled water Data (Design)	
	Inlet temperature	12.8 °C
	Outlet temperature	7.2 °C
	Flow rate	27.2 l/m
	Pressure drop at 27.2 l/m	13.7 KPa
	Maximum working pressure	690 KPa
	Unit water volume	5.7 litres
4.	Condensing water data (design)	
	Heat rejection rate	25 KW
	Inlet temperature	29.7 °C
	Outlet temperature	37.5 °C
	Flow rate	45.4 l/m
	Pressure drop at 45.4 l/m	28.6 KPa
	Maximum working pressure	690 KPa
	Unit water volume	11.3 litres
5.	Cooling tower Data(design)	
	Maximum heat rejection	31 KW
	Minimum sump temperature	24 °C

From table 5.4 it is seen that the COP which is the ratio of design delivered capacity (10.5 KW) to the design input hot water requirements (14.6 KW) comes out of to be 0.72. The performance data (delivered capacity) on Arkla 3-ton absorption chiller for different hot water inlet and outlet temperatures, condensing water temperature of 29.4 °C is shown in table 5.5. From this table it is seen that the capacity of unit changes from 1.09 tons (3.83 KW) to 3.50 (12.31 KW) by varying different parameters. It is also seen that the chiller shows 3-ton capacity when the hot water inlet temperature is 90.6 °C, outlet temperature is 85.5 °C, inlet condensing temperature is 29.4 °C and the heat rejection rate is 25.20 KW.

Table 5.5 Performance data of Arkla 3-ton LiBr -H_2O absorption.

Hot water temperature (°C) Inlet temp.	Outlet temp.	Energy input (KW)	Heat rejected (KW)	Cooling capacity tons	(KW)
79.5	77.3	6.30	10.10	1.09	(3.84)
85.0	81.5	10.52	18.02	2.13	(7.50)
90.6	85.5	14.65	25.20	3.00	(10.55)
96.2	90.0	17.81	30.12	3.50	(12.31)

Inlet condensing water temperature	29.4 °C
Condensing water flow rate	45.4 l/m
Hot water flow rate	41.6 l/m
Chilled water flow rate	27.2 l/m
Chilled water leaving temperature	7.2 °C

5.3.8 Yazaki absorption chiller

Yazaki Corporation of Tokyo[50], Japan is manufacturing LiBr-H_2O absorption chillers of various capacities ranging from 4.6 KW (1.3 ton) to 174.5 KW (49.6 ton). The specifications of these absorption chillers needing hot water from 75 C to 100 °C which can be supplied by solar flat plate collectors suitable for residential airconditioning or industrial airconditioning are given in table 5.6. In these systems the chiller is water cooled and heat is rejected through a seperate cooling tower. The chilled and hot water circulating pump is built-in to each chiller for simple

Table 5.6 Design specifications of water heated absorption chiller suitable for solar operation produced by Yazaki Corporation, Japan

Chiller Model	WFC-400	WFC-600	WFC-2300	WFC-3000	WFC-4600	WFC-6000	WFC-9000	WFC-120000	WFC-150000
Refrigeration Capacity (KW)	4.6	7.0	26.2	34.9	52.4	69.8	104.7	139.6	174.5
Chilled water outlet temp.(℃)	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
Chilled water inlet tempt (℃)	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0
Generator heat input (KW)	7.7	11.6	37.4	49.8	49.8	74.8	99.6	149.4	249.9
Generator inlet temp.range(℃)	75-100	75-100	75-100	75-100	75-100	75-100	75-100	75-100	75-100
Generator hot water circulation(1/h)	1116	1656	6408	8568	12816	17136	25704	34272	42840
Generator pressure loss (KPa)	9.8	12.8	24.5	39.2	24.5	39.2	39.2	39.2	39.2
Cooling water inlet temp.(°C)	29.5	29.5	29.5	29.5	29.5	29.5	29.5	29.5	29.5
Cooling water outlet temp.(°C)	34.5	34.5	34.5	34.5	34.5	34.5	34.5	34.5	34.5
Cooling water circulation (1/h)	2124	3204	10800	14400	21600	28800	43200	57600	72000
Pressure loss in cooling water circuit (KPa)	14.7	22.6	68.6	68.6	68.6	68.6	68.6	68.6	68.6
Maximum pressure in 3 circuits (KPa)	588	588	588	588	588	588	588	588	588

installation. The chiller reaches to its full capacity and optimum efficiency after 30 minutes.

5.3.9 Hitachi Solar Powered absorption airconditioner.

Hitachi Ltd. of Japan has recently introduced[135] a solar energy operated 25 ton, LiBr-H_2O absorption airconditioner suitable for large buildings. The prototype model is schematically shown in fig.5.19. The main features of the prototype are as follows:

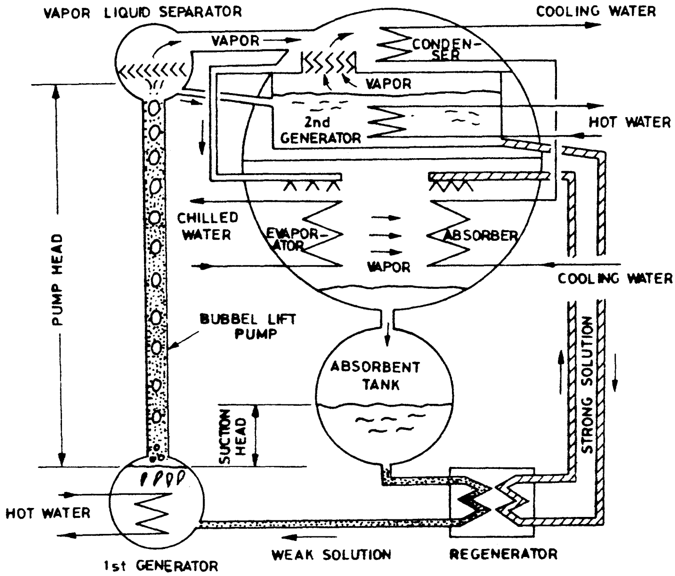


FIG.5.19 SCHEMATIC DIAGRAM OF ABSOPTION AIR-CONDITIONER DEVELOPED BY HITACHI LTD JAPAN(From tanka and Usui[135])

(i) The solution is circulated using a bubble lift pump instead of electrically operated mechanical pump saving space, cost, and parasitic power.
(ii) Specially designed heat exchangers in generator, evaporator, condenser, and absorber are used which give three times the heat transfer coefficient compared to the bare tubes used elsewhere.

In the first generator, the refrigerant-absorbent solution is heated using solar flat plate collectors at about 90°C, where part of the refrigerant is vapourised.

The bubble action of vapour lifts the refrigerant-absorbent solution to a vapour liquid separation. From here the vapour is removed and solution flows to the second generator. The hot water temperature should be maintained around 90 °C otherwise flow rate will reduce resulting in rapid decrease in cooling capacity. The effect of hot water

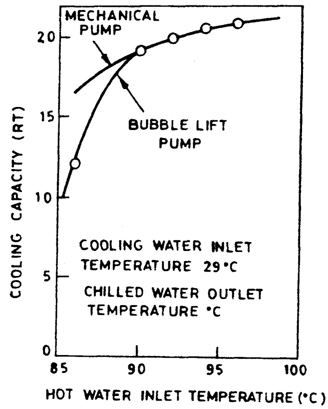


FIG.5.20 EFFECT OF HOT WATER TEMPERATURE ON THE COOLING CAPACITY (From Tanka and Usui[135])

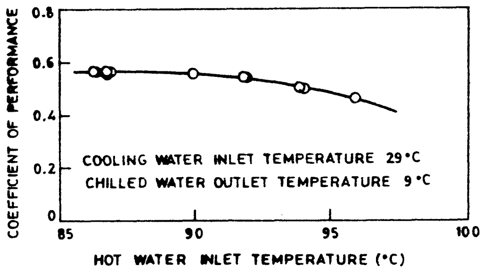


FIG.5.21 EFFECT OF HOT WATER TEMPERATURE ON THE COP OF AIR-CONDITIONING (From Tanka and Usui[135])

temperature on coefficient of performance (COP) of the absorption airconditioner is shown in fig.5.21. The unit had shown a COP of 0.6 with hot water temperature of 90 °C, cooling water inlet temperature of 29°C and chilled water outlet temperature of 9 °C.

5.4 DESICCANT COOLING

Removal of moisture (dehumidification) from room air using either absorbent or absorbent followed by evaporative cooling of air appears to be the most promissing airconditioning method for hot and humid climate[136]. Liquid and solid materials which have property of attracting and holding water vapour known as desiccants can be used to dehumidify the air and can thus be used for cooling the space. Desiccants can be absorbents or adsorbents depending whether during the sorption process there is a physical or chemical change or both such as in some salts or their aqua-solutions, and some inorganic and organic solutions (absorbents) or there is no change during the sorption process such as in some solids. Some of the desired properties of desiccants as listed by Grossman and Johansen[2], and Robinson[137] are: they should be thermally and chemically stable, non-toxic, non-flammable, odourless, non-corrosive, low cost, and readily available. Liquid desiccants should not crystalize and vapourise in the operating range of temperature and concentration, and should be non-viscous and have good heat transfer characteristics. The desiccant should also have desired water vapour pressure characteristics for the required degree of dehumidification and should be operative at temperatures available with solar flat plate collectors.

The pioneer Work on desiccant cooling was done by Pennington[138], Dunkle[139], Baum et al[140], Lunde[141-142] Lof[143], Mullick and Gupta[144], Rush et al[145-148] Hollands[149], Close et al[150], and Kakabaev et al[151-152. Both liquid or solid desiccant either in the open or closed cycles are used for cooling the space. The open cycle cooling system has several advantages such as [153]:

1. The solar collector used in the open cycle can be simple tilted flat plate collector with no glazing, no selective surface, no special fluid channels etc., while in a closed cycle the solar collector should be designed in such a way to give minimum of losses.
2. In a open system the temperature of the solution is to be raised such that its vapour pressure exceeds that of atmosphere. While in a closed cycle the temperature in the generator should be raised to a level that the vapour pressure of the

week solution becomes more than the water saturation pressure in the condenser. Generally the maximum pressure encountered in an open cycle is about one third the maximum pressure in the closed one.

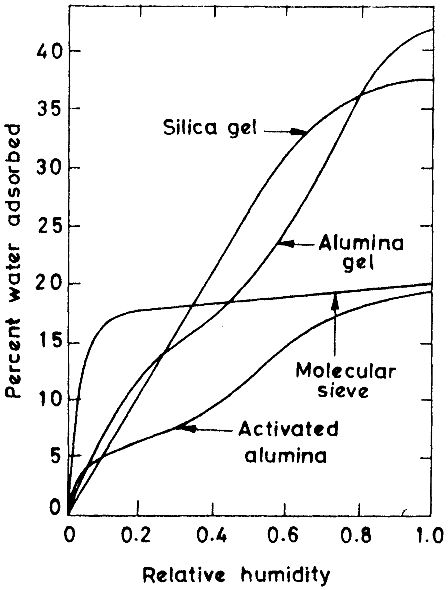
3. In the open system the equipment cost is lower since the the generator and solar collector is a single unit.

There are few problems in the open cycle system such as:

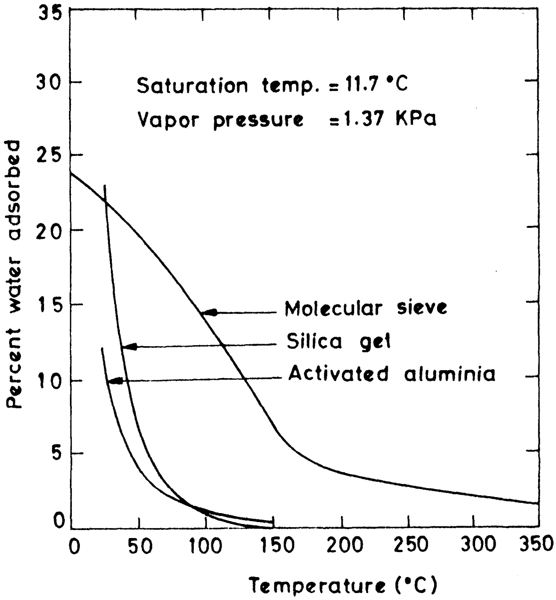
1. The auxiliary power used by vacuum pump required to deaerate the strong solution is more.
2. Since the absorbent-refrigerant solution flows over the collector and is open to the atmosphere, the mixing of dust and other materials from the atmosphere may effect the solution. Similarly tap water used as the refrigerant would effect the performance of the evaporator.
3. Rain may dilute the solution and may therefore effect the performance.

There are several solid and liquid materials which shows high affinity for water vapour as shown in table 5.7 and can be used for dehumidification.

The only liquid absorbents commercially used are aquous solutions of triethylene glycol in the Niagara system[154], and lithium chloride in the Kathabar system[155]. Out of solid absorbents silica gel[156,157] and molecular sieve[158] have been commercially tried in desiccant coolers. Liquid desiccants are preferred over the solid desiccants due to several advantages such as : (i) low power requirement, (ii) use of liquid-to-liquid regenerative heat exchanger improves the efficiency, (iii) heat and mass transfer from a liquid surface is better, and (iv) continuous cooling during desorption. Properties of few solid absorbents are shown[157] in figure 5.22. from fig. 5.22 (a) it is seen that at high relative humidity silica gel and alumina gel have high water adsorption capacity while at low relative humidity molecular sieve has high water adsorption capacity. The effect of temperature on the water adsorption capacity of these four solid absorbent is compared in fig.5.22 (b) at a saturation temperature of 11.7 °C. From this figure, it is seen that molecular sieve has high water adsorption capacity at low as well as at high temperature compared to silica gel and activated alumina. This property of molecular sieve is important during dehumidification since it will absorb more water vapour at a given temperature compared to silica gel. During regeneration, silica gel performs better since it can be regenerated at about 75 °C while molecular sieve is completely regenerated only at 140 °C. The isobar chart for silica gel is shown in fig. 5.23.



(a) Equilibrium curves



(b) Adorption capacity at different temperatures

FIG.5.22 WATER VAPOUR EQUILIBRIUM CURVES FOR SOLID ADSORBENTS. (From Gidaspow et al[157])

Table 5.7 List of few materials used for dehumidification.

Solid Desiccants
Silica gel
Molecular sieve
Zeolites (Natural)
Alumina gel
Activated alumina
Activated charcoal
Solid Absorbents
Calcium chloride
Lithium chloride
Phosphorous petoxide
Inorganic liquid absorbents
Lithium bromide
Lithium chloride
Calcium chloride
Potassium hydroxide
Sulphuric acid
Organic liquid absorbents
Monoethylene glycol
Diethylene glycol
Triethylene glycol
Methyl glycol
Glycerol

As discussed earlier the three important properties of a desiccant required for the design and analysis of a system are: water vapour pressure, temperature of desiccant, and water holding capacity of desiccant. Sometimes dew point of air is used in place of water vapour pressure because both are interrelated. The most commonly used liquid desiccants are aqueous solutions of lithium chloride and triethylene glycol: their equlibrium curves are compared in figure 5.24 for different concentrations. From this figure it is seen that lithium chloride solution is more effective as a sorbent (good dehumidifier) but higher regenerating temperatures are required which can not be supplied from

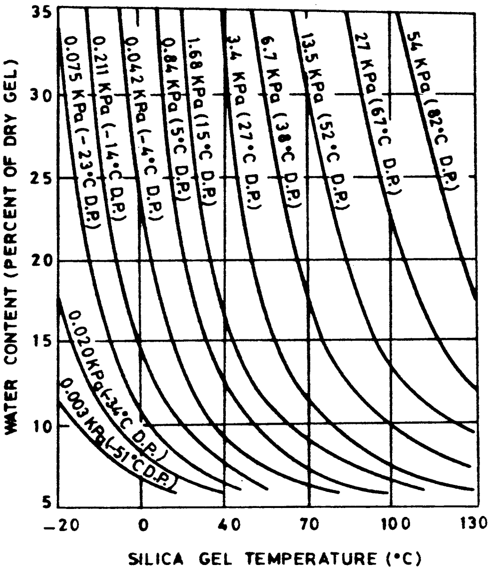


FIG.5.23 EQUILIBRIUM CHART(ISOBARS) FOR SILICA GEL (D.P. = DEW POINT).

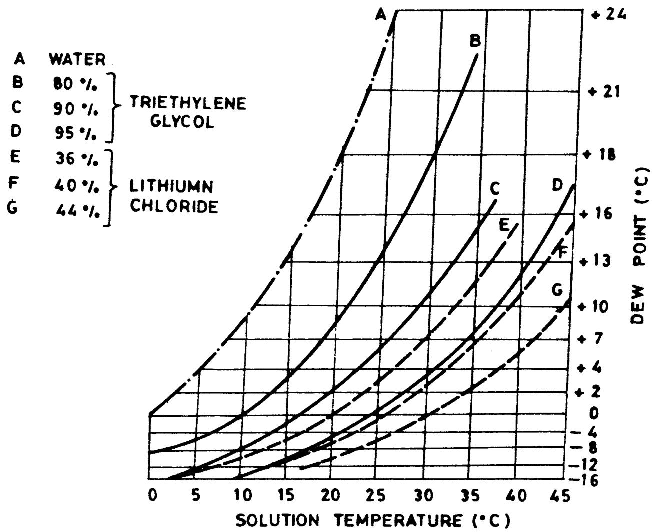


FIG.5.24 EQUILIBRIUM DIAGRAM FOR LIQUID SORBENTS

simple flat plate collectors. Highly concentrated triethylene glycol solutions are also quite effective like lithium chloride solution but require much lower regenerating temperatures. The disadvantages with the use of glycol solutions are its high cost and vapourization resulting in loss of material. The loss of material can be reduced to some extent by condensing the desiccant vapour before leaving to the atmosphere. Figure 5.25 shows the vapour equilibrium surves for four absorbents i.e lithium chloride, calcium chloride, sulfhuric acid, and triethylene glycol. From this figure it is clear that liquid desiccants

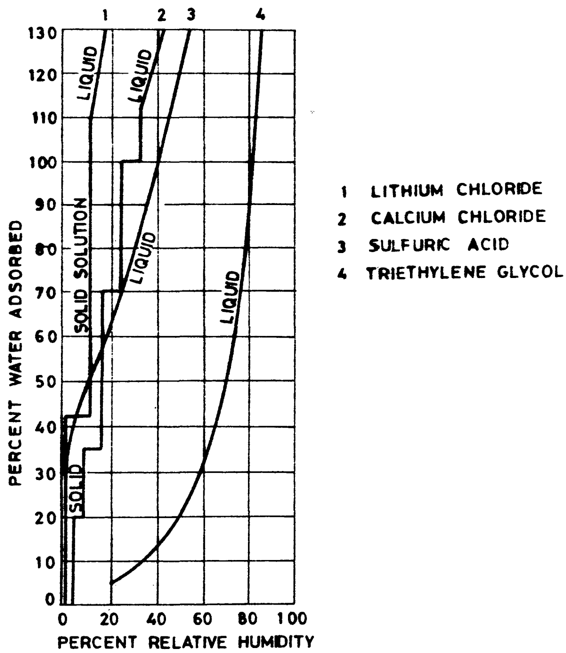


FIG.5.25 EQUILIBRIUM CURVES OF SOME ABSORBENTS.

have higher equlibrium water content capacity in terms of absorbent dry weight compared to solid desiccants. From figure 5.25 it is seen that in case of lithium chloride the relative humility remains very low and constant upto a water content of 42 per cent. The relative humidity of the solution (lithium chloride) remains at 11 per cent upto a water content of 110 percent where the solid changes into a liquid. During the dehumidification process the low value of equilibrium relative humidity is an advantage but higher temperatures are required for complete regeneration in case of lithium chloride desiccant. The equilibrium relative humidity in case of calcium chloride and triethylene glycol is much higher compared to lithium chloride and therefore less dehumidification will be provided with these two materials. However, lower regenerator temperatures are required in case of calcium chloride and triethylene glycol. Like aqueous solutions of lithium chloride and calcium chloride, the aquous solution of lithium bromide can also be used for dehumidifying the air. The lithium bromide like lithium chloride can also dehumidify the air upto a very low relative humidities. While using the aqueous solutions of salt one should provide a controller to control the solutions concentration otherwise there can be a problem of crystallization of the salts.

In the desiccant cooling known as humidification-process, the hot and humid air from rooms is first dehumidified with a solid or liquid desiccant,then cooled by exchange of sensible heat and then finally evaporatively cooled to the desired states. thus in desiccant coolers there are heat and mass exchangers required for dehumidification,heat exchangers, and evaporative coolers. Actually in the desiccant cooling system there are two processes: conditioning (cooling by dehumidification) and regeneration. The conditioning and solar regeneration processes are schematically shown in fig.5.26 and 5.27 respectively alongwith the cycles on the psychrometric charts. Figure 5.26 shows the principle of cooling by dehumidification (conditioning) in which warm humid air from room is first passed through a dehumidifier which is a active desiccant bed. In this process the air temperature is increased and its vapour content decreased as can be seen from line 1 to 2 shown on the corresponding psychrometric chart. From state point 2, the hot dry air is passed through a heat exchanger which is cooled either by cooling tower water or evaporatively cooled air and reaches to state point 3 where the temperature of air is reduced without any change in water content. The warm dry air at state point 3 is now passed through a conventional humidifier (evaporative cooler) and reaches to state point 4 with reduction in temperature but increase in moisture content. This cool humid air at state point 4 is supplied to the room for

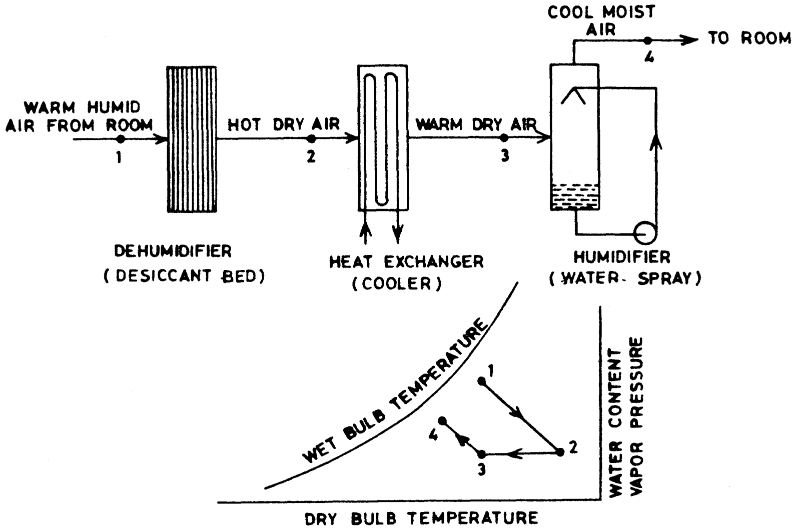


FIG.5.26 PRINCIPLE OF COOLING BY DEHUMIDIFICATION (Conditioning)

cooling. In due course of time the desiccant bed used for dehumidifying the warm humid air becomes inefficient due to absorption of sufficient moisture and hence require regeneration for further use as dehumidifier. The principle of regeneration using solar energy flat plate collector is shown in fig.5.27 alongwith the cycle showing various state points on a psychrometric chart. In the solar regeneration process the ambient air after passing from a heat exchanger passes through the solar collector and gets heated and reaches to state point 3. This hot air passes through the desiccant bed and carries moisture from the desiccant bed and reaches point 4. This warm moist air transfers its heat to the incoming ambient air in the heat exchanger, and finally is exhausted to the atmosphere. The desiccant again becomes active for use in the conditioning cycle.

Recently review articles on desiccant cooling are written by Dunkle and Close[159], Sharma[160], Shelpuk[161], and Satcunanthan and SoBrien[162]. A variety of desiccant cooling systems based on the principle are designed, ranging from simple to complicated designs. The pioneer work in the solar desiccant system is due to Lof[143] who used triethylene glycol as the desiccant.

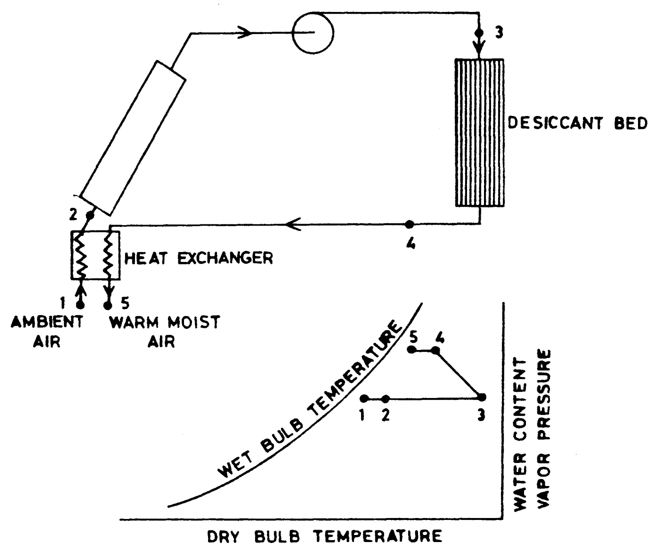


FIG.5.27 PRINCIPLE OF SOLAR REGENERATION

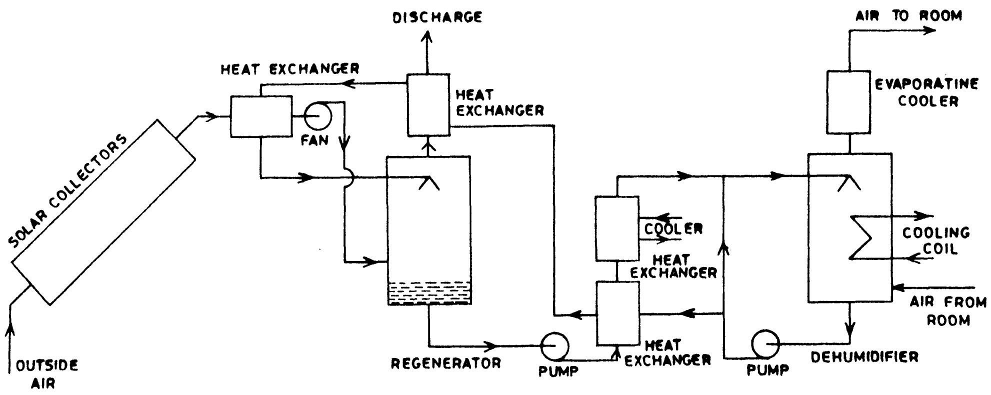


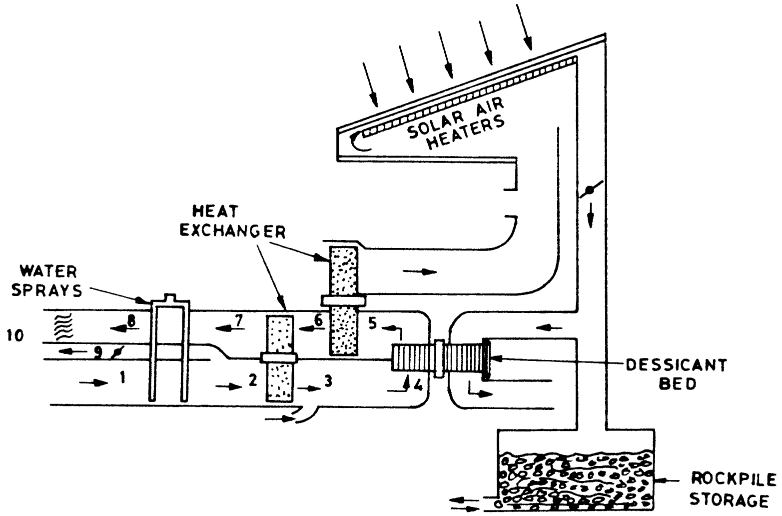
FIG.5.28 SCHEMATIC DIAGRAM OF TRIETHYLENE GLYCOL COOLING SYSTEM USING SOLAR AIR HEATING COLLECTERS FOR REGENERATION AS PROPOSED BY LOF

Figure 5.28 shows the simplified schematic diagram proposed by Lof. The air is dehumidified in a commercial dehumidifier, in which glycol solution is sprayed. From here the glycol is pumped through a heat exchanger to a stripping chamber (regenerator) sprayed counter-curently to solar heated air. The hot air takes a part of moisture from glycol solution and is exhausted to the atmosphere. The concentrated glycol is pumped back through the heat exchanger to the absorber, (dehumidifier). Four heat exchangers are used in the system. One heat exchanger which is liquid-to-liquid heat exchanger exchanges heat from hot glycol solution leaving the regenerator with the cold glycol solution coming from the dehumidifier. Second heat exchanger is used to exchange heat from the warm moist air leaving the regenerator with the glycol solution flowing to the regenerator. The third air-to-liquid heat exchanger is used at the outlet of the solar collector exchanging heat from solar heated air with the glycol solution entering the regenerator. The fourth heat exchanger is a liquid-to-liquid heat exchanger used to cool the glycol solution entering the dehumidifier by the cooling water coming from cooling tower. Eliminators are used to remove the glycol vapours going into the atmosphere.

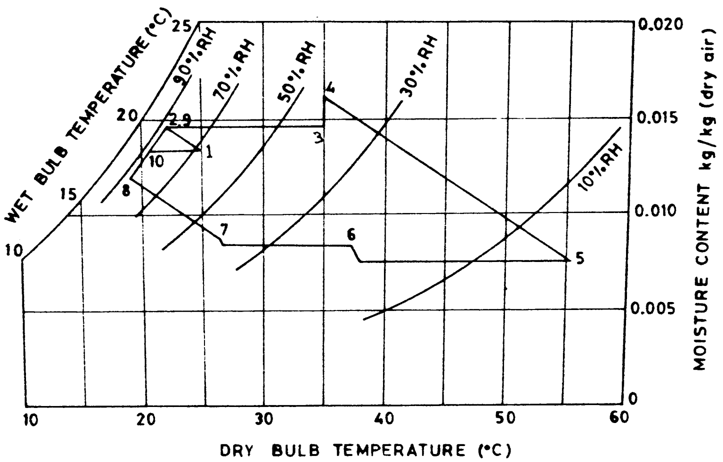
The total solar collector area was 10.2 m^2 which provided hot air in the temperature range of 60 °C to 80 °C able to remove 2.7 to 6.4 Kg water per hour from 90 percent concentrated glycol solution at a solar insolation of 30.6 MJ/m^2 day. This is equivalent to a cooling capacity of 0.5 to 1.2 tons of refregeration per 10 m^2 of solar collector for 8 hours. The overall coefficient of performance (COP) of the whole system starting from solar collector to cooling was of the order of 25 percent. From this experiment it was concluded that for 1 KW of cooling about 3 m^2 of collector area (air type) is required.

The above system can be simplified to a great extent by combining the solar collector and the regenerator. Such a system was first reported by Baum et al[140] and Kakabaev et al[151] where lithium chloride is used as the desiccant. This system is further modified by Mullick and Gupta[144] who used calcium chloride solution as a desiccant.

Dunkle[139] proposed a open cycle solar air conditioning system for operation in humid tropical and sub-tropical regions using a silica gel rotating desiccant system. The silica gel used as desiccant is regenerated using solar heated air and rock pile storage is used to store solar heat. Fig.5.29(a) shows schematically the air conditioning system alongwith solar air heating collectors and rock pile storage and fig.5.29(b) shows the air cooling cycle on a psychrometric chart. In this diagram the state point numbers correspond to the positions on the fig.5.29(a). It is seen from the figure that the outside or



(a) Desiccant cooling system



(b) Thermodynamic cycle

FIG.5.29 SCHEMATIC DIAGRAM AND RELATED THERMODYNAMIC CYCLE OF SILICA-GEL COOLING SYSTEM USING SOLAR HEATING COLLECTORS FOR REGENERATION WITH ROCK BED STORAGE SYSTEM AS PROPOSED BY DUNKLE room air at point 1 is evaporatively cooled to point 2 and a portion is sensibly heated to state point 2 on passage through a rotary regenerative heat exchanger. Here the air gets mixed up with outside air and reaches state point 4. It is then dehumidified by the desiccant bed. During dehumidification the temperature increases and the air reaches state point 5. From here, the hot dehumidified air exchanges heat with outside air (state point 6), then exchanges heat with room air (state point 7) and finally is evaporatively cooled and reaches to state point 8. This cooled air is supplied directly to the rooms after mixing with the part incoming air to provide desired comfort conditions. With the rotating silica gel system the problem of pulverization is anticipated and hence the fixed bed type desiccant system which is periodically regenerated is recommended. It appears that this Dunkle's system has not been practically tried.

