**UNIVERSITY OF THESSALY**

**SCHOOL OF ENGINEERING**

**DEPARTMENT OF MECHANICAL ENGINEERING**

Postgraduate Course 2009-2010:
ENERGY, INDUSTRIAL PROCESSES AND POLLUTION ABATEMENT TECHNOLOGY
in the field of
Processes Design & Analysis

---

# FEASIBILITY STUDY OF THE INSTALLATION OF A HYBRID SOLAR PLANT IN A GREEK DAIRY PRODUCTS FACTORY

*Thesis submitted in partial fulfillment of the requirements*
*for the Degree of Postgraduate Course of Science*
*of the Mechanical Engineering Department*
*by*

**SKRETA OLGA**
Dpl in Electrical Eng., AU of Thessalonica, 1996

*Supervisor Professor:*
Asst.Prof. TASSOS STAMATIS

---

VOLOS 2010

**Εγκρίθηκε**

**από τα Μέλη της**

**Τριμελούς Εξεταστικής Επιτροπής:**


**<u>Πρώτος Εξεταστής</u>:** **(Επιβλέπων)**

**Δρ. ΣΤΑΜΑΤΗΣ ΑΝΑΣΤΑΣΙΟΣ**
*Επίκουρος Καθηγητής Εφαρμοσμένης Θερμοδυναμικής – Θερμικών Σταθμών,*
*Τμήμα Μηχανολόγων Μηχανικών,*
*Πανεπιστήμιο Θεσσαλίας*

**<u>Δεύτερος Εξεταστής</u>:**

**Δρ. ΣΤΑΜΑΤΕΛΛΟΣ ΑΝΑΣΤΑΣΙΟΣ**
*Καθηγητής,*
*Τμήμα Μηχανολόγων Μηχανικών,*
*Πανεπιστήμιο Θεσσαλίας*

**<u>Τρίτος Εξεταστής</u>:**

**Δρ. ΑΝΔΡΙΤΣΟΣ ΝΙΚΟΛΑΟΣ**
*Αναπληρωτής Καθηγητής,*
*Τμήμα Μηχανολόγων Μηχανικών,*
*Πανεπιστήμιο Θεσσαλίας*

# Feasibility Study of the Installation of a Hybrid Solar Plant in a Greek Dairy Products Factory

*Application: Agricultural Corporations Union of Volos (Dairy products Factory)*

by

**SKRETA OLGA**
Dpl in Electrical Eng., AU of Thessalonica, 1996

Supervisor Professor:
Asst.Prof.: **TASSOS STAMATIS**

## ABSTRACT

This study is submitted in partial fulfilment of the requirements for the Degree of Postgraduate Course of Science 2009-2010 of the Mechanical Engineering Department (School of Engineering of University of Thessaly).

Solar plants in the domestic or industrial sector are under continuous development during last years, due to the worldwide serious problems of energy crisis and environmental pollution.

The main scope of this feasibility study is to present a general methodology of the design, modelling and optimization of the installation of a solar energy plant in a dairy products Factory, along with the existing conventional energy plant, in order to cover the total energy requirements.

The base factory is the Agricultural Corporations Union of Volos (EVOL), which lies in the A' industrial Area of Volos in Greece. All energy requirements of the factory are covered by a conventional crude-oil system for heating and by the Greek Public Power Corporation for the machinery's operation and cooling.

Solar applications for energy providing, are very advantageous in an industry with very high energy consumption, since the specific location of the factory, as a typical Mediterranean place, is characterized by strong solar potential. On the other hand, due to the extremely high initial investment cost, the implementation problems of pure solar plants in combination with certain benefits of the operation of the existing conventional energy system, suggests that a hybrid solar plant could be more viable. Therefore a feasibility study would be a useful tool, for the definition of the applicability and the optimum size of equipment, before applying such installation.

The study consists of the following stages:

- Selection of a typical solar technology for heating, cooling and electricity generation, considering technical and economical parameters, the availability in market, the installation suitability for the individual needs and operation conditions of the factory.
- Design of the total hybrid solar plant.
- Developing of a simplified model in steady state of the previously designed hybrid solar plant in EES, along with the financial analysis for a 20-years life.
- Optimization of critical technical and financial variables for the appropriate sizing of the hybrid components for the greater performance of the total plant and the financial profitability of the corresponding investment.
- Parametric analysis of different sized systems, for further discussion about the potentiality of similar applications in dairy products factories.

Although the model is oriented to the specific base factory, it can be applied to similar feasibility analysis for other types of buildings, other climate conditions and other solar technologies considering the particularities of each case. From this point of view, it is expected that, the results and discussion remarks of this study could be useful to similar future studies.

## Acknowledgements

*First of all my greatest debt is to Supervisor Professor TASSOS STAMATIS [Assistant Professor] for his precious guidance and valuable assistance at the duration of my work. His contribution to fulfilment of present thesis was invaluableness.*

*I also want to acknowledge the other members of the final Examination Committee in the Mechanical Engineering Department (School of Engineering of University of Thessaly), Dr Stamatellos Anastasios [Professor] and Dr Andritsos Nikolaos [Associate Professor] for their careful reading of present thesis and their unique suggestions.*

*I appreciate the assistance of Production Manager in Agricultural Corporations Union of Bolos, Mr Tsaganos Basilios [Dpl Electrical Engineer], for his subscription to the receipt of some inceptive data elements, in the field of specific dairy products factory products factory's operation.*

*I am thankful to my good friend Vlaxoutsos Argiris [Dpl Mechanical Engineer] for his contribution to the object selection of present work.*

*Finally, I want to acknowledge all the members of my family for their support and tolerance especially at the last period of my efforts. I dedicate my work to my kids Vaggelis and Xara Stila.*

*Skreta Olga*

# CONTENTS

# List of TABLES

# List of FIGURES

# Abbreviations

| | |
|---|---|
| AC: | Absorber Chiller |
| COP: | Coefficient Of Performance (chillers) |
| CRES: | Centre of Renewable Sources of Energy |
| EES: | Engineering Equations Solver |
| GIEC: | Gross Inland Energy Consumption |
| IRR: | Internal investment's refund rate |
| MC: | Mechanical Chiller |
| Mtoe: | Million tons of oil equivalents |
| NPV: | Net present value of an investment |
| PP: | Pay-Back Period of an investment |
| PV: | Photovoltaic System |
| OHRS: | Operation hours per day |
| RES : | Renewable Sources of Energy |
| SC: | Solar Collector |
| TCI: | Total Investment Cost |
| TH: | Thermal |

*"Turn to the sun and you will left all shadows behind"*

*German Saying*

# CHAPTER 1:

# INTRODUCTION

## 1.1 In General

Energy, which is necessary for economic development and human's prosperity, must be physically and economically worldwide accessible. The gradually exhaust of fossil fuels and the increasingly voracious energy consumption, has give to production and use of energy a political dimension, mainly after the oil crises of 1985, in all countries of developing world.

