Solar energy: state of the art

Furbo, Simon; Shah, Louise Jivan; Jordan, Ulrike

Publication date:
2003

Document Version
Publisher's PDF, also known as Version of record

Link back to DTU Orbit

Citation (APA):
Furbo, S., Shah, L. J., & Jordan, U. (2003). Solar energy: state of the art. (BYG Sagsrapport; No. SR 03-14).

BYG·DTU

Editors:
Simon Furbo
Louise Jivan Shah
Ulrike Jordan

Solar Energy
State of the art

DANMARKS
TEKNISKE
UNIVERSITET

Internal Report
BYG·DTU SR-03-14
2003
ISSN 1601 - 8605

Solar Energy
State of the art

Editors:
Simon Furbo
Louise Jivan Shah
Ulrike Jordan

Nordisk Energiforskning

Department of Civil Engineering
DTU-bygning 118
2800 Kgs. Lyngby
http://www.byg.dtu.dk

2003

PREFACE

In June 2003 the Ph.D. course Solar Heating was carried out at Department of Civil Engineering, Technical University of Denmark.

The participants worked out state-of-the-art reports on 6 selected solar topics.

This report is a collection of the reports.

The course was sponsored by Nordic Energy Research.

July 2003

CONTENTS

Solar Combisystems

Helena Gajbert, Lund Technical University, Sweden & Frank Fiedler, Solar Energy Research Center SERC, Högskolan Dalarna, Sweden…….. Page 1-25

Thermal Storage for Small Solar Heating Systems

Søren Knudsen, Department of Civil Engineering, Technical University of Denmark, Denmark & Mike Dennis, Centre for Sustainable Energy Systems, Australian National University, Australia………………………………………………………...... Page 1-25

Solar Collector Materials

M. Reza Nejati, Energy Systems Engineering Group, Department of Mechanical Engineering, K.N. Toosi University of Technology, Iran & V. Fathollahi, Center for Renewable Energy Research and Application CRERA, Iran…………………………………………... Page 1-10

Solar Cooling

Wimolsiri Pridasawas, Royal Institute of Technology, Sweden and Teclemariam Nemariam, Royal Institute of Technology, Sweden…. Page 1-31

Solar Thermal Technologies for Seawater Desalination

Jenny Lindblom, Renewable Energy Systems, Luleå University of Technology, Sweden……………………………………………... Page 1-17

Review of the State of the Art: Solar Radiation Measurement and Modeling

Pongsak Chaisuparasmikul, College of Architecture, Illinois Institute of Technology, USA & Thomas Bache Andersen, Department of Civil Engineering, Technical University of Denmark, Denmark……………………………………………………….. Page 1-19

Ph.D. Course SOLAR HEATING

DEPARTMENT OF CIVIL ENGINEERING
TECHNICAL UNIVERSITY OF DENMARK (DTU)

SOLAR COMBISYSTEMS

A STATE OF THE ART REPORT

HELENA GAJBERT & FRANK FIEDLER

Lund Technical University
Solar Energy Research Center SERC, Högskolan Dalarna

JULY 04, 2003

Nordisk Energiforskning

TABLE OF CONTENTS

1 Introduction 3

2 Background 4

3 Latest research on improvement of combisystems 8
  3.1 Improvements from Task 26 8
  3.3 New storage medium 10
  3.4 Improved storage insulation 11
  3.5 Collectors adapted for seasonal load 11
  3.6 Compact systems 14
  3.7 Advanced control strategy for solar combisystems 16

4 Evaluation methods for combisystems 18
  4.1 The CTSS method 18
  4.2 The DC test method 18
  4.3 The CCT method 20
  4.4 The characterization tool FSC 20
  4.5 Calculation model for assessment of reliability in solar combisystems 21

5 Summary 23

Literature 24

1 Introduction

Due to the increasing concentration of greenhouse gases and climate changes, the need for renewable energy sources is greater than ever. This has now attracted attention from the European Commission that has set up targets to increase the share of renewable energy supply in Europe in order to reduce greenhouse gas emissions. By using solar thermal systems, a significant part of the space heating and hot water demand can be covered. In order to increase the market for solar systems, technical improvements of the systems has to be made, so that the systems can be more cost efficient. By using solar combisystems that deliver heat for both hot water and for space heating, a greater part of the total energy demand can be covered by solar energy. The share of solar combisystems has increased lately in many countries. In the middle and northern European countries the heat demand for space heating is still significantly higher than for hot water only. Therefore, the use of combisystems could significantly reduce the use of fossil fuel in these countries.

Within this report the newest developments of solar combisystems in terms of technical improvements and trends are investigated and discussed in order to provide a general overview of the situation of solar combisystems today. As sources for this investigation served articles from the Solar World Congress 2003 in Gothenburg as well as reports from the IEA task 26 “Solar Combisystems”. Solar combisystems in the context of this report are systems for delivering hot water and space heating for residential buildings, mostly for detached houses built for one or two families. Larger systems e.g. systems with seasonal heat storage or combination of heat pumps and solar collectors are not investigated. Within the framework of task 26 typical industrial made solar combisystems in the participant countries have been identified and systematically investigated in terms of their design and performance.

2 Background

Today about 82% of the world's primary-energy requirements are covered by coal, natural gas, oil and uranium. Approximately 12% comes from biomass and 6% from hydroelectric power. A reduction of greenhouse gases throughout the world of about 50 % is required in the next 50-100 years, according to many experts. In order to achieve this, a reduction of greenhouse gas emissions of approximately 90% per capita in the industrial countries, will be necessary. If we shall be able to change our energy supply system and reduce greenhouse gases, we need to use renewable energy sources, and solar energy is one of the most environmentally safe energy sources. The European Commission has set up goals, described in the White Paper "Energy for the future, renewable sources of energy", to increase the market share of renewable energy sources from 6%, as it is today, to 12% by the year 2010 [17]. The energy consumption in the building sector represents around 40% of all end energy consumption in the EU, of which 75% is required for hot water and space heating. Until 2000 only around 0.11% of the total requirement for hot water and space heating are covered by solar thermal systems. According to the White Paper, this share should be increased to 1.18% until 2010 [16]. Experts have estimated that an annual increase of 20% of installed collector area will be necessary in order to accomplish this goal. Today the installed collector area in the EU is approximately 18 million m^2 (of which approximately 7 million m^2 are collectors in combisystems) and by the year 2010 100 million m^2 would have to be installed (20 million m^2 for combisystems). In order to accomplish these goals, the solar heating technologies need to be further developed [17].

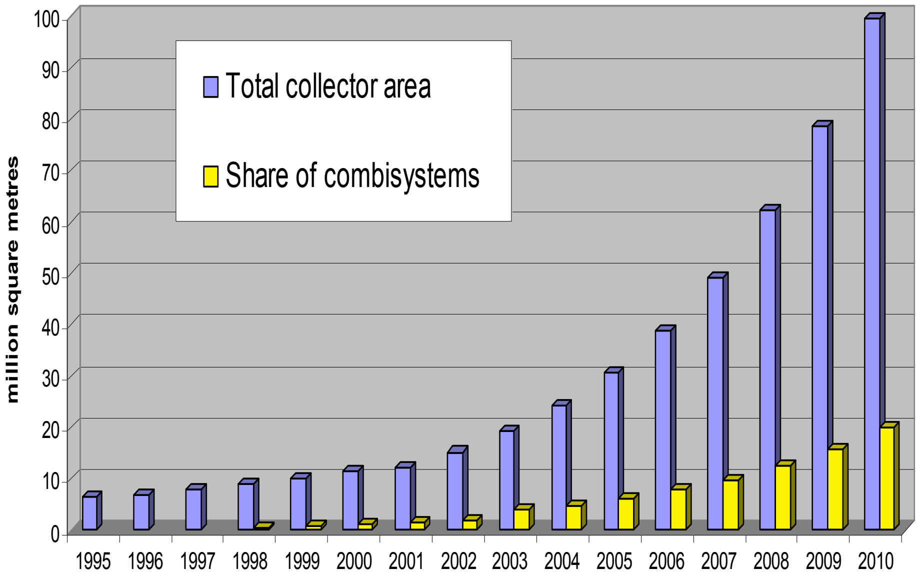


Figure 1. The predicted area of solar collectors that needs to be installed within the EU until 2010 in order to accomplish the goals, according to many experts. The part of collector area for combisystems are shown in yellow. The figure is from Weiss [23].

The demand for combined solar heating systems for both heating of hot water and space heating is increasing in many countries, and it's shown that even in northern European climates, space heating is possible. The market penetration of solar thermal systems differs drastically between the EU member countries in terms of number of installations and type of systems in use. Most systems are designed to produce hot water only, especially in the south European countries, where space heating is often not required. During the latest decades, hot water systems has represented the largest share of solar thermal installations in the building sector in the middle and north European countries. However, combisystems, providing heat for both hot water and space heating, become more and more popular. The potential for combisystems is large due to the fact that the energy required for space heating for residential buildings in middle and northern climates is 3-5 times higher than for hot water. In Austria, Switzerland and the Scandinavian countries Denmark, Norway and Sweden the number of installed combisystem in 2001 was equal to, or higher than the number of installed hot water system [17].

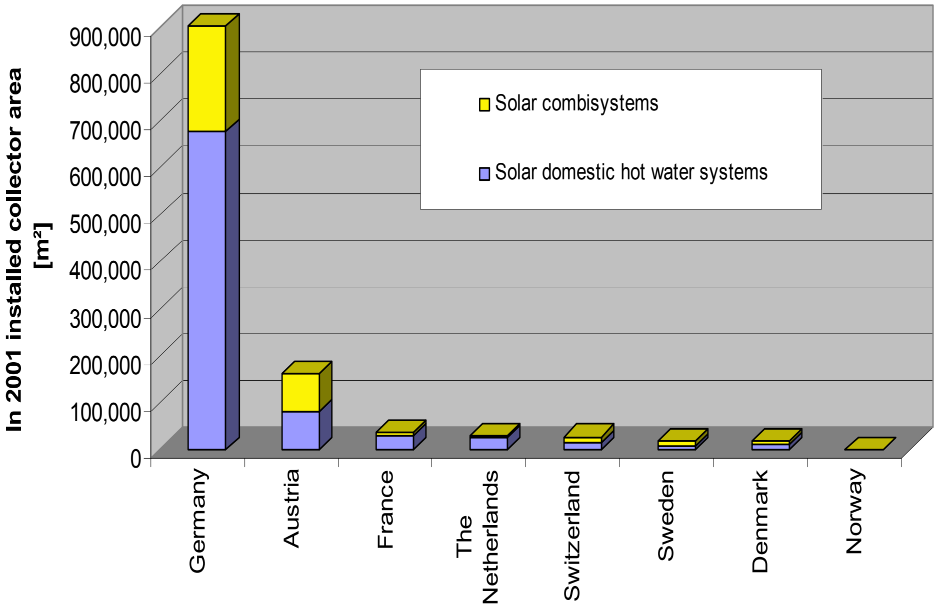


Figure 2. Installed collector area in different European countries. The yellow parts represent the collectors for combisystems. The figure is from Weiss [23].

Combisystems are rather complex systems since they consist of several components such as heating devices, heat exchangers, pumps etc. Therefore, the cost of solar combisystems is generally higher than the cost of solar domestic hot water systems. A general problem at higher latitudes is that the energy demand of a building is largest during the winter, when very little solar energy is available, and smallest in summer, when the solar energy supply is largest. For a domestic hot water system, this is not as big a problem as it is for combisystems, since the hot water demand is rather similar throughout the year, whereas the heating demand of a building is strongly depending on the seasonal changes, as can be seen in figure 3. On the contrary, the daily variations in heating demand for space heating are very small whereas the hot water demand varies significantly during the day. This seasonal load mismatch of energy demand and solar energy causes difficulties in optimising a solar combisystem with a high solar fraction. In order to profit from as much irradiation as possible during a year, the solar systems are often over dimensioned, which can result in overheating and stagnation in the summer time. The solar fractional savings that can be achieved with combisystems for the space heating load are generally lower than the solar fractional savings for hot water with a solar hot water systems as the heating demand and solar irradiation mismatch over the year. Typical industrial manufactured solar hot water systems for the north and middle European market cover 50% and more of hot water demand, depending on size, design and climate. Combisystems cover usually between 10 and 30% of the space heating load, but larger solar fractions can be achieved for houses with a high insulation quality.

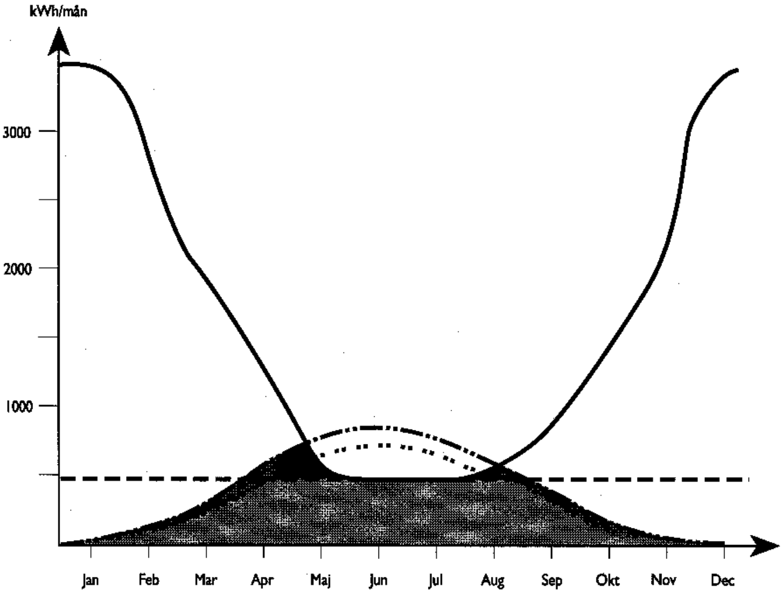


Figure 3. Illustration of the energy supply from two solar collector of 15 and 10 m^2, the heat load from hot water demand (the dotted horizontal line) and the total demand from both hot water and space heating (the unbroken line) for an general Swedish household in Swedish climate. The vertical axis shows energy in kWh/month. The mismatch of energy demand and supply often causes overheating and stagnation problems in solar system during summer. The figure is from "Solvärmesystem för småhus"(1998).

The differences in irradiation over the day and year can to some extent be compensated by using seasonal storage in smaller or bigger scale. This is however very expensive, causing large heat losses, and is unreasonable for an ordinary household. The summer heat can be stored to the winter in large reservoirs (60-130 m^3) if included in a solar heating plant or a very large system connected to district heating. That way the heat losses can be within acceptable range. For a small system, as one of the studied systems, storage can be used to store the heat for a few days, and the hourly variations can be compensated for by the building's thermal capacity.[17]

When installing a combisystem the collector area needs to be larger than if a DHW system where to be installed, since the energy demand is higher. Another difference between the systems is that the space heating loop of the combisystem has small temperature differences; a relatively low delivery temperature (30-50°C) but a relatively high return temperature (25-45°C). In the DHW system, the differences are much larger with high hot water temperatures (45-60°C) and low return temperatures (4-20°C). The energy needed by a DHW system is usually between 10 and 40% of the total heating demand for space heating and hot water together. The main difficulties in creating a well functioning combisystem is to achieve a good balance between the requirements of the space heating system and the hot water system and also to balance it to the consumers interests, so that the highest benefit is achieved from the collectors. The heat store is one of the most important components of a combisystem. The two heat loops of the combisystem require fluids of different temperatures and it is possible to use two different tanks, but it is also suitable to combine the systems into one storage tank with a high level of vertical stratification, with the hot water in the top and the cold water in the bottom. These systems can be constructed in a variety of ways. The input and output of heat to the storage tank can be achieved either by direct inlets to the tank or by heat exchangers inside the tank. However, the use of heat exchangers often creates unwanted uniform temperature zones above the heat exchanger, which can destroy the stratification, which is a very important characteristic of the tank. Direct inlets and outlets to the tank could be a better option in order to achieve a good stratification level, but this requires thorough system planning and adjusting of the inlet and outlet heights in the tank to the optional levels. In figure 4, an example of a combisystem is shown.

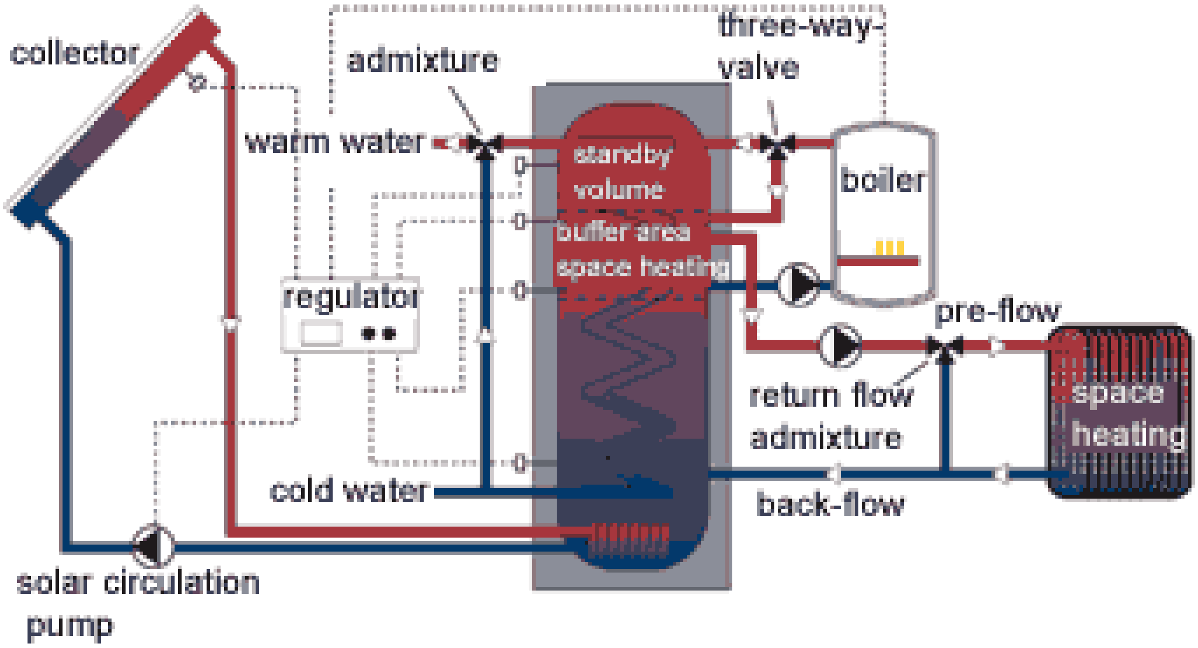


Figure 4. Design of a combisystem. The collector loop heats the tank through a heat exchanger, the hot water is heated through another heat exchanger and the water for space heating is supplied from the tank, which is stratified with hot water in the top and cold water in the bottom. An auxiliary heat source is also connected directly to the tank.

In different countries, different designs of combisystems are used more frequently than others. Small systems are most common in countries where the auxiliary heat source is gas or electricity, whereas the larger systems, which also have the largest solar contributions (heat demand covered by solar energy) are more common in countries where the major auxiliary heat source is pellets or oil.

The most important issues of the hydraulic layout of combisystems are, according to experts within Task 26, that the system delivers solar energy to the heat store with as low heat loss as possible, that it distributes all the heat needed to hot water and space heating demand and that it reserves sufficient store volume for auxiliary heating taking into account minimum running time for the specific heater. It should also have low investment costs, low space demand and be easy and failure-safe during installation.

3 Latest research on improvement of combisystems

3.1 Improvements from Task 26

IEA, the International Energy Agency, was founded in 1974 within the OECD (Organisation for Economic Co-operation and Development), and consists of 24 member countries who co-operate working with energy related issues. The Solar Heating and Cooling programme (SHC) is one of the first research and development implementing agreements of the IEA. Within the SHC, 27 different tasks have been undertaken. The aim with Task 26, *Solar Combisystems*, was to further develop and optimise solar combisystems, both for detached single-family houses, groups of single-family houses and multi-family houses, all with their own heating installation. 25 experts from IEA member countries and 11 solar industries worked with the task 26 from 1998 to 2002. They have developed standardised classification and evaluation processes and design tools for combisystems and made proposals for the international standardisation of combisystem test procedures.

Within the IEA Task 26 "Solar Combisystems" and other recent research projects the typical system designs of each country have been investigated. 21 Generic solar combisystem was defined in the beginning of the project and they have been investigated to different extents. Since it is important to compare different combisystems, the reference conditions of the systems have been determined as a standard. These are input data such as flow rate, temperatures etc. and fixed parameters, as for example the heat transfer coefficients, pump power and conductivities etc. Eight of the combisystems were investigated in more detail using the characterization tool FSC, which is further described in chapter 4.3. Simulation studies with the dynamic simulation tool TRNSYS have been performed to investigate the performance of the systems and to be able to improve them. Climate data from the program Meteonorm where used in these simulations. Methods allowing comparisons between the systems for different locations and load conditions have been developed.

Characterisation

Some of the aspects when dimensioning a combisystem are the collector area (which here divides the systems into small, medium or large systems), the number of tanks, the system using storage of auxiliary heat or not, the type of collector fluid, the inlet- and outlet heights, the type of heat exchanger, the flow rate, whether there is a stratifier or not, dimensions of components in the system, control algorithms, etc. According to Task 26, the two main aspects of classification of combisystems are: 1) the method to store the heat produced for space heating by the collectors, and 2) how the heat produced by the auxiliary heater is stored and how the heater is controlled.

Four categories for heat storage and stratification methods have been defined for different systems:
A: No controlled storage device for space heating
B: Multiple tanks and/or multiple inlet/outlet pipes and/or 3- or 4- way valves
C: Natural convection in the storage tanks, no built-in stratification device
D: Natural convection in the storage tanks and built-in stratification devices

Three categories are defined for the heat produced by the auxiliary heater:
M: Mixed mode – Both solar collectors and an auxiliary heat source supply a combined heat storage tank, which feeds the space heating loop.
P: Parallel mode - Space heating is supplied by heat from either the collector tank or the auxiliary heat storage tank.

S: Serial mode – The space heating is supplied with heat from either both the collector heat storage tank and the auxiliary tank (series connected) or the auxiliary heater only.

Additional characteristics are defined:
d: - Drain back system.
i: – Integrated gas or oil burner into the storage tank.
l: – Wood burners may be used which require long burning time and fixed power. Large storage is required.

Two of the characterized systems are shown in figure 5. To the left is a CM*i*-rated system that is often used in Finland, manufactured by the company Fortum. It uses an integrated oil or gas burner as auxiliary heat source, placed in the middle of the storage tank. The solar collectors are coupled to the system by an immersed heat exchanger in the bottom of the store. A radiator heating system or a floor heating system is directly connected to the tank, whereas the domestic hot water load is provided through an immersed heat exchange in the top of the store. To the right is a design of a BM*l*-rated Austrian system which has an early design with a large number of components. This system is more difficult to install, but it can also be very efficient.

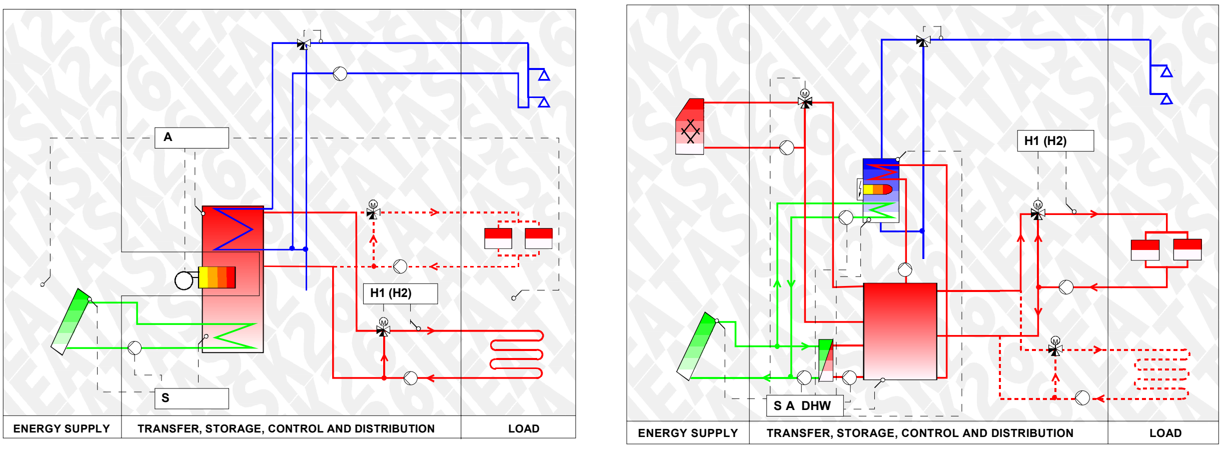


Figure 5. To the left a CMi-rated finish solar combisystem and to the right is a BMl-rated two- stores combisystem with fixed-power auxiliary heater, which is common in Austria. [17]

Results from the comparisons

The calculations performed have shown that the best performing system with the highest efficiency of the auxiliary heater, of the 8 systems that were studied in more detail, was a German DM*il*-rated system constructed as a compact unit where both an auxiliary gas condensing burner and a DHW flat plate heat exchanger are integrated in the storage tank. An immersed low-flow heat exchanger and stratifying tubes provide the solar energy input to the tank. Besides the condensing gas boiler, other auxiliary heaters can easily be connected to the system. The French AP-rated system is second best performing according to the results. In this system, the space heating loop is coupled both to the collector and to the auxiliary heater, so that the floor heating works throughout the heating season. A special tank has been developed with heat storage for DHW and hydraulic decoupling of collector, auxiliary heater and heating floor loops. The system is assembled in compact units. This system is easy to install and well suited for new houses. Se more details about the systems in [17].

3.3 New storage medium

Almost all solar heating systems for space heating and domestic hot water use water as storage medium in a buffer tank, hot water tank or combined tank. The availability, the harmlessness to the environment, the low price, the possibility to use the same medium in more than one circuit and the relatively high heat capacity of water are the main advantages of using water as the heat storage medium. The development of solar combisystems with high solar fractions discovers the limitations for water as storage medium. In order to achieve higher solar fractions the heat storage capacity need to be increased in order to store more surplus energy from days/seasons with high radiation and low load. The physical properties of water allow only the use of its sensible heat – in a temperature range between 10 and 95°C. The total heat capacity of the storage can of course be increased by increasing the storage size, but in residential buildings it is often not economically feasible, since the space is usually limited and the additional cost of an enlargement of the storage often is very expensive.

This is the challenge for other storage materials with better matching physical properties to be used in solar heating systems. Indeed, a lot of research is carried out to develop so called Phase Change Materials (PCM). The main idea behind the development of these materials is to use the latent heat to increase the heat capacity. Usually the phase change from solid to liquid is of highest interest because the phase change takes place under relatively small volume changes. The most appropriate materials have their melting point in the range of the operating temperature for the space heating and/or the hot water temperature, between 45°C and 60 °C.

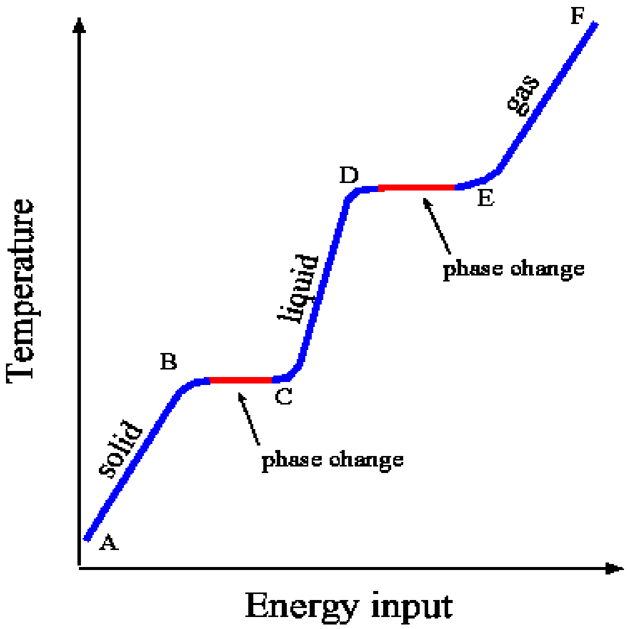


Figure 6. General graph of phase changes and energy input from solid state to gas state.

Storages using storage mediums such as paraffin drastically reduce the necessary storage volume compared to a water based store. Some storages with PCM are already available on the market, e.g. the store from the German company Powertank. Nevertheless a lot of work still has to be done to make them suitable for application in large scale. More information about the recent development can be found in the state of the art report of Dennis and Knudsen for this PhD course.

3.4 Improved storage insulation

In a solar heating system the storage is the weakest component in terms of heat losses. Between 5% and 10% of the total energy yield is needed to cover the storage losses, even for a rather well insulated storage tank. If the heat losses are not beneficial to the house it is necessary to reduce these losses to a minimum. This can be achieved by a thicker insulation but the necessary additional space demand is often limited. To improve the storage insulation, vacuum insulation materials, which are developed in the recent years for other applications such as building insulation or insulation for stoves and boilers, could be used. With vacuum insulation materials it should be possible to reach thermal conductivity values between 5 to 10 times lower than for ordinary insulation materials, depending on the temperature range. Within a Swiss research project, first prototypes of these new insulation panels have been produced and tested in different applications.

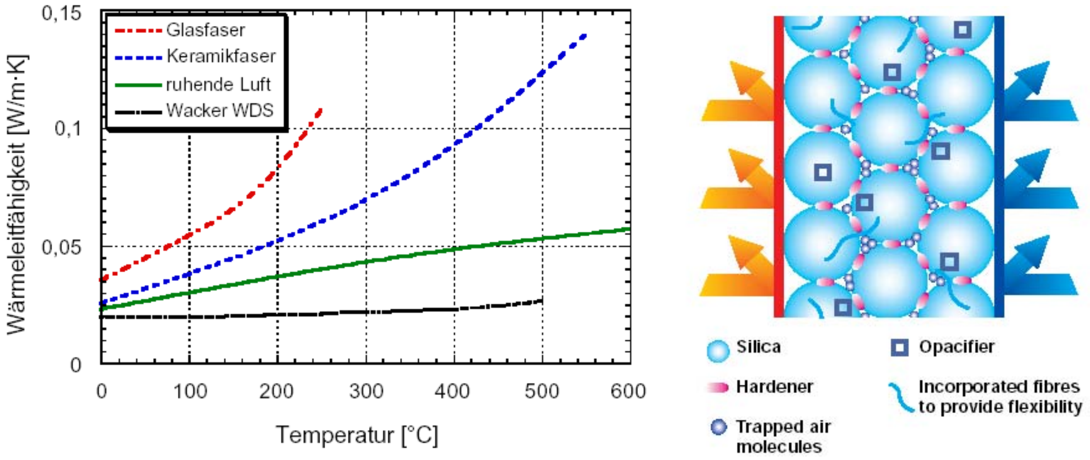


Figure 7. To the left a diagram of heat conductivity of vacuum insulation compared to other insulation materials. To the right is an illustration of the interior of a vacuum insulation material.

Vacuum insulation panels consist in general of a base material that is placed in a volume surrounded by gas-tight foils. The vacuum inside these panels has a key function due to the fact that the thermal conductivity of an insulation material depends mainly on the heat conduction of the gas inside the material. By evacuation the conductivity of the composite structure will be reduced. The base material is a kind of silicon acid with a very small pore size. It is produced in under low pressure and packed in panels and covered with a gas-tight foil [21]. First simulations with this kind of insulation material have been carried out but still have to be verified by measurements [18]. However, the material has not been produced for these applications, but there is a potential to lower the heat losses by using vacuum insulation.

3.5 Collectors adapted for seasonal load

As described above and shown in figure 3, there is a general load mismatch with high heating loads in the winter and high solar energy supply in the summer, which causes difficulties in optimising a solar combisystem with a high solar fraction without overheating the system in the summer. Newly developed collector designs provide possible solutions to this problem.

Load adapted collector (LAC)

One solution to reduce overheating and to adopt the output of the solar system to the demand has been proposed by Nordlander from the Solar Energy Research Center (SERC) in Sweden [14]. The collector uses a simple geometry with internal reflectors and a ordinary absorber fin.

The geometry of the reflector and the absorber implies a low optical efficiency in the summer when the heating demand is small and the solar elevation angles are high. In spring and autumn, when the solar elevation angles are low, the geometry implies a higher optical efficiency and thus a higher collector output. In figure 8, a simplified illustration of the collector and its optical performance can be seen.

Besides the improved match to the load conditions this type of collector also has improved thermal properties. Due to the fact that the absorber area is only a third of the glazing area the heat losses from the collector are lower than for a flat plate collector. The design offers also potential for cost reductions, since the price for reflector material is approximately three times cheaper than the absorber material [13]. Prototypes of LAC collectors have been built, tested and evaluated at SERC. The results showed good correspondence to the theoretical calculations [4]. Nevertheless some optimisation work is still necessary to commercialize this type of collector.

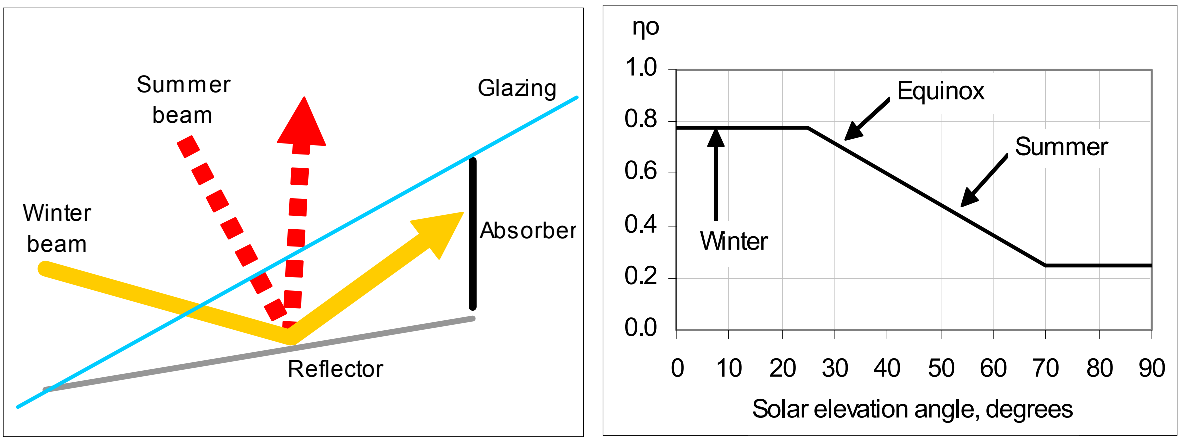


Figure 8. To the left, the schematic design of a LAC collector, to the right, the optical efficiency of the collector as a function of the solar elevation angle.

The Maximum Reflector Collector (MaReCo) design

Another collector design has been developed at Vattenfall Utveckling AB in Sweden in order to avoid the problem of overheating during the summer [6]. The spring-/fall- MaReCo collector is a concentrating collector that suppresses the summer performance, selecting irradiation from certain solar altitude angles. The collector, shown in figure 9, consists of an asymmetric reflector trough with a bifacial absorber, which will receive irradiation from both sides when the concentrator is working. The reflector is partially parabolic with the optical axis tilted 45° to the horizontal. This implies that irradiation coming from solar altitudes of 45° and lower will be reflected onto the backside of the bifacial absorber. During summertime, the concentrating reflector is not working, since irradiation is coming from solar altitudes higher than 45°, and the absorber will only receive direct irradiation from the upper side. This way, the collector yield in spring and fall can be increased in relation to the summer yield, which is lowered, thus creating optional conditions for the solar heating system by , preventing overheating and stagnation in summer. This collector is designed for roof-integration. Since the concentrated irradiation increases the energy output per absorber area when working, it will be sufficient to use less absorber area than in an ordinary solar heating system. The material of the reflector is cheaper than the absorbers and the lowered use of absorber area will lower the total cost of the collector. As it is intended for roof integration, savings in building material are also expected.

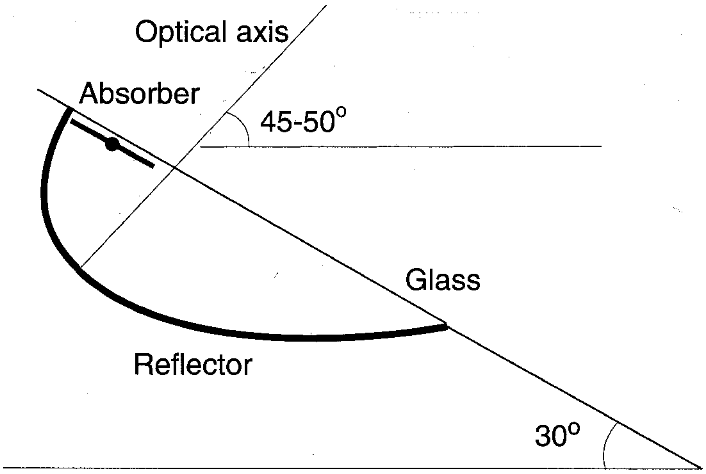


Figure 9. Design of a spring-/fall-MaReCo collector

Vattenfall Utveckling AB has evaluated the performance of a prototype of the Spring/Fall-MaReCo. This study shows that the efficiency decreases, as wanted, during the summer. According to energy simulations, the annual energy output is estimated to approximately 222 kWh/m² per glazed area, when the operating temperature is 50°.

Façade collectors

Another approach to overcome overheating problems would be to mount the collectors with a higher tilt. From figure 10 can be seen that the distribution of radiation for a collector with 90° tilt is more even than for a collector with a tilt of 45°. During the summer, when overheating usually occurs, the amount of radiation on the collector is significantly smaller. It thus implies a lower annual energy yield from the collectors. In central Europe the annual solar irradiation on a façade is about 30 % less than the irradiation on a south-facing roof with a 45° slope [22], so a larger collector area is needed in order to produce the same amount of energy as a roof collector would produce, but for combisystems with large façade collector areas no or less overheating will occur.

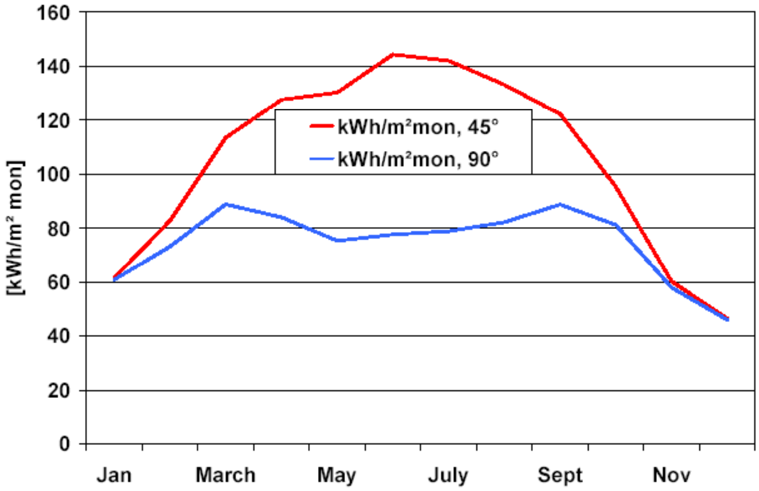


Figure 10. Solar irradiation on a 45 °tilted surface and a 90° tilted surface, Graz; Latitude 47°. Figure from Weiss [22].

3.6 Compact systems

Some years ago solar combisystems had a rather complex design with separate components as the solar collectors, a hot water storage, a storage for space heating, a boiler and a controller. Due to the complexity of the system hydraulic problems and difficulties to control the system in terms of a good interplay of all the components were very common. This was reason for a rather poor overall efficiency of these systems. In order to optimise the systems, the manufactures have increased their efforts in the recent years to improve the components and to simplify the systems by integrating the components into one devise, the heat storage (figure 11).

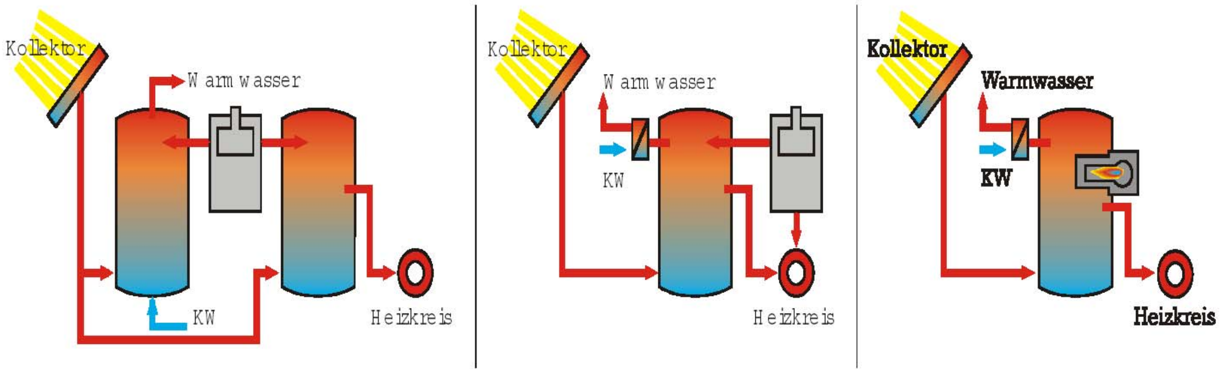


Figure 11. Complete integration of all components in one device (source Solvis).