The above system has been subsequently modified by Close and Dunkle[163] to include two beds of desiccant (silica gel) used for energy storage and dehumidification. At a given time one bed is used for dehumidifying the room air, and the other bed is generated by solar heated air.

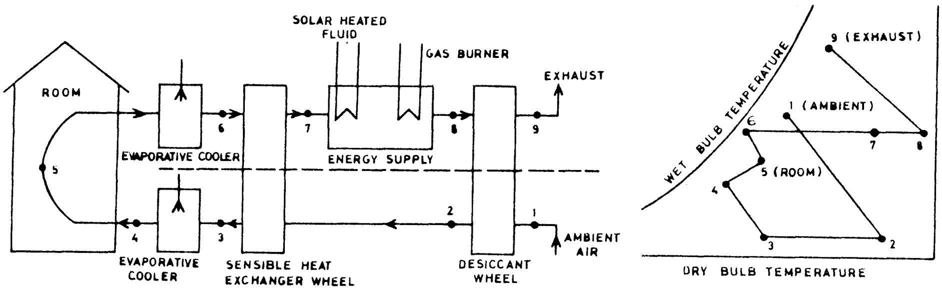


FIG.5.30 SCHEMATIC OF SOLAR MEC SYSTEM IN VENTILATION MODE AND ASSOCIATED CYCLE ON PSYCHROMETRIC CHART

The Institute of Gas Technology and its subsidiary, the Gas Development Corporation has developed an adsorption system which has been studied in great detail by Rush[145-148] and Nelson[165,166] using solid adsorbents. The system known as Solar-MEC (Munters Environmental Control) system as described by Rush can be used in two basic modes. In the ventilation mode (figure 5.30) fresh outside air is continually introduced into the conditioned room, and in the recirculation mode (figure 5.31) the exhaust air from the conditioned room is reconditioned and supplied to the conditioned room. The cycles on the psychrometric chart are also shown alongwith the corresponding schematic diagrams.

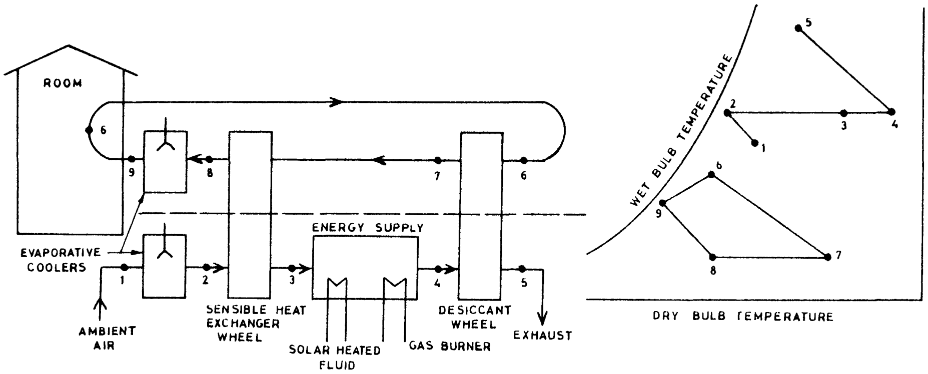


FIG.5.31 SCHEMATIC OF SOLAR MEC SYSTEM IN RECIRCULATION MODE AND ASSOCIATED CYCLE ON PSYCHROMETRIC CHART

In the ventilation mode as is seen from figure 5.30 there are two air ducts one for conditioning the air and another for regenerating the air placed side by side and connected by a rotary desiccant wheel (regenerator/dehumidifier) and a rotating sensible heat exchanger wheel. The desiccant wheel uses molecular sieve as desiccant material and the sensible heat exchanger wheel is made up of a aluminium honeycomb-shaped matrix. The wheel is built-up in sections. Two commercially available evaporative pads with high humidification ability, low pressure drop, and good anti-fouling characteristics are used alongwith two fans,one for each duct. In the regenerator stream, an energy supply arrangement which is a heat exchanger containing a finned-tube heat exchanger through which solar heated fluid exchanges heat with the air stream and a second element through which heat from direct fired gas heater is supplied to the air if required is used.In ventilation mode as shown in figure 5.30 the warm humid ambient air at State 1 passes through a rotating desiccant wheel from where the air gets dehumidified and heated in the adiabatic process and reaches State 2. From State 2 the air passes through a sensible heat exchanger getting cooled with change in moisture and reaches State 3. It then enters through an evaporative cooler, gets cooled to State 4 and enters the room to be conditioned. At this State 4 the air is at a lower temperature and water content than the room air at State 5, and therefore the sensible and latent loads can be met. The exhaust cooled air at State 5 from room enters the evaporative coolers and gets at State 6, cools the incoming fresh air through the sensible heat exchanger wheel and reaches State 7. Now this warm humidified air is further heated in a heat exchanger either by solar heated fluid, or if required by gas burners and reaches state 8 to regenerate the heat and mass exchanger wheel before it is exhausted to outside air at state 9.

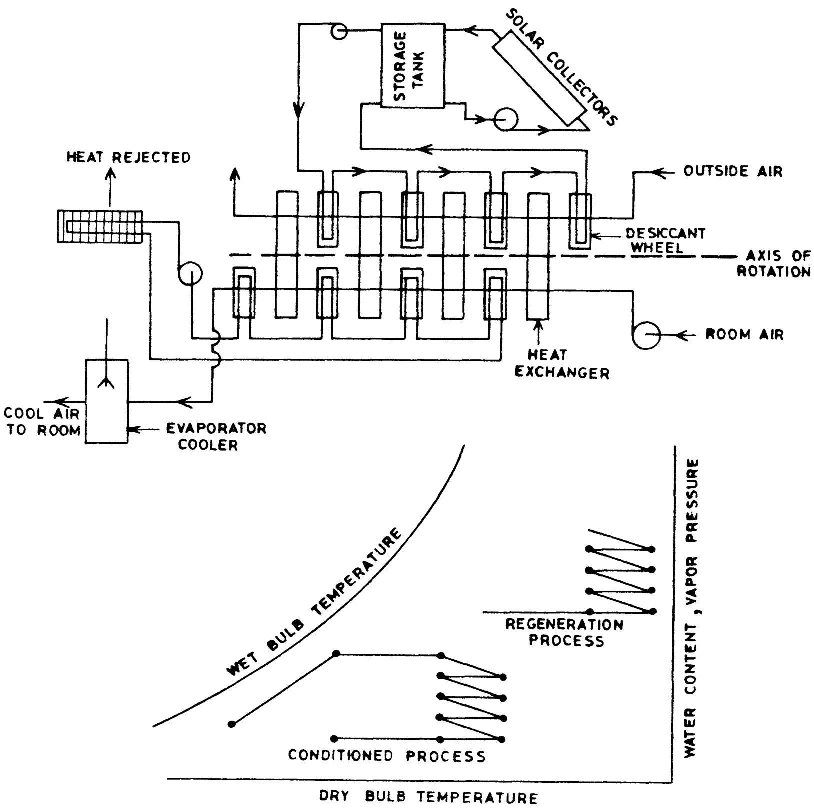


FIG.5.32 SCHEMATIC OF DESICCENT SYSTEM USING INTERSTATE HEAT TRANSFER AND ASSOCIATED CYCLE ON PSYCHROMETRIC CHART AS PROPOSED BY LUNDE

In the circulation mode as shown in figure 5.31 the room air is recirculated again and again and ambient air is used only for regeneration. In this mode the building return air is first dehumidified in the desiccant wheel, then cooled sensibly in the sensible heat exchanger wheel, then evaporated and supplied to the conditioned room. The difference between the recirculation mode and the ventilation mode is that in the MEC-recirculation mode, no fresh air is supplied, while in the MEC-ventilation system fresh air is supplied. Simulation, studies are conducted for cooling a room with floor area 140 m^2 at Miami (USA) with a sensible heat load of 3.8 KW and latent heat load of 1.0 KW using 45 m^2 collector area with selective coating and two glazings and a 3.375 m^3 hot water storage tank. It was observed that 95 percent of the energy required is met by the system and hot air at less than 60 °C is required for 60 percent of the time to regenerate the desiccant bed. Hot air at more than 80 °C is required for hardly 5 percent of the time.

The coefficient of the performance as determined experimentally, not including the efficiency of the solar collector, is found to be of the order of 0.55. It has also been observed while comparing the ventiliation mode and the recirculation mode, that the ventilation mode uses about one-half the auxiliary energy compared to recirculation mode. Dunkle and Close[159] have recently compared theoretically the MEC-ventilation system, MEC-recirculation system, and the system proposed by Dunkle[139] and found that the Dunkle system is superior in terms of thermodynamic performance and required system mass flow rate.

It was observed that due to large temperature swing, the adsorption and desorption capacity of the desiccant bed decreases. The adsorption and desorption capacity of the bed can be increased by using interstage cooling and heating of the bed as proposed by Lunde[141,142,167]. The concept is schematically shown in fig.5.32 alongwith the conditioned and regeneration process on the psychrometric chart. The warm humid room air is blown through a silica-gel-bed which consists of four rotating disc with four intermediate heat exchangers between them. Thus the air is passed through a series of heat exchangers and desiccant beds. The dried air coming out of four-stage heat exchanger is finally cooled in an evaporative cooler and supplied to the room. In the desorption mode solar heated air is used to regenerate the silica-gel-bed and is introduced in four inter-stage heat exchangers. The interstaging helps to achieve desorption at temperatures as low as 65 C and therefore this technique is efficient for providing cooling since inexpensive flat plate collectors can be used to reactivate the drier bed.

As mentioned earlier if the desiccant bed is maintained at a reasonably low temperature then the adsorption capacity can be increased considerably as shown by group of scientists at the Illinois Institute of Technology, USA[157,168,169]. In this cross flow desiccant cooling system as is shown schematically in fig.5.33,there are two identical fixed bed dehumidifiers, one of which dehumidifies the process air while the other is being regenerated by solar heated air. Cross cooling is achieved with evaporatively cooled air flowing through rectangular channels and the process stream flows in perpendicular channels lined with paper-like sheets made up of small sized silica-gel particles held in a Teflon Web. The system can be either in the recirculation mode or in the ventilation mode.

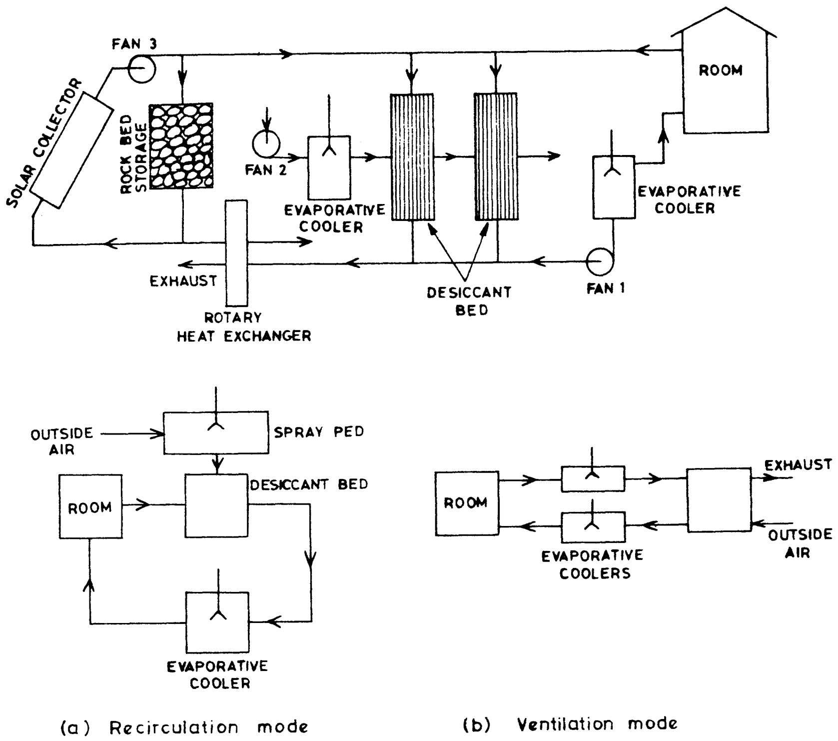


FIG.5.33 SCHEMATIC DIAGRAM OF STATIONARY CROSS DISICCANT BED COOLING SYSTEM DEVELOPED AT ILLINOIS INSTITUTE OF TECHNOLOGY

As is seen from the fig.5.33, fan 1 is used to blow evaporatively cooled room air through the desiccant channels while fan 2 circulates ambient air cooled by adiabatic humidification through the cross cooling channels. Simultaneously the second desiccant bed is desorbing with hot air either directly from the collector or through the storage tank by using fan 3. The regenerating air used in this bed is exhausted to the atmosphere because it contains high humidity. The heat from this hot high water content air is transferred to the ambient air going to the collector using a rotary heat exchanger. Small experimental studies and computer analyses have shown that the coefficient of performance of this cooling system excluding efficiency of the collector at a regeneration temperature of 82 C is about 0.52.

As discussed earlier there is a continuous open cycle desiccant cooling system which was earlier studied by scientists in the USSR[151,152] in which the weak liquid desiccant solution regenerated by losing refrigerant to the earth's atmosphere instead of to the condenser as is done in closed cycle. This regeneration of weak desiccant solution takes place either through regenerative type flat plate collector[170], or packed tower, or by spray chamber. These open-cycle desiccant cooling system where condenser is eliminated holds great promise compared to absorption and Rankine cycle cooling methods and have been studied by Kakaebaev[151,152], Baum[140], Collier[153], Lof[171], Lanz and Lof[172], Leboeuf and Lof[173], Johannsen[174], and Kimura and Tamura[175].

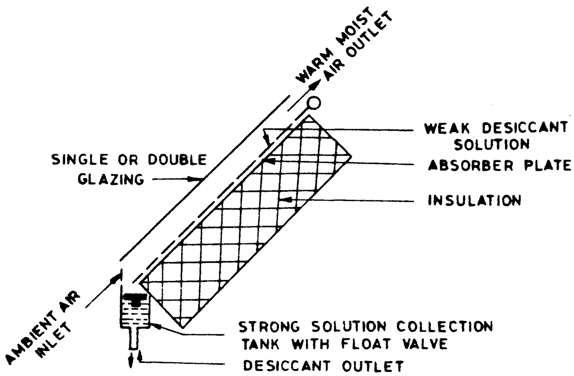


FIG.5.34 REGENERATIVE TYPE FLAT-PLATE COLLECTOR USED FOR REGENERATION OF LIQUID DESICCANT

A regenerating collector which is extensively studied by Mullick and Gupta[144], Johannsen[174], and Applebaum and Wood[176] where solar energy collection and regeneration of desiccant take place in the collector itself appear to be more effective. The regenerative type flat plate collector as is schematically shown in fig.5.34, consists of a flat or corrugated metallic sheet painted black absorbs solar radiation and insulted on the rear side and single or double glazing on the exposed side. The desiccant solution flows in a very thin layer from top to bottom and the ambient air enters through the opening in the lower side of the collector, picks up moisture and is heated and goes out from the upper opening of the collector. The desiccant solution

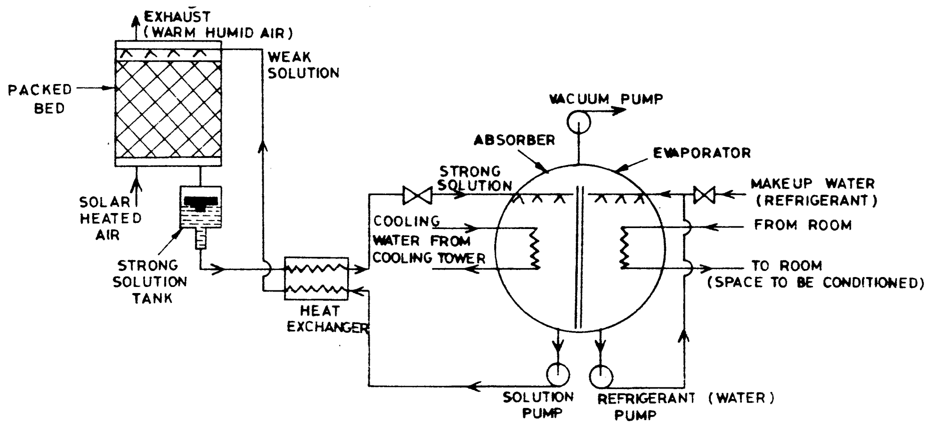


FIG 5.35 SCHEMATIC OF OPEN CYCLE ABSORPTION COOLER USING PACKED BED FOR REGENERATION OF DESSICANT AS PROPOSED BY LOF

flows down due to gravity and the concentrated solution is collected in the storage tank attached to the bottom of the collector. The ambient air moves up in the space of absorbing plate and glazing due to thermosyphon action. Due to simplicity of the collector, no pump requirement and production of highly concentrated desiccant solution, the regenerated collector is preferred.

The open cycle absorption cooling unit using packed bed (concentrator) for regeneration of desiccant solution as proposed by Lof[171,172] and using regenerative collector for regeneration of desiccant solution as proposed by Collier[153] are schematically shown in fig.5.35 and 5.36 respectively. The only difference between this unit and the closed cycle absorption unit is that here the weak solution is regenerated by allowing the refrigerant (water) to go to the atmosphere instead of recovering it in the condenser. The weak solution is regenerated in packed bed (fig.5.35) using solar heated air from flat plate collectors. The strong desiccant solution from the packed bed is allowed to pass through heat exchanger and then through a valve where the pressure is reduced from atmospheric levels and then

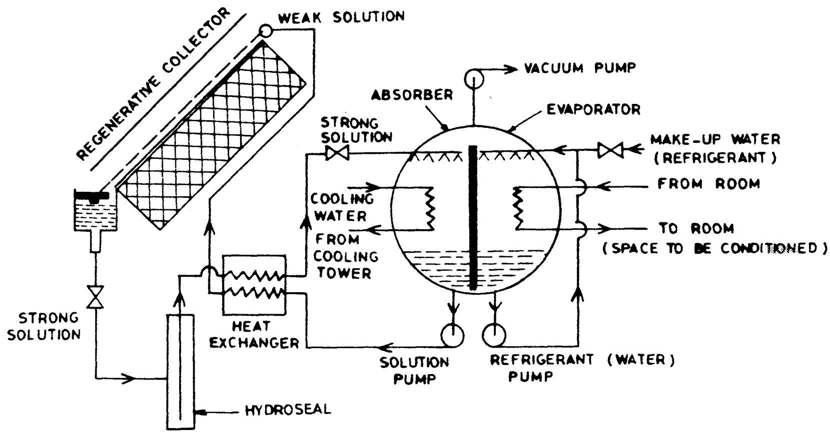


FIG.5.36 SCHEMATIC OF OPEN CYCLE ABSORPTION COOLER REGENERATIVE FLAT-PLATE COLLECTOR USED FOR REGENERATION OF LiCl AS STUDIED BY COLLIER

goes to the absorber where the solution absorb water from the evaporator maintaining the reduced pressure required by the evaporator[177]. The heat of adsorption is removed from the absorber using cooling water from cooling tower. In the evaporator the make-up water from an external source is evaporated at a reduced pressure taking heat supplied by the air coming from the room which is to be cooled. The weak desiccant solution is pumped back from the absorber through the heat exchanger at atmospheric pressure to the packed bed. The vacuum pump is used initially to bring the absorber and evaporator assembly at predetermined vacuum level. The effectiveness of the cooling unit depends on the amount of the water evaporated in the evaporator. It has been observed that for 1 kg of water evaporated in the packed bed, 1 kg of water can be evaporated in the evaporator and absorbed in the absorber.

The most important part of a desiccant cooling system is the dehumidifier, the design of which is different for liquid desiccant and solid desiccant. Several designs of liquid desiccant dehumidifiers and solid desiccant dehumidifiers are described by Grossman and Johansen[2]. The dehumidifier should be designed in such a way that during dehumidification the heat of adsorption should be removed at a fast rate providing isothermal conditions. The liquid desiccant dehumidifiers as described by Grossman and Johansen[2] are of three types: packed tower, spray chamber, and sprayed-coil chamber. In a packed tower dehumidifier, the weak desiccant solution is sprayed from the top of a packed tower which flows down and hot air is blown in the opposite direction. In the spray chamber dehumidifier, the weak desiccant solution is atomized in the air through high pressure nozzles and the hot air is blown in upward direction. In the sprayed coil dehumidifier the dehumidification and sensible cooling is done in the same unit. The weak desiccant solution is sprayed on to the outer surface of the finned heat exchanger through which cooling water flows. The hot air is blown from the bottom which picks up the moisture passing through the outer surface of the finned heat exchanger. There are advantages and disadvantages of each of the three liquid desiccant dehumidifiers but the sprayed coil dehumidifier is considered to be most effective.

Two types of solid desiccant type dehumidifiers namely the stationary bed type and rotary bed type developed by Illinois Institute of Technology (IIT)[157,169], USA and Airesearch Manufacturing[178,179] company are shown in fig.5.37 and 5.38 respectively. The IIT has developed a cross-cooled dehumidifier using silica-gel as a desiccant and same is shown in fig.5.37. The dehumidifier consists of a paper like sheet of silica-gel in a Teflon matrix[177]. The heat of adsorption is removed by evaporatively cooled air which is allowed to pass through alternate channels of the heat exchanger in cross flow arrangements.

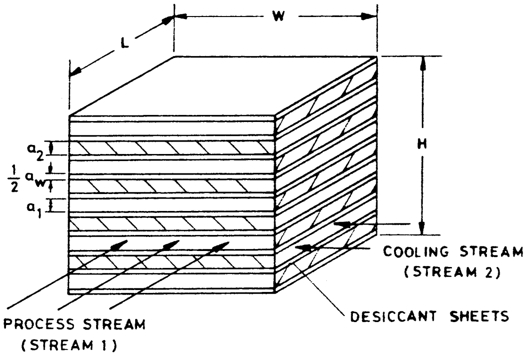


FIG.5.37 SCHEMATIC OF CROSS-COOLED DEHUMIDFIER (From Rousseau [178])

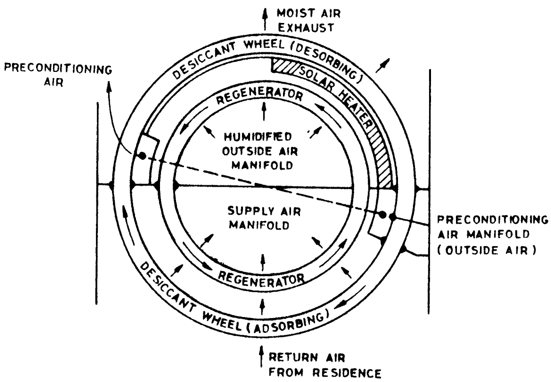


FIG.5.38 CROSS-SECTION OF ROTARY RADIAL-FLOW TYPE DEHUMIDIFIER (From Rousseau[178])

The rotary radial-flow dehumidifier is developed by the Airesearch Manufacturing Company, California, USA is shown in fig.5.38[178,179]. In the design the parasitic power requirement is reduced by increasing the face areas of the dryer and regenerating heat exchanger and reducing the path length of air flow. Silica-gel is used as a desiccant material and the regenerator matrix is a fine screen of galvanized iron. The desiccant bed and regenerator are two counter-rotating concentric drums. The dehumidifier drum is 86 cm in diameter, 86 cm high, 3 cm long air flow length along the desiccant and rotates with a speed of 5 rpm. The prototype dehumidifier is designed for a 5.3 KW cooling capacity with parasitic requirement of 0.5 KW providing COP of 0.52.

5.5 VAPOUR COMPRESSION REFRIGERATION

As discussed earlier the Rankine cycle operated solar cooling system has an advantage that the system apart from providing cooling can be used in the heat pump mode, and can also be used for electricity generation. Although conventional vapour compression refrigeration machines are being used widely and studied for the last 100 years and machines ranging from small domestic unit of 0.5 ton capacity to airconditioning plant of 300 tons capacity are available, solar energy operated vapour - compression cooling machines are comparatively recent. The main problem lies in the efficient and economical conversion of solar energy into mechanical energy and the storage of solar energy. In a conventional vapour compression cycle, a suitable vapour is compressed, then condensed to a liquid, following which the pressure gets dropped and fluid is evaporated at a low pressure producing cooling. Thus in this process heat is extracted from the space to be cooled used to evaporate fluid at low pressure, and is rejected in condensing it at high pressure. The necessary pressure rise between evaporator and condenser is provided by a pump operated by a heat engine. The future of solar Rankine air conditioning system will depend on the economic conversion of solar energy into mechanical power. The vapour compression refrigeration systems are energy efficient system compared to desiccant cooling and absorption cooling systems but this is possibly at the expense of somewhat lower reliability.

5.5.1 History

Although vapour compression refrigerators and airconditioners are in use for the last 100 years, the studies on Rankine - cycle operated solar refrigerators and air conditioners are quite recent. Recently very good reports[9,180-184] of notable advance in this area have appeared. Some theoretical studies on solar powered Rankine cycle airconditioning systems are conducted by Teagen and

Table 5.8 Details of some Rankine cycle solar cooling projects in USA (From Koai et al[184])

Firm	Power cycle fluid	Engine	RPM	Compressor	Cooling cycle fluid	Solar supply temp. (°C)	Cooling capacity, (tons)	Comments
1	2	3	4	5	6	7	8	9
Barber-Nichols/ Honeywell, Inc.	R-113	Turbine	39,500 52,000	3600 RPM Reciprocating	R-12	102-149	3,14,25,50	
Carrier, MTI	R-113	Turbine	20,000	Centrifugal	R-113	143	25	Single-shaft turbo-compressor
Foster-Hiller Assoc.	R-22	Recipro-cating	1,200	Reciprocating	R-22	60 +	20	+ For waste heat use; common cylinder block condenser
Garrett-Air Research	R-11	Turbine	24,800 82,600	Centrifugal	R-11	93	3,25,75	Single-shaft turbo-compressor; heat pump
General Electric Corpn.	FC 88 (flour-inated hydro-carbon)	Rotary vane expander	1,200	Reciprocating	R-22	100-140	3,10	Heat pump

Table 5.8 cont.

1	2	3	4	5	6	7	8	9
United Technologies Research Centre	R-11	Turbine	45,000	Centrifugal	R-11	143	18	Single-shaft turbo-compressor; heat pump
University of Pennsylvania	Steam	Turbine	15,300	1750 RPM commercial (Trane) open, reciprocating; or other	R-22	100	25	Hybrid cycle; 20% of the energy supplied by fuel of superheat steam to 600 C

Sargent[185], Beekman[186], Olson[187], and Curran et al [188]. The use of solar energy for providing medium temperatures and then fossil fuel to attain higher temperatures in vapour compression cycles known as solar powered/fuel assisted vapour compression cooling systems (hybrid solar/fuel-Rankine cycle) can achieve high COP and reliability. These systems are studied extensively by Curran[189], and Lior et al[183,184,190,191]. Several solar Rankine airconditioning prototypes have been developed by Batton and Barber[192], Werner[193], Abbin[194], Biancardi and Meader [195], Barber[196,197], McDonald[198], English [199], Kolenc et al[200], Graf[201], Scarborough[202], Lior and Yeh[203] and Biancardi et al[204]. The details of some of the projects are given[184] in table 5.8.

5.5.2 Principle of vapour compression cycle

The solar operated Rankine cycle airconditioning system consists of three loops as shown in fig.5.39. In the solar loop solar energy collectors are used to heat water or any other working fluid and the heat is stored in the storage tank. In the Rankine loop, there is a heat exchanger in which the working fluid is vapourised, the vapour is expanded in the expander (heat engine), condensed

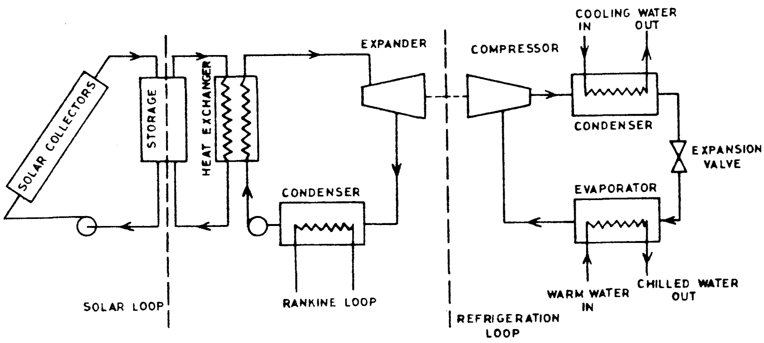


FIG.5.39 SCHEMATIC OF SIMPLE RANKINE CYCLE AIR CONDITIONING SYSTEM

in the condenser and returned to the heat exchanger by a feed pump. This expander drives the compressor of a conventional vapour compression refrigerator. The refrigeration or vapour compression loop consists of a compressor, a condenser, an expansion valve for throttling and an evaporator. The principle of operation of vapour compression refrigeration cycle can be well understood with the help of the pressure - enthalpy (p-h) diagram as shown in fig.5.40. In this figure, the refrigerant pressure (N/m^2) is shown on ordinate and enthalpy (KJ/Kg) on abscissa. The various thermodynamic processes are as follows:

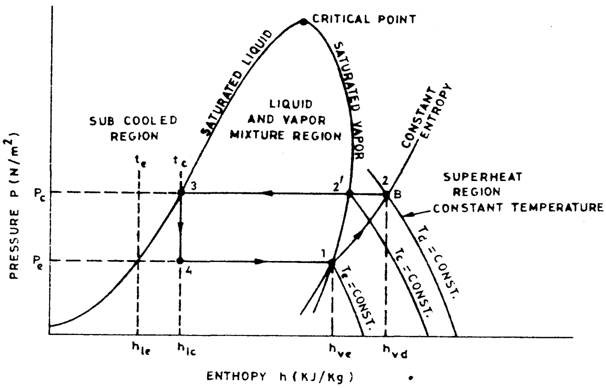


FIG.5.40 SIMPLE VAPOUR COMPRESSION CYCLE ON PRESSURE ENTHALPY DIAGRAM

(i) the process 1 to 2 represents the compression of refrigerant vapour from pressure Pe to Pressure Pc. This is isentropic compression where entropy remains the same (S =S). The work of compression W_C (KW) is given as :

$$W_C = \dot{m} \quad (h_{vd} - h_{ve}) \tag{5.19}$$

where $\dot{m}$ is the mass of refrigerant circulated (Kg/s), and h_{vd} and h_{ve} are enthalpy (KJ/Kg) vapour at compressor discharge and evaporator outlet respectively.

(ii) the process 2 to 3 represents the desuperheating and condensation of refrigerant. First the sensible heat is removed at constant pressure P_c from process 2 to 2' from temperature T_d to T_c and then in the condenser latent heat at constant temperature T_c at condenser saturation pressure P_c is removed from process 2' to 3. The total heat rejected q_c (KW) in the condenser is given as :

$$q_c = \dot{m}\,(h_{vd} - h_{lc}) \tag{5.20}$$

where h_{lc} is the enthalpy of liquid refrigerant at condenser temperature.

