Grant agreement no.

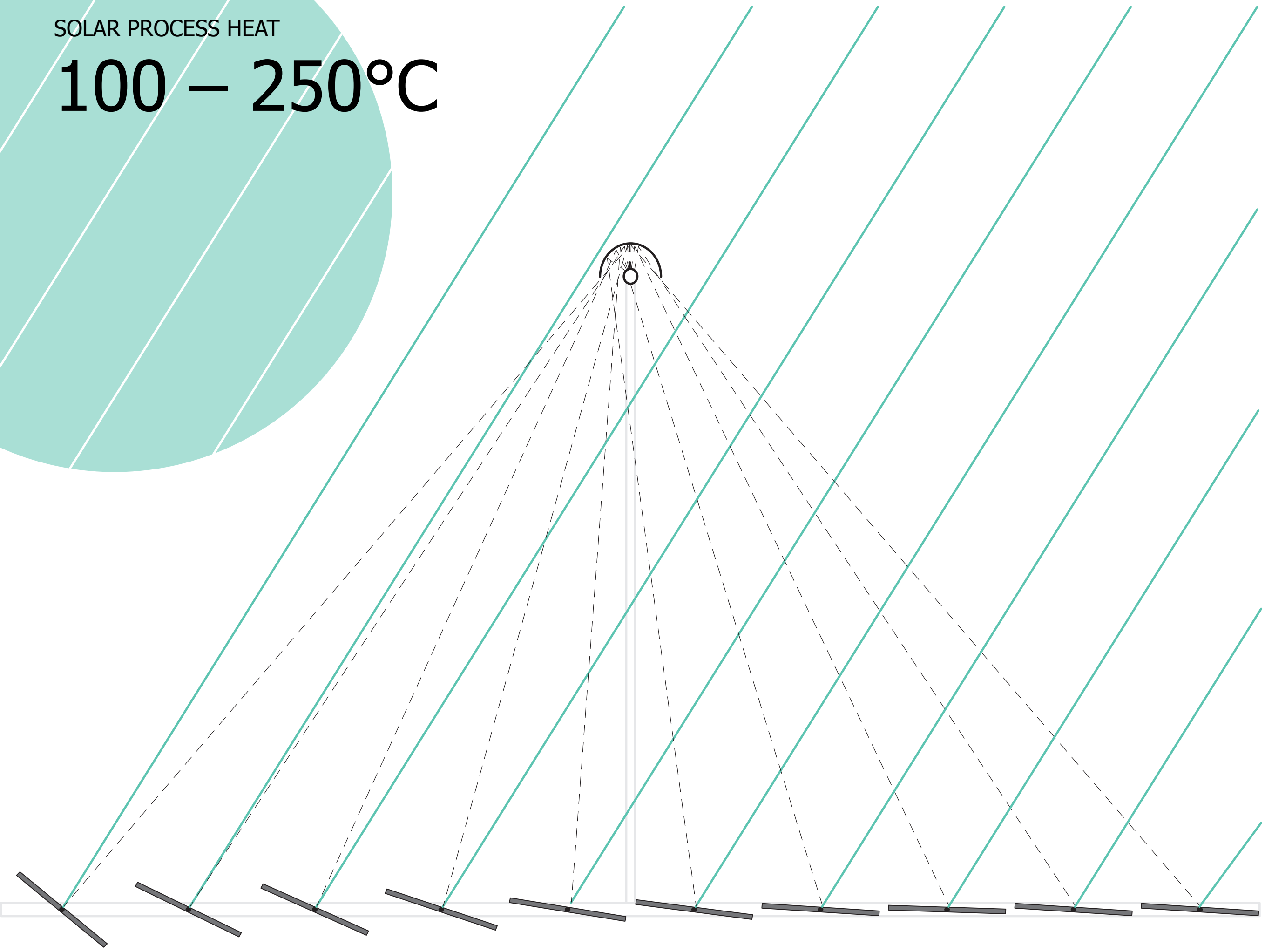
308792

FRESH NRG

SPH

SOLAR PROCESS HEAT

100 – 250°C



HANDBOOK

FOR SOLAR INDUSTRIAL APPLICATIONS

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ABBREVATIONS AND ACRONYMS

ACRONYM	DEFINITION
ΔT	Temperature Difference
BoP	Balance of Plant
CAD	Computer-Aided Design
COP	Coefficient Of Performance
CPC	Compound Parabolic Concentrator
CSP	Concentrated Solar Power
DN	Diameter Nominal
DNI	Direct Normal Irradiation
DSG	Direct Steam Generation
FP7	7th Framework Programme
GA	Grant Agreement
HTF	Heat Transfer Fluid
IAM	Incidence Angle Modifier
JER	Dr. Jakob Energy Research GmbH & Co. KG
LCA	Life Cycle Analysis
LFC	Linear Fresnel Collector
MTS	Multi Tank Systems
ORC	Organic Rankine Cycle
PCM	Phase Change Materials
PLC	Programmable Logic Controller
PT	Parabolic Through
SWOT	Strength Weakness Opportunities Threats
T	Task

1. INTRODUCTION

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The following handbook is the main outcome of Task T7.4 (Deliverable D7.4 Handbook for solar industrial applications) of the FRESH NRG project, which is an FP7 research project supported by the European Commission under GA no. 308792. The project addresses the integrated development of a low cost solar concentrated collector that can be operated at 250°C with collector efficiency above 50%.

Potential market size and environmental benefits make the use of solar heat in industrial applications a promising and vast area for future application of solar thermal systems. Many industrial thermal processes can combine their traditional heat sources with solar energy. Solar concentrating collectors can generate process heat like steam, thermal oil or hot water, both directly or indirectly depending on what is best for each industrial process. The handbook is presenting a general overview of suitable technologies for solar industrial process integration. Starting with differnet collector types and technologies as well as common working fluids and moreover several system applications as storage types. Furthermore, general system design settings assisted with a few rule of thumbs for the design process are given including some calculation examples.

Besides a scope of application and further potentials of process integration also three detailed case studies, which had been elaborated in an earlier Task T5.2 of the project, are presented. The three case studies address industrial and commercial applications including solar cooling as well as hot water and steam generation for various countries (Italy, Chile and Jordan) to show the application potential of the new FRESH NRG collector. In general, there are three major findings for the new FRESH NRG collector:

- Price list reduction of -15% between existing parabolic trough collector (PTMx) and FRESH NRG collector
- Collector aperture area reductions of 7-15% for the real collector area as well as 12-34% for the gross installation area are determined for the FRESH NRG collector against the other investigated solar concentrator collectors
- Preliminary system costing shows that for the solar cooling application lower system costs up to 31-42% or specific costs of 1,132 EUR/kW can be achieved with the FRESH NRG collector. The investigated solar process heat systems show up to 32-38% cost reduction or specific costs of 412 EUR/m^2 for process heat applications

2. TECHNOLOGIES

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In this chapter we will address the known and commercial technologies for solar thermal collectors relevant to this handbook.

2.1. CONCENTRATING AND NON-CONCENTRATING SOLAR THERMAL COLLECTORS

In this section solar thermal collectors are categorized into two categories as follows:

2.1.1. CONCENTRATING SOLAR THERMAL COLLECTORS

They are the ones that use reflective surfaces (or any other methods like refractive methods) to concentrate solar energy to the absorber of the collector. This is mainly done for the purpose of increasing the temperature of the working fluid. For the temperature ranges considered in this handbook the following concentrating solar collectors are considered:

- Parabolic Trough: Where the reflector surface-concentrating surface is a Parabolic Mirror concentrating the direct solar beam to a linear absorber and concentrating solar radiation up to 80-100 times in commercial large scale parabolic trough to obtain temperatures around 450°C. Nonetheless, for the temperature ranges considered here usually a (micro Parabolic Trough) collectors are used that concentrate Direct Solar Radiation to 20-40 times. The absorbers tubes in these collectors can be either enclosed in a glass tube or free to the atmosphere. If they are enclosed in glass tubes this can be evacuated or non-evacuated tubes. The axis of this concentrator can be East West, North South, or any orientation angle between them and a tracking mechanism moves the trough to track the solar beam accordingly.

- Linear Fresnel Collectors (LFC): Where the concentrator is basically a group of flat or slightly curved reflectors placed on ground level and controlled in a way to concentrate solar direct beam to linear absorber above them that is basically an absorber tube. Usually there is a secondary mirror above the absorber tube to redirect escaping radiation to the absorber tube. Similar to the parabolic trough there are large scale LFC and micro scale LFC based on the concentration ratio. The micro scale LFC and generate the temperatures up to $300°C$ covering the range of interest in this hand book. Also similar to the parabolic trough the absorbers tube(s) can be covered by glass or freely open to the atmosphere. The absorber tube can also be enclosed in evacuated or non-evacuated tubes. Furthermore, similar to the Parabolic trough The axis of this concentrator can be East West, North South, or any orientation angle between them and a tracking mechanism moves the trough to track the solar beam accordingly.

- Compound Parabolic Concentrator (CPC): In this concentrator a combination of two (or more) parabolic trough sections are arranged together to reduce or even eliminate the need to track the sun. Available commercial systems with evacuated tubes are available with concentrating rations between 4-6 that can generate heat up to a +-130$°C$ although in theory large concentrator ratios are theoretically available but there are no large commercialized products for large scale CPC.

- Other types of concentrating thermal collectors are point concentrators which include parabolic dish, heliostat-tower systems, Fresnel Dish and Fresnel Point Concentrators. But these concentrator are used for much higher concertation ratios

- (around 1000x sun) and to generate temperatures beyond the scope of this handbook.

2.1.2. Non-Concentrating Solar Thermal Collectors

- They are the ones that absorb the solar radiation directly and do not use any concentrators. As such, these collectors are receptive to the global solar radiation and not only to the direct radiation have beam and can heated the working fluid even in a partially cloudy day or a day with thin white clouds but not to the full level of the sunny days. The known types of non-Concentrating solar Thermal Collectors include the following:

- Flat Plate Collectors: These are the most used solar thermal collectors and also are very easy to manufacture. The working principle will be discussed in section 2.1.4 in this hand book. They can heat working fluid up to 80$°C$ in a stagnation mode in warm day with good sunshine. However, there average operational temperature range is between 40$°C$ - 60$°C$ in an average summer day. Their efficiency depends on the selective coating of their absorber area and the quality of glass, pipes, insulation and other components. In all cases their efficiency drops largely at low ambient temperature and high wind.

- Evacuated tubes Collectors: These have growing market share collectors due to their favorite characteristics. Their stagnation temperature at a sunny day can reach up to 200$°C$. However, they can reach operation temperate up to 90$°C$ on a continuous form in sunny days. As the absorber surface is enclosed in an evacuated tube its efficiency is not much affected by the ambient temperature nor by the wind. In general its operation efficiency varies between 50%-80%. There are various types of heat transfer techniques form the absorber surface to the working fluid mainly the heat pipe technique, the thermosiphon technique, and the conductive U tube technique.

- Low Temperature PVC collector: This is mainly used for swimming pool heating and it is basically a collection of black UV treated PVC tubes that heat the working fluid (mostly water) by exposing it to the sun shine.

In this handbook we will consider in details the Parabolic Trough and the Linear Fresnel Collector from the concentrating thermal collector category and the flat plate from the non-concentrating category.

2.2. Working Fluids

For engineering systems the heat collected from the absorber of the Solar Thermal Collector to the application where it is used by a fluid hence called Heat Transfer Fluid (HTF) or the Working Fluid. Such fluid can in general any fluid that is suitable for the temperature range and characteristics needed. In this section we will consider the most common types for temperature ranges considered here which are:

- Water
- Water-Glycol Mixture (Solution)
- Thermal Oil
- Air

Their Temperature levels, heat capacities and costing will be further discussed.

2.2.1. Temperature Levels

Each Working has a temperature level that is suitable to it. We will discuss each of them accordingly. A summary of the recommendations resulted from the discussion here can be found in Table 1 in addition to other properties.

- Water: In fact water can be used as a working fluid in tow of its phases; the liquid phase as regular liquid water, and in the gaseous phase (steam). In its liquid state pure water boils at 100$°C$ if it is open to the atmosphere at Sea level. It also freezes at $0°C$ in these conditions. However, if compressed water will boil at higher temperatures allowing to operate as a working fluid for temperatures higher than 100. For example if water is compressed up to 10 bars then it can be used as a liquid working fluid for 150$°C$-170$°C$. To reach 250$°C$ water should be pressurized up to more than 40 bars before it boils. Since solar systems for industrial applications form large scale collectors area it may not be practical to use liquid water to be pressurized at 40 bars going into hundreds or thousands of meters of pipes. As such, either another working fluid is used or water is allowed to evaporate to form a steam inside the absorber of the Solar Collector (mostly concentrated). This is called Direct Steam Generation (DSG) and is suitable to be used in a Fresnel System (LFC) since its absorber is basically not moving in space. It should be mentioned that water (or steam) may not be only the working fluid but in some application they may be the application also. By that we mean water (or steam) is not used only as an HTF but also we use the heated water in the process directly and new water is fed in. In this case scaling has to be considered. Liquid Water is used mainly in the non-concentrating flat plate collector or in the concentrating collectors if temperatures below 100$°C$ (or 120$°C$ if pressurized) are needed. Steam can be used with temperatures up to 450 if a DSG systems.

- Water-Glycol Mixture (Solution): To increase the boiling point of liquid water (and to decrease its freezing point) its mixed with a Glycol to form a solution often called anti-freeze. This allows water-Glycol liquid to boil at temperatures above $100°C$. For example a 50% solution of Ethylene Glycol and Water will boil at $107°C$ at Sea Level while a 60% Ethylene Glycol and Water will boil at $111°C$ at Sea Level. A 90% Concentrated solution will boil at $142°C$. Glycol by itself boils at $197°C$ at 1 bar. However, using Glycol alone or very high concentration of it is not feasible for HTF due to economical and some other considerations as will be mentioned later.

- Thermal Oil: This is one of the most used HTF when high temperate ranges are needed. Thermal oil is a byproduct from all oil refineries is available there at relatively low prices. However, it is recommended to use thermal oil for Solar Thermal Systems to guarantee long life, low viscosity and environmental effects. Thermal oil for medium range temperature applications has a maximum operation temperature around $315°C$ at atmospheric pressure while special types can stand up to $450°C$ for high temperature applications. Although thermal oil can be used in an open cycle it is recommended to use it in a closed pressurized cycle to reduce its oxidization at high temperature when in contact with air. Thermal oil is seldom to be used as an application by itself as it is in the case of water so mostly steam is generated or water is heated by a heat exchanger when thermal oil is used as a Working Fluid.

- Air: Air is basically the cheapest and most available Working Fluid in nature. However, it is not commonly used due to its low heat capacity and its gaseous state which makes it harder to handle. However, air is used as a working fluid for space heating using a flat plate collector in many cases. Furthermore, it used as a working fluid in the Stirling Engine-Parabolic Dish system and in current researches it is used in Solar Powered Gas Turbines. The temperature range where air can be used as a Working Fluid for Solar Thermal System is between $30°C$ - $1500°C$ but other factors has to be considered as mentioned before.

2.2.2. Heat Capacities of Working Fluids

The Heat Capacities of each HTF plays an important role in its selection and its mode of usage. The heat capacity is characterized by the specific heat factor usually refer to as (C_P) and is measured by $kJ/(kg * K)$ in the SI units or by $(BTU/lb\,°F)$ in the British Units, in this handbook we will use only the SI units. This factor is important for the sensible heat gained by the Working Fluid or taken from it. In general the total heat given to HTF or taken from it is given by

$$Q = \dot{m} * C_P * (T_{OUT} - T_{IN}) \tag{1}$$

where:

Q : is the heat flux given to the HTF or taken from it in kJ or kW depending on the units of $\dot{m}$.

$\dot{m}$: is the mass or mass flow rate of the HTF in kg or kg/s accordingly.

C_P : is the specific Heat as defined above.

T_{out}, T_{in}: the temperature of the HTF inlet or outlet form the system in $°C$ or $°K$.

- As it can be seen form the equation above the more the C_P is the lower the mass flow rate of the HTF for the same temperature difference and heat flux. In many application it is favorable to reduce this flow rate to reduce the pressure drop and equipment size hence higher heat capacity is preferred. In other applications it is desired to obtain the highest increase in temperature for the same heat flux hence a lower heat capacity is preferred.

- In general C_P does not change drastically with temperature for liquids. However, temperature and pressure has considerable effect on C_P of steam and air while some minor effects on liquids as will be discussed. Finally, as mentioned earlier the equation above is for the sensible heat where no change of phase occurs. For the change of phase flow or two-phase flows the latent heat should be considered and thermodynamic tables need to be consulted to find the heat gained or given to or from the HTF. The heat capacities of the thermal fluids considered in this handbook are discussed herewith:

- Water: Pure liquid water has one of the highest heat capacities among liquids and is the highest among the commonly known ones. Its commonly used value is 4.18 $kJ/(kg * K)$ but it varies slightly with temperature and even pressure. As such, for very accurate calculations one needs to consult thermodynamic tables if the temperature and pressure variation is very high. Water is the most favorable fluid as heat capacity is concerned if the temperate ranges are suitable.

- Steam: Steam is the gaseous state of water. Unlike water its heat capacity is highly sensitive to the operation temperature and pressure. For the boiling stage the latent heat is to be considered while the sensible heat once it is in the full gaseous phase the enthalpy is to be used from the thermodynamic tables.

- Water-Glycol Solution: Since water is sometimes mixed with Glycol to allow for higher temperature range as discussed earlier, consequently the heat capacity also changes. In fact it is reduced form 4.18 $kJ/(kg * K)$ to 3.600 $kJ/(kg * K)$ at 50% concentration or to 3.475 $kJ/(kg * K)$ at 60% concentration. Mixing water with Glycol requires the increase of flow to compensate for the drop in C_P if the same heat flux is desired. Usually the flow is increased by 14% on average but a calculation of flow is needed for accurate results.

- Thermal Oil: As mentioned thermal oil has various commercial brands and specifications depending on the additives and the type of Oil. The values presented in this handbook are for a common commercial brand that should form the norm for most types to give an indication for the reader. However, for a more accurate calculation one should consult the vendor catalog of the specific heat capacity for that specific thermal oil. Table-1- list some values for the C_P of thermal oil. It varies between 2.3 to 3 $kJ/(kg * K)$ depending on the operational temperature.

- Air: Air is always in the gaseous state for the temperature ranges considered here. At atmospheric pressure and temperature the C_P of air is usually around 1.005 $kJ/(kg * K)$. However, due to the sensitivity of the heat capacity of air to operational pressure and temperature one needs to consult with the thermodynamic tables or ideal gas relations to calculate the C_P for the specific operation conditions.

To compare heat capacities and temperature levels for the discussed working fluids Table 1 demonstrates such comparison to summarize the discussion above.

TABLE 1: SUMMARY AND RECOMMENDATIONS FOR TEMPERATURE AND HEAT CAPACITIES OF VARIOUS WORKING FLUIDS

Thermal Fluid	Recommended Temperature Range $[°C]$	Heat Capacity $[\frac{kJ}{kg * K}]$
Water (liquid)	5-100 (If compressed up to 220)	4.18 On average
Steam	100-450	Depends on Temperature and Pressure Consult with Thermodynamic Tables
Water-Glycol	-50 – 111 (if compressed can go higher)	3.475 at 60% concentration 3.600 at 40% concentration
Thermal Oil	50-315 (if compressed up to 450)	2.3 (at 100$°C$) to 3.0 (at 315$°C$)
Air	30-1500 (If application is suitable)	1.005 at ambient conditions (Depends on Temperature and Pressure Consult with Thermodynamic Tables)

2.3. Flat Plate Collectors

Characteristics

The Flat plate collector is one of the most used thermal collectors worldwide. Figure 1 below shows the basic concepts of these collectors. Its basic principle of operation is based on the principle of the (Green House Effect) where glass allows short wave solar beam to pass through it while it prevents the long wave radiations from the absorber surface from passing back. As such, this would result in the entrapment of the solar radiation and heating up the surface and consequently heating up the Working Fluid passing in the pipes below it. The efficiency of the collector depends no some factors including:

1- The quality and the efficiency of the selective coating of the absorber surface.
2- The thickness of the thermal insulation materials.
3- The conductivity of heat from the absorber thickness to the working fluid.
4- The quality of the top glass (low iron glass with anti-reflecting coating is better from regular glass.
5- Ambient temperature and wind conditions.

Other characteristics of the flat plate collectors are discussed in section 2.1 and the Strengths, Weaknesses, Opportunities and Threats are discussed in the SWOT analysis below.