In Greece, energy consumption requirements (Figure 1) have been increasing by 50% between 1990 and 2006 and in average was increasing by 3% yearly, accordingly to economy development indicator (Table.1). The inability of domestic production to satisfy energy requirements has the result of external dependence, which is increasing and costly to Greek economy [1].

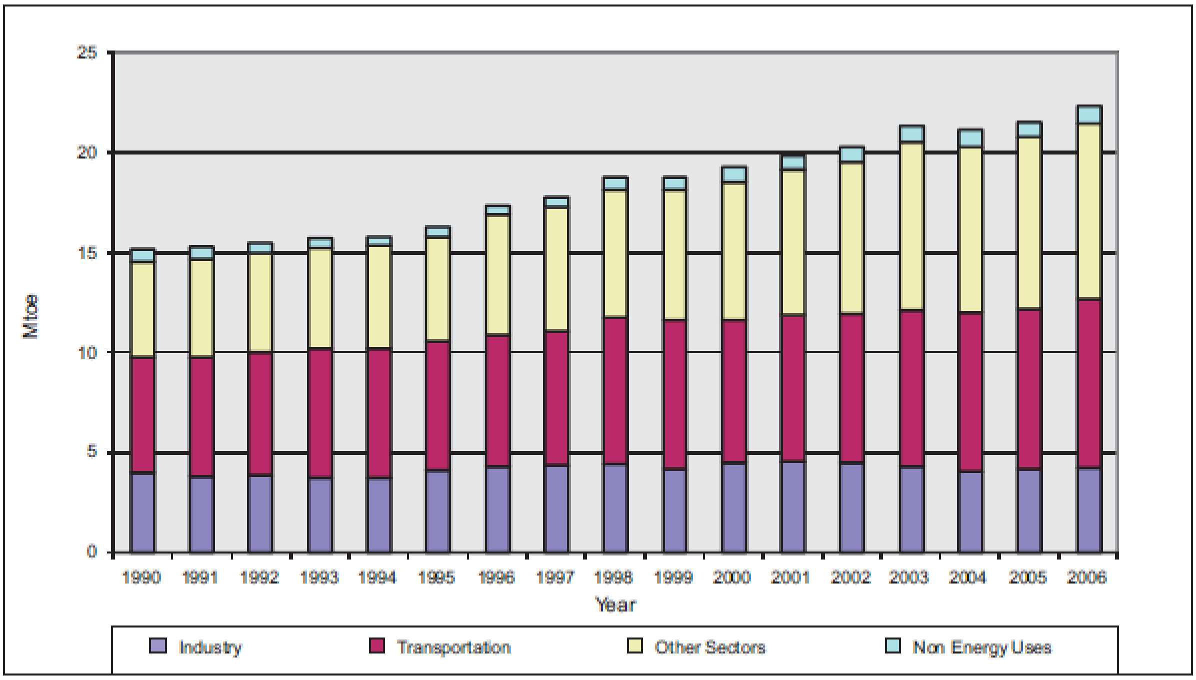



**Figure 1: Final energy consumption by sector in Greece [1]**

| | 1996 | 1997 | 1998 | 1999 | 2000 | 2001 | 2002 | 2003 | 2004 | 2005 | 2006 | 1995-2006 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| GDP | 2,4% | 3,6% | 3,4% | 3,4% | 4,5% | 4,2% | 3,4% | 5,6% | 4,9% | 2,9% | 4,5% | 3,9% |

**Table 1: Economic Development in Greece [1]**

On the other hand, the extended, worldwide, environmental pollution is a major problem, which menace even the planet viability. Environmental pollution is indissoluble connected to the energy production and use, due to the intensive use of conventional technologies.

In the major of cases, energy requirements are satisfied:

- Heating by conventional thermal systems of fossil fuels,
- Cooling loads by mechanical driven chillers (electrical consumption) and
- Electricity is produced in huge thermal conventional stations.

The emissions of these activities are responsible for the greenhouse effect and intensify the vicious circle of the climate changes.

In Figure 2, is shown the contribution of each economy's sector to CO2 emissions, due to the use of fossil fuels, in Greece (2006).

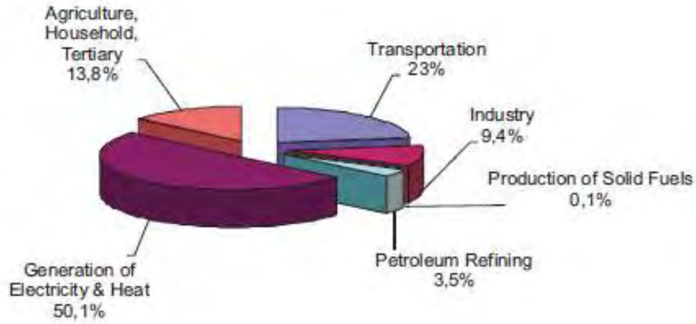



**Figure 2: Contribution to CO2 emissions by sector.**

Energy crisis and environmental factors impose the need of organizing special Programs, in international level, which affair the adopting of renewable sources of energy technologies, in all sectors are concerned.

Renewable Sources of Energy (RES) are:

- practically inexhaustible,
- with great variety (solar, geothermal, wind, biomass, hydro e.g.) and
- can be used for generation of electricity, heat or as fuels in transportations

Technologies by (RES) are:

- energy efficient and
- environmentally safe.

In Figure 3, is shown the contribution of each source of energy to the energy consumption, in Greece (2006).

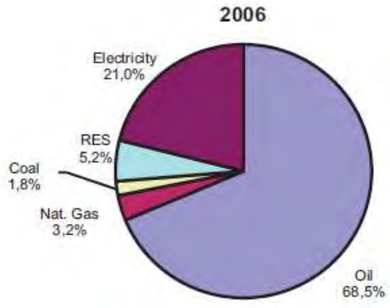



**Figure 3: Final energy consumption by fuel, in Greece (2006)**

The contribution of RES to national energy balance was about 5.3% in 2006, on level of total GIEC and about 18% on the level of primary energy production [1].

As far as solar technologies concerned, Greece is a country, which has the great advantage of landing in Southern Europe, which almost year-round (especially during summer months, when is occurred the energy peak demand of electrical power due to the operation of air-condition units) is exceptional considering the falling solar irradiation (Figure 43: Appendix A). Due to that reason, the adopting of solar technologies in Greece, is a very attractive concept.

Solar energy can provide thermal energy by energetic solar thermal systems and electricity by PV systems or other energetic solar plants. In last decades has been done a rapid advancement in the Science and in Technology by the development of systems with higher performance and reliability. The main goal is “zero” emissions of the greenhouse gases in the energy production and leakage of the harmful cooling fluids.

Although a large potential market exists for solar technology, existing solar systems are not directly economically competitive with conventional, mainly because of the high initial investment cost of solar systems and the still low prices of conventional fuels. Another disadvantage is the size of solar plants, competitive to conventional (for example: solar plants requires big areas of collectors or PV panels to be installed) and the fact that such systems are in great dependence by the hourly, daily and monthly fluctuations of solar radiation.