(iii) the process 3 to 4 represents the isenthalpic expansion where hot liquid refrigerant at condensing pressure P_c passes through the expansion valve and pressure drops to the pressure in the evaporator P_c. If x is percentage mass of liquid vapourised then

$$h_{lc} = h_{ve} + h_{le}\,(1-x) \tag{5.21}$$

or
$$x = \frac{h_{lc} - h_{le}}{h_{ve} - h_{le}} \tag{5.22}$$

(iv) the process 4 to 1 represents the vapourisation of liquid refrigerant at constant pressure Pe and the transfer of heat from the water being chilled to the refrigerant in the evaporator. The refrigerating effect per Kg of refrigerant q_r (KJ/Kg) is given as

$$q_r = (h_{ve} - h_{le})\,(1-x) \tag{5.23}$$

$$= (h_{ve} - h_{lc}) \tag{5.24}$$

The coefficient of performance COP of a refrigeration machine is given as the ratio of energy removed at the evaporator (refrigeration effect) to energy supplied at the compressor. This $(COP)_c$ is given as

$$(COP)_c = \frac{h_{ve} - h_{lc}}{h_{vd} - h_{ve}} = \frac{q_r}{W_c} \tag{5.25}$$

and COP for heating $(COP)_h$ is given as

$$(COP)_h = \frac{h_{vd} - h_{lc}}{h_{vd} - h_{ve}} \tag{5.26}$$

Between these two temperature limits i.e. T_c and T_e, the COP of a carnot cycle is given as

$$(COP)_{carnot} = \frac{T_e}{T_c - T_e} \qquad (5.27)$$

In practice the (COP) is about 0.75 to 0.85 to that of $(COP)_{carnot}$ and largely depends on the properties of the refrigerant fluid.

5.5.3 Refrigerants

Refrigerants are those fluids which absorb the heat energy from the low temperature source and convey it to the high temperature sink either in the form of sensible heat or in the form of latent heat. The refrigerants which carry the heat and dissipate the heat in the form of latent heat only are more efficient than those refrigerants which carry and dissipate the heat in the form of sensible heat. There is a large variety of refrigerants for use in vapour compression systems but before deciding on a particular refrigerant, their thermodynamic, chemical and physical properties and safety characteristics must be carefully studied. The refrigerant properties are : i) the condensation and evaporation pressure should not be very high, ii) the critical temperature should be high but the freezing temperature should be low, iii) the latent heat of vapourization and vapour specific heat should be high, iv) it should have low viscosity and high film heat conductivity, v) the boiling temperature and specific heat of liquid should be low and the specific volume of vapour should be low. Apart from the above thermal properties, some of the desirable properties are: i) it should be easily available at low cost, ii) it should be non-toxic, non-flammable and non-explosive, iii) it should be non-corrosive to construction materials and should not effect the lubricant used, and iv) it should be easily detactable on leaking.

The most important property of a refrigerant for use in vapour compression system is the normal boiling point since this helps in selecting a fluid which will be above atmospheric pressure on the low side. Most widely used refrigerants are the halogenated fluids (such as R-11, R-12, R-22) and NH_3. A comprehensive list of refrigerants is prepared by Arora[20] and the same is given in Table 5.9 The vapour pressure of some of the refrigerants as a function of temperature is plotted in fig.5.41. This figure is useful in selecting a particular refrigerant for a desired pressure provided the condenser temperature and the temperature required in refrigeration is known. The performance of some of the commonly used refrigerants used in the vapour

Table 5.9 Properties of some refrigerants (From Arora[20])

Refrigerant	Chemical Formula	Designation	Molecular weight	Boiling point (°C)	Critical temperature (°C)	Critical pressure (MPa)	Critical volume (1/Kg)	Freezing point (°C)	$\frac{C_p}{C_v}$
1	2	3	4	5	6	7	8	9	10
Inorganic Refrigerants									
Water	H_2O	R 718	18.0	100.0	374.1	21.71	3.26	0.0	1.33
Ammonia	NH_3	R 717	17.0	-33.3	132.4	11.08	4.13	-77.7	1.31
Carbon Dioxide	CO_2	R 744	44.0	-78.5	31.0	7.23	2.15	-56.6	1.30
Sulphur Dioxide	SO_2	R 764	64.0	-10.0	157.2	7.72	1.92	-73.2	1.26
Nitrous-Oxide	N_2O	-	44.0	-88.4	36.5	7.13	2.18	-80.2	-
Sulphur Hexafluoride	SF_6	-	146.0	-63.8	45.5	3.67	1.35	-50.8	1.06
Organic Refrigerants									
Monofluoro Trichloro Methane	$CFCl_3$	R 11	137.4	23.7	197.7	4.28	1.80	-111.0	1.13
Difluoro Dichloro Methane	CF_2Cl_2	R 12	120.9	-29.8	112.0	4.04	1.79	-136.0	1.14

Table 5.9 cont.

1	2	3	4	5	6	7	8	9	10
Trifluoro Monochloro Methane	CF_3Cl	R 12	104.5	-81.5	28.7	3.80	1.72	-180.0	-
Trifluoro Monobromo Methane	CF_3Br	R 13 B1	148.9	-58.7	67.8	3.97	-	-142.0	1.11
Tetrafluoro Methane	CF_4	R 14	88.0	-128.0	-45.5	3.68	1.50	-194.0	1.22
Monofluoro Dichloro Methane	$CHF Cl_2$	R 21	102.9	8.9	173.5	5.06	1.91	-135.0	1.16
Difluoro Monochloro Methane	$CHF_2 Cl$	R 22	86.5	-40.8	96.0	4.84	1.90	-160.0	1.16
Trifluoro Methane	CHF_3	R 23	70.0	-82.2	-	-	-	-160.0	-
Dichloro Methane	CH_2Cl_2	R 30	8.9	39.2	235.4	5.85	-	-96.7	1.18
Methyle Chloride	CH_3Cl	R 40	50.5	-23.7	143.1	6.55	2.7	-97.6	1.20
Trifluoro Trichloro Ethane	$CFCl_2$-CF_2Cl	R 113	187.9	47.6	214.1	3.35	1.73	-36.6	1.09

Table 5.9 cont.

1	2	3	4	5	6	7	8	9	10
Tetrafluoro Dichloro Ethane	CF_2Cl-CF_2Cl	R 114	170.9	3.5	145.8	3.21	1.71	-94.0	1.107
Pentafluoro Mono-Chloro ethane	CF_2ClCF_3	R 115	154.5	-38.0	80.0	3.17	1.68	-106.0	1.09
Difluoro Monochloro ethane	CH_3-CF_2Cl	R 142	100.5	-9.2	136.4	4.04	2.30	-130.0	1.13
Trifluoro Ethane	CH_3-CF_3	R 143	84.0	-47.6	73.1	3.71	2.30	-111.3	-
Difluoro Ethane	CH_3-CHF_2	K 152	66.0	-25.0	113.5	4.40	2.74	-	-
Ethyl Chloride	C_2HCl	R 160	64.5	12.0	187.2	5.15	3.03	-138.7	1.16
N-perfluoro Butane	C_4F_{10}	-	238.0	-2.0	113.2	2.28	1.58	-	-
<u>Hydrocarbons</u>									
Ethane	C_2H_4	R 170	30.1	-88.6	32.1	4.83	4.70	-183.2	1.25
Propane	C_3H_8	R 290	44.1	-42.1	96.8	4.17	4.46	-187.1	1.13
N-butane	C_4H_{10}	-	58.1	-0.5	153.0	3.46	4.29	-135.0	-
Isobutane	$(CH_3)_3CH$	-	58.1	-11.7	133.7	3.62	-	-159.6	-

Table 5.9 cont.

1	2	3	4	5	6	7	8	9	10
Unsaturated Hydrocarbons									
Propylene	C_2H_4	-	42.1	-47.7	94.4	4.51	4.20	-185.0	-
Dichloroethylene	$C_2H_2Cl_2$	-	96.9	90.0	243.0	5.38	-	-56.6	1.14
Aliphatic Amines									
Methylene	CH_3NH_3	-	31.1	-6.7	156.9	7.31	-	-92.5	1.18
Ethylamine	$C_2H_4NH_3$	-	45.1	7.0	164.6	5.36	-	-93.0	1.15

Table 5.10 Comparative performance of refrigerants evaporating at 5°C condensing at 40°C (From Jones[205])

Name of refrigerant	Suction Temperature (°C)	Evaporating pressure (KPa)	Condensing pressure (KPa)	Compression ratio	Refrigerating effect (KJ/Kg)	Specific vol. of vapor (m^3/Kg)	Compressor displacement (litre/s KW)	Power in KW per KW of refrigeration	% Carnot cycle efficiency
Water	5	0.90	7.4	8.22	2370.0	147.0	62.0	0.1355	92.9
Trichloromono-fluoromethane	5	49.6	174.7	3.52	157.0	0.332	2.12	0.1395	90.2
Ammonia	5	516.0	1555.0	3.01	1088.0	0.243	0.214	0.1456	86.4
Dichlorotetra-fluoroethane	12.7	106.2	337.3	3.18	106.2	0.122	0.14	0.1484	84.8
Dichlorodifluoro-methane	5	362.6	960.7	2.65	115.0	0.047	0.409	0.1502	83.8
Trichlorotri-fluoroethana	10.4	18.8	78.3	4.16	129.5	0.652	5.03	0.1511	83.3
Monochlorodi-fluoromethane	5	583.8	1534.0	2.63	157.8	0.040	0.255	0.1518	82.9
An azeotropic mixture	5	667.8	1677.0	2.51	101.0	0.026	0.259	0.1631	77.1

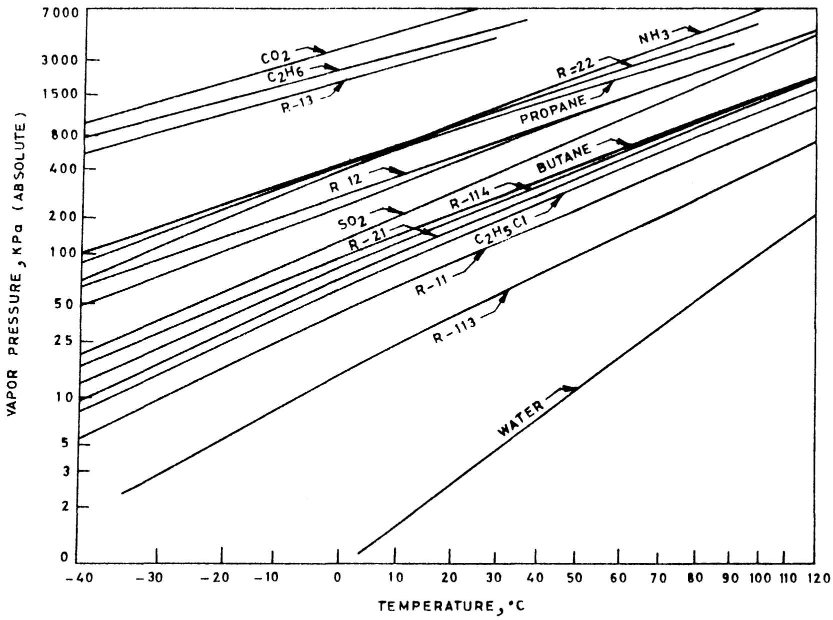


FIG. 5.41 VARIATION OF PRESSURE WITH TEMPERATURE FOR REFRIGERANTS

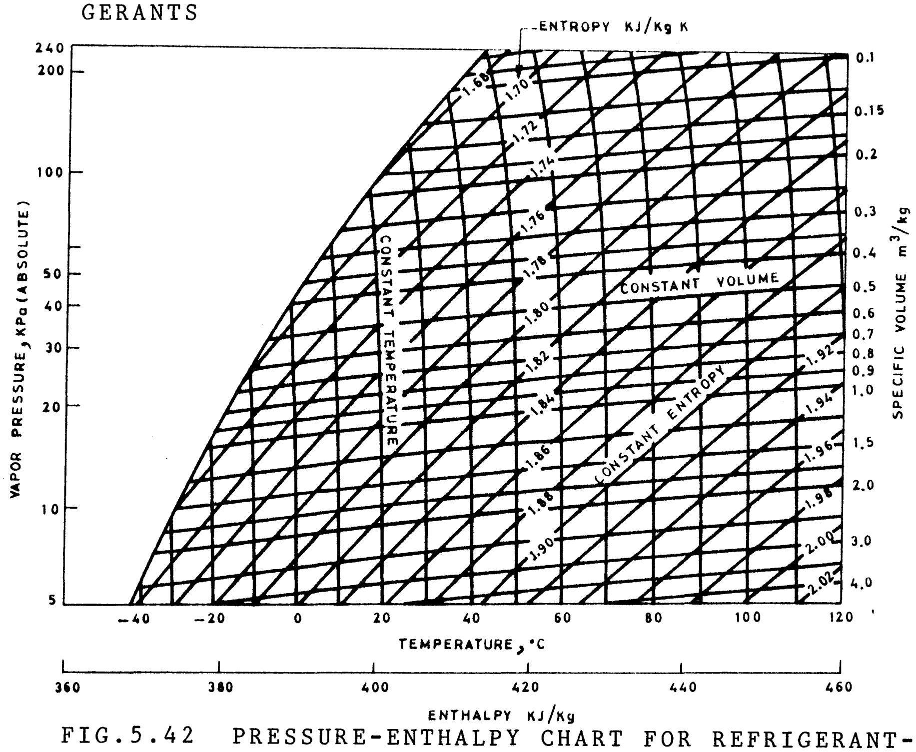


FIG. 5.42 PRESSURE-ENTHALPY CHART FOR REFRIGERANT-11

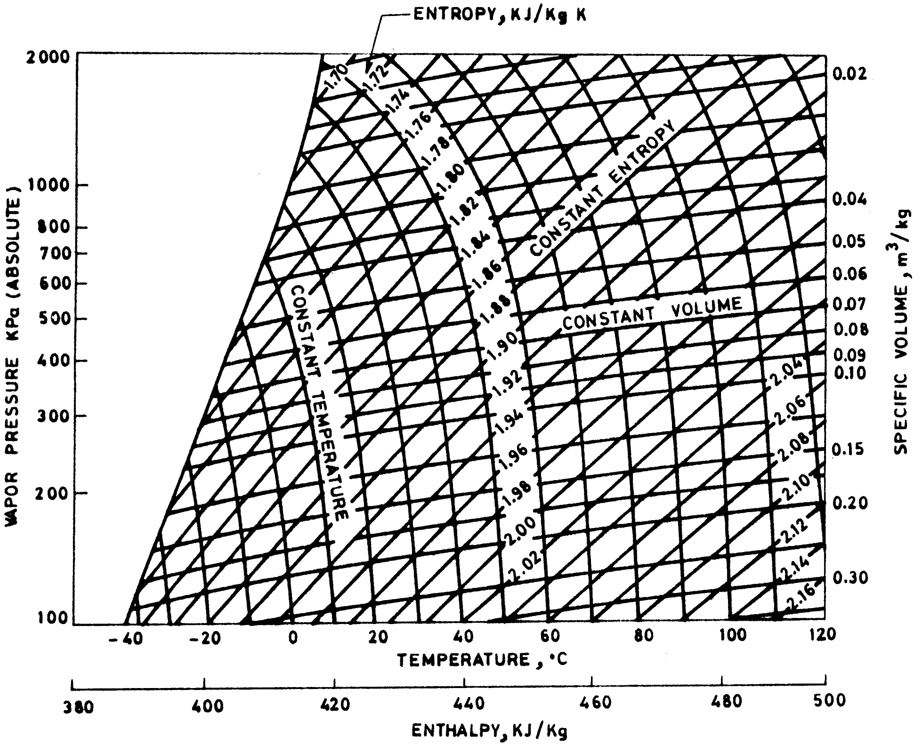


FIG.5.43 PRESSURE-ENTHALPY CHART FOR REFRIGERANT-22

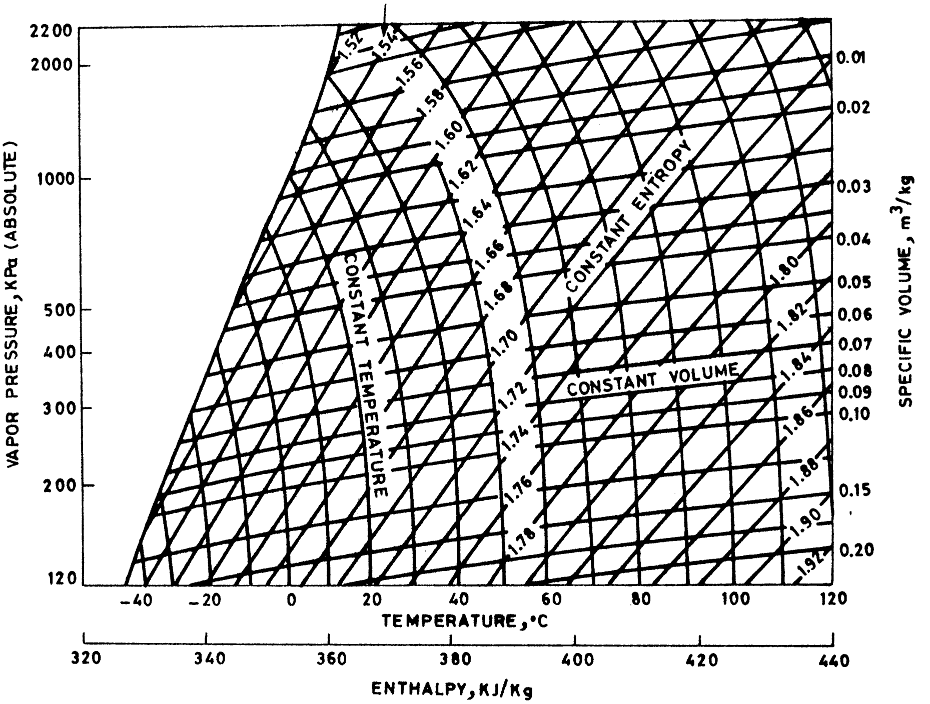


FIG.5.44 PRESSURE-ENTHALPY CHART FOR REFRIGERANT-502

compression cycle is compared[205] in table 5.10. While doing the comparison it has been assumed that vapour enters in the compressor in a saturated condition at 5 °C and the condensing temperature is 40 °C.

Extensive tables and charts showing the properties of refrigerants are given in many handbooks like ASHRAE Hand book of Fundamentals[83]. Pressure-enthalpy charts for R-11, R-12, and R-502 are given in figure 5.42, 5.43 and 5.44 respectively.

5.5.4 Typical designs

As discussed earlier, recently several studies are conducted[206] on the design, development and construction of solar operated vapour compression cooling devices. Prigmore and Barber[9] described a Honeywell 'Mobile Solar Research Lab.' developed at the Barber-Nichols Engg. Co., Colorado, USA which consists of a trailer with 58 m^2 flat plate solar collectors and a 3-ton (10.6 KW) Rankine cycle cooling unit. Flat plate collectors are used to heat the water upto 100 °C which is used to heat the working fluid R-113 which is the working fluid in the vapour power cycle and the same is expanded in a high speed turbine of 50,000 rpm. In the condenser cold water at 29 °C is used. The efficiency of the Rankine cycle was of the order of 11 percent. The shaft of the turbine rotates a conventional 12 piston compressor at a speed of 3600 rpm (or an electric generator if required) of the refrigeration unit. The refrigerant used in the cooling cycle is R-12 with the evaporator temperature of 7 °C. The overall coefficient of performance was about 0.74.

The General Electric Co. of USA[201] has developed solar operated vapour compression systems. One 3-ton system has been installed at Southern Methodist University and two 10-ton systems have been delivered to TVA for their installation and a 20 ton unit is being fabricated. The solar energy is collected by GE TC - 100 evacuated tubuler collectors operating at temperature of 160 °C. In the solar loop a mixture of ethylene glycol and water is used and heat is stored in a storage tank. The working fluid in the Rankine cycle loop is flourinated hydrocarbon, FC 88, which is circulated by a feed pump through the vapour generator and into the expander. A modified GE Weathertron Compressor which uses R-22 as the working fluid is used in the vapour compression loop.

The Barber-Nichols Engg. Co.[193] , Colorado, USA has built a solar powered Rankine cycle (63.4 KW capacity) to drive a conventional 100-ton water chiller for Honeywell, Inc. USA. The system designed will meet the 60 percent of the energy for heating and cooling and 100 percent of hot water requirement for its eight storey office building in

Minneapolis, USA. Advanced solar concentrating collectors of about 1881.3 m^2 are used. Caloria HT-43 is used to transfer heat and the Rankine cycle uses a working fluid R-113. The expander was designed and manufactured by Barber-Nichols Engg. Co. which provides 63.8 KW shaft power with 148.9°C oil and 29.4°C cooling water at 15.6 percent cycle efficiency. The firm had made several solar Rankine cycle airconditioners (23-ton capacity) which can be operated using flat plate collectors operating at about 80°C.

5.5.5 Solar cells operated vapour compression cooling system

In remote areas or in isolated villages in developing countries the photovoltaic operated refrigerators can play a significant role. These refrigerators can be used in hospitals and health centres for storing vaccines and sensitive medicines. These may also be used to some extent for preservation of food materials. The World Health Organisation (WHO) has chalked out an ambitious programme for installation of solar photovoltaic refrigerators for use in hospitals in rural areas and has identified a potential market of about 10000 solar photovoltiac refrigerators per year over the next 10 years.

In the small photovoltaic powered refrigerators, there is a photovoltaic array, series of storage batteries, voltage regulator and the conventional vapour compression refrigerator. Rosenblum et al[207] have estimated that a photovoltaic panel of 100 watts (peak power) will be able to provide 0.14 m^3 refrigerator capacity and the same can be used for preserving a small amount of food. In Arizona[207] a photovoltaic panel of 88 m^2 area of 3.5 KW peak power output is used to operate refrigerators and for generating electricity for use in other applications. In all, 15 refrigerators each of 0.13 m^3 are installed in a building for domestic use in the Arizona Project. One unit consists of 3 refrigerators which is powered by single compressor of 93 watts, 120 Volt DC permanent magnet motor. Each refrigerator is heavily insulated with an automatic door closer. In the Arizona Project fifty two 2380 ampere-hour storage batteries are used to store the electricity. Derrick and McNelis[208] have pointed out many problems which have been experienced with the photovoltaic refrigerators installed in rural areas. It has been pointed out that although photovoltaic solar refrigerators are commercially available in many countries, there is a scope for further development and improvement. Starr and McNelis[209] have discussed the technical and economic requirements of photovoltaic refrigerators and concluded that photovoltaic refrigerators are cost competitive with the main alternatives in remote areas and large markets are foreseen when suitable systems become available.

In many developing countries photovoltaic power plant of different capacities are installed ranging from 200 watts to 100 KW. In Indonesia[210] 25.5 KW photovoltaic power plant is installed in the fishing village of Cituis, Regency Tungerang (West Jawa) and the power plant output is used to drive a reverse osmosis desalination plant (1500 litres per hour) and for ice making (50 Kg per hour). The crushed ice is used by the fishermen for fish preservation and water for drinking.

The Pollar products[211] firm of Torrance, California, USA is manufacturing and marketing photovoltaic power refrigerators which use two separate compressors, and two independent refrigeration circuits. If one compressor goes wrong then the other circuit can maintain the temperature in both the freezer and refrigerators compartments. This photovoltaic refrigerator also uses three batteries (300 amprs. hour total capacity at 12 VDC).

The batteries in the photovoltaic generators can be eliminated since during the day time when sun is available, the ice can be formed which may maintain the temperature in the freezer required by the products to be stored. Thus it can be said that photovoltaic power operated refrigerators are quite reliable, cost competitive in rural areas of developing countries.

The use of boiling collectors where the working fluid of Rankine cycle is directly heated is proposed by Hedstrom[211] for use in vapour compression air conditioning systems. This system eliminates a heat exchanger, a pump and a storage tank required in the conventional solar air conditioning system and may outperform the conventional solar system by 28 percent depending on the type of collector used and location. Recently some studies are conducted by Ayyash et al[213,214,215] on the feasibility, energy balance and economics of cooling systems. Chinappa et al[216] have also made a comparison of solar absorption and vapour compression of two cascaded hybrid airconditioning systems: vapour absorption-vapour compression and vapour jet-vapour compression for three Australian locations. It has been shown that cascaded systems are economical only where cooling is required for all the time.

5.6 PASSIVE COOLING

Building can be cooled by natural means without the use of any mechanical or electrical power. Such systems where the thermal energy flow is by natural means i.e. by conduction, natural convection, and radiation are known as passive solar energy systems. The state of the art for passive cooling is much less developed than for passive space heating and it may not always be possible to use natural means to provide comfortable temperatures in buildings. As a matter of fact passive solar cooling techniques are not strictly 'Solar' but the performance of these techniques depend directly or indirectly on solar energy and requires no mechanical or electrical power. These natural cooling techniques at least can reduce to some extent the peak cooling load of a building, thereby reducing the size of airconditioning plant. Recently several review articles on passive cooling of buildings namely by Yellott[217], Givoni [218] , Balcomb[219], Joncich[220], Gupta[221], and Sodha and Bansal[222] are published, discussing various passive cooling concepts. The natural means which can be used for passive cooling are : shading, ventilation, evaporation, radiation cooling, ground coupling and dehumidification. Recently Carroll et al[223] have made a detailed technical assessment of the above passive cooling strategies. The Department of Energy (DOE), USA[224] has prepared a passive solar design handbook in three volumes discussing various passive solar heating and cooling design concepts. The passive cooling systems have many advantages such as : i) the system is oflow cost, and maintenance free compared to active system, ii) the system is simple and easily repairable and understood, iii) the operation is natural and aesthetically attractive, iv) the system remains operative and effective as and when required, and v) the system is more useful if incorporated at the design stage of the building but can be retrofitted also.

5.6.1 Shading

The most effective way of cooling a building is to shade the window, walls and roof of the building from direct solar radiation. Heavily insulated walls and roof need less shading than poorly insulated walls and roof. The maximum heat flow in a building takes place through windows. By using vertical, horizontal, and inclined louvres; shades; screens and by planting trees etc.; the entry of direct solar radiation through windows can be controlled to some extent. Windows on the east and west direction can be shaded only by using vertical louvres. The window on the south wall can easily be shaded by using properly sized horizontal louvres. The length of the louver depends on the place, and height of the window. The louver can be designed to admit direct radiation in winter and exclude radiation in summer (fig.5.45). The building facets can be shaded by using overhangs and awnings on the outside of the building. Reflective glass or sun strips are also effective means to eliminate the solar gain during summer months while visual contacts may remain maintained. Operable shading devices are even more versatile but if provided outside the building are difficult to maintain and may deteriorate with time.

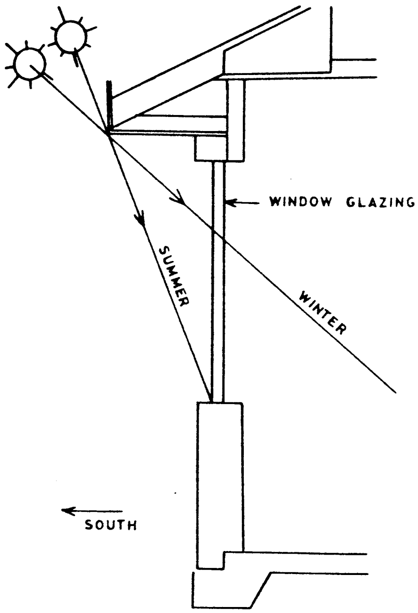


FIG.5.45 SHADING OF WINDOW USING OVERHANG

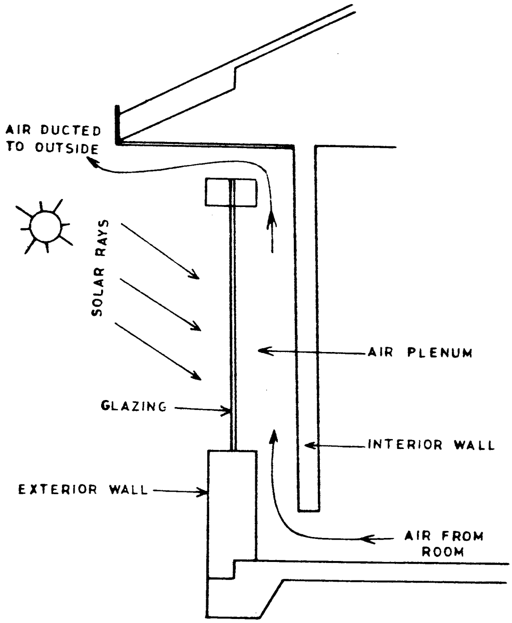


FIG.5.46 VENTILATION THROUGH A BUILDING

These external shades can be fixed type or adjustable type. The adjustable type is more versatile since one can control more precisely the amount of light and heat entering a room. Care should be taken that the external shade if used should not interfere with the night time radiative cooling. The roof of a building can effectively be shaded by using a removable canvas cover which can be used during the day time and rolled up during the night time to allow radiative cooling. Painting of the walls and roof with light colours will help in the reduction of heat gain. Whitewash which can cheaply be done on walls and roof is an effective way of reducing heat gain.

5.6.2 Ventilation

It is well known that movement of air across human skin at temperature below 37 °C provides a cooling sensation. If a new building is to be designed, then the speed and direction of the prevailing local winds should be considered for proper siting and orientation of the building. By properly positioning the windows and opening them air movement can be created in the rooms. The pattern of air flow in the building will depend on the air inlet (window) and outlet (window) location, their sizes and wind speed. At high altitude stations where the outside temperatures are quite moderate during the day and nights time, the Trombe wall concept can be used to draw room air through the bottom opening (airvent) as shown in figure 5.46. This 'Solar Chimney' effect if properly designed can be quite effective for cooling.

A wind tower (or wind catcher) as is used in Iranian Architecture can also be used to cool a building. The one end of the wind tower goes up to the basement of the building and the other end goes above the roof. The wind tower operates, according to the time of the day (day or night) and the presence or absence of the wind. During the day the walls of the tower absorb solar heat which heats the air at the top of the tower. Thus the air pressure at the top of the tower gets reduced and creates an upward draft. This results in an air movement in the building. During windy period the movement of the air is in the opposite direction down the chimney. On calm days, the walls of upper part of the tower are cooled (at night), cooling the outside air which sinks down into the tower creating down draft and cooling the space.

In desert regions where the sky is clear and night temperatures in summer are quite low, the coolness of night air can be stored either in the building structure (building mass) or in a separate rock bed (crushed rock) storage. During the night by using an electric fan or thermosyphon effect (natural convection) the outside cool air is passed either through the building or through the rock bed storing the coolness. During the following day the stored cold energy in the building or rock bed may keep the room cool.

5.6.3 Evaporation

Evaporation of water from the roof top of the building appears to be one of the most effective ways of cooling a building. Evaporative cooling can be achieved by using a roof pond, thin water film, flow of water, and spraying of water over the roof. A water film over the roof does not allow the roof to get hot and its evaporation,the rate of which depends on wind speed, solar radiation and roof temperature, lowers the temperature of the roof which in turns cools the space below it. This type of evaporative cooling is more effective in dry regions and for building where the roof is thin. Intermittent spraying of water over the roof is observed to be more effective than the roof pond.

Direct type of evaporative coolers (desert coolers) where the supply air is simultaneously cooled and humidified by bringing in contact with water are very popular in residential and industrial buildings in many developing countries. Direct evaporative cooling due to high humidity and excessive flow rate, is not desireable physiologically. The evaporation of 5.2 Kg of water per hour at 23.9°C produces 1 ton of refrigeration equivalent. Inside massive buildings, courtyard fountains provide cool spaces and are used in many countries. Indirect coolers, where evaporative cooled air exchanges heat with the supply air through heat exchanger are more effective and does not increase the humidity of air.

5.6.4 Radiation cooling

Cooling occurs when a body (building) at a higher temperature radiates energy to the body (sky) at lower temperatures. The amount of cooling by radiation depends on many parameters like the temperature of the body, its emittance and orientation; the sky temperature and the pressence of other materials like trees, building etc. between the building and sky. A review of cooling of building by nocturnal radiation is done by Givoni[218] . A horizontal surface like roof of a building which is exposed to the sky is the most effective long wave radiator. There are three methods of cooling of building by nocturnal radiation : the roof pond with moveable insulation (the skytherm system) by Hay and Yellot[225] , the 'cool pool' system with hinged insulation by 'living' Systems'[226], and 'roof radiation trap' by Givoni[227] .