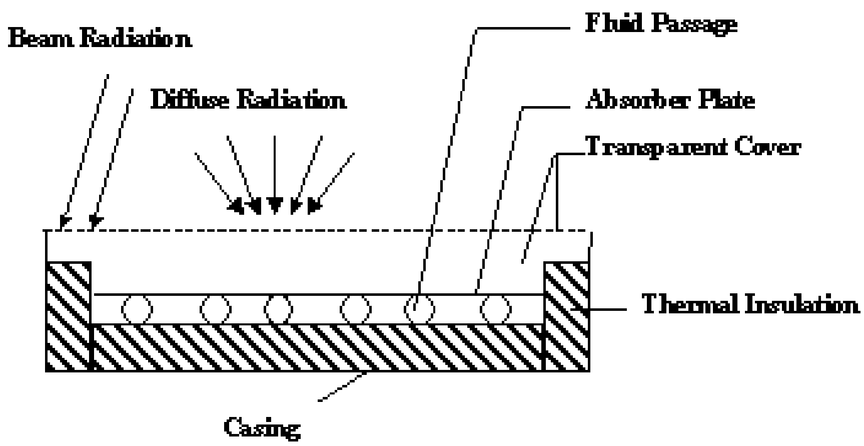
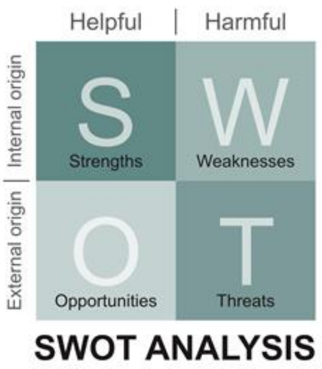


FIGURE 1: PRINCIPLE OF OPERATION OF FLAT PLATE SOLAR COLLECTORS (MUTAH)

TABLE 2: SWOT ANALYSIS OF FLAT PLATE COLLECTOR



	Strengths	**Weaknesses**
	• Ease of manufacturing. • Low cost • Ease of installation • Ease of operation • No moving parts. • Matured and reliable technology	• Low efficiency in winter • Low temperature levels • Degradation of performance with time. • Limitted application
Opportunities • Industries that need low temperature process heat. • Resedential and commercial applications. • Emerging of technologies that uses low temperature such as solar cooling	**Opportunity-Strength Strategies** • Establishment of local system integrators to provide economical solution for industries. • Obtaining long term finances due to technology maturity and reliability.	**Opportunity-Weakness Strategies** • Taking advantage of high demand in industry to improve performance. • taking advantages of finance to increase quality and reduce degradation.
Threats • Emmerging of new technologies that has better performance like evacuated tubes. • Rapid drop in prices of competing technologies. • Bad installation practices that damage reputation.	**Threats-Strength Strategies** • Use ease of manufacturing to produce low cost local manufacturing systems. • Use its maturity and reliability to overcome competition.	**Threats-Weakness Strategies** • Conducting more R&D to improve efficiency and performance. • Searching for more applications.

2.4. Parabolic Trough

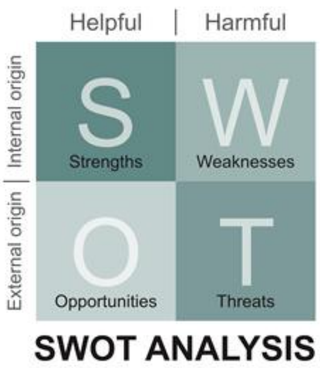
Characteristics

The Parabolic Trough is a Linear Concentrator as described in section 2.1 before that can heat working fluids up to $450°C$ for large spans (around 6 meters) and up to $300°C$ for medium spans (around 1.5 meters). Figure 2 shows a matrix of Parabolic Trough Solar Collectors installed in Mutah University for the tri-generation system. This system composed of 40 collectors each with 1.5 m span and 2 m length connected in a matrix of four parallel rows each composed of 5 collectors in series. The temperature of the thermal oil in this receiver reached above $260°C$.

FIGURE 2: MATRIX OF PARABOLIC TROUGH COLLECTORS INSTALLED AT MUTAH UNIVERSITY, JORDAN (SOURCE: MUTAH)

The parabolic trough solar collectors are the oldest used concentrated collectors where the first known was operated in Egypt in 1913 and it is the most bankable and mature technology of the CSP. More characteristics of the Parabolic Trough Collectors are described in section 2.1 and the **S**trengths, **W**eaknesses, **O**pportunities and **T**hreats are discussed in the SWOT analysis below.

TABLE 3: SWOT ANALYSIS OF PARABOLIC TROUGH (PT) SOLAR COLLECTOR

	**Strengths** ● High optical and thermal efficiency. ● Acheives required temperature easily ● Ease of operation ● Relatively low cost. ● Matured and bankable technology. ● Can be used in a wide range of applications. ● Can be fitted on roof tops for some situations.	**Weaknesses** ● Reflectors cannot be manufactured easily worldwide. ● Very high wind load as a result of thellarge cross sectional area. ● Requires very accurate tracking of the sun. ● The cost is not low enough. ● Shipment of P.T. reflector needs special care. ● Need to be cleaned thoroughly and continuously.
Opportunities ● Prices of oil is increasing and subsidies of oil products are being lifted in an increasing rate around the globe. ● Many industries use such levels of temperatures that P.T. can provide. ● Many applications for such system in commercial buildings is available like solar cooling, heating, ..etc.	**Opportunity-Strength Strategies** ● Increasing production rates to match the high demand and lower the cost. ● Obtaining long term finances due to technology maturity and reliability. ● Installing successful pilots to demonstrate other new applications. ● Unlike the Fresnel there is no end-effect losses.	**Opportunity-Weakness Strategies** ● Taking advantage of high demand in industry to produce models dealing with the weaknesses at low cost. ● Taking advantages of finance to increase quality and provide a full solution with financial packages..
Threats ● Emmerging of new competing technologies. ● Low efficiency for electricity generation. ● Limitted system integrators, installers and operators.	**Threats-Strength Strategies** ● Use poly-generation to increase total system efficiency when electricity is needed. ● Use its maturity and reliability to overcome competition.	**Threats-Weakness Strategies** ● Conducting more R&D to overcome weaknesses. ● Train and support more system integrators and operators.

2.5. LINEAR FRESNEL CONCENTRATORS (LFC)

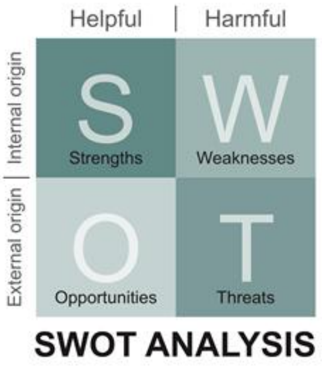
Characteristics

The Fresnel Solar Collector is emerging as a competitor to the Parabolic Trough Collector but with fewer problems. The concentrator is named after Augustin-Jean Fresnel (1788-1827) who proposed that an imaging smooth reflector can be replaced by strips of non-imaging reflectors with more practicality of manufacturing. As such, the Parabolic Trough can be replaced with strips of mirrors placed at ground level each moving with an angle to reflect the sun beam to a secondary CPC concentrator placed above the receiver as shown in Figure 3. Large scale Fresnel systems exist to heat the working fluid up to 450$°C$ but also smaller size collectors are now widely used to produce temperatures up to 300$°C$ which is in the scope of this handbook.

FIGURE 3: A PROTOTYPE OF THE FRESH NRG FRESNEL SYSTEM (SOURCE: MUTAH)

Although thermal oil is used in the Fresnel collectors a direct steam generation can be used easily due to the fact that its receiver is not moving and usually it is large enough. However, the Fresnel System is suffering from the relatively low optical efficiency and the end effect which reduce its total efficiency. However, many argue that it needs less total space than the Parabolic Trough. The **S**trengths, **W**eaknesses, **O**pportunities and **T**hreats are discussed in the SWOT analysis below.

TABLE 4: SWOT ANALYSIS OF LINEAR FRESNEL COLLECTOR (LFC)



	Strengths	**Weaknesses**
	• Low wind resistance. • Acheives required temperature easily • Ease of manufacturing of all parts worldwide. • Relatively low cost. • Ease of mobility and shipment of reflectors. • Can be used in a wide range of applications. • Can be fitted on roof tops for most situations. • Less accurate tracking of the Sun is needed.	• Lower optical efficiency. • Secondary reflector hard to be cleaned. • end-effect losses exist due to the geometry imposing geometrical and area constraints • The cost is not low enough. • Need to be cleaned thoroughly and continuously. • Emerging technology not widely used and needs more awareness. • The technology is not as mature as parabolic trough and more R&D is needed to reach total maturity stage for bankability.
Opportunities • Prices of oil is increasing and subsidies of oil products are being removed in an increasing rate around the globe. • Many industries use such levels of temperatures that LFC can provide. • Many applications for such system in commercial buildings is available like solar cooling, heating, ..etc.	**Opportunity-Strength Strategies** • Encouraging of local manufacturing lines to reduce cost. • Increasing production rates to match the high demand and lower the cost. • Obtaining long term finances due to technology maturity and reliability. • Installing successful pilots to demonstrate other new applications.	**Opportunity-Weakness Strategies** • Conducting more R&D to reach to a lower cost systems. • Taking advantage of high demand in industry to produce models dealing with the weaknesses at low cost. • Install more systems with lucrative outcome to increase its bankability and financing attraction.
Threats • Not yet appealing for finance and bankability. • Low efficiency for electricity generation. • Limitted system integrators, installers and operators.	**Threats-Strength Strategies** • Use poly-generation to increase total system efficiency when electricity is needed. • Use its maturity and reliability to overcome competition.	**Threats-Weakness Strategies** • Obtaining more international certificates for banks. • Train and support more system integrators and operators.

2.6. STORAGE

In many cases energy storage is needed as the system may be used after sunshine hours. As thermal energy is much cheaper to be stored than electrical energy this forms a considerable advantage of thermal solar energy systems over PV systems. If the application requires certainty of thermal heat production then solar thermal system either need a backup from another source of energy (like boilers) or a sufficient storage systems. Storage systems also have the advantage of using of all solar energy even if the solar collectors produce more energy than the application can utilize at a certain time. In such case, storage will insure full utility of the solar thermal system. In this section we will discuss the types of storage systems that can be used in the temperature ranges considered in this hand book. Furthermore, we will discuss the basic principle of storage capacity design. Although there are many references for thermal storage we refer the reader to Ref. [1], [2], and [3].

2.6.1. STORAGE TYPES

Various types of storage can be used to store thermal energy and to be used upon need. Some of the most common types are explained as follows:

- Hydronic thermal storage: When the working fluid is liquid one of the most used methods of thermal storage is simply to store hot fluid in hydronic tanks. Hydronic thermal storage method is based on storing energy in the sensible heat Mode. This is widely used with hot water but it should be noted that above $100°C$ (or boiling point) water needs to be compressed or mixed with Glycol for temperatures up to $110°C$. For temperatures above that it is recommended to use thermal oil or even molten salt when very high temperature is needed. The hot working fluid is stored in heavily insulated thermal tank and used upon need. Stratification of the liquid inside that tank is used to keep the temperature gradient inside it. Off the shelf thermal storage tanks commercially exist or custom made ones can be designed and constructed according to need but a special care needs to be considered for the thermos fluid movement inside such tanks. Although this method of thermal storage is usually the cheapest and most widely used a main disadvantage occurs when high temperatures are needed in very narrow spectra of temperature ranges. For example if the temperature range needed are strictly between $240°C$ and $250°C$ thermal oil is necessary, which needs to be heated from near ambient ($30°C$) to $250°C$ or above. In this case the energy needed to heat the fluid is nearly 22 times the energy required. This means that the need for large volumes of hydronic systems and carful controlling of its flow rate to make sure that the outlet thermal fluid is not below the strict lower limit. In such cases the Phase Change Materials can be used as explained below.

- Phase change material (PCM) storage system: Where the hydronic thermal storage system is based on storage is based on sensible heat energy storage the PCM is based on the latent heat thermal energy storage. It is based on the latent energy of fusion of certain materials where energy is stored during the melting process and is regained during the solidification process. For temperature ranges around 200$°C$ a work is demonstrated in [4]. The advantages of the PCT storage system is that these materials melt and solidify at constant temperature hence very suitable for narrow-strict spectra of temperature levels. As the sensible heat capacity of the PCM is chosen to be low compared to the latent heat then the wasted energy used to bring the PCM to melting point where energy is stored is much less than the stored energy. Usually it is 10% - 20% of the stored energy. Another advantage of the PCM storage system is that it occupies much less space than hydronic systems. One main disadvantage is its high cost and that it is limited to certain temperature level and is manufactured accordingly.

- Steam storage system (steam accumulators): When steam is the working fluid as discussed in section 2.2.1 in a DSG system and a need to store energy in this form arise the steam storage system can be used as discussed in [2]. This means storing pressurized steam in insulated tanks with controls. As steam has much lower density compared to liquids like water or thermal oil then the volume of such tanks needs to be large if storage for long period were needed. Such systems are used as steam accumulators to overcome intermittency of solar system for a short period of time.

- Solid mass storage: This is a rising technology of low cost effective thermal storage for the temperature ranges considered. Its basic principle is to store heat in solid masses (like concrete blocks or thermal stones) to be used later in heating the working fluid. For example a system composed of steel pipes embedded in concrete blocks insulated from the outside where hot working fluids (like thermal oil) passes through the pipes and heats the concrete block hence storing thermal energy in it. When needed, cold working fluid is passed through the same pipes regaining heat from the concrete blocks and retaining the energy. Although the concrete need to be especially treated to withstand the temperatures of the working fluid this is attainable for the temperatures considered here. However, one main disadvantage is the relatively large difference between storing energy and regaining energy of such systems as compared to PCM or hydronic storage systems. The main advantage is it potentially low cost when large storage is needed. In other application hot air is used to heat rock beds (like basalt) and when needed cold air is passed across these rock bed to regain the energy back.

- Thermo-chemical & thermo-physical energy storage: When long term seasonal energy storage is needed one solution can be the thermo-chemical or thermo-physical energy storage. This is based on storing thermal energy in the bond energy between molecules. By this an endothermic--exothermic reversible reaction is used to store thermal energy. Such reaction can be either physical reaction (like adsorption or absorption) that absorbs heat in separating the bond between molecules of materials and dissipates heat as this bond is brought back. For example lithium bromide (LiBr) – water solution absorbs energy and heat in the desorption process between water and LiBr while it dissipates heat when the absorption process between these two materials. Similar behavior occurs in the adsorption process where (for example), the methanol activated carbon pair dissipate energy in the adsorption process and store energy in the desorption process. These reactions (adsorption and absorption) are called physical reaction since no chemical reaction occurs. On the other hand, there are other reversible chemical reactions that can lead to the same results. For example ammonia (NH3) can be dissociated into nitrogen (N2) and hydrogen (H2) by absorbing heat at high temperature with the existence of certain catalyst. On the other hand, hydrogen and nitrogen can react together with the existence of certain catalysts to produce NH3 and release energy at high temperatures near the values in the initial reaction. While both nitrogen and hydrogen can be stored at room temperature for long periods such pair of reversible reaction can be used to store energy. Such technologies are under developments and a commercially available storage system based on physical or chemical reactions are investigated.

It should be mentioned that occasionally some of these methods of energy storage are used in combination with each other like PCM and hydronic thermal storage system or PCM and solid mass systems.

2.6.2. Storage Capacity Design

Thermal energy storage systems are adding cost to the solar system consequently need to be rationally designed. Over estimating the storage capacity system will reduce project feasibility and under estimating the storage capacity will not lead to satisfactory performance. In designing the storage capacity for solar systems one need to take into consideration the following factors:

- The source of heat and the temperature levels needed to be stored.
- Selection of the most appropriate thermal storage type to suite the solar system.
- The demand capacity and form of the thermal energy. For example does the application need a certain temperature continuously for a certain period or does it need a certain amount of energy regardless of the temperature.
- The existence (or non-existence) of backup system for solar system.
- The complexity of the control of energy used.
- The availability of solar system during sunshine.
- Other factors related to the especial design.

As the storage capacity design is highly dependable on the specifications of each case an example is provided here for a certain case:

EXAMPLE I

A factory installed a Fresnel system to generate steam with a temperature of 150$°C$ for a total of 200 kg per night with the needed heat of 2700 kJ/kg of water. If the temperature is strictly needed but the instantaneous flow rate of steam is not restricted and the steam is generated by supplying the heat exchanger with thermal oil at temperature no less than 190$°C$. The Fresnel system can provide as much thermal oil during the day as you need for a temperature of 230$°C$. If the specific heat coefficient of the thermal oil (C_p) is 3 $kJ/(kg * K)$ and density 830kg/m^3 design the basic hydronic storage system and the storage capacity of that system.

SOLUTION

The storage system is a hydronic insulated tank for thermal oil with a pump and three way mixing valve with a controller to flow the oil for a heat exchanger at a temperature of 190$°C$ to the heat exchanger. The size of the storage tank can be calculated as follows:

The total energy needed to generate the 2000 kg of steam:

$$2000\ kg * 2700\ kJ/kg = 540{,}000\ kJ$$

Heat given by the oil:

$$Q = \dot{m} * C_P * (T_{OUT} - T_{IN}) \qquad (1)$$

As such:

$$\dot{m} = \frac{Q}{C_P * (T_{OUT} - T_{IN})} \qquad (2)$$

The total amount of mass of thermal oil needed to generate such energy:

$$m = \frac{540{,}000\ kJ}{3\ kJ/(kg*K) * (230°C - 190°C)} = 4{,}500\ \text{kg}$$

To compensate for some losses 10% is needed. Mass of oil to be stored

$$m_{loss} = 1.1 * 4{,}500\ \text{kg} = 4{,}950\ \text{kg}$$

The volume of the tank

$$V = \frac{4{,}950\ \text{kg}}{830\ kg/m^3} = 5.96\ m^3 \approx 6\ m^3$$

Note: This is assuming that the solar system can provide the storage system with around 5 tons of hot oil at 230$°C$.

3. General System Design Settings

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Although we have mentioned an example of how to design the storage system we will here mention the basic general design parameters to be considered when designing the thermal solar system mentioned here. Since each case has a different design requirements and constrains we will mention here the basic settings and parameters to be considered for the design. Some examples will be mentioned but by now mean these examples are conclusive as they describe very specific cases. Each case shall have its own design solution based on the parameters that are described here. This section is not a substitute of a specialized designer for such solar systems and only presents an introduction about the general system design parameters and settings.

3.1. Design

In this section we are discussing the solar matrix design and size. In section 3.2 a discussion of the piping system setting will be discussed and in section 3.3 the control systems will be discussed. In all of these sections a distinction between the concentrating systems (parabolic trough and linear Fresnel) and the non-concentrating (flat plate) solar systems is made.

The parameters affecting the design of the solar matrix are basically the temperature needed for the solar system and the energy needed. The energy needed is basically of two folds; the first is the total energy needed per year or per average day or per certain days while the second is the peak power obtained from the solar system.

In general there are constrains that affects the design of the solar matrix and they are as follows:

- The Solar Radiation level: This can be the global horizontal radiation (especially needed for flat plate collectors), the DNI for concentrating solar systems and solar irradiation. As these parameters vary from day to day or from hour to hour in each days the needed values for these levels depend on the design requirement. For example a system with back up boiler and needs to minimize the fuel consumption per year the average yearly DNI or Solar Insulation is needed. On the other hand if the solar system is to be based on its maximum output the peak value of DNI or Irradiation is needed. However, for a full accurate design all of the solar radiation parameters are needed. The more accurate and detailed the more optimum design can be obtained.

- The efficiency of the solar system: Which reflects how much energy can be obtained from the specific solar systems required? This efficiency can be given as a single value like that for a flat plate or in components like it is the case for Fresnel and parabolic trough. Such components can be optical efficiency parameters, thermal efficiency parameters and total efficiency parameters. As this efficiency is also a function of other parameters the efficiency is described in form of its affecting factors. A careful study of the technical specifications and efficiency curves of the chosen solar collectors should be studied to choose the efficiency used for the required performance. For example to calculate the peak output form the solar matrix the peak efficiency should be used while to calculate the average yearly output then the average yearly efficiency should be used.

- Space limitation and orientation: In many cases there are space limitation for the solar matrix and orientation of space forcing the solar matrix to be oriented in a tilted fashion. A such this will affect the performance and the output of this solar matrix. Specialized software is available commercially for such systems and a help from the vendor will be needed in certain cases to calculate the outcome form such systems.

- Temperature: As mentioned earlier the needed output temperature from the solar matrix sets constrains on the design. This usually fixes the type of the collectors and the number of collectors in each row. As the solar matrix is composed in collectors in series (rows) and in parallel columns the number of collectors in series of each row usually sets the temperature of the thermal fluid output. Another factor affecting the temperature is the flow rate. However, for optimum efficiency the flow rate across each collector should be within a certain ranger.

The existence of back up or storage system: This will defiantly affect the design of the solar matrix as it is obvious.

EXAMPLE II

If a Fresnel system is to be used to generate the needed thermal oil of example (1) during the day. If the average DNI in the worst day of the year normalized to the orientation of the Fresnel system is 4.8 $kWh/m^2/day$ (i.e. the total Direct Irradiation normal to the plane of the mirrors of the Fresnel System). The average total incident angle modifier (IAM) based on the DNI is 60% and the thermal efficiency of the receiver is 65%. Furthermore, assume that the end effect losses for this day are 5% of the system. Calculate the total needed area of the collector's mirror to meet the demand at this day.

ANSWER

To heat up a 4950 ≈ 5000 kg of oil in that day from 190°C to 230°C the total energy needed is

$$Q = 5000\ kg * 3\ kJ/(kg * K) * (230°C\ - 190°C\ ) = 6{,}000{,}000\ kJ$$

The thermal and optical efficiency of the Fresnel collector:

$$\eta_{OPT} =\ 65\% * 60\% = 39\%$$

Taking the end effect loss for this case the total average efficiency of that Fresnel Collector in that day:

$$\eta_{OPT} =\ 39\% * (1 - 5\%) = 37.05\%$$

This means that 37.05% of the solar energy is converted into thermal energy for that collector. The total solar irradiating needed to heat up 5000 kg of oil per day is:

$$\frac{6{,}000{,}000\ kJ}{0.3705} = 16{,}194{,}332\ \frac{kJ}{day}$$

The daily solar radiation normal to the plane of the collector per square meter is:

$$4.8 \frac{kWh}{m^2 * day} * 3600\ s\ = 17{,}280\ \frac{kJ}{m^2 * day}$$

Then the total area of the collector mirrors:

$$\frac{16{,}194{,}332\ \ kJ/day}{17{,}280\ \ kJ/m^2 * day} = 937.17\ m^2$$

Assume a commercial collector is available at 371.3 m^2 net collecting surface per collector then the number of collectors needed is:

$$\frac{937.17\ m^2}{371.3\ m^2} = 2.524 \approx 3\ collectors$$

Further calculations of these collectors performance are shown later.