The idea of an integrated system is based on a single stratified storage tank as one central unit. The heat storage provides heat for hot water as well as for space heating from the appropriate temperature layer in the store, through direct outlets from the tank or through a heat exchanger. To avoid mixing in the store each energy input, from the collector or the auxiliary heater, is stored at the right temperature layer in the store. By integration of the components into a compact system significantly less pipe connection are required and the total pipe length is reduced drastically. The installation of compact system is therefore easier and faster, reducing the cost of installation and avoids wrong connections during installation. Due to the shorter pipe length and the smaller number of connections, the heat losses are reduced.

The design of a compact auxiliary burner that can be integrated into a solar store is in progress. Meanwhile integrated solution for oil and gas boilers are on the market and used in several systems, e.g. the combisystem of the German company Solvis. In some European countries pellet boilers are commonly used as auxiliary heat source. The existing pellet burners that can be integrated into the store are not yet satisfying in terms of space requirement, maintenance and efficiency. A prototype of an integrated pellet burner has been developed by Lorenz at SERC and was presented at the ISES conference 2003 in Gothenburg. It can be mounted in any body of water, including a storage tank. The so called Pellet Integral, shown in figure 12, has a diameter of 25 cm and a length of 50 cm only and comprises a burner, a combustion chamber and a heat exchanger. [10]

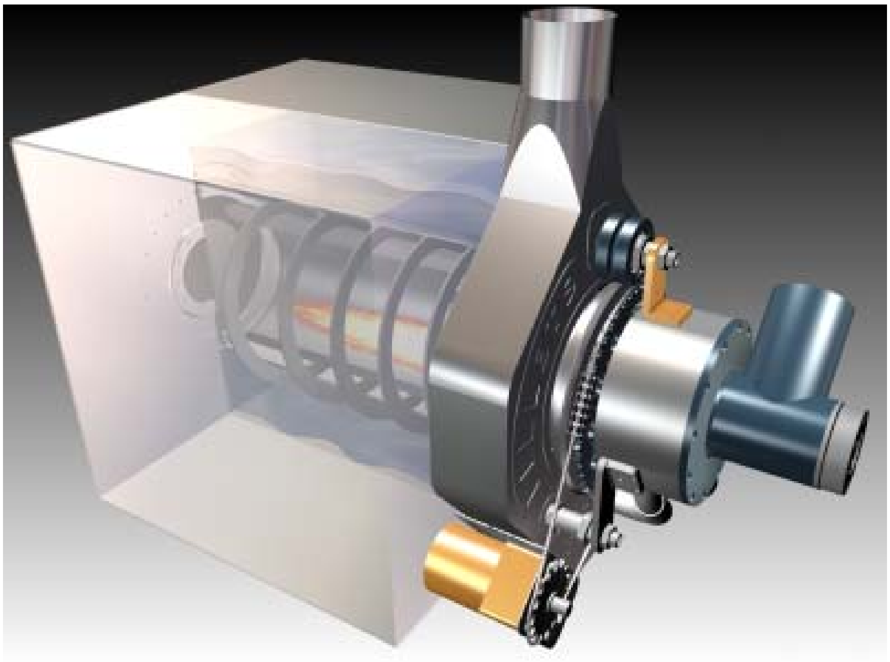


Figure 12. Cut away view of the Pellet Integral mounted into a prototype boiler.

An important innovation of this burner is that it automatically removes ash and cleans the inside of the heat exchanger. The pellets are automatically fed into the combustion chamber. These two design features lower the maintenance requirement significantly compared to the existing burners on the market.

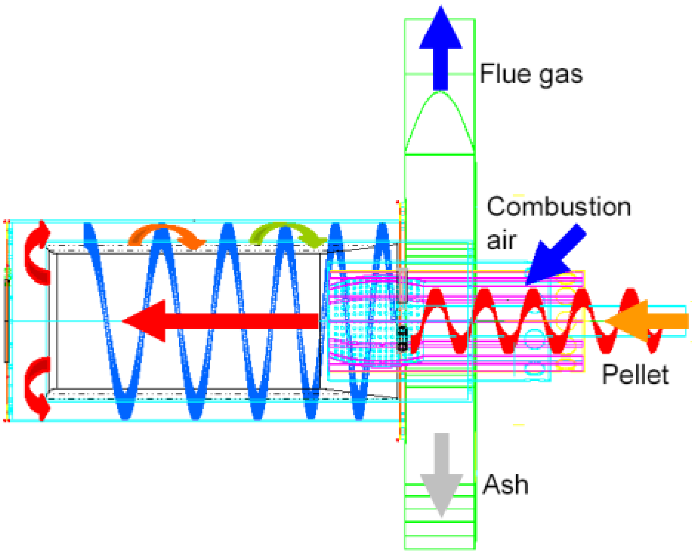


Figure 13. Cross section of the Pellet Integral combustion chamber and heat exchanger.

The combustion chamber has a feed unit (on the right) that serves as supplier for pellets and combustion air as well as removal for ashes (bottom) and exhaust gases (top).The exhaust gases are forced by the outer spiral screw to take a 7 m long path towards the outlet at the top of the unit, resulting in a good heat transfer to the surrounding water and a very low flue gas temperature, which allows the use of a plastic chimney.

3.7 Advanced control strategy for solar combisystems

An advanced predictive control strategy for solar combisystems has been developed by Prud'homme and Gillet [15]. The strategy treats the energetic optimization of a building and a combisystem as a unique system. A dynamic model of a combisystem and a building has been developed in order to compute an optimal profile of the energy to be dissipated into a building over one day. Weather forecasts are required in the model. In order to adapt the model to buildings of different types than the building used in the developed model, an automatic parameter identification strategy has been developed as well. An illustration of the system considered in this work is shown to the left figure 13. The pump, collector and the heat exchanger of the collector in the storage tank are considered as a closed loop and the hot water passes through another heat exchanger in the tank. In this case a gas burner serves as auxiliary heat source and this gas burner is either switched on or off, just as the collector pump. The space heating loop is connected to the tank and the power that is dissipated into the building is controlled with a three-way-valve.

The dynamic model of the combisystem can often easily be obtained from the manufacturer. In this control strategy, four parameters of the combisystem model are controlled in order to optimize the energetic performance of the system and also the comfort in the building. The four parameters are the power of the auxiliary gas burner, the flow rate of the two pumps (for the collector loop and for the space heating loop) and the aperture angle of a three-way valve in the space heating loop. The input data of the four variables are put together in a time-dependent vector, u (t). The disturbances, various climate data (such as for example the solar radiation and ambient temperature), are grouped together in another variable, w (t). Every component is described by different nodes, as can be seen in the picture. The building is just divided in to nodes, one for the walls and floors and one for the air in the building. The energy balance of each node is considered, leading to one differential equation for each node in the system (49 in this model). In the different parts of the system, the temperature of the liquid varies gradually, and all the temperatures are grouped together in a vector that depends on the previously defined vectors, $x(t) = f(x(t), u(t), w(t))$. The power that is dissipated into the building is then computed by an optimization algorithm. Therefore the system is referred to as a predictive system. The control strategy is illustrated to the right in figure 14.

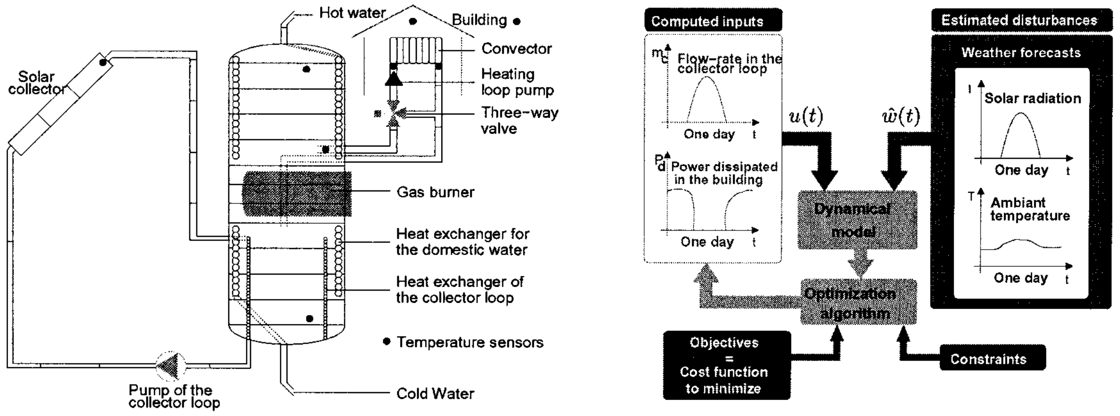


Figure 14. To the left, the design of the combisystem in the model and the nodes for modeling. To the right is an illustration of the control strategy.

Conventional PID-controllers will control the pump of the heating loop and the three-way-valve. That way the power dissipated in the building will be close to the power optimized by the algorithm. An objective function, J (shown below), aims to minimize the gas consumption and maximize the comfort of the users. T_b is the temperature inside the building, α is a trade-off factor, T_{set} is the desired temperature in the building and P is the power of the pump, burner and collector, respectively.

$$J = \int_{24h} \left[P_{burner} - \left(P_{sol} - P_{pump} \right) \right] dt + \int_{24h} \alpha \left(T_b - T_{set} \right)^2 dt$$

A closed loop mechanism is introduced in order to make sure the optimization procedure is repeated whenever there is a change in the weather forecast. It is thus important to have access to frequent measurements of climate data. The optimized results of the system, the variables in the vector *u(t)*, have been used as input to a real solar combisystem where an ordinary control strategy and this advanced strategy has been compared. Both weather forecasts and measured data has been available and used to compare the behavior of the two systems. A reduction in gas consumption of 13% has been measured for the system with the advanced control strategy and the comfort in the building was significantly higher than for the ordinary control strategy. This is due to better regulation performance and due to the anticipation of passive gains, since this controller knows in advance when the weather is going to be fine and then it can turn off the heater in time. However, additional tests should be performed in order to demonstrate the economical advantages of this control strategy.

4 Evaluation methods for combisystems

Combisystems consist of several input and output units to produce, deliver and store heat, and to control the interplay between these components. A determination of the characteristic parameters of the system is necessary to compare the different systems and system concepts and to generate performance predictions. The existing standard test method CTSS was so far used to test the systems and identify the parameters. Due to the fact that this method is very time consuming, alternative test methods have been developed. There are several approaches to identify the characteristic parameters of solar combisystems by using different test methods. The most known are described in this report:

- The CTSS test method – component testing/system simulation
- The concise cycle test (CCT) method – a twelve day system test
- The DC (direct comparison) test method

4.1 The CTSS method

This method was the standard test method before the CCT and DC methods had been developed within the framework of the IEA Task 26. It has been further developed by the ITW in Stuttgart during a two year project about solar combisystems. The characteristic parameters determined by the CTTS-procedure can be used to evaluate the performance of a combisystem for defined reference conditions as climate or load profiles, by using them in annual system simulations [3]. The CTTS method has been validated by long term measurements [7], which showed that the simulation results using the output parameters from this method don’t differ more than 5% from the measured data. The main advantage of this test method is its flexibility due to its component oriented approach. It is possible to apply the CTSS method to nearly every system configuration.

4.2 The DC test method

The DC test method is based on the thermal store test procedure called combitest that has been developed at the Solar Energy Research Center in Borlänge [1]. The combitest was built up by knowledge gained from other short term test as the six-day test [11], the CTSS test method and the STF method [2]. To test a system according the DC test method, the system of interest has to be set-up completely. Therefore it is especially interesting for factory made systems.

The test can be carried out at indoor or outdoor facilities. In order to simplify the set-up, the heat sources, the collector and the auxiliary heater, are often emulated by electrical heating elements with variable output. Another solution for the solar input would be to use a solar simulator. In case the input are emulated, detailed parameters for information about the control function of the collector and auxiliary heater should be available. The DHW as well as the heating load are usually emulated by an electrical cooling device. The emulation has to take specific climate conditions and the specific temperature of the heat distribution system into account.

The test is performed for at least 8 days, where the first two days are used to achieve thermal conditions in the store similar to how the conditions will be after the tests are completed. In the core phase, during the remaining 6 days, the system is tested with typical conditions for two winter days, two summer days and two autumn/spring days. During the tests all heat flows in and from the storage are recorded [3].The method can be applied for systems with up to 20 m^2 collector area and 2000 liters heat store volume. For these system sizes, the predicted error of the annual system performance is smaller than 5 %.

Figure 15 shows the principle of how the method has been evaluated by Naron and Visser at TNO [17]. On the right hand side of the scheme, the solar combisystem (SCS) model is used to carry out simulated tests for a chosen 6 days core sequence. The identified parameters are then applied to calculate an annual performance prediction. On the left hand side, the annual performance prediction is a direct result of a simulation using the solar combisystem model. A comparison of both predictions gives an idea about the accuracy of the performed test.

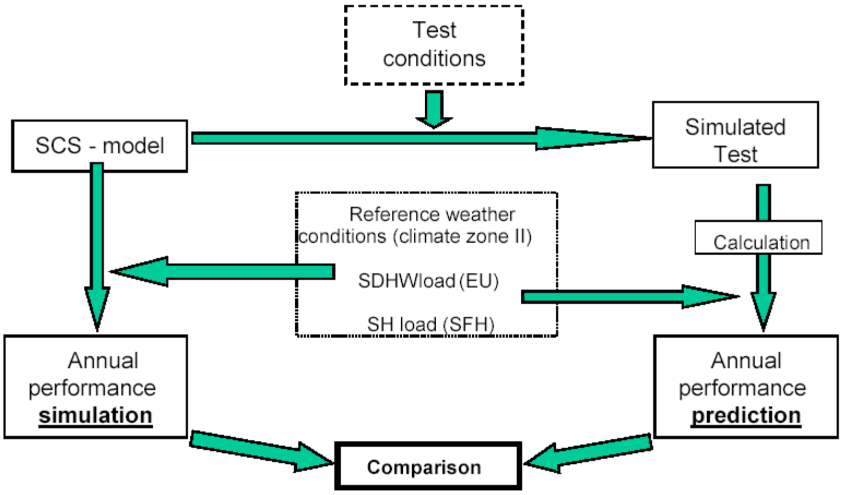


Figure15. Scheme for evaluation of the test quality [3].

Tests of larger systems revealed too high prediction errors. It is possible to derive correction factors, but they can be used exclusively for systems tested with the same the reference conditions. Generally the method reaches its limits when different climate conditions and demands for space heating and domestic hot water should be applied. The test results are only valid for weather conditions that correspond to the test weather conditions and similar heat loads. Therefore three climate zones and three space heating and hot water loads have been derived.

A main advantage of the DC method compared to the CTSS method is that the controller function also is tested, since the system is completely set-up. The latest work on the DC test method has been carried out by the work of TNO in the Netherlands with the intention to be passed on to CEN as a work item for the Technical Committee 312 that works with this item [17].

4.3 The CCT method

The **C**oncise **C**ycle **T**est Method has been developed within Task 26 by Swiss research institute SPF and is somewhat similar to the DC test method. The core phase consists of 12 days instead of 6 days in order to achieve more accurate performance predictions. In contrast to the DC method the heating load of the building is simulated online and the system with its controller(s) decides how the heat is supplied to the building. A significant advantage of this feature is that all of the systems functions may be assessed. The disadvantage is that there is no uniform or predictable energy use for the space heating, which complicates the characterization of the systems energetic performance. Unlike the DC method, the CCT method can in principal be used to characterize solar combisystems, where the system uses the thermal mass of the building to optimize its heat storage strategy, e.g. when there is a heavy heating floor such as in many French direct solar floor systems [20].

At the present neither the DC test method nor the CCT method have the status of a preliminary standard. Both test procedures still need validation and more practical experience. More information on the DC and CCT method can be found in technical reports of task 26 possible to download on:
http://www.fys.uio.no/kjerne/task26/handbook/tech_reports.html.

4.4 The characterization tool FSC

In Task 26 eight combisystems with different components and operational parameters from different countries have been investigated. Usually the fractional energy savings, F_{sav}, where the auxiliary heat demand of the investigated combisystem is compared with a reference system without solar collectors, is used as measure of the system performance. This method only allows systems with the same climate and loading conditions to be compared. To be able to compare the simulation results for the systems in Task 26, it was necessary to develop a new characterization tool. The so called Fractional Solar Consumption (FSC) method takes simultaneously into account the climate, the space heating and hot water loads, the collector size, its orientation and tilt angle, but does not depend on the studied systems. With the FSC method the maximal fractional solar gain for the given reference conditions (load + irradiation), is calculated by dividing the usable solar energy in terms of radiation on the collector area by the heat demand that has to be covered by the system. The FSC value gives somewhat the upper limit of how much solar energy can be delivered to the system [9].

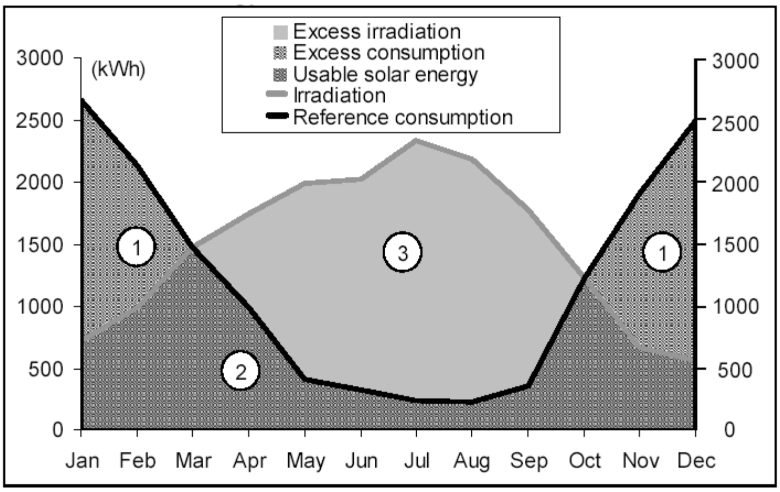


Figure 16. Monthly plot of the final energy consumption for a reference system and the solar radiation on a specific collector area.

By calculating the FSC values, according to the equation on the next page, for different locations and for different loads and by plotting them in a diagram with the fractional savings as the Y-axis, a graph for each system can be obtained.

$$FSC = \frac{Q_{solar,usable}}{E_{ref}}$$

A plot of the graphs of several systems in the same diagram allows a comparison of the performance of the systems, as is can be seen in figure 17.

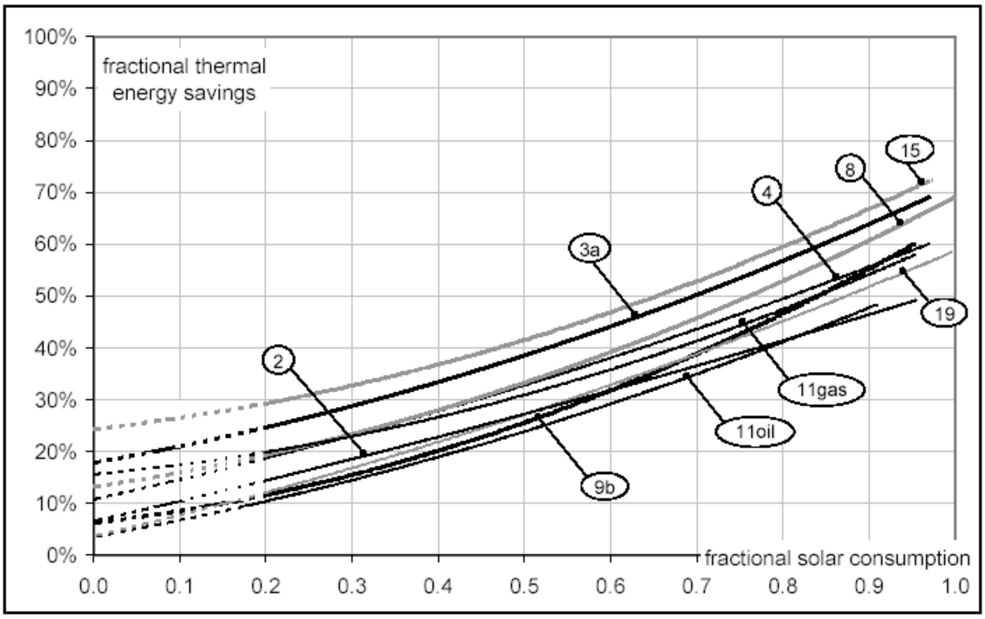


Figure 17. Fractional thermal energy savings as a function of fractional solar consumption for systems simulated in task 26. The upper graphs are for systems with the best performance.

4.5 Calculation model for assessment of reliability in solar combisystems

A new calculation model has been developed within the task 26 in order to get a suitable tool for assessment of reliability of the combisystems, which can be used in the development of new combisystems. When designing combisystems, the thermal performance is often the main focus. However, other parameters as economy, user friendliness and reliability are being considered more and more. This model, developed in Sweden, 2000, and described in [8] [5] and [24], has two kinds of analyses; one that compares the system complexity of the different generic systems defined within Task 26(described in [24]), and one calculation model considering the quality, maintenance frequency and lifetime of the system components. The model has some limitations. It does not account for the different criticalities to the system function of single components, but treats every component with the same lifetime as if they're equally critical. A component either works or doesn't work. Smart controllers have not been assessed. The system borders of the model include all components related to the collector loop, but it doesn't include components related to the space heating loop, nor to the auxiliary heater. The reliability, here defined as the probability that a unit functions as intended or better, is reduced with time both for the components and the systems they form together, and therefore the lifetime of the components plays a major role in this model. The model is described in an excel file, where different components of combisystems are listed and for each of them lifetimes and their distributions are described for three different quality levels. The related maintenance intervals are described as well. Information of lifetimes has been collected from researchers and manufacturers, whereas data of the related distribution and maintenance intervals are based on assumptions. The generic system number 12 of Task 26 is used as a test system in this model. It is a Swedish system with two immersed heat exchangers from the collector loop and one heat exchanger for the hot water, which are used together with an electric heater inside the tank. For every component of the system, the reliability is calculated and then the reliability of a whole system is calculated, using equations described in [8], [5] and [24]. Here is also described how to adapt this model to combisystems different from the tested system. The outputs are results of nine different combinations of quality and maintenance for the system, which are calculated as “system lifetime averages” of a reliability indicator. The variation of this indicator with time can then easily be plotted in a graph and compared to other situations of quality/maintenance for the same system. The model files, component list and a report [5] of this work can be downloaded from the link below.

ftp://ftp.sp.se/public/solar combisystems reliability calculation

5 Summary

Solar heating systems for hot water and space heating provide the possibility to boost the share of renewable energy sources for heat production in residential buildings. According to the EU White paper “Energy for the future, renewable sources of energy” the installed collector area in the EU member countries has to be increased drastically until 2010. In the middle and northern European countries the heat load for space heating is still significantly higher than for hot water only, even for well insulated houses. Combisystems will therefore play a major role to reduce the amount fossil fuels used for heating in residential buildings.

Solar combisystems are no longer customer made systems comprising a lot of different components and using several control units. Today many factory made systems are on the market with, of course, country adopted system concepts. Within the IEA Task 26 “Solar Combisystems” and other recent research projects the typical system designs of each country have been investigated. Simulations have been performed to investigate the performance of the systems and to improve them. Methods allowing comparisons between the systems for different locations and load conditions have been developed. The results from these projects are a milestone in the development of solar combisystems.

Nevertheless it is desired to reach higher solar savings than 10 to 30 %, as for typical combisystem with typical load conditions. The focus is therefore on a further development of the systems, both the system concepts as well as the components of the systems. Some important possibilities to further improve the performance of solar combisystems are described in this report. However, this is only an extract of the research work performed in several research institutes and enterprises. It can be expected that the large effort in research in many countries will lead to further milestones of the development of solar combisystems.

Literature

1 Bales C., Thermal store testing, Evaluation of test methods, Chalmers University of Technology, Göteborg, Sweden, 2002.

2 De Geus and Visser H., An evaluation method for solar energy systems tested in the TPD system test facilities, TNO Institute for Applied Physics, Delft, Holland, 1987.

3 Drück H. and Bachmann S., Performance testing of solar combisystems – Comparison of the CTSS with the ACDC test procedure, IEA SHC-Task 26 Solar Combisystems, Subtask B, November 2002.

4 Fiedler F., Optical and thermal performance of load adapted solar collectors, Master thesis, Högskolan Dalarna, Sweden, 2002.

5 Johannesson K. and Persson J., Reliability analysis of solar combisystems. A method and a model. Swedish National Testing and Research Institute Work Report SP AR 2001:37.

6 Karlsson B. and Helgesson A., System testing of a MaReCo with suppressed summer performance, Proceedings ISES Solar World Congress, Göteborg, June 2003.

7 Kerskes H., Validation of the CTSS test procedure – Procedure by in-situ measurements, IEA SHC-Task 26 Solar Combisystems, Subtask B, December 2002.

8 Kovács P., A calculation model for assessment of reliability in solar combisystems, IEA SHC – Task 26 Solar combisystems, subtask A.

9 Letz T., Bales C. and Perers B., A new concept for combisystems characterisation, The FSC Method, Proceedings ISES Solar World Congress, Göteborg, June 2003.

10 Lorenz K. and Bales C., Pellet integral – Auxiliary pellet burner integrated into solar stores, Proceedings ISES Solar World Congress, Göteborg, June 2003.

11 Lorenz K., Bales C. and Broman L., Performance comparison of combitanks using a six-day test, North Sun ’97, Espoo, Finnland, 1997.

12 Naron D. and Visser H., Development of the direct characterisation test procedure for solar combisystems, IEA SHC-Task 26 Solar Combisystems, Subtask B, 2002.

13 Nordlander S. and Lorenz K., Economic analysis of combisystems for high solar fractions, Proceedings ISES Solar World Congress, Göteborg, June 2003.

14 Nordlander S. and Rönnelid M., Load adapted solar collectors, North Sun 2001, Leiden, The Netherlands, 2001.

15 Prud'homme T. and Gillet D., Advanced control strategy for solar combisystems. Proceedings ISES Solar World Congress, Göteborg, June 2003.

16 Schönherr M., Solar thermal in the EU, Refocus – The international renewable energy magazine, March/April 2003.

17 Suter J. et. al., IEA SHC-Task 26 Solar Combisystems, overview 2000. Bern, Switzerland, July 2001.

18 Tepe R. and Bales C., Simulation study of a dream system, IEA SHC-Task 26 Solar Combisystems, Subtask C, April 2003.

19 Visser H. and Naron D., Direct characterisation test procedure for solar combisystems - 5th draft, IEA SHC-Task 26 Solar Combisystems, Subtask B, March 2003.

20 Vogelsanger P., The concise cycle test method – A twelve day system test, IEA SHC-Task 26 Solar Combisystems, Subtask B, November 2002.

21 Wacker G., Product information brochure, Kempten, Germany, 2003.

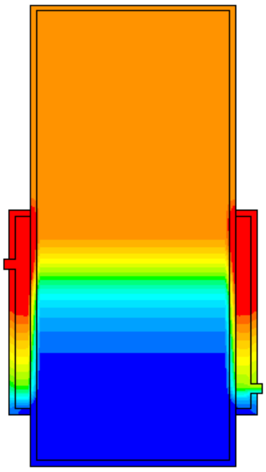
22 Weiss W., Solar heating systems – Status and recent developments, Proceedings ISES Solar World Congress, Göteborg, June 2003.

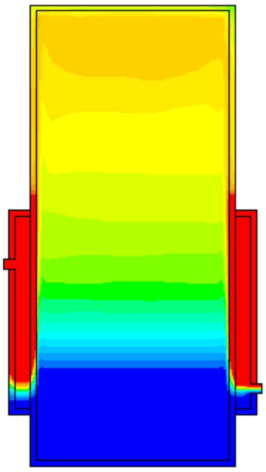
23 Weiss W. et. al., Comparison of system design for solar combisystems, Proceedings ISES Solar World Congress, Göteborg, June 2003.

24 Weiss W (ed.), Solar heating systems for houses – A design handbook for solar combisystems, James & James Science Publishers Ltd., London, 2003.

Thermal Storage for Small Solar Heating Systems

State of the Art Report





27th June 2003

Søren Knudsen

Department of Civil Engineering
Technical University of Denmark
Lyngby, Denmark

Mike Dennis

Centre for Sustainable Energy Systems
Australian National University
Canberra, Australia

Preface

This report is carried out as a part of the Ph.D. Course Solar Heating held at Department of Civil Engineering, Technical University of Denmark June 2-27, 2003.

The report presents state-of-the-art for Thermal storage for small solar heating systems based on literature study and presented papers at ISES 2003 Solar World Congress, June 16-19, Göteborg, Sweden.

Kgs. Lyngby, Denmark, 27 June 2003

Mike Dennis

Søren Knudsen

Table of contents

Preface 2

Introduction 4

Thermal Energy Storage – Part 1 5

Water based heat storage 5

Introduction 5

Types of Hot Water Storages 6

New trends and concepts presented at ISES 2003 10

Conclusion and Future Developments 13

Thermal Energy Storage – Part 2 14

Phase Change Materials (PCM) 14

Introduction 14

Charge and Discharge of PCM 17

Practically issues of PCM Materials 18

Applications to a Domestic Hot Water System 18

Opportunities for Future Work 21

References 22

Introduction

Many domestic water heaters do not use thermal storage. They heat water when and where it is required. This is very energy efficient but provides problems of peak energy network loading and the inability to effectively access solar energy.

Hot water systems that utilise solar contribution must necessarily use energy storage when the loads are to be drawn beyond the solar day. This is typical of domestic hot water loads. Traditionally, water based storage is used and this provides the great majority of thermal storage in domestic hot water systems around the world.

Thermal storage technologies may be grouped into three main classes:

- Sensible heat stores
- Latent heat stores
- Chemical energy stores

This paper reviews water based (sensible heat) storage technologies and provides an overview of an alternative technology based on phase change materials (latent heat). Chemical stores are not yet well developed. Systems that offer combined hot water and space heating (combi-systems) are not reviewed.

The performance of a thermal storage system is primarily defined by its efficiency to cost ratio and energy density. The efficiency of a thermal store relates to the amount of heat lost over the duration of energy storage. The paper reviews factors and research that influence these performance parameters.

Thermal Energy Storage – Part 1

Water based heat storage

Introduction

Water is very often used as the storage medium in solar heating systems. The water is inside a tank (typical made of steel) and the tank is connected through pipes to a solar collector array. In some parts of the world the pipes from the storage tank are connected directly to the solar collector and water from the tank circulates through the collector to get heated. In colder climates were there is a risk of the fluid freezing in the collector, an antifreeze heat transfer fluid (typically a mixture of propylene glycol and water) is used, and some heat exchange arrangement between the collector fluid and the water in the tank is required. This means that different types of systems and different types of hot water storage tanks are used in various locations and climates.

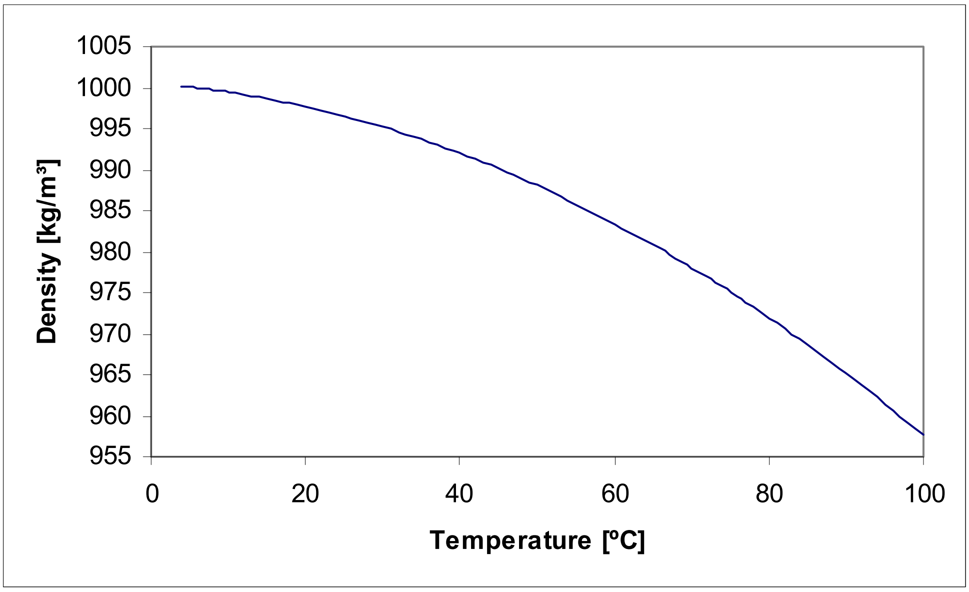


Figure 1. Density of water as a function of temperature.

Thermal storage plays an important role in securing a high thermal performance of the different systems. Some of the relevant performance parameters are:

- It is important to minimise the heat loss from the heat storage by insulating the storage and avoiding thermal bridges in the upper (hot) part of the tank.
- Inside the hot water tank it is important to secure a high degree of thermal stratification, that is, with the top of the tank hotter than the bottom. A high degree of thermal stratification increases the thermal performance of the solar hot water system. Figure 1 shows the density of water and it is seen that it is varying with temperature and that makes it possible to create density driven stratification with water as heat storage medium.
- The heat storage often has an auxiliary energy supply system to supply heat when there is insufficient solar energy available and the auxiliary energy supply system has to be efficient.

Types of Hot Water Stores

In this section an overview over the most common types of heat storages for solar hot water systems is given, disseminated by country.

Australia, Greece and Israel

Thermosyphon systems:

In countries such as Australia, Greece and Israel where there is a lot of sun available and high ambient temperatures, thermosyphon systems are often used. In a thermosyphon system, the flow in the pipes between the solar collector and the tank is driven by natural convection. To achieve natural circulation during the day and to minimise risk of reverse circulation at night, the tank must be located above the collector. As water in the collector is heated in the collector it rises naturally into the tank, while cooler water in the tank flows down to the bottom of the collector, causing circulation throughout the system. The heat storage in this thermosyphon system is a horizontal tank and the tank is placed outside on the roof just above the solar collector (figure 2). In Australia, new water tanks must be located outside, either on the ground or roof.

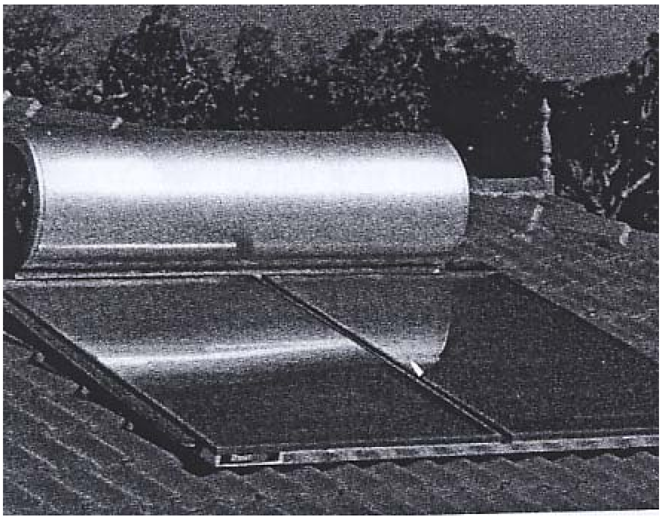


Figure 2. Thermosyshon system with horizontal tank on roof.

Sometimes the water from the tank is heated directly in the solar collector. In Australia antifreeze solution is often used in the collector loop, and the collector loop has to be isolated from the water in the storage tank. A horizontal mantle heat exchanger is then used (figure 3). The inlet to the mantle in the horizontal mantle heat exchanger is placed at the bottom. Rosengarten (2000) has investigated a side inlet configuration and a top inlet configuration. The side inlet configuration had a lower heat transfer rate to the inner tank due to absence of the impinging jet region and the top inlet configuration was not sufficient in periods with low inlet temperatures. These investigations were pure numerical and in reality the low inlet temperature maybe not occur that often because the flow is self-regulating. However, with a bottom inlet to the mantle, the horizontal mantle heat exchanger does not promote thermal stratification because the highest heat transfer rates are near the inlet at the bottom (Morrison et al., 1998). The horizontal tanks also have a low height/diameter ratio, which makes it difficult to build up a high degree of thermal stratification. On the other hand, thermosyphon systems with a horizontal heat storage placed on the roof just above

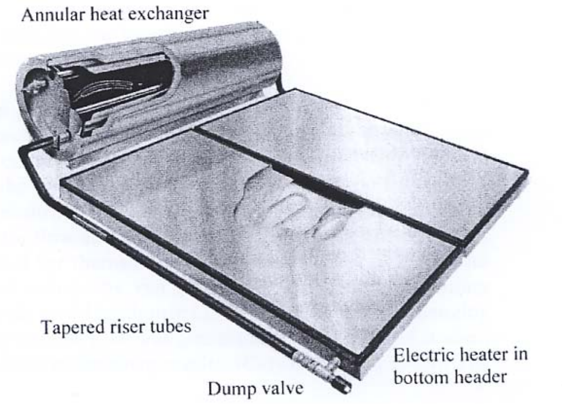


Figure 3. A system with a horizontal heat exchanger.

the solar collector does not need a pump and controller, which makes them more reliable and they have a longer life than pumped systems (Solar Energy – state of the art, 2001).

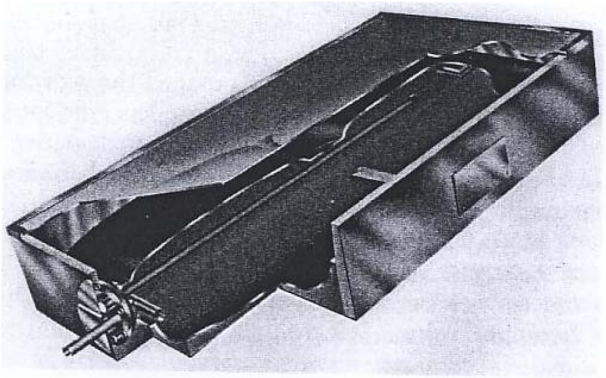


Figure 4. Integral tank-collector system.

Integral tank-collector systems:
Another type of hot water storage used in warm climates is the integral tank-collector systems (ICS) where the tank and collector is combined into one unit (figure 4). These systems are simple, effective and low cost. However, due to high heat loss at night they only provide hot water during the day and early evening (Solar Energy – State of the art, 2001). Thermal protection of the storage tank of the ICS system is difficult, as a significant part of their surface is used for the absorption of solar radiation. Double-glazing, selective absorbing surface coatings and transparent insulation materials are used for the thermal protection of the storage tank (Souliotis and Tripanagnoustopoulos, 2003). A main limitation of the ICS system concept is that it is only a pre-heater and hence must be connected in series with a conventional water heater if a 24-hour water supply is required.

China

Figure 5. Water-in-glass solar water heater.

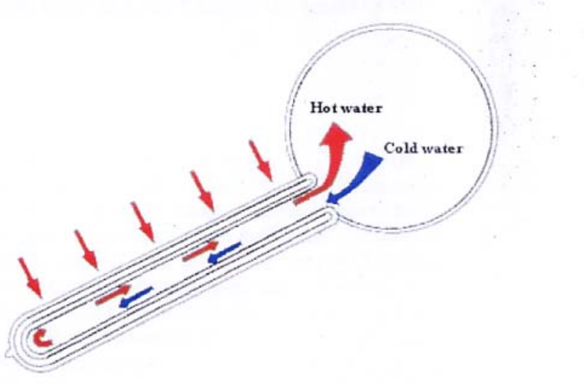


Figure 6. Heat transfer mechanism and the resulting natural convection in water-in-glass collector.

In China all-glass evacuated tubes are now produced in very large quantities, mainly for wet tube domestic water heaters. A water-in-glass solar water heater consists of all-glass vacuum tubes inserted directly into a horizontal storage tank, with water in direct contact with the absorber surface. Figure 5 shows water-in-glass solar water heater. Heat extraction from a water-in-glass evacuated tube is driven by natural circulation of the fluid between the collector and the storage tank (figure 6).