In the 'Skytherm' system, black plastic water bags kept

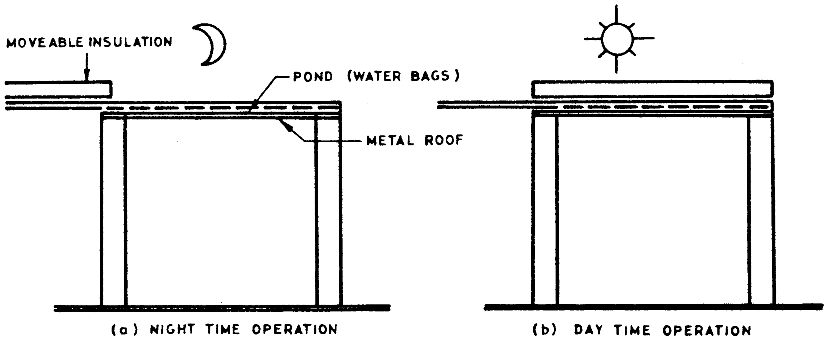


FIG.5.47 SKYTHERM RADIATION COOLING SYSTEM

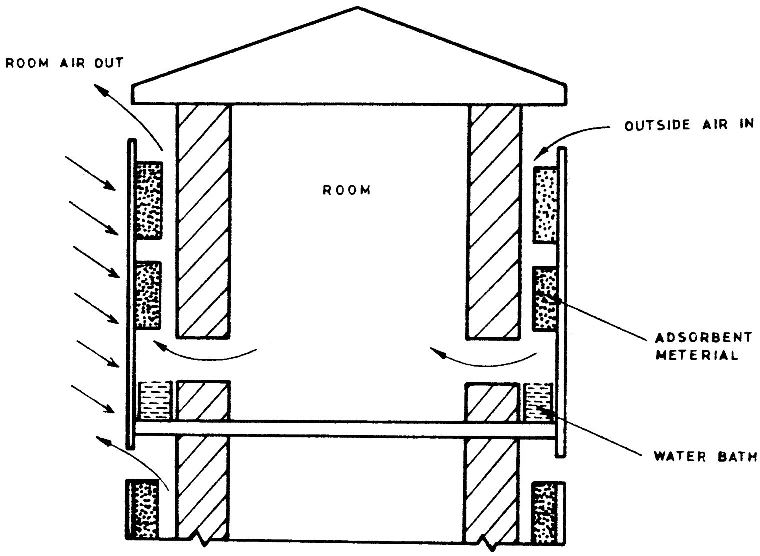


FIG 5.48 A SOLAR COOLED HOUSE BY DEHUMIDIFICATION AND EVAPORATIVE COOLING

over a metal roof are exposed to the sky at night (fig.5.47a). The water gets cooled by Nocturnal radiation. The cooled water cools the house by absorbing heat from the roof which comes from the space below the roof. Additional cooling by evaporation if required can be provided by flooding the bags with water. During the day time the pond is shaded by moveable insulation panel (fig.5.47b). Experiments on such Skytherm systems have been conducted in many countries. The concept is found very effective in full size house experiment in California where the out door air temperature varied from 31 °C to 12 °C and the temperature of water in the bag varied from a18 to 20 °C. The room air temperature remained practically constant throughout the day at about 21 °C, when a 5 cm thick polyurethane insulating panel was used to cover the ponds during day time.

Hammond[226] of 'living system' in California developed a roof pond system for heating and cooling of buildings using a water layer over the roof and openable insulation cover. In this system water is contained in shallow galvanized iron pans with asphalt coating and covered by transparent plastic sheet. The insulating panels are hinged having a reflective layer on its bottom side. These panels are closed and opened by a hydraulic ram. In summer, the pond is exposed to sky during night and covered by insulation during the day and cooling is done as in the Skytherm system. In winter during the day time the reflected radiation falls on to the water resulting in enhancement of solar radiation.

In the 'roof radiation trap' system as developed by Givoni[227] the insulation layer is covered by a corrugated metal sheet which is painted white on the external side. During the day time the system remains as such. During night the roof gets cooled due to nocturnal radiation, cooling the air below. Through these corrugations of the metal roof air is sucked with the help of a fan and this cooled air can be supplied either directly to the room for cooling or can be stored in a rock bed system.

5.6.5 Ground Coupling

In summers when cooling is required, the ground temperature is always lower than the air temperature. This lower temperature of the ground can be used for cooling a building by partially sinking it into the ground. As we go deeper and deeper in the earth more comfortable conditions can be obtained in the room but construction would be more expensive. Earth bermed structures where the walls are earth bermed as high as possible is another way of maintaining constant temperature in the building. Earth air tunnels can also be used for heating and cooling of buildings. Air can be drawn through these earth tunnels and the cold air supplied directly to cool the room. These systems generally have problems with condensation.

5.6.6 Dehumidification

Active methods of cooling by dehumidifying the air is described earlier. Here the moisture content of the air in the room is reduced, thereby increasing the allowable maximum dry bulb temperature while still remaining in the comfort zone. An idea for passive cooling using solid absorbent materials in the walls of a room was giving by Altenkirch[229]. This idea was discussed by Dannies[230] in details. In this idea as shown schematically in fig.5.48,the building orientation is such that in the morning when the east wall is heated by solar radiation the incoming air from the west wall side first gets dried by the solid adsorbent material and then evaporatively cooled by passing over the water baths and flows into the room. The hot air going out of the room regenerates the solid adsorbent material on the east wall side. In the evening, when the west wall is heated by sun, reversal of the air flow occurs. The system appears to be attractive but depends on solar radiation and relative humidity and puts a restriction on the building design.

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CHAPTER - 6

PASSIVE SOLAR HOUSE HEATING

6.1 INTRODUCTION

For heating the space using solar energy, the solar energy is to be collected, stored, and distributed properly in the space to be heated. In an active solar space heating system, the solar energy is collected using some kind of separate solar energy collectors; solar energy stored in a separate storage unit may be in sensible heat storage materials, latent heat storage materials, or in chemical reactions; and the energy is distributed in the space using electrically operated pumps and fans using radiators etc. While in a passive solar heating system all the three functions of solar energy collection, storage and distribution is done by natural means and generally no electrical or mechanical power and electronic controls are used. In the passive house heating system, various elements of the buildings like walls, roof, windows, partitions, etc. are so selected and so architecturally integrated that they participate in the collection, storage, transport and distribution of thermal energy. Thus in a passive solar house heating system the building becomes an 'energy machine' and its structural and architectural details become integral parts of the components of the energy system.

In passive solar heating and cooling of house, the natural on-site energy sources (for heating) such as solar radiation, outside air and internal metabolism are used while sinks (for cooling) used are sky and space, outside air, and wet surfaces. Both direct and diffuse solar radiation is used for heating the house. When the outside air is at more than 24 $^{\circ}$C it is used as a heat source other-wise it is used as a heat sink. All the appliances, light, cooking, motors, people, etc. in the building also add heat to the interior of the building and thus are the heat source. All the wet surfaces behave as heat sinks because when the water evaporates these wet surfaces absorb heat. Both the building elements such as construction materials like stone, bricks, concrete, water, insulation, glazing, shading, reflectors, etc.; and various thermal processes such as thermal radiation, natural and forced convection, conduction, air stratification, evaporation, thermosiphoning etc. are combined in various ways in passive designs. Depending on the particular need which also depends on the site, climate, and on activity, the designer or architect if aware of different building elements and thermal processes can take decisions for each particular design project.

The building materials like brick, stone, concrete, phase change materials, water, etc. form the building envelope but if these materials are heavier (thick walls, roofs, etc.) then these materials may store sufficient thermal energy and may help in time delay and lowering the heat wave amplitude. During the daytime, the excess heat may be stored in these materials, which may be released during night time when required. Insulation which shows high resistance to heat flow is another important component used in passive designs. In cold climates it reduces the conductive heat loss from the building to the outside while in hot climates it reduces the heat gain. Sometimes insulation is used over the windows during night time (movable insulation) which also helps in reducing the heat loss through windows. Glazing or window is an integral part of passive house and therefore its design, orientation, position and area is extremely important. It not only traps the solar radiation acting as solar collector but also is a source of lights indoors. Sometimes reflectors are employed to enhance the solar radiation falling on the window and sometimes these are used to shade the building during the summer. In hot climates or in certain times of the year the shading of part of walls, roof, and windows help in keeping the building cool. There are several ways of shading a part of the building such as by vegetation, trees, overhangs, shutters, balcony, parapet walls, etc.

Heat transfer by conduction is an important aspect in heating and cooling of buildings by passive means. In this process heat is transferred due to temperature difference from warmest place to the coolest place. The heat transfer by convection helps in distributing the heat from the source at high temperature generally at a lower level to the cooler places generally at an higher elevation. In heat transfer by convection also, a temperature difference is required and transferred by air movement(in buildings) from hot spot to cold spot. Heat transfer by radiation exchange between building faces is another important energy transfer mechanism. People sitting in a room will exchange thermal radiation with its various surfaces like walls, roof, floor, and other surfaces. The radiation exchange between a human body and surfaces of a room is as important as room air temperature for providing the comfort. The evaporation of water, the rate of which depends on vapour pressure differential between its surface and surrounding air, is made use for cooling the space and therefore is an important mechanism used in passive cooling of building. In evaporating of water, heat is required which is supplied from the contact surface and therefore a cooling effect is experienced. The property of the air that when it is heated, its density decreases and becomes lighter and more buoyant than cooler air, is also made use of cooling and heating of building. In summer, the hot air from the room is vented inducing the fresh cool air from outside to the room resulting in cooling effect. However, this air stratification is undesirable in the winter. Sometimes the thermosyphon effect i.e. in a closed loop when a fluid is heated it becomes lighter and moves up and the cold fluid replaces the heated fluid is also used for storing the heat and also for distributing the heat in the rooms. Probably the least recognised and understood is the contribution of sky radiation in the passive cooling of buildings. The average sky temperature is always lower(10 °C to 25 °C depending on moisture in air) than the outside air temperature and therefore the building facets always radiate heat to the sky.

From the above it is clear that practically all the buildings collect some amount of the solar radiation for heating to some degree in winter and reject the heat to some extent in summer, but when the building structure is designed in such a way that maximum amount of solar radiation is collected and used which is a significant proportion of heating load of the building and similarly if a building structure is designed in such a way that the heat is rejected to an extent that the cooling produced is a significant proportion of cooling load then one would use the term passive solar building in practice. Understanding of the behaviours and properties of different building elements and thermal processes will help the designer to arrange them in an appropriate fashion such that they behave according to the cooling and heating requirements of a building with the sun as a major heat source and night sky as a major heat sink.

6.2 HISTORY

Heating of residences using solar energy is as old a practice as building habitations. Aristotle advised that houses should have large fenestrations on walls facing south and as small as possible fenestrations on the north side, with horizontal shading devices above the south facing windows to allow the winter sun to shine indoors and cut the direct radiation to enter the room in summer season. There are several examples of ancient architecture, where thermal comfort in the buildings is provided by shaping, sizing, and designing the building. Sometimes the advantage of local vegetation, trees, water circulation, site, orientation is also taken. Socrates (Circa 400 B.C.) also described a technique for passive heating of buildings by providing large fenestration with an horizontal louver on the south side wall and little fenestration on the northside wall. One of the most significant example of passive solar heating is the Indian cliff dwelling popularly known as 'Montezuma's Castle' which is about 128 Km. north of Phoenix, Arizona. This building which is of masonary structure, tucked into the side of a south facing white limestone cliff, is protected with summer sun with the help of overhangs. In winter season, the south side walls of the apartment house are heated by direct sunlight and this absorbed heat is released to the inside during night time. During daytime, the sunlight which directly enters through small windows and the residual heat in the walls was enough to provide comfortable conditions. The elevation of this Montezuma's Castle is about 1220 m above sea level and hence the natural ventilation in summer season was adequate for comfortable conditions. This building was started in AD 700 and completed in AD 1300. It appears that during the Mohenjo-Daro civilizations also the same principles for heating and cooling the buildings by natural energies were used. Ancient Iranian architecture used the concepts like clustering of buildings, use of thick walls, tree and shrub plantations, rooms, basement, wind tower, evaporative cooling, etc. for providing cooling of buildings in summer. Passive solar techniques are used in the construction of dwellings in Chaco Canyon and Mesa Verde by American Indians as long as AD 1100. The old Indian forts, domes and historical buildings are very good examples of ancient Indian architecture where several concepts like flow of water near the buildings; water fountains; use of thick walls made of stone, mud, limestone, etc; large ceiling height; overhangs; low fenestration area on south and west side; planting trees; tower in the centre of the building; clustering of houses; etc., are used for keeping the building warm in winter and cool in summer. The early work on passive heating and cooling of buildings was not based on scientific experiments but on common sense and experience and therefore their thermal performances were far from satisfactory and even such examples are rare and isolated.

The first scientific work on solar heating began in 1881 when Professor E.L. Morse in Salem[1], Massachusetts was granted a patent who actually built a dark coloured wall covered by glass sheet with both top and bottom vents and dampers for heating a room. A patent on thermosiphoning air panel was taken by E.M. Morris[2] in 1898. Both the ideas of Morse and Morris were neither evaluated for their performances nor quantitatively assessed for more than 60 years. During the year 1947, a thermal wall similar to the one proposed by Morse was used by Hollingsworth[3] in the MIT Experimental House. During the years 1972-1976 several homes at Odeillo in Pyrenees, France was built by Felix

Trombe[4,5,6] of Odeillo, France alongwith architect Jacques Michel using thermal storage walls based on the Morse's concept and which was later known as 'Tromble Wall'. In 1947 Simon[7] published a book entitled, 'Your Solar House' which contained designs of several direct gain type solar houses, one suitable for every state of USA and which were prepared by famous architects. During the years 1952 George and William Keck[8] and other architects designed several solar houses in USA in which large southern 'Picture Windows' and glass walls admitted large quantities of solar radiation into the building interior in winter while in summer the horizontal overhangs were shading the entire south wall. New interest in passive solar heating again arose in the sixties when three remarkable buildings one Wallasey School built by Morgan[9] in England, second the Trombe House built by Trombe[10] in France, and the third Atascadero House built by Hay[11,12] in California, USA each with different approach were built. Each of the above three houses were analysed and evaluated by a independent team of scientists. The Wallasey School was analysed by M.G. Davies and A.D.M. Davies[13]. The Trombe House in which J.F. Robert lives was analysed by Trombe, Robert, Cabanet, and Sesolis[14]. The Hay's House in Atascadero, California, USA was analysed and evaluated by Haggard et al[15]. The first National Passive Solar Heating and Cooling Conference[16] and Workshop was arranged by the scientists of Los Almos Scientific Laboratory at Albuquerque, New Mexico during May 18-19, 1976 where the work carried out so far in this field was reviewed and was attended by many scientists. This conference was the real turning point in the field of Passive solar heating and cooling and this topic became so important that almost every year[17-20] a conference was arranged in USA. Recently several reviews[21-28] on Passive Solar Heating and Cooling are published dealing with different concepts, modelling, designs, etc. Some comprehensive reports on Passive Solar Heating are also prepared[29-33] which not only discuss the various passive concepts and designs but also the simulation models and simplified performance prediction models. Looking to the importance and application of the subject since many thousands of Solar Passive Houses are made in different parts of the world, books dealing only with the Passive Solar Heating and Cooling are recently published[34-39].

6.3 TYPES OF PASSIVE SYSTEMS

There are two schemes for classifying passive solar systems. In one of the schemes based on functional or generic classification, the passive solar buildings are classified as: (i) direct gain, (ii) indirect gain, and

(iii) isolated. In the second classification which is widely used and is based on physical classification, the passive solar buildings are classified as: (i) direct gain, (ii) thermal storage wall, (iii) attached sunspace, (iv) thermal storage roof, and (v) convective loop. All the passive solar buildings fall in the above five categories. Each of the above main classes can be further sub-divided into many different specific system depending on the use. These main classes are briefly described below:

6.3.1 Direct gain

The simplest passive solar heating concept is direct gain system as shown in fig.6.1 where following concepts are employed.

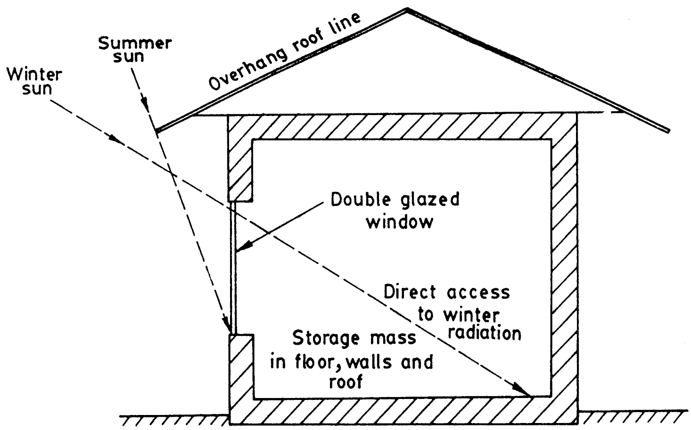


FIG.6.1 DIRECT GAIN PASSIVE SOLAR HEATING SYSTEM

(i) A double glazed window facing south or the entire south facing wall is double glazed through which direct radiation in winter enters and strikes the floor, walls, or other objects in the room. Almost all the solar radiation entering the room is converted into the heat. The heat loss from the room is reduced by using a double glazed window.

(ii) An appropriate overhang above the windows or at the roof level where south wall is glazed, which shades the window or the wall during summer when the elevation of the sun is high

(iii) The floor and/or walls are made of massive construction to increase the thermal mass of the construction which helps in storing the heat during daytime when sufficient heat is available and releasing the same during night time, thereby reducing the large oscillations in the room air temperature.

(iv) If some kind of thermal insulation is used to cover the windows during periods (night) when heat loss is more than heat gain through windows, then the performance can be further improved,

6.3.2 Thermal Storage Wall

Inspite of heavy thermal mass provided in the direct type passive heated rooms, there is a large oscillations in the room air temperature. A more effective way of heating the rooms and reducing the large oscillations in the room air temperature is the use of a thermal storage wall between the double glazing(facing south) and the room. In this category following concepts are employed.

(i) The entire south facing wall is covered by one or two sheets of glass or plastic with some air gap between the wall and the inner glazing. In this air gap hot air moves from bottom to top generally due to natural convection.

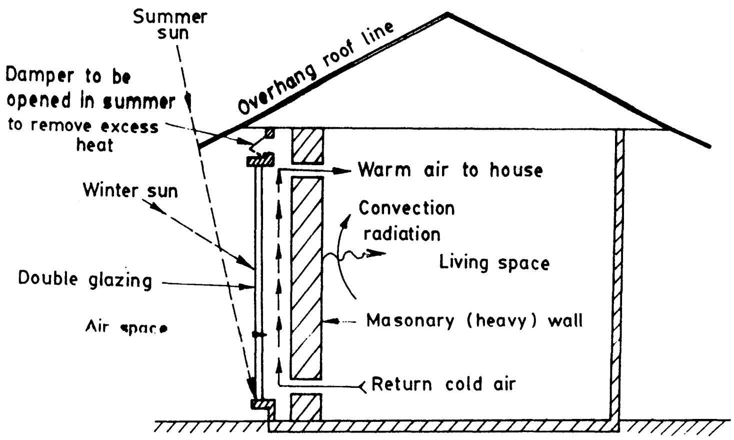


FIG.6.2 TROMBE WALL OR THERMAL MASS WALL

(ii) A south facing thermal storage wall made of masonary or concrete with the outer side facing the sun blackened (fig.6.2). The solar radiation after penetration through the glazing is absorbed by the wall and therefore the wall gets heated. Thus the air in the gap between glazing and wall gets heated, rises and goes in the room through the upper vent while the cool air from the room through the bottom vent enters in this gap. This circulation continues till the wall goes on heating the air. The flow of heat in the room can be changed by adjusting the air flow through dampers provided at the inlet and outlet vents. The room is also heated by convection and radiation heat loss from the inner surface of the wall facing the room. Thus this thermal storage wall collects, stores, and transfers the heat to the room. This wall is known as Trombe wall.

(iii) In some cases the thermal storage wall instead of masonary or concrete wall is made up of drums or barrels or other suitable containers full of water stacked over each other which collects, stores, and distributes the heat and is termed as water wall or drum wall (fig.6.3). The faces of the drums facing the glazing are painted black to increase the heat absorption. The air temperature swing in the room in case of water wall is much less compared to thermal storage wall(Trombe wall).

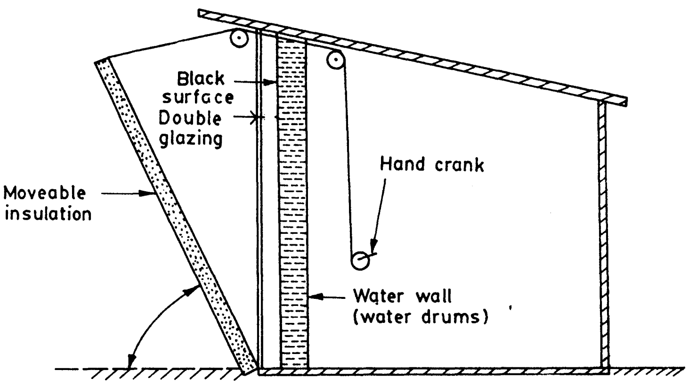


FIG.6.3 WATER WALL (DRUM WALL) TYPE PASSIVE SOLAR HEATING SYSTEM

(iv) An air vent in the bottom of the trombe wall and top of the trombe wall opening in the room as well as another air vent at the top of the trombe wall opening outside the room are important. The shutters on the inlet and outlet vents are also used which control the air flow. When heating is not required, the excess heat goes out from the air gap through the vent opening outside the room.

(v) An insulating shutter with a reflector lining on its upper face (Fig.6.3) is also sometimes used in the trombe wall and water wall systems. This insulating shutter when opened during the daytime when sun is shining, reflects a part of solar radiation and thus enhance the solar radiation input in the wall. During off sunshine hours, the insulating shutter is raised with the help of either hand crank or automatic device photocell device and which reduces the heat loss from the storage wall to the outside.

6.3.3 Attached greenhouse (Sun Space)

In this passive solar heating approach the concepts of direct gain and indirect gain (mass thermal wall) are combined as shown in fig.6.4. The main components of the concept are as follows:

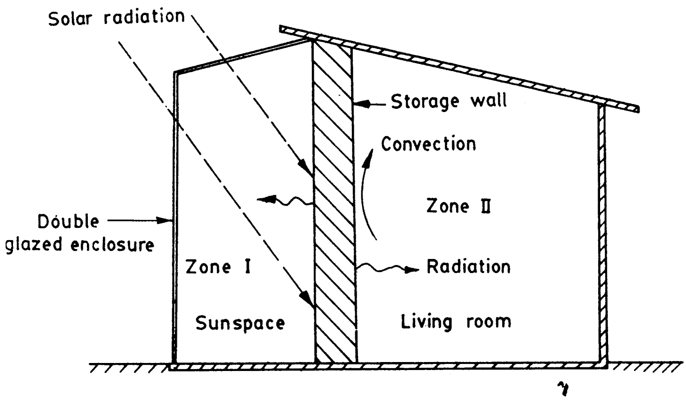


FIG.6.4 ATTACHED GREENHOUSE (SUNSPACE) APPROACH TO PASSIVE SOLAR HEATING

(i) There is sunspace (zone I) on the extreme south facing side of the house covered with single or double layers of glass sheets or plastic sheets which becomes a green house and can be used either for raising vegetables or flowers and also can be used as a sunny space for living. In this attached greenhouse there is a large air temperature swing. In summers, the air temperature in the greenhouse may be very high and therefore large vents for air circulation may be used.

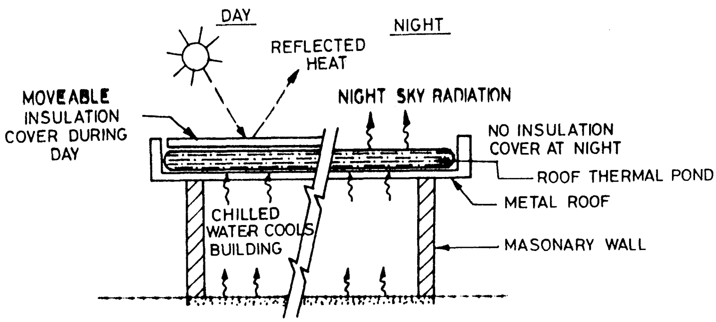
(ii) There is a thermal storage wall facing south in between the room (living space, zone-2) and greenhouse (sunspace, zone-I). The thermal storage wall gets heated by direct absorption of solar radiation coming through greenhouse transparent cover. The living room gets heated through convection and radiation heat loss from the thermal wall. The heat loss from the thermal wall to the outside in this case is low because of greenhouse and sometimes by using a moveable insulation over the walls of the sunspace.

6.3.4 Thermal storage roof

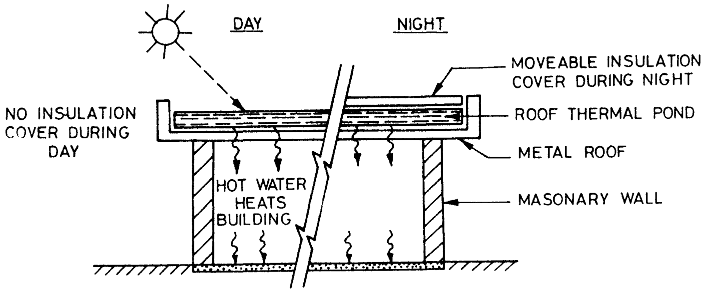
The thermal storage roof concept for passive solar heating was developed by Hay and Yellot in Arizona[12] and is similar to the thermal storage wall except that the interposed thermal storage mass is on the building roof instead of a wall. This concept uses the following components:

(i) A metal roof which conducts the heat effectively.

(ii) Water bags made of transparent or black plastic sheet and filled with water or any other massive materials and are put over the metal roof. In



(a) Cooling During Summer



(b) Heating During Winter

FIG.6.5 THERMAL STORAGE ROOF (ROOF POND) APPROACH FOR PASSIVE COOLING AND HEATING OF BUILDING

winter when room heating is required during daytime when sun is shining, the water in the bags gets heated, store the heat which further heats the room below during daytime as well as during night time when sun is not shining (fig.6.5b).

(iii) Movable insulating shutters are used over the water bags. In winter, during off sunshine hours these insulating shutters are slided over the bags (fig.6.5b), reducing heat loss from the water bags to the outside. During daytime when sun is shining the insulating shutters are pulled back allowing the solar radiation to fall on the water bags. Thus,the water begs keep the room warm in winter during night and day.

(iv) The same thermal storage roof system can be used for cooling the room in summer as shown in fig. 6.5a by simply reversing the process. In this case the insulating shutters are pulled back during night time allowing the cooling of water bags and roof by thermal radiation heat loss to the outside resulting in cooling the room. During daytime, the shutters are slided over the water bags avoiding the heating of water bags from direct sunshine. The cool water mass keeps the room below cool during day and at night.

6.3.5 Convective loop

The convective loop approach for passive solar heating is similar to the active heating approach in the sense that there is a separate collector and storage system but no mechanical or electric pump for the circulation of fluid and no electronic control is used. In the convective loop approach (fig.6.6) the components are as follows:

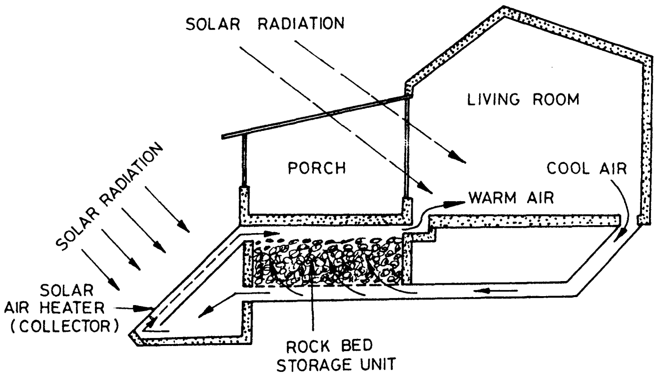


FIG.6.6 CONVECTIVE LOOP APPROACH TO SOLAR PASSIVE HEATING

(i) A liquid or air solar heating flat-plate collector is used either for heating the liquid generally water or air. Simple inexpensive collectors are preferred like solar air heaters.

(ii) A pebble bed storage system of sufficient size kept in the level of the room but above the level of the solar air heaters. Air after getting heated in the solar air heaters goes either directly into the living room to be heated or through the rock bed storage unit heating the storage unit. The cool air from the room or from the bottom of the storage unit enters through the bottom of the solar air heater automatically(natural convection) and again heated, rises, and enters into the room. When room heating is not required, the floor vent can be closed and in this case the storage bed will be charged by the solar heated air.

(iii) When water heating collectors are used, then water heated in the collectors circulates automatically through the pipes imbedded in or under the floor.