Hybrid Solar Plants, mainly in the industrial sector, if they are appropriate sized, could overcome the objections of pure solar systems, contribute to the direction of decreasing emissions of greenhouse gases and leakage of harmful fuels, do the operation of total (solar and conventional) system more efficient and avoid major problems in the electric supply when energy peak demand occurs. The basic concept of hybrids is the usage of two or more technologies (for economy reasons the applied technologies are limited to two) in order to cover all the energy requirements in the most profitable way, exploiting each time, the advantageous characteristics of both.

A dairy products Factory is a food industry, with high energy consumption for:

- the climate of production places in specific temperature and humidity conditions,
- carrying on the total processes of fluids (heating and refrigerating) and
- the operation of the rest mechanical equipment.

In particular, the energy needs in a dairy products factory are:

- heat: for thermal processes of fluids and heating spaces,
- cool: for freezing processes of fluids and air-conditioning of spaces and
- electrical energy: for the operation of the installed machinery.

An investment of a hybrid solar plant in order to satisfy the energy requirements of a dairy products factory seems to be a very attractive scenario in the field of RES development in Greece.

A hybrid solar plant, could cover the above mentioned energy requirements by the operation of a solar in cooperation with a conventional system.

For the appropriate design of a hybrid solar plant should be considered the available solar technologies in market, the factory's potential and other particularities (such as meteorological data of situated region, present industry conventional installation) etc.
In any case a detailed feasibility analysis must be attained, in order to evaluate the profitability of each investment.

## 1.2 Scope of the study

In this study, it is analyzed the viability of a hybrid Solar Plant in an existing dairy products factory, which is situated at the neighborhood of the Greek city of Volos.

So far, in the Agricultural Corporations Union of Volos (EVOL), all energy requirements, according to its form, heat or electrical, are covered by existing conventional, crude-oil combustion chamber and by Hellenic Electricity Supplier (Public Power Corporation S.A.).

The objective of the thesis is to access a feasibility study of the design, modeling and simulation of a prosperous solar plant comprised by a solar thermal, a solar cooling and a PV system which in combination with part of the existing conventional system will cover all the energy needs of the dairy products factory. Discussion on the effect of critical factors to the profitability of the plant operation and the viability of the investment would come up to interesting concluding remarks.
Specifically, the feasibility study comprises by the following steps:

- selection of the most promising solar technology and system layout,
- simulation of system layout,
- sizing of its components, e.g. size of the solar collector field,
- calculation of efficiency and exploitation values like the coefficient of performance of the cooling system, the solar fraction of covering the loads by the solar thermal system,
- calculation of consumption figures (electricity, fuel),
- estimation of financial key figures and of the primary energy savings,
- optimization of the system studied.

Although the study refers to the specific dairy products Factory of Agricultural Corporations Union of Volos (EVOL), the developed simulation model can be applied for the analysis of similar projects, which are related with the design and modeling of energy systems in general. It should be mentioned that in any case all the proportions and particularities of another case project, related to the location, the desired solar technology, other special operation, financial parameters etc, could be considered and adapted in the model, by the Researchers or Engineers. Thus, it is expected that the methodology and the results of the thesis could be useful for similar feasibility studies in the future.

## 1.3 Principles and Considerations

Of great importance in the accession of this feasibility study is the definition of principles and considerations taken into account.

In particular, the study was carried out in regard with the followings:

- Data about the present state of the applied dairy products factory (the data were given by the Production Manager of EVOL)
- Available solar technologies in market [the source was mainly special links in web (all are reported in the Reference list and Links Appendix)].
- Relative available Literature [the source was special links in web (all are reported in the Reference list and Links Appendix)].
- Principles of Thermodynamics and Financial Analysis.
- Meteorological data for the evaluation of the energy loads or solar irradiation data provided by the official web-site of Hellenic Meteorological Service. These data (such as ambient temperature and relative humidity, solar radiation etc) are given in mean monthly values (mean-max, mean-min, mean-monthly etc.), referring to the location of the industry. They are statistical results of a research covering a time span of at least 30 years.
- Steady state analysis of operation.

## 1.4 Description of Application Factory – Existing Plant

The Agricultural Corporations Union of Volos (EVOL) is situated at A΄ Industrial Area of Volos in Greece (Longitude of 22.884° and Latitude of 39.374°).

A photomap of the application factory is given in Figure 4.

**Figure 4: Photomap of Agricultural Corporations Union of Volos (EVOL) - Source: Meoronorm 6. online)**

The factory comprises by the following buildings:

- Building A: main production building, app. 3.070m2 (in the middle of photomap – figure 4). A ground plan of Building A is given in Figure 46.
- Building B: office-building, app. 400m2 (in the left hand side of photomap – figure 4).
- Building C: stowage building, app. 460m2 (down and right side of photomap – figure 4).
- Building D: subside-building for boilers etc, app. 170m2 (up and right hand side of photomap – figure 4).

In Figure 45 is presented a general lay-out of the dairy products factory.

In the factory, are working 130 employees. It operates by two (2) staffs per day, eight (8) hours per staff and six (6) days per week. So, if we assume an average of 25 days per month, then the yearly working days of dairy products factory will be 300 and the monthly and yearly working hours will be 400 and 4800 respectively.

The installed electric power (electromechanical machinery) is 706kW.

Nowadays, the quantity of processed milk per day is 25tons or 625tons per month. It produces pasteurized full cream and skim milk, in bottles and in one-way containers, Greek cheese ("Feta ") with whey and yoghurt.

# CHAPTER 2:

# FIELD OF STUDY

A feasibility analysis of the installation of a hybrid solar system for a dairy products factory is a very demanding and complicated task in all stages of the study (design, modelling, simulation and optimization). Such systems are strongly depended upon several design, climatic, modelling and economic parameters which are not facile to be determined and in many cases is almost impossible to be evaluated.

The main difficulties of the study are:

- The way the total plant in a feasibility study is properly designed, sized, modelled and optimized is depending upon the initial selection of the applied technology. A hybrid solar plant should cover all energy requirements in the factory by the installation of different energy systems (for heating, cooling and electricity generation). Each energy system combines the operation of two different technologies: a solar assisted one and the existing conventional one. Besides, the rapid development of Science and Technology in the field of Solar Systems has introduced nowadays in the market, a number of possible applying structures, technologies and equipment, with different technical and financial characteristics. The selection of the appropriate technology is not obvious and there is no a unique safe decision methodology available. Many different technologies and plants are also impossible to be compared in the framework of a unique study.
- The modelling and simulation of the performance or viability of a hybrid or pure solar system, is indeed too complex, because of the fluctuations of climatic parameters (between different regions and time of the year). The analysis could be carried-out after certain considerations (e.g. steady state operation) and the adoption of several empirical or semi-empirical relations in the simulation model.
- The design, the structure and the size of a solar system are depended also on the total energy requirements or other particularities (type of installation, environmental limitations, valid prescriptions etc) of the studied application. The total energy loads which should be covered are usually fluctuant with time quantities and it is necessary to be evaluated.