3.2. Piping

Piping of the solar matrix is very much dependent on the specific design. The main idea is to choose the pipe diameter and circulating pump to match the flow at the peak supply of that system. The connectivity of pipes to the collectors should be considered along with the recommended flow rate through the collector as per the vendor's recommendation. Once the solar matrix design is set and connectivity pipes are put forward the choice of the pump capacity should match the maximum flow rate for the system at the peak load. Based on these data either manual calculations of the pressure drop based on the viscosity of the thermal fluid are used or a Software CAD system can be used. To calculate the peak flow rate the following example is presented.

EXAMPLE III

To design the piping system for the Fresnel system in example (2) the maximum flow rate of thermal oil is needed. Calculate the maximum Flow rate of that system based on the following conditions at noon of July day: The peak irradiation is $0.90\ kW/m^2$. The total IAM at this point is 1.0. The end effect losses at this point are 2%. The thermal efficiency of the receiver is 65%.

ANSWER

The peak thermal load that the solar matrix can provide is:

$$Peak\ Load = Irradiation * Efficiency * no. of\ collectors * Area\ of\ Collecting\ Surface$$

$$= 0.90\ \frac{kW}{m^2} * 1 * 0.65 * (1 - 0.02) * 3\ collectors * 371.3\ m^2 = 638.6\ kW$$

To heat oil from 190°C to 230°C the following flow rate is needed to absorb this thermal energy.

$$\dot{m} = \frac{638.6\ kW}{3\ kJ/(kg * K) * (230°C\ - 190°C\ )} = 5.32\ \frac{kg}{sec} = 19{,}158\ \frac{kg}{hr}$$

The volumetric flow rate is calculated based on the density of oil to be 830 kg/m^3. The flow rate for the pump $= \frac{19{,}168\ (kJ/day)}{830\ kg/m^3} = 23\ m^3/hr$. As this might be very large flow through a single collector it might be better to connect these three collectors in parallel hence the flow rate through each collector is:

$$\frac{23\ m^3/hr}{3\ collectors} = 7.67\ \frac{m^3}{hr}\ for\ each\ collector$$

That's a more practical solution especially if each collector can obtain the needed temperature rise through it.

3.3. Control

The control of the solar system aims basically for two things: the first is for the safety and protection of the collector and the second is the control of heat flow form the solar system. The control is conducted via certain techniques based on the specific design of the solar system. It is basically conducted via the controlling of the pump circulating the working fluid through the collectors via frequency and speed modulation or via an on-off control, controlling valves (three way mixing valves of limiting on-off valves or regulating valves) to control the flow amount and directions and finally the controlling of the reflectors and concentrators to focus or de-focus the solar beam on the receiver. The controller logic is based on measurement of various parameters as discussed below:

- Temperature measurements: this can be a ΔT measurement to control the circulating pumps for accumulating thermal energy. Such system is widely used in flat plate collectors. Temperature of working fluid outlet from the collector (especially for the concentrating collector) is important to control the flow rate via the circulating pumps or valves. In certain cases ambient temperature is measured to control safety issues. Furthermore, temperature of the working fluid is also used for focusing or de-focusing the concentrators when needed.

- Flow measurements: This can be done via flow meter or limiting switch that sense the existence or non-existence of flow. Measurement of flow rate and temperature differences is important to calculate and control the total energy obtained from the system and for safety consideration also.

- Wind speed measurements: This is especially important for parabolic trough and Fresnel system where if a high wind speed is deducted the concentrators are oriented in the stow position to safeguard the collectors.

- Pressure measurements: This is especially vital when pressure build up is a risk in the system and alarms are needed. Furthermore, pressure drop in the system can be useful in many controlling applications. However, pressure measurements are usually a demand for the application of the solar system especially when steam is to be generated.

- Radiation measurements: Radiation and irradiation is an important input for the control and performance calculation of the collector. Not only the Direct Normal Irradiation (DNI) should be measured but the diffused and global radiation is of importance. Furthermore, the direction of the direct beam is as important as its magnitude. This is especially important to calculate the efficiency, system output, and direction of reflectors. The irradiation measurement coupled with temperatures can be used to determine the flow rate and the by-pass of the system. In addition for low irradiation the system might also be set off.

Many control algorithm and systems are commercially available to meet the specific needs of each system.

4. Scope of Applications (Technology)

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After discussing the various technical aspects of the thermal solar systems to provide temperatures between 100°C - 250°C we will discuss in this chapter some applications of these systems. These applications are valid mostly for the concentrating solar collectors linear Fresnel and parabolic trough. The flat plate can be used for hot water generation basically below 100°C.

4.1. Solar Process Heat

In many industries process heat is needed in the form of hot water or hot liquid fluid. These industries varies from textile industries where hot water is used to wash the fabrics or for other applications like bleaching and dying. Temperatures above 100°C can also be used for process heat especially in certain food industries. For example a sesame frying pans factory to manufacture Tahini uses hot oil with temperatures of 200°C to flow around the frying pans and dry the sesame. A solar system can be used for that to replace the existing diesel boilers to heat the thermal oil. Similarly plastic industries need hot oil up to 180°C.

Solar Steam Generation

In many industrial applications energy in the form of steam is needed. Many industries can use solar generated steam especially as these industries are usually located in places far from the city and in low cost land where a lot of space is available. Some of the most famous industries that uses steam in this ran age of temperature are listed below:

- Dairy industry: Dairy products uses steam up to 150°C for the pasteurization of milk and washing and cleaning of equipment. Such industry is widely spread worldwide and usually already exist around farms with large availability of space.

- Pharmaceutical industries: This industries need steam between 180°C-230°C in there process for many reasons.

- Poultry product industries: Steam is needed to sterilize and moisture the mixture of poultry feed. Furthermore, steam with 120°C is widely used in the slaughter houses of Poultry. These farms and facilities have to be in dry lands far away from the population where land is cheap and sunny.

- Paper industries: Steam with temperatures between 130°C and 180°C is used in paper industries.

- Food and beverage: these industries use steam intensively in the cooking process and in sterilizing process. The temperature needed is between 120℃ - 180℃.

- Water desalination: As solar desalination is becoming a viable solution to solve the fresh water scarcity in areas where brackish or sea water is available, solar distillation using steam generated form the concentrating system is becoming more and more an attractive solution. With temperatures between 120°C to 150°C high efficiency water thermal desalination techniques like MSF can be used to generate fresh water at feasible prices.

- Poly-generation: Although generating electricity at temperatures around 250°C is not efficient and hence not so commercially viable the poly-generation for utilizing heat cascading technique is proven to be feasible both technically and financially. The poly-generation system in Mutah University generated electricity, water distillation, cooling and heating. Such systems are shown to be financially feasible.

4.2. Solar Cooling

As the heat from the sun is the main reason why space cooling is needed solar cooling is a very smart solution to use the sun to power the cooling systems. An obvious method that may occur to someone mind is to use PV system to generate electricity to run traditional refrigeration cycle. However, this method suffers from the problem of electricity storage and the large area needed. Heat driven chiller are known for hundreds of years and now are being developed to be combined with solar thermal collectors. A brief description including some of the most known technologies are presented here with:

- Three stage absorption chillers: These are very efficient absorption chiller that requires steam at temperatures around 220°C or more. The coefficient of performance of these chillers are around 1.8 to 2.2 (that is the cooling capacity generated are 1.8 to 2.2 of the thermal energy input of these chillers). As such these chillers need concentrating solar collectors and systems that generate high temperature steam. The used working pair is lithium bromide and water as refrigerant. One of the main drawback of these chillers is that only few manufacturers exist worldwide hence there cost is relatively high.

- Two stage absorption chiller: This is a widely used technique for solar cooling that uses steam up to 120°C. However, its coefficient of performance is around 1.3 making it less efficient than the three stage absorption chillers but at a much lower cost. It requires less temperature hence the solar system cost can also be reduced. Usually a lithium bromide and water refrigerant pair is also used.

- Single stage absorption chiller: although this system has a low coefficient of performance around 0.7 it can use hot water as low as $70°C$ to operate that chiller when a lithium bromide and water pair is used. This would largely reduce the cost of storage and working fluid. However, such chillers suffer from the issue of crystallization and special control or backup systems need to be used to overcome such problem. It should be mentioned that there exist an aqua ammonia refrigerant pair chillers in this category. These chillers require high temperature than $70°C$ but there are no crystallization problems and they can be air cooled.

- Adsorption chillers: To overcome the problem of the crystallization at lower temperature and to accommodate other technical issues in single stage absorption chiller adsorption chillers can be used. These are chillers based on the adsorption principle and operate at low temperatures as low as $50°C$ without being affected in the fluctuation of the hot water temperatures. It is very useful for low capacities where lithium bromide absorption chillers are not viable. A well spread refrigerant pair is the silca gel and water adsorption chillers that have a coefficient of performance around 0.6. However, the cooling side of this chiller cannot exceed $35°C$ and is recommended to be $25°C$ hence a wet cooling tower is needed. To overcome such problem in areas where water is scarce and precious a two-stage adsorption chiller is patented and is now begging commercialized. It uses a pair of methanol and activated carbon and can operate at temperatures around $70°C$. It can opiates at ambient temperature up to $50°C$ without the need of a wet cooling tower. However, its coefficient of performance is around 0.33. This chiller is not widely commercial yet but it has great potential especially as it can use low cost solar collectors.

- Desiccant wheel: In this system the aim is basically to dehumidify the air by absorption (or actually adsorbing) the humidity in it and then using the solar heat to desorb the humidity. It is basically composed of a rotating wheel where filled with adsorbing (desiccant material) allowing air to flow around it. In some part of the rotation the air that is needed to be dehumidified passes around the dry adsorbing material hence the humidity from the air is adsorbed and the coming air is dry. This dry air can be cooled down by evaporative cooling or be mixed with cooled humid air to reduce its humidity ration. As the wheel turns the desiccant material now filled with humidity is exposed to solar heat by one way or another (air heated by solar collectors can be one method) hence the humidity is desorbed form the desiccant material and it becomes dry again to be rotated towards the humid air again. The cycle then continues.

5. POTENTIALS OF PROCESS HEAT INTEGRATION

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5.1. BACKGROUND

Thinking of the future and its current environmentally sensitive energy scenario, the limited conventional energy resources, and the climate change it is evident that there is a need to walk a more environment-friendly path. Hence, the global interest in alternative energy resources like solar power, wind, biomass and hydropower is growing. With these alternative forms of energy production it is possible to reduce the $C0_2$ emission, conserve some of the non-renewable natural resources and, for some countries, reduce dependency on imported energy and have an economical benefit. In order to achieve a meaningful impact, the main energy consuming players need to act, such as the industry sector.

One third of the total energy demand in European and developed countries is consumed by the industrial sector [6]. In some industrial Asian countries, such as China, the industry consumes more than half of the total energy [7]. Due to environmental restrains and the demand of the reduction of CO_2, the interest of the industry in enhancing their energy efficiency without compromising their competiveness is growing. Next to enhancing energy efficiency methods by re-using waste heat and optimizing their processes, the implementation of renewable energy sources are becoming more attractive. At least half of the energy consumed by the industry is used to provide heat for processes with temperatures below $300°C$. The vast amount and scale of energy use for industrial heating represents a unique opportunity for implementing solar process heat technologies at a medium and medium-high temperature level $(80°C - 300°C)$.

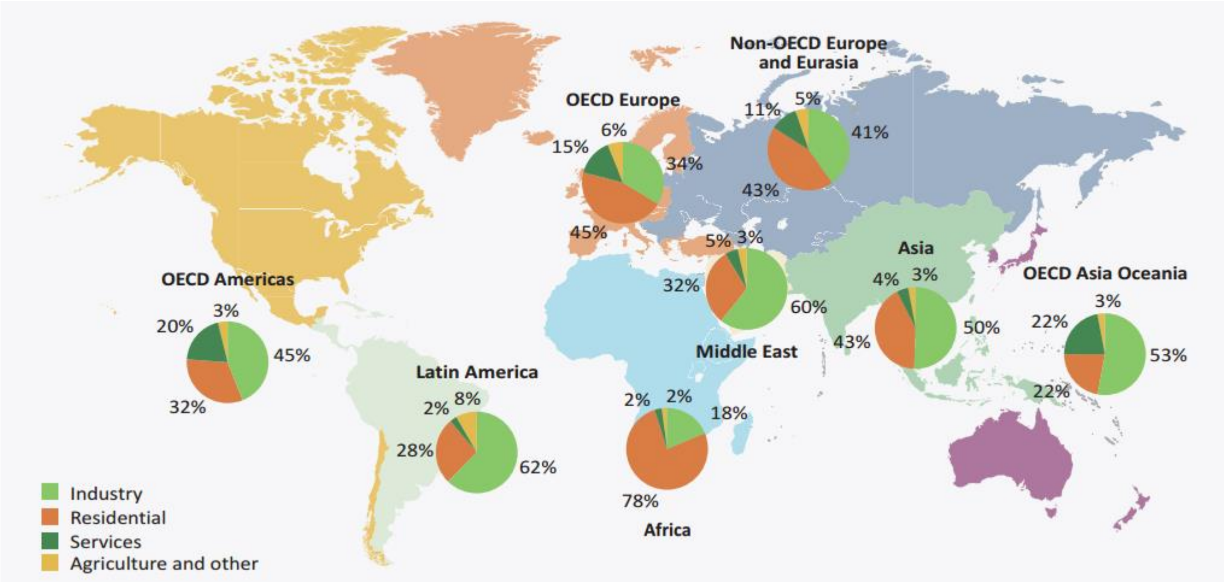

FIGURE 4: GLOBAL HEAT CONSUMPTION BY REGION IN VARIOUS SECTORS (SOURCE: DATA BASED ON 2009 TAKEN FROM © OECD/IEA 2015 [8]).

5.2. OPPORTUNITY FOR SOLAR PROCESS HEAT

Solar process heat has a huge potential for solar thermal applications and up to this point has remained largely untapped. Worldwide exist about 152 operating plants providing process heat for industrial purposes [9, 10] with a total capacity of $100\,MWTh$ (143,000 m^2).The potential for the application of solar thermal systems for process heat depends overall on the consumption of the selected processes, their demand for temperature and importantly, on the amount of solar irradiation at the given location. In Table 5 the main industrial processes that will benefit from solar heat technologies and the corresponding temperature ranges are listed. The most suitable processes that can be found in several industrial branches are pre-heating of raw materials, pasteurization and sterilization as well as washing, drying, boiler feed water and supply of hot water and steam as well as space heating in industrial buildings.

TABLE 5: OVERVIEW OF THE TEMPERATURE RANGES FOR DIFFERENT INDUSTRIAL APPLICATIONS [7, 11]

Industry	Process	Temperature range [°C]
Food and Beverage	Drying	20-180
	Washing	40-80
	Pasteurizing	60-110
	Cooking	70-170
	Sterilization	110-160
	Heat treatment	40-60
Textile	Washing	40-80
	Bleaching	60-100
	Dyeing	100-160
Chemical Industry and pharmaceutical	Cooking	95-105
	Distill	110-300
	Other chemical processes	120-180
Paper	Drying	60-100
	Boiler Feed water	40-90
	Bleaching	130-150
Automobile	Painting	160-220
	Drying	80-100
Other sectors	Preheating of water	30-100
	Heating of industrial space	30-80

According to several studies five sectors have been identified with the highest potential for solar process heat with temperature below $300°C$. These sectors are: food and beverage (including the tobacco industry), textiles and leather as well as transport equipment, machinery and mining [12]. The highest potential for solar process heat is in the food and beverage industry, which plays a critical role in the less developed countries where food security is a critical issue. Here, solar process heat can play an important role in the modernization of this industrial sector.

Solar Thermal collector technology

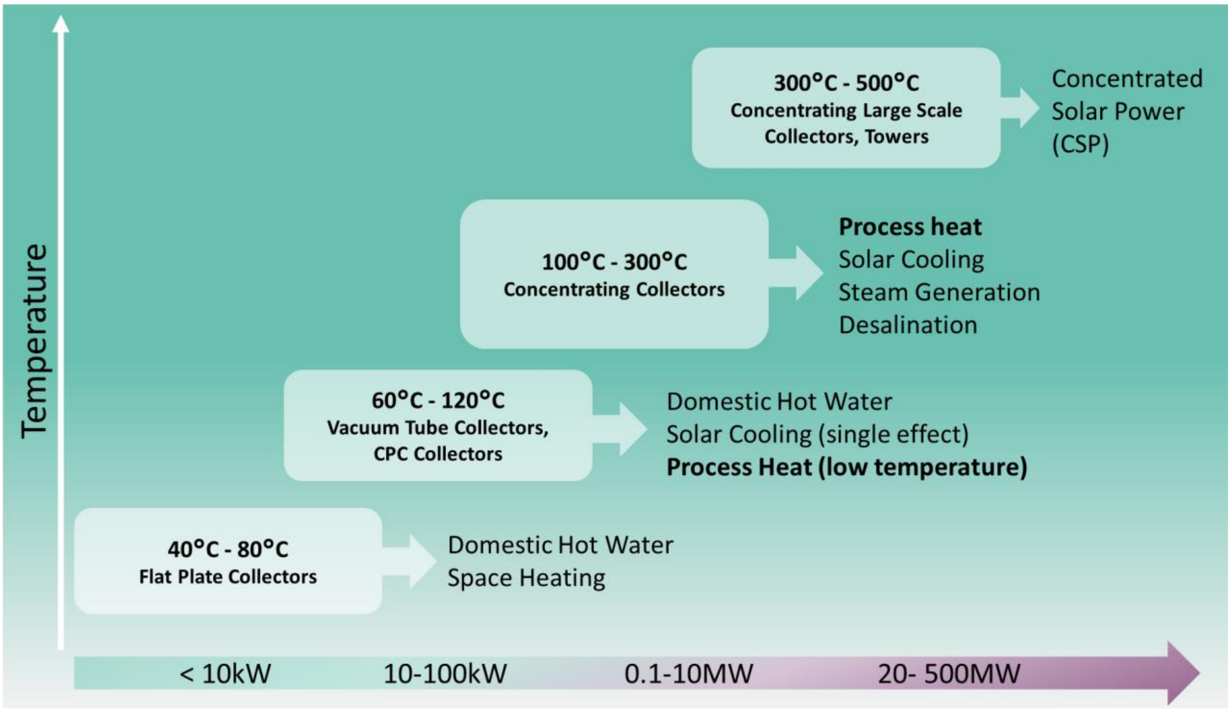

FIGURE 5: OVERVIEW OF THE DIFFERENT COLLECTOR TECHNOLOGIES AND THEIR APPLICATION FIELD AT DIFFERENT TEMPERATURES (SOURCE: HSR – SPF)

The low temperature levels for process heat (below $120°C$) are covered by flat plate and evacuated tube collectors (Figure 5). In order to achieve middle-high temperatures $(120 - 300°C)$ concentrating solar thermal collectors, such as parabolic trough and Fresnel collectors are necessary. In the past years new developments regarding those collectors have emerged, which allow the deployment of solar process heat into a wider temperature range. These new concentrating collectors are especially suited for regions with a high solar irradiation and can be used for solar cooling, steam generation and desalination as well. For more details about each technology refer to the designated previous chapters.

5.3. Perspective

Solar thermal technology has many strengths and opportunities to become an established technology for the application for process heat. As any other technology there are also some obstacles that need to be overcome.

Strengths	**Weakness**
Environmental friendly	High technical requirement
Positive Image	Shortage of knowledge and experience
Supply guarantee (combined with storage)	Cost effectiveness
Fuel saver	Lack of information for energy consultors and investors
Stable energy costs	

The strengths using solar thermal energy technology are evident and, in combination with new and future storage technologies, solar thermal energy is becoming a technology with a wide field of applications. However, for the the current generation of solar collectors the investment costs need to be reduced significantly in order to become competitive. If in the future the carbon prices rise again, the economic competiveness of solar thermal energy in industry will be very positively affected and the companies will benefit from the stable and well-predictive operational cost of solar thermal energy. However, a reduction of the first investment costs should be reduced as well, to enable industry to have a more cost-neutral transition to the solar energy source. In order to reduce the collector costs, new development projects, for example the EU Project FRESH NRG, are aiming into a more cost effective fabrication and higher efficiency of the modules.

Next to the costs, the high technical requirements for the integration of solar thermal energy into an existing system poses a challenge. In order to support solar process heat for industrial processes, many research institutes are working on the implementation, monitoring and evaluation of plants with solar thermal collectors in different industrial sectors (overview of plants see reference [10]). In addition, detailed guidelines for process heat integration for planners, energy consultants and process engineers were developed [13] and will help to overcome technical and integration challenges.

Solar thermal energy has a great potential beyond the low temperature applications for domestic hot water and space heating. The most promising application field is solar heat for industrial processes up to medium-high temperatures ($< 300°C$), resulting in a worldwide demand for this innovative technology.

6. Application studies

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Three detailed application studies specific to the FRESH NRG collector have been explored by JER. The three detailed studies address the applications with the highest industrial potential as identified by the high-level assessment done by MUTAH and Soltigua. Investigations therefore cover finally three case studies on solar cooling, solar process heat and solar steam generation in order to prove the full potential of the new FRESH NRG collector.

6.1. Investigated solar collectors

For the three studies the following concentrating collector systems are chosen for different comparisons. The FRESH NRG collector, the Soltigua FTM-36 and the Soltigua PTMx-36 are subject to the research. Specific data sheets can be found in the Appendix 8, 0 FRESH NRG Collector, 8.2 Soltigua FTM Collector, 0 PTMx Collector. The collectors are all single axis self-tracking the sun. The FRESH NRG and the FTM are Fresnel collectors and the PTMx is a parabolic trough collector.