In any of the building, several of the passive solar heating systems can be combined. Moreover, these can also be combined with the different natural cooling systems such as

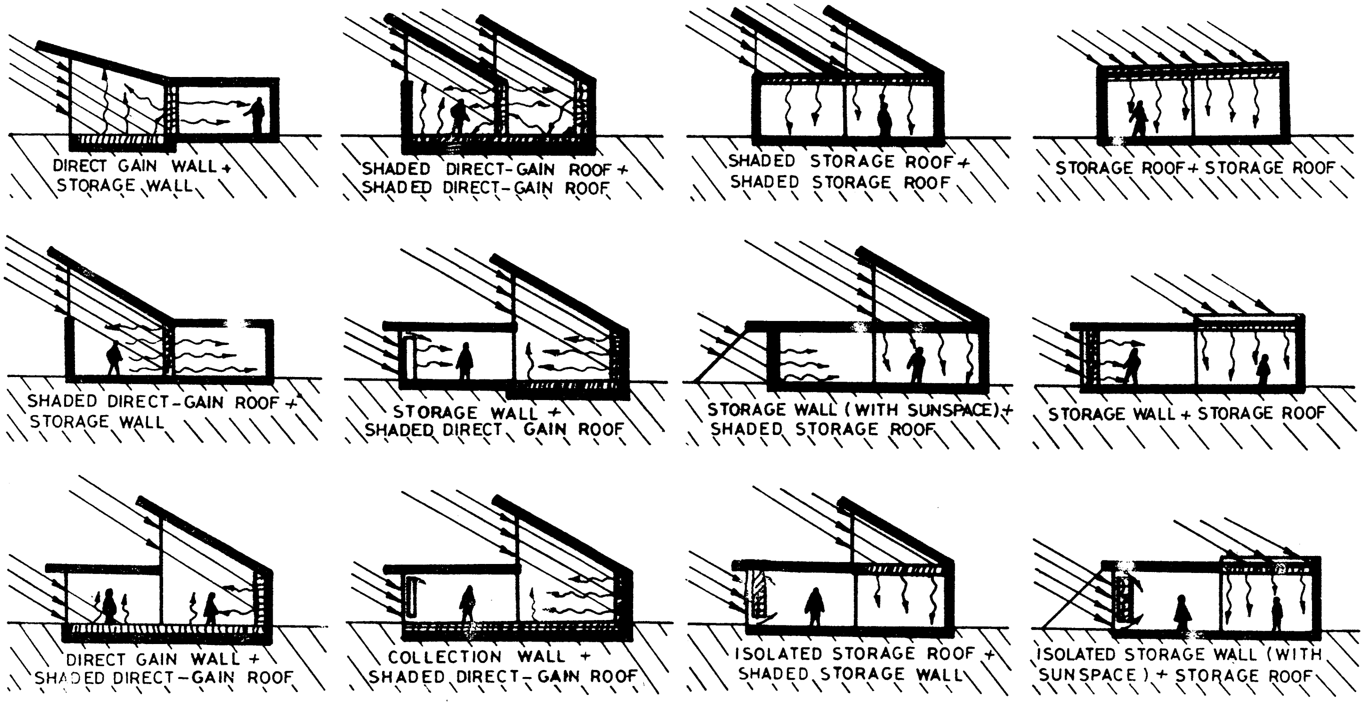
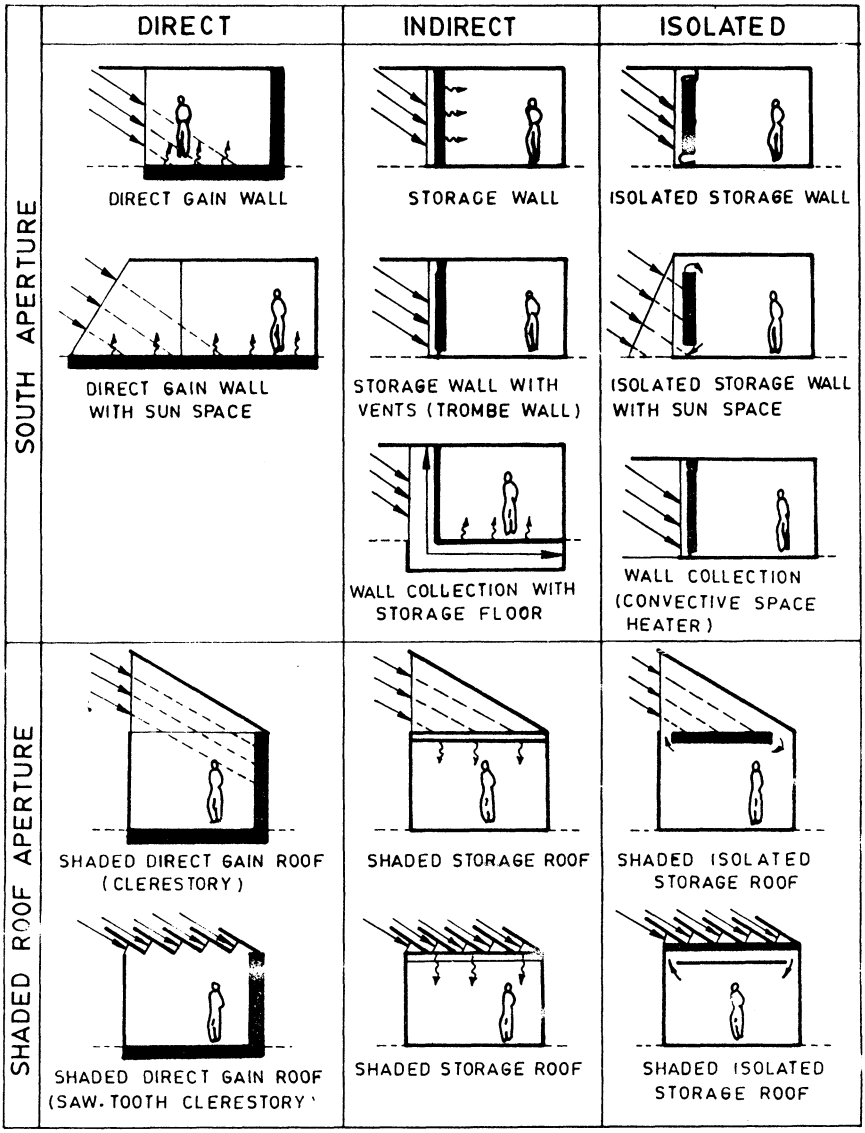
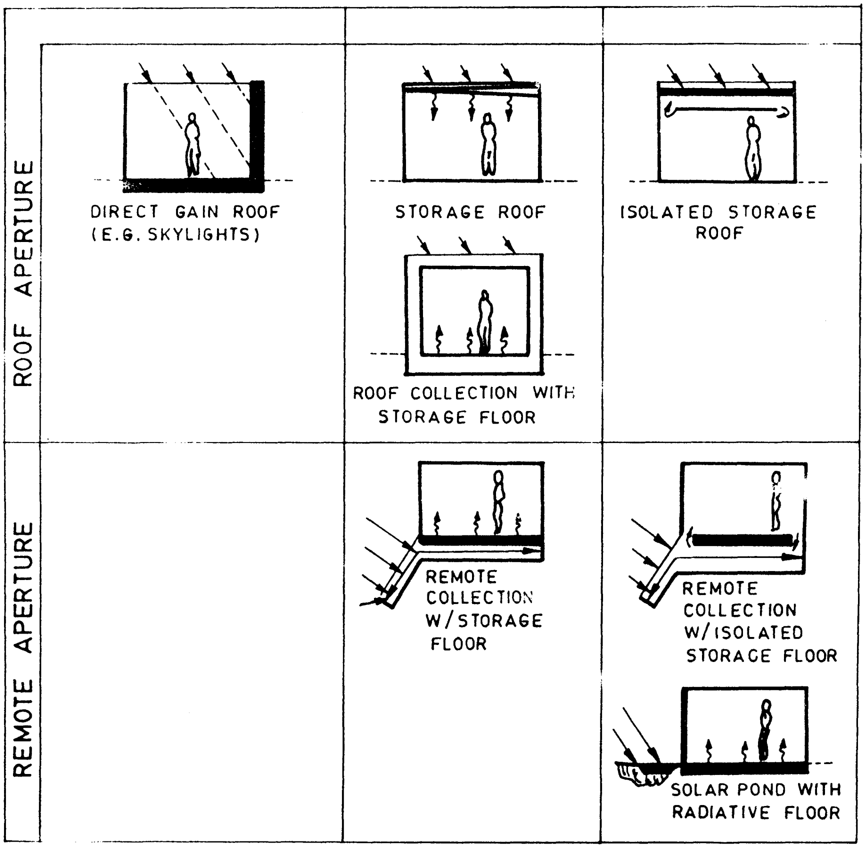


FIG.6.7. SOME INTEGRATED PASSIVE FEATURES (From DOE[40])



(a)

evaporative cooling, noctural longwave radiant cooling, etc. It is possible that by incorporating many passive features in a building (integrated-passive systems), comfortable temperatures can be achieved. A few examples of combining various passive features are shown[40] in fig.6.7. As discussed earlier there are several passive solar heating features and by making small variations many more can be made but mainly these are grouped as direct gain, indirect gain, and isolated type. Variations in all the three types are shown[40] in fig.6.8(a & b). Out of these three systems, the direct gain system is thermally most efficient but in this case there is a large fluctuation in room air temperature. The room air temperature is more controllable in case of isolated systems, but these are thermally less efficient.



(b)

FIG.6.8 VARIATION OF PASSIVE SOLAR HEATING FEATURE (From DOE[40])

In certain climates, it is not possible to provide a high fraction of heating requirements of a building by passive systems alone may be due to large requirement of storage of heat or large requirement of window area. If in such cases the active heating systems where energy can be stored more efficiently is also combined with the passive systems, then the temperature will be controllable, storage capacity will be efficient, active system size will be reduced, and hence the cost will also be lowered. Such a combined heating system is known as hybrid' system.

6.4 TYPICAL PASSIVE HEATING BUILDINGS

There are several thousand passive solar heated buildings in different countries of the world representing different climatic zones like cold, tropical, arid, hot, and humid. Most of these buildings utilize integrated passive heating approach where two or three passive heating approach are combined and therefore the final result is the combination of these approaches. But there are some typical installations where only one single approach of passive heating is used and the house is experimentally studied to see the effect of the concept. Here only some special characteristics of a few typical installations for each of the passive approach are discussed.

6.4.1 Direct gain installation.

Several Passive Solar Heated Houses are described by Shurcliff[41] and in the Proceedings of the National Passive Solar Conferences[16-20]. Perhaps the most impressive direct gain passive heating building is at Santa Fe County[36°N], New Mexico, USA which is at an altitude of 2028 m and is described in a Sandia Publication[29]. This is a single storey simple direct gain house illustrated[29] in figs. 6.9 and 6.10 with rectangular shape and long axis oriented east-west. In this house as shown in fig. 6.10, two bed rooms and a full bath is located on the west end. A kitchen, pantry, and a utility room is located on the east end. One garage is located on the north end with access through the pantry. The floor area of the house is 117.5m^2. There are five clearstory windows each 1.17m high and 1.93m wide facing south inclined at 20° from vertical admitting solar radiation which falls on the north adobe wall of the house. There are seven more windows each of 1.7m high and 1.93 m wide in the south wall of the house. The solar radiation entering these windows falls on the floor and furnishings of the house. The total area of the window is about 28.8 m^2. There are no overhangs on the windows to cut the direct solar radiation in summer but projecting roof beams protect the window in summer. There are seven small (25 cm x 71 cm) openable windows below the seven big windows in the south wall of the house used for ventilation.

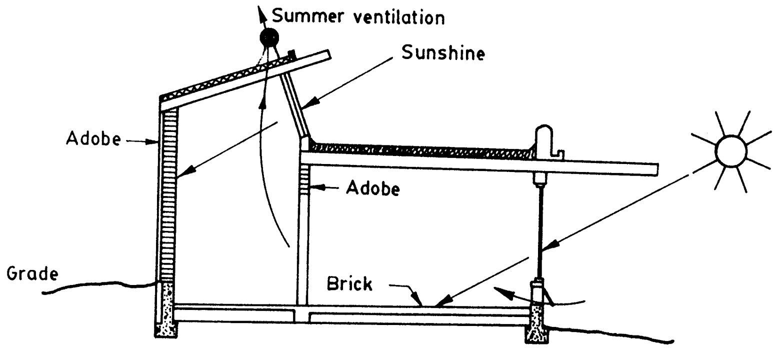


FIG.6.9 CROSS-SECTION OF WILLIAMSON HOUSE(From SANDIA[29])

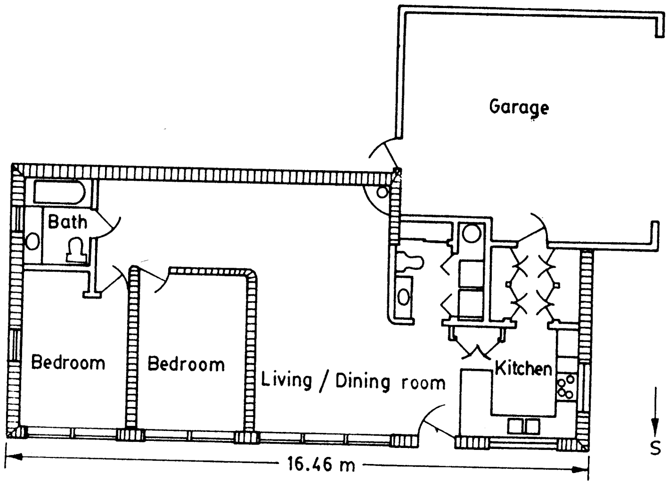


FIG.6.10 FLOOR PLAN OF WILLIAMSON HOUSE (From SANDIA[29])

All the exterior walls of the house are 25 cm thick adobe with 5 cm of polystyrene. The north end wall is additionally insulated with 2.5 cm urethane foam. The north wall is bermed to a depth of about 75 cm. The exterior finish of all the walls is stucco while the interior finish is brown in colour. The bedroom walls are also made of adobe. The roof is a wooden beam structure with 2.5 cm thick lumber, 10 cm polystyrene covered with celotex and sealed with asphalt coating. The floor is made of red brick laid in sand. From the above it appears that the house has sufficient of direct gain window aperture and thermal storage mass (adobe walls and brick floor) which helps in absorbing, storing, and distributing the heat by the convection and radiation to the living space. Auxiliary heat to the house is provided by electric baseboard heating units and a fireplace which is located on the north wall. Drain down type solar flat collectors are used to provide heated water at about 60 °C for domestic use.

The design temperature for the house is -17.7 °C and the heating load is 3388 °C-day. The solar insolation on the horizontal surface on a typical day in the month of January is 12.38 MJ/m^2day. Because of large thermal mass in the house, the air temperature fluctuation in the house is moderate and this helps in keeping the living space warm at off sunshine hours. Due to large window area, sometimes overheating is experienced. The excess heat is flushed out by opening the small windows provided below the main windows on the south wall and also by opening the turbine vents provided in the roof.

Temperatures at various points in the house were recorded from December 26, 1978 to January 8, 1979 which includes few sunny days, cloudy days, and very cold nights. The various data recorded is shown[29] in fig. 6.11. From this figure it is seen that outside air temperature varies from -17 °C to 4 °C. Instead of measuring the air temperature in the living room, globe temperature was measured which is a combined effect of room air temperature and the radiation exchange between the black copper globe and the various room surfaces. From fig. 6.11, it is seen that the globe temperature varies from 18 °C to 32 °C on various testing days. A thermocouple embeded 5.0 cm deep in the north wall was used to measure its temperature and its variation is also shown in fig. 6.11. Since north wall also receives direct solar radiation entering through the clearstory windows, it shows dirunal variation during sunny day and almost no variation on cloudy days. The floor temperature was also measured at a depth of 1.25 cm, 5.0 cm, and 15.0 cm at a point 2.25 m away from the south wall at a point where direct solar radiation is never reaching. Diurnal variation in floor temperatures at a depth of 1.25 cm and 5.0 cm is observed but at a depth of 15.0 cm this variation is completely damped out. The floor temperature at various points is also measured and the same is shown[29] in fig.6.12.

It is seen that the floor temperature near the south window which is exposed to solar radiation shows large diurnal variation with a maximum temperature reaching to 37 °C. During the daytime the floor temperature rises very

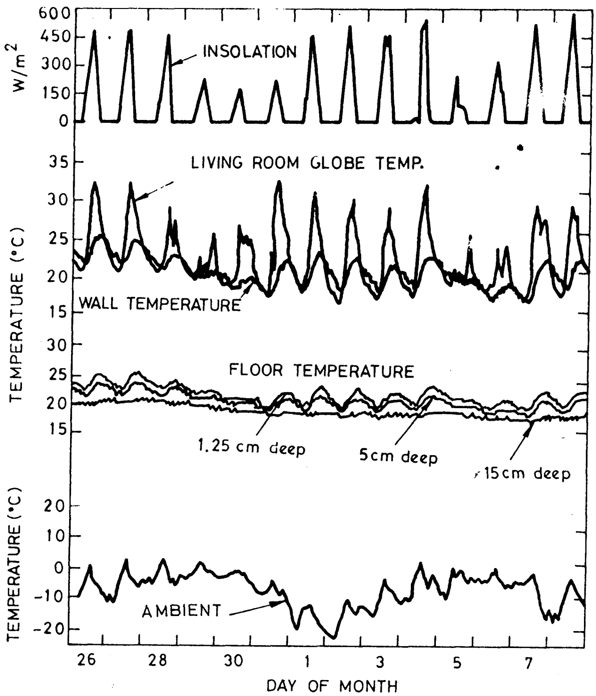


FIG.6.11 EXPERIMENTAL RECORDED DATA FOR WILLIAMSON HOUSE (Dec.-Jan.1978-79) (From SANDIA[29])

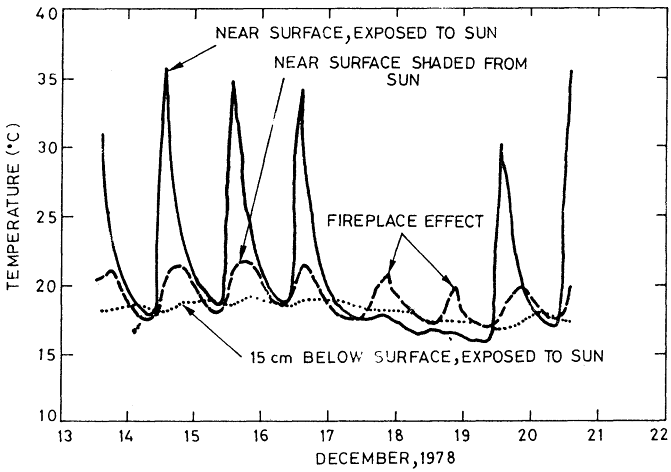


FIG.6.12 FLOOR TEMPERATURE IN THE WILLIAMSON HOUSE (From SANDIA[29]

fast due to direct absorption of solar radiation while during the night time it gives out heat to the room till it becomes equal to the room air temperature. December 17-19 were cloudy days and hence there is no diurnal variation in temperature. The diurnal variation of floor temperature at a point away from the south wall is also shown in fig. 6.12. Here the diurnal variation is small because it does not receive direct solar radiation. But here also the floor temperature varies from 18 °C to 23 °C. On December 17-19 which are cloudy days, a diurnal variation in temperature is observed at this point which is due to heat from the fire-place which was in operation during evening hours on these days. No fluctuation in temperature is observed at a depth 15 cm below the floor due to large thermal mass.

In summers, comfortable temperatures in the house were observed. Sometimes overheating was experienced and the excess heat was exhausted through turbine vents. In winter mornings, sometimes auxiliary heating system was required.

6.4.2 Thermal Storage Wall (Trombe Wall) Installation. (From SANDIA[29])

An impressive Trombe wall house with a small green house attached to it is in Princeton (40° 21'N), New Jersey at an elevation of 30.5 m and is owned by Doug and Meg Kelbaugh and therefore known as 'Kelbaugh House'. This is a small 172 m² two-storey house with no attic or wooden frame construction. On south side there is a trombe wall made of concrete of 38 cm thick, the outer side of trombe wall (facing outside) is coated with selective black coating. There are windows in the trombe wall which are used to see outside things. Figs. 6.13 shows the cross-section of this house showing the movement of heat by radiation and convection from trombe wall in the two storey house [29,42]. The trombe wall on the outer side is covered with double glazing. The spacing between the inner glazing and the trombe wall is about 15 cm. The total glazing area on the south side is about 57 m². The eastern half of the first floor, is a lean-to greenhouse with a total glass area including end walls of about 29 m².

The floor plans (I floor and II floor) are shown in fig.6.14. On the ground floor there is one large continuous room with the kitchen on the east side and partially separated from the living and dining area by the staircase. All the three bed rooms are on the second floor towards the concrete wall (trombe Wall). On the east of the trombe wall there is a small double glazed green house which lies above the basement which contains auxiliary furnace and hot water heater. All the walls, east, west and north walls are well insulated. All the windows are small and triple glazed. The two windows on the north side have 2.5 cm thick styro

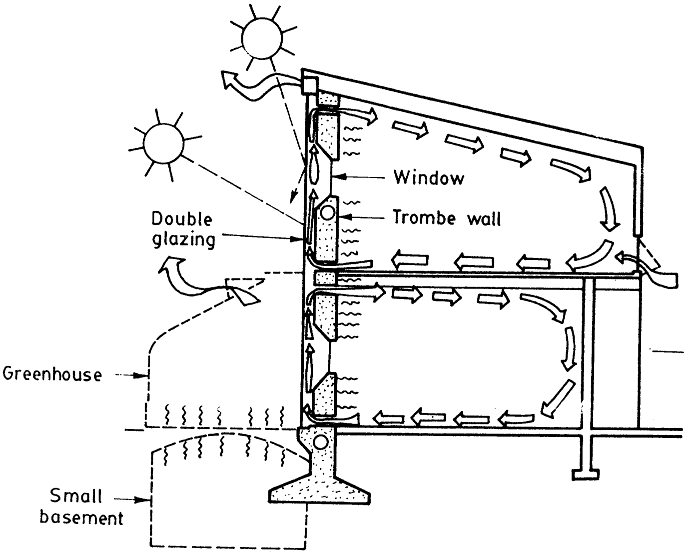


FIG.6.13 CROSS-SECTION (LOOKING WEST) KELBOUGH HOUSE (From SANDIA[29])

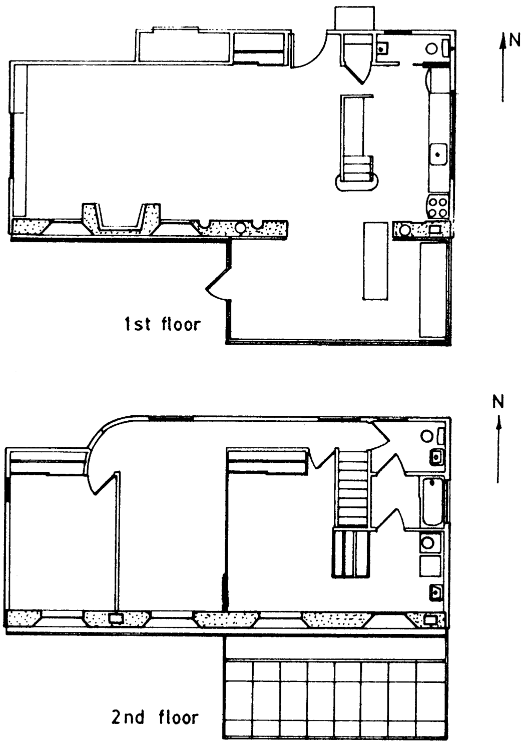


FIG.6.14 FLOOR-PLANS OF KELBOUGH HOUSE (From SANDIA[29])

foam panels which can be used at night time to reduce heat loss. The roof is heavily insulated containing 25 cm cellulosic fibre. Mainly in this house, the heat is stored in the trombe wall which is directly heated by the sun rays. Some heat is also stored in the 205 litres drum of water kept in the green house, green house floor made of concrete, and also by concrete floor of the house. Auxiliary heat in the house is supplied either by the gas fired, forced hot air furnace or by a heatilator wood fireplace. Hot water for domestic use is supplied by a gas-fired heater. Heat supplied to the house is controlled by dampers and blower fans. In summer the excess heat is exhausted out.

In winter season, the direct sunrays strike the trombe wall after penetrating through the double glazing and heats the wall. The warm wall and glazing heat the enclosed air which rises, moves up, goes into the room through the upper vent and cool room air from the bottom vent goes in the enclosed space. The warm air supplied to the room gradually cools, goes down near the floor and then enters in the enclosed space and further gets heated. The heat from the surface of the concrete wall also moves by conduction on its other side. Thus the inner side of the wall also radiates heat to the room. During night time since wall is already heated continue to radiate heat to the room, thereby keeping the space warm. On cloudy days, the room thermostat will automatically switch on the gas fired forced circulation furnace heating the room by auxiliary energy. In summer, the angle of incidence of sun rays with the glazing becomes quite large and hence most of the rays are reflected back.

The design temperature of the house is -17.7 $^{\circ}$C and the heating degree days are 2833 $^{\circ}$C-day. The solar insolation on a horizontal surface on a typical January day is 7.25 MJ/m^2 day.

Two problems were encountered during the experimentation. One was of reverse thermosiphoning which was solved by using a passive damper consisting of a screen covered by a 12.5 micron thick plastic film and installed on the inside of the concrete wall. This damper allows the air circulation in the inward direction only. The second problem was specific in this design only since the hot air was going fast to the second floor only. This was stopped by using a door at the top of the open stairwell. The house was monitored during the summer and winter season of 1977. The recorded data for July 19-23 and for December 4-10, 1977 is shown in fig.6.15 and 6.16 respectively. It is seen from fig.6.15 that the outside air temperature fluctuates from 14 $^{\circ}$C to 38 $^{\circ}$C while due to heavy thermal mass in the house the inside room air temperature fluctuates only from 20 $^{\circ}$C to 30 $^{\circ}$C. From fig. 6.16 it is seen that in winter season, while the outside air temperature fluctuates from -10 $^{\circ}$C to +14 $^{\circ}$C the inside air temperature fluctuates only from 14 $^{\circ}$C to 22 °C. The air temperature in the green house fluctuates quite high and therefore water storage drums are used to store the excess heat.

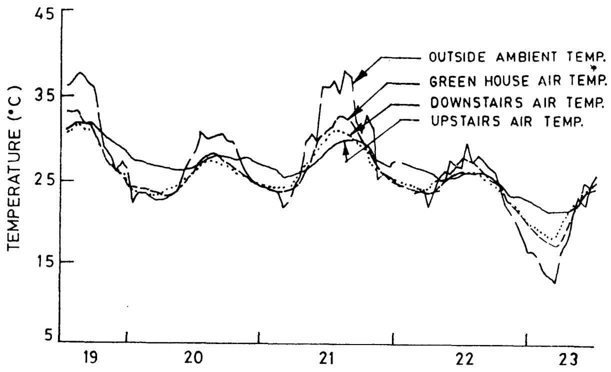


FIG.6.15 EXPERIMENTALLY RECORDED DATA FOR KELBOUGH HOME FOR JULY 19-23, 1977 (From SANDIA[29])

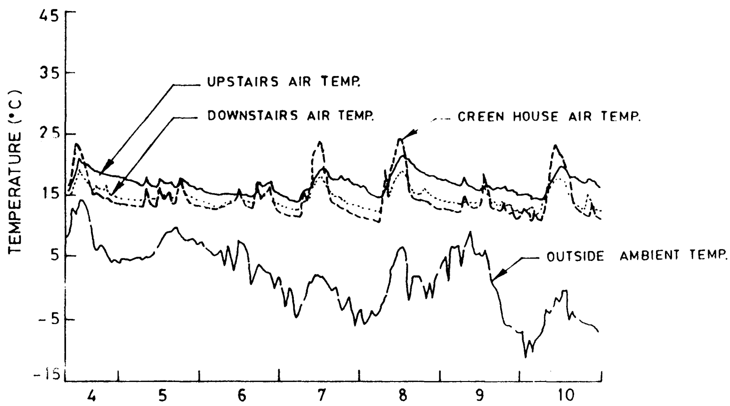


FIG.6.16 EXPERIMENTALLY RECORDED DATA FOR KELBOUGH HOME FOR DECEMBER 4-10, 1977 (From SANDIA[29])

The computed building heat loss coefficient for Kelbaugh house is 7.67 MJ/(°C day). The calculated solar-heating fraction was about 76 percent which was found to be the same by experimental observations also. The heat received by direct gain was estimated to be only 9 percent of the heat absorbed directly from sun rays. The fluctuation in air temperature in the rooms was found to be not uncomfortable. After making several observations and living at least for a year in the house it was observed that the upper and lower vents should be larger, the house should be insulated with at least 5.0 cm thick urethane instead of 2.5 cm used earlier, and some kind of movable insulation should be provided between the greenhouse and the trombe wall particularly during very cold nights.

6.4.3 Thermal Storage Wall (drum wall) Installation.

The drum wall or water wall concept for passive solar heating is similar to the Trombe-Michel wall concept except that here instead of masonary or concrete wall, use of water may be in drums or bags is made where these are stacked in the form of a south facing wall. Water or drum wall appears to be attractive since water is cheap, easily usable, and has high thermal capacity per unit of volume compared to concrete. Moreover, the heat gains by convection is faster and better in case of water wall and much reduced surface temperatures are obtained. The problems with water wall are leaking, corrosion and of containment. Perhaps the first passive heating house using drum wall was constructed by Baer[43] in 1971 in Corrales, New Mexico, U.S.A. who used 100 steel drums each of 210 litres capacity filled with water which are stacked behind double-glazed south facing windows and covered with insulation panels during winter nights.

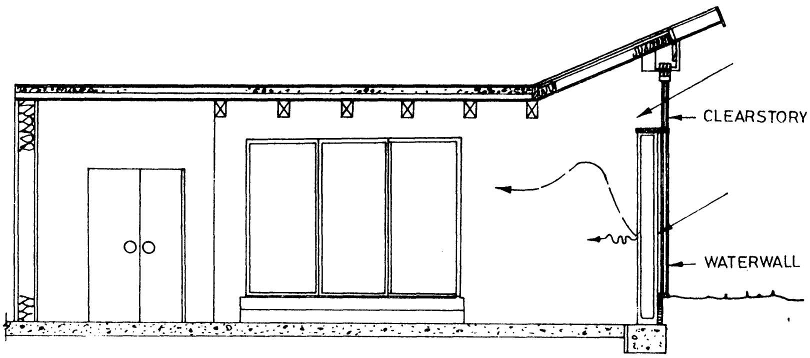


FIG.6.17 CROSS-SECTION OF GUNDERSON HOUSE (From SANDIA[29])

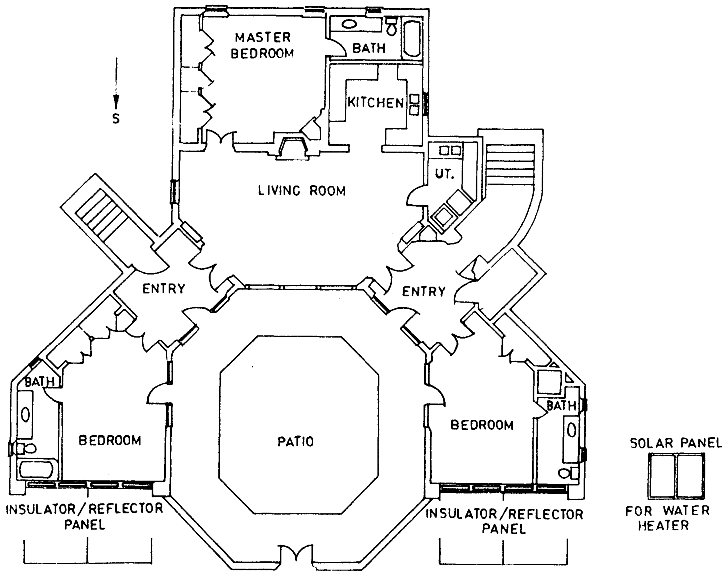


FIG.6.18 FLOOR-PLAN OF GUNDERSON HOUSE (From SANDIA[29])

A direct gain-water wall single storey passive solar heated house was designed by Wayne and Susan Nichols and Bill Lumpkin with a total floor area of 204 m2 and was built at Santa Fe (36°N), New Mexico, USA at an elevation of 2133 m. The house is owned by Dave Gunderson and is known as Gunderson house[29]. The cross-section and floor plan of Gunderson house are shown in figs.6.17 and 6.18 respectively. As is seen from fig 6.18, there is a living cum dining room, master bed room, kitchen, and bath are located on the north side of the house and are heated by direct gain through large clearstory windows. In the centre on the south side of the house there is a large garden and patio area. On east and west sides of this patio, there are two identical wings with each wing having a bedroom and a bath and each is connected to the central section through a small greenhouse. On the south of the patio there is a adobe wall and wooden entry gate. The south walls of each of the two bedrooms are constructed of precast concrete sections having 15 cm thick cavity lined with plastic bag and filled with water. The water side of the wall is painted black and double glazed. Insulating shutters lined with reflecting material are hinged on the lower side of the wall and are kept open during sunshine hours, thereby allowing solar radiation to fall directly on the water wall and also after reflection from the lining of the insulating shutters. During off sunshine hours the insulating shutters are closed manually with the help of crank and shaft arrangement to reduce the heat loss. The east and west exterior walls are of 20 cm thick concrete block with the cells filled with concrete and insulated with 5.0 cm of styrofoam. The wall on the north side is of 30 cm thick concrete block filled with concrete. Some of the interior walls are also of 20 cm thick concrete block with cells filled with concrete. Some other exterior and interior walls are of stud frame construction with 1.25 cm thick cellotex board on the outer side and gypsum board on the inner side with cavity in between. The floor of the house is of 10 cm thick concrete slab insulated with 45 cm wide strip of 5.0 cm thick styrofoam. The perimeter heat loss is further reduced by making the floor at about 1.0 cm below the grade. The roof is made of wooden beams and planks and insulated with 10.0 cm styrofoam and sealed with asphalt and gravel. The windows on the south side bed rooms and clearstory windows are thermopane units while all other windows are of double glazed type with wooden frame.

The north wall, the concrete floor, and masonary interior walls store thermal energy from the direct gain. The two water walls on the south side stores the thermal energy directly. Because of the special construction of the house and the distribution of various thermal masses there is moderate fluctuation of interior air temperature and also the design leads to a high solar-heating fraction. The auxiliary heat is provided with electric baseboard heaters fitted in each room. The hot water for domestic purposes was provided by a solar water heater consisting of 3 m^2 flat-plate collector and a 150 litre preheat tank connected to a 150 litre electric hot water heater. As discussed earlier, the master bed room, bed, kitchen and part of living room receive heat by direct gain through clearstory with glazing area of 12.7 m^2. The window area on the south facing bedrooms is about 8 m^2. The excess heat if any is flushed through vents provided on the walls in the central section near the apex of the tilted roof section. Ventilation can be enhanced by opening the windows on the north wall also. Overhangs on the window are also used which cut the entry of sun rays in summer. The spacing between the glazing and the water wall is unvented and therefore the enclosed air when gets heated gives part of its heat to the concrete wall and part to the glazing which is finally lost to the outside air. The concrete surface of the water wall also receives heat by absorbing the sunrays and conducts this heat through 5.0 cm thick concrete to the water in the cavity. The water in the cavity gets heated rapidly by convective movement and transfers the heat to the concrete wall (5.0 cm thick) on its other side. From the inner surface of the concrete wall facing bedroom the heat is delivered to the bed room by convection and radiation. The bed rooms also receive direct sun rays through small clearstory provided on the upper section of the water wall.