The good knowledge of the available solar technologies in nowadays market, the deep literature study of relative methodologies, results and resolutions of many scientific researches and also the usage of the appropriate software tools, can certainly assist in carrying out the present analysis.

## 2.1 Available Solar technologies

### 2.1.1 Solar Thermal Systems – Solar Collectors

A solar thermal system is composed by:

- Solar collectors area (collects thermal energy by solar radiation)
- An exchanger, for supplying the collector with relative constant temperature water (inlet), in order to retain the collector's performance in acceptable level (this unit can be missed if the energy requirements are high and stream).
- A heat storage tank, for continually water supplying in relative constant output temperature.

Typical installation of a solar thermal system is shown in Figure 5.

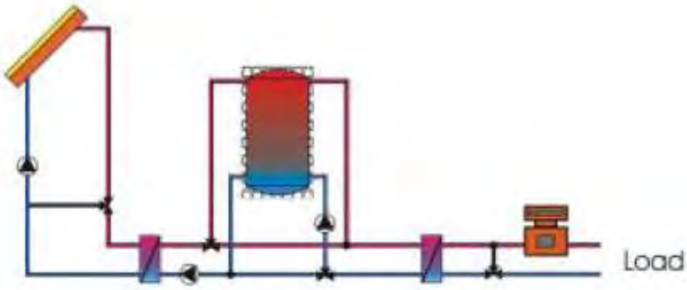



**Figure 5: Typical installation of a solar thermal system [3]**

The basic categories of solar collectors (Table 2), available in market, according to operation principle are:

- ✓ Solar air collector (SAC)
- ✓ Flat-plate collector (FPC - Figure 7
- ✓ Stationery parabolic compound collector (CPC)
- ✓ Vacuum tube collector (EHP, EDF, SYC, ETC)

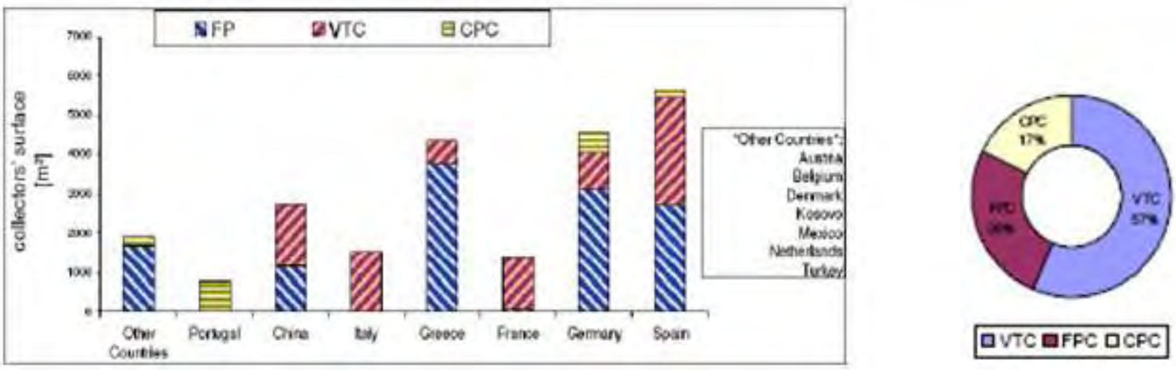



**Figure 6: Worldwide use of different type of solar collectors**

The technology with Vacuum Tube Collectors is the most represented with 57% of the projects (Figure 6). It should be noted that for one project, two technologies could be used at the same time e.g. FPC and VTC. But most generally, only one technology is studied and preferred.

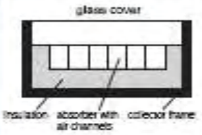
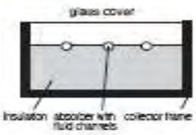
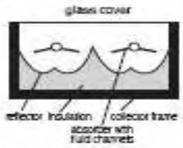
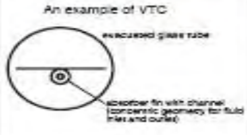
| Collector type | Solar air collector | Flat-plate collector | Stationary parabolic compound collector | Vacuum tube collector (VTC) |
|---|---|---|---|---|
| Short cut | SAC | FPC | CPC | EHP, EDF, SYC: ETC |
| |  |  |  |  |
| |   |   |   |   |
| Principle | Direct heating of air | Heating of a liquid (water, water-glycol) | Heating of a liquid (water, water-glycol); radiation concentration without tracking | Evacuated glass tube for reduction of thermal losses<br>EHP: evacuated tube with heat pipe<br>EDF: evacuated tube with direct flow<br>SYC: Sydney-type evacuated tube with concentrator reflector |
| Main application area | Pre-heating of ventilation air | Domestic hot water preparation | Domestic and industrial hot water preparation | Domestic and industrial hot water preparation |
| Prevalent application in solar assisted air conditioning | Open cooling systems, e.g. desiccant cooling systems | Desiccant cooling systems, Thermally driven chillers (single-stage) with selective absorbers | Thermally driven chillers (single-stage) | Thermally driven chillers (single-stage)<br>Thermally driven chillers (double-stage): SYC |

**Table 2: Overview of basic Solar Collectors types [3].**

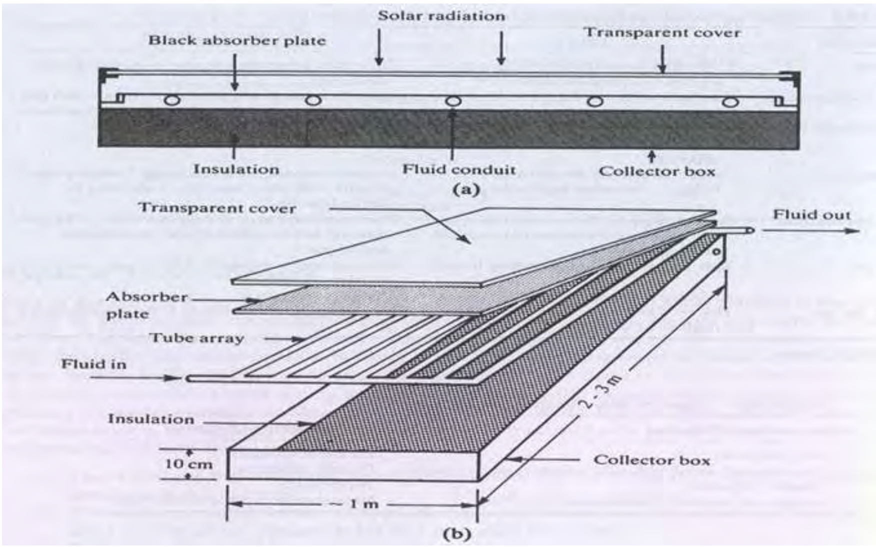



**Figure 7: Structure of Flat Plate Solar Collector with Selective Surface**

The type of solar collectors is selected, considering either the driven temperature which is required by the application and the installation cost indeed. The difference in operation of the solar collectors which applied to a solar assisted cooling system, compared to a solar thermal system for hot water production, is the high temperature level, at which the useful heat has to be provided.