The advantage of the system is that it is a simple and a cheap product. The limitation of the concept is that it can only be used for a low-pressure system, as the tubes can only withstand a few metres of water head and may be sensitive to water hammer (Solar Energy – State of the art). Furthermore, the system does not promoting thermal stratification in the inner tank due to high inlet velocities from the tubes to the storage tank (Budjihardjo et al., 2002).

United States and Canada

In the United States and Canada, vertical storage tanks are used and the systems are based on a pre-heating tank and an auxiliary tank. The energy from the solar collector loop is transferred via an external heat exchanger to a side-arm going from the bottom of the pre-heating tank to the top of the pre-heating tank. The flow in the side arm is driven by natural convection. The pre-heating tank is then connected through a pipe to the second tank with the auxiliary energy supply system. The system concept is shown in figure 7.

This type with a pre-heated storage tank is very simple and is a natural solution for users with existing heating system; furthermore cheap conventional hot water tanks are used. The side arm concept also promotes thermal stratification in the pre-heating tank at high inlet temperatures but the thermal stratification might be destroyed at low inlet temperatures. Two storage tanks increase the overall heat loss and it requires a lot of space.

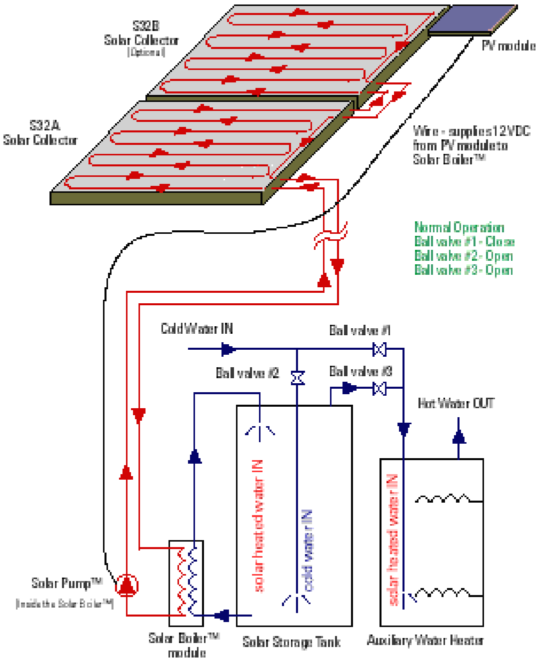


Figure 7. The North-American two-tank system.

Central Europe

Heat storage with a manifold diffuser used in Germany:

To obtain thermal stratification in the storage tank a manifold diffuser is often used. An external heat exchanger is used between the collector loop and tank and the diffuser is mounted inside the tank ensuring that the heated consumption water is distributed near its thermal equilibrium. Figure 8 shows a sketch of the German type of heat storage with a diffuser (Shah, 1999).

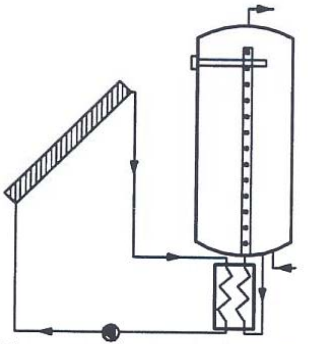


Figure 8 German type with internal diffuser.

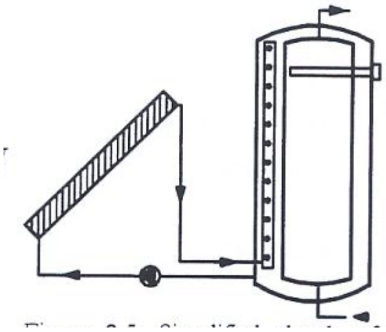


Figure 9. Swiss type with diffuser as a part of collector loop.

Heat storage with a manifold diffuser used in Switzerland:

In Switzerland the heat storage is like a tank-in-tank design and the diffuser can be a part of the solar collector loop ensuring that the solar collector fluid enters the storage near its thermal equilibrium. Figure 9 shows the Swiss tank in tank heat storage design where the domestic water is in the inner tank and the fluid coming from the solar collector enters in the outer tank (Shah, 1999).

The advantage of the two diffuser heat storage designs is that they are having a very high degree of thermal stratification and thus a high thermal performance. On the other hand the systems are very expensive relative to the increased performance (Shah, 1999).

Northern Europe

In northern Europe mainly two different types of heat storage designs are used. The first type uses an internal coil heat exchanger to transfer the heat from the solar collector loop. The internal coil heat exchanger is placed in the bottom of the storage tank as shown in figure 10. The storage tanks with the internal coil heat exchanger do not promote a very high degree of thermal stratification in the storage tank.

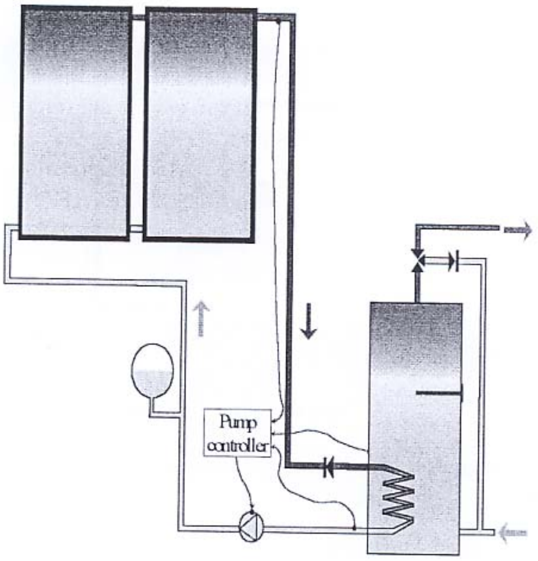


Figure 10. The North European type with internal coil heat exchanger.

The second type uses a vertical mantle heat exchanger to transfer the heat from the solar collector loop (figure 11). The domestic water is in the inner tank and the collector fluid goes into the mantle. The top of the mantle is placed just below the bottom level of the auxiliary energy supply system and the bottom of the mantle is just above the bottom of the tank.

The systems with the mantle tanks use the low-flow principle (the German and Swiss systems with diffusers also use low-flow) with a flow rate in the solar collector loop of 0.15-0.20 l/min/m² collector area and they promote a high degree of thermal stratification. The mantle heat exchanger provides a large heat transfer area and very effective distribution of the collector loop flow over the wall of the tank due to the low flow. Most of the incoming mantle fluid drops down to the thermal equilibrium level in the mantle, and thermal stratification in the mantle and in the inner tank is not disturbed. The cost of the tank is cheap compared to the increased performance initiated by

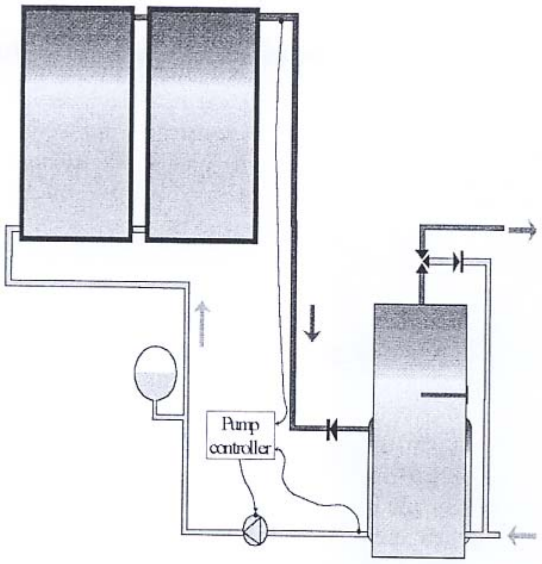


Figure 11. The North European type with a vertical mantle heat exchanger.

the fine thermal stratification (Furbo, 1993). The mantle tanks have a size limit because the heat transfer area gets too small for tanks with volumes over 800-1000 litres.

New trends and concepts presented at the ISES 2003 conference in Göteborg

The papers concerning water based storage for small solar heating systems presented at the ISES 2003 Solar World Congress can be divided into the following parts:

- Investigation of the Chinese water-in-glass concept
- Investigation of a solar-plus-auxiliary single-tank SDHW system with external heat exchanger
- Investigation of a vertical mantle heat exchanger
- Investigation of cold water inlet design
- Control strategies for the auxiliary energy supply system

The first three parts are on specific storage tank concepts while the last two parts are more general and can be applied to most of the storage tank concepts. Heat storage for solar combisystems have not been included in the report, however a few papers on solar combisystems from the ISES 2003 Solar World Congress had a more general approach and conclusions from those will be included in this section.

Water-in-glass solar water heater

Budihardjo et al. (2003) investigated the flow rate in a water-in-glass evacuated tube solar water heater. The investigation was carried out by means of Particle Image Velocimetry (PIV) and Computational Fluid Dynamics (CFD), the experimental and numerical results just at the outlet from the tube to the tank were compared, and good agreement were observed. The fluid from the tubes is entering at a relatively high velocity and these high inlet velocities will result in a fully mixed tank. Inlet velocities between 25-60 mm/s were observed.

The next step is to develop at simpler numerical model of heat transfer and fluid flow in the system based on the CFD results. The further development of the system will be based on this model.

Solar-plus-auxiliary single-tank SDHW system with external heat exchanger

Lin and Harrison (2003) investigated a SDHW system designed around a standard hot water tank with electric heating element and an external heat exchanger with natural convection flow through the tank loop. The motivation for the investigations was an increasing demand for a single-tank system that is more compact and requires less floor area than the two-tank pre-heating system. Figure 12 shows a sketch of the investigated design. The investigations showed that there are problems with flow in the natural convection side mainly at start up in the morning and also problems with small reverse flow in the natural convection loop during the night causing heat loss. To solve the first problem the heat

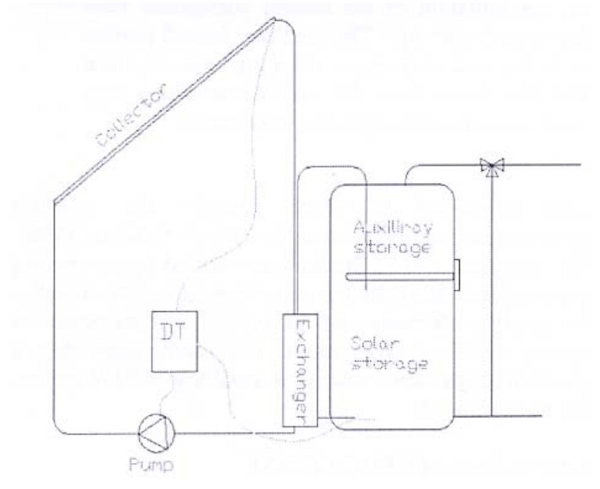


Figure 12. Sketch of the investigated single-tank system.

exchanger-to-tank piping was modified to allow hot fluid returning from the solar collector to heat the return of the natural convection side. Better insulation of the side-arm was proposed to overcome the small reverse flow at night time. However, the system needs more development to work properly.

Vertical mantle heat exchanger
Knudsen et al (2003) investigated the flow structure and heat transfer in a vertical mantle heat exchanger. The investigation was carried out by means of Particle Image Velocimetry (PIV) and Computational Fluid Dynamics (CFD). The investigation showed how the mantle heat exchanger is able to promote thermal stratification in the inner tank both at the mantle level and above the mantle. When the inner tank is heated by the hot fluid in the mantle a slow upward flow (< 7 mm/s) near the tank wall is created and the flow continuous up to the top or to the appropriate temperature level.

Cold water inlet design
Jordan and Furbo (2003a) and Jordan and Furbo (2003b) investigated two different inlet devices for cold water inlets. The two investigated inlet devices were a half ball inlet device with the same diameter as the inlet and flat plate with a diameter to inlet ratio of 7. Thermocouple experiments were carried out and numerical 1-model was developed. The investigation showed a performance reduction rate of 2-9 % for the two inlet devices. Furthermore advanced numerical and experimental investigations of the flow field were carried out. The aim is to find out how stratification depends upon the inlet device geometry for a range of different operating conditions and to predict the influence on the annual performance of the system from different inlet devices.

Advanced control strategies for the auxiliary energy supply system
The idea behind the advanced control strategy is to heat up the water by the auxiliary heater only when the hot water is needed by the user, instead of having a large volume heated to a constant temperature by the auxiliary heater all the time. This strategy leads to a decrease in auxiliary energy supply, decrease in heat loss and an increase in the solar energy contribution. Three different works were carried out on this subject and in all cases the auxiliary energy supply system were designed in a way so it could supply heat to the storage tank from the top and the auxiliary volume could vary.

Furbo et al (2003) proposed two designs, which is shown in figure 13. One system with the auxiliary heating element in a side arm and another system with a horizontal heating element and a vertical heating element inside a pipe, both heating elements are inside the tank.

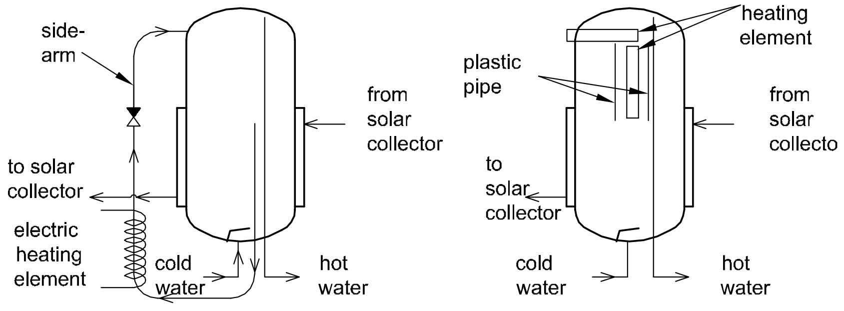


Figure 13. The two smart solar tank designs proposed by Furbo et al. (2003).

Prud'homme and Gillet (2001) proposed a design with a segmented auxiliary heater, the system is shown in figure 14. The auxiliary heater is divided into three vertical elements of different lengths.

The user's needs were predicted every day and weather forecast was used to optimise collector flow rate and minimise the auxiliary heating.

Dennis (2003) also proposed a design with a segmented auxiliary heater. The optimising process is somewhat similar to Prud'homme and Gillet (2001) but the different tariff of electricity during the day was also taken into account.

Al the proposed designs were compared with systems with conventional auxiliary heating. The increase in thermal performance for each system is shown in table 1. Furbo et al. (2003) made the comparison for a lot of different hot water consumption and consumption patters, and the improvement depended strongly on those parameters. A comparison between the proposed designs is difficult because they compared with different systems. Furbo et al (2003) and Prud'homme and Gillet (2001) compared with conventional mantle tank systems with a solar fraction around 35-40 % while Dennis (2003) compared with a typical Australian system with a solar fraction of 20 %. The proposed systems by Furbo et al. had a solar fraction of around 45-60 %, Prud'homme and Gillet's system had a solar fraction of 41 % and Dennis' system had a solar fraction of 37 %.

Figure 14. Smart solat tank design proposed by Prud'homme and Gillet (2001).

	Furbo et al.	Prud'homme and Gillet	Dennis
Solar fraction conventional system	35-40 %	34 %	20 %
Solar fraction of smart systems	45-60 %	41 %	37 %
Increase in thermal performance	5-35 %	19 %	88 %

Table 1. Increase in thermal performance for the different proposed designs compared designs with conventional auxiliary heating.

Trends for solar combisystems

Weiss et al. (2003) presented results from the IEA task 26 concerning solar combisystems. One of the conclusions has interest also for solar domestic hot water systems. There is an emerging trend from very complex systems with a lot piping and connections to a compact system with much less piping and connections. The compact systems might be complex inside but the manufacturer makes this and the installation of the system is relatively simple. Even though solar domestic hot water systems are simpler than solar combisystems they might go in the same direction with natural gas boiler/solar tank units with few connections.

Streicher and Peter (2003) also under the framework of IEA task 26 investigated the energy payback time for solar combisystems. The material demand and accumulated energy expenses (AEE) for the following components were estimated: solar collector, tubing in collector loop (incl. fittings and pumps), heat storage, fluid of collector loop, controller and auxiliary burner (if it was a integrated part of the combisystem). The annual end use energy saved by the different solar combisystems in comparison to a conventional reference system without solar heating were calculated. Based on that the energy payback time was calculated to 1.5-3.5 years for the systems.

The energy payback time would be even lower is it was assumed that the heat storage was necessary in the conventional system. Solar domestic hot water systems will probably have a lower energy payback time.

Conclusion and future developments

A lot of different types of hot water storage tanks are used all over the world. There is lot of variation on how much research have been carried out on the different types. However, all of the different types can be improved further.

For all the different types of storage tanks in order to increase the thermal performance it is important to minimise heat loss by insulation and by avoiding thermal bridges in the upper part of the tanks.

The water based heat storages in the future can be improved if better knowledge about the effect at the cold water inlets is obtained. Advanced control strategy of the auxiliary heater seems to have the potential of being a real improvement in the future. Furthermore the heat storage design can be improved and the installation can be easier if more compact systems (maybe based on natural gas boiler/solar tank units) are developed.

Thermal Energy Storage – Part 2

Phase Change Materials (PCM)

Introduction

As previously mentioned, the two key performance indicators for thermal storage are energy density and relative cost. Increased energy densities allow smaller stores that are cheaper to manufacture and install or may store more energy and hence provide greater flexibility of use at the system level. This part of the report explores a technology used to improve the energy density of a domestic hot water store.

Thermal storage for domestic stores using water relies upon addition of sensible heat to the store. The term "sensible heat" is used when addition of heat to the store results in a rise in temperature of the store. Water has a high sensible heat storage capacity compared to other materials (Table 2). The storage density of water also compares favourably to other less dense materials.

Material	Heat Capacity (kJ/kgK)	Storage Density (kWh/m^3)
Water	4.18	58
Rock	0.88	32
Aluminium	0.96	36
Air	1.0	0.02
Alcohol	2.4	24

Table 2. Storage capacity of materials based on sensible heat gain over 50˚C temperature rise

A change of phase of a material refers to change of state of a material between the phases solid, liquid and gas. The heat absorbed or released as a result of phase change is referred to as latent heat and the change of phase occurs at constant temperature in an ideal material. The change of phase processes are referred to as melting, freezing, boiling, condensing. Some uncommon changes of state may occur eg solid to gas (sublimation) and solid to solid. Considerable exchange of heat is involved in the change of a material's phase. Typical latent heats of phase change are illustrated in Table 3.

Material	Application	Melting Temperature (˚C)	Heat of Fusion (kJ/kg)	Storage Density of melting (kWh/m^3)
Water	Chillers	0	333	93
$CaCl_2.6H_2O$	Space heating and cooling	28	188	89
n-Octadecane Parrafin Wax	Water heating	58	290	57
$Mg(NO_3)_2.6H_2O$	Water heating	61	161	35
Acetamide/Stearic Acid	Water heating	65	218	-
Sodium Acetate	Water heating	58	234	85
PolySiloxane	Flue gas heat recovery	100-200	300	79

Table 3. Storage capacity of materials based on latent heat gain at constant temperature

It is important to note that PCMs have sensible heat exchange when operated at temperatures other than their melting and boiling points and this heat should be included when comparing storage densities to water. When the liquid/gas phase change is used, the boiling point of a material is influenced by the working pressure and hence some flexibility may be gained in the operating state of the system.

The use of PCM is indicated in Figure 15. Once the melting temperature of the PCM is reached, considerable additional heat is absorbed by the material with little rise in its temperature. This means that heat losses from a store using PCM will decrease for the equivalent energy storage with water (which must be raised to a higher temperature).

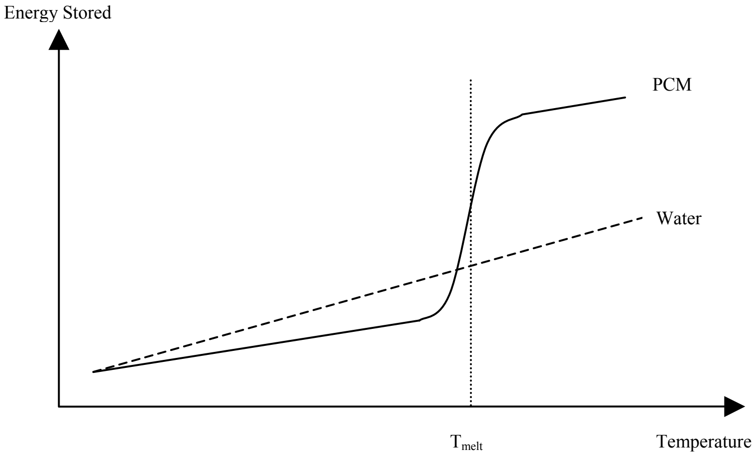


Figure 15. Comparison of PCM and Water based storage

While materials undergo phase changes at a wide range of temperatures, this study will only consider those useful to hot water services. To be useful, the material must undergo a phase change at the hot water service outlet temperature, typically 60˚C. Furthermore, changes of phase from liquid to gas involve large changes in volume of the material and so are difficult to make practical in a closed domestic hot water system. Indeed, there are no attempts to use liquid/gas phase change for PCM storage in the literature. This study now focuses on solid/liquid phase change materials that have a characteristic melting temperature in the range 50-65˚C.

There are two widely used classes of PCM for water heating.

- Paraffin waxes (n-Octadecane)
- Eutectic mixtures of hydrated salts

A range of paraffin waxes may be mixed to give a blend that melts anywhere in the range 50-60˚C. Paraffins are the basis of a cheap PCM, essentially being a waste product of the oil refining industry. Paraffins are non-toxic, although slightly flammable. They encapsulate easily and are non-corrosive. These are significant advantages for domestic water heat storage. The low mass density, low thermal conductivity and high thermal expansion of paraffins limit their energy density and heat transfer capabilities.

The other common PCM family used for this application are hydrated salts. Typically one mixes a eutectic[1] solution of several salts to obtain the desired melting temperature within a range of 5-10˚C. Hydrated salts are also non toxic, non flammable and cheap but can be corrosive due to their electrical conductivity. They have better thermal conductivity and mass density than paraffins. There is some doubt over the long term stability of some salts and additives are often added as preservatives and to promote consistent nucleation of the material during the crystalisation (freezing) process. An example of a commercial product using salts is TEAP TH58 (Table 4).

TEAP TH58 Property	Value	Units
Density of solid	1280	kg/m^3
Heat capacity of solid	4.58	kJ/kgK
Conductivity of solid	0.54	W/mK
Melting Temperature	58	˚C
Latent Heat of fusion	226	kJ/kg
Density of liquid	1450	kg/m^3
Heat Capacity of liquid	2.79	kJ/kgK
Conductivity of liquid	1.09	W/mK

Table 4. Properties of PCM salt TH58

---

[1] A eutectic solution is one in which the melting point of the salt mixture is different (usually lower) than the melting point of either constituent salt

Other non-toxic sodium salts suitable for use with water heating are:

- Sodium Borate.$10H_2O$ (75˚C)
- Sodium Phosphate.$2H_2O$ (75˚C)
- Sodium Thiosulphate.$5H_2O$ (48˚C)

Charge and Discharge of PCM

The charging (melting) of a PCM occurs initially by conduction from the heating fluid, through the heat exchanger wall and into the solid PCM (Lamberg 2003). The charging temperature is usually 10-15˚C above the PCM melting temperature. The thermal conductivity of PCM solids is generally quite low and much lower than water and some sensible heat gain (temperature rise) accompanies the initial charging of a PCM. Once liquid forms, the low thermal conductivity of the PCM creates a convection current within the PCM and this speeds up the heat transfer and melting process. This natural convection is often slowed by the high viscosity of the material or the orientation of the material. The melting is often modelled using the familiar heat diffusion equation although the reader will realise that this is a simplification of the real process.

Discharge (freezing) of a PCM is a much more complicated process. As heat is rejected from the PCM to the heat exchanger, crystallisation occurs near the surface of the heat exchanger. The lower thermal conductivity of the solid PCM then slows further discharge. Typically, discharge of the store takes three times as long as the charging process. Thus an initial power spike is often observed, accompanied by a sensible heat loss as the PCM continues to discharge. A simplified analytical model of this process was attempted by Lamberg (2003) with good results. The displacement of the liquid/solid boundary was predicted with reasonable accuracy.

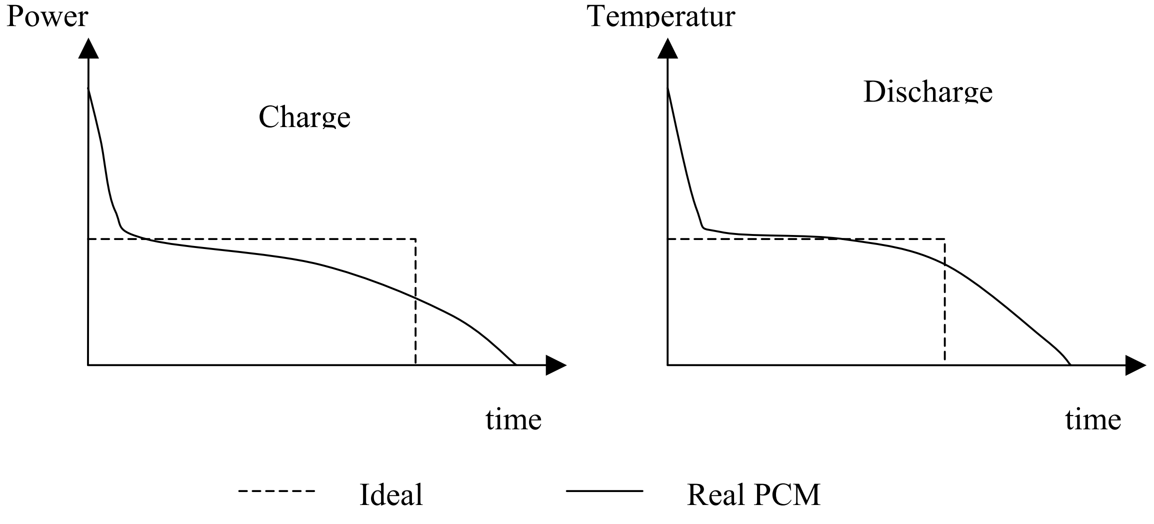


Figure 16. Charge and Discharge characteristics of real PCM materials

Chemical nucleating agents are usually added to the PCM to control the crystallisation process (TEAP, 2003). The aim is generally to prevent stratification in the PCM so that heat transfer can go to complete charge and discharge of the PCM. In practice, heat transfer at the melting temperature is incomplete. Since the PCM melts into a liquid form, a heat exchanger should be used to isolate the PCM and prevent its loss of heat from the store. The design of the heat exchanger is particularly difficult since the discharged PCM is a solid (does not flow). Finned tubes are common although crusting of the PCM has been reported for this type (Lamberg 2003). A novel approach by Santamouris (1988) was to use a rotating heat exchanger, turning at 3 rpm. Crane (1991) and Velradj (1999) used wire mesh and steel rings respectively to modestly enhance heat transfer through the solid material and gain greater heat storage efficiency. Mehling (2000) used PCM impregnated with 12% porous graphite fibre and increased the PCM thermal conductivity from 0.2W/mK to 24 W/mK while only decreasing latent storage capacity by 12%. Commercial products based on aluminium honeycomb and carbon foam encapsulant have reached commercial sale (Pocofoam 2003).

Practicality Issues of PCM Materials

Encapsulation

The role of encapsulation is to contain the liquid PCM while allowing heat exchange between the PCM and water. The most commonly used materials are aluminium, polyolefin and polypropylene. The encapsulant must be able to withstand the stagnation temperature of the solar collector and the corrosive PCM material (hydrated salts) while allowing for cyclic thermal expansion of the PCM. Optimal heat transfer through the encapsulant has lead to the development of micro encapsulation to maximise the surface area to volume of the PCM. More encapsulant is required per unit PCM for this approach.

State of Charge Estimation

The state of charge of a thermal store based on sensible heat may be determined by measuring its temperature profile. The use of PCM complicates estimation of state of charge. If an ideal PCM was used, state of charge would be unknown. A real PCM allows a state of charge guess based on the heat absorption or release *rate* of the PCM. This is a difficult process and warrants further investigation. As optimisation and control is desired for efficient thermal storage, this variable is important.

Stratification

A PCM store's stratification behaviour differs from a conventional store. During times of sensible heat addition, normal stratification is observed although somewhat diminished due to the encapsulation (segregation of the PCM) and its low conductivity and high viscosity. A system designed in this way is intended to remain at or close to the melting temperature at all times. Since the addition or removal of heat does not create a significant temperature change in the material, little stratification and heat transfer occurs between PCM units.

Application to a Domestic Hot Water System

Several attempts have been made to produce a PCM/water thermal storage test facility in the laboratory. At least one commercial unit exists. The interested reader is referred to http://www.latento.de

Nagano (2003) demonstrated a two vessel store where the PCM is stored in a separate vessel to the water and a series of valves and heat exchangers used to store and extract heat. The solar contribution is used as a pre-heater. No auxiliary heating was provided in this system (an oversized collector was used). Auxiliary heating would be applied to the PCM as it is preferable that this be fully charged at all times.

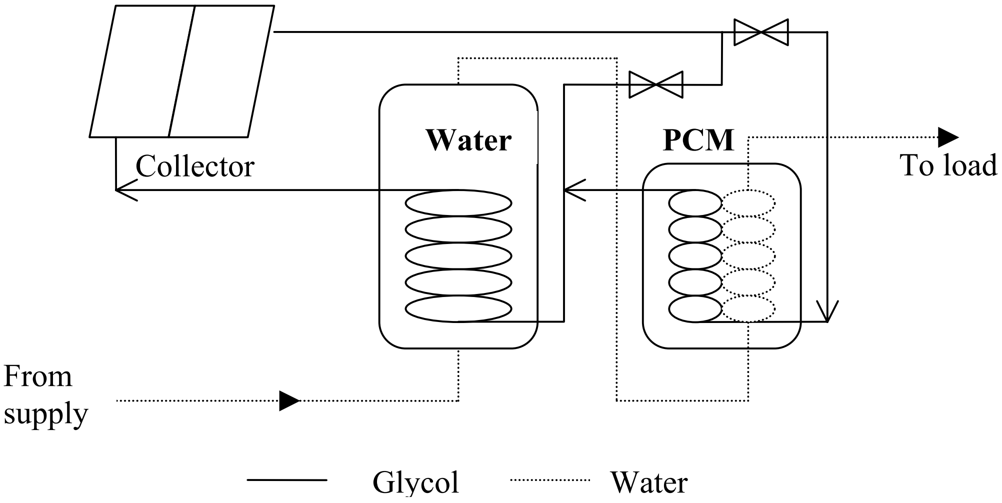


Figure 17. Configuration with separate PCM tank

Auxiliary heating of a PCM store is not a trivial matter. Since heat exchangers must be used with PCM stores, external auxiliary heaters are required. A possible solution is to flood a vessel containing encapsulated PCM material (sitting in baskets) and heat the fluid. Natural convection of this fluid provides the heat exchange and this fluid also circulates through the solar collector and the actual water tank heat exchanger.
Excess heat is diverted to the water tank as this is more efficient than storing the additional energy as sensible heat in the PCM (unless the water temperature exceeds the PCM melting temperature). Poor design of the coil heat exchangers seems to have limited the real performance of this system although modelling predicted a 35% increase in storage capacity for this configuration.

A second method is to use the PCM as an instantaneous heater at the water delivery from the store. This provides the advantage of an integral unit, albeit slightly larger than a normal store. This arrangement (Figure 18.) has the advantage of manufacturability since no modifications are required to the auxiliary heating arrangements and only minor modifications are required to the tank itself. Thermal stratification in the water is preserved well with this method (Mehling 2002).

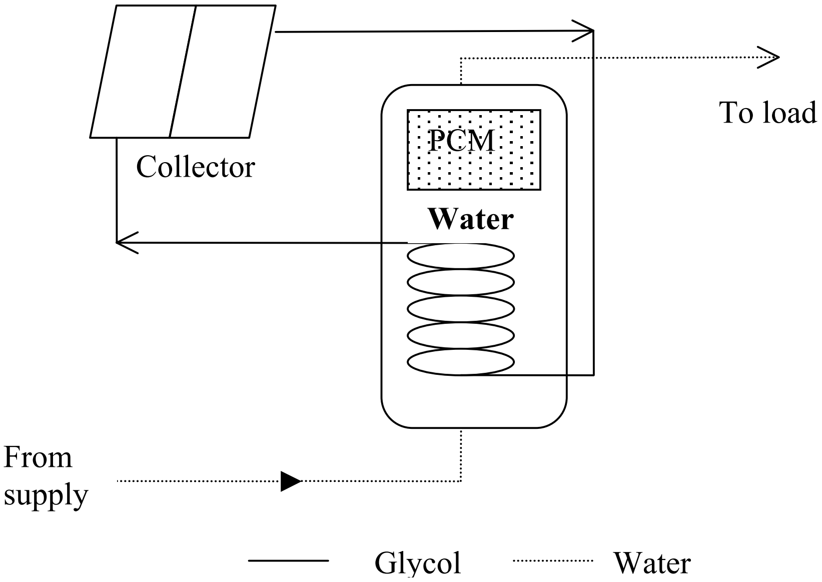


Figure 18. Instantaneous heater from PCM

This configuration is effective at retaining high temperatures at the top of the store for prolonged periods without auxiliary heating. At a hot water draw rate of 10L/min, and a cold store temperature of 20ºC, the PCM would be required to deliver heat at the rate of 27kW. Instantaneous heating of water is very power intensive and this is a substantial challenge for PCM materials with limited thermal conductivity. Materials with enhanced conductivity by means of graphite or metallic foam content show promise (K>25W/mK) although some means of realising consistent freezing behaviour and large heat transfer surface area are required. The space required for this heat transfer may negate the advantages of the compact configuration and thermal stratification achievable and so one of the benefits of PCM use would be lost. Mehling used a small module of 6% tank volume containing sodium acetate impregnated with graphite and demonstrated energy density increases in the range 20-45% and improved localised temperature recovery times..

Supercooled PCM behaviour has been observed in the laboratory whereby the liquid phase has been preserved at temperatures well below the freezing point of the material (Sandnes 2003). Freezing can be triggered in a controlled manner and a very large power release then occurs. The process of freezing occurs by nucleation around a seed crystal. To make use of supercooling, a few seed crystals of the material are preserved during melting, often by high localised pressures that elevate the local melting point (Barett and Benson 1988). When energy release is desired, a mechanical trigger mechanism releases the preserved seed crystals into the melt and freezing occurs rapidly.

This configuration would benefit from stratified top-down heating and advanced control of the auxiliary heating strategy to extract best performance form the PCM.

A third possible configuration, also not found in the literature is to post-heat the solar collectors (Figure 19). In this manner a large fraction of solar heat is recovered since higher collector flow rates may be used and the average collector temperature may be reduced. Solar heated water is heated to useful temperature by the PCM so that the return flow does not adversely affect thermal stratification in the store.

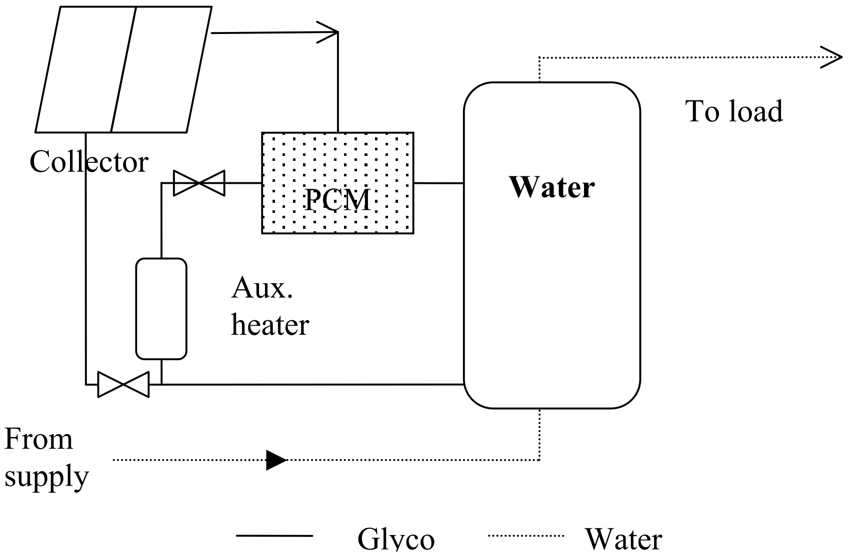


Figure 19. Solar collector post-heat configuration

Again, this configuration will benefit from advanced control. In particular, the PCM would need to be fully charged by sunrise on most days unless there is sufficient insolation to allow the collector to be operated in a mode whereby it can deliver useful heat to the store without *and* not upset any thermal stratification that may be present in the store. The control strategy for this arrangement may be somewhat complicated and warrants modelling.

Opportunities for Future Work

There is a clear requirement for further work on heat exchanger design and enhancement of heat dispersion within the PCM material. This is the most pressing research requirement.

A secondary need is for a method to determine state of charge of PCM stores. Coupled to this is a need to determine the optimum size of PCM capacity for a given set of constraints. Nagano (2003) concluded that the PCM storage capacity should be 30% of the water storage capacity although this is likely to be case specific and dependent upon the collector capacity.

System modelling needs to be carried out to determine the best system configuration for the application(s) of PCM in thermal storage.

It is possible that domestic heat storage using PCM will be surpassed by emerging work on chemical storage. Chemical stores are inherently low temperature devices but their storage efficiency and cost competitiveness are yet to be determined. Thus, work should continue to develop better PCM materials i.e. those of high density and high latent heat capacity.

References

D. Ames, "The Past, Present, and Future of Eutectic Salt Storage Systems", *ASHRAE Journal*, (31:5) pp. 26-28, May 1989.

P. Barrett, D. Benson, “A Mechanism for Nucleating Supercooled Liquids”, *Materials Chemistry and Physics* Vol 20 pp201-208, 1988

D. Buddhi, et al., "Solar Thermal Storage Systems Using Phase-Change Materials", *International Journal of Energy Research,* (12:3) pp. 549-57, July/September 1988.

Budjihardjo I., Morrison G.L. and Behnia M. (2002). Performance of a water-in-glass evacuated tube solar water heater. Proceedings of ANZSES Annual Conference, Newcastle, Australia.

Budjihardjo I., Morrison G.L. and Behnia M. (2003). Measurement and simulation of flow rate in a water-in-glass evacuated tube solar water heater. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

T. Buick, P. O’Callaghan, and S. Robert, "Short-Term Thermal Energy Storage as a Means of Reducing Heat Pump Capacity Required for Domestic Central Heating Systems", *International Journal of Energy Research,* (11:4) pp. 583-92, October/December 1987.

R. Crane and B. Gannesh, "Evaluation of Thermal Energy Storage Devices for Advanced Solar Dynamic Systems" *Journal of Solar Energy Engineering*, (113:3) pp. 138-45, August 1991.

Dennis M. (2003). Optimising energy balance for systems with storage. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

M. Farid and A. Kanzawa, "Thermal Performance of a Heat Storage Module Using PCMs with Different Melting Temperatures", *Journal of Solar Energy Engineering*, (111:2) pp. 152-57, May 1989.

Furbo, S. (1993). Optimum Design of Small DHW Low Flow Solar Heating Systems. Proceedings, ISES Solar World Congress, Budapest, Hungary.

Furbo S., Andersen E., Knudsen S., Vejen N.K. and Shah L.J. (2003). Smart solar tanks for small solar domestic hot water systems. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

A. Ghuniem and S. Klein, "The Effect of Phase Change Material Properties on the Performance of Solar Air-Based Heating Systems", *Solar Energy*, (42:6) pp. 441-447, 1989.

J. Gutherz and M. Schiller, "Passive Solar Heating System for the Perimeter Zone of Office Buildings", *Energy Sources,* (13:1) pp. 39-54, January/March 1991.

M. Hamdan, and F. Elwerr, "Thermal Energy Storage Using a Phase Change Material", *Solar Energy*, (56:2) pp. 183-189, 1996.

D. Hawes, D. Feldman and D. Banu, "Latent Heat Storage in Building Materials", *Energy and Buildings*, (20:1) pp. 77-86, 1993.

Jordan U. and Furbo S. (2003a). Thermal stratification in small solar domestic storage tanks caused by draw-offs. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

Jordan U. and Furbo S. (2003b). Investigations of the flow into a storage tank be means of advanced experimental and theoretical methods. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

Knudsen S., Furbo S., Morrison G.L. and Behnia M. (2003). Analysis of the flow structure and heat transfer in a vertical mantle heat exchanger. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

E. Korin, et al.,"A Novel Passive Solar Greenhouse Based on Phase-Change Materials", *International Journal of Solar Energy,* (5:3) pp. 201-12, 1987.