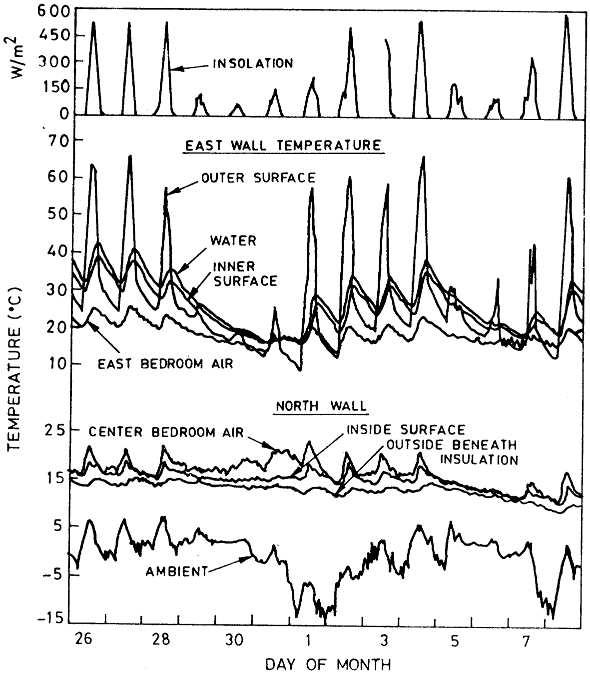


FIG.6.19 EXPERIMENTALLY RECORDED DATA FOR GUNDERSON HOUSE FROM DEC.26, 1978 TO JAN.8,1979 (From SANDIA[29])

The calculated building loss coefficient for the house is about 14.58 MJ/(°C day) or 6.31 KJ/m² (°C day) and the annual solar heating fraction is 72 percent. The experimentally recorded data for this house from December 26, 1978 to January 8, 1979 is shown in fig. 6.19. From this figure it is seen that the ambient temperature varies from 6°C during daytime to -6°C during night time and sometimes it drops to -11°C also. It is also seen from the solar radiation measurement that during the period of measurements some days are clear days followed by cloudy days. The solar insolation on a horizontal surface on a typical winter day is about 12.38 MJ/m². The outer water wall temperature varies from 63°C during day time to 20°C during night time. The water temperature in the water walls varies from 36°C during day time to about 25 °C during night time. On cloudy days, the water wall surface temperature, water temperature, and inside surface temperature drops gradually to 15 °C. In this design the master bed room, bath, and kitchen are completely thermally isolated from the water wall heating side and heated by direct gain only. The temperatures recorded on the north side rooms are also shown in fig. 6.19. From this figure it is seen that the air temperature in the master bed room varies from 22 °C during daytime to 14 °C during night time. The fluctuation in temperatures on the inner surface and on the outer surface beneath the insulation of wall is small. The wall surface temperatures are always lower then the room air temperature and therefore do not contribute in heating the room air.

6.4.4 Attached Greenhouse Installation(SANDIA[29]).

As discussed earlier, the disadvantage of a direct gain passive heating system is large room air temperature fluctuation which can be considerably reduced by the use of a sunspace(attached greenhouse). One of the best known example of attached greenhouse passive Solar Heating unit is unit-I of first village, Santa Fe(36° N, 2347m altitude), New Mexico, USA. This house is also known as Balcomb house and

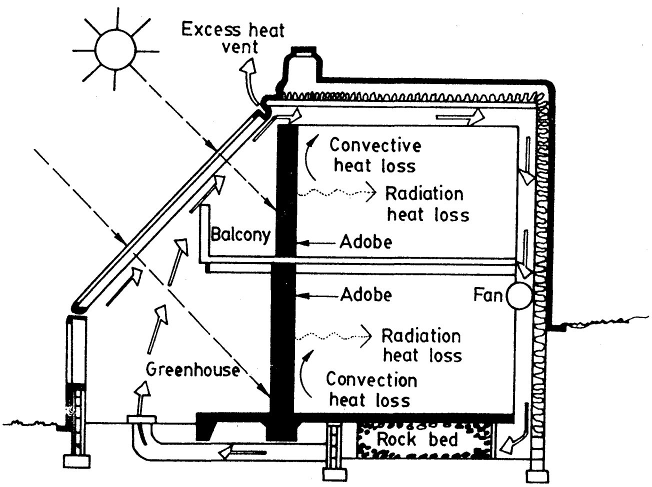


FIG.6.20 CROSS-SECTION OF BALCOMB HOUSE (From SANDIA[29])

is a two storey single family house with a floor area of 213 m^2 with a greenhouse with an aperture of 38 m^2. This house was designed by Susan Nichols and William Lumpkins and built by Wayne and Susan Nichols in 1975 in which Doug and Sara Balcomb live. This is a 'L' shaped house designed in this shape to accomodate green house meant for heating the space. The cross-section and the floor-plans of this house are shown in figs 6.20 and 6.21 respectively. On the first

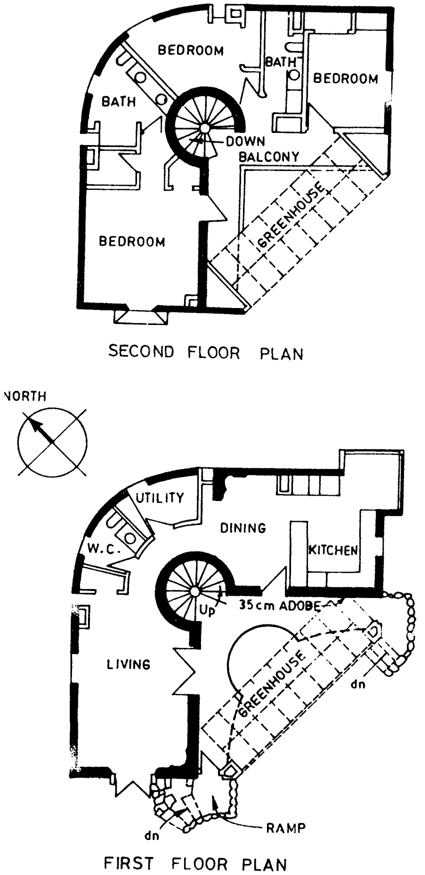


FIG.6.21 FLOOR-PLANS OF BALCOMB HOUSE (From SANDIA[29])

floor of the house there is a living room and the dining room which open into the green house while the window of the kitchen is also open into the greenhouse. A small end corner of the kitchen receives direct solar radiation also from outer side of the house. On the north corner of the kitchen there is a circular stair case for going to the second floor. On the second floor of the house there are three bed rooms and a bath. The house is so built that the greenhouse side is above the grade while the north side is about 1.4m below the grade. The greenhouse on the south side is made of double glazing with an aperture area of 38 m^2 out of which 1/3 area is vertical and 2/3 area sloping about 60° from horizontal. The wall in between the green house and living rooms is 35 cm thick adobe wall on the second floor. The circular stair-case is also surrounded by adobe wall which acts as thermal storage. Heat is also stored in two rock bed units, one below the living room of about 10.7 m^3 capacity and second unit below the dining room of about 8.4 m^3 capacity. By using a 250w motor blower hot air from the top of the green-house is circulated through the rock bed and then leaves at the floor of the green house. The outer walls of the house are of wooden frame with the space filled with fibreglass insulation.

During the daytime when the sun shines, the solar radiation penetrates through the double glazing of the green house and falls on the adobe walls and on the greenhouse floor. Thus the air temperature in the greenhouse increases and the warm air moves up and collected near the roof of the greenhouse from where it is sucked by the fan to the rockbed storage unit. Thus the rock bed storage unit gets heated which further heats the floors of living room and dining room of the first floor. The cool air from the storage unit is supplied to the floor of the green house. A differential controller is used for the operation of the fan. When the temperature difference between the hot air at the top of the greenhouse exceeds by $8^{\circ}C$ from the rock bed temperature, the differential controller makes the fan to circulate the air. The direct heat absorbed by the adobe wall on the first and second floor of the house on the outerside gets conducted on the inner side of the house from where the radiation and convection heat loss from the wall heats the rooms. The time lag for the flow of heat from the outer surface to the inner surface of the wall is about 7 to 8 hours which means that heat absorbed during the daytime will reach on the other side during the evening hours. During the daytime in winter season the doors of the rooms facing the greenhouse are kept open allowing the hot air from the green house to move into the rooms thereby heating them. In evenings or when the air temperature in the greenhouse starts falling these doors are closed. In summers, the excess heat from the green house is exhausted through the vents provided at the top of the staircase and the staircase acts as a chimney forcing hot air to go out rapidly. In summers, when cooling is required during night time, the vents and the doors opening into the greenhouse are kept open. This results in the movement of cold air in the space and the adobe wall also gets cooled. During daytime, the doors are closed and the cooled adobe wall keeps the indoor air temperature at a low level.

Another important feature of the house is the horizontal projection of the roof of greenhouse and the balcony of the house on the south side. This results in cutting the direct sun rays falling on the adobe wall in summer season while allowing it in the winter season. Hot water for domestic use is provided using flat-plate collector, a preheat storage tank, and an electric heated hot water storage tank. Auxiliary heat is provided using base board electric heaters provided in each room.

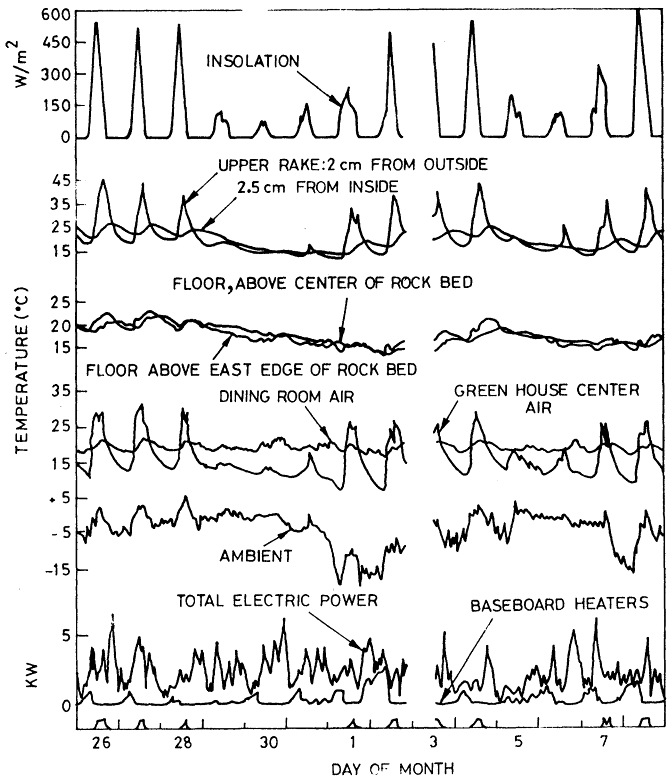


FIG.6.22 EXPERIMENTALLY RECORDED DATA FOR BALCOMB HOUSE FROM DEC.26,1978 TO JAN.8,1979 (From SANDIA[29])

The experimentally recorded data for the Balcomb house from December 26, 1978 to January 8, 1979 is shown in fig.6.22. It is seen from this figure that during the period of observations, there are several clear days which are followed by cloudy days. From this figure it is seen that ambient temperature during this period varies from $-15^{\circ}C$ to $+5^{\circ}C$. The outside wall temperature reaches to more than $45^{\circ}C$ on sunny days. The floor temperature above the rock bed ranges from $15^{\circ}C$ to $22^{\circ}C$. A large fluctuation in the air temperature in the green house is observed while the air temperature in the dining room remains practically constant which is around $20^{\circ}C$.

6.4.5 Thermal Storage Roof Installation.

The roof pond concept for heating and cooling of buildings was invented by Harold Hay[11,12] of USA who along with Kenneth Haggard designed and built a full dwelling based on roof pond system in 1973 at Atascadero(35° 27'N, altitude 260 m), California, USA. The Atascadero house shown in fig.6.23 and the floorplan shown in fig.6.24 is a

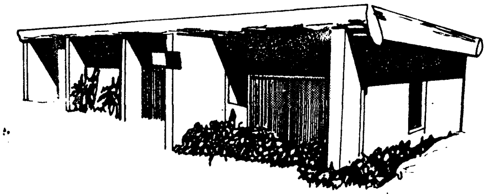


FIG.6.23 THE ATASCADERO, CALIFORNIA 'SKYTHERM' HOUSE

single story building with a floor area of 107 m^2 with the floor made up of concrete slab. The house contains[44] three bedrooms one living cum dining room, two bath house, one kitchen, garage, and patio adjacent to north end, etc. and is working since 1973 and providing comfortable indoor air temperatures in winter and summer without any auxiliary energy. The outer walls of the house are made of concrete blocks. Some of the partition walls are also made of concrete blocks and are sand filled. Some of the concrete blocks of the outer walls are also filled with vermiculite insulation. The south wall is not provided with any window except in bathroom but windows on other walls (21 percent area) are double glazed. The special feature of the house is the steel panel roof of 1 mm thick which holds 4 bags each of 2.7 m by 12.7 m by 20 cm deep made of transparent 0.5 mm thick polyvinylchloride (PVC) sheet. These four bags

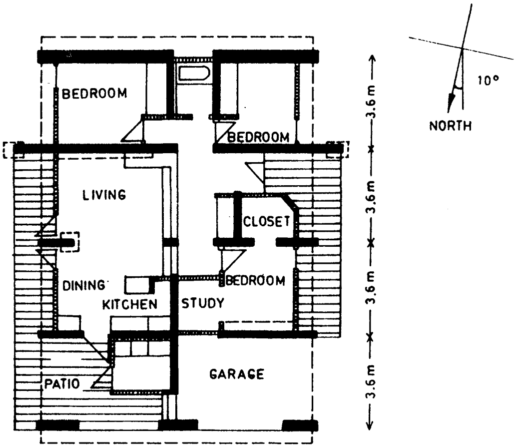


FIG.6.24 FLOOR PLAN OF ATASCADERO, CALIFORNIA 'SKYTHERM' HOUSE

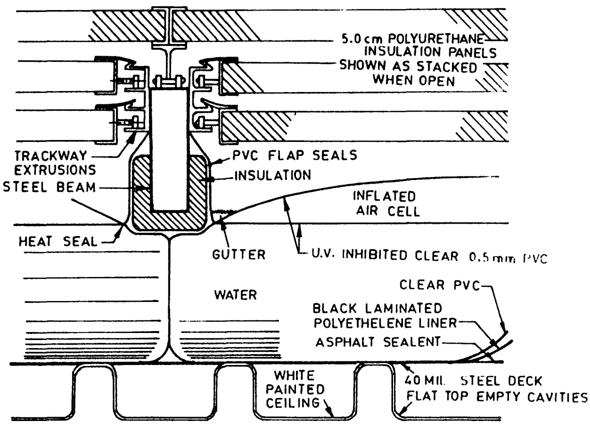


FIG.6.25 CROSS-SECTION OF WATER BAG AND PANEL SYSTEM OF ATASCADERO 'SKYTHERM' HOUSE

lie in the four spaces(alleys, or bays) between five spaced, N-S tracks. These bags contain about 27.43 m^3 of water without the use of any antifreeze solution. Above each of the water bags, a transparent UV-stabilized PVC sheet is sealed along the edges of the bags which is held due to air pressure about 5.0 cm above the bags which reduces heat losses from the top of the bags. Below the water bags there is a black PVC sheet lying over the metal ceiling which helps in absorbing more solar radiation by water in the bags and also heats the metal ceiling. Fig.6.25 shows the cross-section view of water bag on the ceiling and also the movable insulation. The movable insulation consists of 5.0 cm thick rigid polyurethane panels covered with aluminium foil on both the sides of the panel to reduce radiative heat loss. These insulating panels which are rectangular and nine in numbers can be rolled along the roof beams horizontally to expose or cover the roof. The total collector area is about 102.2 m^2. The insulating panels can be moved to cover the water ponds during winter nights and summer days in 10 minutes time by using a 250 W electric motor.

In winter months when heating is required, the water bags are exposed to solar radiation during the daytime. This results in increase in water temperature and metal ceiling temperature. The metal ceiling results in quick transfer of heat into the room and the room gets heated primarily by radiation exchange. During night time, the water bags are covered by insulation panels resulting in reduction of heat loss from water bags to the outside. The heated water in water bags continue to transfer its heat through the metal ceiling to the rooms below. Thus these water bags collect the heat, store the heat, and transfer the heat to the metal ceiling. In summer when cooling of rooms is required, the pressurization of plastic sheet over the plastic bags is discontinued, and the water bags are covered with insulating panels during the daytime and their aluminium foil reflects solar radiation back to the sky. Thus during daytime the cool water in the bags absorb heat from the metallic ceiling, which in turn cools the rooms. During night time, the insulating panels are removed, and the water bags are exposed to the sky resulting in losing heat by radiation and convection. Additional cooling can be provided by spraying some water on the bags during night time which gets evaporated and provides further cooling. This type of thermal storage roof is more suitable at low latitude stations and for dry climates where there is little snowfall and where heating and cooling are of equal importance. Moreover, it is suited to only single story houses.

The over all performance of the house for a period of nine months from 1st February 1974 to 31st October 1974 is

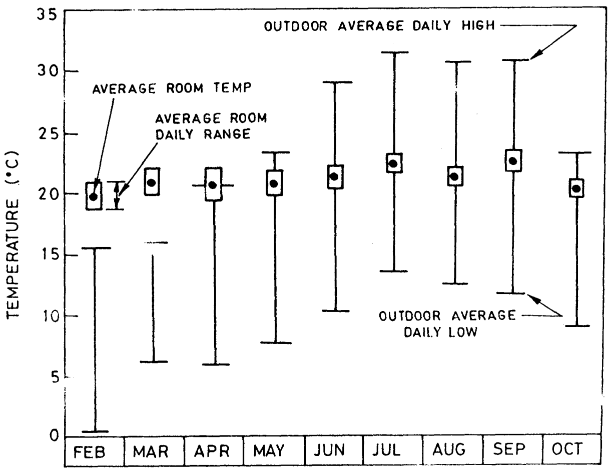


FIG.6.26 MONTHLY AVERAGE PERFORMANCE OF ATASCADERO 'SKYTHERM' HOUSE

shown in fig.6.26. From this figure it is seen that there is only very little variation in indoor air temperature. The indoor air temperature varies from 10 ^{o}C in winter to 23 ^{o}C in summer while outside ambient temperature varies from 1 ^{o}C to 31 ^{o}C. The occupants of the house were very much satisfied since there was very little fluctuation in indoor air temperature throughout the year and no auxiliary energy was used.

6.4.6 Convective Loop Installation(From SANDIA[29])

In a convective loop passive solar heating system, there is some simple, cheap, but efficient solar air heater connected to a rock bed storage unit. The circulation of air between the storage unit and the air heater is generally without the use of fans and takes place by thermosyphon action. For this purpose, the storage unit is to be placed at a higher height relative to solar air heaters. Where this is not possible a small electrically operated fan is used for the circulation of air between the storage unit and the air heater. Although several houses[45,46], are recently made based on the principle of convective loop or thermosiphoning loop but the most studied passive solar heated building using convective loop system is the Jones House which is designed and built by Mark Jones and owned by Mark and Faith Jones and is located at Santa Fe County

(36°N, 2057 m above mean sea level), New Mexico, USA. The house is a single family, single story residence with a floor area of 246 m^2. The house is situated on a south and east facing knoll with excellent east and northeast views. The solar collector array is on the south slope of the knoll and is well below the floor of the rooms allowing the movement of air through collector array by thermosyphon action. The floor plans of the house is shown in fig.6.27. As is seen from this figure, the plan is L-shaped with kitchen and a dining room on the north side and a living room at a slightly lower level on the north side. Several double glazed windows are provided on the east and northeast side to have a good view. The south side is of multilevel with master bed room and bath on the upper level and a second bedroom and a bath on a lower level. There is another bedroom and a bath just below the master bedroom. On this level there is a mechanical room, a rock bed storage unit, a green house, and solar collector array.

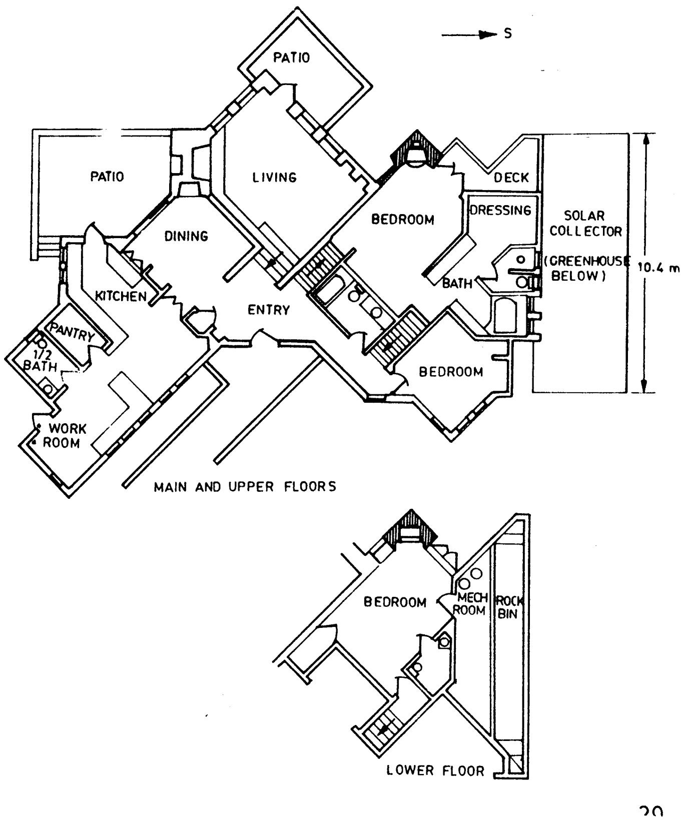


FIG.6.27 FLOOR PLAN OF MARK JONES HOUSE (From SANDIA[29])

The building walls are made of double layer wood frame, with 2X6 studs on the outside, 2X4 studs on the inside with 5.0 cm dead air space in between. The fibreglass insulation used in walls has an R value of 30. Part of the walls on the west and northwest are bermed. The walls which are below grade are insulated with 5.0 cm of polystyrene. Minimum window area is provided on the north side to reduce heat loss. Several partition walls are made of 35 cm thick solid grouted concrete block to increase the thermal mass. The flat roof is supported with 15 cmX25 cm wooden beams over which is placed 25 cm thick fibreglass layer supported on both sides by plywood sheets. The outer exposed surface is water proofed with built up asphalt roofing.

The solar heating part consists of a solar collector array, a storage unit, and an attached greenhouse. The storage unit is 1.2 m deep and 10.4 m long with a volume of 14 m^3 containing 27200 kg rock of about 4.0 to 7.5 cm diameter. The bin is of wood frame construction with steel tension cross ties and plywood/drywall liner and insulated to R-22. The solar collector array is oriented due south with a tilt angle of 45^o with a net effective absorber area of 49 m^2. Into the lower portion of the array, a 5 m^2 collector area is used for heating the water for domestic use. Some sun light through the transparent windows of the collector array goes to the greenhouse situated below the collector array. Solar collector array consists of a plywood backing, and a single glazed Kalwall Sunlite 1 mm plastic glazing. The absorber plate is galvanized sheet metal pans of 'U' shape of 7.6 cm deep with 3-layers of 1.0 cm mesh wire lath set at the top of the pans. The absorber surface is painted with black paint. The area below the collector array is used as a green house with 10 m^2 of glazed area. Fresh air into the collector enters through this green house only. Several varieties of vegetables are grown in this greenhouse and heat is stored in it using drums of water.

Hot air from the collector array top enters by natural means to the plenum above the rock storage bin. Manually operated hinged damper doors are used in the ducts to check the reverse flow during night time or in cloudy weather. During sun up hours the plenum above the rock bin develops sufficient pressure due to hot air from the collector outlet to drive the hot air down through the rocks. Cooled air from the bottom of the rock bin enters the space below the collector and reenters the collector at its base. Under free flow conditions air flow rates of 0.5 to 0.6 m/s are measured. The cross-section of the collector and storage unit is shown in fig.6.28. Heat from the storage bin to the house is transferred by using an electric operated fan with a capacity of 470 l/s and is operated automatically using a single thermostat fitted in the house. Cool air from the house reenters the bin bottom plenum, completing the circuit, and the flow is upward in the rock bin during the heating period. When the rock bin is being charged, the thermosiphon-powered flow is downward through the rock bed. This temperature stratification in the rock bin with the hottest rock-air at the top helps in delivering the hottest air to the room. An electric coil(15KW) is fitted in the upper plenum of the rock bin which is automatically switched on when hot air from the rock bed alone is not able to heat the rooms.

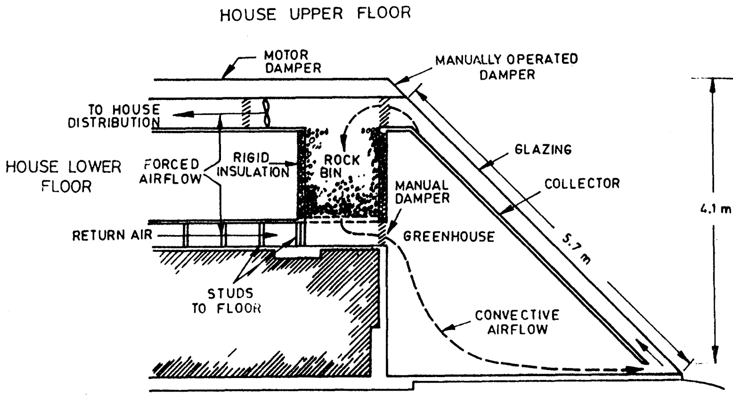


FIG.6.28 CROSS-SECTION OF SOLAR HEATING SYSTEM OF MARK JONES HOUSE (From SANDIA[29])

More than 28 thermocouples are used to measure temperatures at various positions in the collector, rockbin, and other places during the two weeks period in winter in 1979 of experimentation. The recorded data is shown in fig.6.29. The highest collector temperature recorded during this period was 90 °C. It is seen from fig.6.29, that the clear days are followed by cloudy days. It is also seen that as far as rock bin is concerned, there is a large fluctuation in temperature at the top of the rock bin compared to at the bottom of the rock bin. At the bottom the temperature fluctuates between 10 to 15 °C while at the top, the temperature reaches to sometimes at 65 °C. The temperature in the rock bin increase from bottom to the top as expected. Although the collector inlet temperature varies from 5 °C to 15 °C, the collector outlet temperature varies from 10 °C to 75 °C. The ambient temperature varied from -17 °C to +7 °C but with the solar heating system, the house was maintained at a temperature of 18 to 22 °C during

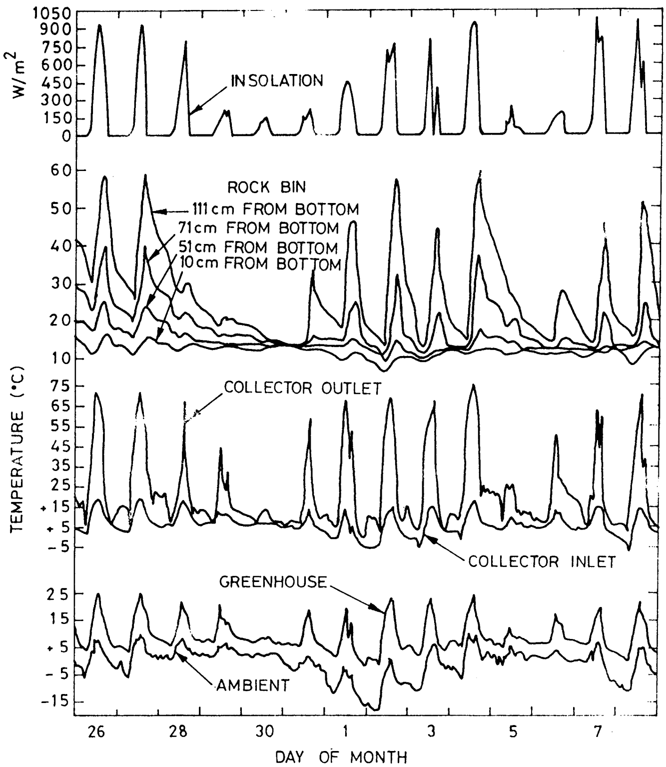


FIG.6.29 EXPERIMENTALLY RECORDED DATA FOR MARK JONES HOUSE FROM DEC.26 TO JAN.7, 1985 (From SANDIA[29])

the heating season without the use of any auxiliary energy. The temperature in the green house varied from 0 ^{o}C to 28 ^{o}C. The overall collector efficiency was observed to be about 30 percent. The overall solar heating fraction was 84 percent out of which more than 74 percent was provided by solar collector and storage unit.

6.5 CRITICAL PARAMETERS FOR DESIGN OF PASSIVE HEATING SYSTEMS.

If passive solar buildings are carefully designed, then solar energy can provide a large solar heating fraction. The two main problems in passive heating systems are the sufficient energy storage and the control and distribution of temperature and heat. By providing sufficient energy storage, the variation in indoor air temperature which may take place due to variable nature of solar radiation, can be reduced. By selecting building materials of appropriate thermal properties, the indoor temperature can be conveniently controlled. Therefore it is essential for a designer of a passive solar building to know the dynamic thermal principles on which such a building operates. Passive Solar buildings should be designed on year round basis to provide comfort for all the twelve months and provision of some kind of 'thermal flywheel' to damp out diurnal variation[47] in indoor air temperature is essential. In all the passive heating concepts there are some key design parameters the effect of which on the performance can be studied either experimentally or by using mathematical models. In this section the discussion will be on the importance of these key design parameters.

6.5.1 Direct Gain System

As discussed earlier, in a direct gain system the solar radiation directly enters the building through suitably oriented windows and gets absorbed by floor, various walls, and other materials in rooms. The heat absorbed is subsequently available for heating the indoor air whenever it is required. Some of the key design features of a passive heating system are: The south facing windows should be as large as possible with double glazing; some provision of insulating the window at night time to reduce heat loss should be provided; some kind of overhang or other sun control device on the external side of the window should be provided so that solar radiation in summer does not enter the building; a floor of sufficient thermal capacity like that of concrete with some light colour at the top which helps in distributing the incoming solar radiation, and insulation below should be used; Walls of adequate thermal capacity with some insulation on the outer side should be provided; and provision of some kind of ventilation arrangement may be by using weather-sealed openable windows particularly in summer to avoid excessive heating is essential.

The window should be designed in such a way that it should miximize the solar gain in the building in heating season, the loss of heat by conduction through window should be as small as possible, excessive ventilation losses should be avoided, and suitable overhang should be used to reduce direct solar gain in summer. One would like in a direct gain system that the solar transmittance through window should be as large as possible and heat loss through it should be as small as possible. There are several ways of increasing the overall transmission coefficient through window such as by reducing the convective loss through the space between the glazings, by increasing transmission using better quality glass or plastic sheet and by using heat mirror coatings. Convection heat loss through air space in double and triple glazing becomes quite substantial and is about 40 percent of the total heat loss through double glazing. This convection heat loss can be reduced either by optimizing the air spacing between the two glazing or by evacuating the space between the two glazing or by introducing some heavy gas in the space. All the above three methods appear to be effective but not practical. As is seen[31] from table 6.1 for ordinary 3.0mm thick single sheet of window glass the transmittance for solar radiation is 0.87 while if one uses a low iron content glass the transmittance will be 0.92. The transmittance for double glazing in case of ordinary glass is 0.76 while for low iron content glass it is 0.85. Thin plastics if used in place of glass will give higher transmission for solar radiation but most plastics are transparent for longwave radiation and also and deteriorate with time. Teflon FEP is a special plastic produced by Du Pont company of USA which is about 50 micron thick with a refractive index of 1.33 and solar transmission of 96 percent is recommended as a suitable window material. But teflon is also not completely opaque to longwave radiation. Thus the advantage in increased solar transmittance gets lost in increased radiation loss. Some transparent insulations are also being developed by Suntek Inc., California by coupling the teflon with another transparent material which is opaque to long wave radiation. The transmission for this combination is 0.95.