The parabolic trough collector system is using a trough-shaped parabolic mirror to reflect irradiation in a concentrated way onto the absorber tube. The parabolic mirrors are tracking the sun on a single axis. Due to the height of the collectors, the modules need to be installed by considering the shading factor of the single collectors. The collectors can provide temperatures up to $250°C$ and are therefore adequate for the use in process heat generation. Parabolic trough as well as Fresnel collectors require direct radiation. Diffuse radiation cannot be reflected by the mirrors.

In contrast to the parabolic system Fresnel collectors consist of many single mirrors, each facing a specific reflection angle to a fixed secondary reflector. The secondary reflector concentrates the reflections onto the absorber tube. The single mirrors are self-tracking the sun on a single axis. Due to the multiple small mirror stripes the Fresnel collectors requires more gross area compared to the parabolic trough collectors, though the Fresnel modules can be installed closer to each other. This fact has a huge impact on the gross installation area and makes the Fresnel way more efficient in the use of land area [14].

6.2. Case Study #1 Solar Cooling Southern Italy

For the first case study the potential and the performance of a solar cooling system for the FRESH NRG collector combined with a double-effect water/lithium bromide absorption chiller for shopping mall applications in southern Italy is investigated. As location Naples is chosen (latitude $40°50'N$ and longitude$14°15'E$) with almost one million inhabitants, the third largest city in Italy, which is located on the west coast of Italy at the Mediterranean Sea (Figure 6). The objective is to investigate a solar cooling system, which will be added to existing cold distribution system.

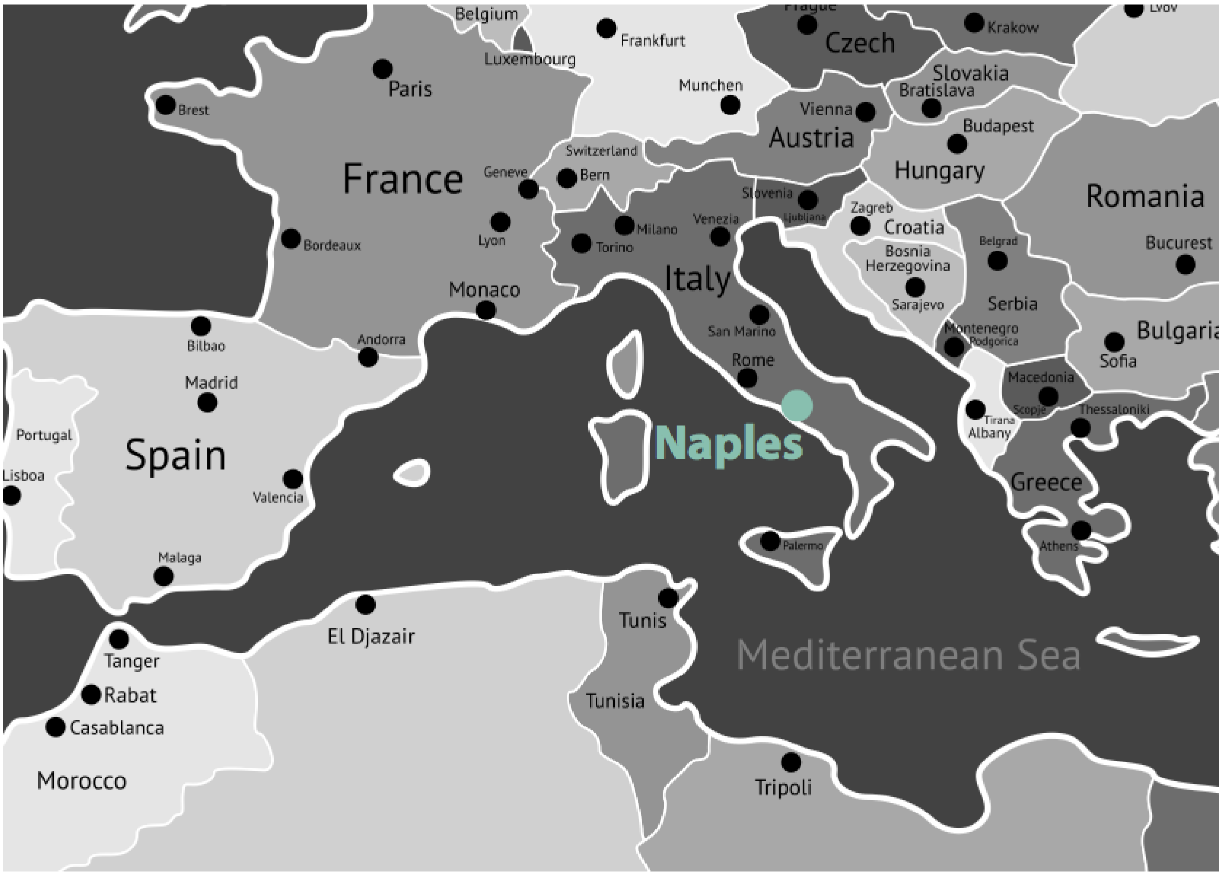


FIGURE 6: LOCATION OF NAPLES, ITALY (SOURCE: JER)

6.2.1. Energy concept

The energy concept for the shopping mall application is based on the fact, that there is a usually a high cooling demand of about $1\,MW$. The assumed operating hours of the shopping malls are between 10 am until 8 pm. Therefore, solar cooling should cover $1/3$ of the total cooling capacity of the shopping mall (Figure 7). One or more electrical chillers will serve as back up for peak load demand.

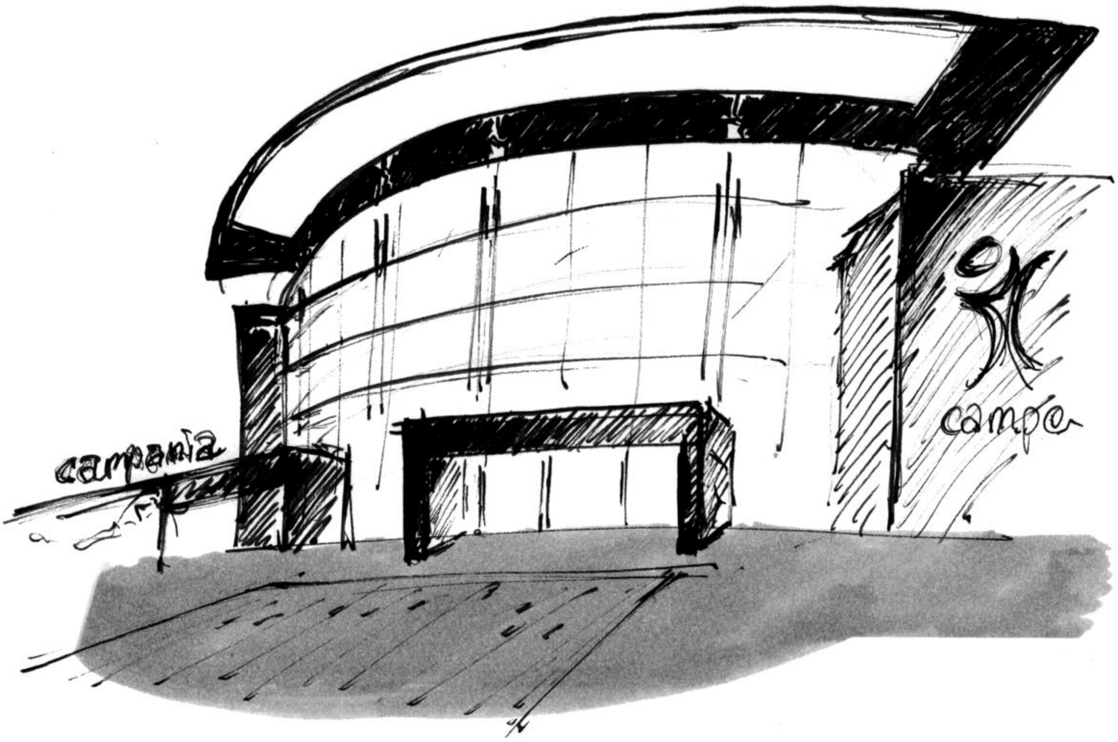


FIGURE 7: EXAMPLE OF A SHOPPING MALL IN NAPLES, ITALY (SOURCE: JER)

A double-effect water/lithium bromide absorption chiller with about 350 - 400 kW cooling capacity (100 RT) is foreseen for the solar cooling system to cover the cooling demand. The space for the solar collectors is either on the flat roof of the shopping mall or besides the building. The other parts of the solar cooling system should be installed either partially in the technical room of the building (absorption chiller, storage tank) and/or on the flat roof/beside the building (including wet cooling tower).

The existing compression chillers should be complemented (in parallel) to reach facility climate parameters inside the shopping mall. Ideally no heat storage should be integrated in the solar cooling system to use the solar heat directly for the absorption chiller.

6.2.2. Solar cooling system

Solar Collector

For this study the following solar concentrating collectors are investigated as heat supply component of the solar cooling system. The FRESH NRG collector, the Soltigua FTM-36 and Soltigua PTMx-36 are subject to the research. All three collectors are on single axes self-tracking the sun (technical details see Appendix 8).

Heat storage

In this study a buffer tank is chosen with thermo oil as heat transfer/storage fluid. Ideally there should be no heat storage included in the solar cooling system, only as short-time buffer.

Absorption chiller

The core component of the proposed solar cooling system is a double-effect water/lithium bromide absorption chiller (suggested cooling capacity of 350 kW).

TABLE 6: DOUBLE-EFFECT ABSORPTION CHILLER

Absorption chiller facts	Operation weight	Electrical connection	Power consumption
Double-effect Absorption	4,400 kg	400 V 3~ 50 Hz	3.2 kW
World Energy SWH100 **352 kW, COP = 1.36**	 FIGURE 8: ABSORPTION CHILLER (SOURCE: COURTESY OF WORLD ENERGY ABSORPTION CHILLERS)		

In this study one World Energy SWH100 double-effect water/lithium bromide absorption chiller with a nominal cooling capacity of 352 kW (100 RT) is chosen as chiller for the solar cooling system to provide 350 kW cooling capacity to the cold distribution of the shopping mall. One or more additional compression chillers can be used for peak loads or as back-up (in parallel).

Heat rejection

The waste heat of the solar cooling system should be rejected by a wet cooling tower (32.0/37.5$°C$ inlet/outlet temperature). The maximum wet bulb temperature for Naples is about 27.8$°C$. An example of a wet cooling tower is shown in Figure 9.

FIGURE 9: EXAMPLE OF WET COOLING TOWER (SOURCE: EWK)

Instead of a wet cooling tower also a hybrid cooler can be used with approximately 70% less water consumption for the recooling process. Such heat rejection technology is a combination of open-loop (evaporative) and closed-loop (dry) cooling. But the investment costs are about three times higher, then of wet cooling towers.

6.2.3. PRELIMINARY SYSTEM DESIGN

Based on the fixed cooling capacity of 350 kW the solar cooling system is designed assuming an average radiation of 800 W/m^2. For three different solar collector types (FRESH NRG collector, Soltigua FTM-36 and the Soltigua PTMx-36) are investigated. Tables 6.5 and 6.6 show the preliminary collector field, storage, chiller and heat rejection data. For the calculation a wet cooling tower is used for the heat rejection. The preliminary system calculation includes 5% of thermal field losses for every case.

TABLE 7: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Collector axis orientation (from North-South)	90°	90°	90°
Number of required modules	2	4	7
Design efficiency	0.656	0.561	0.608
Total aperture area	586 m^2	686 m^2	632 m^2
Total area required for installation	842 m^2	1,117 m^2	1,281 m^2
Heat transfer fluid	thermo oil	thermo oil	thermo oil
Design supply temperature	200°C	200°C	200°C
Design temperature difference	20 K	20 K	20 K
Row flow rate	14.38 m^3/h	7.19 m^3/h	4.11 m^3/h
Field supply rate	28.76 m^3/h	28.76 m^3/h	28.76 m^3/h
Operating pressure	max. 2 bar	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	800 W/m^2	800 W/m^2	800 W/m^2
Supply/return piping diameter (heating)	DN 100	DN 100	DN 100
Design storage time	0.5 h	0.5 h	0.5 h
Storage tank size	14 m^3	14 m^3	14 m^3
Total annual solar system yield	638 MWh/a	638 MWh/a	638 MWh/a

The resulted total aperture areas for the different collector types (FRESH NRG, FTM, PTMx) are 586 m^2, 686 m^2 and 632 m^2, respectively. It can be seen that for the same cooling load the FRESH NRG collector requires the least aperture area and has also the smallest foaotprint in relation to the total area required for installation. The short-term storage buffer tank size for 0.5 h of storage time is 14 m^3.

TABLE 8: PRELIMINARY ABSORPTION CHILLER AND HEAT REJECTION DATA

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Total rated cooling capacity	350 kW	350 kW	350 kW
Annual COP	1.2	1.2	1.2
Design chilled temperature	7°C	7°C	7°C
Design temperature difference	5 K	5 K	5 K
Supply/return piping diameter (chilled)	DN 125	DN 125	DN 125
Heat rejection capacity	642 kW	642 kW	642 kW
Design cooling temperature	32°C	32°C	32°C
Design temperature difference	5.5 K	5.5 K	5.5 K
Supply/return piping diameter (cooling)	DN 200	DN 200	DN 200

For the location Naples with the weather data of Naples, Italy the following preliminary annual performance at design conditions and annual yields for the solar cooling application are simulated as listed in Table 8.

TABLE 9: ANNUAL PERFORMANCE OF COLLECTOR TYPES

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Number of modules	2	4	7
Real collector area	601 m^2	798 m^2	640 m^2
Annual system specific yield	1.06 MWh/m^2/a	0.80 MWh/m^2/a	0.99 MWh/m^2/a
Annual solar yield	638 MWh/a	638 MWh/a	638 MWh/a
Annual recooling system yield	1,913 MWh/a	1,913 MWh/a	1,913 MWh/a
Annual chilled water yield	1,275 MWh/a	1,275 MWh/a	1,275 MWh/a

The comparison of the collector types with three different preliminary system designs for the solar cooling application is showing the significant achievement of the FRESH NRG collector (Figure 10). Not only that its technical performance and efficiency is just better than the PTMx-36 or FTM-36, but also its area efficiency is, due to the more compact design and the larger modules, more profitable.

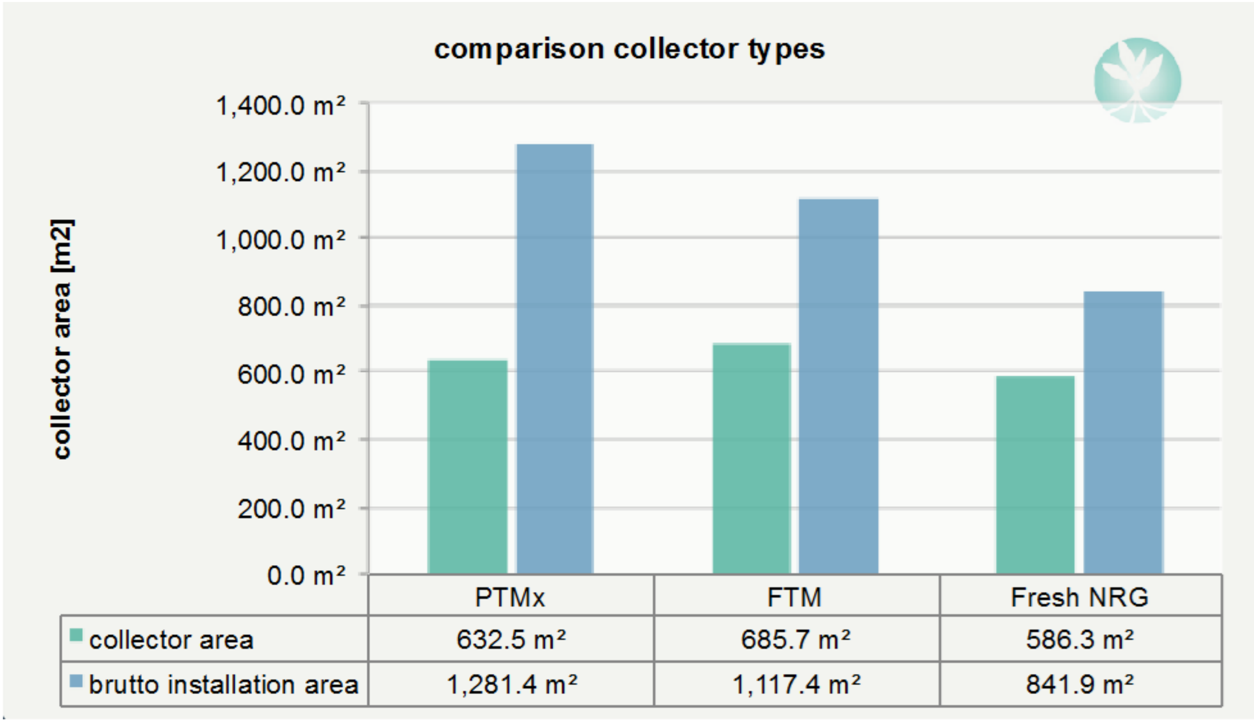


FIGURE 10: PRELIMINARY DESIGN OPTIONS (SOURCE: JER)

Furthermore, in the study different storage sizes are examined to get a better understanding, how much storage volume is required in each case for the different storage times (Figure 11) and with that operating strategies.

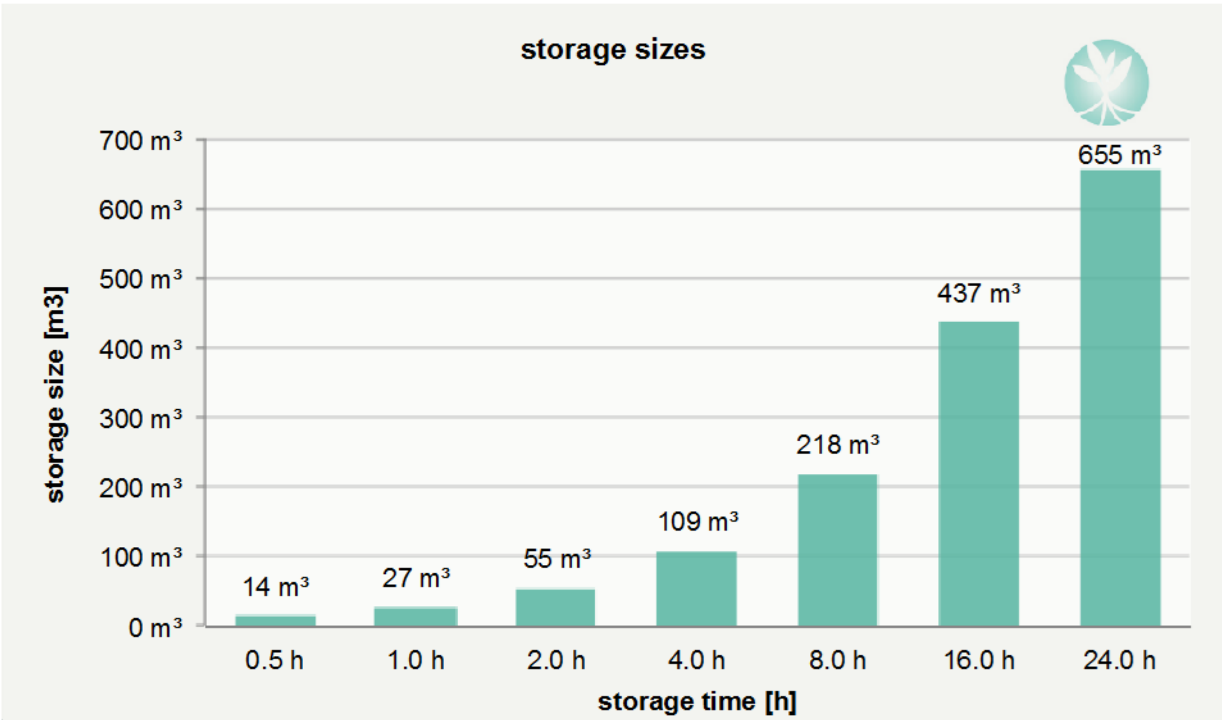
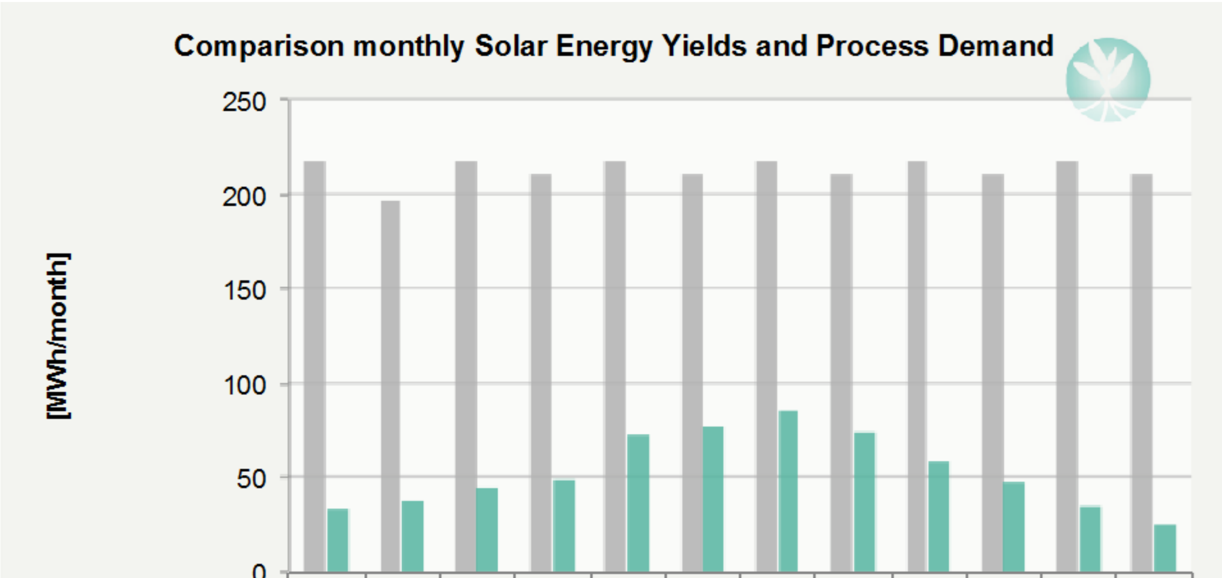


FIGURE 11: VARIABLE STORAGE TIMES (SOURCE: JER)

6.2.4. SOLAR FRACTION AND CO2 SAVINGS

The solar fraction of the system is defined to the cover of provided solar energy according to the whole solar heat demand for the absorption chiller. The aim of the designed solar system is to substitute a maximum of the conventional chilled water generation. The annual running time (3,120 h) of the air-conditioning and the corresponding heat demand of 0.3 MW for the double-effect absorption chiller means a total heat demand of 2,548 MWh per year, if the absorption chiller would running from 10am until 8pm.