For thermally driven chillers, the driving temperature is mainly above 80°C, lowest values are 60°C. For desiccant cooling systems, the driving temperature is above 55°C up to 90°C. Flat-plate collectors with preferable surface, can source single-stage chillers and DEC-systems with lower cost than vacuum or parabolic collectors. The mean driven temperature (output temperature of thermal solar system) is reaching up to 90ºC, in regions with high solar radiation. Consequently, standard flat-plate collectors and solar air collectors may be implemented with most benefit in solar assisted desiccant systems. In configurations using an adsorption chiller or a single-effect absorption chiller, the use of selectively coated flat plate collectors is limited to areas with high irradiation availability. For other areas and for chillers requiring higher driving temperatures, high efficient collectors are to be implemented, e.g. evacuated tube collectors.

### 2.1.2 Solar Cooling Systems

The heat source to the cooling system is provided by solar thermal plant in combination with the conventional thermal system in hybrids.

There are two basic categories in solar cooling systems:

- ✓ closed systems (thermally driven chillers),
- ✓ open systems (DEC-systems - the "refrigerant" is always water, since it is in direct contact with the atmosphere).

There are many variations within these types: hot or cold side energy storage, continuous or intermittent process, different types of collectors, various operating temperature ranges, different control concepts etc.

In Table 3, is shown an overview of the most common technologies in solar cooling.

The first criterion to classify the different projects is the technology: absorption, adsorption and DEC.

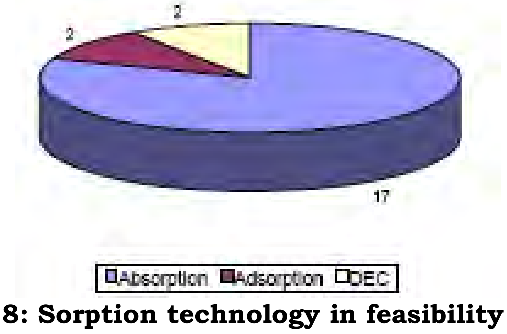



**Figure 8: Sorption technology in feasibility studies**

The principal technology is absorption with more than 80% of the feasibility studies (Figure 8). This leading position is well in accordance with the existing solar cooling systems in Europe (Figure 9): the proportion reaches 60%. The explanation of such dominating position is mainly due to the majority of centralized cooling distribution systems made of chilled water instead of cold air, especially in Mediterranean countries.

| Method | Closed cycle | | Open cycle | |
|---|---|---|---|---|
| Refrigerant cycle | Closed refrigerant cycle | | Refrigerant (water) is in contact with the atmosphere | |
| Principle | Chilled water | | Dehumidification of air and evaporative cooling | |
| Phase of sorbent | solid | liquid | solid | liquid |
| |  |  |  |  |
| Typical material pairs | water - silica gel | water - lithium bromide ammonia - water | water - silica gel, water - lithium chloride | water - calcium chloride, water - lithium chloride |
| Market available technology | Adsorption chiller | Absorption chiller | Desiccant cooling | Close to market introduction |
| Typical cooling capacity (kW cold) | 50 – 430 kW | 15 kW – 5 MW | 20 kW – 350 kW (per module) | |
| Typical COP | 0.5 – 0.7 | 0.6 – 0.75 (single effect) | 0.5 – >1 | > 1 |
| Driving temperature | 60 – 90 °C | 80 – 110 °C | 45 – 95 °C | 45 – 70 °C |
| Solar collectors | Vacuum tubes, flat plate collectors | Vacuum tubes | Flat plate collectors, solar air collectors | Flat plate collectors, solar air collectors |

**Table 3: Overview of solar cooling systems [4]**

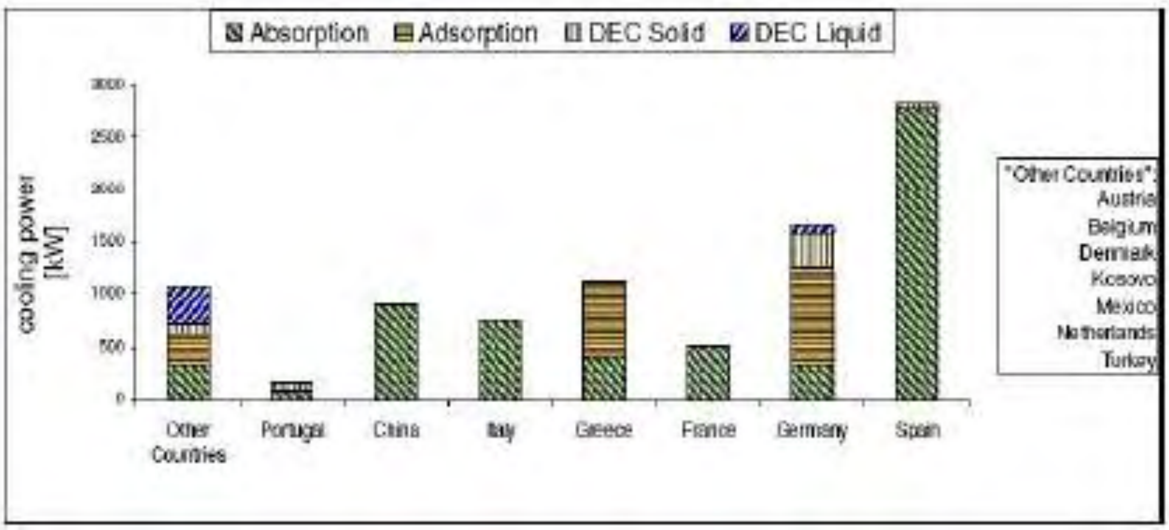



**Figure 9: World-wide use of different solar cooling technologies [21]**

Between adsorption chillers, desiccant coolers and the absorption systems the later has the highest market penetration. The market share of adsorption systems is significantly lower. The desiccant cooling technique has the advantage of the lowest driving temperatures and therefore has a large potential for market penetration [21].

#### ***2.1.2.1 Closed systems - Thermally driven chillers***

General principle of the operation of thermally driven chillers (Figure 10) is the consumption of energy to transfer heat from a source at a low temperature to a sink at a higher temperature:

$Q_{low}$ is the heat rejected from the chilled water in the evaporator of the chiller (chilling power), $Q_{high}$ is the required heat in the generation part to drive the process (delivered either by the solar system or by backup heat sources), and the amount of $Q_{intermeiate}$, the sum of $Q_{low}$ and $Q_{high}$, has to be removed at a medium temperature level. For the heat removal, in most cases, a wet-cooling tower is used.