A. Kurklu, "Energy Storage Applications in Greenhouses by Means of Phase Change Materials (PCMs): A Review", *Renewable Energy*, (13:1) January 1998.

P. Lamberg, K. Siren, "Approximate Analytical model for Solidification in a finite PCM Storage with Integral Fins", *Applied Mathematical Modelling 27* 2003, pp 491-513

Lin Q. and Harrison S.J (2003). Modeling and evaluation of solar-plus-auxiliary single-tank SDHW system with an external, natural convection heat exchanger. Proceedings of ISES 2003 Solar World Congress, Göteborg, Sweden.

H.Mehling, S.Hiebler, F.Ziegler "Latent Heat Storage Using a PCM-Graphite Material", *TERRASTOCK 2000,* Stuttgart, Germany, September 2000.

H.Mehling, L. Cabeza, S.Hiebler, S. Hippeli, "PCM Module to Improve Hot Water Heat Stores with Stratification", *Renewable Energy* Vol 28, pp699-711, 2003

E. Mehmet, "Thermal Performance of a Solar-Aided Latent Heat Store Used for Space Heating by Heat Pump", *Solar Energy*, (69:1) pp. 15- 25, 2000.

Morrison, G.L., Nasr, A., Behnia, M. and Rosengarten, G. (1998). Analysis of horizontal mantle heat exchangers in solar water heating systems. *Solar Energy* 64, pp. 19-31.

K. Nagano et al, "Development of a High Performance Solar Hot Water Supply System using Evacuated Solar Tubes and Latent Thermal Energy Storage", *Proceedings of International Solar Energy Society,* Gotegorg 2003

K. Peippo, P. Kauranen, and P. Lund,"A Multicomponent PCM Wall Optimized for Solar Heating", *Energy and Buildings,* (17:4) pp. 259-70, 1991.

Pocofoam, www.pocothermal.com/html/whatis.html accessed 22/06/2003

Prud'homme T. and Gillet D. (2001). Advanced control strategy of a solar domestic hot water system with a segmented auxiliary heater. *Energy and Buildings* 33, pp. 464-475.

Y. Rabin, et al., "Integrated Solar Collector Storage System Based on a Salt-Hydrate Phase-Change Material", *Solar Energy*, (55-6) pp. 435-444, 1995.

Rosengarten, G. (2000). An experimental and numerical study of mantle heat exchangers for solar water heaters. PhD Thesis, School of Mechanical and Manufacturing Engineering, University of New South Wales.

W. Saman, F. Bruno, E. Halawa, "Thermal Perrformance of PCM Thermal Storage Unit for Roof Integrated Solar Heating System", *Proceedings of International Solar Energy Society,* Gotegorg 2003

I. Salyer, A. Sircar., "Advanced Phase-Change Materials for Passive Solar Storage Applications", *Proceedings of the 20th Intersociety Energy Conversion Engineering Conference,* Miami Beach, FL, August 18-23, 1985, pp. 3.699-3.709.

I. Salyer and A. Sircar, "A Review of Phase Change Materials Research for Thermal Energy Storage in Heating and Cooling Applications at the University of Dayton from 1982 to 1996" *International Journal of Global Energy Issues*, (9:3), 1997.

B. Sandnes "Triggering crystallisation in Supercooled Liquids", Department of Physics, University of Oslo, N-0316 Oslo, Norway

M. Santamouris and C. Lefas, "On the Coupling of PCM Stores to Active Solar Systems", *International Journal of Energy Research,* (12:4) pp. 603-10, October/December 1988.

K. Sasaguchi and R. Viskanta, "Phase-Change Heat Transfer During Melting and Resolidification of Melt Around Cylindrical Heat Sources/Sinks", *Journal of Energy Resources Technology*, (111:1) pp. 43-49, March 1989.

Shah, L.J. (1999). Investigation and modeling of thermal conditions in low flow SDHW systems. PhD Thesis, Department of Buildings and Energy, Technical University of Denmark. Report-034.

Solar Energy – state of the art (Chapter 5), 2001. International Solar Energy Society. James & James Ltd.

Souliotis M. and Tripanagnoustopoulos Y. (2003). CPC Type ICS Solar Water Heaters. Proceedings ISES Solar World Congress, Göteborg, Sweden.

Sthreicher W. and Peter M. (2003). Material demand and energy payback time of solar combisystems. Proceedings ISES Solar World Congress, Göteborg, Sweden.

M. Syed et al., "Thermal Storage Using Form-Stable Phase-Change Materials" *ASHRAE Journal*, (36:5) pp. 45-50, May 1997.

TEAP, *www.teap.com.au*, accessed 02-06-2003

J. Tomlinson, C. Jotshi and D. Goswami, "Solar Thermal Energy Storage in Phase Change Materials", *Solar '92: American Solar Energy Society Annual Conference,* Cocoa Beach, FL, June 15-18, 1992, pp. 174-79. Available from the American Solar Energy Society, 2400 Central Avenue, Suite G-1, Boulder, CO 80301; Phone: (303) 443-3130; Email: ases@ases.org; World Wide Web: www.ases.org .

M. Vandini, "The HVAC System Choice with Cool Storage" *Energy Engineering,* (87:6) pp. 34-45, 1990.

R. Velradj, R Seeniraj, B.Hafner, C.Faber,K.Schwarzer, "Heat Transfer Enhancement in a Latent Heat Storage System", *Solar Energy*, Vol65 No3, pp171-180

Weiss W., Suter J.M. and Letz T. (2003). Comparison of system designs of solar combisystems. Proceedings ISES Solar World Congress, Göteborg, Sweden.

D. Wylie, "Evaluating and Selecting Thermal Energy Storage", *Energy Engineering,* (87:6) pp. 34-45, 1990.

I. Zsigrai et al, "Thermochromic Complex Compounds in Phase Change Materials: Possible Application in an Agricultural Greenhouse", *Solar Energy Materials & Solar Cells*, (51:3-4), February 1998

Solar Collector Materials

State of the Art and Further Development

M. Reza Nejati
Energy Systems Engineering group, Department of Mechanical Engineering,
K.N.Toosi University of Technology, Iran
and
V. Fathollahi
Center for Renewable Energy Research Application (CRERA) AEOI, Iran

Solar collector materials, State of the art and further development

M. Reza Nejati[a], *V. Fathollahi*[b]
[a] *Energy Systems Engineering group, Dept. of Mechanical Engineering, K.N.Toosi University of Technology, P. O. Box 16765-3381, Tehran, Iran*
[b]*Center for Renewable Energy Research and Application (CRERA), AEOI, P.O. Box 14155-1339, Tehran, Iran*

Introduction:

Most of the research on solar materials has been analytical or developmental, with very little new theoretical work available. Most optical theories and microstructure models are based on work done by physicists quite some time ago. This study covers the primary solar materials including selective absorbers, transparent low emittance coatings, and antireflection films.

Solar absorbers are modelled by several theories, classical interference-film theory is used to model semiconductor/metal tandems. Effective-medium theories are used to model graded composites and optical-trapping surfaces are modelled by scattering theory. Multilayer transparent low emittance and anti reflection films rely on interference-film theory. Semiconductor low emittance films and reflectors are characterized by Drude free-electron theory. In the following sections, we would briefly review various photo thermal conversion mechanisms in solar absorber surfaces. Also, we would outline the recently developed fabrication techniques and superior structures for each mechanism.

1-Absorber Surfaces:

Solar absorber surfaces have been researched the most extensively over the last decade compared to the other collector materials. Much of the theory and modelling of materials has also been confined to this area. There are two categories of absorbers: **selective and non-selective**. The selective absorber has optical properties that vary distinctly from the solar to the thermal infrared spectral regions. The optical properties of a selective absorber are responsible for its high solar absorptance and low thermal infrared emittance. An idealized spectral response of a selective absorber is shown in figure (1)

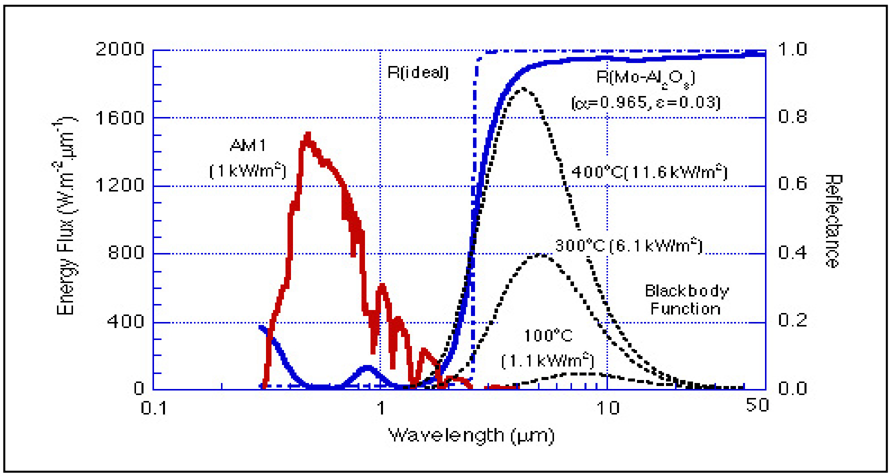


Fig-1 the solar radiation spectrum at AM 1 and black body radiation at temperatures of 100, 300 and 400 C, Reflection Spectrum for an ideal solar selective absorber and reflection spectrum for a simulated Mo-Al_2O_3 selective absorber

In contrast to the selective absorber, the non-selective absorber has optical properties that varies slightly over the solar and thermal infrared spectra. There are numerous examples of non-selective absorbers. Most of them belong to the categories of paint, conversion coatings, and carbon filled plastics. The application of non-selective absorbers has been confined to low temperature processes. The theory and modelling of these materials have not yet been explored.

Solar selective absorbers can be classified as follows:

- Absorber-Reflector tandems: Semiconductor/Metal tandems, Graded-composite/Metal Tandems
- Interference multilayers
- Optical-trapping surfaces (wave front discrimination)

Many detailed reviews have been written on all classes of solar absorbers. As an example niklasson and granqvist provided an excellent annotated bibliography on solar absorbers that covers to the end of 1981.[1]

1-1 Absorber-Reflector tandems:

An absorber-reflector tandem is a combination of two materials each of which having certain optical function and together make a solar selective absorber. Such absorber surfaces are made of a highly reflecting metallic substrate in the IR wavelength range on which a coating with high absorptance in the solar wavelengths have been deposited.

Usually the underlying layer is made up of Cu, Ni, Mo, Al, SS or Ag, which are of low thermal emittance of about 0.1 at 300 C. These IR reflector layers can also play the role of a protective layer for the underlying metal against corrosion. This kind of absorbers necessarily should have high thermal conductivity to keep its surface temperature as low as possible.

1-1-1 Semiconductor/Metal tandems:

Some semiconductors such as Si, Ge and PbS whose transition wavelength is about $2\mu m$, can be used as solar absorber layer in absorber-reflector tandems. In semiconductors, solar absorptance is the result of interaction of incident photons whose energies are higher than semiconductor's band gap. The absorbed energy of photon is utilized to transfer an electron from valance to conduction band. Obviously, photons with lower energy would be transmitted through the material with no effect. Since the suitable semiconductors for absorbing the solar radiation are of higher refractive index, one should use antireflection films to prevent high surface reflectance of the semiconductor film. This kind of absorber is shown in figure 2 schematically.

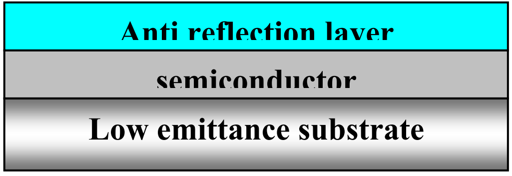


Fig-2 schematic diagram of semiconductor/Metal tandems as an selective absorber

Si can be deposited both in crystalline and amorphous form by CVD technique (2-4) this coating are highly resistant to temperature up to 800C and have a solar absorptance of 0.8 and thermal emittance of lower than 0.1 at 500 C.

1-1-2 Graded and ungraded composite (cermet)/Metal absorbers:

Recently, cermet composites (ceramic- metallic composite) have been extensively investigated for application as solar selective coatings. These kinds of composites are formed by embedding small metal particles (5-10 nm) in a dielectric (ceramic) matrix. Au-MgO, Ni-Al_2O_3, Cr-Cr_2O_3 (black chrome), Pt-Al_2O_3, Mo-Al_2O_3, SS-AlN, Ni-NiOx (black Nickel)… are just examples of this category which are resistant to temperatures up to 400 C however their thermal emission is higher than 0.1. Following figure (fig-3) shows a schematic cross-section of two commercial absorbing surfaces, which use cermets as solar absorber. One of these absorbers is composed of metal particles that are uniformly dispersed in anodised aluminium (Ni-Al_2O_3) and the other shows a composite whose metallic component is graded throughout the cermet layer (Ni-NiO_x). The latter is deposited by sputtering technique.

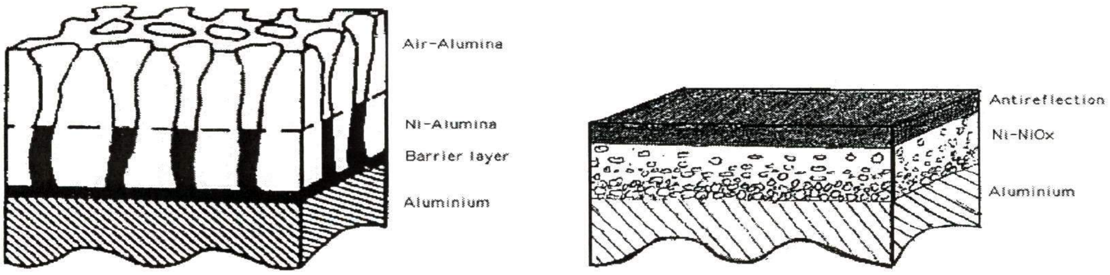


Fig-3 Schematic cross section of two commercial selective absorber surfaces. The left figure shows an absorber surface (Ni-Al_2O_3), which has been fabricated by electroplating. The right figure shows a cermet absorber surface, fabricated by sputtering technique (Ni-NiO_x) in contrast to the electroplated sample, metallic particles in this absorber has been graded.

It is clearly seen from the results obtained by investigators over the last three decades[5-7] that absorbers with one homogeneous cermet layer have low emittance, but the absorptance is not usually high enough for practical applications as a selective surface. To achieve high absorptance (>0.90), graded composite absorbers have been used by most workers. However, these absorbers incur higher thermal emittance[8-10].

For high temperature solar energy conversion applications, cermet components should have a high decomposition temperature. This is why metals such as W, Mo and SS are commonly used as the metallic component of cermets and dielectrics such as Al_2O_3 and AlN as the ceramic component of cermets[11, 12].

Cermets for solar applications usually consist of nanometer-sized metal particles (~1-20 nm) embedded in a ceramic binder. This material is deposited over a metallic substrate and may be covered with an anti-reflection coating to enhance the solar absorptance. Also, it has been found that grading the concentration of the metal particles from low metal particle density at air/cermet interface to high metal particle density at the cermet/substrate interface, improves the spectral selectivity of the coatings. Recently Q. Zhang has shown that, a system with two cermet layers with different metal concentrations provides the best optical results.

The selectivity of cermets is based on a tandem effect: the cermet itself absorbs radiation strongly in the region before the cut-off wavelength (between 2 and 3 micrometers) and is almost transparent in the IR wavelengths. Meanwhile, the metallic substrate provides high IR reflectance and contributes a small portion of the absorption in the ultraviolet (UV) and visible regions.

During the last decade, DC and RF reactive sputtering technologies have extensively been used for the deposition of cermet selective coatings. A major disadvantage of reactive sputtering is that one should be able to precisely control the reactive gas partial pressure, which in turn needs an accurate gas analyser. On the other hand, non-reactive sputtering is much simpler due to lack of reactive gas. However in this mode of sputtering, compound-sputtering targets should be available. It should be noted that compound dielectric targets could only be sputtered by RF sputtering systems [18].

Due to the variety of available materials as metallic and ceramic components of a cermet and also due to the possibility of varying optical properties of the resultant cermet composed of two certain materials, cermets have been extensively investigated in optical research in recent years. Besides, recently scientists have paid much attention to this category of materials, their fabrication methods and also their applications as solar selective absorber coatings. If the metal grading of the cermet instead of air ends to a dielectric whose refractive index is between air and composite, the solar absorptance would be improved further. For this reason some anti reflection dielectric films, such as SiO_2, Al_2O_3 and AlN are commonly used. Farooq and Hutchins 2003 have shown that the grading modes of metallic contents in cermet layer have strong effect on optical properties of absorber[19,20]. By using a mathematical series they have proved that a cermet layer that is primary graded cermets would have better optical properties than linear graded cermets.

1-2 Paint coatings:

Paint coatings are very attractive due to the ease of fabrication and application and also their reparability. However, their application is limited due to the rather high thermal emission, which is originated from the optical characteristics of polymer binder.

The selective paint coatings have been recently commercialised and Z-solarect from Slovenia and SolkoteHI7SORB-II from United States are two well-known trademarks.

A method in designing solar selective paints is using coated metallic flakes. In fact, in this design the spectral selectivity is obtained by these coated metallic flakes, which could be considered as many absorber-reflector tandems dispersed in the medium.

An example of pigments (mostly metal oxides) deposited on aluminium substrate is FeMnCuOx with a silicon binder (siloxane), which has a solar absorptance of 0.92 and thermal emittance of 0.13. The selectivity of such paint coatings not only depends on the thickness of the film and volume fraction of pigments but also depends on the dispersion and pigment size in the matrix**.**

This kind of paint, which absorbs solar radiation and reflects thermal IR radiation and prepared from a mixture of various pigments, represents a great challenge for those working on development on new pigments, resins, paint additives and formulation of new types of solar absorber paints.

In the past most of the work was devoted to the formulation and preparation of thickness sensitive spectrally selective (TSSS) paint coatings, which gain their selective properties in combination with the optical properties of the substrate. To exploit the low emittance of the metal substrate the paint coatings need to be transparent for thermal IR radiation but absorbing enough for solar radiation. Nowadays, this is achieved with the use of coarse-grained inorganic pigments incorporated in the durable and high enough IR transmitting resins. The best results have obtained with heat-treated aryl or alkyl substituted polysiloxane resin binder systems because they exhibit a remarkable stability and good adhesion to the Al substrate. However due to the need for controlling the thickness of the paint layer, this type of TSSS coatings is not widely used. It should be mentioned that only a newly developed coil-coating deposition technique could fulfil all the requirements for application of the paint at sufficiently high temperatures (300C) and short heat treatment times (about 30 s) [21].

The new developed selective paint coatings insensitive to thickness (TISS) can be a better choice and has more extensive applications. It has been shown that metallic pigments with large lateral dimensions (about up to 50 micrometers) are the primary conditions for obtaining low emittance of paints. This new TISS paint coatings having Al flakes with an outer layer of Iron Oxide allow to satisfy two requirements for coatings of solar building facades: the colour of the coatings would be much attractive than the usual black/grey and thermal emittance would go down to 0.5-0.58. Unfortunately due to the non-black colour of the coatings, the solar absorptance is usually about 0.8, which is not of interest. Although the value of absorptance is sacrificed, the large area of the building faced could sufficiently compensate the decrease in alpha value. Besides it may be possible to roughen the surface of paint coatings at certain conditions to obtain higher values for solar absorptance, but no systematic work has been done yet to evaluate the effect of surface roughness on the spectral selectivity of TISS paint coatings[21].

1-3 Multilayer interference films (Multilayer Stack):

Multilayer thin films, called multilayer stack interference films, are a complex compound of metal/dielectric films that work as a selective filter for solar radiation absorption. The effect of multilayer interference films is trapping the solar energy within the alternating semi-transparent metal-dielectric-metal films through the phase interference phenomena. In this kind of absorbers, the desired effect is resulted by frequent passing of solar radiation through the dielectric film sandwiched between two reflecting metal surfaces.

The upper metallic film is semi transparent which lets the radiation pass through at first. Then it acts as a mirror for the reflected radiation. The exact determination of semi transparent layer thickness and choosing suitable material for multilayer are two important factors that should be considered. Naturally, other wavelengths, which do not correspond with the absorption frequency of this multilayer, are reflected. A multilayer of Al_2O_3-Mo-Al_2O_3 is an example of such absorbers. other examples are outlined in table 1 and their optical results in figure (5)

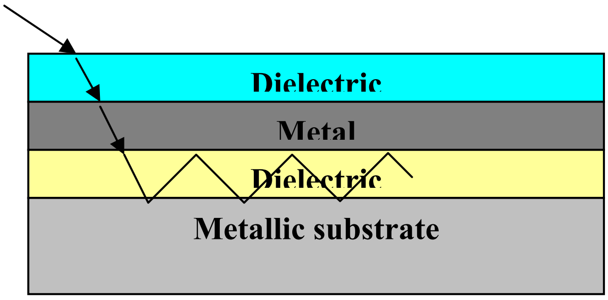


Fig-4 schematic diagram of a multilayer interference absorber

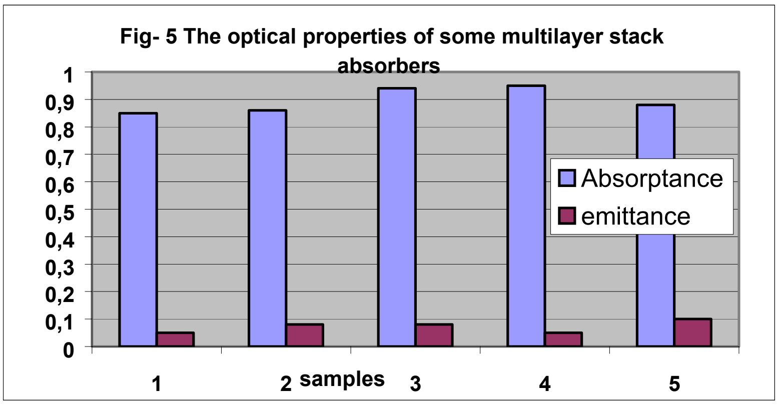


Fig- 5 The optical properties of some multilayer stack absorbers

Table-1 some samples of interference multilayer absorbers

1	MgF2-Mo-CeO2/Mo sheet VE
2	SiO2-Mo-SiO2/Mo VE
3	Al2O3-Mo-Al2O3/Mo (VE)
4	SiO2-Al-SiO2 multilayer stack (VE)
5	SiO/Cr/SiO (VE)

1-4 Wave front discrimination:

Spectral selectivity of a metal surface can only be developed by geometrical methods. Surface irregularities such as porous and grooves with the size comparable to the incident radiation wavelength would easily increase the solar absorptance by multi reflections. Meanwhile thermal IR emission would be slightly increased[22-31].

The irregularities with size comparable to the transition wavelength (the maximum wave length which should be absorbed) only increase the solar absorptance and have a little effect on thermal emission. This different optical behaviour in visible and IR wavelength range can be explained by "wave-front discrimination" phenomena. For visible wavelengths, which are small in comparison with real irregularities, the surface seems rough and radiation would be trapped through forward reflections and partial absorption in micro size grooves. But for IR thermal radiation, as the wavelengths are greater than the irregularity's size, the surface looks like a smooth mirror and IR radiation could be reflected without any absorption.

In the following figure a schematic diagram of solar and thermal radiation and the size of grooves have been shown(figure 6).

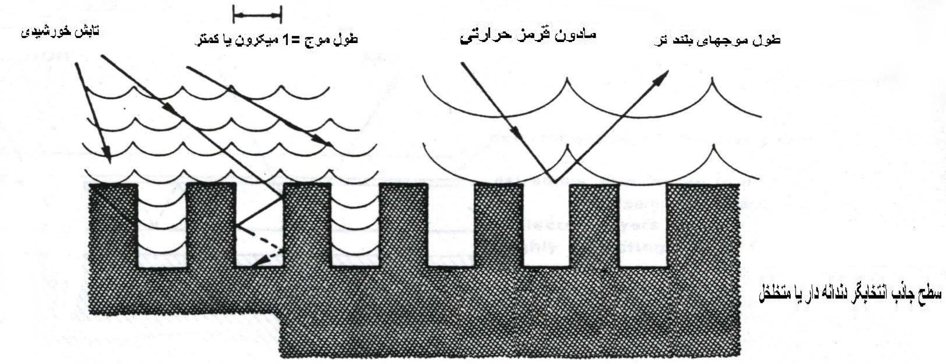


Fig-6 effect of wave front discrimination by roughly absorber surface

Some examples of wave front discrimination textures are: dendritic porous and needle shaped textures. A schematic view of radiation trapping in dendritic absorber has been shown in figure (7).

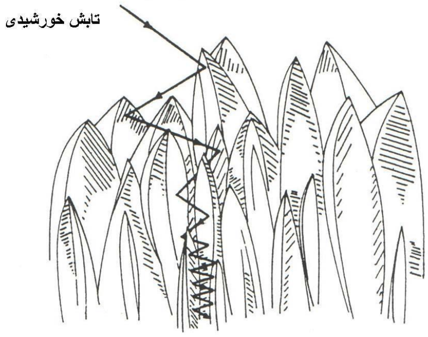


Fig-7 A schematic view of solar dendritic absorber. The incident radiation might be reflected up to 15 times before complete absorption.

Since a thin film deposited on a textured surface has the same roughness of the substrate, one can conclude that in a solar absorber film, the spectral selectivity could be improved by proper choice of the substrate's roughness.

A newly developed low cost absorber using wave front discrimination phenomenon has been recently described by P. Konttinen et al [33]. The material of their absorber is ($C/Al_2O_3/Al$) in which, carbon clusters are adhered to the Al sheet. In their work, the pure Al substrate has been mechanically grinded by some grinding pad. As schematically shown in figure (8), the Al substrate sheet moves back and forth under the grinding pad in a relatively slow motion. During the grinding process the grinding pad is saturated with carbon dust, which is bound to the pad by static electricity.

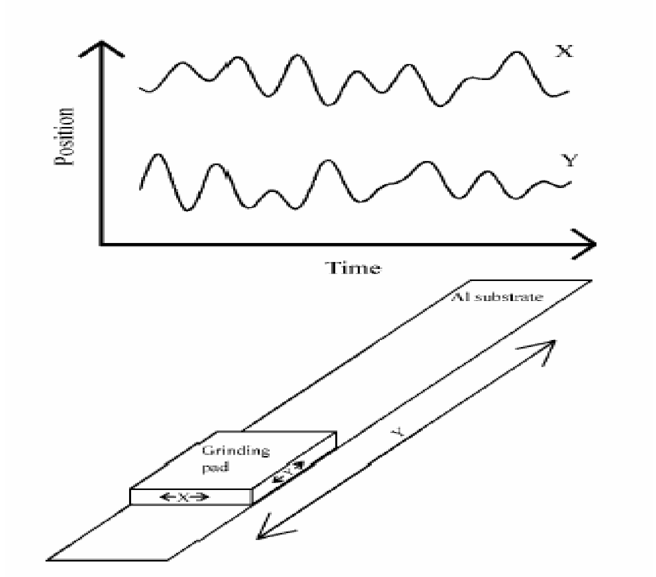


Fig-8 The non-correlating classical random noise used for moving the grinding pad in X/Y directions across the substrate (left and right). The substrate moves back and forth slowly in y-dimension

Carbon dust reacts with the surface being scratched (containing Al_2O_3 and unoxidized Al) and atmospheric oxygen forming a matrix structure on the final surface. This structure contains mainly elements Al, O and C, and it covers the surface as a black or dark grey layer. Changing the size of the grinding particles, grinding pressure, speed and time has strong effect on the optical properties of the forming surface. By this method, a solar absorptance of 0.90 and a thermal emittance of 0.22 have been achieved so far.

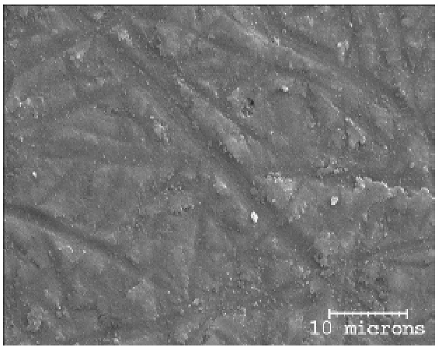


Fig-9 SEM photograph of a typical grinded surface 2000 X

1-5 Intrinsic Selective Material:

The number of materials, which are itrinsically selective, is very limited. are two examples of these materials. The pure tungsten is one of the best naturally selective metals. As shown in figure 10 although W is a relatively good intrinsic selective absorber, its reflection spectrum is far from that of an ideal selective absorber.

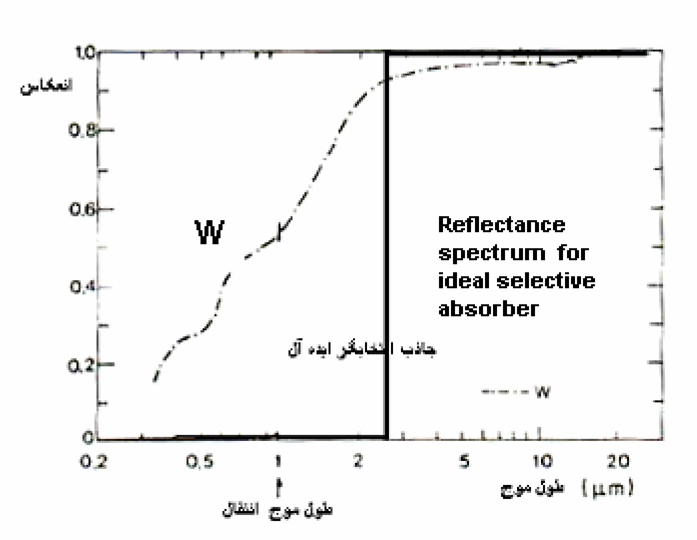


Figure-10 the reflectance spectra of an Ideal solar selective absorber (solid line) and that of a 0.98 pure W sample (dotted line)

As can be seen in the above figure the transition from low reflectance region to high reflectance regions occurs in lower wavelengths $(1\mu m)$ in comparison to an ideal selective surface. In contrast for most semiconductors, this transition occurs in higher wavelengths.

1-6 Heat Mirrors:

Sometimes instead of using a selective absorber, an ordinary black surface is used. However to obtain a spectral selectivity, a material transparent in visible wavelengths but highly IR reflector, is deposited over the ordinary black absorbing surface. A schematic diagram of such a surface has been shown in figure (11). This coating prevents the radiation loss of black surface but at the same time lets the solar radiation to reach the black absorbing layer and be absorbed. This kind of coatings is known as heat mirrors. Creating the spectral selectivity by using heat mirrors would be a good interchange for solar selective absorbers, although the heat loss in heat mirrors is comparatively high in comparison with solar selective absorbers.

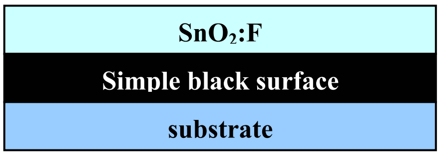


Fig –11 the schematic diagram of layers in Heat mirror

1-7 Non- or Semi-selective Coatings:

To reduce the fabrication cost of solar collector, there are various absorber surfaces other than selective absorbers that can be used for domestic applications, called non-or semi-selective coatings. This kind of absorber has the lowest price and can be easily used. As an example black opaque paints (organic or inorganic), ceramic or organic enamels and chemically or electrochemically modified metallic coatings are of this kind absorbers. One of the most important characteristics of absorber surfaces using black paints is that these surfaces have high solar absorptance even in not normal incident angels. But unfortunately they are of high thermal emittance as well. For the same reason the criteria for choosing non-selective coatings are : higher absorptance, reasonable stability and lower price. The pigments, which are conventionally used as black paints are: Black carbon, Fe_3O_4, Amorphous graphite and asphalt.

The ceramic enamels are of higher stability at higher temperatures compared to organic enamels or paint coatings and have higher selectivity as well. Chemical conversion coatings are of several advantages. These coatings are inexpensive and readily produced in a few minutes.

Different kinds of such coatings on various semi selective substrates such as steel, copper or aluminium can be easily fabricated. Chemical conversion is dipping a metallic surface in a strong oxidizing or sulphidizing solution that would lead to the formation of an oxide or sulphide coating on a metal substrate.

Results:

1-The best fabrication method of solar selective absorber, considering the economic factors depends on the application of the collector.

2- In sequenced graded structures, using double optimised cermet structure, would lead to better optical properties than that of graded cermets.

3-Integral graded structure using prime grading has better optical properties than the linear graded one.

4-Choice of proper roughness and morphology of the surface would lead to higher selectivity and if it is combined with the successful structures outlined before, an ideal solar absorber coating may be achieved.

5-If the cost is so important to us then the use of antireflection coating together with the novel design recently proposed by Konttinen would be the best choice.

References:

1-Niklasson, G. A. and C.G. Granqvist. 1983a. Solar Energy Mat. 7:501.

2- Seraphin, B.O., J. Jpn. Soc. Appl. Phys. 44, 1975, PP.11.

3- Idem, Thin Solid Films 39, 1976, PP. 87.

4- Janai, M.; Allred, D.D.; Booth, D. C. and Seraphin, B.O. Sol. Energy Mater. 1, 1979, PP. 11.

5- Niklasson G. A. and Granqvist C. G., J. Appl. Phys., 55, (1984), 3382.

6- Garnich F. and Sailer E., Sol Energy Mater., 20, (1990), 81.

7- Thomas L. K. and Tang C., Sol Energy Mater., 18, (1989), 117.

8- Zhang Q. C. and Mills D. R., Solar Energy Mater. Solar Cells, 27, (1992), 273-290.

9- Zhang Q. C., Solar Energy Mater. Solar Cells, 52, (1998) 95-106.

10- Zhang Q. C. and Mills D. R., J. Appl. Phys., 72, (1992), 3013-3021.

11-Zhang Q. C., J. Phys. D: Appl. Phys., 34, (2001), 3113-3120.

12- Zhang Q. C. and Mills D. R., J. Appl. Phys., 72, (1992), 3013-3021.

13- Zhang Q. C. and Mills D. R., Solar Energy Mater. Solar Cells, 27, (1992), 273-290.

14- Zhang Q. C., Solar Energy Mater. Solar Cells, 52, (1998) 95-106.

15- Zhang Q. C. and Mills D. R., J. Appl. Phys., 72, (1992), 3013-3021.

16- Teixeira V. et al., Thin Solid Films, 392, (2001), 320-326.

17- Zhang Q. C., J. Phys. D: Appl. Phys., 34, (2001), 3113-3120.

18- Ohring M., "The Materials Science of Thin Films", (1992), Academic Press Inc.

19- Farooq M. and Hutchins M. G., Solar Energy Mater. Solar Cells, 71, (2002), 73-83.

20- Farooq M. and Hutchins M. G., Solar Energy Mater. Solar Cells, 71, (2002), 523-535

21-B. Orel, A. Vilcnik, B. Jelen, M. Kohl and Franz Brucker, Procceding ofISES 2003 coneference Gotheborg, sweden

22-Cohen, R.W. ; Cody, G. D. ; Coutts, M. D. and Abeles, B. Phys. Rev. B8 PP. 3689, 1973.

23-Craighead, H.G. and Buhrman, R.A. , J. Vac. Sci. Technol. 15, PP. 269, 1978.

24-Tabor, H. , Transactions of the Conference on the Use of Solar Energy, Vol.2, Tucson, Arisona, PP. 1A, 1956.

26-Mar,H.Y.B. and Liu, J. H. , Optical Coatings for Flat Plate Solar Collectors, Final Report ERDA, 1975.

27-Lampert, C.M. and Washburn J., Sol. Energy Mater. 1, PP.81, 1979.

28-Sowell, R. R. and Mattox, D. M. , Plat. Surf. Finish. 65, PP. 50, 1978.

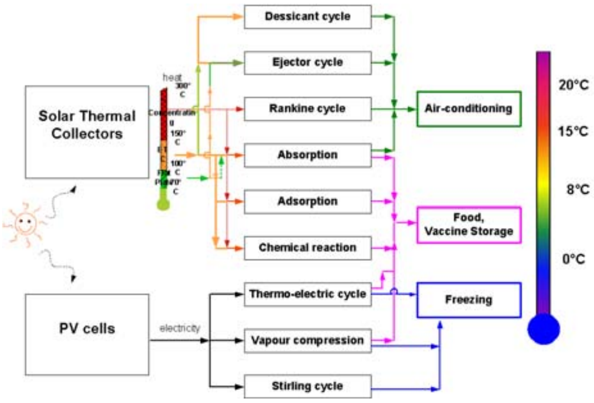
29- Ignatiev , A. ; Oneill , P. ; Doland, C. and Zajac, G., Appl. Phys. Lett. 34, PP. 42, 1979.

30- Granqvist. G. and Niklasson , G. A. , J. Appl. Phys. 49, PP.3512 , 1978.

31-Hollands, K. G. T. , Sol. Energy 7, PP. 108, 1963.

32-Konttinen P., Kilpi R., Thin solid Films 425 (2003) 24-30.

Solar C☼☼ling



Wimolsiri Pridasawas
Royal Institute of Technology, Stockholm, Sweden
and
Teclemariam Nemariam
Royal Institute of Technology, Stockholm, Sweden

June 27, 2003

Assignment for Ph.D. Course: Solar Heating
Technical University of Denmark (DTU)

TABLE OF CONTENTS

1. Introduction 3

2. Performance of the solar-driven cooling system 4

3. Solar cooling technologies 6

Thermal/work driven system

3.1 Absorption refrigeration cycle 8
3.2 Adsorption refrigeration cycle 11
3.3 Chemical reaction refrigeration cycle 13
3.4 Desiccant cooling cycle 14
3.5 Ejector refrigeration cycle 16
3.6 Rankine's cycle 17

Electricity (Photovoltaic) driven system

3.7 Stirling refrigeration cycle 19
3.8 Thermo-electric refrigeration cycle 22
3.9 Vapour compression refrigeration cycle 24

4. Review of the papers from the ISES conferences on solar cooling Technology 26

References 30

1. INTRODUCTION

The demand for refrigeration increases due to the basic need of food and vaccine storage, for human thermal comfort and the development of the global technology. The international Institute of Refrigeration (IIR) has estimated that approximately 15% of all electricity produced worldwide is used for refrigeration and air-conditioning processes of various kinds (Lucas, 1998). Regional shortage of electricity forces the price of electricity to be high. Global warming and ozone depletion are the main global environmental issues. Due to the fact that the cooling load is generally high when solar radiation is high together with the existing technology that solar energy can be converted to either electricity or heat. These make it possible to build a solar-driven refrigeration machine.

The first recorded solar-driven machine was in 1872 by Albel Pifre, in Paris, for producing a small amount of ice. At the beginning of the 20th century, many countries were interested in using solar energy but the technology was focused only on the heating of water or air. Research in 'solar air-conditioning' increased after 1965. During the first oil crisis in 1973, the air conditioning system was considered as a luxury and an unnecessary system. The attention was focused on improvement of design and efficiency of the refrigeration system rather than investigation of it as a novel system.

The absorption refrigeration system using lithium chloride (LiCl) was developed, but it was considered as an inappropriate system for a residential sector due to its complexity. In 1976 around 500 solar-driven air conditioning systems were installed in USA, most of them were absorption systems using lithium bromide (LiBr). In a hot and humid area such as India, the adsorption system was applied using a molecular sieve material such as zeolite or calcium chloride ($CaCl_2$) for reducing the water content in the air.

The International Energy Agency (IEA) set up the 'Solar Heating and Cooling' program in 1977, which is still active as of present (2003). The solar cooling is focused on task 25, "Solar Assisted Air Conditioning of Buildings" which was started on June 1, 1999 and will end on May 31, 2004 (IEA, 2003). In 1982, the International Institute of Refrigeration (IIR) held a conference in Jerusalem on the topic 'Solar energy for refrigeration and air conditioning'. There were about 200 attendees with 40 papers discussed (IIR, 1982).

Photovoltaic (PV) technology has been integrated with a vapor compression system by using a DC-compressor adapted from a DC-pump. Research in a Peltier's cooling effect integrated with PV has also developed, primarily for the cold chain project of the World Health Organization (WHO) and the international Health Organizations. Most of the PV-refrigerators are used for vaccine storages and medical services. WHO started to become active in the development of solar refrigeration by photovoltaic panels in 1979. The first specification of a solar refrigerator for medical use was published by the 'Expanded Programme of Immunization (EPI). In 1996, WHO has concluded that the solar-refrigerator has significant benefits fulfilling immunization project. Furthermore, the photovoltaic power system can be shared with other applications in a medical centre. However, WHO decided to avoid an implementation programme that focuses exclusively on the solar vaccine refrigerator, reasoning that it cannot compete with gas-powered units in terms of investment and incurring costs (WHO, 1996).