	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
demand 0.3 MW	217	196	217	210	217	210	217	210	217	210	217	210
solar supply	33	38	45	49	72	77	86	74	59	47	34	24

FIGURE 12: SOLAR ENERGY YIELDS AND DEMANDS FOR SHOPPING MALL (SOURCE: JER)

The Figure 12 is showing the monthly system demand of energy for the absorption chiller and the yield of the solar system in comparison. The grey bars are showing the constant heat demand through the whole month by 0.3 MW load as heat input for the absorption chiller (corresponding cooling capacity is 350 - 400 kW). In comparison to that the green bars are showing the solar fraction of the heat generation per month. The solar fraction for the process heat varies in a range of minimum 11% in December and up to maximum of 40% in July. It need to be mentioned that this calculation does not include additional heat losses through the heat storage as well as that between 5 p.m. and 8 p.m. usually no solar heat will be available to run the absorption chiller, only if solar heat is stored during operation, but this requires a large heat storage.

The annual yield of the pre-designed solar system, including thermal field losses, amounts to 638 MWh/a, which corresponds to 1,275 MWh/a of annual chilled water yield. The CO_2 emission for Italian electricity mix is about 406 CO_2 g/kWh[1]. Each conventionally produced MWh of hot water, which is substituted by solar hot water generation, means savings of 0.406 $tons$ (CO_2). Therefore the greenhouse gas savings will reach 173 $tons$ of CO_2 per year (Table 10).

TABLE 10: CO2 EMISSIONS AND ELECTRICITY SAVINGS

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Annual chilled water yield	1,275 MWh/a	1,275 MWh/a	1,275 MWh/a
Conventional air-conditioning COP off setting	3.0	3.0	3.0
Electricity savings	425 MWh/a	425 MWh/a	425 MWh/a
CO2 savings	173 tons	173 tons	173 tons

[1] from SunEarthTools.com: CO_2 Emissions: http://www.sunearthtools.com/de/tools/CO2-emissions-calculator.php, (22nd February 2015)

6.2.5. Generic hydraulic scheme and control

A generic hydraulic scheme has been developed for the proposed solar cooling system, which consists of the solar collector field, a heat storage tank, the double-effect absorption chiller and the heat rejection unit as shown in Figure 13.

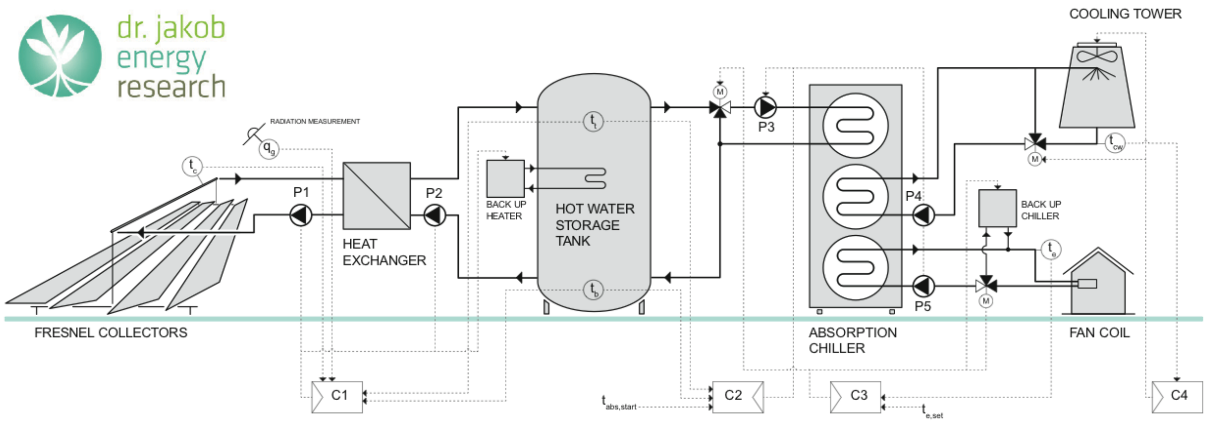


FIGURE 13: CONTROL EXAMPLE OF A COMPLETE CLOSED LOOP SOLAR COOLING SYSTEM WITH FOUR DIFFERENT CONTROL LOOPS (SOURCE: JER)

The control of a solar cooling system is a very important issue, because the system efficiency highly depends on that. If single controllers for each control loop are used, then the overall system efficiency is likely to be lower than if an overall master system controller is used. This is because multiple single control loops not always operate together in the optimum combination. A master controller combines different control loops into a single controller unit. It is recommended for solar cooling system control. The system includes four control loops C1, C2, C3 and C4, each with a different function. Controller C1 is responsible for the solar collector circuit and controls the mass flow. The second controller C2 controls the mass flow to the Absorption chiller, whereas controller C3 is responsible for the chilled water management and C4 for reccoling the chiller. These control loops are usually programmed in a PLC (programmable logic controller). PLC's are commercially available. Alternatively, a preconfigured system controller designed especially for solar cooling is offered by manufacturer SolarNext from Germany.

6.2.6. Preliminary system costing

The preliminary costing for the solar cooling system has been investigated as shown in Table 6.11. The costing is based on manufacturer information and estimates and has an estimated uncertainty of approximately ± 30%. In Table 6.11 it has been assumed that the balance of plant (BoP) including installation costs is 30% of total equipment cost. The cost figures do not include transport cost and VAT.

TABLE 11: PRELIMINARY COSTING FOR SOLAR COOLING DESIGN

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Number of collectors	2	4	7
Solar collector cost	180,300 EUR	399,000 EUR	320,000 EUR
Storage cost	10,500 EUR	10,500 EUR	10,500 EUR
Absorption chiller cost	73,000 EUR	73,000 EUR	73,000 EUR
Heat rejection cost	41,040 EUR	41,040 EUR	41,040 EUR
Total equipment cost	304,840 EUR	524,540 EUR	445,540 EUR
Total BoP incl. installation cost	91,452 EUR	157,362 EUR	133,662 EUR
Total system cost	396,292 EUR	681,902 EUR	597,200 EUR
Specific system cost	1,132 EUR/kW	1,948 EUR/kW	1,655 EUR/kW

The solar cooling system with the FRESH NRG collector comes at a system cost of about 396,292 EUR. The specific system costs are 1,132 EUR/kW for an installed solar cooling system.

6.3. Case Study #2 Solar Process Heat Chile

For the second case study the potential and the performance of a solar process heat system for the FRESH NRG collector for Chilean copper mines is investigated. A location in the in the desert highlands is chosen (latitude 23°48'S and longitude 69°03'W), which is located in the region of Antofagasta east from the Pacific Ocean (Figure 14). The objective is to investigate a solar process heat system, which will be added to existing hot water system.

FIGURE 14: LOCATION OF ANTOFAGASTA, CHILE; (SOURCE: JER)

6.3.1. Energy concept

For this study the objective is to investigate a solar process heat system, which will be added to a conventional hot water system. Hot water is used for a mining process with a peak-heating load of 3 MW at 90$°C$ (Figure 15). Main focus of the investigation is to compare different collector temperatures at 90$°C$ and 160$°C$, respectively and to proof the feasibility of a heat storage management. For the heat storage two different fluids, water and thermal oil, are investigated. The process demand of heat is set to 24 hours, 7 days a week.

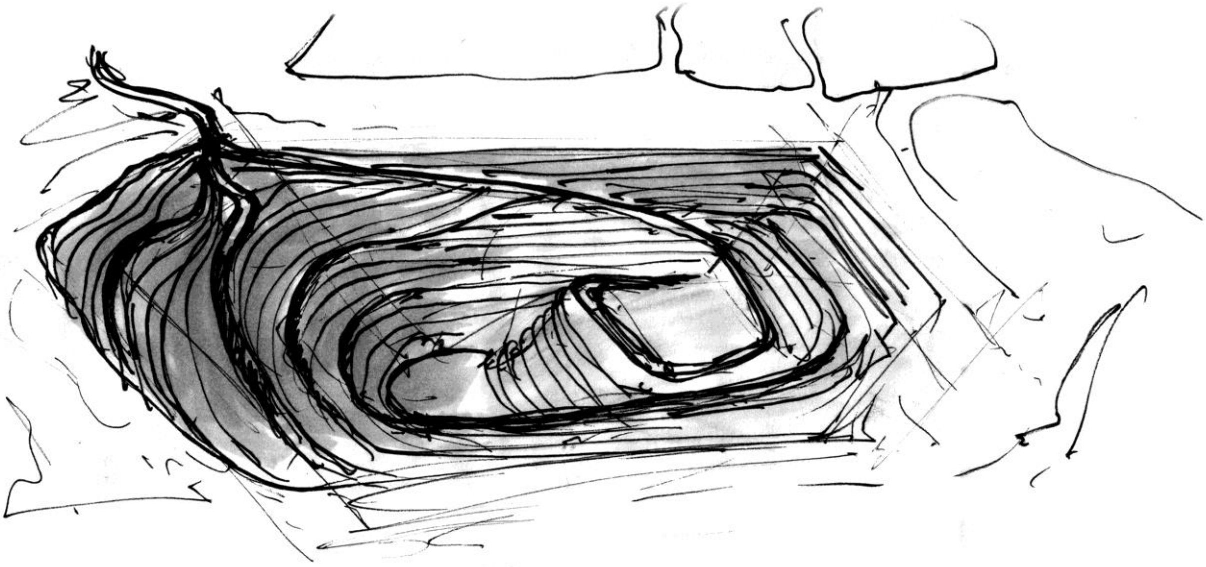


FIGURE 15: COPPER MINE IN THE REGION OF ANTOFAGASTA, CHILE (SOURCE: JER)

The target of the case is to provide a maximum cover of solar fraction to the hot water demand and to minimize the operating time of the conventional hot water diesel generation. Different collector types (FRESH NRG, Soltigua FTM-36 and Soltigua PTMx-36) will be investigated due to this analysis. A main focus of the case is to optimize the combination of storage size and collector filed size for a maximum share of process heat. Due to the heat storage management the number of tanks as well as a control strategy for filling and unfilling the different tanks will be investigated. As a result of the estimated solar fraction, possible savings in CO_2 will be calculated.

6.3.2. Solar process heat system

Solar Collector

For this study the following concentrating collector systems are chosen for comparison. The FRESH NRG collector, the Soltigua FTM-36 and the Soltigua PTMx-36 are subject to the research (technical details see appendix 8).

Heat storage

Large-scale heat storage can be used in applications such as commercial buildings, industry or district heating/cooling systems. The higher the cost for these systems is usually prohibitive for small applications. Large water heat storage tanks can be freestanding above the ground or buried underground. The state-of-the-art are freestanding, insulated steel tanks. Underground tanks also have been constructed, but in much smaller numbers than freestanding tanks. Several demonstration projects with underground tanks are currently on going to investigate the technical and economic feasibility of these tanks. Sometimes solids are mixed in with the water, such as water-sand-gravel mixture, to enhance the mechanical stability of large underground tank structures. The surface volume ratio is more favourable of larger tanks, hence the thermal losses are smaller. Even the proportions of height and diameter of those tanks affects the amount of thermal heat losses [15].

In this study freestanding steel tanks are chosen for the investigation, but with different heat transfer/storage fluids.

6.3.3. Preliminary system design

Based on the hot water generation and due to the approach to cover a total heating load of 3 MW or 2 MW (partial load), four different system configurations were investigated. Each has the aim to cover a maximum of solar fraction throughout the day. The systems are designed to assume an average radiation of 800 W/m^2. In every variation there is a comparison of the different solar collectors types (FRESH NRG collector, Soltigua FTM-36 and the Soltigua PTMx-36), any other boundary condition are set identical. The following tables are showing different heat generation and transfer strategies with their preliminary collector fields and corresponding storage datas. The preliminary system calculation includes 5% of thermal field losses for every case.

Besides the different heat load options (3 MW and 2 MW) for the preliminary system design, this case study is mainly investigating the supply of different heat levels for the process and its impact on the system design itself. Therefore heating temperatures of 90/70°C and 160/140°C are investigated (Figure 16).

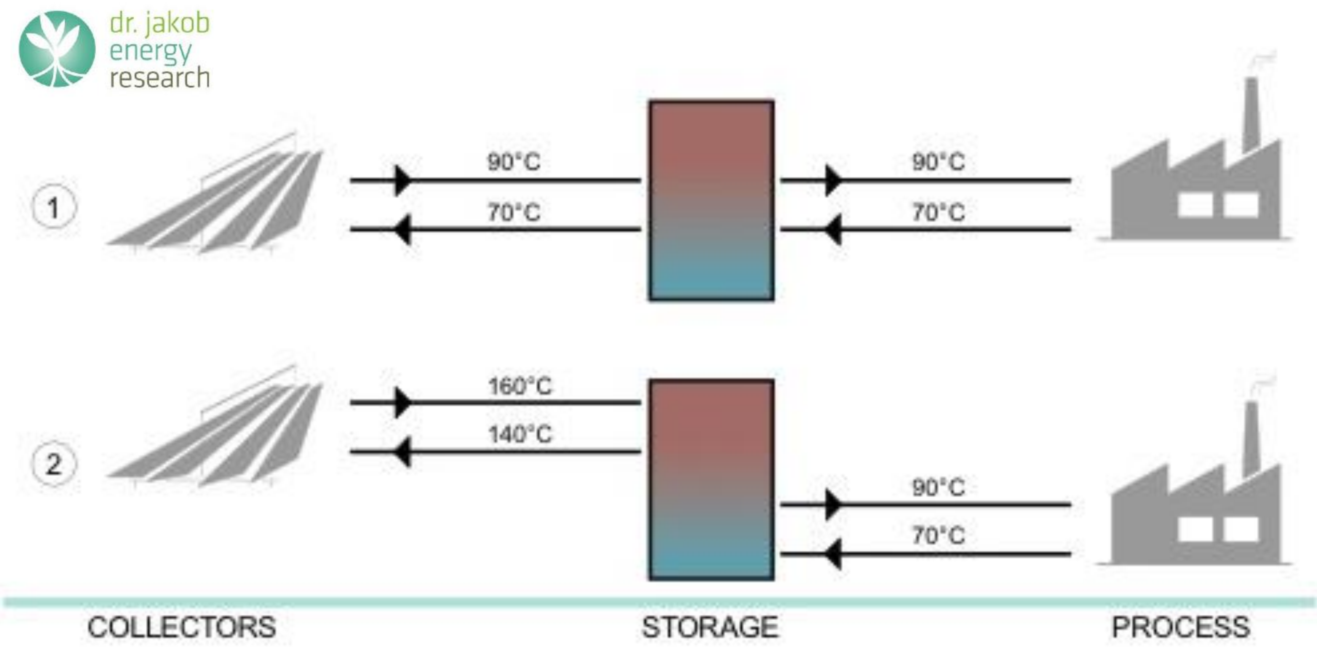


FIGURE 16: SYSTEM HEATING TEMPERATURES (SOURCE: JER)

The following tables: Table 12, Table 13, Table 14, Table 15 show the preliminary collector field and storage data for the four investigated system configurations.

System configuration 1: 3 MW peak load / heating temperature 90/70°C

TABLE 12: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA FOR 3 MW AND 90/70°C

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Collector axis orientation (from North-South)	90°	90°	90°
Number of required modules	20	32	62
Design efficiency	0.667	0.617	0.696
Total aperture area	5,909 m^2	6,381 m^2	5,658 m^2
Total area required for installation	8,419 m^2	8,939 m^2	11,350 m^2
Heat transfer fluid	water	water	water
Design supply temperature	90°C	90°C	90°C
Design temperature difference	20 K	20 K	20 K
Row flow rate	6.95 m^3/h	4.34 m^3/h	2.24 m^3/h
Field supply rate	138.97 m^3/h	138.97 m^3/h	138.97 m^3/h
Operating pressure	max. 2 bar	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	800 W/m^2	800 W/m^2	800 W/m^2
Supply/return piping diameter (heating)	DN 200	DN 200	DN 200
Design storage time	1 h	1 h	1 h
Storage tank size	132 m^3	132 m^3	132 m^3
Total annual solar system yield	7,436 MWh/a	7,436 MWh/a	7,436 MWh/a

The resulted total aperture areas for the different collector types (FRESH NRG, FTM, PTMx) are 5,909 m^2, 6,381 m^2 and 5,658 m^2, respectively. It can be seen that for the same peak load the PTMx-36 collector requires the least aperture area, but looking at the total area required for the installation the FRESH NRG collector is the best with the smallest footprint. The storage tank size for 1 h of storage time is 132 m^3.

System configuration 2: 2 MW partial load / heating temperature 90/70°C

TABLE 13: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA FOR 2 MW AND 90/70°C

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Collector axis orientation (from North-South)	90°	90°	90°
Number of required modules	14	22	42
Design efficiency	0.667	0.617	0.696
Total aperture area	3,940 m^2	4,254 m^2	3,773 m^2
Total area required for installation	5,893 m^2	6,146 m^2	7,688 m^2
Heat transfer fluid	water	water	water
Design supply temperature	90°C	90°C	90°C
Design temperature difference	20 K	20 K	20 K
Row flow rate	6.62 m^3/h	4.21 m^3/h	2.21 m^3/h
Field supply rate	92.65 m^3/h	92.65 m^3/h	92.65 m^3/h
Operating pressure	max. 2 bar	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	800 W/m^2	800 W/m^2	800 W/m^2
Supply/return piping diameter (heating)	DN 200	DN 200	DN 200
Design storage time	1 h	1 h	1 h
Storage tank size	88 m^3	88 m^3	88 m^3
Total annual solar system yield	4,958 MWh/a	4,958 MWh/a	4,958 MWh/a

The resulted total aperture areas for the different collector types (FRESH NRG, FTM, PTMx) are 3,940 m^2, 4,254 m^2 and 3,773 m^2, respectively. It can be seen again that for the same partial load the PTMx-36 collector requires the least aperture area, but looking at the total area required for the installation the FRESH NRG collector is again the best with the smallest footprint. The storage tank size for 1 h of storage time is 88 m^3.

System configuration 3: 3 MW peak load / heating temperature 160/140°C

TABLE 14: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA FOR 3 MW AND 160/140°C

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Collector axis orientation (from North-South)	90°	90°	90°
Number of required modules	20	35	68
Design efficiency	0.661	0.617	0.636
Total aperture area	5,967 m^2	6,804 m^2	6,193 m^2
Total area required for installation	8,354 m^2	9,526 m^2	12,386 m^2
Heat transfer fluid	thermal oil	thermal oil	thermal oil
Design supply temperature	160°C	160°C	160°C
Design temperature difference	20 K	20 K	20 K
Row flow rate	15.26 m^3/h	8.72 m^3/h	4.49 m^3/h
Field supply rate	305.6 m^3/h	305.6 m^3/h	305.6 m^3/h
Operating pressure	max. 2 bar	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	800 W/m^2	800 W/m^2	800 W/m^2
Supply/return piping diameter (heating)	DN 315	DN 315	DN 315
Design storage time	1 h	1 h	1 h
Storage tank size	85 m^3	85 m^3	85 m^3
Total annual solar system yield	7,436 MWh/a	7,436 MWh/a	7,436 MWh/a

The resulted total aperture areas for the different collector types (FRESH NRG, FTM, PTMx) are 5,967 m^2, 6,804 m^2 and 6,193 m^2, respectively. It can be seen that for the same peak load but at higher heating temperatures of 160$°C$ the FRESH NRG collector requires the least aperture area and has the smallest footprint in relation to the total area required for installation. The storage tank size for 1 h of storage time is 85 m^3.

System configuration 4: 2 MW partial load / heating temperature 160/140°C

TABLE 15: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA FOR 2 MW AND 160/140°C

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)	Option 3 (Soltigua PTMx-36)
Collector axis orientation (from North-South)	90°	90°	90°
Number of required modules	14	23	45
Design efficiency	0.661	0.617	0.640
Total aperture area	3,979 m^2	4,537 m^2	4,129 m^2
Total area required for installation	5,571 m^2	6,352 m^2	8,258 m^2
Heat transfer fluid	thermal oil	thermal oil	thermal oil
Design supply temperature	160°C	160°C	160°C
Design temperature difference	20 K	20 K	20 K
Row flow rate	14.55 m^3/h	8.86 m^3/h	4.53 m^3/h
Field supply rate	203.75 m^3/h	203.75 m^3/h	203.75 m^3/h
Operating pressure	max. 2 bar	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	800 W/m^2	800 W/m^2	800 W/m^2
Supply/return piping diameter (heating)	DN 250	DN 250	DN 250
Design storage time	1 h	1 h	1 h
Storage tank size	57 m^3	57 m^3	57 m^3
Total annual solar system yield	4,958 MWh/a	4,958 MWh/a	4,958 MWh/a

The resulted total aperture areas for the different collector types (FRESH NRG, FTM, PTMx) are 3,979 m^2, 4,537 m^2 and 4,129 m^2, respectively. It can be seen that for the same partial load but at higher heating temperatures of 160$°C$ the FRESH NRG collector requires the least aperture area and has the smallest footprint in relation to the total area required for installation. The storage tank size for 1 h of storage time is 57 m^3.