By using Drude mirror or heat mirror coating, the longwave radiation losses by reflecting them back to the room can be lowered thereby increasing the overall effectiveness of the window. Several firms in USA and Germany are producing heat mirror coatings on glass and plastics in small sizes only. By using Drude mirror or heat mirror coatings on glass, the solar transmittance gets a little reduced, the effectiveness of this combination improves, but the cost becomes quite high.

The heat loss through the window which is generally expressed in terms of 'U' values which is the overall heat loss coefficient can be reduced by using multiple glazing with the above modification suggested or by using a movable insulation over the window during off sun shine hours. The 'U' values for different kind of windows with and without insulation are shown[31] in table 6.1. The insulating materials which can be used are several and any one of the following can be used:

(i) Rigid expanded polystyrene sheets or any other material sheet can be manually inserted at night and removed in the morning.

Table 6.1 Overall heat loss Coefficient U ($W/m^2\ ^\circ C$) for glazing with movable insulation(From Ref.31)

1	Single * Glass 2	Double Glass 3	Triple Glass 4
Solar Transmission Values			
Nominal Solar 'Transmittance'	0.87	0.76	0.66
Approximate seasonal Transmittance	0.80-0.85	0.64-0.72	0.51-0.61
'U' Value (winter)**			
Nominal U-Value	6.52	3.12	1.98
With R-4 insulating cover	1.19	0.96	0.79
Average U-Value with R-4 cover in place, 16 hr/day (3/4 of the degree days)	2.55	1.53	1.13
Average U-Value with R-4*** Cover in place,12 hr/day (2/3 of the degree day)	2.95	1.64	1.19
With R-10 insulating cover	0.51	0.48	0.44

Table 6.1 cont.

1	2	3	4
Average U-Value with R-10 cover in place, 16 hr/day (3/4 of the degree day)	2.04	1.13	0.85
Average U-Value with R-10 Cover in place, 12 hr/day (2/3 of the degree day)	2.49	1.36	0.96

* for 3.0 mm grade B window glass only

** Values are slightly different in summer

*** R is the resistance offered by the fabric, R-4 means resistence offered is 4 $ft^2\,hr\,^{o}F/Btu$. Since this is a standard practice to give R value in FPS units only hence the same are used here.

(ii) Roller shade devices made out of wood or plastic slats can be used.

(iii) Framed and hinged insulating panels on the outer side which can be operated either manually or automatically using sun sensitive system.

(iv) Roller shade devices made out of aluminized mylar in combination of cloth or other suitable material.

(v) By using beadwall in which the space between glazings is filled by polystyrene beads at night using blowers and removing the same during sunshine hours.

(vi) By using 'skylid' in which insulating louvers are opened and closed automatically during sunshine hours and off sunshine hours respectively.

These insulating panels of shutter will be effective only when these are sealed properly when closed. The effect of number of glazings, NGL, the resistance of moveable night insulation, Rn, and the allowable temperature swing, ΔT, about the 20 ^{o}C reference value for a room of thermal storage mass equal to 1098 kg/m^2 of glass area and glazing area to building load ratio equal to 0.57 m^2 ^{o}C/W on the solar fraction is computed by Wray and Balcomb[48] and the results are shown in fig. 6.30. In this figure, eight

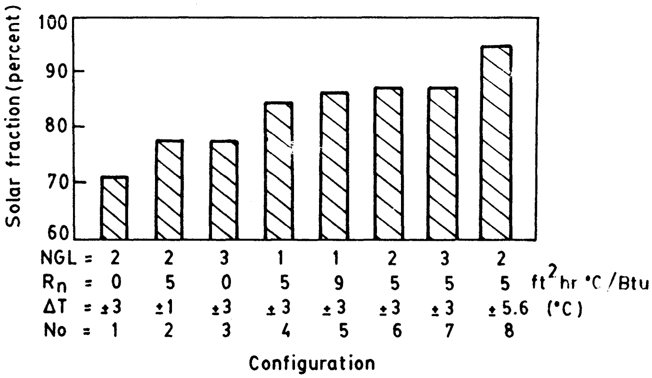


FIG.6.30 THE EFFECT OF SOME DESIGN PARAMETERS IN DIRECT GAIN SYSTEM (From Wary and Balcomb[48])

configurations are compared. By comparing configuration 1 and 3, the effect of number of glazings can be seen which is quite substantial. If the number of glazings are more then the effect of movable insulation becomes marginal. It is also seen that the solar fraction can be increased considerably by the use of movable insulation. From configurations 4 to 7, it is seen that for the temperature swing of $\pm$ 3 ^{o}C, there is not significant improvement in the solar fraction since there is atleast a night insulation of R-5. A slight improvement can be achieved if movable insulation of R-9 value is used. Here the effect of number of glazings also gets masked due to the presence of movable insulation. It is also seen that if the allowable temperature swing in the room is increased, the solar load fraction also gets increased as is clear from configurations 2, 6, and 10.

In summers, the use of glass on the south side, may cause overheating even in cold or temperate countries. This overheating can be avoided either by using shading devices, or by using building material of high thermal capacity or by using adequate ventilation in summers. Generally in a passive heated building, all the three concepts are used to avoid summer overheating. The shading devices which are more effective can be of fixed type, adjustable type, or retractable type. These shading devices can be used either on the external side of the window which includes overhangs; awnings; shutters; and a variety of louvers may be vertical, horizontal, or combination of both; or on the internal side of the window which includes venetian blinds, roller blinds, curtains, etc. These external and internal shading devices can be of variety of architectural shapes and geometric configurations. In architectural practice the geometrical principles of shading designs are well established and it is easier to design a fixed overhang to keep the sun off from the south wall but it is difficult to assess its overall energy implications. The effectiveness of different external shading devices are discussed by Givoni and Hoffman[49]. If internal shading device is used then its effectiveness will depend largely on how much solar radiation is reflected back by it to the glazing.

Storage in a direct gain system plays an important role since in its absence, there can be an overheating in the room during daytime which can be avoided using ventilation resulting in rejection of heat during daytime which could be used by storage during daytime and used at night to maintain comfortable temperature. Placement of storage relative to the window and its conductivity also affect the performance. Mazaria et al[50] and Balcomb et al[51] have done some studies on the positioning of storage mass relative to the window in a direct gain system and concluded that thermal storage mass exposed to solar radiation is four times more effective in capturing heat and distributing it than when the thermal storage mass is placed in some other unexposed position in the room. It is also concluded that the lower position of the storage mass with large exposed area and increased thermal conductivity will lower the temperature fluctuation in the room. Properties like density, specific heat, and thermal conductivity of some building storage materials are listed in table 6.2. Water in container as a storage material is proved to be the best material to reduce the fluctuation in air temperature. This may be due to the fact that water gets heated by convection currents and mixed up and gives uniform temperature. Brick structure as a storage has also given good results (small temperature fluctuation) which could further be improved if brick has some magnesium additive. Since adobe has low conductivity, it has given largest temperature fluctuation. As a thumb rule for each 1 m^2 of glazing area a thermal storage mass to store 625 KJ per degree centigrade of temperature change is required.

Table 6.2 Thermal Properties of Storage Materials

Material	Density, ρ (Kg/m^3)	Specific heat Cp($KJ/Kg\,^\circ C$)	Thermal conductivity K($w/m\,^\circ C$)
Concrete	2240	0.84	1.70
Marble	2600	0.88	2.90
Limestone rock	2450	0.92	0.93
Brick(ordinary)	1920	0.84	0.72
Brick(Magensium added)	1920	0.84	3.80
Wood (pine)	496	2.80	0.16
Dry sand	1521	0.79	0.32
Adobe	1700	1.00	0.52
Water (Isothermal)	1000	4.19	0.59

Recently Wray[52] using the monthly solar load ratio (SLR) method analysed the performance of heavy mass direct gain buildings. A thermal storage mass of 920 KJ/per degree centigrade of temperature change for one m^2 of glazing area is assumed to be kept on the floor. For a typical building at Albuquerque, New Mexico, the solar savings fractions are calculated as a function of thickness and ratio of mass surface area to glazing area (Am/Ag = 2,3,6,10) and the same are shown[52] in figs.6.31 and 6.32 for no night insulation case and with night insulation case respectively. The reference design conditions i.e. thickness equal to 0.15m and Am/Ag equal to 3 are shown in these figures by small solid circle. From these figures it is seen that there is not much change in solar savings fraction after the mass thickness of 0.1 m. This is true irrespective of configuration, location, and mass surface area. In the thickness range of 0.05 m to 0.10 m which can be called the transition range, a slight reduction in thickness may effect the solar savings fraction considerably. The thickness below 0.05m is not recommended because here the solar savings fraction is very low and it falls very rapidly with the decrease in thickness.

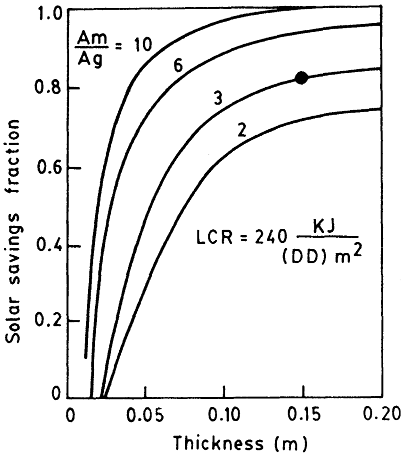


FIG.6.31 SOLAR SAVINGS FRACTION FOR DIFFERENT THICKNESS AND RATIO OF STORAGE SURFACE AREA TO GLAZING AREA IN CASE OF NO NIGHT INSULATION FOR ALBUQUERQUE (From Wray[52])

Apart from the thickness of the storage mass, its thermophysical properties like density ρ , capacity C, and conductivity K also affect the solar savings fraction considerably. Fig. 6.33 shows the effect of ρCK and LCR on the solar savings fraction in case of night insulation (R-9

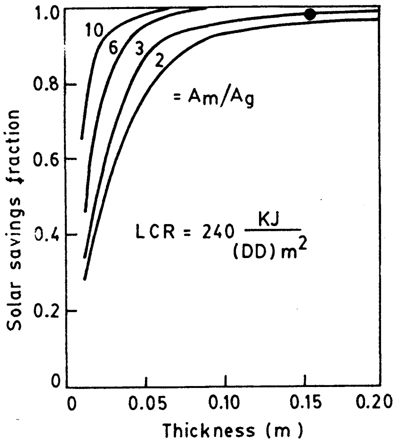


FIG.6.32 SOLAR SAVINGS FRACTION FOR DIFFERENT THICKNESS AND RATIO OF STORAGE SURFACE AREA TO GLAZING AREA IN CASE OF NIGHT INSULATION (R-9) FOR ALBUQUERQUE (From Wray[52])

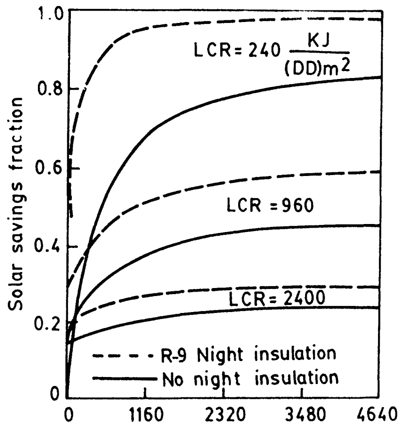


FIG.6.33 EFFECT OF ρCK AND LCR ON SOLAR SAVINGS FRACTION FOR ALBUQUERQUE (From Wray[52])

value) and no night insulation. From this figure it is seen that by increasing the value of ρCK beyond 2320, there is no gain in solar savings fraction. A value of ρCK below 1160 should not be used since in such cases, the solar savings fraction becomes very low. Values of ρCK between 1160 and 2320 is recommended. However, in case where low value of

ρCK is to be used such as in case of adobe walls, the solar savings fraction can be increased by using large ratio of mass surface area to glazing area (Am/Ag > 3).

6.5.2. Thermal storage wall system

The principle of thermal storage wall for space heating in winter and for drawing cool air through the house in hot weather was invented and patented by E.L.Morse[1] in USA in 1881 and was later developed and used by F.Trombe[4-6] and his collaborator J.Michel of France in 1956 and therefore the solid thermal storage wall is generally named as 'Trombe wall'. The thermal storage wall may either consist of cans or drums of water stacked to form a wall known as water wall; or solid wall made of brick or concrete, or stone, or adobe or phase change storage materials (PCMs) without any thermo circulation ; or Trombe wall in which case wall of suitable thickness is made of brick or concrete or PCMs with vents for thermo circulation. In all these cases the surface(facing south) is painted black and suitable glazing with some air space between wall and glazing is provided. Sometimes glazed windows are provided in the thermal storage wall itself to allow light into the room which also provide some direct gain. The analysis of thermal storage wall was first presented by Balcomb, Hedstrom and McFarland[26] and the same was used by Perry[53] who derived useful results. Like that of direct gain system, here also the conductivity plays a significant role. Perry[53] did parametric studies and determined the effect of various parameters on the annual solar heating fraction. The effect of storage heat capacity and number of glazings with and without night insulation on the annual solar heating fraction is shown[53] in fig. 6.34. From this figure it is seen that after 195 $KJ/m^2\ ^{o}C$ of storage heat capacity there is no gain in annual solar heating fraction. A single glazing without any night insulation provides only 30 percent annual solar heating fraction even at a very high storage heat capacity and therefore it is not recommended. Single glazing with night insulation improves the performance considerably and is even higher then the double glazing without any night insulation. By providing night insulation in case of double glazing, the performance can be improved, but the improvement is not so significant as in case of single glazing with night insulation. In any case night insulation improves the performance. The effect of ratio of glass area to building load on the annual solar load fraction is shown[53], in fig.6.35. It is obvious that if the building load is increased by a factor of two by using poor insulation, the glass area and the storage wall area is also to be increased by a factor of two to have the same annual solar heating fraction. Here also there is a significant

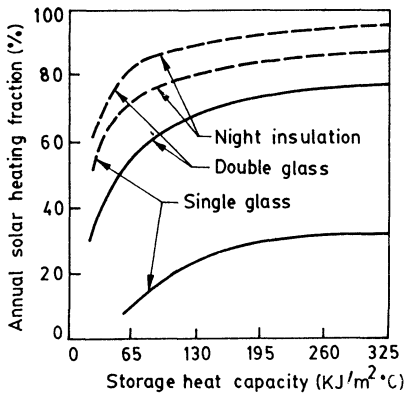


FIG.6.34 EFFECT OF STORAGE CAPACITY (KJ/m^2 °C), NIGHT INSULATION, AND NUMBER OF GLAZINGS ON THE ANNUAL SOLAR HEATING FRACTION IN CASE OF WATER WALL (From Perry[53])

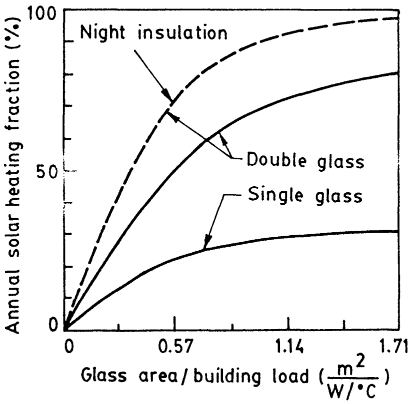


FIG.6.35 EFFECT OF BUILDING LOAD (W/ °C) GLASS AREA (m^2), NUMBER OF GLAZINGS AND NIGHT INSULATION ON THE ANNUAL SOLAR HEATING FRACTION IN CASE OF WATER WALL (From Perry[53])

difference in the annual solar heating fraction by using a single or double glass with and without insulation. The effect of wall thickness and wall conductivity on the annual solar load fraction is shown[53] in fig.6.36. It is seen

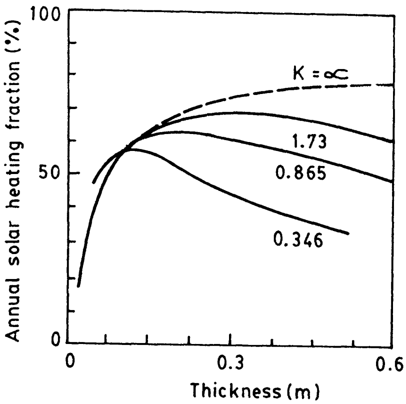


FIG.6.36 EFFECT OF WALL THICKNESS (m) AND ITS CONDUCTIVITY (W/m °C) ON THE ANNUAL SOLAR HEATING FRACTION. THE HEAT CAPACITY WAS KEPT CONSTANT EQUAL TO 190 KJ/m^2 °C (From Perry[56])

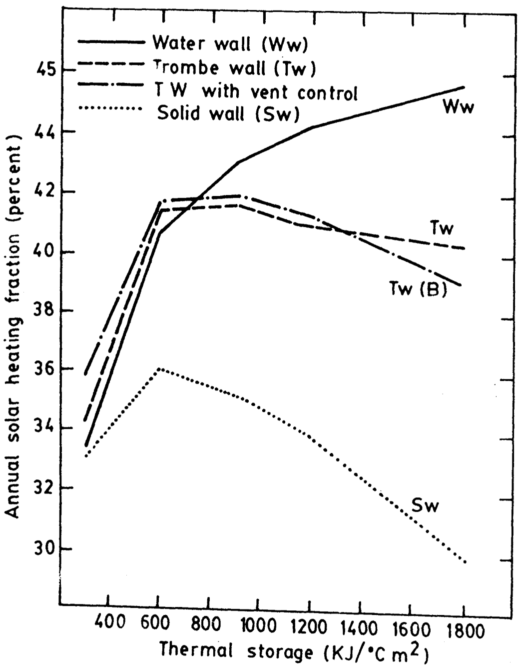


FIG.6.37 EFFECT OF THERMAL STORAGE MASS ON THE ANNUAL SOLAR HEATING FRACTION FOR VARIOUS THERMAL STORAGE WALL TYPES

that as the thermal conductivity increases, the annual solar heating fraction increases. It is also seen that after 30 cm of wall thickness there is no gain in annual solar heating fraction. In case of concrete wall of 30 cm thick the annual solar heating fraction is 68 percent while for the same storage capacity water wall, the annual solar heating fraction is 73 percent. The effect of storage mass per unit of glazing area in case of water wall(Ww), Trombe wall with no reverse flow(Tw), Trombe wall with thermostatic vent control (Tw(B)), and solid wall on the annual solar heating fraction is shown[53] in fig.6.37. From this figure it is seen that water wall gives the highest annual solar heating fraction amongst all the four walls. Trombe wall shows significantly high annual solar heating fraction compared to solid wall. The maximum annual solar heating fraction in case of trombe wall is obtained when a storage of about 600 KJ/ $^{\circ}$C per m^2 of glazing area is provided.

The area of the thermal storage wall should be equal to the area of the glazing and should be as close facing south as possible. The glazing selected should be able to transmit maximum solar radiation. The colour of the wall receiving solar radiation should be dark may be black or dark red or green. Somekind of overhang of suitable dimension on the outer side for shading the glazing in summers should be provided. The wall thickness of a particular material should be so selected so that it delays the heat entry to the other side of the wall by 8-10 hours.

6.5.3 Attached greenhouse system

As the name suggests, it is an integration of direct gain and indirect gain system. In the direct gain unit known as greenhouse (zone I, fig.6.4),the south, east and west wall and roof are made up of glass sheet (single, double or sometimes triple) and the north wall is a thermal storage wall made up of brick or concrete. The living room which is zone II (fig.6.4) where temperature swing is reduced considerably due to the thermal storage wall on its south side has all other features as of any other room. The greenhouse may have single or multiple glazings and depends on the ambient temperature. If outside (ambient) air temperature rarely goes below 0 $^{\circ}$C then single glazing is adequate. But if the ambient temperature goes very low then double and even triple glazing can be used. Generally double glazing green house is preferred and if the climate is very bad then double glazing with movable insulation is used. In case of multiple glazings, light gets considerably reduced and therefore, in such cases high quality glass or plastic films are used. Generally the temperature swings in the green house are quite large, typically 20 to 25 $^{\circ}$C during winter days, and the excess heat can be either stored directly by placing water drums or rocks in the greenhouse or by blowing air in the greenhouse and passing the hot air into the rock bed storage system in the basement and thus storing the excess heat.

Since the living space gets heated by convection and radiation heat loss directly from the thermal storage wall, the material and its size is very important. Direct air connection between sunspace and living space is not provided. Recently Sodha et al[54] have evaluated the thermal performance of solarium(attached greenhouse system) using the periodic solution technique and predicted the temperature of air in the sunspace and the thermal flux from the south thermal wall into the room as a function of time for climatic conditions of Boulder, Colorado, USA. The results are summarized[52] in table 6.3. From this table it is seen that if the thickness of the storage wall increases, the temperature in zone I increases. It is also seen that if the area of the storage wall is increased by changing its width only and thereby increasing the floor area, the increase in temperature is relatively smaller than in the case when the area of the storage wall is increased by changing its height. From table 6.3 it is also seen that as the thermal storage wall thickness decreases, the average heat flux entering the living space increases and the corresponding swings in the heat flux also increases. For a wall thickness of 0.45m a small fluctuation in heat flux occurs resulting in good load levelling. The use of movable insulation also enhances the thermal performance.

6.5.4 Thermal storage roof system

The roof pond or skytherm system pioneered by Hay[11,44] and Hay and Yellot[12,55] is a successful passive heating and cooling method. As discussed earlier, this system is based on the absorption and storage of heat in water during day time (for heating) and noctural radiation heat loss during night time (for cooling). Further cooling can be provided by evaporative cooling using water sprinkled on water bags during summer. This system can provide comfortable temperatures all the year round at places where the diurnal temperature variation is large and humidity is low. In the heating mode in winter months, the ponds are exposed to sun during daytime to absorb solar heat and are covered by movable insulation at night so that the heat is radiated to the rooms below through metallic roof. In the cooling mode in summer months, the pond is exposed to sky during night time dissipating heat to the air and sky and covered by movable insulation during daytime to avoid heat gain from sun. Here the cooled water absorbs room heat providing radiant cooling to the rooms. Roof ponds have many advantages such as it can be used both for heating and

Table 6.3 Temperature in sunspace and heat flux in living room in case of attached greenhouse system (From Sodha et al[54])

Isothermal mass(kg)	Wall area (m^2)			Thickness of mass wall(m)	Temperature of zone I (°C)			Heat flux into zone 2 (w/m^2)		
	South	Floor	East/West		Max.	Min.	Average	Max.	Min.	Average
0.0	9	9	9	0.15	54.0	17.7	35.9	206.4	29.0	100.6
0.0	9	9	9	0.30	56.1	20.6	39.0	94.8	53.3	73.5
0.0	9	9	9	0.45	57.5	22.1	40.8	63.1	51.9	58.0
232	9	9	9	0.30	56.0	20.8	39.0	94.8	51.7	73.5
0.0	13.5	13.5	9	0.30	55.1	20.2	39.8	95.7	53.2	74.7
0.0	13.5	9	13.5	0.30	59.8	21.4	40.3	97.1	53.3	75.9

cooling, little swing in room air temperature and no building orientation constraints. There are many problems also with the roof pond system such as: suitable to one storey structure or at the most two storey structure only; roof configuration should be specially designed for this purpose, movable insulation requires planned storage space for the retracted insulation panels, requires that all the rooms are radiatively coupled to the roof pond via the ceiling, effectivenss of concept of the weathering of plastic bags which generally get deteriorated, corrosion of metallic roof effects the lifetime, accumulation of dust which is difficult to remove, effects performance, susceptibility of plastic bags against wind, etc. Apart from climatic conditions, the heating and cooling by roof pond system depends on many paramters like properties of movable top insulation, thickness of air, absorption and transmission properties of plastic bag, water depth in bags, properties of steel ceiling, air gap between bags and steel ceiling, movable bottom insulation, etc. All these components should be carefully coordinated to get results. Analysis of roof pond system is carried out by Niles[56]; Clark and Allen[57]; and Tavana et al[58]. Tavena et al[59] have developed a simulation model using BLAST(Building Loads Analysis and System Thermodynamics) computer programme and used it for studying the effect of movable top insulation, water depth movable bottom insulation and evaporative layer of water for heating and cooling potentials of roof ponds for four places viz. Washington, Atlanta, Sacramento, and Phoenix. Here the performance of roof pond system is expressed in terms of hourly heat flow rates per unit of deck

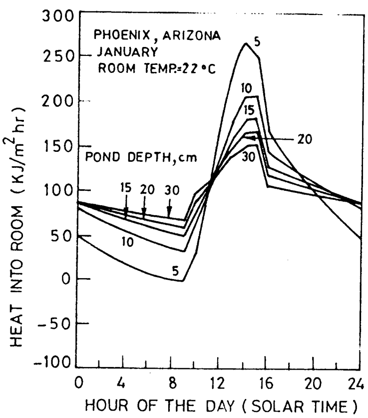


FIG.6.38 EFFECT OF POND DEPTH ON THE HEAT GAIN INTO THE ROOM (From Tavena[58])

area into the room (KJ/hr m^2. The effect of depth of water in roof pond for heating the room in the month of January at Phoenix is shown[58] in fig.6.38. From this figure it is seen that if the pond depth is increased from 5 cm to 15 cm, the daily heat gains into the room increases by about 15 percent. If the pond depth increases beyond 15 cm, there is no net gain of heat in the room. But as the pond depth increases the range of variation of heat flux into or out of the room decreases. The effect of top insulation thickness (K = 0.09 KJ/m hr °C) on the hourly heat flow rates per unit of deck area into the room (KJ/hr m^2) is shown[58] in fig.6.39. From this figure it is seen that as the top

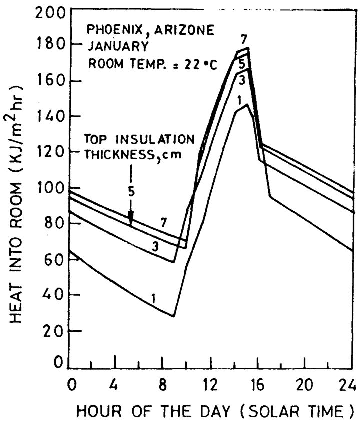


FIG.6.39 EFFECT OF TOP INSULATION THICKNESS ON THE HEAT GAIN INTO THE ROOM (From Tavena[58])

insulation thickness increases, the heat flow into the room increases. It is also seen that after an insulation thickness of 3.0 cm there is not much increase in the heat flow into the room. Thus the top insulation is critical to the roof pond performance while the bottom insulation(below the metal ceiling) has not shown any effect on the performance. In the cooling mode also the increase in thickness of top insulation increases the cooling performance. Increasing the water pond depth improves the daily cooling performance also but the effect is small. But a deeper pond may provide more stable comfort conditions. The effect of flooding the roof pond for providing evaporative cooling was also studied for Phoenix in August. It is concluded that more than 10 times cooling can be provided by evaporative cooling (flooding of water) compared to simple roof pond system.

6.5.5 Convective Loop System

The convective loop passive heating system uses some kind of solar air heater installed at a low level compared to the floor of the house and a rock bed storage system which is located between the levels of collector and floor of the house. The air flow from the house to the storage unit to the air heater and from the air heater to the storage unit and then to the house is by natural means known as thermosyphon action. Only at very few places away from the densely built areas, such sites where collectors can be placed below the level of the house are available. Only in rural areas or suburban areas, the prospects of convective loop systems are better. One of the most important requirement of the system is low flow resistance through the storage and the air heater otherwise the stack forces may prove inadequate and also the air heater collectors should be placed below the level of the house. To overcome the flow problems, generally forced flow circulation (active distribution) in the hybrid passive heating system utilizing solar air heaters and storage unit along with other passive features in the house is employed. With this hybrid passive/active air thermosyphon system comfortable temperatures between 18-21 $^{\circ}$C have been obtained in heating season in many buildings in USA.

In all these natural convection rock-storage systems, a reverse circulation during the night occurs which can be avoided using suitable dampers. To avoid overheating during summers, vents should be provided at a higher level with a cross-sectional area of atleast half the total collector channel cross-section. The vents can be provided on the down-wind side of the system or on both the sides. If vents are not provided, then solar collectors should be covered.

6.6 PASSIVE SYSTEM PERFORMANCE PREDICTION

In recent years several mathematical techniques are developed for predicting and optimizing the passive houses. The first step in this direction is the prediction of heating load of a building which depends on local climatic conditions, solar intensity, building type and size, thermophysical properties of materials used, degree of building use, and infiltration loss. A simple steady state method for predicting heating load of a building is described in chapter1, section 1.8 of this book. Some of the methods or models used for predicting the passive heating building performances are given below.

6.6.1 Degree-Day method

The degree-day method is used all over the world by architect and engineers for finding out the annual fuel consumption of buildings. This degree day method first fixes an indoor design temperature T_b which is independent of building type. This design indoor air temperature for united states is 18.3° C (65° F) and for U.K.15.6° C (60° F). The outdoor design temperature is not the lowest minimum temperature which will unnecessary increase the heating load but is taken as the hourly minimum temperature which are exceeded by 95 percent of the time. If long term outdoor weather (air temperature) data is not available then outside design air temperature can be taken as the mean outdoor air temperature. Since the heat loss is proportional to the indoor to outdoor air temperature difference and the degree day is also considering this difference, hence degree day method can be used for heat load calculations. If the indoor air base temperature is 18.3° C and the design outdoor air temperature is 3° C, then the temperature difference is 15.3° C. If this summation of daily average temperature differences over a month is done then this total degrees difference is called the degree-days for the month. Thus the number of degree days for a given day is[65]

$$d = \begin{cases} 0 & \text{if } T_b > T \\ T_b - T & \text{if } T_b < T \end{cases} \qquad (6.1)$$

where T is the design indoor temperature taken as 18.3° C and T is the mean of outside air temperature

$$T = 0.5 \ (T_{max} + T_{min}) \qquad (6.2)$$

Where T_{max} and T_{min} are the maximum and minimum temperature of the outdoor air respectively. For a month or a season of N days, the number of degree days (DD) is given as

$$(DD) = \sum_{1}^{N} d \qquad (6.3)$$

Thus the monthly heating load (Q_s) is given as

$$(Q_s) = UA(DD) \qquad (6.4)$$

Where UA is the product of overall building heat loss coefficient and the area of the building. The heating degree days for various places are given in text books[66,67,68]. The value of UA can be calculated by several ways. For structures[66] whose thermophysical properties are known, UA can be calculated by procedure as described in ASHRAE handbook of fundamentals. For existing buildings, where a record of conventional fuel consumption is kept, UA can be written as:

$$UA = \frac{N_F H_F \eta_F}{(DD)} \tag{6.5}$$

Where N_F is the units of fuel consumed, H_F is the heating value of the fuel consumed, η_F is the furnace efficiency. (UA) can also be calculated as

$$UA = \frac{\text{Design Heating load}}{\text{Design temperature difference}} \tag{6.6}$$

The step wise calculation procedure is given below:

(i) Select a suitable design outdoor and indoor air temperature from the local weather data. The indoor design air temperature may be taken as $18.3^\circ C$.

(ii) Calculate areas of walls, roof, floor, windows, and doors.

(iii) Calculate the total conduction, infiltration, and ventilation loss and sum it to get the total heating load of the building.

(iv) Express this as a design heat loss rate per degree day.

6.6.2 Steady State Method (From Sodha et al[37])

Steady state methods for the calculation of heat loss from buildings have been widely used[63]. These methods are approximates one and can give an idea of annual fuel consumption in a building. The steady state methods assume a constant outside/inside temperature difference and does not take into account the lag effect which is associated with massive construction under changing thermal conditions generally encountered in passive buildings. These estimates have been traditionally used for sizing the conventional heating systems. Moreover, the steady state procedure assumes that the temperatures at various points in the building do not change with time and therefore not suitable in buildings which are intermittenlly heated. However, a simple steady state procedure for predicting the performance of some passive features of building heating is described.