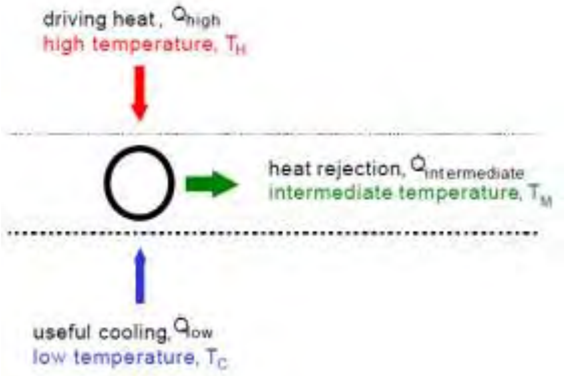



**Figure 10: Schematic diagram of energy flows in a thermally driven machine**

The efficiency of a thermally driven chiller is described by the thermal Coefficient Of Performance (COP), defined as the fraction of heat rejected from the chilled water cycle ('delivered cold') and the required driving heat, i.e. $COP_{thermal} = Q_{low} / Q_{high}$. This is different to the $COP_{conv}$ of a conventional electrically driven compression chiller, defined by $COP_{conv} = Q_{cold} / E_{electric}$, with $E_{electric}$ representing the electricity consumption of the chiller.

Thermally driven chillers may be characterized by three temperature levels:

- a high temperature level at which the driving temperature of the process is provided;
- a low temperature level at which the chilling process is operated;
- a medium temperature level at which both the heat rejected from the chilled water cycle and the driving heat have to be removed.

*Absorption chillers*

Absorption chillers are the most distributed chillers worldwide. Typical chilling capacities of absorption chillers are several hundred kW and only few absorption chillers with capacities below 50 kW are available.

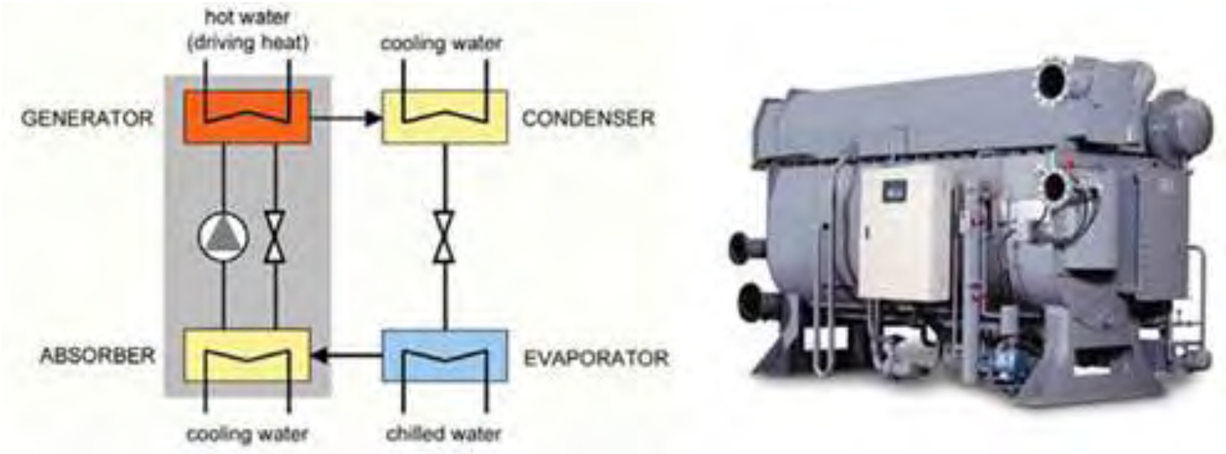



**Figure 11: Absorption chiller**

The main components of an absorption chiller are shown in Figure 11. Absorption systems use solar energy from thermal collectors as the driving force. A thermal compression of the refrigerant is achieved by using a liquid refrigerant/sorbent solution and a heat source, thereby replacing the electric power consumption of a mechanical compressor. Absorption refrigerators use two working substances, a refrigerant and an absorbent for that refrigerant. The cooling effect is based on the evaporation of the refrigerant (water) in the evaporator at very low pressures.

For chilled water above 0°C, as it is used in air conditioning, typically a liquid $H_2O$/LiBr solution is applied with water as refrigerant. At present the market of sorption refrigeration systems is dominated by LiBr–H2O systems, which are normally used for air-conditioning applications. In the operation of an $H_2O$/LiBr absorption chiller, a crystallisation of the solution has to be avoided by an internal control of the heat rejection temperature in the machine.

The required heat source temperature is usually above 80°C for single-effect machines and the COP is in the range from 0.6 to 0.8. Double-effect machines with two generator stages require driving temperature of above 140°C, but the COP's may achieve values up to 1.2. There are available absorber chillers with one, two or three generator stages, with different operation and performance characteristics.

- Single-effect: Gen. temperature: 80 – 100 C, COP: 0.6 – 0.8
- Double-effect: Gen. Temp 100 – 160 C, COP 1.0 – 1.2
- Triple-effect: Gen. Temp 160 – 240 C, COP ABOUT 1.7

In the last decade, research has been focusing on solar-driven absorption refrigeration cycles with unconventional fluids, exhibiting improved behavior. One class is ammonia–salt solutions, such as ammonia–lithium nitrate (NH3–LiNO3) and ammonia–sodium thiocyanate (NH3–NaSCN). These systems provide certain advantages such as: lower generator temperatures (which T. Tsoutsos et al. / Applied Thermal Engineering 23 (2003) 1427–1439 1429 ARTICLE IN PRESS allow operation with simple flat-plate collectors), lower evaporation temperatures in comparison with H2O/LiBr systems and higher coefficient of performance in comparison with NH3/H2O systems [21].

*Adsorption Chillers*

Here, instead of a liquid solution, solid sorption materials are applied. Market available systems use water as refrigerant and silica gel as sorbent.

The machines consist of two sorbent compartments (denoted as 1 and 2 in the figure 12), one evaporator and one condenser. While the sorbent in the first compartment is regenerated using hot water from the external heat source, the sorbent in the compartment 2 (adsorber) adsorbs the water vapour entering from the evaporator; this compartment has to be cooled in order to enable a continuous adsorption. The water in the evaporator is transferred into the gas phase being heated from the external water cycle; here actually the useful cooling is produced. If the cooling capacity reduces to a certain value due to the loading of the sorbent in the adsorber, the chambers are switched over in their function.

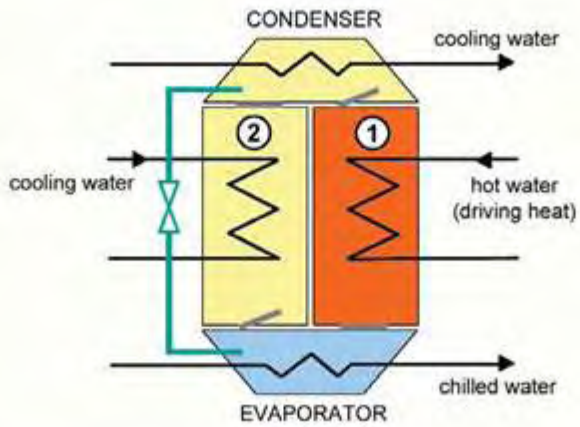



**Figure 12: Adsorption chiller – main components**

To date, only a few Asian manufacturers produce adsorption chillers. Under typical operation conditions with a temperature of the driving heat of about 80°C, the systems achieve a COP of about 0.6, but operation is possible even at heat source temperatures of approx. 60°C. The capacity of the chillers ranges from 50 kW to 500 kW chilling power.