The specific annual performance indicates the annual solar yield of the different design strategies based on the real collector area (Table 16 to Table 18). The real collector area is calculated by the required aperture area of the system design and the specific collector sizes of the single modules.

TABLE 16: ANNUAL PERFORMANCE FRESH NRG COLLECTOR

	3 MW – 90°C	2 MW – 90°C	3 MW – 160°C	2 MW – 160°C
Number of modules	20	14	20	14
Real collector area	5,909 m^2	3,940 m^2	5,967 m^2	3,979 m^2
Annual system specific yield	1.26 MWh/m²/a	1.26 MWh/m²/a	1.25 MWh/m²/a	1.25 MWh/m²/a
Annual solar yield	7,436 MWh/a	4,958 MWh/a	7,436 MWh/a	4,958 MWh/a

TABLE 17: ANNUAL PERFORMANCE OF SOLTIGUA FTM-36

	3 MW – 90°C	2 MW – 90°C	3 MW – 160°C	2 MW – 160°C
Number of modules	32	22	35	23
Real collector area	6,381 m^2	4,254 m^2	6,804 m^2	4,537 m^2
Annual system specific yield	1.17 MWh/m²/a	1.17 MWh/m²/a	1.09 MWh/m²/a	1.08 MWh/m²/a
Annual solar yield	7,436 MWh/a	4,958 MWh/a	7,436 MWh/a	4,958 MWh/a

TABLE 18: ANNUAL PERFORMANCE OF SOLTIGUA PTMX-36

	3 MW – 90°C	2 MW – 90°C	3 MW – 160°C	2 MW – 160°C
Number of modules	62	42	68	45
Real collector area	5,658m^2	3,773 m^2	6,193 m^2	4,129 m^2
Annual system specific yield	1.31 MWh/m²/a	1.31 MWh/m²/a	1.20 MWh/m²/a	1.20 MWh/m²/a
Annual solar yield	7,436 MWh/a	4,958 MWh/a	7,436 MWh/a	4,958 MWh/a

Two different strategies were investigated for the different cases of heat load (3 MW and 2 MW). Generally it can be seen, that the generation of a 3 MW heat load is clearly requiring a larger scale of collector area. The analysis of the different collector types shows the highest required collector area for the Soltigua FTM-36 (Figure 17).
The solar hot water generation on a heating level of 90$°C$ is reveals that the PTMx-36 gets along with the lowest amount of required collector area. Its efficiency is significantly higher for low temperatures than it is for high temperatures. Beside this fact the FRESH NRG collector is providing the highest efficiency for high temperature levels as it can be seen for the system configurations 3 and 4.

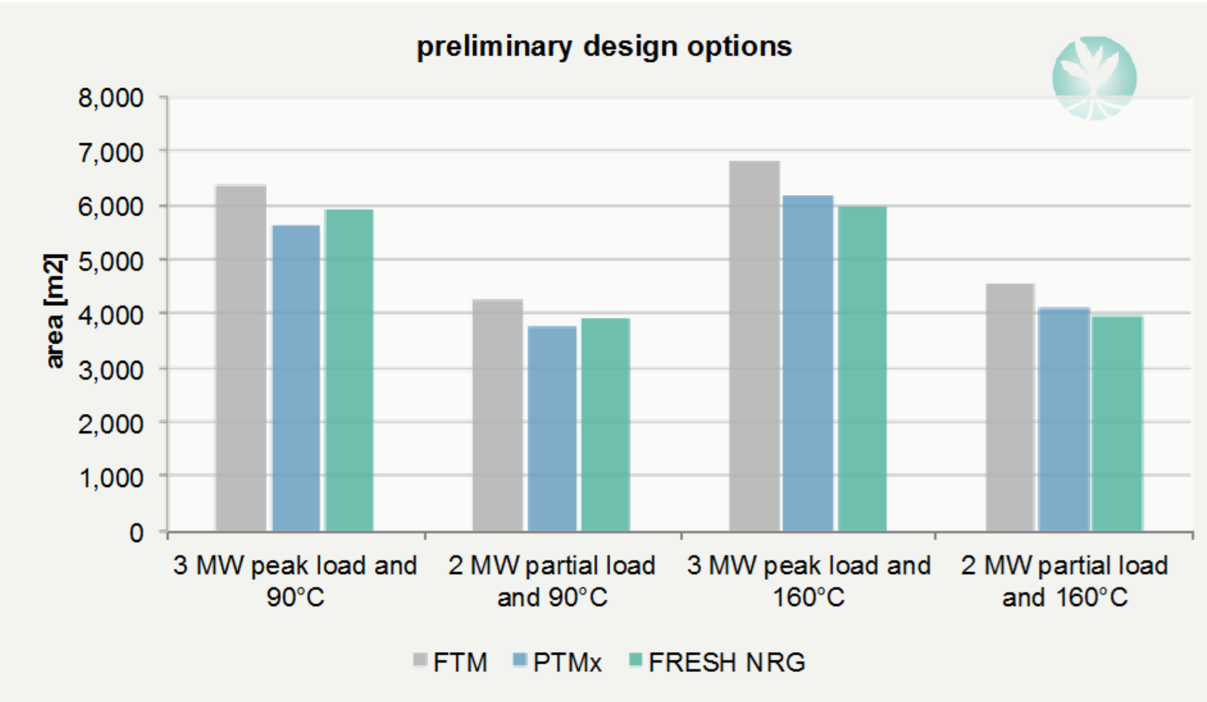


FIGURE 17: PRELIMINARY DESIGN OPTIONS (SOURCE: JER)

Although the difference and improvements between the parabolic trough (Soltigua PTMx) and the new developed FRESH NRG collector does not seem so high. It need to be meant, that the technology of the FRESH NRG collector is way more efficient in the use of the gross area, which includes beside the net required collector area also installation and shading factors for the specific collector technology. The gross installation factor for parabolic trough collectors can at least set to 2, which means that the required collector area need to be doubled to get the gross area for the collector field. The gross installation factor for the FRESH NRG collector can be set with 1.4, which means just a 40% increase of the gross collector field, compared to the net required collector area.

6.3.4. Heat storage

Storage management

A major point in the solar hot water system is the matter of heat storage. This case also investigates the question of heat storage management. Therefore the meaning of parallel and serial charging and / or discharging of heat storage tanks is essential. As well as the storage sizing and the possibility of splitting storage volume into multi tank systems (MTS).

As part of likewise examinations a scientific report has been presented on the International Conference on Solar Heating and Cooling for Buildings an Industry in 2013. A published paper deals with the content of “Variable-volume storage systems for solar heating and cooling system” and has come to the following result:

„In this paper, two novel multi-tanks system strategies are evaluated with the aim at managing the MTS thermal storage capacity as a function of the combinations of solar radiation availability and user thermal / cooling energy demands. In order to provide design and operating guidelines, case studies for four Southern and Northern Italy locations are presented. Simulation results show that, in terms of primary energy, which only includes the consumptions due to heating and cooling scopes, a better management of the storage volume is achieved by adopting the parallel charging / discharging operation strategy compared to the single tank of the first configuration. On the contrary, the adoption of a series charging and parallel discharging operation strategy does not determine any significant difference in the operation with respect to the single tank layout. […] In particular, besides the installation advantages that this modular system offers, the analysed MTS systems do not determine significant improvements in terms of economic and energetic SHC system efficiencies” [14].

This elaboration was investigating three different configurations for heat storage (Figure 18).

a. Single Tank
b. Multi Tank system, proposed parallel charging/discharging strategy
c. Multi Tank system, series charging and parallel discharging strategy

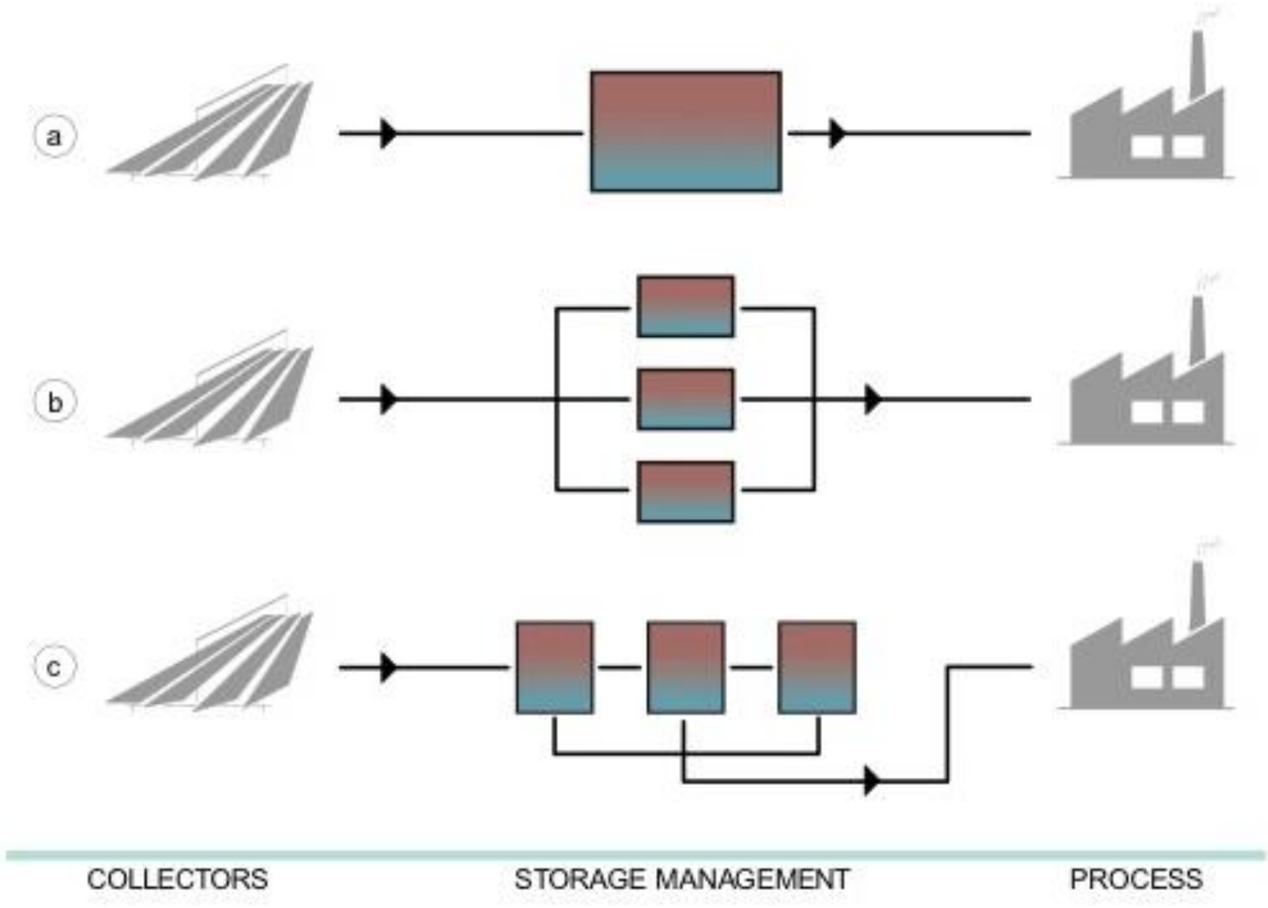


FIGURE 18: HEAT STORAGE MANAGEMENT (SOURCE: JER)

Simulations have shown that none of the alternatives beside the standard single tank configuration has brought significant achievements in heat storage management. Due to this fact this case is rather looking on the variable heat storage sizes, which are the result of different heat storage charging temperatures and varying storage fluids.

Storage sizes

In consideration of four different investigated system configurations for heat generation and transfer, each with different boundaries, the result is showing also four different storage sizes. Each provides a maximum storage capacity time of one hour. High temperature levels, above 100$°C$, are realised with thermal oil. In fact thermal oil has a lower heating capacity than water, but it can be used for high operating temperatures, at bulk temperatures up to 315$°C$. Furthermore the heat capacity of thermal oil is increasing with rising temperature. Therefore it can be used without having the problematic of high vapour pressures as it has to be considered with water. Due to this fact high temperature storages can be realised without invests in expensive high-pressure systems.

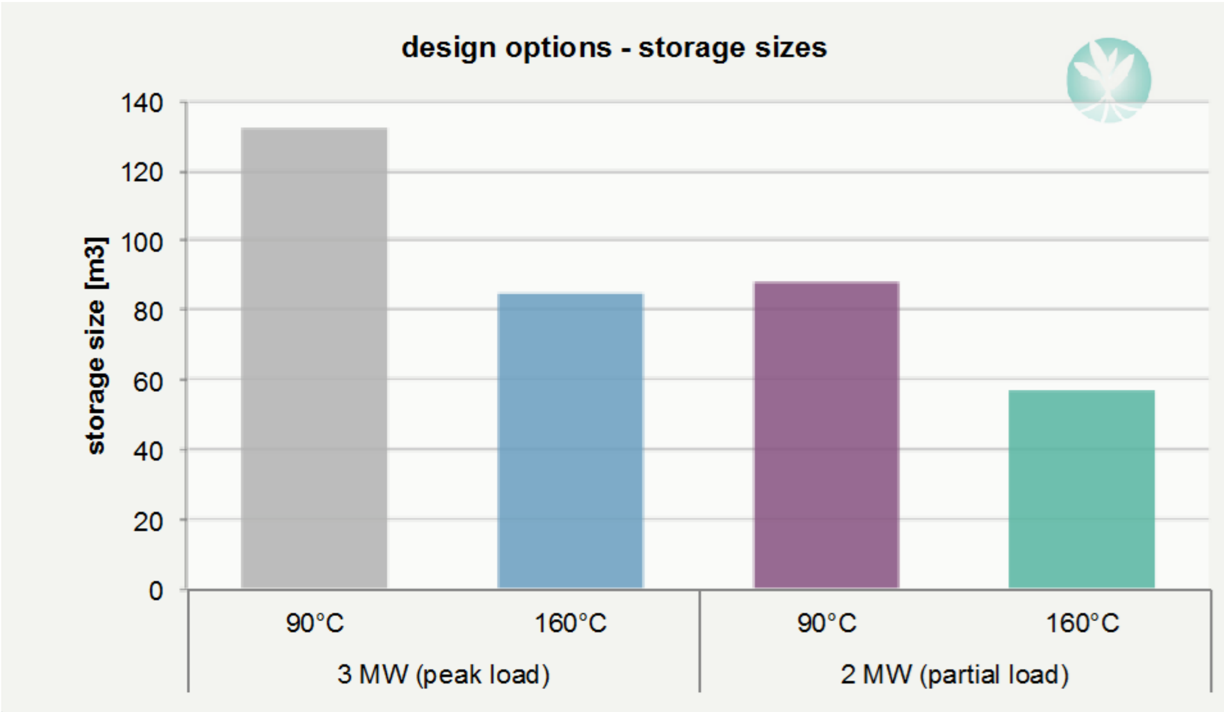


FIGURE 19: STORAGE SIZES OF THE PRELIMINARY SYSTEM DESIGN (SOURCE: JER)

Temperatures below 100$°C$ can be easily realised by conventional hot water storages. Water has a significant higher heat capacity than thermal oil. Besides the preferences of different storage fluids also the load and unload temperature levels have to be considered for the process integration of storage applications. Hereby a higher load temeperature of the storage enables the implementaion of a wider temperature inclination (ΔT) in the storage process. This effects the realization of smaller tank volumes. A scheme for different storage load temperatures can be seen in Figure 16 on page 53. In this case, increasing heat losses due to higher design temperatures are negligible compared to the savings in storage volume, because lower load temperatures, small (ΔT), are effecting an increase of the storage volume. Due to this fact the smallest tank size for 3 MW as well as 2 MW (partial load) can be realised by high temperatures and thermal oil as storage fluid (Figure 19). The heat storage volume is designed for one hour of heat supply.

Storage costs

In general it can be said that the total costs for heat storages are significantly degreasing for increasing storage sizes. The following Figure 20 presents this characteristic of storage costs which bases on several projects. The presented storage costs are without VAT.

For smaller storage sizes the costing for tanks are estimated by a logarithmic regression of different manufacturer´s prices (Figure 20). Beside the maximum volume of the storage the heat and pressure level are significant for the costs of the storage tanks. The different temperature levels, however, can have a major impact on the possible use of the tank materiality and therefore on the trend of the costs itself. As a result of that the cost for heat storage, up to 100°$\boldsymbol{C}$, is significantly lower, as storage units for high temperatures like 160°$\boldsymbol{C}$.

Table 19 shows estimated costs for heat storages in different sizes ($T_{storage}$ < 100°C).

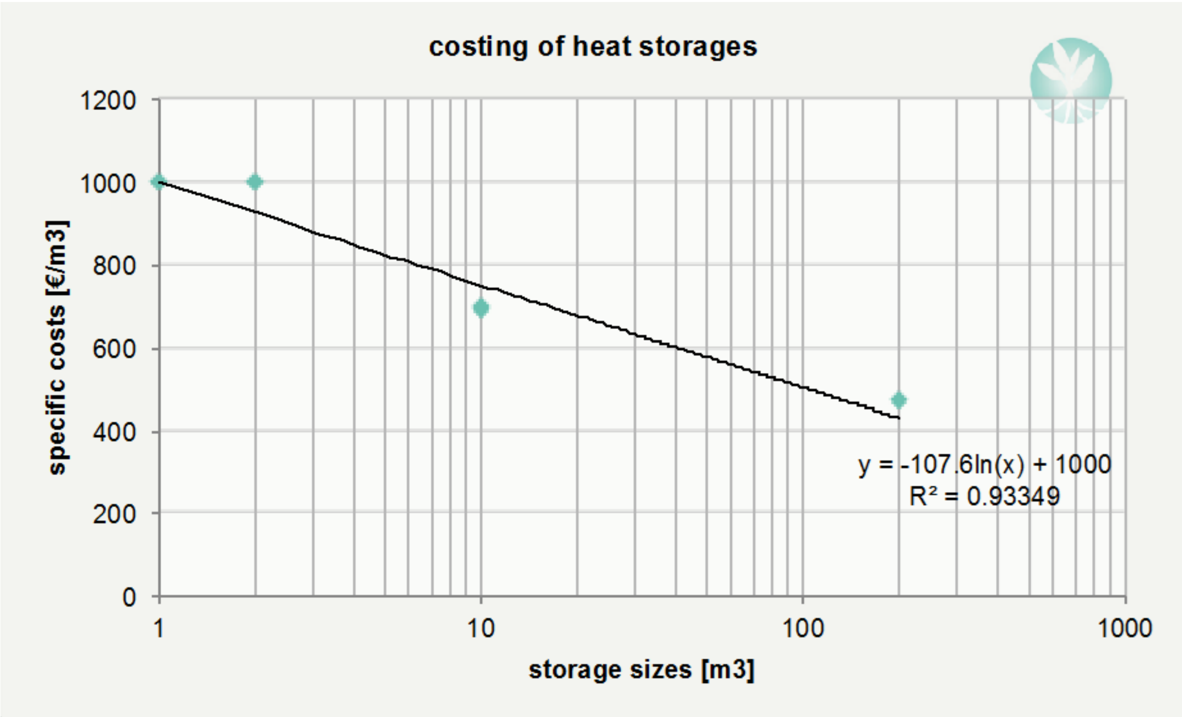


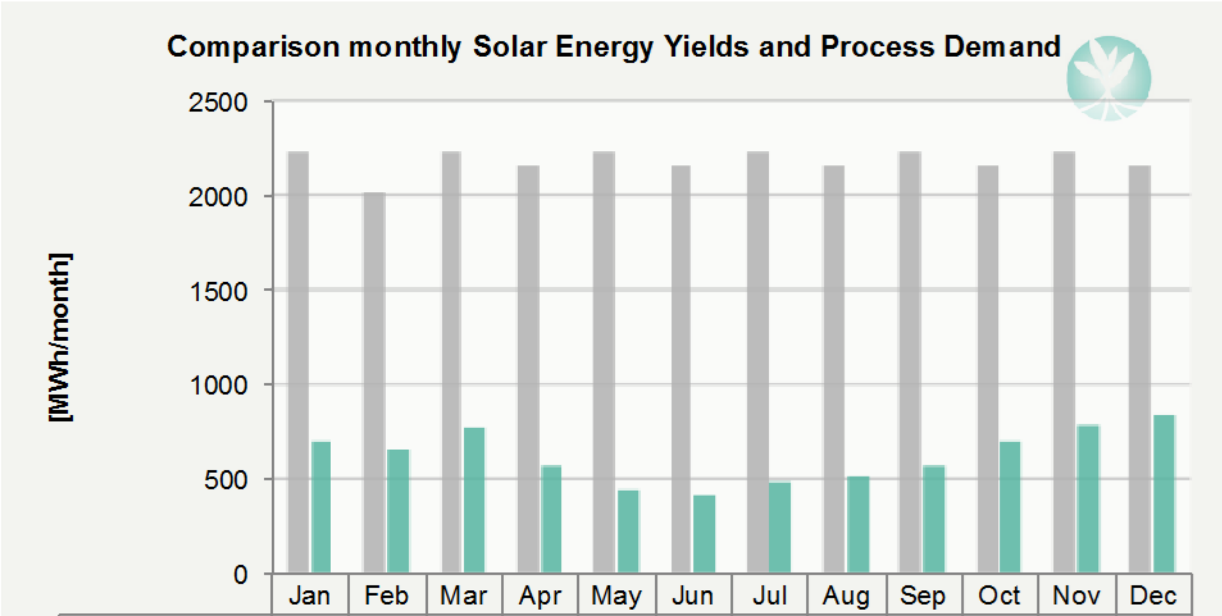
FIGURE 20: STORAGE COSTS FOR TANK VOLUMES OF 1 TO 1,000 M3 (SOURCE: JER)

TABLE 19: COSTS FOR HEAT STORAGES AT VARIABLE SIZES

Storage size V	1 m³	10 m³	50 m³	100 m³	1,000 m³
Specific costs [€/m³]	1,000	752	579	505	257
Total costs [€]	1,000	7,520	28,950	50,500	257,000

6.3.5. SOLAR FRACTION AND CO_2 SAVINGS

The solar fraction of the system is defined to the cover of provided solar energy according to the whole process heat demand. The aim of the designed solar system is to substitute a maximum of the conventional hot water generation. The annual running time (8,760 h) of the process and the heat demand of 3 MW (peak load) means a total heat demand of 26,280 MWh per year.