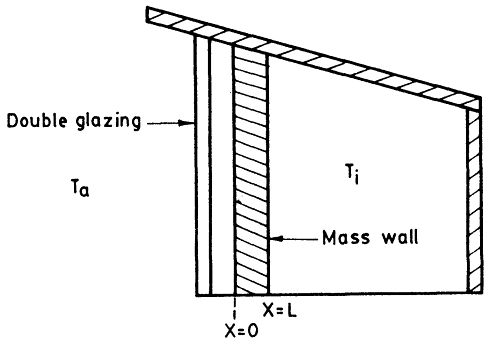


FIG.6.40 THERMAL MASS WALL

6.6.2.1 Thermal mass wall

For a thermal mass wall as shown in fig. 6.40 which can be considered as a Trombe wall without vents, the average heat flux entering through the mass wall into the room is given as:

$$\dot{Q} = \zeta\alpha H_T - h_{OS}\,(T_{X=0} - T_a) \qquad (6.7)$$

Where $\dot{Q}$ is the average heat flux entering the room, ζ and α are transmittance of glazing and absorptance of thermal mass wall respectively, H_T is the total radiation incident on the glazed surface, h_{OS} is the overall outside heat loss coefficient from outer surface of the mass wall to the outside air at temperature T_a, and $T_{X=0}$ is the outer surface temperature of the wall. Under steady state conditions this flux conducted goes into the room by convection and radiation and therefore the relevant expressions are:

$$\dot{Q} = \frac{K}{L}\,(T_{X=0} - T_{X=L}) \qquad (6.8)$$

and $$\dot{Q} = h_{is}\,(T_{X=L} - T_i) \qquad (6.9)$$

Where K is the thermal conductivity of wall material, L is thickness of mass wall, h_{is} is inside surface coefficient, $T_{X=L}$ is the inside surface temperature of mass wall, and T_i is the room air temperature at which the room is maintained. From equation (6.7) to (6.9), we get

$$\dot{Q} = U \left[T_a + \frac{\zeta\alpha H_T}{h_{os}} - T_i \right] \tag{6.10}$$

$$= U (T_{sa} - T_i) \tag{6.11}$$

Where U is the overall heat loss coefficient from room inside air to outside air and is given as

$$\frac{1}{U} = \frac{1}{h_{os}} + \frac{L}{K} + \frac{1}{h_{is}} \tag{6.12}$$

and T_{sa} is the sol-air temperature and is given as

$$T_{sa} = T_a + \frac{\alpha\zeta H_T}{h_{os}} \tag{6.13}$$

From equation (6.11) the average heat flux entering into the room through the thermal mass wall can be calculated by knowing the thermophysical properties of the material used in the wall and glazing, and also the climatic conditions.

6.6.2.2 Water Wall

Refering to fig.6.41 and assuming T_P as the average outer surface temperature (metallic drum) of the drum wall, the average heat flux can be written as:

$$\dot{Q} = \zeta\alpha H_T - h_{os} (T_P - T_a) \tag{6.14}$$

$$\dot{Q} = h_1' (T_P - T_W) \tag{6.15}$$

$$\dot{Q} = h_2' (T_w - T_{is}) \tag{6.16}$$

$$\text{and } \dot{Q} = h_{is} (T_{is} - T_i) \tag{6.17}$$

where h_1' and h_2' are the heat transfer coefficients between outer surface of water wall to water and from water to inside surface of the water wall respectively, T_W is the water temperature in the water wall, and T_{is} is the inside surface temperature of the water wall. From equations (6.14) to (6.17) we get

$$\dot{Q} = U (T_{sa} - T_i) \tag{6.18}$$

$$\text{where } \frac{1}{U} = \frac{1}{h_{os}} + \frac{1}{h_1'} + \frac{1}{h_2'} + \frac{1}{h_{is}} \tag{6.19}$$

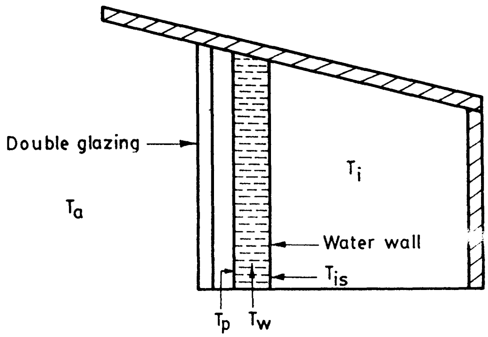


FIG.6.41 WATER WALL OR DRUM WALL

6.6.2.3. Attached Sunspace

Referring to fig. 6.42, where there is no direct heat transfer between absorbing surface and the ambient, the equations for average heat flux $\dot{Q}$ is given as:

$$\dot{Q} = \zeta\alpha H_T - h_{os} (T_o - T_a) \qquad (6.20)$$

$$\dot{Q} = \frac{K}{L}(T_{x=o} - T_{x=L}) \qquad (6.21)$$

and $$\dot{Q} = h_{is} (T_{x=L} - T_i) \qquad (6.22)$$

The temperature T_o in the sunspace is related with the ambient air temperature T_a as follows:

$$h_{os} (T_o - T_a) = h_{ws} (T_{x=o} - T_o) \qquad (6.23)$$

Where h_{ws} is the heat transfer coefficient from mass wall and the sunspace. From equation (6.23)

$$T_o = h_m \left(\frac{1}{h_{os}} T_{x=o} + \frac{1}{h_{ws}} T_a \right) \qquad (6.24)$$

where $$\frac{1}{h_m} = \left(\frac{1}{h_{os}} + \frac{1}{h_{ws}} \right) \qquad (6.25)$$

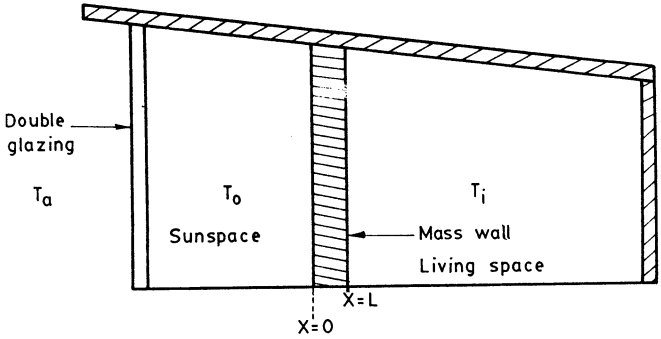


FIG.4.42 ATTACHED SUN-SPACE

with the help of above equations, the average heat flux is given as:

$$\dot{Q} = U \left[\frac{\zeta \alpha H_T}{h_m} + \frac{h_{os}}{h_m} \left(1 - \frac{h_m}{h_{ws}} \right) T_a - T_i \right] \qquad (6.26)$$

where $$\frac{1}{U} = \frac{1}{h_m} + \frac{L}{K} + \frac{1}{h_{is}} \qquad (6.27)$$

6.6.3 Solar Load Ratio (SLR) Design method

For estimating the thermal performance of collector storage wall (Trombe wall and water wall) passive heating systems, Balcomb and McFarland[69] devised a simple correlation method known as Solar Load Ratio (SLR) method. This simple method calculates the monthly performance for Trombe or water wall heating systems to an accuracy of 8 percent and the annual performance of the heating system to an accuracy of 3 percent as compared with hour by hour simulations. This correlation method is based on detailed hour by hour simulations of storage wall systems for 29 locations and for 6 building loads for each location and the results were correlated in terms of the ratio of absorbed solar energy to loads. This method is applicable to Trombe wall and water wall passive heating systems with and without night insulation with structure having characteristics as given[69] in table 6.4

The monthly solar load ratio (SLR) is defined as:

$$SLR = \frac{\text{Monthly solar energy absorbed by storage wall}}{\text{Monthly building load including storage wall}}$$

$$= \frac{\zeta\alpha\, \bar{H}_T N\, A_C}{(UA + A_C U_W)\,(DD)} \qquad (6.28)$$

where ζ = transmittance of glazing
α = absorptance of storage wall surface
$\bar{H}_T$ = Average daily total radiation intensity on glazing
N = Number of days in the month
A_C = Collecting area of storage wall
U = Building loss coefficient excluding storage wall
A = Building area excluding storage wall
U_W = loss coefficient of storage wall
(DD) = Number of heating degree days in a month.

Table 6.4 Parameters used in the development of SLR design method.

Temperature range in the room	$19^\circ C - 24^\circ C$
Storage Capacity	$0.92\ MJ/m^2\ ^\circ C$
Time of night insulation used	5.0 pm to 9.0 am
Resistance of night insulation (if used)	$1.6\ m^2\ ^\circ C/W$
Conductance from wall to room	$5.68\ W/m^2\ ^\circ C$
Thermal conductivity of storage wall	$1.73\ W/m\ ^\circ C$
Thermal capacity of storage wall (ρc)	$2.0\ MJ/m^3\ ^\circ C$
Double glazing	$\zeta = 0.747$, $L/K = 2.012$
Thermal mass wall has vents with backdraft dampers	

The values of heat loss coefficient U_W which includes thermal wall (Trombe on water wall), glazing, and insulation averaged over the day in case of water wall and Trombe wall (45 cm thick) are given[69] in table 6.5.
The equation (6.28) can also be written as

$$SLR = \frac{SCI}{MLCR} \tag{6.29}$$

Where SCI is known as solar capability index and depends only on the weather and given as

$$SCI = \zeta \bar{H}_T \, (DD) \tag{6.30}$$

and MLCR is known as modified load collector ratio and depends only on the building structure and given as

$$MLCR = \frac{(UA + A_C U_W)}{\alpha A_C} \tag{6.31}$$

Table 6.5 Average values of loss coefficient U_W of collector storage wall used in SLR method.

Wall type	Heat loss coefficient U_W ($w/m^2\,^\circ C$)	
	Double glazed no night insulation	Double glazed with R9 night insulation ($R = 1.6\ m^2\,^\circ C/W$)
Water wall	1.87	1.02
Trombe wall (45cm)	1.25	0.68

Thus in this method first of all building loss coefficient including collector storage wall is determined and then by using equation (6.28), the monthly value of SLR is determined. Now using fig. 6.43, monthly solar heating fraction SHF is determined corresponding to the value of SLR. The solar heating fraction SHF is given as:

$$SHF = 1 - \frac{\text{monthly auxiliary energy used}}{\text{monthly heating load}} \tag{6.32}$$

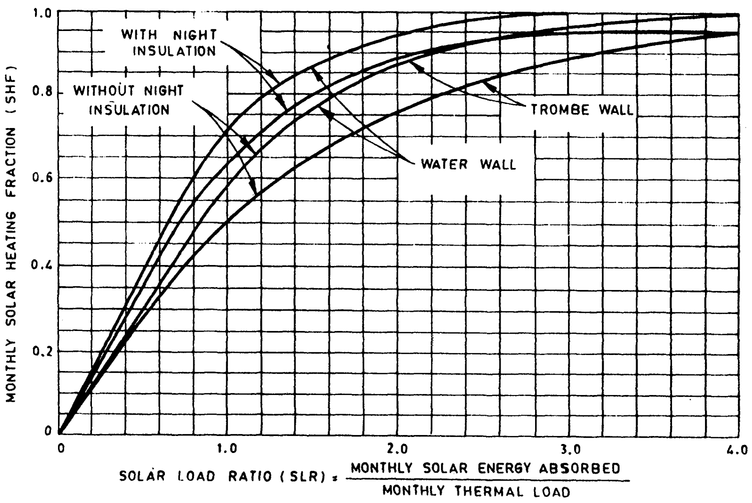


FIG.6.43 A CORRELATION BETWEEN MONTHLY SOLAR LOAD RATIO AND MONTHLY SOLAR HEATING FRACTION FOR COLLECTOR STORAGE WALL SYSTEMS (From Balcomb and Mc Farland [69])

Therefore the monthly auxiliary energy used is (1-SHF)x (load). The annual auxiliary energy requirement will be the sum of monthly auxiliary energy requirements.

The curves of fig.6.43 can be mathematically put in the following forms[69]:

$$SHF = A\ (SLR) \quad \text{for } SLR \leq E \tag{6.33}$$

$$SHF = B - C \exp[-D\ (SLR)] \quad \text{for } SLR \geq E \tag{6.34}$$

The values of A,B,C,D and E are given in table 6.6.

As discussed earlier the above method is based on large number of hour by hour calculations for several building types under a variety of climatic conditions. However, one should be careful in using the above method since it will provide accurate results only when it is applied to the appropriate types of system.

The SLR method requires calculation of monthly solar heating fraction for each heating month of the year before

Table 6.6 Correlation Coefficients used in equation 6.33 and 6.34

Storage wall type	A	B	C	D	E
Trombe wall without night insulation	0.4520	1.0137	1.0392	0.7047	0.1
Trombe wall with night insulation	0.7197	1.0074	1.1195	1.0948	0.5
Water wall without night insulation	0.5995	1.0149	1.2600	1.0701	0.8
Water wall with night insulation	0.7642	1.0102	1.4027	1.5461	0.7

finding out the yearly solar heating fraction or yearly auxiliary heating requirement. This requires a set of calculations to be made. An annual load collector Ratio (LCR) method is developed by the Scientists of Los Alamos Scientific Laboratory to determine the annual performance of the collector storage wall heating systems using monthly values of SLR. Balcomb and McFarland[69,70] have defined LCR as:

$$\mathrm{LCR} = \frac{\text{Building loss coefficient(KWH/DD)}}{\text{Solar Wall collector area}} \tag{6.35}$$

Thus for a particular location and building monthly values of SLR are determined and from which LCR is determined. A set of curves are given by Balcomb et al[32] relating SLR, LCR and monthly solar savings fraction. Balcomb and McFarland[69] have also prepared tables for finding out annual solar heating fraction for various values of LCR and for various cities and passive systems like Trombe wall, without night insulation, Trombe wall with night insulation, water wall without night insulation, water wall with night insulation, and Direct gain with and without night insulation.

6.6.4 The Un-Utilizability Design Method

The Solar Load Ratio (SLR) method is limited in scope since it can not be used to study the effect of number of glazings, R-Values, night insulation, building thermal storage capacity, solar absorptance, and different room set temperatures. The correlation method developed by Monsen et al[71,72] is based on the solar radiation statistic utilizability and is known as un-utilizability method which requires somewhat more computations than the Solar Load Ratio method but covers a much larger range of design parameters. The difference between the yearly solar load fractions computed from the TRNSYS[73] simulation method and the above un-utilizability method was found to be less than 4 percent.

In this method of un-utilizability, the monthly average auxiliary (Q_{aux}) energy requirement of building using collector storage (Trombe) wall heating system is estimated using upper and lower theoretical limits to system performance. The collector storage wall is assumed to be a part of the building with an allowable room temperature swings of 6 ^{o}C. Two assumptions are made which simplify the problem considerably. According to the first assumption the annual auxiliary energy requirement of a vented and unvented Trombe wall is nearly the same. This assumption is confirmed by Utzinger et al[74,75]. According to the second assumption the thermal storage capacity of a collector-storage wall over a month is negligible compared to the total energy flow through the wall over a month which results in linear variation of monthly average temperature profile through the wall on a monthly basis. Out of the energy absorbed by the collector-storage wall, a part gets lost from wall water surface through glazing to outside air and the rest goes in the room through the wall.

$$\overline{H}_T\ (\overline{\zeta\alpha}) = \overline{U}_L\ (\overline{T}_w - \overline{T}_a)\Delta t + U_k(\overline{T}_w - \overline{T}_i)\ \Delta t \tag{6.36}$$

Where $\overline{H}_T$ is the monthly average daily radiation incident on the glazing, $(\overline{\zeta\alpha})$ is the monthly average transmittance-absorptance product of glazing-wall combination, $\overline{T}_w$ is the monthly average outer wall surface temperature, $\overline{T}_a$ is the monthly average outside air temperature, $\overline{T}_i$ is the monthly average room air temperature, Δt is the number of seconds in a month, $\overline{U}_L$ is the monthly average loss coefficient(w/m^2 ^{o}C) from collector-storage wall outer surface at temperature $\overline{T}_w$ to outside air at temperature $\overline{T}_a$, and U_k is the conductance from the outer wall surface to the room air and is given as

$$U_k = \frac{h_{is}\ K}{K + h_{is}\ L} \tag{6.37}$$

Where h_{is} is tho ilwqts bspe eoplwcso uhsccquqsle cohn inside surface of wall to the room air, K is the thermal conductivity of wall, and L is the thickness of wall. Now from equation (6.36) the monthly average outer surface temperature of wall $\overline{T}_w$ is given as:

$$\overline{T}_w = \frac{\overline{H}_T\ (\overline{\zeta\alpha}) + (U_k T_i + \overline{U}_L \overline{T}_a)\Delta t}{(U_k + \overline{U}_L)\ \Delta t} \tag{6.38}$$

The energy going into the room(Q_{in}) through the collector storage wall is given as

$$Q_{in} = U_k A\ (\overline{T}_w - T_i)\ \Delta t\ N \tag{6.39}$$

Where A is the collector-storage wall area, and N is the number of days in the month. A stepwise procedure as described by Monsen et al[72] for finding out the solar fraction for collector-storage wall is given here:

Step 1: Calculate the monthly-average daily total radiation $\overline{H}_T$ incident on the glazing. This can be determined from standard equations described earlier[64]. The monthly average value of $(\overline{\zeta\alpha})$ can be determined by the method given by Klein[76].

Step 2: Calculate the monthly average building load, L_a, excluding the contribution of the collector storage wall using following equation:

$$L_a = (UA)_a\ (DD)_b \tag{6.40}$$

Where $(UA)_a$ is the overall conductance-area product for the building excluding the collector-storage wall, $(DD)_b$ is the monthly total degree days evaluated at the balance point temperature T_b which is $[T_i - \dot{g}/(UA)_a]$, and $\dot{g}$ is the rate of energy generation from appliances, occupants, lights etc. in the building. Now calculate L_w which is the monthly average energy loss from room to the outside through collector-storage wall when no solar radiation is absorbed by wall using following equation:

$$L_w = U_w\ A(DD) \tag{6.41}$$

Where DD is the monthly degree days at the base temperature T_i, and U_w is the average wall conductance from inside the room to outside air and is given as:

$$U = \frac{1}{\dfrac{1}{U_L} + \dfrac{1}{h_{is}} + \dfrac{L}{K}} \tag{6.42}$$

Step 3: Calculate Q_{in} the net monthly heat transfer through the wall using equation (6.39) by making use of equation (6.38) and (6.37).

Step 4: Calculate Q_{dump} which is the monthly total energy which would have to be removed from the building if there were no storage capacity in the system by using following equation:

$$Q_{dump} = \left(\frac{U_k\ A(\overline{\zeta\alpha})}{U_L + U_k}\right)\overline{H}_T N\ \overline{\Phi} \qquad (6.43)$$

Where $\overline{\Phi}$ is the monthly-average radiation utilizability function and which can be determined either by using graphs or empirical relations for a critical intensity I_c described earlier[64].

Step 5: Calculate the solar fraction F by calculating the values of F_i the upper limit of solar fraction when the building has infinite thermal capacity, and Y the storage dump ratio. The F_i is calculated using the following equation:

$$F_i = 1 - \frac{Q_{aux,\,i}}{L_a + L_w} = \left(\frac{L_w + Q_{in}}{L_a + L_w}\right) \qquad (6.44)$$

and Y is calculated as:

$$Y = \frac{S_b + 0.047\ S_w}{Q_{dump}} \qquad (6.45)$$

Where S_b is the storage capacity of building given as:

$$S_b = C_b\ (\Delta t_b)\ N \qquad (6.46)$$

and S_w is the storage capacity of wall given as:

$$S_w = \rho C_p\ LA\ (\Delta T_w) N \qquad (6.47)$$

Where C_b is the effective building storage capacity, ΔT_b is the allowable indoor air temperature swing, ρC_p is the product of density and specific heat of collector-storage wall, and ΔT_w is the monthly average temperature difference between the centre of the wall and the inside wall surface. Now the solar fraction F is calculated using the following equation:

$$F_i = \text{MIN}\ [PF_i + 0.88\ (1-P)\ (1-e^{1.26F_i}),\ 1.0)] \qquad (6.48)$$

where $P = (1 - e^{-0.144Y})^{0.53}$ (6.49)

The auxiliary energy used Q_{aux} is calculated from the following equation:

$$Q_{aux} = (L_a + L_w)\ (1-F) \qquad (6.50)$$

For finding out the annual solar fraction or annual auxiliary energy used all the above steps 1 to 5 are repeated for all the months.

The above method is examined by Monsen et al[72] for a large range of system parameters as shown in table 6.7.

Table 6.7 Range of parameters examined in Un-untilizability method.

Parameters	Range
Location in USA	Madison, Albuquerque, Caribou, Nashville, Ely, Seattle.
$(UA)_a$	83 - 667 w/°C
Collection-storage wall area	10-50 m^2
Night insulation 'R value'	1.59 m^2 °C/w
Number of glazings	1 - 3
$(\overline{\tau\alpha})$	0.5 - 0.8
(ρc_p)	0.5 - 4.0 MJ/m^3 °C
K	0.851 - 2.22 W/m^2 °C
L	0.1 - 0.9 m
U_k	2.08 - 8.33 W/m^2 °C
C_b	0.4 MJ/ C - 1.0 GJ/°C
Room temperature swing	0 - 20 °C
Low set point temperature	10 - 21 °C

6.6.5 The Admittance Design Method.

The admittance procedure was developed by the U.K.Building Research Development[77,78] for the determination of daily room air temperature fluctuations in non-conditioned buildings without the use of computers. In this method in place of U-values of materials, the thermal admittance values of materials which takes into account the dynamic response of these materials to heat flow are used. Since in this method all thermal masses within the room are lumped, therefore the change in room air temperature fluctuations caused due to sizing of thermal masses can not be accounted. The average room air temperature $\overline{T}_i$ using simple energy balance is given as:

$$\overline{T}_i = \frac{\overline{Q}}{\Sigma AU + C_V} \tag{4.51}$$

Where Σ AU is the sum of the product of heat loss coefficients and surface areas of roof and walls, C_V is the ventilation conductance, and $\overline{Q}$ is the average heat gain given as:

$$\overline{Q} = \overline{Q}_s + \overline{Q}_g + \overline{Q}_i \tag{4.52}$$

Where $\overline{Q}_s$ is the average solar gain through solid elements and is given as:

$$\overline{Q}_s = A_s U_w \overline{T}_{sa} \tag{4.53}$$

and $\overline{Q}_g$ is the average solar gain through solid elements and is given as:

$$\overline{Q}_g = \overline{H}_T f A_g \tag{6.54}$$

and $\overline{Q}_i$ is the average internal gains and is given as:

$$\overline{Q}_i = (U_g A_g + C_V) \overline{T}_a \tag{6.55}$$

Where A_s and A_g are the areas of solid element and glass respectively, U_w and U_g are the transmittance of solid and glass element respectively, $\overline{T}sa$ and $\overline{T}a$ are the average solair and ambient air temperature respectively, $\overline{H}_T$ is the 24 hour mean solar radiation on each surface, f is the solar gain factor, and C_V is the ventilation conductance which is given as

$$\frac{1}{C_V} = \frac{3}{NV} + \frac{1}{4.8 \Sigma A} \tag{6.56}$$

Where N is the number of air changes per hour, V is the volume of room, and ΣA is the total surface area of the room.

The swing of room air temperature about its mean value at any time t is given as

$$\tilde{T}_i(t) = \frac{\tilde{Q}(t)}{\Sigma AY + C_V} \qquad (6.57)$$

where ΣAY is the sum of the product of room admittances and surface areas, and $\tilde{Q}(t)$ is the cyclic heat gain at any time and is given as

$$\tilde{Q}(t) = \tilde{Q}_s(t) + \tilde{Q}_g(t) + \tilde{Q}_u + \tilde{Q}_{gc} \qquad (6.58)$$

The cyclic heat gain $\tilde{Q}_s(t)$ through solid surface is given as:

$$\tilde{Q}_s(t) = AUf_s\, d_f[Tsa(t-\varphi) - \overline{T}sa] \qquad (6.59)$$

The cyclic heat gain $\tilde{Q}_g(t)$ through glass is given as:

$$\tilde{Q}_g(t) = A_g f_s[H_T(t) - \overline{H}_T] \qquad (6.60)$$

The cyclic heat gain/loss $\tilde{Q}_u$ through ventilation is given as:

$$\tilde{Q}_u(t) = C_u[T_a(t) - \overline{T}_a] \qquad (6.61)$$

and the cyclic heat gain/loss $\tilde{Q}_{gc}$ by convection through glazed area is given as:

$$\tilde{Q}_{gc}(t) = A_g U_g\,[\,T\,(t) - \overline{T}_a\,] \qquad (6.62)$$

Where f_s is the cyclic solar gain factor, d_f is the decrement factor, φ is the time lag(seconds) (= $1.38\, L\sqrt{1/\alpha}$), L is the thickness of wall and α is the diffusivity of material. The actual value of room air temperature is determined by adding the average room air temperature $\overline{T}_i$ and the swing of room air temperature $\tilde{T}_i(t)$. The values of decrement factor and admittance for various building materials are given by Milbank and Harrington[77].

6.6.6. The Periodic Method (From Sodha et al[37])

Since solar radiation and ambient temperature changes with time in a periodic manner and the heat conduction through building elements (walls, roof) of a room take place under periodic condition, the net flow can be assumed one directional. The one dimensional heat flow equation can be written as:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \tag{6.63}$$

Where α ($= K/\rho C$) is the thermal diffusivity of building element. The solution of above heat conduction equation is:

$$T(x,t) = T_o(x) + \sum_n T_n \exp(inwt) \tag{6.64}$$

Now substituting equation (6.64) in equation (6.63) and equating the coefficients of equal powers of exp(inwt), one gets:

$$\frac{d^2 T_o}{dx^2} = 0 \tag{6.65}$$

$$\frac{d^2 T_n}{dx^2} = \frac{inw \rho c}{K} \tag{6.66}$$

Now the temperature at any distance x at time t from equations (6.65, 6.66 and 6.63) is given as:

$$T(x,t) = A_o + A_1 x + \sum_n [\lambda_n \exp(\alpha_n x) + \lambda_n \exp(-\alpha_n x)] \exp(inwt) \tag{6.67}$$

where $$\alpha_n = -(1 - i)\left(\frac{nw \rho c}{2K}\right)^{1/2} \tag{6.68}$$

The constants A_o, A_1, λ_n, λ'_n are determined from the appropriate boundary conditions. The solar intensity and ambient temperatures are also expressed as Fourier series with a frequency w as follows:

$$f(t) = A_o + R_e \sum^{\infty} A_n \exp(inw + \Phi_n)t \tag{6.69}$$

where $A_n = \sqrt{a_n^2 + b_n^2}$

and $$\Phi_n = \tan^{-1} \frac{b_n}{a_n} \tag{6.70}$$

These can also be expressed as

$$f(t) = \sum_{n=-\infty}^{\infty} C_n \exp(inwt) \tag{6.71}$$

$$\text{Where } C_n = \frac{a_n + ib_n}{2}, \; n < 0 \qquad (6.72)$$

$$= \frac{a_n - ib_n}{2}. \; n > 0$$

Here λ_n is the decrement factor and Φ_n is the angular displacement.

6.6.7 PASOLE Computer Program

The PASOLE computer program was developed by McFarland [79,80] at Los Alamos Laboratory, U.S.A. to perform simulation of passive buildings. The program utilizes a general thermal net work approach in which uniform temperature regions are interconnected, these being denoted by a node and there is a heat transfer between various nodes illustrating different temperature regions and each heat transfer is defined by a thermal conductance. To explain the method, one can consider a three node thermal net work as shown in fig.6.44 by defining.

K = Conductance (U.A) between nodes($w/m^{\circ}C$),

T_i = Temperature at node,

M_i = heat capacity of node i
(Sp. heat x density x volume = $J/^{\circ}C$),

S_i = heat source at node i,

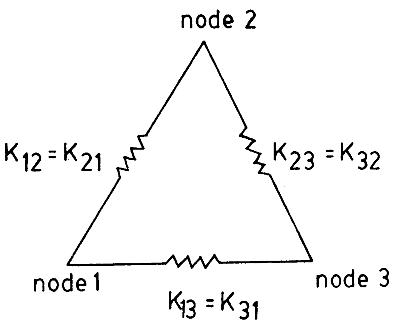


FIG.6.44 THREE-NODE THERMAL NETWORK DIAGRAM

t = time (seconds),
N = number of nodes,

At any instant, the rate of heat flow at node 1 is balanced and expressed as

$$K_{12}(T_1 - T_2) + K_{13}(T_1 - T_3) + M_1 \frac{dT_1}{dt} = S_1 \qquad (6.73)$$

Very often, the heat source is taken to be a linear function of temperature of the node representing it as

$$S_i = A_i + B_i T_i \qquad (6.74)$$

In generalised form, therefore, the heat balance equation at the node i can be written as

$$\sum_{j=1}^{N} K_{ij}(T_i - T_j) + M_i \frac{dT_i}{dt} = A_i + B_i T_i \qquad (6.75)$$

Where

K_{ij} = thermal conductance between the node i and j (w/ °C)

T_i = Temperature of the node i (°C)
M_i = Heat capacity of the node i, and
N = Number of the nodes.

In a building represented by the thermal net work mode, many of K_{ij} may be zero, essentially for j = i, K_{ij} = 0. Many nodes may have no heat source term i.e. $A_i = B_i = 0$ and many have essentially no capacity i.e. $m_i = 0$.

Expressing equation in small time interval Δt and integrating over the time interval from t^o to t, one obtains the node temperature change over time increment by t i.e.

$$T_i - T_i^o = \int_{t^o}^{t} \frac{A_i + B_i T_i'}{M_i} dt + \sum_{j=1}^{n} \int_{t^o}^{t} \frac{K_{ij}}{M_i}(T_j - T_i)dt \qquad (6.76)$$

where the superscript zero denotes condition at the old time i.e. $t^o = t - \Delta t$.

To perform the integral, the integrand is assumed to be a linear function of their values at times t^o and t. If F is a general integrand.

$$\int_{t}^{t_o} F\,dt = \overline{F}\,\Delta t \qquad (6.77)$$

where $\overline{F} = \beta F + (1-p) F^{o}$; β may be varied in the program and may take any values between zero and unity; usually a value of 0.5 is used for β since it gives most accurate results and results in stable solutions.

Some nodes in the thermal net work may have fixed or known temperatures at a given time such as ambient temperature. The other nodes may have unknown temperatures. If N be the total number of nodes out of which NV are the unknown temperatures and NF the fixed or known temperatures, then substituting Eq(6.77) into Eq.(6.76) and rearrangement leads to the following set of linear simultaneous algebric equations i.e.

$$\sum_{j=1}^{NV} a_{ij} T_j = b_i, \; i = 1, NV \qquad (6.78)$$

where

$$a_{ij} = -\beta K_{ij} \quad \text{for } j \neq i$$

$$a_{ij} = \frac{M_i}{\Delta t} + \beta \left(\sum_{j=1}^{N} K_{ij} - B_i \right) \text{ for } j = i$$

and

$$b_i = M_i \frac{T_i^{o}}{\Delta t} + \beta \left(A_i + \sum_{j=1}^{NF} K_{ij} T_j \right)$$

$$+ (1-\beta) \left(\frac{M_i}{M_i^{o}} \right) [A_i^{o} + B_i^{o} T_i^{o}]$$

$$+ \sum_{j=1}^{N} K_{ij}^{o} (T_j^{o} - T_i^{o}) \qquad (6.79)$$

For $M_i = M_i^{o} = 0$, b_i reduces to

$$b_i = \left(A_i + \sum_{j=1}^{N} K_{ij} T_j \right) \qquad (6.80)$$

and β cancels out of the equation.

The set of linear algebric equations given by Eq.(6.78) can be solved by using standard techniques such as the matrix inversion method explained by Williams[36] or by using standard library subroutines.

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