The combination of an adsorption chiller with solar collectors offers a technically simple and energy saving solution, especially in Southern European regions such as Greece. The simple mechanical construction of adsorption chillers and their expected robustness is an advantage. No danger of crystallization is given and thus no limitation in the heat rejection temperatures is existed. An internal solution pump does not exist and hence only a minimum of electricity is consumed. A disadvantage is the comparatively large volume and weight. Furthermore, due to the small number of produced items, the price of adsorption chillers is currently high.

### *2.1.2.2 Open systems*

*Desiccant cooling systems (DEC)*

Desiccant cooling systems (Figure 13) are basically open cycle systems, using water as refrigerant in direct contact with air.

The thermally driven cooling cycle is a combination of evaporative cooling with air dehumidification by a desiccant, i.e. a hygroscopic material. For this purpose, liquid or solid materials can be employed. The term 'open' is used to indicate that the refrigerant is discarded from the system after providing the cooling effect and new refrigerant is supplied in its place in an open-ended loop. Therefore only water is possible as refrigerant since a direct contact to the atmosphere exists. The common technology applied today uses rotating desiccant wheels, equipped either with silica gel or lithium-chloride as sorption material.

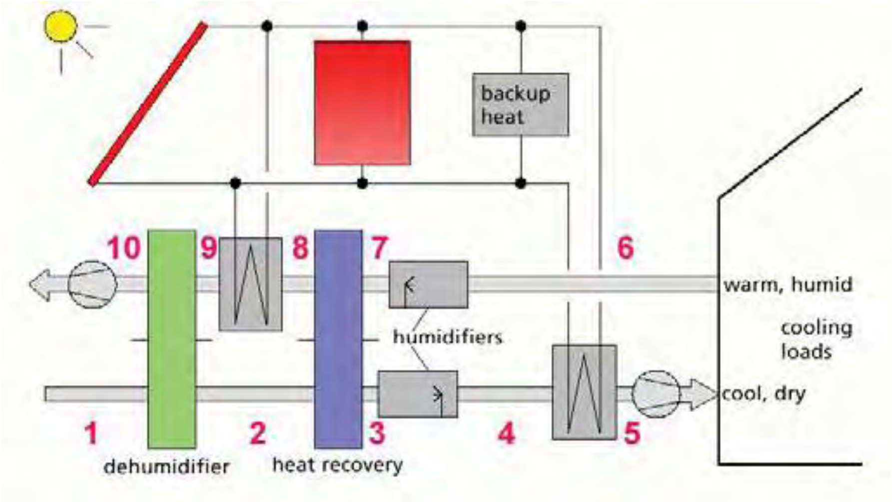



**Figure 13: DEC system with rotating wheels – main components**

Flat-plate solar thermal collectors can be normally applied as heating system in solar assisted desiccant cooling systems. Special design of the desiccant cycle is needed in case of extreme outdoor conditions such as e.g. coastal areas of the Mediterranean region. Here, due to the high humidity of ambient air, a standard configuration of the desiccant cooling cycle is not able to reduce the humidity down to a level that is low enough to employ direct evaporative cooling.

*Open systems - Liquid Desiccant cooling systems*

A new development, close to market introduction, are desiccant cooling systems using a liquid Water/Lithium-Chloride solution as sorption material. This type of systems shows several advantages like higher air dehumidification at the same driving temperature range than solid desiccant cooling systems, and the possibility of high energy storage by storing the concentrated solution. This technology is a promising future option for a further increase in exploitation of solar thermal systems for air conditioning.

### 2.1.3 Photovoltaic Systems

Photovoltaic (PV) is a method of generating electrical power by converting solar radiation into direct current electricity using semiconductors that exhibit the photovoltaic effect. The photovoltaic effect refers to photons of light knocking electrons into a higher state of energy to create electricity. The term photovoltaic denotes the unbiased operating mode of a photodiode in which current through the device is entirely due to the transuded light energy. Virtually all photovoltaic devices are some type of photodiode.

Photovoltaic power generation employs solar panels comprising a number of cells containing a photovoltaic material. Materials presently used for photovoltaic (Figure 14) include monocrystalline silicon, polycrystalline silicon, amorphous silicon, cadmium telluride, and copper indium selenide/sulfide.

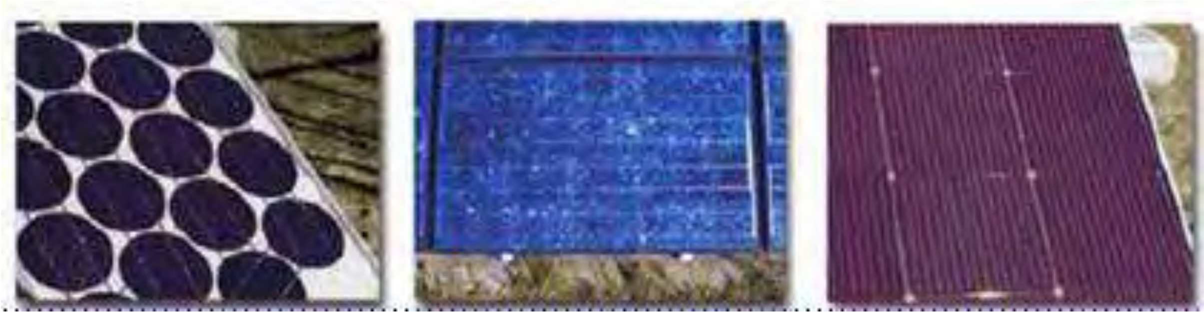


**Figure 14: Types of photovoltaic panels (monocrystalline silicon, polycrystalline silicon, amorphous silicon)**

A photovoltaic installation typically includes an array of solar panels, an inverter, batteries and interconnection wiring.

Solar cells produce direct current electricity from sun light, which can be used to power equipment or to recharge a battery. Photovoltaic systems are used for either on-grid or off-grid applications, and for solar panels on spacecraft. They are often associated with buildings: either integrated into them, mounted on them or mounted nearby on the ground. Building-integrated Photovoltaics (BIPV) are increasingly incorporated into new domestic and industrial buildings as a principal or ancillary source of electrical power. Typically, an array is incorporated into the roof or walls of a building.

For best performance, terrestrial PV systems aim to maximize the time they face the sun. Solar trackers aim to achieve this by moving PV panels to follow the sun. The increase can be by as much as 20% in winter and by as much as 50% in summer. Static mounted systems can be optimized by analysis of the Sun path. Panels are often set to latitude tilt, an angle equal to the latitude, but performance can be improved by adjusting the angle for summer or winter.

Due to the growing demand for renewable energy sources, the manufacture of solar cells and photovoltaic arrays has advanced considerably in recent years. As of 2010, solar photovoltaics generates electricity in more than 100 countries and, while yet comprising a tiny fraction of the 4800 GW total global power-generating capacity from all sources, is the fastest growing power-generation technology in the world. Between 2004 and 2009, grid-connected PV capacity increased at an annual average rate of 60 percent, to some 21 GW. Such installations may be ground-mounted (and sometimes integrated with farming and grazing) or built into the roof or walls of a building, known as Building Integrated Photovoltaics or BIPV for short. Off-grid PV accounts for an additional 3–4 GW.