	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
demand 3 MW	2232	2016	2232	2160	2232	2160	2232	2160	2232	2160	2232	2160
solar supply	702	656	766	573	449	412	479	516	565	698	779	838

FIGURE 21: SOLAR ENERGY YIELDS AND DEMANDS FOR COPPER MINE (SOURCE: JER)

Figure 21 shows the monthly system demand of energy and the yield of the solar system in comparison. The grey bars are showing the constant heat demand through the whole month by a 3 MW load. In comparison to that the green bars are showing the solar fraction of the heat generation per month. The solar fraction for the process heat varies in a range of minimum 19% in June and up to maximum of 39% in December. It needs to be mentioned that this calculation does not include the additional heat losses through the heat storage itself.

Conventional heat generation

TABLE 20: CONSUMPTION AND EMISSIONS

	1 h	1 d	1 week	1 month	1 year
Litres	337 l/h	8,082 l/d	56,575 l/week	244,887 l/m	2,949,964 l/h
CO_2	0.8 t CO_2	19.2 t CO_2	134.6 t CO_2	582.5 t CO_2	7,016.8 t CO_2

To provide a constant heat demand of 3 MW the conventional heating process takes up to 8,082 litres fuel per day. Burning fossil fuels emits greenhouse gases in to the atmosphere. The most common greenhouse gas is CO_2. Other greenhouse gases can be rated by a CO_2 equivalent factor. The conventional hot water generation system is fired by a diesel boiler which has an enormous impact on CO_2 emissions. The emissions are roughly about 19 tons[2] of CO_2 a day, according to the operating grade. Calculated for an operating time of 24h a day and 7 days per week this means a total maximum amount of 7,016 $tons$ of CO_2 a year.

The annual yield of the pre-designed solar system, including thermal field losses, amounts to 7,434 MWh/a. Each conventionally produced MWh of hot water which is substituted by solar hot water generation means savings of 0.267 tons of CO_2. Therefore the greenhouse gas savings will reach 1,985 tons of CO_2 per year.

[2] IPCC, 2006: The Emission Factors. Online available: http://www.eumayors.eu/IMG/pdf/technical_annex_en.pdf, (14th January 2015)

Coverage of heat demand

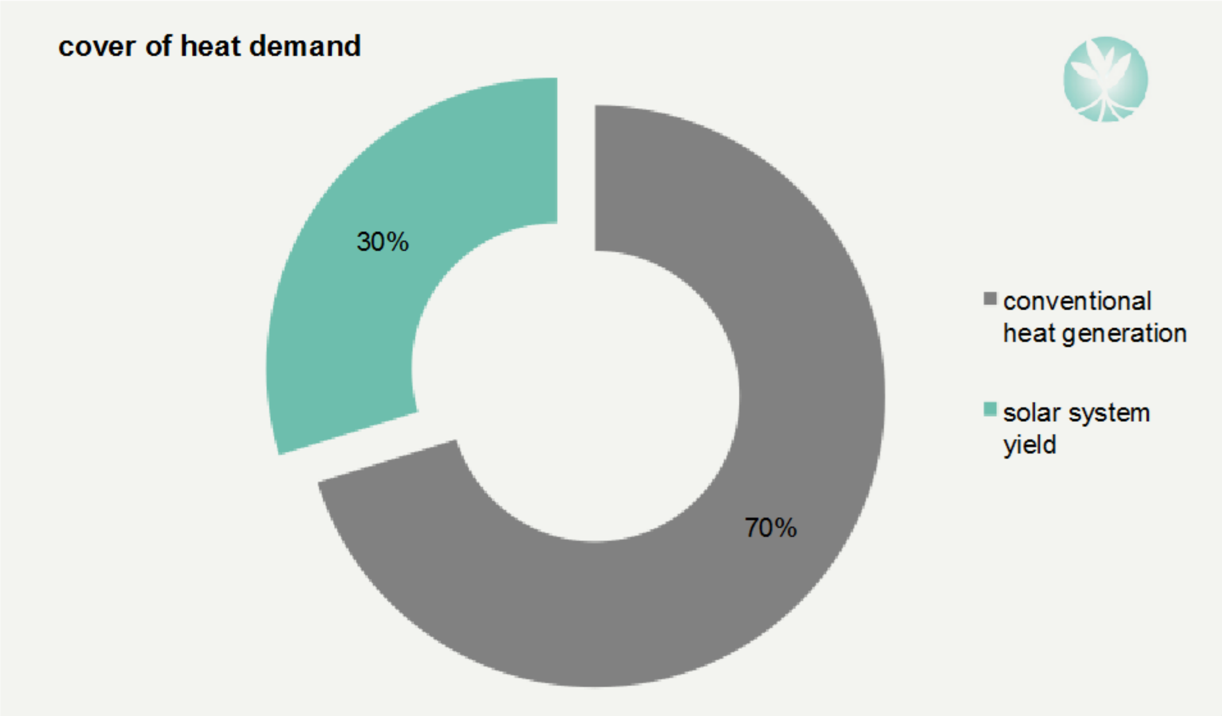


FIGURE 22: SHARES OF HEAT GENERATION (SOURCE: JER)

heat demand	3	[MW]
operating time	8760	[h]
total demand	26,280	[MWh]
solar fraction	7,434	[MWh]
conventional	17,520	[MWh]

Considering not only burning processes of fossil fuels (diesel) but also life cycle analysis of the fuel, emissions are calculated by a carbon dioxide equivalent. The savings of CO_2 emissions can be calculated by the LCA emission factor of 0.305 $[tCO_2 - eq/MWh]$[3]. Regarding to the solar fraction this means annual greenhouse gas savings of 2,267 $[tCO_2 - eq]$ by the use of solar energy.

[3] from ELCD, The European reference Life Cycle Database: The Emission Factors. Online available: http://www.eumayors.eu/IMG/pdf/technical_annex_en.pdf, (14th January 2015)

6.3.6. GENERIC HYDRAULIC SCHEME AND CONTROL

A generic hydraulic scheme has been developed for the proposed solar hot water generation system, which consists of the solar collector field and a heat storage tank.

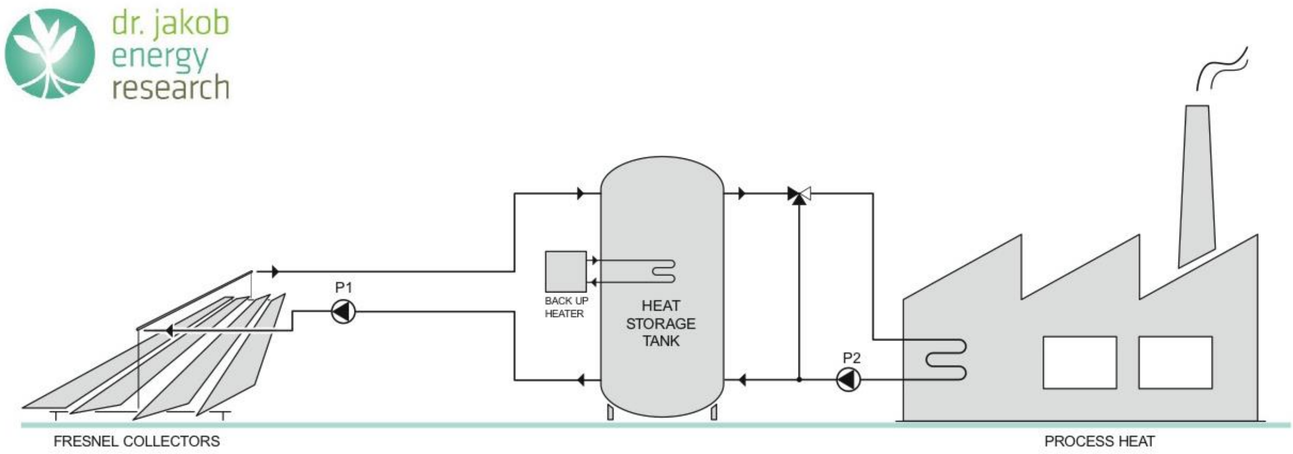


FIGURE 23: GENERIC HYDRAULIC SCHEME (SOURCE: JER)

According to the different design options, thermal oil or water can be used as heat transfer fluids in the solar field (primary loop). The collector filed is designed as a low-pressure installation with maximum pressure of 2 bar this makes it much easier for the design and finally the components are cheaper than for high-pressure installations. The fluid is passed through the storage tank for charging the heat storage. The storage tank also serves as a hydraulic compensator. The secondary loop is feeding the hot water process itself. In this case water or thermal oil, like Mobiltherm 605 (specific pour point of $-12°C$ and flash point of $230°C$), can be used. The chosen thermo oil can be used in indirect heating and cooling installations in all kinds of industrial processes operating at bulk temperatures up to $315°C$ [4].

The controller for the primary loop is responsible for the solar collector circuit and controls the mass flow of pump P1 to maintain a constant temperature difference between collector and storage tank. The larger the collector fields the more thermal mass and slower reaction time. Radiation-based control is typically recommended for collector fields smaller than $100\ m^2$. Field sizes larger than that should use temperature-based control.

[4] Data from product data sheet Mobiltherm 605, Mobil Oil Company Limited. Online available: http://www.ulei-mobil.ro/pdf/MobilIndustrieDataSheet/Mobiltherm%20605%20pds.pdf

6.3.7. Preliminary system costing

The preliminary costs for the solar process heat system varies with regard to different solar field sizes and the variable storage sizes. Even the selection of the storage fluid can drive the total system costs. The cost estimations are based on manufacturer information, estimated as well as data based on experiences. The calculation has an estimated uncertainty of approximately ± 30%. The cost figures do not include transport costs.

Option 1: 3 MW - 90°C (FRESH NRG)

Option 2: 3 MW - 90°C (PTMx)

Option 3: 3 MW -160°C (FRESH NRG)

Option 4: 3 MW -160°C (PTMx)

TABLE 21: PRELIMINARY SYSTEM COSTING FOR DIFFERENT SYSTEM CONFIGURATIONS

	FRESH NRG 90°C	PTMx 90°C	FRESH NRG 160°C	PTMx 160°C
Solar collector cost	1,800,000 €	2,852,000 €	1,800,000 €	3,128,000 €
Storage cost	62,700 €	62,700 €	44,370 €	44,370 €
Fluid cost	370 €[5]	370 €[5]	195,000 €	195,000 €
Total equipment cost	1,863,070 €	2,915,070 €	2,039,370 €	3,367,370 €
Total BoP cost	372,614 €	439,014 €	407,874 €	673,474 €
Total installation cost	242,199	291,507	265,118	336,737€
Total system cost	2,477,833	3,645,591	2,712,362	4,377,581
Specific system cost	412 €/m²	642 €/m²	451 €/m²	703 €/m²

The total equipment cost results of the specific system design compilation for the different options. Out of that the total BoP cost is calculated by a rate of 20% of the total equipment cost. The total installation cost can be calculated via a percentage rate of the total costs of the equipment with 10% for the PTMx and 13% for the Fresnel collectors. The percentage rate for the installation of the Fresnel is clearly higher because to the more complex installation of the Fresnel itself. These installation costs are only estimated values. The total system cost for the different design options are varying from 2.5 Million EUR to 4.4 Million EUR.

[5] Bárbara A. Willaarts Alberto Garrido M. Ramón Llamas: (2014) Earthscan Studies in Water Resource Management, Routledge: Water for Food Security and Well-being in Latin America and the Caribbean. Social and Environmental Implications for a Globalized Economy

6.4. Case Study #3 Solar Process Heat Jordan

For the third case study the potential and the performance of a solar process heat system for the FRESH NRG collector to produce steam at 10 bars in Jordan is investigated. As location Amman is chosen (latitude 31°56'N and longitude 35°55'E), which is located in the north-west of Jordan to the east of the Dead Sea (Figure 24). The objective is to investigate a solar process heat system, which will be added to existing steam system.

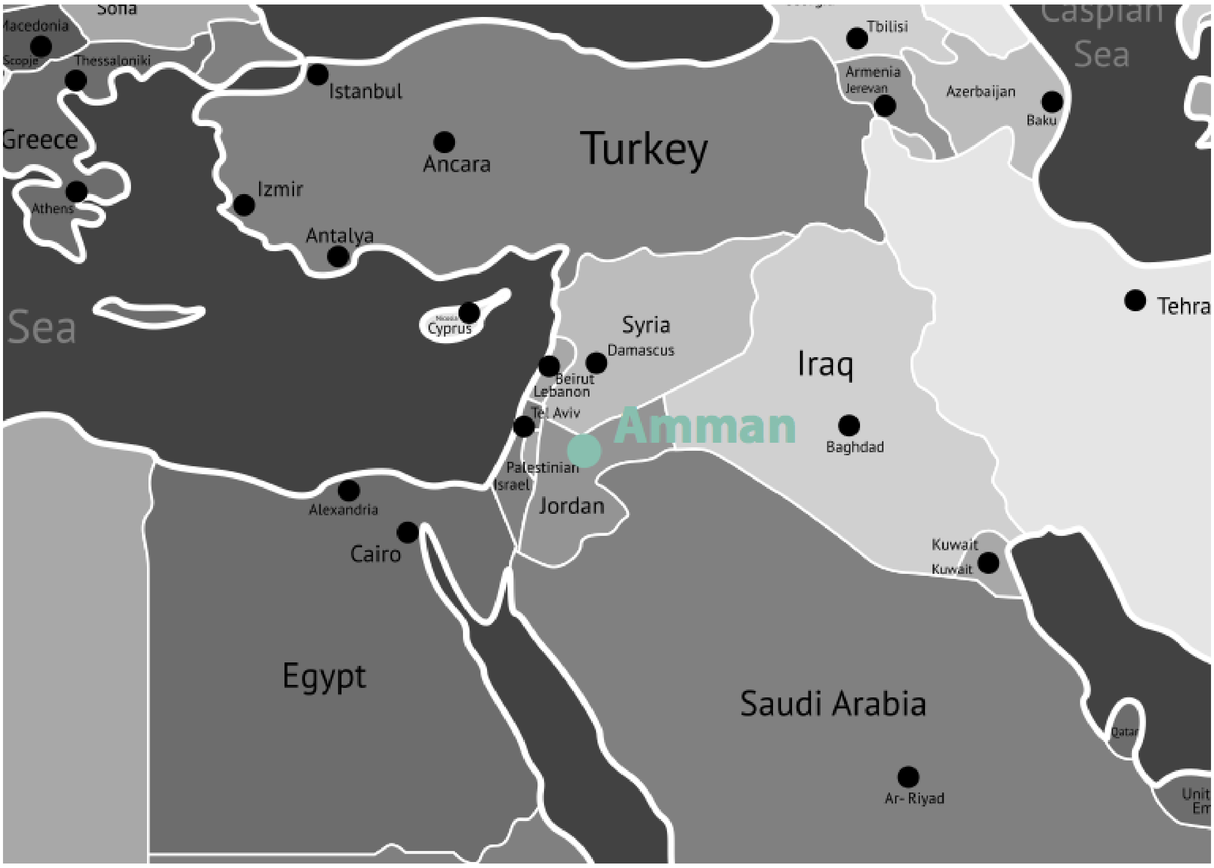


FIGURE 24: LOCATION OF AMMAN, JORDAN (SOURCE: JER)

6.4.1. ENERGY CONCEPT

For this study the objective is to investigate a solar process heat system, which will be added to a conventional steam system of a paper factory (Figure 25) to provide a constant demand profile of 1 MW (1.4 $tons/h$ of steam). Its usage time is set to 24 hours, 6 days a week. The temperature level for the process heat is 180°C. The target of the case is to provide a maximum cover of solar fraction to the steam process to minimize the operating time of the conventional diesel steam generation. Storage of steam is not considered. Overproduction due to oversize and high solar yields can be transported to dump heat into the environment by a dry-recooler.

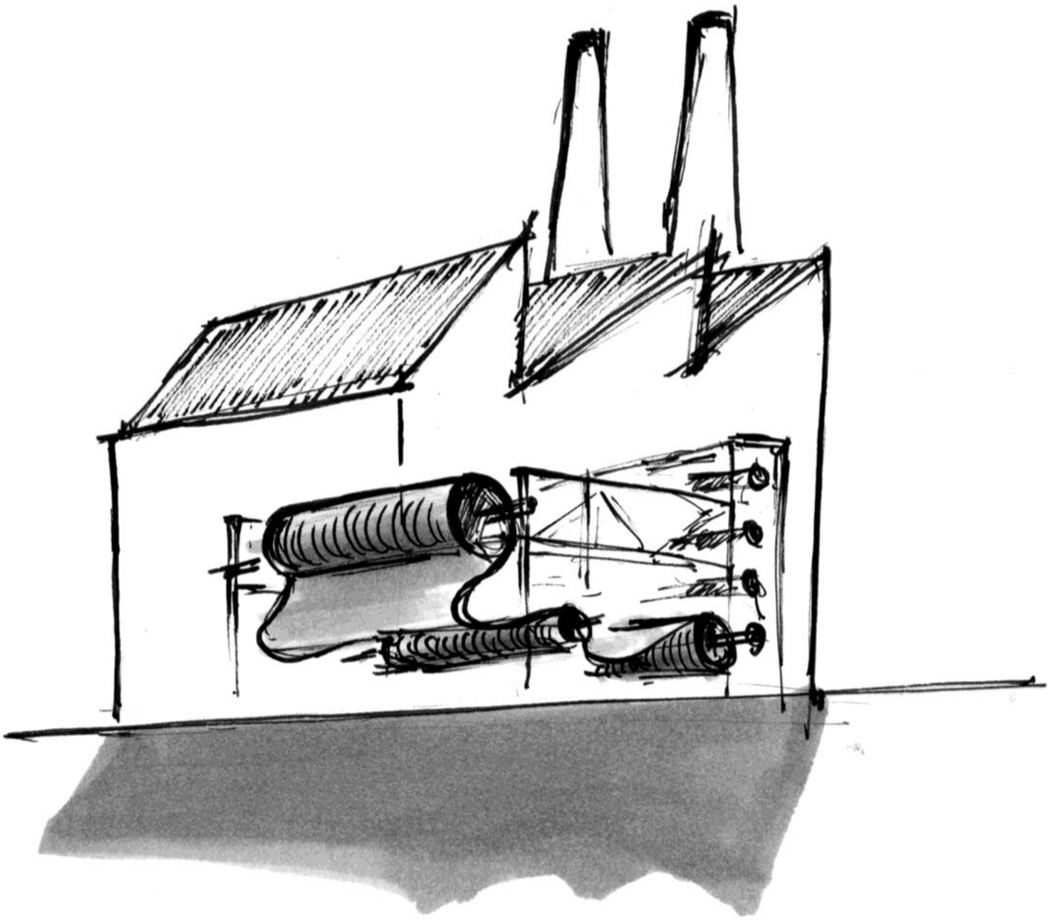


FIGURE 25: PAPER FACTORY IN JORDAN (SOURCE: JER)

Main focus of the case is to give a statement, about the required space of the collector area. Different collector types (FRESH NRG and Soltigua FTM-36) will be investigated due to this analysis. Beneath an estimated system design for the cogeneration-steam production, savings in comparison to a conventional steam-production-system will be calculated.

6.4.2. SOLAR STEAM GENERATION SYSTEM

Solar Collector

For this study the following concentrating collector systems are chosen for comparison. The FRESH NRG collector and the Soltigua FTM-36 Linear Fresnel Collector are subject to the research. Both collectors are on single axes self-tracking the sun (technical details see appendix 8).

Heat storage

In this study a freestanding steel tank is chosen for the investigation with thermo oil as heat transfer / storage fluid.