2. PERFORMANCE

Performance of the refrigeration system is presented as a ‘coefficient of performance (COP)’. It shows how much heat can be removed from a cold region (Q_e) for each unit of work or energy used (Q_g or W).

$$COP = \frac{Q_e}{W} \text{ or } COP = \frac{Q_e}{Q_g} \tag{1}$$

For the solar-driven systems, the performance can be written as the product of the COP and the solar collector efficiency (η_c). Besides, it can be defined as a ratio of the refrigeration effect and the solar energy input (I) for the thermal-driven systems, which is called ‘system thermal ratio (STR)’.

$$\eta_{system} = COP \times \eta_c \tag{2}$$

$$STR = \frac{Q_e}{I \cdot A} = \frac{Q_e}{Q_g} \times \frac{Q_g}{I \cdot A} = COP \times \eta_c \tag{3}$$

The STR of a mechanical refrigeration system (vapour compression and Rankine cycle) varies at 0.3-0.6 based on COPs at 3. At the COPs about 1.35 of the absorption cycle (two-stage devices with parallel-flow heat regeneration and 140-190°C heat input) STR can be reach at 0.5. (Gordon, 2000) The installation cost of the absorption and the desiccant system is reported on the National Association of Home Builders, USA website at US$ 4 000 to US$ 8 000 per ton of refrigeration effect.

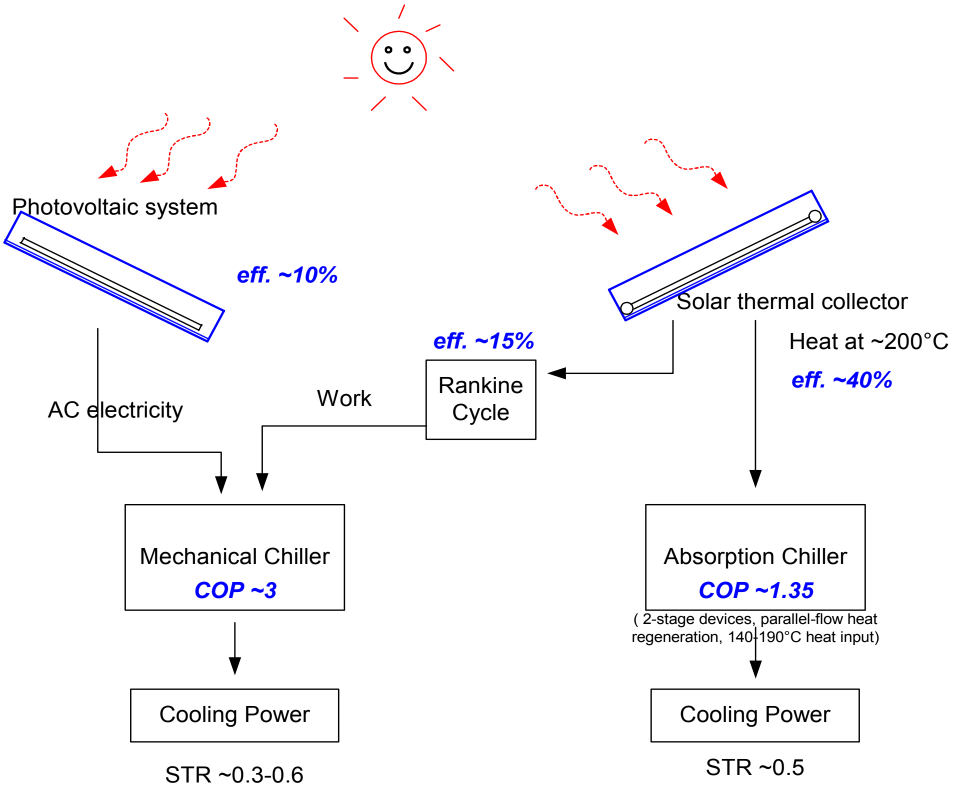


Figure 2.1 High efficiency commercial solar refrigeration

The limitation of the solar cooling system is mainly due to the technical availability, solar flux and price. However, other environmental factors such as surrounding temperature, air humidity and location affect the efficiency and feasibility of the system. The STR that showed on figure 2.1 is based on 10% of Photovolatic efficiency (from solar to AC electricity) and 40% of solar thermal collector (from solar energy to heat at 200°C).

The cost of the Photovoltaic refrigerator for the heath care, estimated by WHO, 1993, roughly shows between $US 3 500 to $US 6 000 without installation cost for the 12-24 V compression refrigerator, 0-8°C. For the thermal-driven refrigerator, the price is between $US 4 000 to $US 5 000 without transportation and installation cost. The retail price of the medical refrigerator, driven by PV system (DC compressor) is listed in the following table.

Table 2.1 The retail price for the PV-driven medical refrigerator

Size (litre)	Type	*Amps/day	Retail price (US$)
330	Refrigerator-Freezer	28	~ 1 900
280	Refrigerator only	15	~ 1 500
280	Freezer only	55	~ 1 600
110	Refrigerator-Freezer	13	~ 1 300
110	Refrigerator or Freezer only	9	~ 1 300
110	Vaccine Refrigerator-Freezer	13	~ 1 500
30	Vaccine Storage	6	~ 1 000

Remark: * power used per 24 hour period at 70°F or 23°C
Source: http:www.worldbank.org/html/fpd/energy/subenergy/solar/medical.htm , 16th July 2001.

A large scale solar driven air-conditioning system is commercial available in Germany however it is not yet economical due to the low efficiency of the system (Henning, 2003).

3. SOLAR COOLING TECHNOLOGIES

All heat, mechanical work and electricity can drive refrigeration machines. The appropriate refrigeration cycle depends on the cooling demand, available form of energy input, and temperature level of the refrigerated objects and the environment. The solar-powered cooling system generally comprises three main parts: the solar energy conversion equipment, the refrigeration system, and the cooled object (e.g., a cooling box). A number of possible "paths" from solar energy to the "cooling services" are shown in figure 3.1

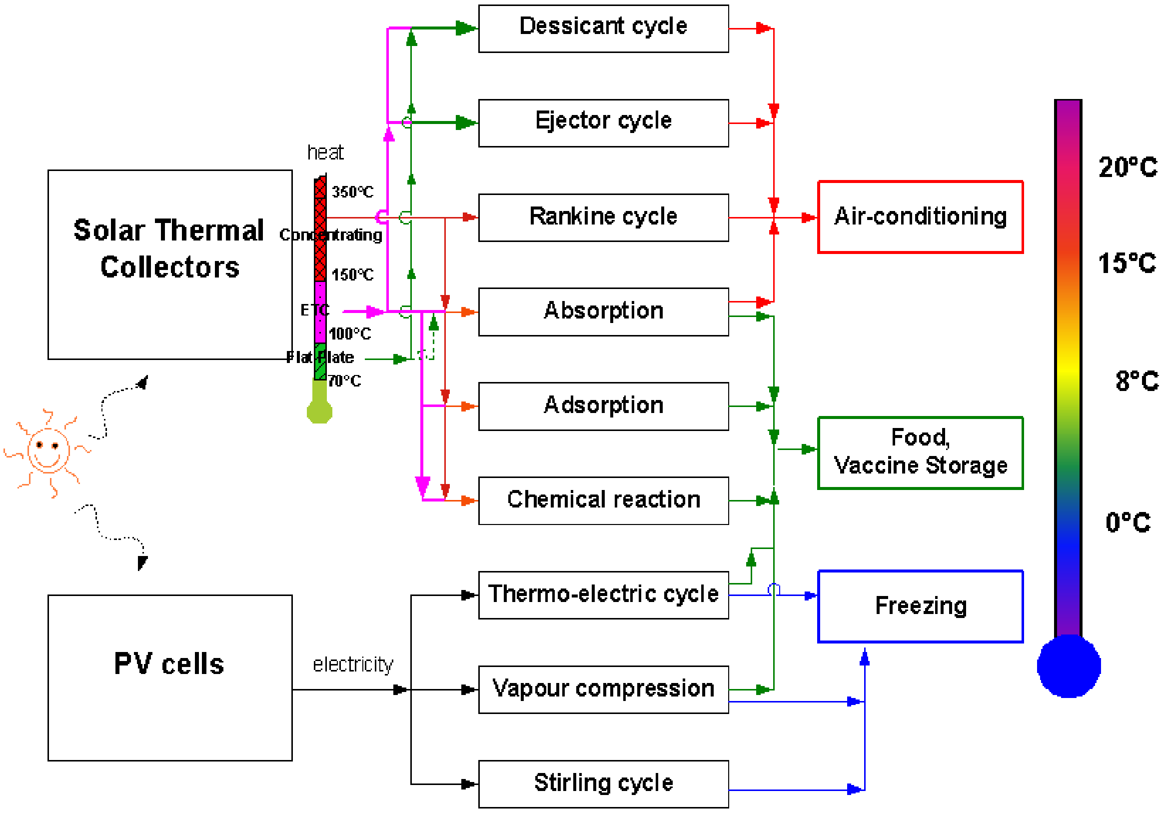


Fig. 3.1. Solar cooling path (Pridasawas, 2003)

For air conditioning, at a service temperature of 15-20°C, cooling capacity is generally high and energy removed from the chilled space has a low potential to convert to a useful energy. The solar thermal-driven system is more suitable for the air-conditioning system than the photovoltaic-driven one due to a lower installation cost. Furthermore, the performance of the thermal driven refrigeration machine is high at high evaporation temperatures. The solar thermal-driven air-conditioning system could be the absorption chiller, the adsorption chiller, the desiccant cooling system, the Rankine system or the ejector refrigeration system.

For the low temperature requirement for food storage of 0-8°C, many systems can be applied, e.g., the vapour compression system, the thermoelectric system (Peltier), the absorption system, the adsorption system or the chemical reaction system. For an application that needs temperature below 0°C, e.g., freezing boxes or ice production units, a vapour compression chiller, an absorption chiller, an adsorption chiller, a chemical reaction chiller and a Stirling chiller can be used. Typically the performance of an electricity-driven refrigeration system is quite high but it requires photovoltaic panels, which are expensive and have low efficiencies. But these systems can be built in small sizes, thus making them suitable for small applications such as vaccine transportation or cooling boxes. Most PV-driven systems are used for vaccine storage of the WHO-project.

The solar-driven refrigeration system is mainly classified in to 2 main groups depending on the energy supply: thermal/work driven system and electricity (Photovoltaic) driven system. Each group can be classified as the following,

1. Thermal/work driven system
    - Absorption refrigeration cycle
    - Adsorption refrigeration cycle
    - Chemical reaction refrigeration cycle
    - Desiccant cooling cycle
    - Ejector refrigeration cycle
    - Expansion refrigeration cycle
2. Electricity (Photovoltaic) driven system
    - Stirling refrigeration cycle
    - Thermo-electric refrigeration cycle
    - Vapour compression refrigeration cycle

Table 3. 1: Existing solar-driven refrigeration systems

Thermal-driven systems				***Applications***	
Systems	**Gen./ Regen. Temp. (°C)**	**COP$_{cycle}$**[c]	**Working Fluid**	**Refrigeration**	**A/C**
Absorption	80-190	0.6-0.8 (single stage) ≤ 1.3 (2 stages)	NH_3/H_2O, $H_2O/LiCl$, $H_2O/LiBr$	✓	✓
Adsorption	80-300	0.3-0.8	H_2O-Zeolite, Methanol-Activated Carbon	✓	
Chemical reaction	80-300	0.1-0.2	$NH_3/SrCl_2$	✓	
Duplex-Rankine	>120	0.3-0.5	water, R114, Toluene, Organics fluid		✓
Desiccant	40-100	0.5-1.5	water		✓
Ejector	80-150	0.3-0.8	water, butane, R141b, etc.		✓

Electricity-driven systems				***Applications***	
Systems	**Power for 1 W of the cooling effect (W)**	**COP$_{cycle}$**[c]	**Working Fluid**	**Refrigeration**	**A/C**
Vapor-compression	12-50	3 - 5	R134a, R290, etc.	✓	
Thermo-electric	a few W	0.5[a]	-	✓	
Stirling	3 - 17	3[b]	He, H_2, N_2	✓	

Remarks:
a*;* International Institute of Refrigeration (IIR), (1999)., based on the sunlight of 5 kWh/m^2 day
a**;* International Institute of Refrigeration (IIR), (1999)., based on the theoretical calculation of ETC solar collector surface of 1.8-1.2m^2/kWh day, chilling at –5°C and condensation temperature at 35°C.
b*;* Globalcooling, (2001)
c*;* COP of the refrigeration sub-system.

Thermal-driven System

3.1 Absorption Refrigeration Cycle

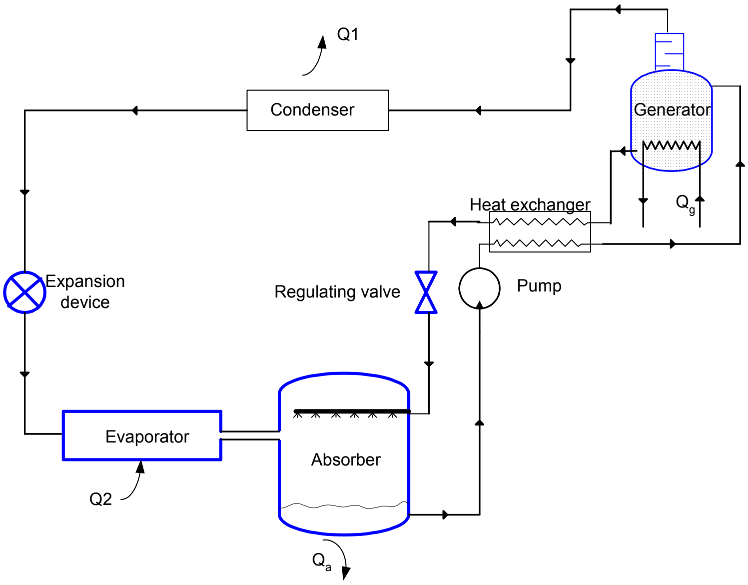


Fig. 3.1.1 Simple diagram of the absorption refrigeration system.

The main components of the absorption refrigeration system are an absorber, a generator, a condenser, an expansion valve, a heat exchanger and a pump. Two kinds of working medium are used at the same time in a refrigeration and an absorption processes. The refrigerant vapour flows to the condenser passing through a vapour-trap and condensed. Liquid refrigerant from the condenser goes through an expansion valve while the pressure is decreased to an evaporation pressure. At the evaporator, cooling effect is achieved by the vaporisation of the refrigerant at a low temperature. Refrigerant vapour from the evaporator continues to an absorber and dissolves in a weak refrigerant solution, and it becomes a stronger refrigerant solution, which called “rich solution”. A pump is the only moving part in this system. The “rich solution” is pumped to a generator. At the generator, the rich solution is heated up; the refrigerant is separated from the solution. The refrigerant is vaporised and goes to the condenser while the weak solution is passed through a heat exchanger and returned to the absorber to absorb the refrigerant vapour. The refrigeration process and the regeneration process operate at the same time as the continuous process, producing a continuous cooling effect.

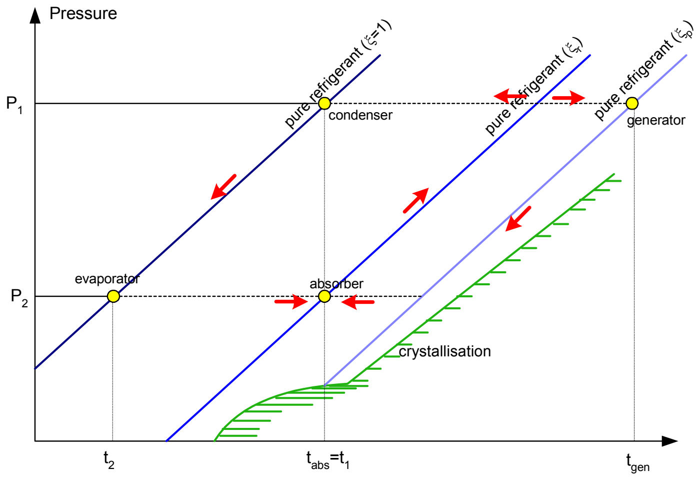


Fig. 3.1.2 The pressure-temperature diagram of the absorption refrigeration system.

A flat plate solar collector can maintain the operating condition at the generation temperature about 75-100°C but very efficient heat exchangers are required. An evacuated solar collector with higher generation temperature can be used with lower refrigeration temperature in the Platen-Munters cycle (Valizadeh 1996). A solar pond is one technique that can supply the heat to this system (Sierra 1993).

Working Media

Several pairs of working media have been used for absorption refrigeration system e.g. a pair of ammonia-water (ammonia is the refrigerant and water, water is the absorption medium), a pair of water-lithium bromide (water is the refrigerant and lithium bromide is the absorbent) and a pair of water-lithium chloride. (water being the refrigerant and lithium chloride is the absorbent). Both ammonia and water have good heat transfer characteristic. In addition water separator (rectifier) is needed to be install in order to prevent water from passing to the condenser with pure ammonia.

For the air-conditioning system, LiBr-H_2O is preferred than using NH_3-water due to the safety issue. Lithium bromide is a hygroscopic salt, which easily absorbs water and a vapour pressure is almost zero, any separator (or rectifier) between LiBr and water is not needed due to a very large difference of the boiling point of water and lithium bromide. However, the problem of crystallisation can be happened since at the low temperature, the salt crystals can be formed at high concentration of LiBr. This crystallisation problem limits the ability of the system. In addition, water has a low saturating pressure or huge vapour volume, affecting the system geometry. Water-lithium bromide mixture is quite a corrosive solution, some inhibitors may require together with a suitable selective material.

Table 3.1.1 Advantages and disadvantages of the absorption refrigeration system

Advantages	Disadvantages
1. Require little maintenance	1. Low COPs
2. Only one moving part (pump) and might be no moving part for a small system	2. It cannot be applied for a very low evaporating temperature (when water-LiBr are used).
3. No auxiliary energy for operation of the small system	3. High heat release to the ambient.
4. Solar thermal collector is used, that is cheaper than Photovoltaic cells.	4. A continuous and big system need pump which is not solar thermal energy dependent.
5. Low energy cost (for pump only). A small system might not require pump.	5. Quite complicated system and require advanced knowledge for maintenance.
6. Low-temperature heat supply	6. For the big system such as an air conditioning unit, it requires a large area of solar collector which meant a very high installation cost and a large installation area.

Platen-Munters cycle

This system is well known as the Electrolux refrigerator and principally was invented from the division of applied thermodynamics and refrigeration, KTH, Sweden. It is developed from the Carré absorption cycle but operating without pump. It can be called no-moving part and no-auxiliary energy supply system. Hydrogen is used to maintain the total pressure in the whole system to be constant. The refrigerant partial pressure is allowed to be low at the evaporator, achieving the refrigeration effect. Ammonia is conventionally used as the refrigerant, water is used as the absorption media and hydrogen is used as the inert gas.

The principle of the cycle is similar to absorption cycle, however total pressure in the whole system is constant. Hydrogen is circulated between the evaporator and the absorber, compensating the pressure difference between the high and low-pressure side. Ammonia vapour evaporates in the generator and then condenses in the condenser before flowing to the evaporator. The ammonia poor aqueous solution is then back to the absorber by the gravitational flow. At the evaporator, the liquid-ammonia is exposed into the hydrogen atmosphere, and evaporates due to a low partial pressure (of ammonia). The ammonia-hydrogen mixture continues to the absorber (passing through the heat exchanger), in which ammonia is absorbed in the water solution. The hydrogen returns to the evaporator through the heat exchanger while aqueous ammonia solution forwards to the generator by a thermosyphon pump. The generator temperature is typically varied between 120 to 180°C, depending on the operating temperature. The conventional energy sources are natural gas, kerosene or electricity. The practical COP varies between 0.2 and 0.3 at 25 and 100 W of cooling capacity (Granryd, 1998). Large capacity system is difficult to be achieved.

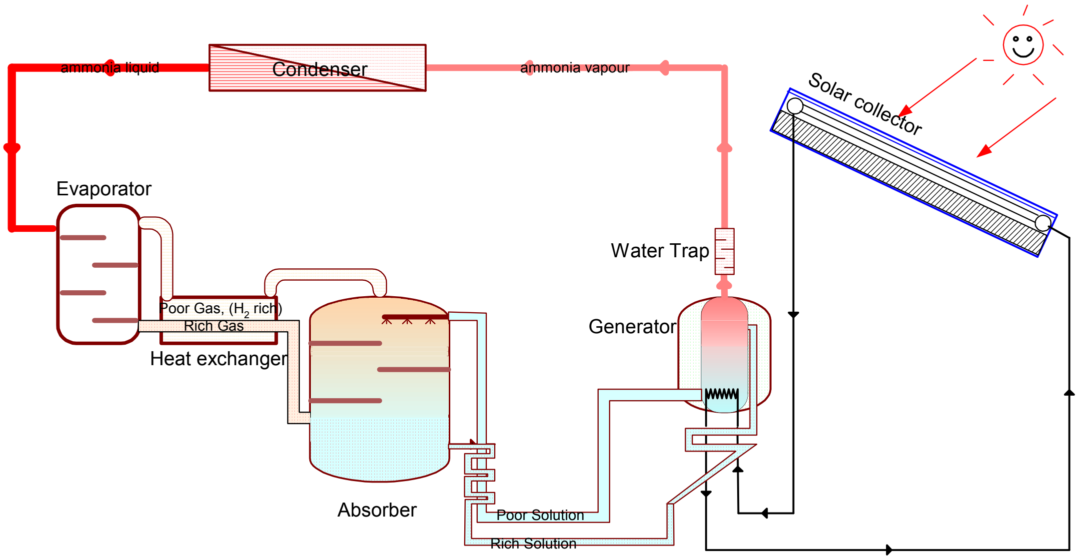


Fig. 3.1.3 The solar-operating Platen-Munters refrigeration system.

3.2 Adsorption refrigeration cycle

An adsorption, also called a solid-sorption cycle, is a preferential partitioning of substances from a gaseous or liquid phase onto a surface of a solid substrate. This process involves the separation of a substance from one phase to accumulate or concentrate on a surface of another substance. An adsorbing phase is called an 'adsorbent'. Material, which is accumulated, concentrated or adsorbed in another surface, is called an 'adsorbate'. The sticking process *should not change any macroscopic of the adsorbent* except the changing in adsorbent's mass. Both adsorption and absorption can be expressed in term of sorption process. The adsorption process is caused by the Van der Vaals force between adsorbates and atoms or molecules at the adsorbent surface. The adsorbent is characterised by the surface and porosity.

In the adsorption refrigeration cycle, refrigerant vapour is not be compressed to a higher temperature and pressure by the compressor but it is adsorbed by a solid with a very high microscopic porosity. This process requires only thermal energy, no mechanical energy requirement. The principles of the adsorption process provide two main processes, adsorption or refrigeration and desorption or regeneration.

The refrigerant (water) is vaporised by the heat from cooling space and the generator (absorbent tank) is cooled by ambient air. The vapour from the cooling space is leaded to the generator tank and absorbed by adsorbent (zeolite). The rest of the water is cooled or frozen. In the regeneration process, the zeolite is heated at a high temperature until the water vapour in the zeolite is desorbed out, goes back and condenses in the water tank, which is now acting as the condenser.

For a discontinuous process, the desorption process can be operated during daytime by solar energy, and the adsorption or the refrigeration process can be operated during night-time. The solar energy can be integrated with a generator. The single adsorber is required for a basic cycle. The number of adsorbers can be increased to enhance the efficiency, which depends on the cycle. This process can also be adapted to the continuous process

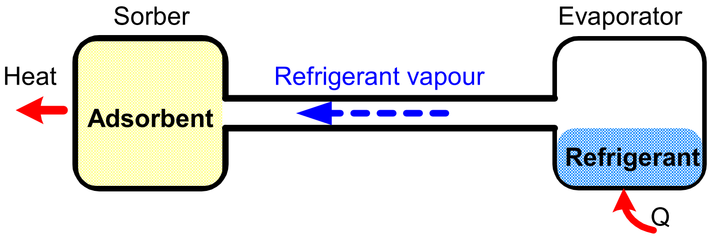


Fig. 3.2.1 The adsorption (Refrigeration) process

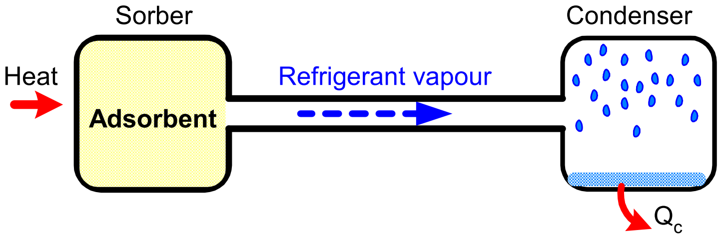


Fig. 3.2.2 The desorption (Regeneration) process

Working Media

Typical and commercial adsorbents are made from silica gels, zeolite and activated carbons. The adsorbates (refrigerant fluid) could be water, ammonia or methanol. The famous pairs that have been used commercially are zeolite '13x'/H_2O for the temperature above 0°C and activated carbon 35/methanol for the temperature below 0°C. The others are ammonia/$SrCl_2$, water/silica gel or air/silica gel in the open cycle (Stegou-Sagia et al., 1997).

Table 3.2.1 Advantages and disadvantages of the adsorption refrigeration system

Advantages	Disadvantages
Require little maintenance	High heat release to the ambient.
No moving part	The high weight of absorbent, not suitable to build in the high capacity.
Thermal COP is not so low (~0.4 at T_e 0°C,T_c 40°C,T_{ad} 35°C and T_{de} 100°C)	Poor thermal conductivity of the solid adsorbent which cause the long-term problems.
Solar thermal collector is used, that is cheaper than Photovoltaic cells.	Low operating pressure requirement, which is difficult to achieved air-tightness. For Activated carbon 35/Methanol, the operating pressure is around 50 mbar and around 6 mbar for zeolite/water.
Low operating temperature can be achieved.	Low energy density. The quantity of the cycled gas (kg gas/kg solid) is very low, around 0.13 for activated carbon/methanol pair and the quantities of ice recovered per 1 kg per activated carbon 35 is 0.26 kg.
	Very sensitive to low temperature especially the decreasing temperature during night-time.

3. 3 Chemical reaction (solid-sorption) refrigeration cycle

A chemical reaction refrigeration cycle is a solid-gas adsorption process with a chemical reaction. It is an intermittent system. The principle of chemical reaction solid-sorption process is similar to the adsorption process. The same analogies of these two systems are:

- They are intermittent processes since the cold cycle is not continuous produced.
- They are heat-driven refrigeration cycles (The mechanical work is required in some cases to blow out the vapour). The sorption latent heat from the gas phase is the driving energy.

The differences between adsorption and chemical reaction solid-sorption refrigeration cycle are:

1. The physical process (the major important difference)
   - In the adsorption process, the physical adsorption process on the surface of the adsorbent does not cause deformation or changes any macroscopic structure of the adsorbent (or solid).
   - In the chemical reaction process, the macroscopic structures of the adsorbent are changed after the chemical reaction takes place.

2. The thermodynamic operation of the cycles
   - Adsorption is a bivalent process. To complete the adsorption and desorption cycle, heat supply is required to the adsorber to increase the temperature of the absorbent from the minimum reactor temperature (T_o) to the maximum adsorber temperature only.
   - Chemical reaction is a monovalent process. To complete the cycle, more heat supply to the adsorption cycle is required to achieve high kinetics of reaction. The volume of the sorber is also changed significantly.

3. The refrigerant
   - Many pairs could be used such as ammonia/activated carbon, methanol/activated carbon, water/silica gel and etc. in the adsorption system.
   - There are two main groups of the refrigerant pairs to use in chemical reaction cycle, (Meunier,1998)
     −Ammonia salts with alkaline compounds such as $BaCl_2$, $MnCl_2$, $SrCl_2$, etc.
     −Methal hydride with low-hysteris intermetallic or mishmetal compounds.

4. The number of the sorbers
   - For the adsorption cycle, the number of the sorber depends on the cycle. One sorber is enough for the basic cycle. Enhancing the efficiency of the cycle can be achieved by increasing the number of sorbers.
   - For methalhydride pairs, hydrogen does not change to the liquid phase, two sorbers that show the temperature lift are used.

The COP of the intermittent cycle is always much less than one. For the solar-driven sorption cycle, the efficiency of the solar thermal collector must be multiplied with the COP of the intermittent cycle, which make its efficiency lower.

Table 3.3.1 Advantages and disadvantages of the adsorption refrigeration system

Advantages	**Disadvantages**
Require little maintenance	High heat release to the ambient.
No moving part, it is a static system	Low COPs.
Low operating temperature can be achieved.	Poor thermal conductivity of the solid adsorbent which cause the long-term problems.
Solar thermal collector is used, that is cheaper than Photovoltaic cells.	The high weight of adsorbent, not suitable to build in a high capacity.
Large energy density	Low operating pressure requirement, which is difficult to achieve air-tightness. For activated carbon 35/Methanol, the operating pressure is around 50 mbar and around 6 mbar for Zeolite/water.
Less bulk than an adsorption cycle for a given capacity.	Very sensitive to low temperature especially the decreasing temperature during night-time.
	Not easy to design the system especially in the volume of the adsorber that changed when the chemical reaction occurred.
	Low operating pressure at the lower temperature, difficult to achieve air-tightness.

3.4 Desiccant Refrigeration System

A desiccant cooling system is based on an open-cycle dehumidification process. This system is also called “Lizzy” or “Pennington” cycle. Heat and water are needed to operate this system. Water is commonly used as a refrigerant since it is cheap and environmentally friendly. A desiccant material can be either liquid or solid. This cycle consists of one drying process, one heat exchanging process and one humidifying process. There are three major components, operating in an atmospheric pressure. These components are a dehumidifier, an evaporative cooler and a regenerator. Heat exchangers are also used as the additional components to increase the system efficiency.

A drying process can be performed in a desiccant wheel when solid desiccant (such as silica gel or zeolite) is used, or it can be performed in an absorption tank when liquid desiccant is used. A heat exchanging process occurs in a heat exchanger and the humidifying process is performed in a saturated pad or humidifier. The rotor wheel is widely used as the heat exchanger wheel and the drying wheel.

A simple diagram is shown in figure 3.4.1. Outdoor air is dehumidified with a solid or liquid desiccant where some of the moisture is removed, resulting in rising of the air temperature and decreasing of the humidity. The air is then cooled by exchanging sensible heat to the returned air in the heat exchanger and humidified to the desired humidity before supplied to the cooling space. The temperature of the supply air is further lowered by the humidifier or the evaporative cooler before entering the cooling space.

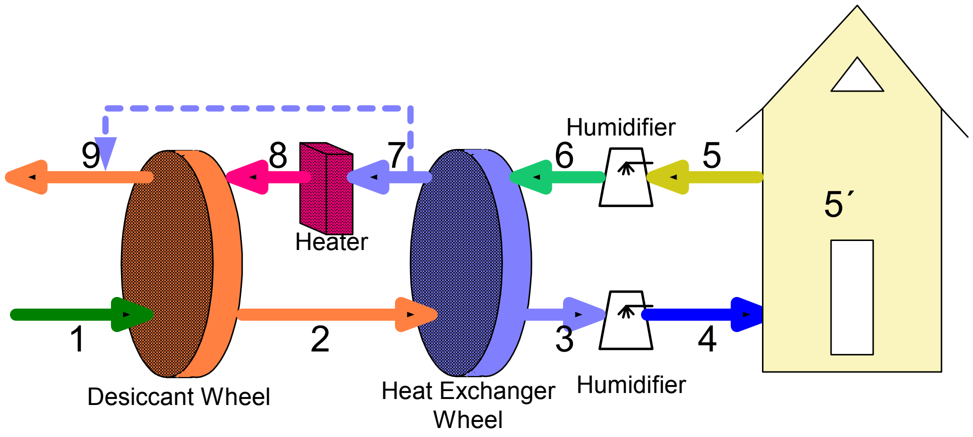


Fig. 3.4.1 The solid desiccant cooling machine

The returned air from the cooling space is returned to the evaporative humidifier. It is humidified to a lower temperature at the same enthalpy but with a higher humidity. The cooled air enters the energy recovery unit where acting as the cooling medium for the supply air. The air temperature is increased after passing the heat recovery (heat exchanger wheel). It is then passed through a heater, where it is further heated, and then enters the reactivation sector of the desiccant rotor to reactivate the desiccant. The wet air leaves the rotor as the exhaust air. This process is shown in the Mollier's diagram in figure 3.4.2.

The energy supply to the heater depends on the temperature of the return air entering the desiccant wheel at stage 8. The humidity of the entering air and the effectiveness of the desiccant affect the amount of energy supply. The low-temperature heat can be supplied to the heater such as solar energy from flat plate solar collector, waste heat from industry or geothermal energy. A small amount of electricity is required for rotating the wheels.

The desiccant materials for a solid-desiccant system are usually silica gel or Zeolite. For a liquid desiccant system, the desiccant dehumidifier's hygroscopic aqueous solution can be triethylene glycol (TEG), $CaCl_2$-H_2O, LiBr-H_2O, LiCl-H_2O and etc.

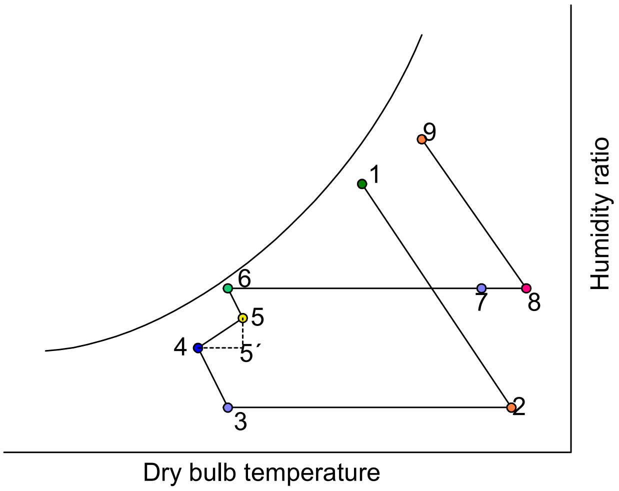


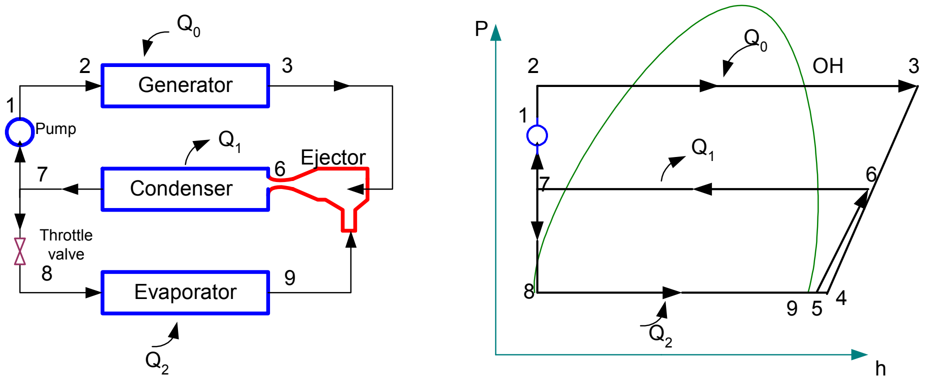
Fig. 3.4.2 Desiccant cooling process

Table 3. 4.1 Advantages and disadvantages of the desiccant refrigeration system

Advantages	Disadvantages
Environmentally friendly because water is used as the working fluid.	Can not get the low temperature in the humid region.
Can be integrated with a ventilation and heating system.	Required maintenance because of moving part in the rotor wheel of the solid desiccant system.
A thermal collector can be used, which is cheaper than PV cells.	Can be contaminated easily.
Low heat release to the ambient	Difficult to design for a small application.
	Require dehumidifier.

3.5 Ejector refrigeration cycle

An ejector refrigeration cycle is one of the heat-operating cycles. The interesting advantage is as a 'low temperature heat supply' air conditioning system. With this outstanding, the research and development of such a system has been considered increasingly since the energy crisis 1970s. Solar energy (as a renewable source) and waste heat from a heat-operated process such as from truck engines can be integrated with the ejector refrigeration system. The simplicity in installation, design and operation are advantages. The pump is the only moving component in this system. The ejector and the pump are used to maintain the pressure differences in the system. Low efficiency is a drawback of this system, however when the generating temperature is low, the COP of the ejector cycle is higher than the corresponding COP of an absorption system; moreover low-graded heat can be applied.



Ejector Refrigeration System

Fig. 3.5.1 Schematic diagram for a simple ejector refrigeration system

The major components in the solar-driven refrigeration system are an ejector, a condenser, a generator, an evaporator, an expansion device and a pump. The vapour from the low temperature evaporator is sucked into the high velocity vapour stream in the ejector. The high velocity vapour stream goes through a converging-diverging nozzle in the ejector resulting in the vapour being sucked from the low temperature evaporator. The suction occurs, as the pressure is low at the narrowest section of the ejector. The stream from the evaporator reaches subsonic velocity. A mixing occurs in a mixing zone at the end of the converging section. After mixing, a combined stream becomes a transient supersonic stream, and the velocity of the combined fluid must be high enough to increase the pressure after deceleration in the diffuser to a suitable condensing pressure. After the pressure build-up, the stream from the ejector goes to the condenser, condenses and heat is rejected to the environment. After the condenser, one part of the fluid is pumped to the generator and the rest goes to the evaporator, reaching the evaporating pressure through the expansion device.

Many refrigerants can be used with an ejector refrigeration system such as water, R113, R114, R141b, R134a, R11 and R12.

Table 3.5.1 Advantages and disadvantages of the ejector refrigeration system

Advantages	Disadvantages
Low temperature heat source can be supplied.	Low COPs
Low operating and installation cost	Difficult to achieve low evaporating temperature
Easy to design and install	Superheated is required for some refrigerant such as NH_3 or water.
The system is not complex	Difficult to design an ejector
High reliability.	
Required less maintenance, less interrupted service.	
A large overload capacity can be achieved.	
The solar collector can be used to supply heat, which is cheaper than the Photovoltaic cells.	

3.6 Rankine-driven refrigeration cycle

A Carnot heat engine is the most efficient engine to produce work from heat. Heat in the Carnot engine transfers from a higher temperature to a lower temperature. Generally, the Carnot heat engines can not be operated since the mechanical problems such as erosion or cavitation of turbine blades, when operating in a two-phase region. The adaptation of the Carnot heat engine in the one-phase region is called “Ranking”cycle.

The reversible of the Rankine heat engine is called the ‘Rankine refrigeration cycle’ or the ‘vapour compression cycle’. Work from the turbine of the power cycle drives the compressor of the refrigeration cycle.

Any excess energy can be used to produce electricity and reserved as a backup energy when sunshine is lacking or it can be connected to a grid system.

The efficiency of the Ranking power cycle can be obtained generally about 0.09 (IIR, 1999), the COP of a refrigeration cycle can be estimated around 3 (at an evaporating temperature –5°C and a condensing temperature 35°C). The efficiency of solar collector systems is around 0.5 for a flat-plate solar collector and 0.6-0.7 for an ETC collector. The theoretical system thermal ratio (STR) can be estimated around 0.13. IIR has estimated the STR for a selective panel or the ETC at 0.08 to 0.12 respectively at the clear day and 1.8 to 1.2 m^2/kWh day of the solar collector surface.