6.4.3. PRELIMINARY SYSTEM DESIGN

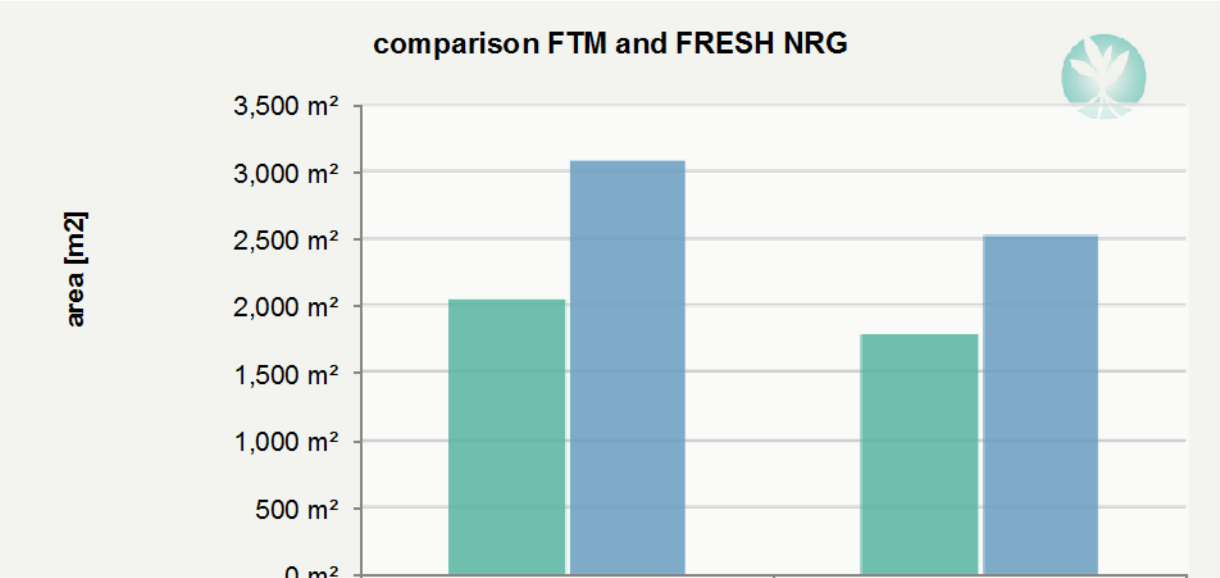
Based on the considered steam generation and due to the approach to cover a constant heat load of 1 MW, with maximum cover of solar fraction throughout the day, the system is designed to assume an average radiation of 900 W/m^2. In the first step there is a comparison of the two different solar collectors (FRESH NRG collector and the Soltigua FTM-36), any other boundary condition is set identical. The Table 22 below is showing the preliminary collector field and storage data for the two options. The preliminary system calculation includes 5 % of thermal field losses in both cases.

TABLE 22: PRELIMINARY COLLECTOR FIELD AND STORAGE DATA

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)
Collector axis orientation (from North-South)	90°	90°
Number of required modules	6	11
Design efficiency	0.656	0.563
Total aperture area	1,804 m^2	2,195 m^2
Total area required for installation	2,526 m^2	3,073 m^2
Heat transfer fluid	thermal oil	thermal oil
Design supply temperature	220°C	220°C
Design temperature difference	20 K	20 K
Row flow rate	16.2 m^3/h	8.8 m^3/h
Field supply rate	96.67 m^3/h	97.15 m^3/h
Operating pressure	max. 2 bar	max. 2 bar
Design radiation for pipe sizing	900 W/m^2	900 W/m^2
Supply/return piping diameter (heating)	DN 200	DN 200
Design storage time	0.5 h	0.5 h
Storage tank size	47 m^3	47 m^3
Total annual solar system yield	2,788 MWh/a	2,788 MWh/a

Due to the fact, that both systems are designed to provide the same heat load of 1 MW, they are both resulting with the same storage size of 47 m^3 and identical field supply rate as well as identical supply/return diameter. Also both systems are running on thermal oil as the heat transfer fluid. Only the fact of different collector efficiencies varies the specific required collector area. This affects the total amount of required collectors.
The resulted total aperture areas for the different collector types (FRESH NRG, FTM) are 1,804 m^2 and 2,195 m^2, respectively. It can be seen that the FRESH NRG collector requires the least aperture area and has also a smaller footprint in relation to the total area required for installation.

The comparison of the collector types with two different preliminary system designs is showing the significant achievement of the FRESH NRG collector (Figure 26). Not only that its technical performance and efficiency is just better than the FTM-36, but also its area efficiency is, due to the more compact design and the larger modules, more profitable.



	FTM	FRESH NRG
collector area	2,046 m²	1,778 m²
gross installation area	3,073 m²	2,526 m²

FIGURE 26: PRELIMINARY DESIGN OPTIONS (SOURCE: JER)

The higher design efficiency of the FRESH NRG collector compared to the FTM-36 itself is saving 268 m^2 of required net area. The also larger module sizes of the FRESH NRG collector, which increased from 200 m^2 (FTM-36) to roughly 300 m² (FRESH NRG), is saving gross area in a range of 547 m^2.

The specific annual performance indicates the annual solar yield of the different design strategies based on the real collector area (Table 23). Hereby, the real collector area is calculated by the required aperture area of the system design and the specific collector sizes of the single modules.

TABLE 23: ANNUAL PERFORMANCE OF FRESH NRG COLLECTOR AND FTM-36

	Option 1 (FRESH NRG)	Option 2 (Soltigua FTM-36)
Number of collectors	6	11
Real collector area	1,804 m^2	2,195 m^2
Annual system specific yield	1.55 $MWh/m^2/a$	1.27 $MWh/m^2/a$
Annual solar yield	2,788 MWh/a	2,788 MWh/a

6.4.4. Solar fraction and CO_2 savings

The solar fraction of the system is defined to the cover of provided solar steam according to the whole process heat demand. The aim of the designed solar system is to substitute a maximum of the conventional steam generation. The annual running time (7,488 h) of the process and the heat demand of 1 MW means a total heat demand of 7,488 MWh per year.

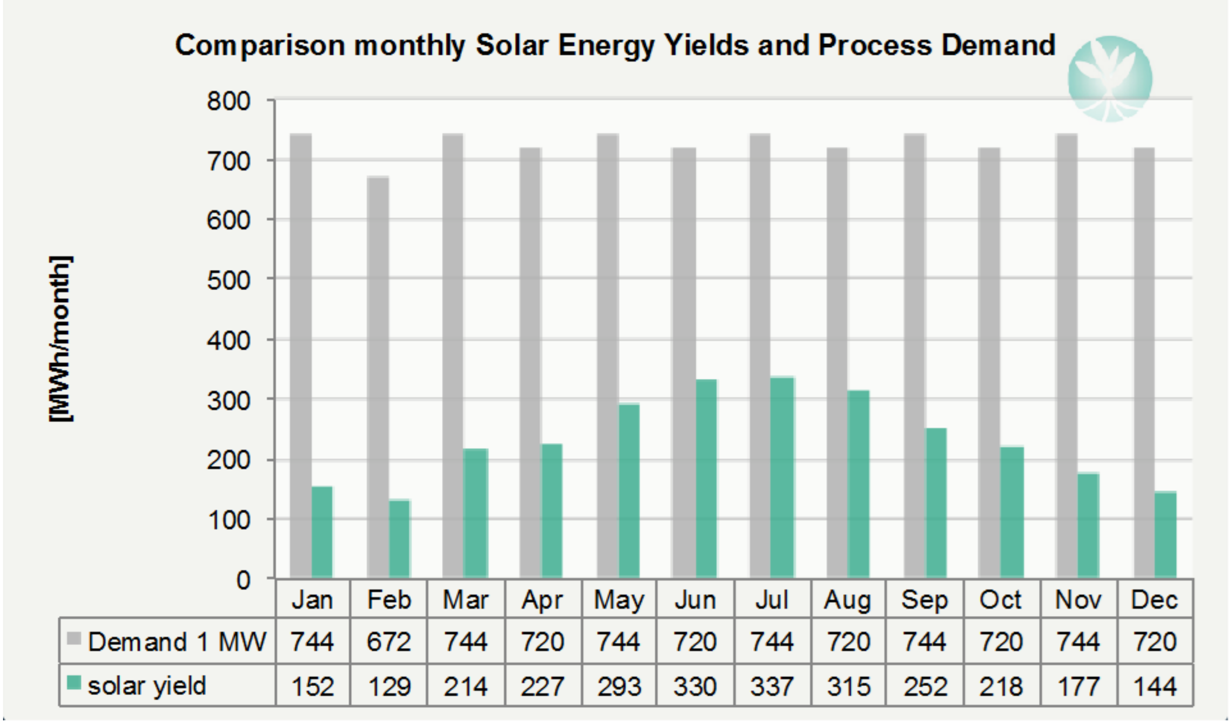


FIGURE 27: SOLAR ENERGY YIELDS AND DEMANDS FOR PAPER MILL (SOURCE: JER)

The Figure 27 shows the monthly system demand of energy and the yield of the solar system in comparison. The grey bars are showing the constant heat demand through the whole month by a 1 MW load. The green bars are standing for the heat demand during the daytime, when solar energy can actually be used. In comparison to that the green bars are showing the real solar fraction of heat generation per month. The solar fraction for the process heat varies in a range of minimum 19% in February and up to maximum of 46% in June. It needs to be mentioned that this calculation does not include the additional heat losses through the heat storage itself.

6.4.5. Conventional steam generation

TABLE 24: CONSUMPTION AND EMISSIONS

	1 h	1 d	1 week (6 d)	1 month (26 d)	1 year (7488h)
litres	126 l/h	3,032 l/d	18,191 l/week	78,826 l/m	945,914 l/a
CO_2	0,3 t CO_2	6.4 t CO_2	38.4 t CO_2	166.6 t CO_2	1,999 t CO_2

Burning fossil fuels emits greenhouse gases in to the atmosphere. The most common greenhouse gas is CO_2. Other greenhouse gases can be rated by a CO_2-equivalent factor. The conventional steam generation system is fired by a diesel boiler and has an enormous impact on CO_2 emissions. The emissions are roughly about 6.4[6] tons of CO_2 a day. Calculated for an operating time of 24h a day and 6 days per week this means a total amount of 1,999 tons of CO_2 a year. In this case the use of solar energy to produce steam can achieve great savings of CO_2 emissions.

The annual yield of the pre-designed solar system, including thermal field losses, amounts to 2,800 MWh/a. Each conventionally produced MWh of steam which can be substituted by the solar steam generation means savings of 0.267 tons of CO_2. Therefore the greenhouse gas savings will reach 747.6 tons of CO_2 per year.

6.4.6. Coverage of heat demand

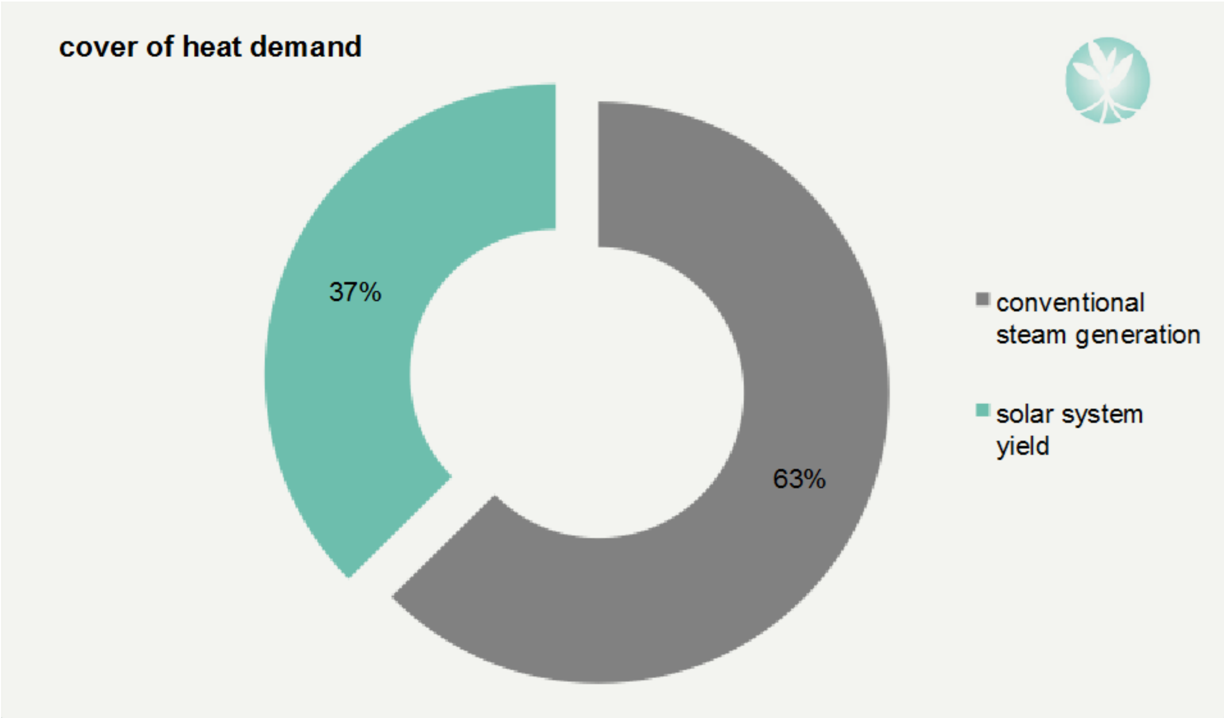


FIGURE 28: SHARES OF HEAT GENERATION (SOURCE: JER)

heat demand	1	$[MW]$
operating time	7,488	$[h]$
conventional	4,688	$[MWh]$
solar fraction	2,800	$[MWh]$

[6] IPCC, 2006: The Emission Factors: online available: http://www.eumayors.eu/IMG/pdf/technical_annex_en.pdf, (14th January 2015)

Considering not only burning processes of fossil fuels (diesel) but also life cycle analysis of the fuel, the CO_2 emissions are calculated by a carbon dioxide equivalent. The savings of CO_2 emissions can be calculated by the LCA emission factor of 0.305 $[t\ CO_2 - eq/MWh]$[7]. Regarding to the solar fraction this means annual greenhouse gas savings of 854 $[t\ CO_2 - eq]$, by the use of solar energy.

6.4.7. Generic hydraulic scheme and control

A generic hydraulic scheme has been developed for the proposed solar steam generation system (Figure 29), which consists of the solar collector field and a storage tank including back-up.

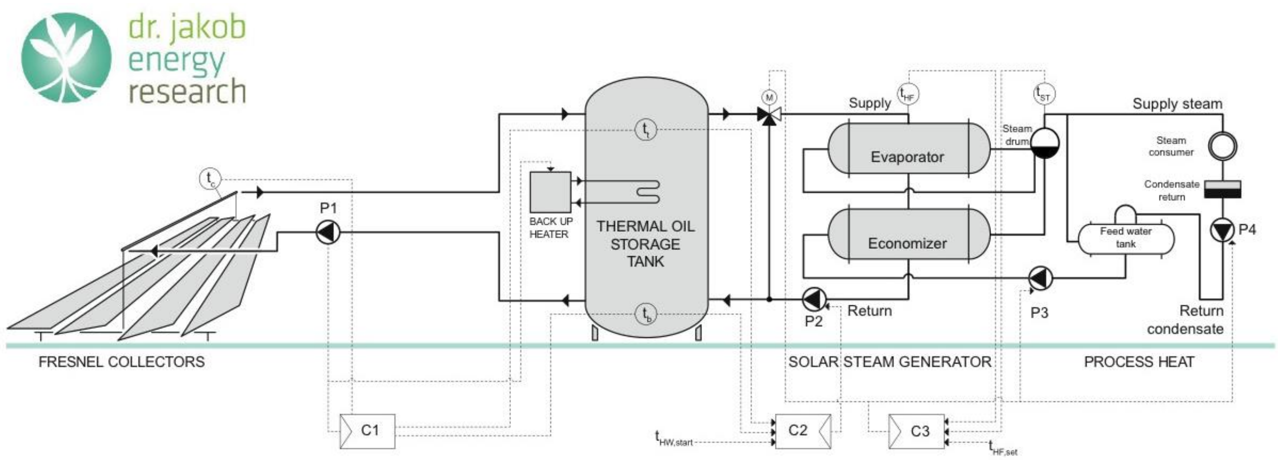


FIGURE 29: GENERIC HYDRAULIC AND CONTROL STRATEGY SCHEME (SOURCE: JER)

Thermal oil is used as heat transfer fluid in the solar field (primary loop) and is passed through the storage tank (used as a hydraulic compensator) in to a secondary loop to produce steam. The operating pressure of thermo oil was chosen at max. 2 bar (compared to 16 – 20 bar, if water was used as a heat transfer fluid), which makes it much easier for the design of the primary loop and finally the components are cheaper. Thermal oil can be used in indirect heating and cooling installations in all kinds of industrial processes operating at bulk temperatures up to 315$°C$. Generated steam is led into the steam drum and further to the steam consumer itself. By the consumption of the heat in the process, steam is going to condensate.

The system control includes three control loops C1, C2, C3, each with a different function (Figure 29). These control loops are usually programmed in a PLC (programmable logic

[7] from ELCD, The European reference Life Cycle Database: The Emission Factors http://www.eumayors.eu/IMG/pdf/technical_annex_en.pdf, (14th January 2015)

controller). PLC's are commercially available. Controller C1 is responsible for the solar collector circuit and controls the mass flow to maintain a constant temperature difference between collector and storage tank. The storage operates as a hydraulic buffer in the system. Controller C2 controls the mass flow of the process of steam generation depending on the hot water storage tank temperatures. In the end controller C3, which has an integrated set point for switching the steam supply circuit on/off, controls the steam supply to the factory.

LIST OF FIGURES

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7. BIBLIOGRAPHICAL REFERENCES

[1] Hardon, Jean-Christophe, “Storage Solution for Solar Thermal Energy”, online available: http://www.viking-house.ie/downloads/Long%20Term%20Heat%20Stores,%20Swiss%20Study.pdf

[2] Laing, D. “Solar Thermal Energy Storage Technologies”, DLR, 2008. Online available: http://www.dlr.de/Portaldata/41/Resources/dokumente/institut/thermischept/Solar_Thermal_Energy_Storage_Technologies_Hannover2008.pdf

[3] IRENA, ETSAP report, Jan. 2013. Online available: https://www.irena.org/DocumentDownloads/Publications/IRENA-ETSAP%20Tech%20Brief%20E17%20Thermal%20Energy%20Storage.pdf

[4] A. A. Al-Maaitah, A. Olimat, and A. Salaymeh, “Designing building and testing of a thermal energy storage container utilizing phase change material concept” GCREEDER 2013 conference, September 10-12, Jordan University, Amman, Jordan.

[5] A. A. Al-Maaitah, “Utilization of CSP system in Tri-generation of Electricity, water, cooling and heating in the Southern Part of Jordan”, in Proceedings of SOLAR2013 conference, American Solar Energy Association, Baltimore, MD, USA April 16-20, 2013. Online available: http://proceedings.ases.org/wp-content/uploads/2014/02/SOLAR2013_0155_final-paper.pdf

[6] T. Naegler Int. J. Energy res. 2015; 39:2019-2030, 2015 John Wiley & Sons, Ltd. SOI:10.1002/er

[7] S.Mekhilef et al. Renewable and Sustainable Energy Reviews 15 (2011) 1777-1790

[8] Based on IEA data from the *Energy Technology Perspectives 2012* © OECD/IEA 2015, www.iea.org/statistics. Licence: www.iea.org/t&c.

[8] Solar Heat Itegration in Industrial Process SHC Task 49 http://task49.iea-shc.org/

[9] C. Brunner AEE Intec, 2015, IEA Task 49

[10] Database for solar heat plants http://ship-plants.info/

[11] B. Schmitt, Integration thermischer Solaranlagen zur Bereitstellung von Prozesswärme in Industriebetrieben,Dissertation Univ. Kassel, 2014

[12] Renewable Energy in Industrial Applications, an assessment of 2050 potential, (UNIDO)

[13] Solar Heat Integration in Industrial Processes, IEA SHC Task 49, SolarPaces, Dec 2014

[14] Paul Kohlenbach, Uli Jakob: (2014) Solar Cooling. The Earthscan Expert Guide to Solar Cooling Systems. Routledge Chapman & Hall

[15] Annamaria Buonomano, Francesco Calise, Gabriele Ferruzzi and Laura Vanoli, Energy Procedia 48, SHC: (2013) Variable-volume storage systems for solar heating and cooling system. A case study for different Italian climates

Figures

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World Energy Absorption Chillers (2016): World Energy Absorption Chillers Europe Ltd. Sky Business Centres - Unit 21, Port Tunnel Business Park - Clonshaugh - Dublin 17 – Ireland – *Permission granted (02.05.2016)*

8. APPENDIX

Appendix 1: Data Sheet: FRESH NRG Collector

Appendix 2: Data Sheet: Soltigua FTM Collector

Appendix 3: Data Sheet: Soltigua PTMx Collector

8.1. FRESH NRG COLLECTOR

The following Table 25 shows the technical data of the FRESH NRG collector given by Soltigua (status as of August 2014).

TABLE 25: TECHNICAL DETAILS FRESH NRG COLLECTOR

Type	FRESH NRG collector
Length	37.35 m
Width	8.05 m
Height	4.95 m
Optical efficiency η_0	0.67
Linear loss coefficient a_1	0.032 W/m² K
Quadratic loss coefficient a_2	0.00018 W/m² K²
Temperature range	water / steam: ≤ 220 °C thermal oil: ≤ 320 °C
Picture	 Source: JER

8.2. Soltigua FTM Collector

Table 26 shows the technical data of the Soltigua FTM (-36) collector (status as of June 2013).

TABLE 26: TECHNICAL DETAILS SOLTIGUA FTM-(36) COLLECTOR

Type	Soltigua FTM (-36)
Length	38.08 m
Width	5.24 m
Height	3.59 m
Optical efficiency η_0	0.65
Linear loss coefficient a_1	0.41 W/m^2 K
Quadratic loss coefficient a_2	-
Temperature range	water / steam: ≤ 220 °C thermal oil: ≤ 250°C
Picture	 Source: Soltigua

8.3. Soltigua PTMx Collector

The technical data of the Soltigua PTMx (-36) collector is shown in Table 27 below (status as of August 2013).

TABLE 27: TECHNICAL DETAILS SOLTIGUA PTMX-(36) COLLECTOR

Type	Soltigua PTMx (-36)
Length	38.62 m
Width	2.37 m
Height	2.57 m
Optical efficiency η_0	0.747
Linear loss coefficient a_1	0.64 W/m^2 K
Quadratic loss coefficient a_2	-
Temperature range	hot water: ≤ 110 °C thermal oil: ≤ 250°C
Picture	 Source: Soltigua