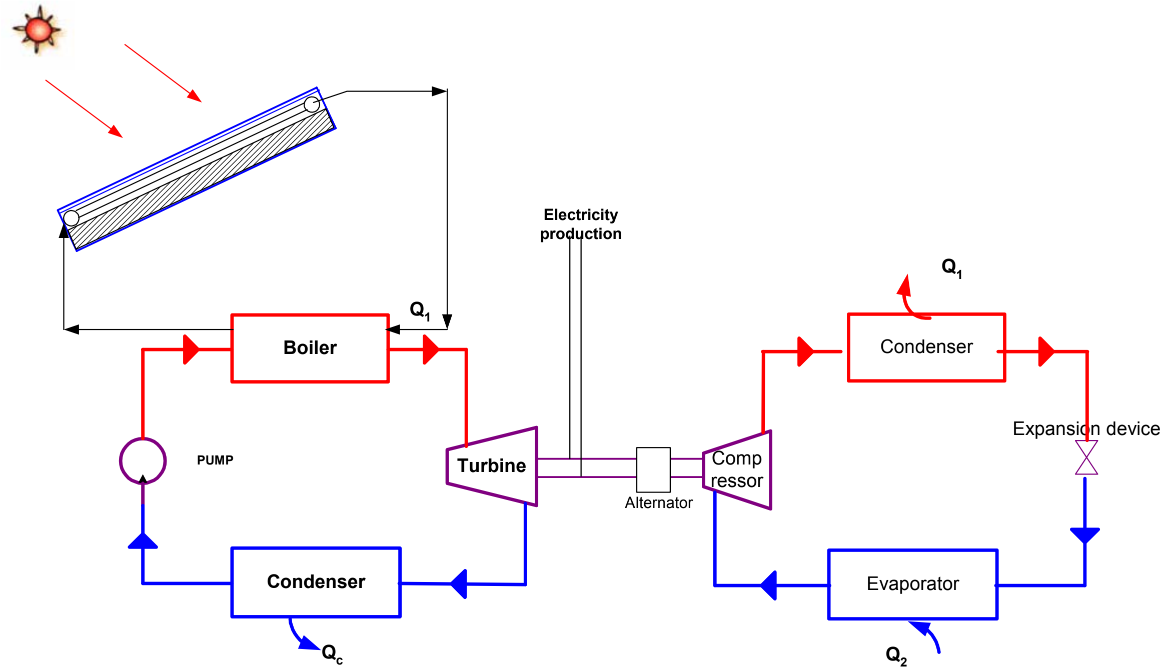


Figure 3.6.1 Solar driven Rankine cycle

A solar-operated Rankine cycle is not much different from a conventional power plant, using water as the working fluid. To increase the efficiency and prevent the erosion of the turbine blades, superheating and extraction processes are used. The working fluid in the Rankine power cycle and the refrigeration cycle can be different. The suitable refrigerant in the solar operating system should be chosen to avoid moisture in a turbine. Superheating is not preferred since the increasing of the collector temperature requirement. The extraction is not economic for a small system. Working fluids such as R114 that give a positive slope of the saturated vapour line on a T-S diagram, the outlet temperature from the turbine is significantly higher than the condensation temperature gives the benefit to preheat the working fluid before it enters the boiler. However R114 is not environmental friendly; it has an ozone depleting potential due to a Chlorine atom.

The speed of the turbine and the compressor should be analogous. The alternator or other equipment that used to adjust the speed should be installed with the system.

Table 3.6.1 Advantages and disadvantages of the Rankine's driven refrigeration system

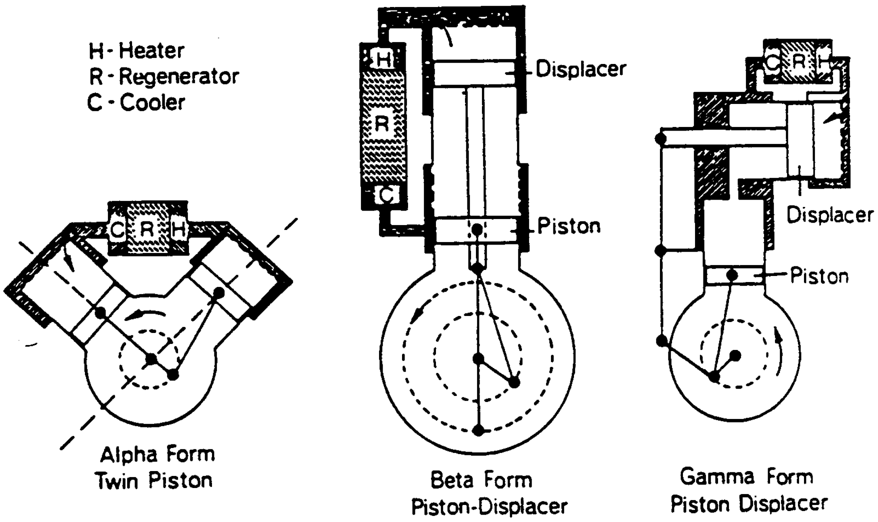
Advantages	Disadvantages
Excess energy can produce electricity.	High installation cost
Suitable for high capacity system	Large system
The thermal collector can be used as the heat supply, which is cheaper than PV cells.	Required maintenance because of moving parts.
	Low capacity
	Working fluids are easy to contaminate and are harmful for environment.

3.7 Stirling refrigeration cycle

A possibility of a Stirling cycle has generated wide interest in refrigeration technology. It can be applied for both engine and refrigerator. Almost all refrigeration applications, over a large operating temperature range, from a domestic refrigerator to a cryogenic application, can be provided by the Stirling engines Electricity can drive this cycle as well as heat. There are many outstanding points such as high efficiency, light weight and environmentally friendly, thus make a large attention by engineers and scientists now.

A principle of the Stirling refrigeration cycle is based on a changing of pressure and temperature of gas when heated or cooled at the constant volume. This cycle is similar to both internal combustion engine and a Rankine and closed-cycle Brayton machines. It has pistons like the internal combustion engine and the refrigerant container is separated from the heat source and sink as the Rankine and closed-cycle Brayton machines. The ideal Stirling cycle consists of two isothermal reversible processes and two constant-volume reversible processes. The refrigerant using in the Stilring unit is environmentally friendly, no Chlorofluorocarbon (CFC) or Hydrochlorofluorocarbon (HCFC). Gas (e.g. helium, hydrogen, nitrogen or air) is used as the working medium at an elevated pressure. The Stirling cooling unit consists of a hermetically sealed capsule with 2 moving parts inside, a piston and a displacer, containing working medium inside. The piston is used to compress and expand the working fluid and the displacer is used to shuttle the working gas back to and out of between the cold side and the warm side. Heat is absorbed at the cold side during expansion of the working fluid and rejected to the hot side during the compression process. A regenerator (or a heat storage heat exchanger) is an important unit, used for inputting and rejecting heat from the working gas, and store heat during the changing of the cycle's phase. It is made from a porous solid material.

The examples of the Stirling engines or refrigerators are shown in Figure 3.7.1



Alpha Form
Twin Piston

Beta Form
Piston-Displacer

Gamma Form
Piston Displacer

Fig. 3.7.1 The example of Stirling engines or refrigerators in different arrangements
(Lundqvist, 1993)

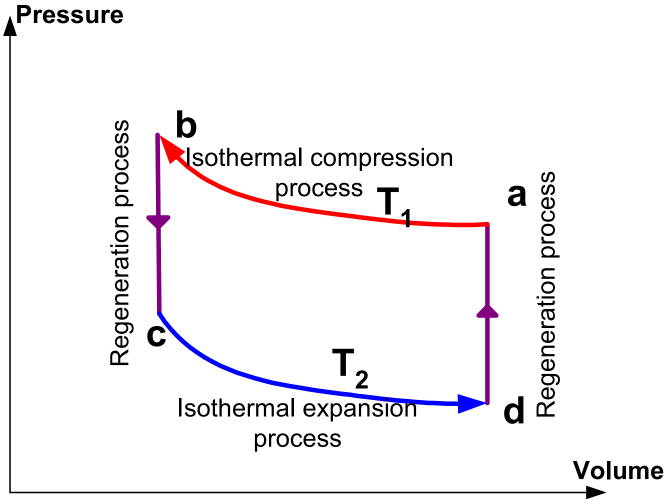


Fig. 3.7.2 Pressure-Volume diagram of a Stirling refrigeration process

If we apply heat to a high temperature heat exchanger (heater), the machine will rotate in the same direction, taking up heat from the heater to the cooler and creating the mechanical work; this is called the 'Stirling engine'. If the temperature at the heater decrease or no heat supply to the heater more, the machine will stop. However, if we still supply work (e.g. by an electrical motor) to the piston and let it continue rotates in the same direction, heat from the heater will be taken to the lower temperature until the temperature of the heater decrease. The heater is now changed to be the cooler (or the freezer) and the machine will continue pumping heat from the lower temperature heat source and reject to the higher temperature heat sink. The Stirling machine, which is driven by mechanical energy is operated as the cooler or heat pump.

The operating temperature of the Stirling cycle is wide comparing to other refrigeration cycles. Gas is used as a refrigerant, which gets rid of phase change from liquid to vapour, thus the operating temperature range is wider. A few gases can be used as the refrigerant such as helium, nitrogen and air.

The Stirling cooling machine is widely used in cryogenic application. A few applications are used such as for cooling of sensors and electronics devices, a small scale production of liquid air (for cooling of infrared sensors at the cooling temperature range of –160°C to –200°C, by Phillips company), a thermal imager (Stirling cooler and all the electronics are integrated into a single camera head, by the Information System Division of Mitsubishi Eletronics America, Inc.) Some of them are produced by Sunpower company (USA) for a specific application such as the Orbital Refrigerator/Freezer (OR/F) for store experimental samples (e.g. blood and urine) in the Shuttle mid-deck.

Solar driven Stirling refrigerators.

Some research was funded by the Dutch Consortium of Environmental Groups and Utilities to build a Stirling cooled, solar powered, battery-free super efficient refrigerator in November 1993, (Mennink and Berchowitz, 1999). This prototype unit was first demonstrated in April 1994 at the 12th European Photovoltaic Conference. The overall COP is close to 3.0 at ambient temperature of 25°C. Two sizes of the refrigerators have been studied, 365 litres and 30 litres (1 cu. ft.) The capacity are varies from 8 to 50 W at 0 to 30°C. At 200 litres PV-refrigerator requires approximately 50kWh/yr whereas the average energy consumption of the typical European refrigerator is about 300kWh/yr. It is a free-piston machine driven by a linear motor, AC voltage source drives. To avoid the need of batteries, 5 litres of water ice is produced for no-sun operation time 24 hours. Helium is used as the working fluid and hermetically sealed to protect the leakage.

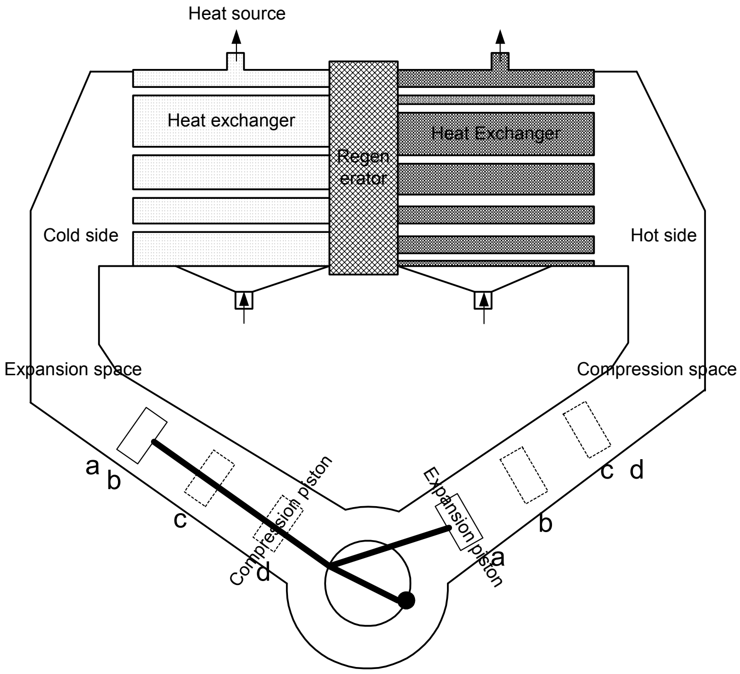


Figure 3.7.3 Stirling refrigerator, twin piston [Adapted from IIR, 1999]

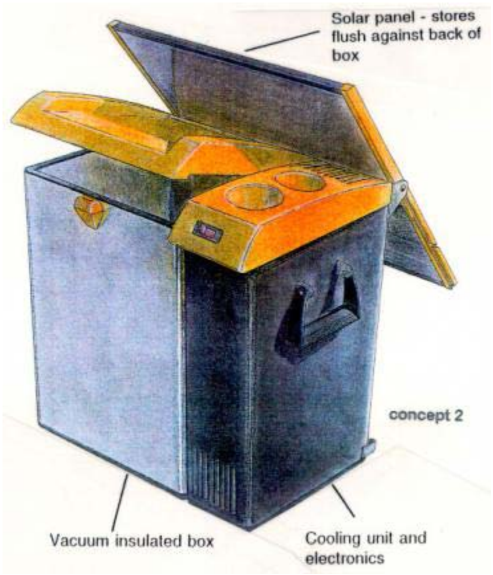


Fig. 3.7.4 Solar driven Stirling cooler box by Global Cooling BV (Netherlands)

[Source: Berchowitz, 1996]

Table 3.7.1 Advantages and Disadvantages

Advantages	**Disadvantages**
High COP for high temperature difference. The COP can be as high as 3.0 between 0 to 30 °C at the power consumption between 8 to 50 W.	High production cost; the purchasing price of a 4 kW Stirling refrigerator is today around 30,000 US$. (Sunpower, 1999).
Can be used for cryogenic applications and more mechanically simple than other application for low temperature operation.	Need photovoltaic (PV) cells which has high installation cost. However, it needs small power consumption, so the area of the photovoltaic cell is not so large.
Environmental friendly, No CFC, no ODP (Ozone Depletion Potential) and no GWP (Green House Warming Potential) working fluid.	Need more research and studies. It is quite new technology and not widely used. The studying in this field should be attended more in order to increase its performances.
Available for large operating temperature range	High-pressure cycles which makes the gas containment more difficult.
Fully modulatable; the refrigeration temperature can be adjusted as much or as small as needed. Small temperature variation and the refrigeration in the refrigerator stay at one exact temperature.	Lack of Stirling engines for such application. Complexity in design and technology; if something was wrong during operation in the rural area, it is impossible for the local operator to repair and maintenance. The units of today are hands built.
Can be operated in a high ambient temperature. The ambient temperature of around 50°C can be operated without problem.	
Mobility It is solid structure and can be made in a small scale. The mobile cooling application such as an ice-cream car can be applied.	
Light weight	
The vibration at the cold end of the cooler can be quite low.	

3.8 Thermoelectric refrigeration cycle

A thermoelectric refrigerator is also called a Peltier refrigerator. This technology is suitable for moderate and quite low temperature applications. Electron that moves in the system is used as the heat carrier to remove heat from the cooling space instead of a refrigerant. The practical unit has found in some applications such as a cooling box and a freezing box. This cycle has no moving part, no refrigerant and portable. Nevertheless, the coefficient of performance is quite low and it needs more improvement of the thermoelectric element.

Electric potential difference can be measured from different metal junction when these two junctions have different temperature because electric current moves from the higher potential to the lower potential one. This phenomenon is called the Seebeck effect and also known as a thermocouple. The reverse Seebeck effect was found by a French watchmaker named Peltier in 1834. If the electric current is forced through an electric circuit providing the different metal junctions (or conductors), the different temperatures between the ends of these metals can be measured; this phenomenon is called the Peltier effect. The different temperatures and the absorbed heat are proportional to the electrical potential difference, resulting from the amount of the electricity flows cross the junctions.

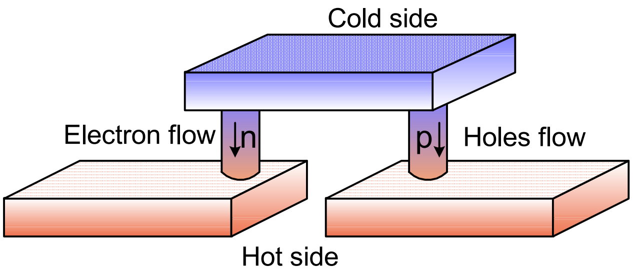


Fig. 3.8.1 Peltier's element

Figure 3.8.1 shows a Peltier's element which is composed of N and P typed semiconductor materials; the electric conductors are made out of copper. The upper electric conductor represents the cold side and the lower represents the hot side. Electrons flow through N-doped semiconductors and holes flow through the P-doped semiconductors. Now Peltier's elements are made from the semi-conductor materials, such as Bi_2Te_3.

Table 3.8.1 Advantages and disadvantages of the Peltier's refrigeration system

Advantages	Disadvantages
No working fluid	Low COPs
Quiet	For a PV system, installation cost is high and it requires battery for energy backup.
No moving part	Difficult to achieve a low refrigeration temperature
Small size and can be made a portable unit	Low reliability especially when the power supply is cut
Insensitive to orientation	
Low cost in mass production	
Light weight	

Relevant Researches

Sofrata, (1996) built the thermoelectric refrigerator driven by Photovoltaic cells. The minimum temperature was reached at –3°C. He proposed the alternatives to reject heat from the system by using single fan, double fans and natural chimney.

Hara et al., (1998) made the thermoelectric cooling headgear driven by a solar cell (amorphous type) for cooling the forehead as shown in the figure below. The forehead temperature can be reduced by 4 K in order to satisfy the thermal comfort.

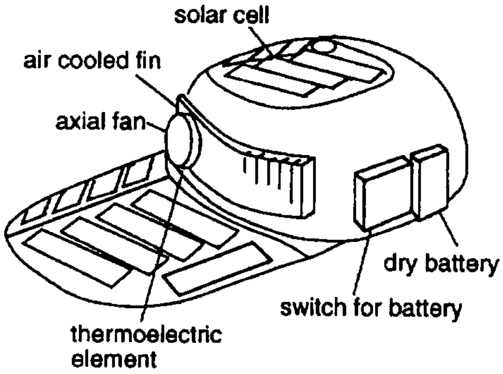


Figure 3.8.2 A thermoelectric cooling headgear driven by solar cells, proposed by Hara et al., (1998).

World Health Organisation (WHO) has proposed vaccine storage and transportation using the Peltier-effect refrigerator (IIR, 1999). This system can operate at low voltage thus making it possible to connect with the solar cells. However, there are some problems regarding to the temperature changes, which is limited at 5°C at the ambient temperature 30°C and the system heats up when the electric supply is cut off since the Peltier's element becomes a thermal bridge. Furthermore, the reliability of power output is still a problem when the power cut down. The theoretical COP of this system is about 0.5, which is quite low.

3.9 Vapour compression refrigeration cycle

A vapour compression refrigeration system is the most widely used cooling system because of high efficiency and reliability. Electricity, as the main energy source, is used as the driven-energy for almost vapour compression system. Solar energy can be integrated with vapour compression cooling system by both Photovoltaic cells and solar thermal collectors with the Rankine engines.

Energy supply by the photovoltaic system for the vapour compression refrigeration system has been presented since 1970s (IIR, 1999). The commercial products are available now and normally driven by a direct current compressor. A medical refrigerator is the largest market following by household uses. It was proposed to the World Heath Organisation for vaccine storage of the Expanded Programme on Immunisation (EPI) since the late 1970s.

The main components of the vapour compression refrigeration system are a compressor, a condenser, an expansion device and an evaporator. Refrigerant is circulated in a closed system among these components. In the compressor, the appropriate pressures between high and low pressure are maintained at two temperature levels. At the lower pressure and temperature, in the evaporator, liquid refrigerant is allowed to vaporise when it absorbs heat from the surroundings and creates the refrigeration effect. Vapour is compressed and pumped to the higher temperature and pressure. The high-pressure vapour from the compressor is then condensed, heat is transferred to the surrounding and vapour becomes the liquid refrigerant. The liquid refrigerant goes to the evaporator passing through the expansion device; the refrigerant pressure has falling down to the evaporator pressure.

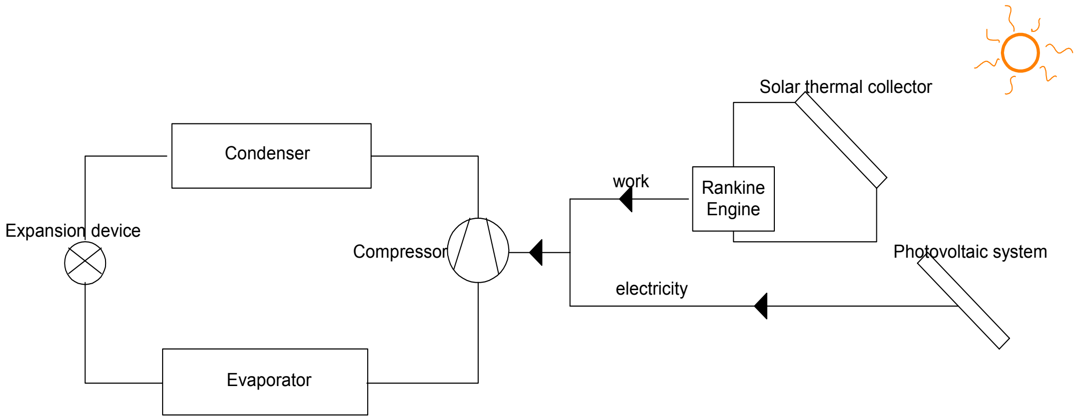


Fig. 3.9.1 Solar-driven vapour compression refrigeration cycle

The compressor for the solar-driven system usually is a direct current (DC, 12 or 24 volts) compressor since the electricity output from the PV cell is the direct current. Inverter is needed to convert DC electricity to be AC electricity when using AC-compressor.

Battery is needed to prolong the cooling period when there is lack of sunlight. Battery's capacity is generally 340 Amp-hour (Foley, 1993). The size of the PV array depends on the available insolation of each area. The small application such as a vaccine box or a cooling box is more economic than the large one. Other electric equipments can be shared electricity from solar cells such as lighting, radio transmitter, ventilation fan or television; thus make the system more efficient.

The power-driven compressor requires Rankine engine to convert heat from the solar thermal collector into a useful work for the compressor. A high technology solar collector is needed since the temperature requirement for the Rankine machine is quite high.

Table 3.9.1 Advantages and Disadvantages for the solar vapour compression refrigerator

Advantages	**Disadvantages**
High COPs	For a PV system, installation cost is high and it requires battery for energy backup.
Simplicity for the refrigeration system	Noisy from compressor
Long term experiences that is easy to maintenance when the problem happens	Required high technical knowledge for PV system
Low price	Refrigerant can be leaked.
Require little maintenance	
Low heat loss	
Can be used with various refrigerants	
Widely commercial available	
Adjustable from a small to a large system	

5. Review of the papers from the ISES congress on solar cooling Technology

Solar assisted air-conditioning of buildings –an overview

Authors: Hans-Martin Henning and Edo Wiemken

The overview and the state of the art of the solar assisted air-conditioning are presented including the potential, economic analysis and the limits. Available cooling technologies are an absorption, an adsorption and a desiccant cooling cycles. There are some large-scale applications (above 40 kW) available but there is no market available for small-scale system.

The COP for the single step absorption machines can be obtained about 0.7 at the driving heat of 80-100°C and about 1.15 for the double-stages chiller at 140-160°C. Water-LiBr pair is usually used as the working medium.

The adsorption system usually runs as a quasi- continuous operation system. Water is commonly used as the refrigerant and silical gel as the sorbent. The COP can be got at 0.6 at the driving temperature about 80°C.

The open cycles – desiccant cooling system is usually integrated with the ventilation system. Water is used as the refrigerants and Silica gel or LiCl is used as the sorption material.

From the economic analysis, the authors show that the solar assisted cooling is not economically feasible for any of the configurations studied (Freiburg, Madrid and Athens). Improvements in the performance of the thermal-driven cooling machines can play an important role in order to approach this limitation.

Technical options for a solar-driven cooling system

Authors: Wimolsiri Pridasawas and Per Lundqvist

An overview of state of the art solar cooling technology towards sustainability is introduced. The technical consequences of the solar-cooling technology from energy source to service, for both solar thermal-driven systems and photovoltaic-driven systems are presented. The importance of local conditions and demands is introduced as a guideline for choosing a suitable solar cooling system. The principle of cooling technique is briefly explained including advantages and disadvantages of each system.

Desiccant cooling technology powered by solar thermal air collector systems

Authors: Ursula Eicker, Martin Huber, Uwe Schurger, Jurgen Schumacher, Andreas Trinkle

Type of the system: Solid- Desiccant cycle
Refrigerants: Water/LiCl + Silica gel matrix
Cooling Load: 12 000 m^3/h
Regeneration temperature: 70°C
Efficiency: COP=0.5-0.6

Solar-collector technology: Flat-plate
Note: integration with the cooling of the PV facade

Prototype for a novel solar powered ejector air-conditioning system in Mazunte, Mexico

Authors: J.L. Wolpert, S.B. Riffat and S. Redshaw

Type of the system: Ejector cycle
Refrigerants: HCF77
Cooling Load: 13 kW
Regeneration temperature: 140°C
Efficiency: COP=0.67 (from simulation)
Solar-collector technology: Evacuated Heat Pipe
Note: not yet installed

Development of an optimised solar driven Diffusion-absorption cooling machine

Author: Uli Jakob, Ursula Eicker, Ahmed H. Taki, Malcon J. Cook.

System: Diffusion-absorption system
Refrigerant: Ammonia-water and helium
Cooling load: 2.5 kW
Generating temperature: 120 – 140 °C.
Evaporating temperature: -10 – 5 °C
Solar collector: commercial vacuum tubes
Efficiency: 0.1 to 0.3 (COP) at 1.5 kW actual capacity of the first prototype. But no detail is given for the second one.
Purpose: For refrigeration system
Three loops: solution circulation (ammonia-water), auxiliary gas circulation (helium) and solar circulation.

Project description solar cooling system in Pristina/Kosovo

Authors: Ernst MeiBner and Christian Holter

System: Absorption system, two each 45 to 55 kW capacity
Backup chiller: Electric driven vapour compression system of 30kW capacity
Cooling tower: 220 kW
Refrigerant: Water-lithium bromide
Cooling load: 110 – 120 kW
Heating load: 170 kW
Generating temperature: 75 – 95 °C
Evaporating temperature: 8 °C
Solar collector: flat plate solar collectors of total area of 227 m^2
Efficiency: COP = 0.7 at 91 °C heating temperature and 26.5 °C cooling water temperature
Purpose: Air conditioning.
Storage tank: 4.0 m^3
Cold storage tank: 1 m^3

Average COP of the cooling machine was 75%

On sunny days both machines can be operated by solar energy only.
For morning and evening air conditioning the backup chiller is used.

Solar assisted Cooling of the New Federal Environment Agency Building in Dessau

Authors: Edo Weimken and Hans-Martin Henning

System: Adsorption
Refrigerant: water (refrigerant) and silica gel (sorption)
Cooling load: 70 kW
47% by evaporating cooling via the cooling tower.
The remaining cooling load is covered by the thermal chiller with 50% heat input supplied by the solar system and the remaining from district heat supply.
Generating temperature: 55 °C – 90 °C
Evaporating temperature: 8 °C
Solar collector: Vacuum tube collector of 350 m^2 solar fraction decreases when flat plate collectors are used.
Storage tank:22 m^3
Efficiency: COP below 0.65
Purpose: air conditioning

Evaluation of a Zeolite-water Solar adsorption Refrigerator

Authors: Miguel Ramos, Rafael L. Espinoza, Manfred J. Horn, Antonio Pralon, Ferreira Leite

System: Adsorption system
Refrigerant: water (refrigerant, mineral zeolite (adsorber of 4 mm average diameter)
Max adsorption capacity: Zeolite water pair about 0.3 kg of adsorbate/kg of adsorbent
Regeneratetemperature: 200 – 300 °C
Evp temperature: -8 – 0 °C
Solar collector: Parabolic solar collector
Purpose: Refrigeration

Thermo chemical Accumulator - TCA

Authors: Frederik Setterwall, Chris Bales, Gøran Bolin

System: Energy storage
Thermo chemical Accumulator: Heat for cooling and heating purposes is stored in a thermal energy storage. The energy density is based on the volume of the salt solution. The maximum energy density, excluding the volume of the condenser is1190 MJ/m^3 (330 kWh/m^3). It includes the vessels, main reactor, condenser and internal heat exchangers. Storage capacity=75 kWh (94%), cooling load=50 kWh (63%), heating capacity=71 kWh (88%)
Efficiency: For cooling, COP is 0.67. For heating 95% of the charged energy is recovered.
Refrigerants: water, silica gel
Charging temp: 63 - 108 °C
Evaporating temperature: greater than 0 °C

Solar collector: not available
Purpose: air conditioning.

REFERENCES

Bäckström, M., (1970). ***Kylteknikern***, Stockholm. (in Swedish)

Derrick, A., (1993). ***Market Opportunities for Solar Photovoltaic Technology in the Health Sector***, Working Papers on Solar Energy and Health for the World Solar Summit, WHO/EPI/LHIS/93.3, Paris.

Eicker, U., et. al., (2003). *Desiccant cooling technology powered by solar thermal air collector systems*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Fléchon, J., (1999). ***Guide to solar refrigerators for remote areas and warm countries***, *International Institute of Refrigeration (IIR)*, Paris.

Foley, Gerald, (1993). ***Photovoltaics in Primary Heath Care***, Working Papers on Solar Energy and Health for the World Solar Summit, WHO/EPI/LHIS/93.3, Paris.

Granryd, Eric et at., (1998). ***Introduction to refrigerating engineering***, Dapartment of energy technology, Royal institute of technology, KTH, Stockholm.

Gordon, J. M. (2000). *High-efficiency solar cooling*, ***Solar energy*** Vol. 68, No. 1, pp. 23-31, Great Britain.

Hara, T. et al., (1998). *Cooling performance of solar cell driven, thermoelectric cooling prototype headgear*, ***Applied Thermal Engineering***, Vol. 18, pp. 1159-1169, USA.

Henning et. al, (2003). Solar assisted air-conditioing of buildings – an overview, ***Proceeding of the ISES congress***, Gothenburg.

Hug R., (2002). ***Cooling with solar heat: growing interest in solar air-conditioning**, The solar server*, forum for solar energy, Heindl Internet AG, Germany. http://www.solarserver.de/solarmagazin/artikeljuni2002-e.html

International Energy Agency, ***Solar Heating and Cooling Programme***, (2003). Task 25: Solar assisted air conditioning of building. http://www.iea-shc-task25.org/english/hps1/index.html, February 25, 2003.

International Institute of Refrigeration (IIR), (1999). ***Guide to solar refrigerators for remote areas and warm countries***, IIR, Paris.

Jakop, U., et. al., (2003). *Development of an optimised solar-driven diffusion-absorption cooling machine*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Lamp, P. and Ziegler, F., (1998*). Review paper of European research on solar-assisted air conditioning***, International Journal of Refrigeration***. 21(2), 89-99.

Lucas, (1988) IIR news, ***International Journal of Refrigeration***, Vol. 21, No. 2, pp. 88, Great Britain.

Lundqvist, P., (1993). ***Stirling cycle heat pump and refrigeration***, Ph.D. dissertation, Trita Reference Report No. 13/9, ISSN 1102-0245, Royal Institute of Technology, Stockholm.

McVeigh, J.C., (1984). ***Solar Cooling and refrigeration***, Ambient Press Ltd., UK, (1984).

MeiBner, E., et. al., (2003). *Project description solar cooling system in Pristina/Kosovo*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

National Association of Home Builders, (2001) ***Results from the field: Technology iventory, Solar cooling,***
http://www.nahbrc.org/ToolBase/pandt/tech/abstracts/hvacab18.html, Febuary 01, 2001

Pridasawas, W., and Lundqvist, P., (2003). *Technical Options for a Solar-driven Cooling system*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Pridasawas, W., (1999). **Solar-driven Refrigeration system: Msc. Thesis**, Department of Energy Technology, Royal Institute of Technology, Stockholm.

Ramos, M., et. al., (2003). *Evaluation of a zeolite-water solar adsorption refrigerator*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Rannels, J., (2000). *The DOE office of solar energy technologies' vision for advancing solar technologies in the new millennium*, ***Solar Energy***. 69(5),363-368.

Setterwall, F., et. al., (2003). *Thermo chemical accumulator - TCA*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Sofrata, H., (1996). ***Heat rejection alternatives for thermoelectric refrigerators***, Energy Conservation and Management, Vol. 37, No. 3, pp. 269-280, Great Britain.

Thévenot, R., (1979). *A history of refrigeration throughout the world*, ***International Journal of Refrigeration***, Paris.

Tsur, Y. and Zemel, A., (2000*).* ***Long-term perspective on the development of solar energy***, *Solar energy*. 68(5), 379-392.

US Department of Energy (DOE), Energy Efficiency and Renewable Energy Network (2001). ***Active solar cooling and refrigeration***.
http://www.eren.doe.gov/consumerinfo/rebriefs/ac2html, Febuary 2, 2001.

Weimken., E., et. al., (2003). *Solar assisted cooling of the new federal environment agency building in Dessau*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

WHO (1996). ***Solar Energy and Rural Health Care***, Fact Sheet N132, September 1996.
http://www.who.int/inf-fs-en-fact132.html, March 12, 2001.

Wolpert., et. al., (2003). *Prototype for a novel solar powered ejector air-conditioning system in Mazunte, Mexico*, ***Proceeding of the International Solar Energy Congress 2003***, Gothenburg.

Solar Thermal Technologies for Seawater Desalination:

State of the Art

Jenny Lindblom
Renewable Energy Systems
Luleå University of Technology
Sweden

Solar Thermal Technologies for Seawater Desalination: state of the art

Jenny Lindblom

Renewable Energy systems, Luleå University of Technology, SE-97187 Luleå, Sweden.
Tel: +46 920 491330, Fax: Fax: +46 920 491697, Email: jeli@sb.luth.se

ABSTRACT

Solar desalination is a rapidly growing field of research. The coming global oil crisis implies that alternatives to the conventional desalination plants based on fossil fuels must be developed. Solar desalination can be either direct, with collectors and condensers integrated with each other, or indirect, with condensers externally connected to the condensers. Direct solar desalination requires large land areas and has a relatively low productivity compared to the indirect technologies. It is however competitive to indirect desalination plants in small-scale production due to its relatively low cost and simplicity. Indirect solar desalination usually means combining conventional desalination techniques, such as MSF, ME or RO, with solar collectors for heat generation. This state of the art report presents the principles and characteristics of some of the recently developed direct and indirect solar desalination techniques.

1. INTRODUCTION

There is a severe lack of fresh water in the world today. Along with the deterioration of existing water supplies, the growing world population leads to the assumption that two thirds of the population will lack sufficient fresh water by the year 2025 [1]. The areas with the severest water shortages are the warm, arid countries in the northern Africa and southern Asia within the latitudes 15-35ºN [2]. In view of these facts, desalination seems to be the only realistic hope for a new source for fresh water.

The regions in most need of additional fresh water are also the regions with the most intense solar radiation. For this reason thermal solar energy in desalination processes should be the most promising application of renewable energies to seawater desalination [16]. The situation today is, however, somewhat different, since only 0.02% of the global desalination capacity is represented by renewable energy systems [17].

The main problems with the use of solar thermal energy in large scale desalination plants are the relatively low productivity rate, the low thermal efficiency and the considerable land area required. Since solar desalination plants are characterised by free energy and insignificant operation cost, this technology is, on the other hand, suitable for small-scale production, especially in remote arid areas and islands, where the supply of conventional energy is scarce [13-14]. The use of solar energy for driving the desalination plant is also motivated in these areas by the fact that they imply a way for energy independence and water insurance [11]. The low environmental impact as well as the easy operation and maintenance are also incitements for this technology [17-18].

A solar distillation plant may consist of separated or integrated systems for the solar collector and the distiller. Integrated systems are often referred to as “direct solar desalination” and usually involve different types of solar stills. Separated systems are known as “indirect solar desalination” [16], in which the thermal energy from the sun is used for either heating the seawater or generating steam in conventional distillation plants, using for example Multi-Effect (ME), Multi-Stage Flash (MSF) or Reverse Osmosis (RO) systems [19].

2. DIRECT SOLAR DESALINATION

The method of direct solar desalination is mainly suited for small production systems, such as solar stills, in regions where the freshwater demand is less than 200 m^3/day [18]. This low production rate is explained by the low operating temperature and pressure of the steam.

The original solar still can be described as a basin with a transparent cover of e.g. glass. The interior of the still contains seawater and air. When the seawater is heated by solar radiation, it starts to evaporate and the formed vapour is mixed with the air above the water surface. On meeting the inside of the glass ceiling of the still the humid air is re-cooled and some of the vapour condenses on the glass. If the glass cover is tilted, the formed condensation drops will start running down the cover by gravitational forces, and may then be collected at the side of the still (fig. 1).

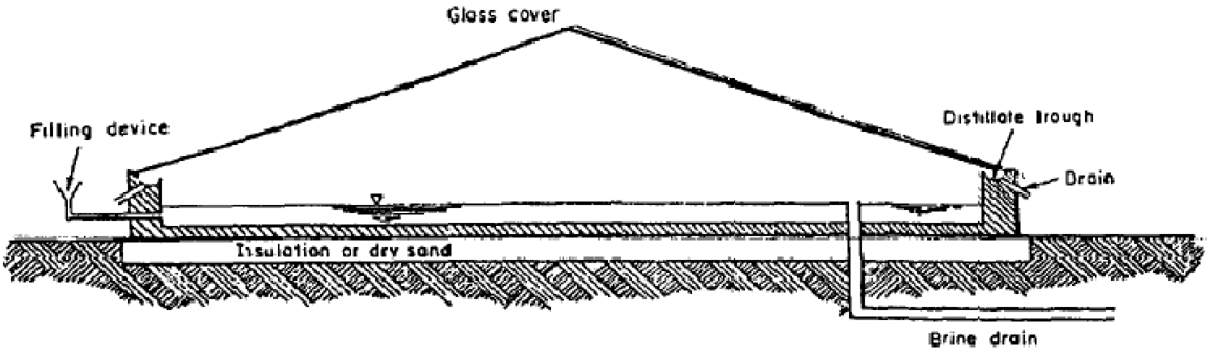


Figure 1: Basic solar still for seawater desalination [18].

One of the main setbacks for this type of desalination plant is the low thermal efficiency and productivity. This could be improved by a number of actions, e.g. injecting black dye in the seawater, reducing the heat conduction through basin walls and top cover or reusing the latent heat emitted from the condensing vapour on the glass cover. Another solution would be to separate the solar collector and the saline water so that corrosion damages, and thereby efficiency losses are avoided [24].

2.1 Single-effect solar stills

In the original solar still construction, also called single-effect solar still, only one layer of glazing covers the still. This enables a large quantity of the latent heat from the condensation process to disappear from the still by conduction through the glazing. Today's state-of-the-art single-effect solar stills have an efficiency of about 30 – 40 % [6] and a production rate in the order of 6 litres per day and square meter of collector surface [11], but research and development on this system continues. The development techniques may be classified as shown in fig. 2 [18]. Passive solar stills utilize the internal heat from the still for the evaporation process, while active stills make use of external sources, such as solar collectors or waste heat from industries. Fath [18] has done a valuable review of the latest development on this topic.

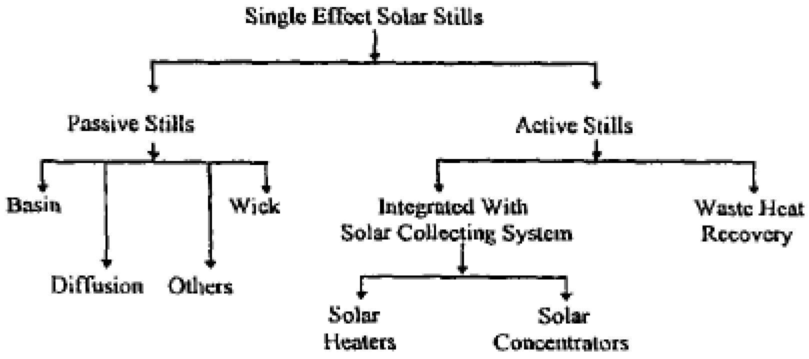


Figure 2: Classifications of developments for single-effect solar stills

2.1.1 Basin stills

Three of the main research topics on the passive basin still are summarised:

1. Single slope versus double slope basin stills: A comparison of a single and double slope solar still, shows that due to the daily and seasonal movement of the sun, a double-sided still can absorb more solar radiation than a single slope basin still. The single slope, on the other hand, has less convection and radiation losses, and the shaded region can furthermore be used for an additional condensation surface. Tiwari et al [20] performed a study of the two configurations and concluded that the single slope basin still performs better than a double slope in cold climate, whereas the opposite result was achieved in warm climates.

2. Still with additional cooling: The evaporation rate increases with increasing temperature difference between the water surface and the glazing. This can be done by for example heating the water more or cooling the cover. Another alternative cooling system for basin type solar stills was presented by Haddad [8], where an external condenser, constructed as a packed bed storage tank, was integrated with the still. The packed bed condenser was cooled during the night, using a radiative cooling panel by circulating water into the packed bed condenser and the radiative cooling panel (fig. 3).

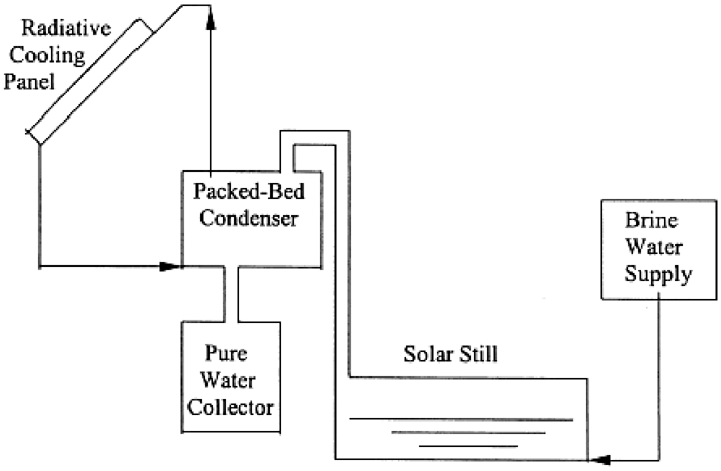


Figure 3: Schematics of the solar still and condenser [8].

The cooling panel utilized the cold effective sky temperature, which normally is 10-25ºC lower than the ambient temperature, in order to cool the rock domain in the packed bed storage during the night. In doing so, the tank temperature was lowered to nearly effective sky temperature. At the beginning of the daylight, water was evacuated from the storage tank, as vapour started to form on the surface of the solar still. By buoyancy forces created in the condenser, the vapour was sucked through the duct between the still and the condenser, which made additional driving forces unnecessary.

Several advantages in this system were noted. Among other things was the heat loss reduced since the temperature inside the still was lowered, and the lower vapour partial pressure in the still contributed in a faster evaporation rate. Also, the low temperature of the condenser enhanced the condensation rate.

3. A typical single effect solar still was recently designed for water purification in remote areas in Venezuela. The solar distiller was of basic design and used for brackish water purification. A community of 100 inhabitants was selected for the experimental set-up. Based on the calculated required daily water consumption the plant was designed to 380 distiller units, with an estimated daily productivity of 5 litres each. Experimental and economic analysis of the system concluded that it should be an attractive alternative to the current method of water distribution by means of cistern trucks [21].

2.1.2 Stills integrated with greenhouses

Greenhouses combined with solar stills represent an interesting possibility for the development of small-scale cultivation in places where only saline or brackish water is available. A version of this system was constructed and analysed by Chaibi et al [3-4], where the south slope of the greenhouse roof was built as a solar still (fig. 4).

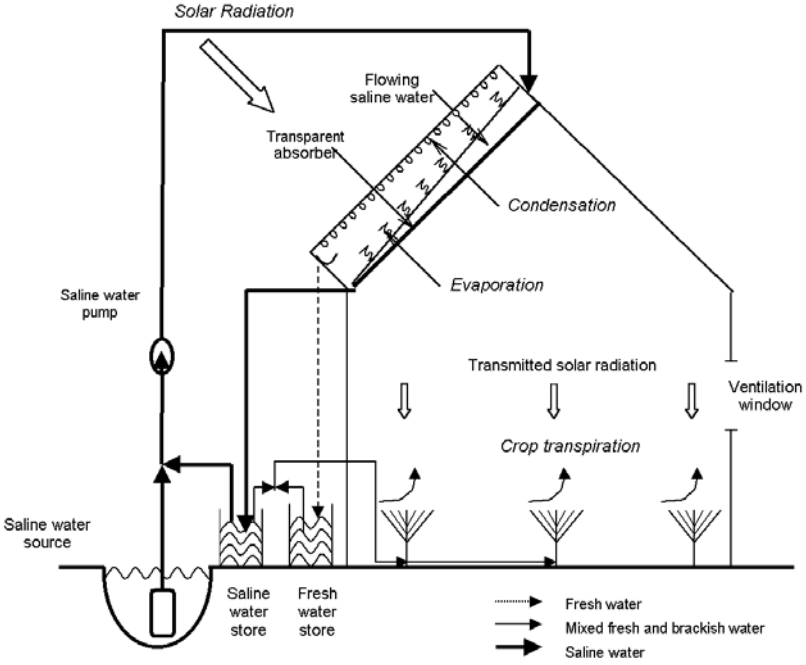


Figure 4: System principle for water desalination integrated in a greenhouse roof. [3]

During the day, saline water was pumped from a reservoir to the rooftop of the greenhouse, from where it was distributed evenly to the evaporation surface in the still. The top cover of the still was a regular glass sheet, while the bottom of the solar still consisted of an only partly light transparent material, which absorbed a substantial amount of the solar irradiation, but transmitted the wavelengths that are favourable for the photosynthesis of vegetation (the photosynthetic active radiation, PAR, has the wavelength interval 380 – 710 nm).

Since most of the heat radiation was absorbed in the still, the temperature of the greenhouse air was lowered, which lead to better climate for the crops and less ventilation requirement. In the end, this lead to a decrease in the water consumption of the crops.

The formed water vapour condensed on the top glazing, ran along the inner wall of the top cover, and was collected in the fresh water store. The residue of the feed water was collected in a separate storage. The returned feed water was partly returned to the feed water duct for another loop in the still, and some of the residue saline water was also mixed with the fresh water before the irrigation to bulk out the supply. The desalination roof was operated during both day and night, as excess heat was stored in the saline water storage.

Compared to a conventional single glass greenhouse, considerable less extreme climate conditions inside the greenhouse could be registered with the roof desalination system. It was also shown that with 50 % of the roof built as a solar still, the irrigation need for low canopy crop could be satisfied, but analyses of crop growth also indicated a lesser yield by approximately 25 %. An economical analysis of the described system, a single-effect desalination system with heat pipe collectors and a solar multiple condensation evaporation (SMEC) cycle process were performed, which showed that the integrated solar still was more economical than the other two by 35 % and 50 %, respectively [4].

2.2 Multi-effect solar stills

Multi-effect solar stills are designed to recycle some of the latent heat from the condensation by using it for preheating either the feed water or the seawater within the still. The former may be accomplished by e.g. using the feed water duct as the condensation surface for the water vapour. The saline feed water is then preheated by the heat released from the condensing vapour, and the condensation surface is kept continuously cool. A multi-effect solar still can in this way produce fresh water up to 20 litres per day and square meter of collector area. The increased production rate that follows by this recycling, must however be measured to the cost for the more complex construction that follows [18].

2.2.1 Solar still with preheating of feed water

An air-blown, double-glazed solar still has been proposed by Mink et al [6]. The solar still is divided into two chambers, the upper one being the evaporator and the lower the condenser. A metal sheet separates the evaporator and condenser along the length of the still, except for a small gap in the upper part of the still (fig. 5).

Feed water is lead upwards through the condensation chamber in a serpentine tube and is let out on to the evaporation surface, consisting of a black porous textile, on the top of the still. The textile absorbs the water evenly, as the water flows down along it. Air enters from the bottom of the evaporator, sweeps away the evaporated vapour from the wetted textile surface and leads it down to the condenser department via the upper gap between the two chambers. The vapour then condenses on the serpentine tubes and preheats the feed water. The distillate is finally collected at the bottom of the condenser.

Experimental investigations on this type of still showed that the solar still performance can be enhanced by further optimising area relations between the evaporator and the condenser. The received daily water production rate was in this experiment approximately 1 kg/m^2.

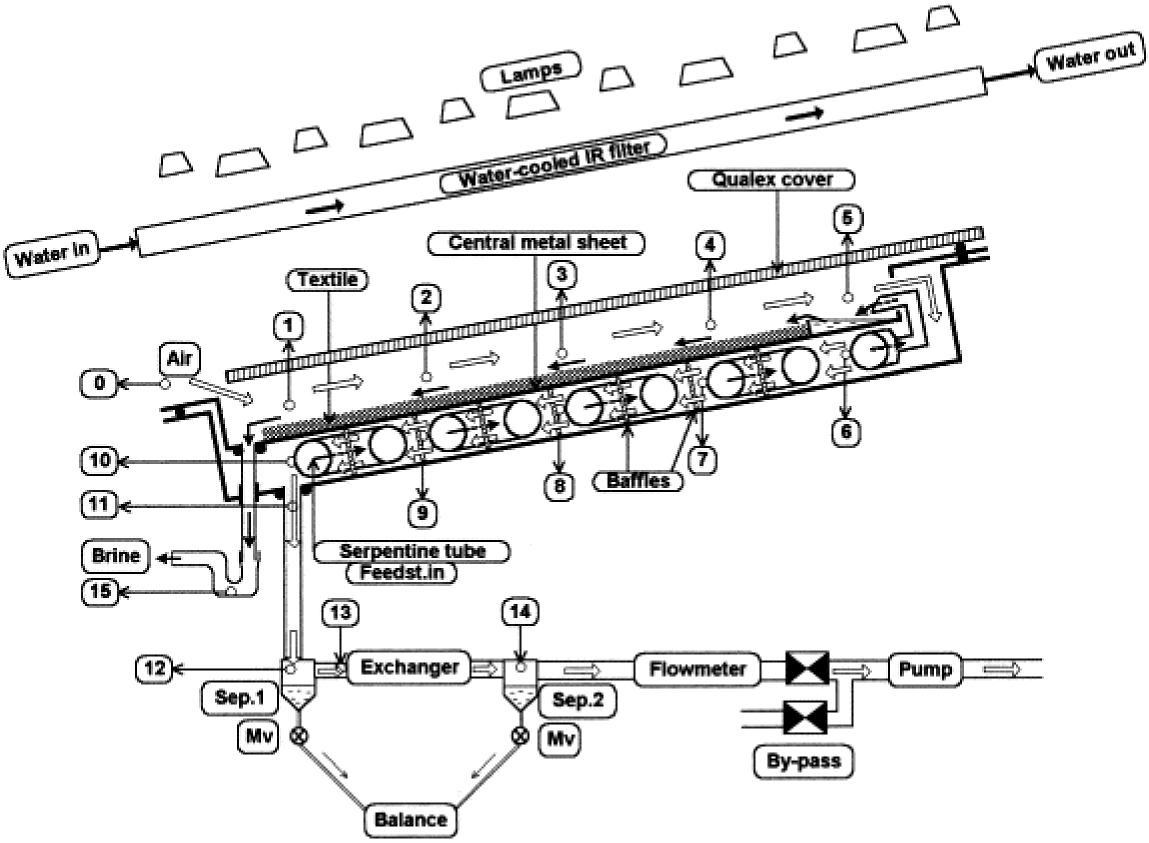


Figure 5: Longitudinal cross-section of the experimental multi-effect solar still set-up [6]

2.3 Water desalination with humidification-dehumidification (HD)

One of the problems that negatively influences the still performance is the direct contact between the collector and the saline water, since this may cause corrosion and scaling in the still and thereby reduce the thermal efficiency [24]. In HD desalination air is used as a working fluid, which eliminates this problem. Systems based on HD consist of a compact unit, containing two heat exchangers for evaporation and condensation, respectively. The constructions are usually lightweight and inexpensive, and work at atmospheric pressure [15]. Due to relatively low desalination capacity, the system performance must however be improved before it can be economically competitive.

2.3.1 Desalination with humidification – dehumidification using an open-air cycle

Dai and Zang [9-10] proposed an open-air cycle desalination system, in which seawater was heated by the sun in a collector and then sprayed on the surface of a honeycomb wall in the humidifier (fig. 6). Air was blown through the humidifier, where it became hot and humid. The air was thereafter led to the condensation area between the feed water tubes, where it was cooled and fresh water precipitated into the collection container. In order to increase the thermal efficiency, the part of the warm seawater that was not picked up by the air in the honey comb was collected and led back into the seawater tank for another round in the humidification-dehumidification process. Since the air flowed in a straight

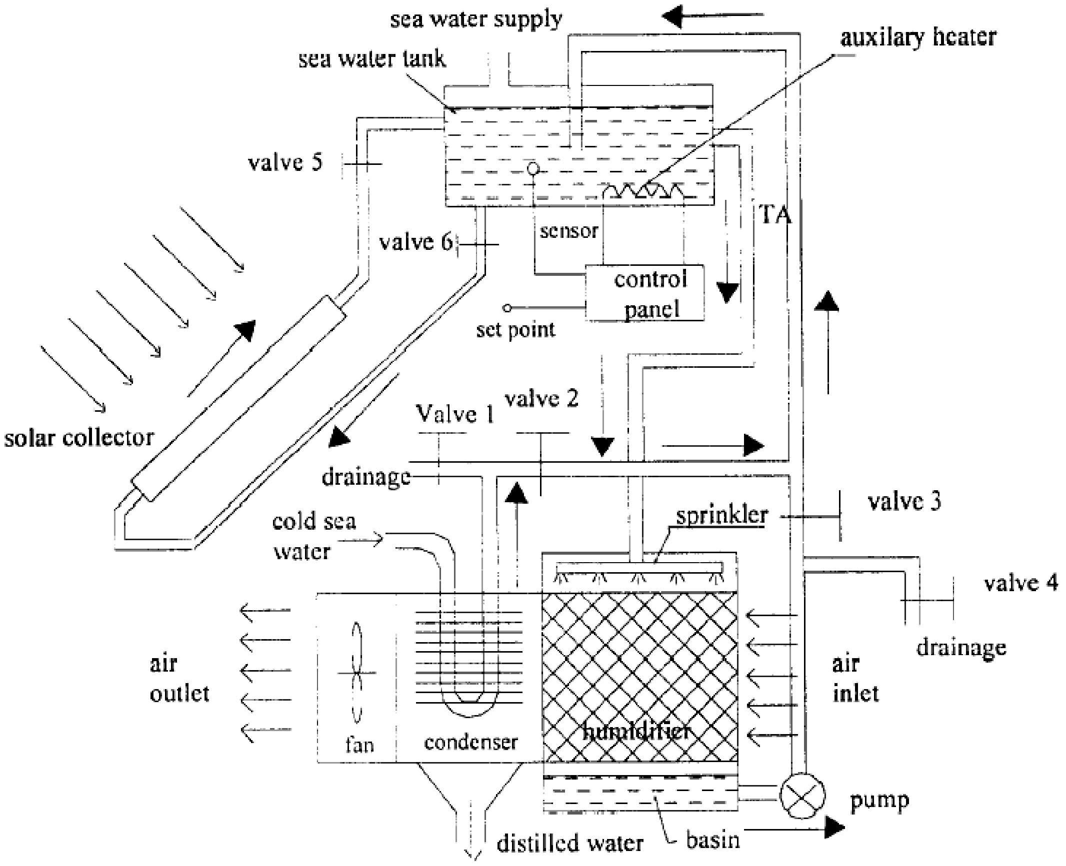


Figure 6: Schematic of solar desalination plant suggested by Dai and Zhang [9-10]

line, unnecessary pressure losses were avoided. During experiments with this desalination system, a freshwater gain of 6.2 kg/m^2 per day was observed for cases when solar irradiation was 700 W/m^2 and the operation time was 8 hours per day. The thermal efficiency of the system was estimated to 85 %.

2.3.2 Solar air-heating with stepwise humidification

The low water production capacity in ordinary solar stills can partly be explained by the small difference in vapour content of the air before and after the condenser [12]. In a single-step system, the only possibility for improving this is either to increase the operating air temperature or flow rate, which leads to unrealistic investments.

An interesting solution to this problem is presented by Chafik [12], where the air is heated and humidified in several steps inside a simple solar collector. This both lead to higher vapour content in the air and a lower required airflow rate. The course of the procedure is as follows (fig. 7): Air with an initial temperature and specific humidity of e.g. 25ºC and 10 g vapour/kg dry air, respectively, enters the solar air-heating collector. During the first step in the collector, the air is first warmed to 50 ºC and then humidified by sprinklers with seawater until the air is approximately saturated.

Due to the heat required for the evaporation of the water, the air temperature sinks to about the wet bulb temperature of 23ºC. In the following step, the air is again heated to 50ºC and then humidified in the same way to a vapour content of 28 g vapour/kg dry air. The air temperature goes down to 31ºC. During the third step the air is heated to 56ºC and humidified to 36 g vapour/kg dry air at 35ºC.

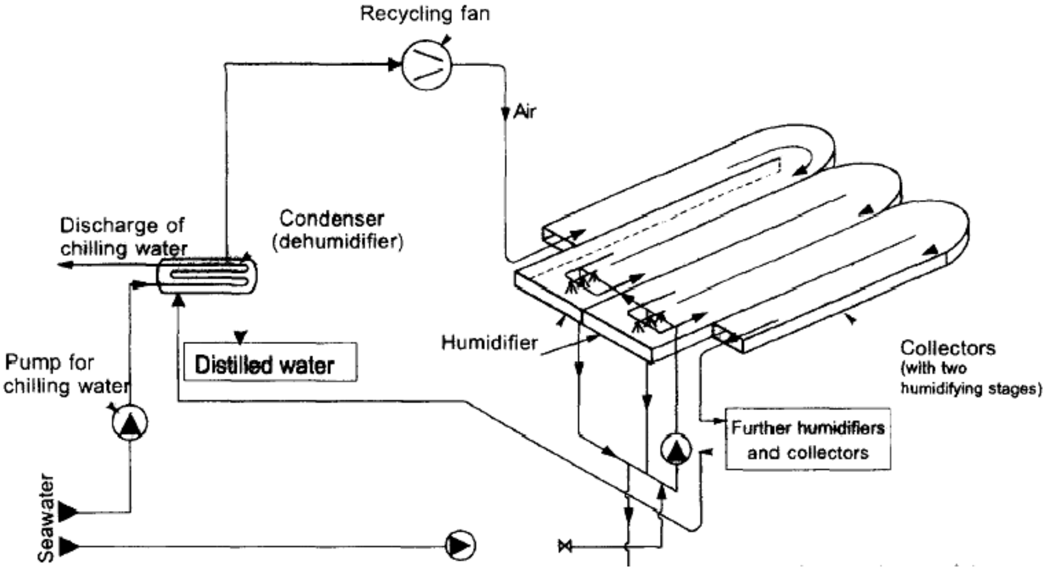


Figure 7: Scheme of the stepwise heating and humidification [12].

The heating and humidification process goes on for 15 steps, after which the air has reached a humidity of 148 g vapour/kg dry air at 60ºC (fig. 8). This high water content is identical to the humidity that can be achieved by injecting water into heated air of about 450ºC. By cooling the warm, humid air exiting the 15th step to 25ºC, 128 g fresh water can be extracted for 1kg of dry air. A pilot plant for this technique is presently running in Bochum, Germany, where 1 m^3 per day is produced.

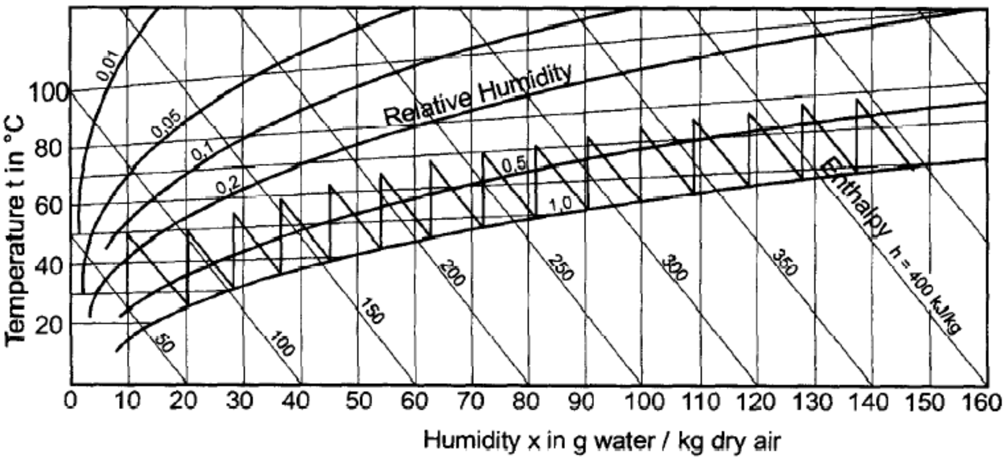


Figure 8: Psychometric chart with stepwise heating and humidification [12]

3. INDIRECT SOLAR DESALINATION

Every day desalination plants around the world produce about $23 \cdot 10^6$ m^3 of fresh water [17]. For this production rate, desalination systems of the industrial scale are required. The majority of the existing desalination plants for this purpose are of the types Multi-Effect (ME), Multi-Stage Flash (MSF) and Reverse Osmosis (RO). Usually these systems use fossil fuels as the energy source for either heating or electric power generation. They are also characterised by high operating cost and the need for highly skilled operation and maintenance personnel.

One of the most promising directions for research and development for these large-scale desalination facilities seems to be the use of solar energy because of its low cost, availability and relatively simple maintenance. An investigation by Garzia-Rodriguez [19] showed that parabolic trough collectors and salinity gradient solar ponds should be the best choices for this purpose.

3.1 Multi-Effect (ME)

The ME is one of the most promising evaporation techniques today [22]. It has long been thought that the ME system is not suitable for production rates lower than 100 m^3/day because of the need for qualified maintenance and electricity supply [5]. However, by the use of solar energy instead of fossil fuels, ME may well be made economically feasible [19].

The essential feature of ME is that saline water stepwise evaporates by heat transfer from condensing steam, transported in a bundle of tubes. The most commonly used configuration of an ME plant is horizontal tube bundles with 8-16 evaporation steps (fig. 9).

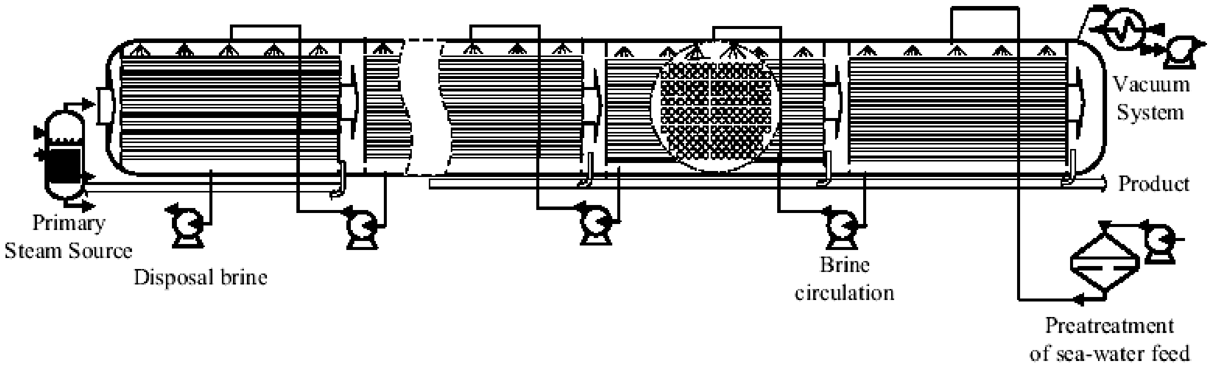


Figure 9: Schematics of a ME distillation plant with horizontal tubes [22]

At the first step, the condensing steam is generated externally, by for example fossil fuel or solar collectors. The produced steam from the evaporation of the brine is then used in the subsequent step, which operates at a slightly lower pressure and temperature, so the energy retrieved from the condensing steam can be used for further evaporation of the brine [23].

Pilot plants for ME desalination with solar heating are being developed and investigated. In the Arabian Gulf, such a plant exists using parabolic through collectors for producing 6000 m^3/day and another ME plant using solar ponds for the production of 30 m^3/day are constructed at the university of Ancona, Italy, to mention just two of them [17].

3.2 Multi-Stage Flash (MSF)

The most common and simple technique for large-scale desalination is at present the Multi-Stage Flash distillation, which globally produces a total amount of about 10 million ton of fresh water every day.

The system works with pressurised seawater that flows through closed pipes and exchanges heat with condensing vapour. When the seawater is heated to a certain degree it is lead into a low-pressure chamber, where it is flash-evaporated. The resulting vapour is then cooled and collected in a tank for freshwater [18].

Several medium scale plants for MSF desalination using solar energy have recently been implemented. One of the most commonly type of solar collectors used are salinity gradient solar ponds, such as the desalination plant in Margarita de Savoya, Italy, with a capacity of 50 – 60 m^3/day, or in El Paso, Texas, with a capacity of 19 m^3/day. Another frequently occurring source for solar thermal energy is the parabolic trough collector, which is used in i.e. a MSF desalination plant in Kuwait for a production rate of 100 m^3/day [17].

3.3 Reverse Osmosis (RO)

In the reversed osmosis system, seawater is forced under pressure through a series of membranes that physically remove salt molecules [5]. In contrast to distillation systems, where separation occurs through difference in evaporation temperatures, the separation process is here determined by size and diffusivity differences [23].

Recent progress have been made on the subject of using RO in seawater desalination, and low pressure RO systems are now able to operate at pressures as low as 20 bar [23]. Ongoing research and development on the subject of combining RO with solar energy predict a good possibility of finding a cost-effective solution. Among the suggested solar driven plants are RO desalination driven by solar produced steam [19].

3.4 Salinity-gradient Solar ponds

As the sun shines over a lake or a pond, the water absorbs some of the irradiation and is warmed. Surface water quickly looses this added heat due to heat and mass convection with the ambient air. Since the underlying water in the pond now is warmer and thereby lighter than the surface, convective circulation begins, where warm water from the bottom rises and the colder water from the surface layer sinks.

The salinity-gradient solar pond is constructed in such a manner that the convective circulation in the pond is prohibited by making the bottom water much denser than the surface water. In doing so, the solar radiation absorbed in the deep water can be stored [16]. The general solar pond consists of three layers of different temperature and salt content (fig. 10).

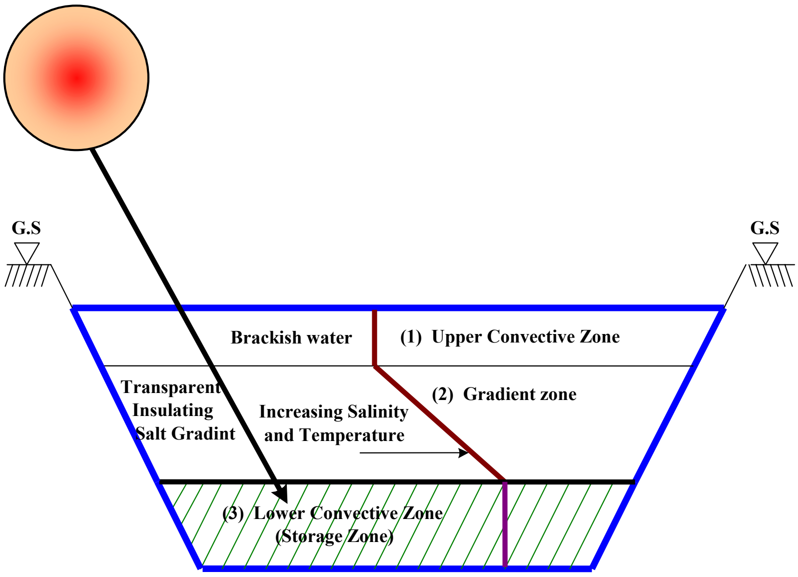


Figure 10: Typical salinity-gradient solar pond [2]

The top layer, usually about 0.8 – 1 meter deep, is at atmospheric temperature and has a low concentration of salt. The second layer is the so-called gradient zone. Here the temperature and salt concentration increase with the water depth, which usually is 1 – 2 meters. The bottom layer in the solar pond, also called the storage zone, is very dense and is heated up to 100ºC [2]. Since the water in the gradient zone cannot rise, due to the light water on top, and cannot fall, due to the dense water beneath, convection is prevented and the heat is stored in the storage zone. The gradient zone could hence be said to work as an insulator for the storage layer. Heat is extracted by passing the brine from the storage zone through an external heat exchanger.

Solar ponds require plenty of land area, water and salt, why it is reasonable to locate them in wastelands or in deserts, close to salt works. Many countries, such as Libya, are greatly dependent on seawater desalination and are in supply of these characteristics. To use solar ponds instead of fossil fuel for heating the desalination plants would mean significantly lower production costs [2].

3.5 Parabolic through collectors

The parabolic through collector uses a trough-shaped reflector to concentrate sunlight on the receiver at up to 60 times its normal intensity. In this way, the receiver tube, which runs along the reflector's focal line, can achieve a much higher temperature than flat-plate or evacuated-tube collectors (fig. 11). The parabolic through collector systems usually include a mechanical control system that keeps the trough reflector pointed at the sun throughout the day. Parabolic-trough concentrating systems can provide hot water and steam, and are generally used in commercial and industrial applications [25].

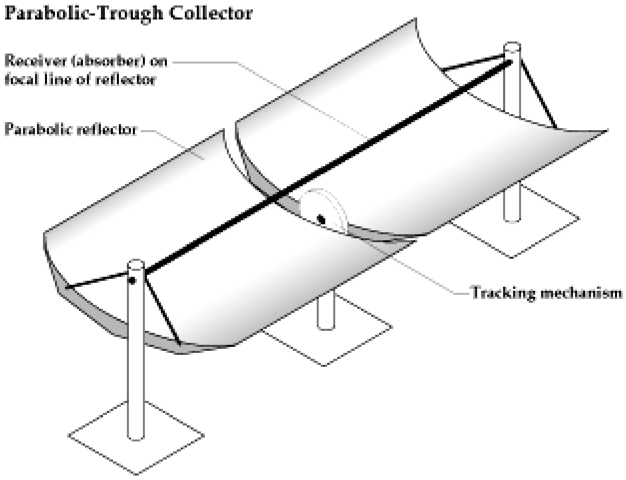


Figure 11: Schematics of a parabolic through collector [25]

4 CONCLUSIONS

The use of solar thermal energy in seawater desalination applications has so far been restricted to small-scale systems in rural areas. The reasons for this have mainly been explained by the relatively low thermal efficiency and production rate compared to the large area required. Of the 23 million m^3 of fresh water produced every day, only 46 000 m^3, or 0.02 %, originates from plant with renewable energy systems.

However, the coming shortages in fossil fuel supply and the growing need for fresh water in order to support increasing water and irrigation needs, have motivated further development of water desalination and purification by renewable energies. The most promising source within this area is, as it seems today, solar thermal or geothermal energy.

If renewable energy could be combined with large-scale desalination plants, such as ME or MSF, to a reasonable cost, this could mean a solution to one of the most pressing environmental issues of today and the near future.

REFERENCES

1. UNEPb (United Nations Energy Programme) http://www.unep.org/geo2000/english/0046.htm [2003-06-09]

2. Hussain A.K.M. Solar energy utilization in Libya for seawater desalination. Proceedings at the ISES Solar World Congress 2003, Gothenburg.

3. Chaibi M.T and Jilar T. System performance and design of roof-integrated desalination in greenhouses. Proceedings at the ISES Solar World Congress 2003, Gothenburg.

4. Chaibi M.T and Jilar T. Economy prospects for roof-integrated desalination in greenhouses. Proceedings at the ISES Solar World Congress 2003, Gothenburg.

5. Virk G.S., et al. Ambient energy for low-cost water desalination, Desalination 2001:137, pp. 149-156.

6. Mink G., et al. Design parameters, performance testing and analysis of a double-glazed, air-blown solar still with thermal energy recycle, Solar Energy 1998:64, pp. 265-277

7. Rahim N.H.A. New method to store heat energy in horizontal solar desalination still. Renewable Energy 2003:28, pp. 419-433

8. Haddad O.M., et al. enhanced solar still performance using a radiative cooling system. Renewable Energy 2000:21, pp. 459-469

9. Dai Y.J., et al. Parametric analysis to improve the performance of a solar desalination unit with humidification and dehumidification. Desalination 2002:142, pp. 107-118

10. Dai Y.J. and Zhang H.F. Experimental investigation of a solar desalination unit with humidification and dehumidification. Desalination 2000:130, pp. 169-175

11. Kunze H. A new approach to solar desalination for small- and medium-size use in remote areas. Desalination 2001:139, pp. 35-41

12. Chafik, E. A new seawater desalination process using solar energy. Desalination 2002:153, pp. 25-37

13. Naim, M., et al. Non-conventional solar stills Part 1: Non-conventional solar stills with charcoal particles as absorber medium. Desalination 2002:153, pp. 55-64

14. Naim, M., et al. Non-conventional solar stills Part 2: Non-conventional solar stills with energy storage medium. Desalination 2002:153, pp. 71-80

15. Borouni K., et al. Water desalination by humidification and dehumidifiation of air: state of the art. Desalination 2001:137, pp. 167-176

16. Garzia-Rodriguez L, et al. Comparison of solar technologies for applications in seawater desalination. Desalination 2002:142, pp. 135-142

17. Garzia-Rodriguez L. Seawater desalination driven by renewable energies: a review. Desalination 2002:143, pp. 103-113

18. Fath H.E.S. Solar distillation: a promising alternative for water provision with free energy, simple technology and clean environment. Desalination 1998:116, pp. 45-56

19. Garzia-Rodriguez L, Gomez-Camacho, C. Perspectives of solar-assisted seawater distillation. Desalination 2000:136, pp. 213-218

20. Tiwari G.N., et al. Comparison of various designs of solar stills. Desalination 1986:60, pp. 191-202

21. Pocaterra, J.E. et al. Design and experimental solar desalination plant to supply potable water to remote areas of the Venezuelan peninsula de la Guajira. Proceedings at the ISES Solar World Congress 2003, Gothenburg.

22. Seimat R. Desalination: Present and future. Water International, Volume25:1, March 2000, pp. 54-65

23. Van der Bruggen B. Desalination by distillation and by reverse osmosis – trends towards the future. Membrane Technology Volume 2003:2, February 2003, pp. 6-9

24. Fath, H.E.S. and Ghazy, A. Solar desalination using humidification - dehumidification technology. Desalination 2002:142, pp. 119-133

25. U.S. Department of Energy, http://www.eere.energy.gov/solarbuildings/techdescr.html [2003-06-23]

Review of the State of the Art:

Solar Radiation Measurement and Modeling

Pongsak Chaisuparasmikul
College of Architecture, Illinois Institute of Technology, USA
and
Thomas Bache Andersen
Department of Civil Engineering, Technical University of Denmark, Denmark

Preface

This report was written at the Department of Civil Engineering at the Technical University of Denmark in connection with the course "Solar Heating". Pongsak Chaisuparasmikul and Thomas Bache Andersen wrote the report in the period from June 2nd to June 27th 2003. Supervisors were Associated Professor Simon Furbo and Associated Research Professor Louise Jivan Shah.

The purpose of this report was to work out a state of the art review report on the subject "Solar Radiation Measurement and Modeling".

Technical University of Denmark,
Lyngby
June 27th 2003

-------------------------------------- ------------------------------------
Pongsak Chaisuparasmikul Thomas Bache Andersen

Table of Contents

PREFACE 1
TABLE OF CONTENTS 1
1. INTRODUCTION 2
2. BACKGROUND AND DEFINITIONS 2
3. SOLAR RADIATION COMPONENTS 3
4. MEASUREMENT OF SOLAR RADIATION 4
  4.1 Measurement of Duration of Sunshine 4
  4.2 Pyranometer 4
  4.3 Pyrheliometer 5
5. MODELS 6
  5.1 Input - Output and Parameterisation 6
  5.2 Component to component Model 7
  5.3 Spectral model 8
  5.4 Meteorological Model 8
  5.5 Auto correlation structure model 9
  5.6 Cross correlation structure model 11
  5.7 Satellite Data 11
  5.8 Stochastic and Multi correlation 12
6. USER PRODUCTION 13
  6.1 Time-Site specific data 13
  6.2 Averaged and Condensed data set 13
  6.3 Data Forecast Map Product 15
7. CONCLUSIONS 16
REFERENCES 17
APPENDIX 1 18
APPENDIX 2 19

1. Introduction

When considering implementation of a solar energy driven application, e.g. solar collector or photovoltaic, the knowledge of the solar irradiance at the specific site and time is an indispensable parameter. Also when designing new energy efficient buildings, whether designed for low cooling or heating loads or high utilization of day lighting, parameters like solar irradiance and illumination are very important. These parameters are significant inputs when simulating the performance of solar energy driven systems or simulation of building energy demands. Therefore measurement and modeling of solar radiation is an important aspect related to solar energy.

2. Background and definitions

Due to the Earths ecliptic orbit around the Sun, the extraterrestrial solar radiation varies slightly during a year (+/- 3%), with a maximum in January and a minimum in July (Gordon, 2001). However the geometry is well known so the distance at any time can be calculated rather accurate. The Solar Constant, sc, is the extraterrestrial solar irradiance at the mean Earth-Sun distance. Through the years it has been measured by different means, with deviating results but today the value is generally recognized to be sc = 1367 W/m^2 (Gordon, 2001).

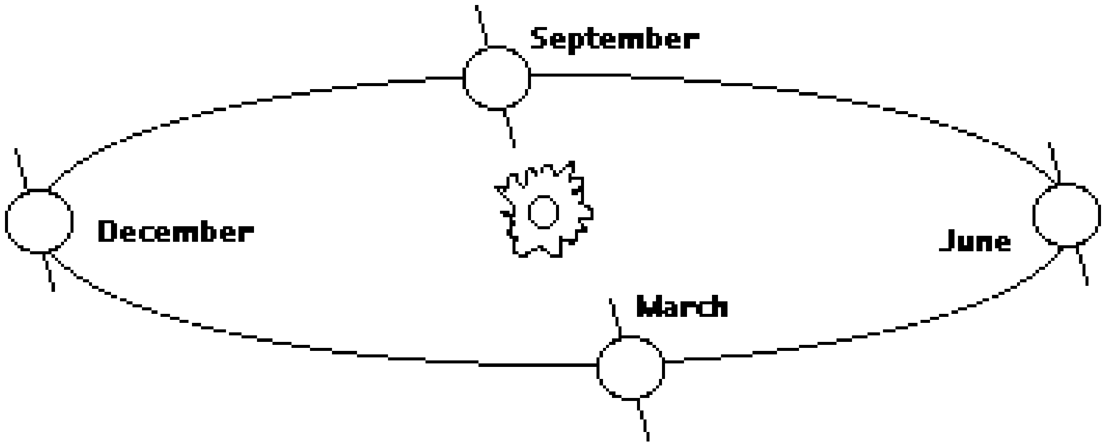


Figure 1: The Earths orbit around the Sun is ecliptic - therefore the extraterrestrial irradiance varies slightly during a year.

It is much more difficult to predict the solar irradiance at the Earths surface. The geometric effects (i.e. time of day, seasons, latitude) can, as mentioned above, be managed by calculations but other parameters like atmospheric absorption, reflection and scattering plus cloud obstructions are not deterministically modellable and will have to be considered otherwise.

3. Solar Radiation Components

Through the atmosphere the solar irradiance decreases and the spectral distribution changes. Ozone absorbs the ultraviolet radiation while water vapor, carbon dioxide and other greenhouse gases absorbs some of the infrared radiation. In the visible spectrum, the shortest wavelengths (i.e. the blue light) is scattered by air molecules and dust. Therefore only part of this radiation reaches the surface, and in the form of diffuse radiation (figure 2). The sum of the diffuse and direct radiation (that travels uninterrupted through the atmosphere, often called beam radiation) is referred to as the global irradiance. The direct normal irradiance is the radiation at a surface normal to the direct radiation. This component is used to define the angle of incidence, θ, as the angle between the direct normal radiation and the normal to the surface of interest.

The irradiance on a surface varies with the cosine of the incident angle:

$$E_g = E_d + E_r + E_n * \cos\theta,$$

Where E_d is the diffuse irradiance, E_r is the ground reflected irradiance and E_n is the direct normal irradiance.

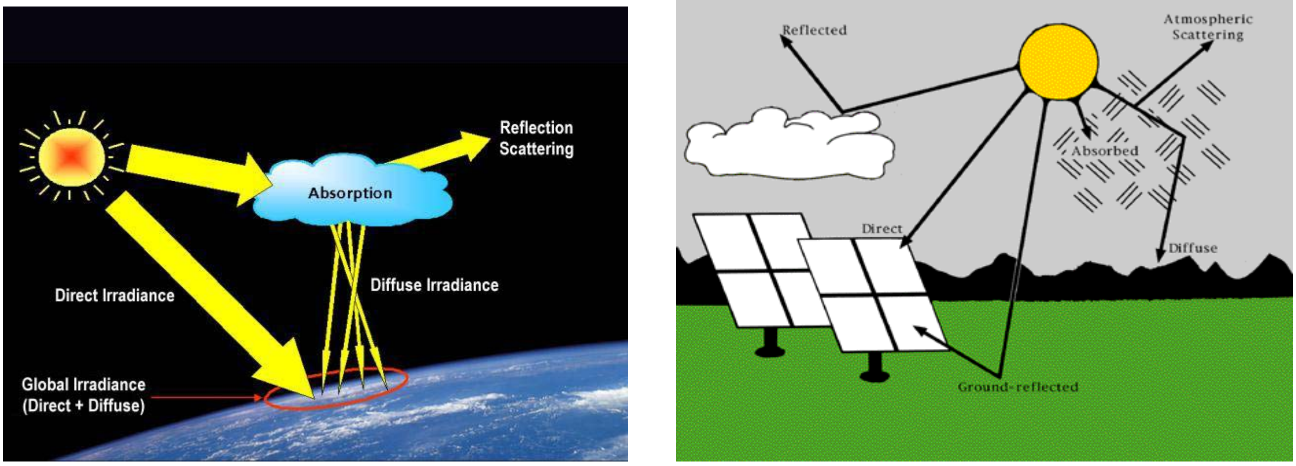


Figure 2: Sketches of the solar radiation through the atmosphere. (From http://www.volker-quaschning.de/fotos/messung/index_e.html and http://rredc.nrel.gov/solar/pubs/shining/page12_fig.html)

For architectural design, the spatial intensity distribution of the extended sky and ground radiation source is of importance. The relationship between the sky radiance, L, and the diffuse radiation is:

$$E_d = \int_{\phi=0}^{2\pi} \int_{\phi=0}^{\pi/2} L \cos\theta \sin\theta d\theta d\phi$$

Where θ and ϕ are integral angles.

For most engineering applications the total value of solar radiation (i.e. integrated over all wavelengths) is of interest. However for some applications (e.g. photovoltaic and day lighting) specific wavelengths are of interest. For that reason the spectral distribution is important.

The spectral distribution with importance for different applications is shown in figure 3. E.g. dealing with day lighting it is the part of the solar irradiance that is sensitive to the human eye (the Photopic Response curve also referred to as the Vλ function) that is of interest. The ratio of day light illuminace to solar irradiance is day light luminous efficacy.

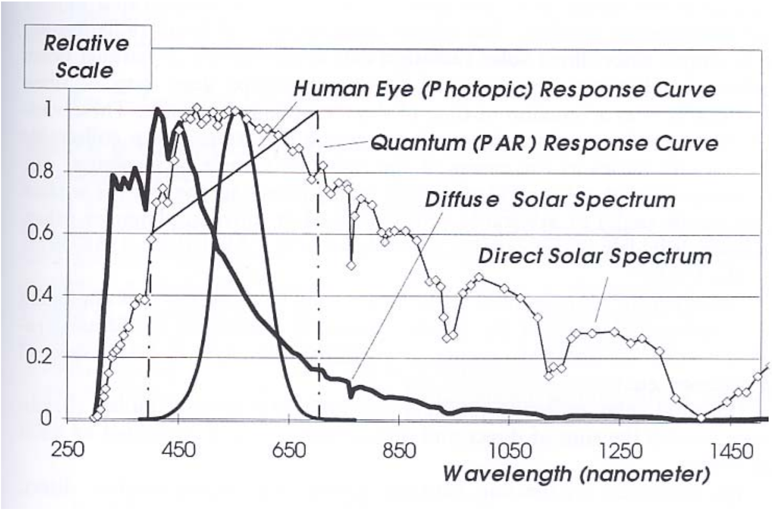


Figure 3: Typical solar spectrum at the Earths Surface (clear day, 45° solar elevation angle) compared with response curves for different applications.

4. Measurement of Solar Radiation

Almost 200 years ago, in the 1830s, the first attempts to measure the intensity of the solar radiation were performed. Since then instruments have been improved considerably encouraged by e.g. the quest for a precise determination of the solar constant (in the early 1900s), the energy crisis and solar energy development impetus in the 1970s, and the need to better understand the global climate change in the 1980s and 1990s (Gordon, 2001)

4.1 Measurement of Duration of Sunshine

The hours of bright sunshine can be used to estimate long-term averages of solar radiation. The Campbell-Stokes sunshine recorder has been widely used. The instrument consists of a solid sphere that has the function of a lens. The lens burns a paper, and the length of the burned paper provides an index of the duration of “bright sunshine”. Uncertainties are associated with this measurement, as the interpretation of what constitutes a burned portion is not clear.
Today the Foster Sunshine Switch is in use. This is a photoelectric sunshine recorder, which incorporates two photovoltaic cells (one shaded and one exposed to beam radiation). When the radiation exceeds a fixed level the sunshine will be recorded, and the output of the cells gives a measure of the duration of bright sunshine (Duffie et.al., 1991).

4.2 Pyranometer

A pyranometer is an instrument used to measure the global radiation. The instrument has a hemispherical view of the surroundings. By adding a shading ring, the direct radiation is excluded by which the pyranometer measures only the diffuse radiation. A pyranometer uses thermal sensors.

One sensor is exposed to the solar radiation, while the other is shaded. The temperature difference registered by the sensors is used as a measurement of the solar radiation.

Figure 4: Pyranometers used for measuring the global radiation (left) and diffuse radiation (right).

4.3 Pyrheliometer

The pyrheliometer is used to measure the beam radiation. The exposed detector is placed in the bottom of a long tube, which gives this instrument a very limited sky view (about 5°). Therefore it must be connected to an accurate solar tracking device.

Figure 5: Pyrheliometer used to measure the solar beam radiation.

In appendix 1 is shown a research class measurement station in Sweden.

5. Models

Models are the mathematical functions or algorithms designed to generate a desired solar radiation component from a set of inputs that can include other solar radiation related components, time and site information.

The need for models arose in order to conduct measurement each time a particular solar radiation component was needed in a particular location. Models is generally linked to a simple, easily available, solar radiation or related parameters to a more complex component that is not available.

The first models were developed to predict global irradiance from more commonly available quantities such as cloud cover and/or sunshine duration (Angstorm, 1924, Kimball, 1919).
In the 1940s and 1950s, ‘component to component’ models came into play with the work of Liu and Jordan (1960). Recently the interest has been growing for models capable of predicting solar radiation from remote sensing (satellite-based) observations.

The Liu and Jordan (1960) model was developed to related monthly averaged values of global and diffuse irradiance. Recently, however, most models have been developed to address hourly, instantaneous, time scales. Hourly time scale is quasi-instantaneous, with a few exceptions; hourly, instantaneous, models are one and the same.

We placed the focus on the hour or less because:

1) From the physical standpoint, relationships between components lend themselves to a sound physical understanding with a known solar geometry – the solar geometry integration process a longer time scales ‘dilutes’ physical interpretation.

2) From the physical standpoint, this is the type of information that is in greatest demand.

We grouped the models into four categories:

a) Transposition models that relate one or more solar radiation components to another.

b) Meteorological models that convert standard meteorological data into solar radiation parameters.

c) Satellite models that convert satellite images into solar radiation parameters.

d) Stochastic models that can generate streams of synthetic solar radiation data from a limited number of parameters.

5.1 Input - Output and Parameterisation

When introducing or validating model, the first task is to specify its input and output parameters. Input to a model consists of:

1) Deterministic parameters, which including solar geometry or the means to derive it, i.e. time of day, time of year and site geometric coordinates, i.e. site elevation.

2) Information parameters, which are available from monitoring the programmes, such as cloud cover, global and direct irradiance.

These inputs can then be processed inside the model, for instance to parameterize insolation conditions, prior to producing the model output. As accurate as the model can be, its operational accuracy depends on the precision of its input. There should be encouragement to perform sensitivity analysis to link input uncertainty to output accuracy. The validation and intercomparison of the models require precise input information, as well as precise output verification measurements. IEA Task IX-B model evaluation, Hay and McKay, 1988, have shown the evaluation results are considerably influenced by the quality and precision of input-output data.

Information parameters should allow a model to adjust to these major influences, Since cloud cover and sunshine duration have been measured by numerous stations, they have been the first quantities that used to parameterize insolation conditions in global irradiation models. The relationship between cloudiness and sunshine duration is far from linear relationship. Sunshine duration has the advantage over he cloud clover of being based on the measurement of the actual direct solar irradiance. However, because of its only reported as exceeding or not the threshold, sunshine duration can not be used to characterize the turbidity of the site

5.2 Component to component Model

5.2.1 **Sunshine duration models (V.Estrada-Cajigal)** fall under the empirical models consisted of simple regression approximating the real radiative transfer. Angstrom (1924) model used the mathematical expression for relationship with an empirical coefficient. Angstrom equation has been extensively used and applied to large number of location.

$$H / H_0 = a + bn / N$$

Where H is monthly averaged daily global radiation

H_0 is daily extraterrestrial irradiation
n is monthly average sunshine hours
N is day time length

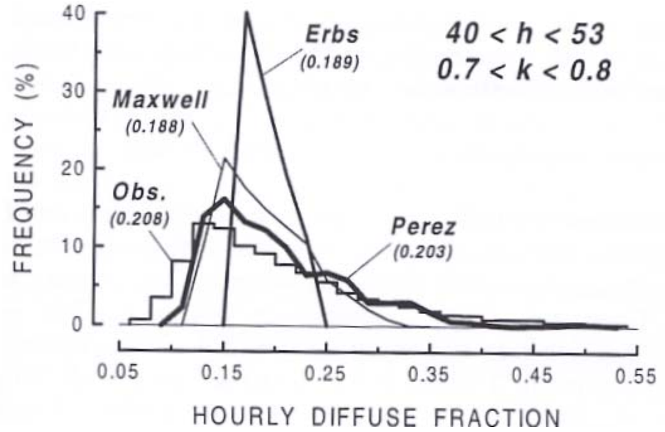


Figure 6: Distribution of 4931 hourly diffuse fractions observed under solar radiation, h and clearness index, k, at Bergen, Norway

5.3 Spectral model

Spectral irradiance data and models are needed in a variety of application spread over different disciplines. Spectral irradiance data are most useful to analyse the energetic response of photovoltaic system, high performance glazing, selective coatings and daylighting applications. Nann and Bakenfelder (1993) describe twelve possible uses of Spectral radiation models for solar energy system and buildings applications.

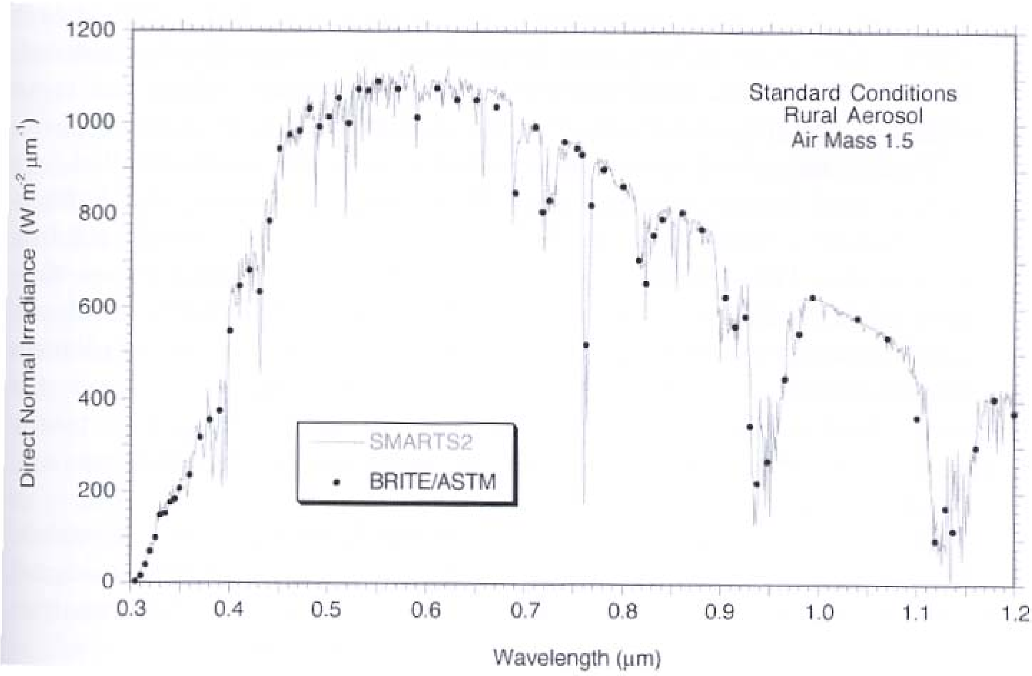


Figure 7: Beam spectral irradiance predicted by SMARTS2 and BRITE for a standard air mass 1.5 atmosphere with rural aerosol, as defined in ASTM (1987a) or ISO

5.4 Meteorological Model

Meteorological Model attempts to estimate solar radiation from information provided by weather stations. This approach is attractive because weather measurement sites vastly outnumber solar radiation measurement sites. Frequency distribution: most system responds non-linearly to weather parameters, average, frequency and normal distribution.

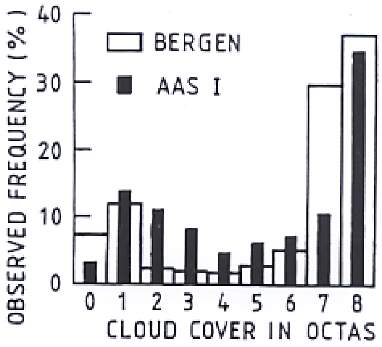


Figure 8: (1965-79) and AAS (1968-79)

at 60º N, based on observation at 7, 13, 19 CET

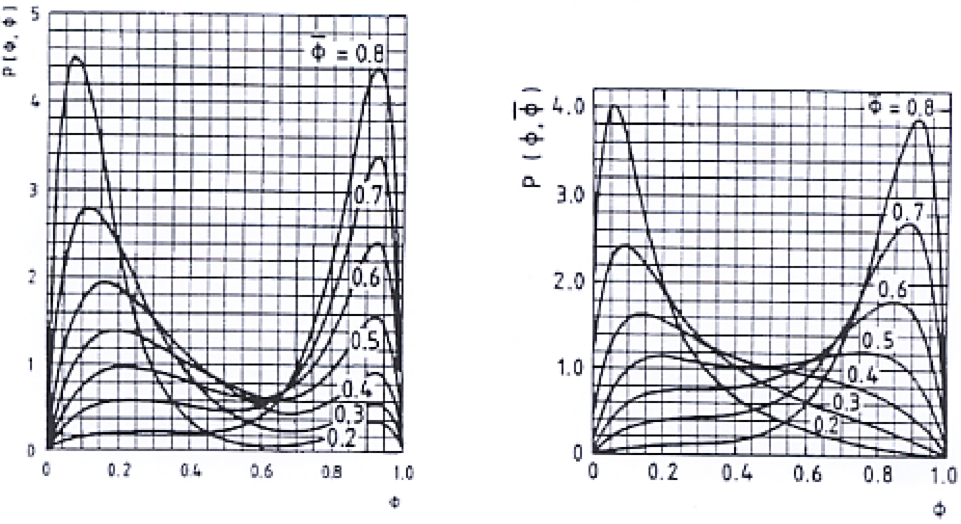


Figure 9: Pobability density function P of hourly (left) and daily (right) normalized clearness index

5.5 Auto correlation structure model

Most buildings and solar systems possess a dependent memory that response to hourly irradiances, spectrum of time scale occur, which yields a positive autocorrelation for most of the weather parameters. Spectrum of time scale occur, which yields a positive autocorrelation for most of the weather parameters.

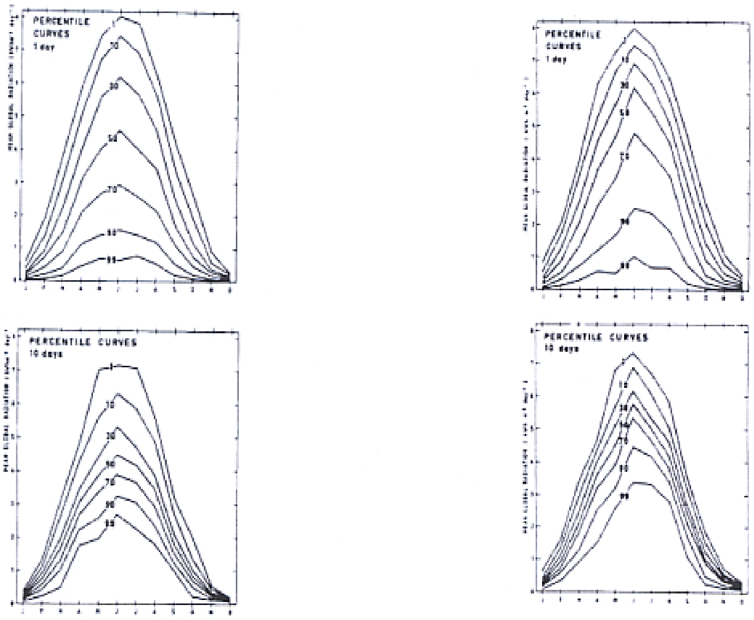


Figure 10: Percentile curves of daily and ten days global irradiation at Bergen (left) and AAS (right)

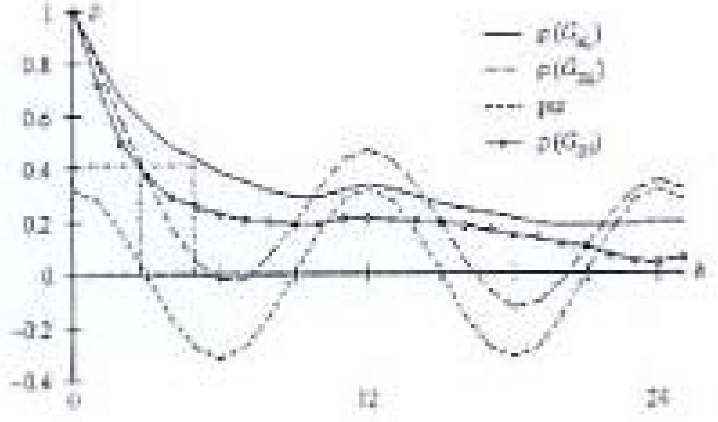


Figure 11: Autocorrelation hourly function (average, variance, quadratic deviation) of the periodical alternating beam and diffuse irradiance

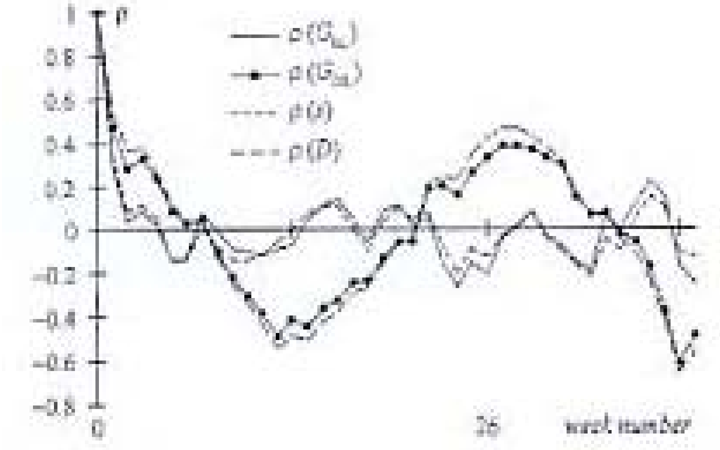


Figure 12: Autocorrelation weekly function (average, variance, quadratic deviation) of the periodical alternating beam and diffuse irradiance (summer 1999-2002)

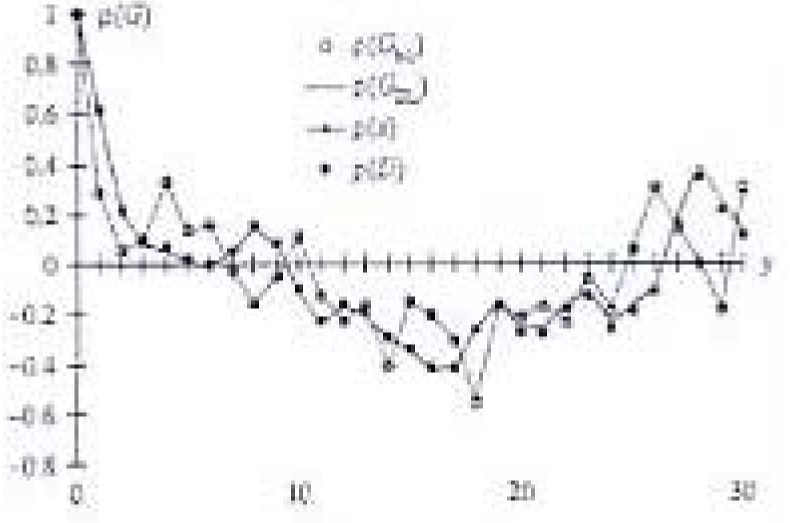


Figure 13: Autocorrelation monthly function (average, variance, quadratic deviation) of the periodical alternating beam and diffuse irradiance (1955-2001)

5.6 Cross correlation structure model

Solar systems response to any single weather parameters depends on the simultaneously and past values of other parameters.

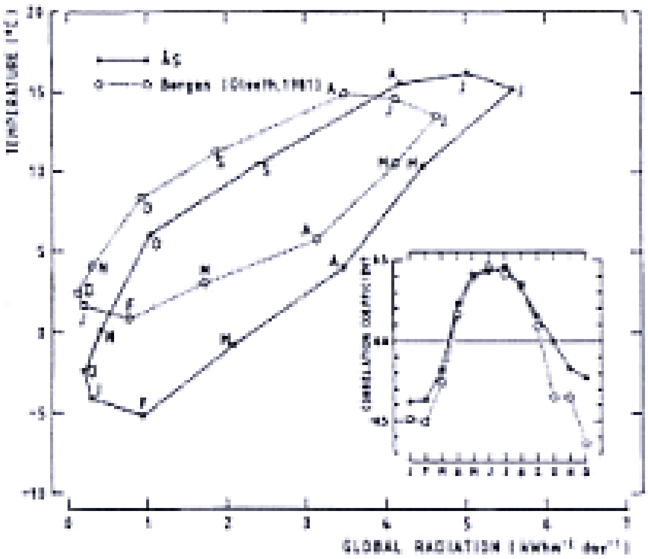


Figure 14: Average monthly of air temperature vs global radiation at Bergen and AAS

5.7 Satellite Data

An important function of meteorological satellites is detecting cloud fields and monitoring their evolution in time over extended regions of the world

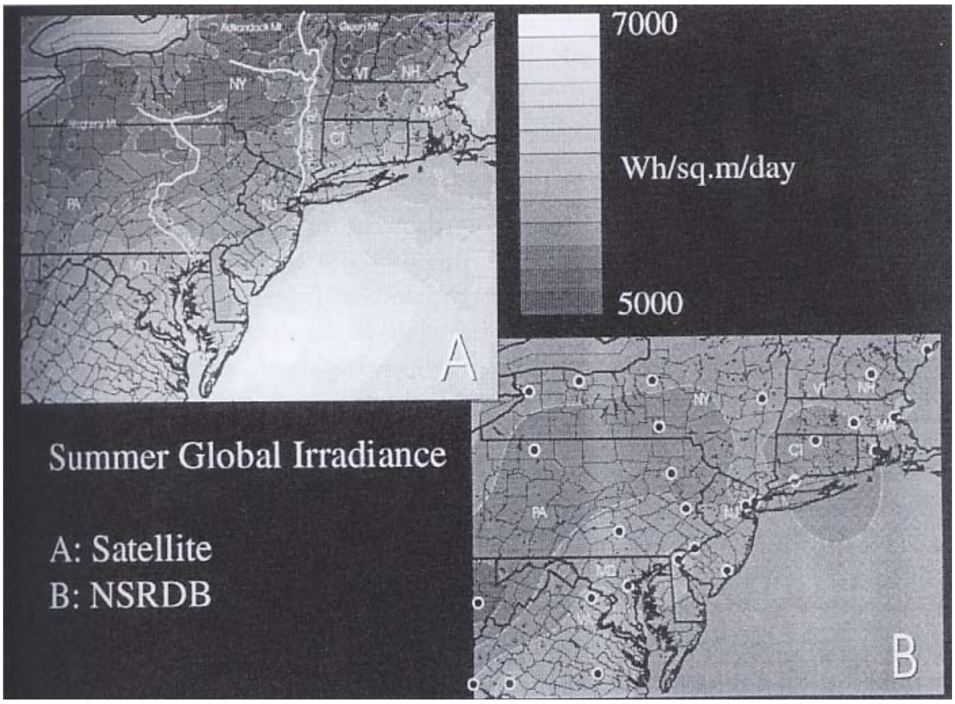


Figure 15: Compare in the north-eastern U.S.A. from satellite image (A) and U.S. National Solar Radiation Database (B). NSRDB is based on combination of ground irradiance measurements and irradiances derived from meteorological observations

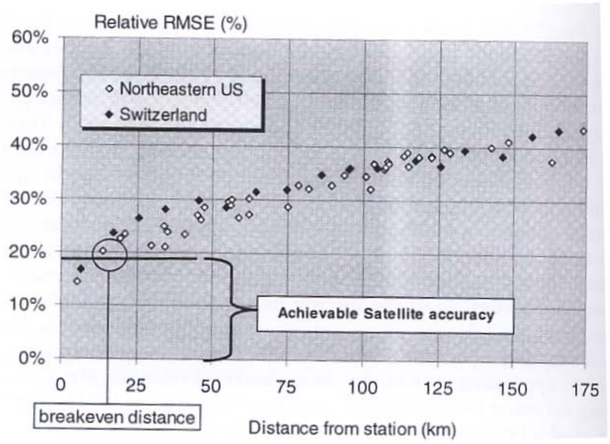


Figure 16: Hourly globala irradiance extrapolation RMSE error as a function of distance from station compared to observed satellite prediction error

5.8 Stochastic and Multi correlation

Stochastic of daily and hourly time series of global irradiation and sunshine fraction were explored in the late 1970s. These models become the useful tool for improving the use that can be made from observed data. Substitute missing and spurious values, extending the length of an observation series, and data compression. The followings are the suggested models:

5.8.1 Auto Regressive Moving Average (ARMA) Model

$$X\,(t + \Delta t) = \phi_1 x(t) + r(t)$$

Where *r(t)* is a random uncorrelated variables

x and r have zero mean and normally distributed

Variance of r is $\sigma^2(r) = 1- \phi_1^2$

5.8.2 Markov Transition Matrix Model (MTM).Equation:

$$\sum_{i=1}^{m} Pki \geq u(t)$$

Where probability of transition P_{ij} from state i to j are computed for all case i, j = 1,…., N, yielding the MTM matrix, starting from x(t) to a state k, the next state m is determined with the aid of a random uncorrelated variable u, uniformly distributed.

6. User Production

6.1 Time-Site specific data

Time-Site specific data is necessary when the location and timing are critical.
Include is the conventional measurement networks, satellite remote sensing, and combination.

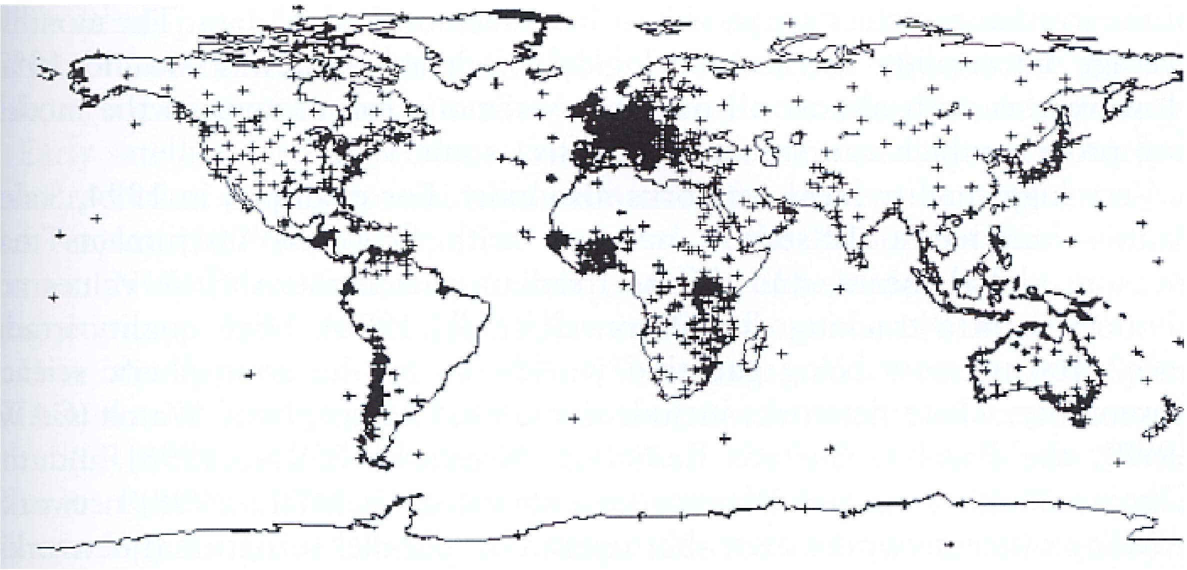


Figure 17: Location of stations reporting solar radiation data to the World Radiation Data Center

6.2 Averaged and Condensed data set

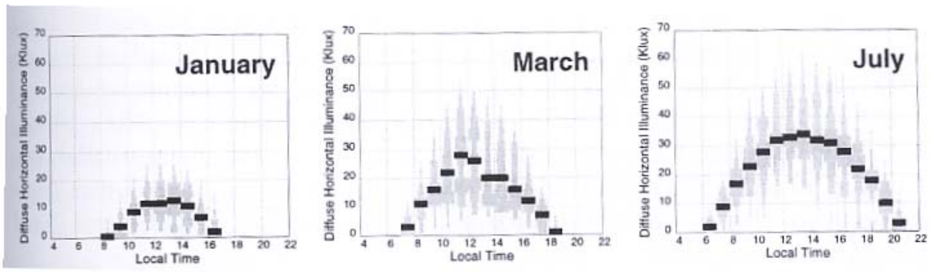


Figure 18: Frequency analysis of hourly illuminance data as a function of time of day, the median (> 50%) is represented in dark line.

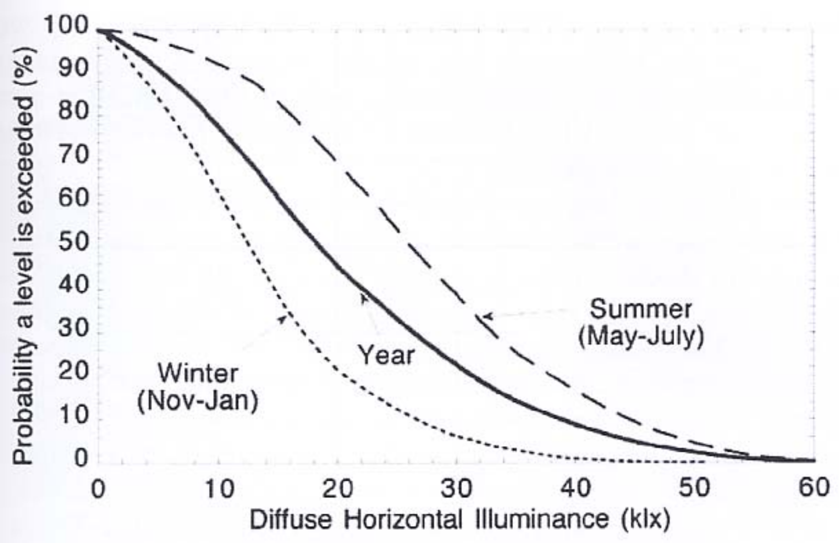


Figure 19: Cumulative probablity distribution for diffuse illuminance availability in Lyon, France

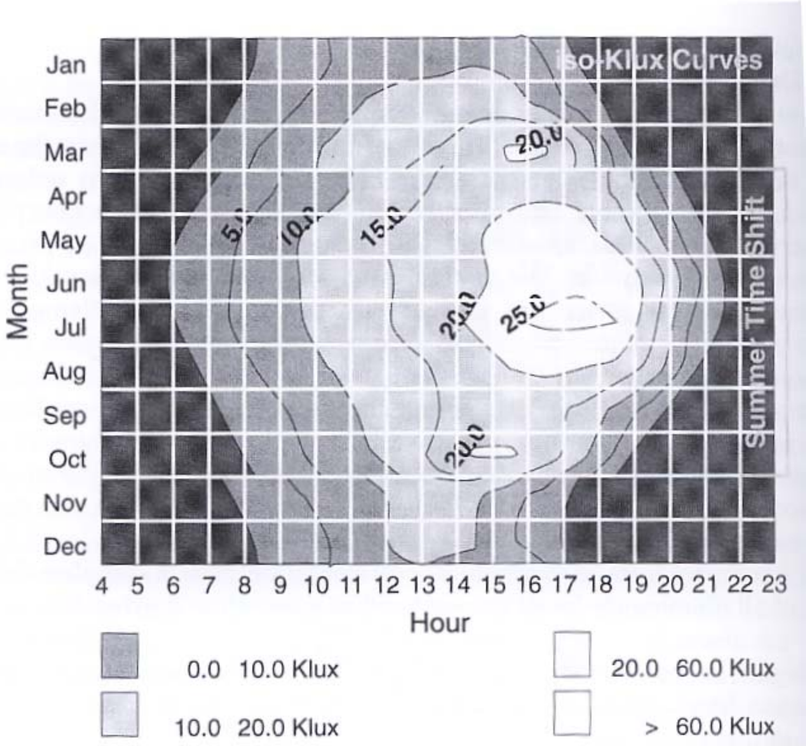


Figure 20: Month-hour diagram for diffuse illuminance available in Lyon, France, on a vertical surface facing west

6.3 Data Forecast Map Product

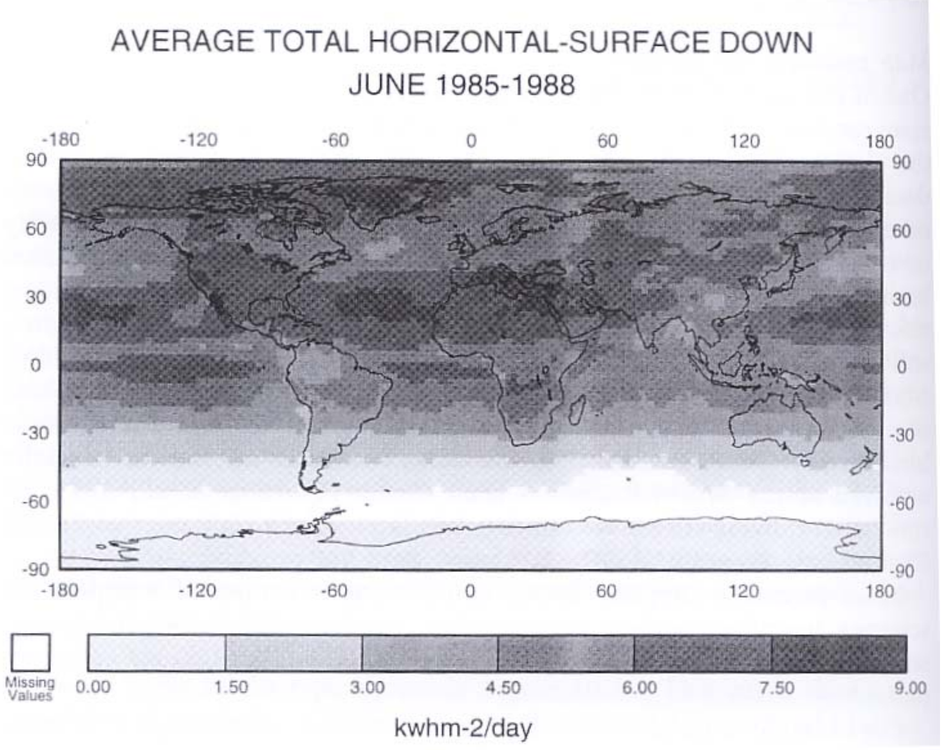


Figure 21: Example of NAS´-SRB map

In appendix 2 is a briefly describes of the Meteonorm software.

7. Conclusions

1. Definition of the measurement to end use data products

2. State of the art and significant in this field

3. Emphasis on the models and methodologies capable of transforming input information into customized data for solar energy

4. Identify the need for development of solar resource assessment

References

DNMI, 1994:
DNMI-Rapport
Report no. 11/94 Klima
The Norwegian Meteorological Institute, 1994

Duffie et.al, 1991:
Duffie, J.A. and Beckman, A.
Solar Engineering of Thermal Processes
John Wiley & Sons, Inc., 1991

Gordon et.al, 2001:
Jeffrey Gordon (editor)
Solar Energy – The State of the Art, ISES Position Papers
ISES, 2001

Goswami et.al., 2000:
D. Yogi Goswami, Frank Kreith and Jan F. Kreidler
Principles of Solar engineering
Taylor & Francis, 2000

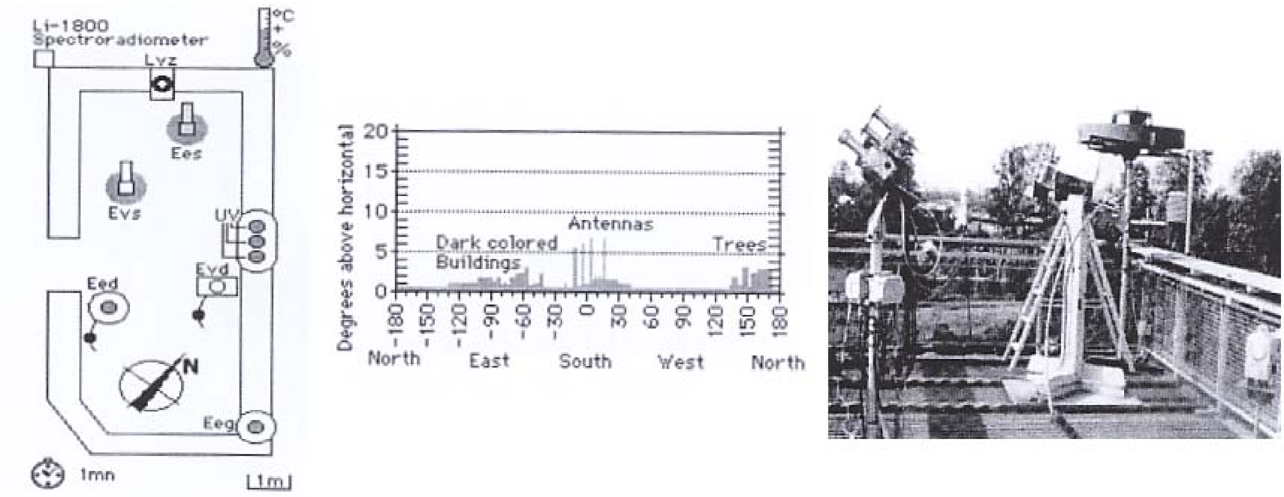
Melikov et.al, 2003:
Enn Melikov, Teolan Tomson, Ain Kallis and Vivi Russak
Steady-State Variability of Solar Irradiance
ISES 2003 paper

Milutiene et.al., 2003:
Edita Milutiene, Vida Auguliene and Arunas Marsalka
Solar Radiation Monitoring in Vilnius
ISES 2003 paper

Remund et.al, 2003:
Jan Remund and Stefan Kunz
The New Version of the Worldwide Climatological Database Meteonorm
ISES 2003 paper

Vazquez et.al, 2003:
Manuel Vazquez and Jose M. Santos
Preliminary Analysis of the University of Vigo Radiometric Data Station
ISES 2003 paper

Appendix 1



Research class measurement station (IDMP station, Norrköping, Sweden)

The instruments of the IDMP station, Norrköping, Sweden

Sampling Interval: 1 min Recording Interval: 1 min	Illuminance: Global horizontal: Licor LI-210 Direct normal: Licor LI-210 (5.5º angle) Diffuse horizontal: Licor LI-210 North vertical: Licor LI-210 East vertical: Licor LI-210 South vertical: Licor LI-210 West vertical: Licor LI-210
Irradiance: Global horizontal: Kipp & Zonen CM11 Direct normal: Eppley NIP (5.5º angle)Diffuse horizontal: Kipp & Zonen CM11 North vertical: Kipp & Zonen CM11 East vertical: Kipp & Zonen CM11 South vertical: Kipp & Zonen CM11 West vertical: Kipp & Zonen CM11	Others: Zenith luminance: LiCor LI-210 (25º angle)Temoerature: PT 100 Relative humidity: Lambrecht Wind direction: SMHI Wind velocity: SMHI Shadow disks diffuse illuminance: Radius 72.7 cm, width 7.0 cm diffuse irradiance: Radius 72.7 cm, width 7.0 cm

Appendix 2

Meteonorm is a worldwide climatologically database. It contains monthly mean data of 7,400 stations and a set of 8 parameters. The software is capable of generate data in hourly for any place in the world, based on spatial interpolation and stochastic generation of time series.

- Radiation parameters include global, diffuse, beam radiation on inclined planes. Also include temperature, dew point, precipitation or wind speed.

- The chain of algorithms.