## 2.2 Literature Overview

A lot of efforts have been done since today, by several authors in the study of solar assisted systems. Some of them are of great interest and their scientific approach related to the methodology, design, modeling, simulation, and their results have been strongly considered in the present work. Thus, it is worth to mention some representative works.

L.W.Butz, W.A.Beckman and J.A.Duffle (1974) in “Simulation of a solar heating and cooling system” [27] presented thermal and economic analysis of a solar heated and air conditioned house in the Albuquerque climate. The system was consisted by the following components: water heating collector, a water storage unit, a service hot water facility, a lithium bromide-water air conditioner (with cooling tower), an auxiliary energy source, and associated controls. The analysis of the thermal performance indicated the dependence of output on collector area (considered as the primary design variable) and shows, the manner in which annual system efficiency decreases as collector area increases. Based on the computed thermal performance and cost estimates were also shown variations in annual cost as functions of collector area and costs of collector and fuel.

S. R. Swanson and R. F. Boehm (1976) in “Calculation of long term solar collector heating system performance” [26] developed a simplified technique to predict yearly system performance. This simplified method is based on a computation of the system performance for a single "average" day of each month, modifying this result with a correction factor, and summing the results over each month of the heating season. The result is that the system performance can be calculated easily, by hand or with trivial computer cost. A comparison with detailed numerical analyses in the literature for systems in the cities of Phoenix, Charleston, Madison and Boston shows excellent agreement. Results are also presented for systems in Salt Lake City.

D.S.Ward, W.S.Duff, J.C.Ward and G.O.Lof (1978) in “Integration of evaluated tubular solar collectors with Lithium-Bromide absorption cooling systems” [25] by surrounding the absorber-heat exchanger component of a solar collector with a glass-enclosed evacuated space and by providing the absorber with a selective surface, solar collectors can operate at efficiencies exceeding 50 per cent under conditions of $\Delta T/Hr$= 75°C $m^2/kW$ ($\Delta T$ = collector fluid inlet temperature minus ambient temperature, Hr = incident solar radiation on a tilted surface). The high performance of these evacuated tubular collectors thus provides the required high temperature inputs (70 – 88°C) of lithium bromide absorption cooling units, while maintaining high collector efficiency. This paper deals with the performance and analysis of two types of evacuated tubular solar collectors integrated with the two distinct solar heating and cooling systems installed on CSU Solar Houses I and III (USA).

E. Tasdemiroglu and M. Awad (1989) in “Mathematical methods for the optimization of solar collector area in a solar heating system” [24] worked on the developing of three different optimization of solar collector area methods, namely, Newton's, Secant and Regular-falsi for a variety of changes in values for design, climatic and economic parameters in a solar heating system. The ability to use these methods is under discussion from the computational point of view.

Soteris Kalogirou (2002) in “The potential of solar industrial process heat applications” [23] based on TRNSYS simulations, gave an estimation of the system efficiency of solar process heat plants (the temperature requirements of solar industrial process heat applications range from 60°C to 260°C) operating in the Mediterranean climate for the different collector technologies. Five collector types have been considered in this study varying from the simple stationary flat-plate to movable parabolic trough ones. The characteristics of medium to medium-high temperature solar collectors are given and an overview of efficiency and cost of existing technologies is presented. The annual energy gains of such systems are from 550 to 1100 kWh/m2 a. The resulting energy costs obtained for solar heat are from 0.015 to 0.028 C£/kWh depending on the collector type applied.

Theocharis Tsoutsos, Joanna Anagnostou, Colin Pritchard, Michalis Karagiorgas and Dimosthenis Agoris (2003) in “Solar cooling technologies in Greece. An economic viability analysis” [21] carried out an economic evaluation of two types of solar cooling systems an absorption and an adsorption system. Economic analyses of the SCS indicate that these systems will not be competitive compared with standard cooling systems at present energy prices, in Greece. The technology of solar cooling is not presently economically feasible without subsidy, mainly because of its high investment cost. However energy savings (in electricity or gas) may be realized by the integration of SE with cooling systems, so these SCS require lower costs of installation. The analysis shows that SCS are better suited to replacing conventional air-conditioners in remote areas, where there is no connection with the electricity grid and where the conventional fuel used is gas. There is a strong need both for some kind of investment incentive and also for energy tax that would help to reflect the full environmental costs of conventional fuels.

Jay Burch, Craig Christensen, Jim Salasovich and Jeff Thornton (2004) in “Simulation of an unglazed collector system for domestic hot water and space heating and cooling” [22] studied a plant of an unglazed collector system supplying domestic hot water, space heating, and space cooling loads in region of Albuquerque, NW -USA. The system modelled using unglazed collector test results, and they simulated the variation of savings with collector area, storage volume, heat exchanger size, and wind for the specific climate conditions. The over the storage-to-collector ratio range of 40–640 l/m$^2$ collector, annual savings varies only ±15%. It was proved that cooling is sensitive to heat exchanger size, and heating is sensitive to wind velocity.

G. Vokas, N. Christandonis and F. Skittides (2006) in "Hybrid photovoltaic–thermal systems for domestic heating and cooling—A theoretical approach" [20] researched the thermal efficiency of Hybrid photovoltaic–thermal (PV/T) collectors consist of a thermal collector in which a photovoltaic laminate is attached as a thermal absorber. This happens due to the decrease of the photovoltaic temperature. The theoretical study of a photovoltaic–thermal system for domestic heating and cooling has led to the result that the system can cover a remarkable percentage of the domestic heating and cooling demands. The solar coverage percentage of the system is notably affected by the variation of geographical region as well as to different total surface areas of the system.

M. Augustus Leon and S. Kumar (2007) in "Mathematical modelling and thermal performance analysis of unglazed transpired solar collectors" [18] presented the details of a mathematical model for Unglazed transpired collectors UTC using heat transfer expressions for the collector components, and empirical relations for estimating the various heat transfer coefficients. It predicts the thermal performance of unglazed transpired solar collectors over a wide range of design and operating conditions. Transpired collectors are a potential replacement for glazed flat plate collectors. Results of the model were analysed to predict the effects of key parameters on the performance of a UTC for a delivery air temperature of 45–55 °C for drying applications. The parametric studies were carried out by varying the porosity, airflow rate, solar radiation, and solar absorptivity/thermal emissivity, and finding their influence on collector efficiency, heat exchange effectiveness, air temperature rise and useful heat delivered. Results indicate promising thermal performance of UTC in this temperature band, offering itself as an attractive alternate to glazed solar collectors for drying of food products. The results of the model have been used to develop nomograms, which can be a valuable tool for a collector designer in optimising the design and thermal performance of UTC and it also enables the prediction of the absolute thermal performance of a UTC under a given set of conditions.

Huseyin Gunerhan and Arif Hepbasli (2007) in "Exergetic modelling and performance evaluation of solar water heating systems for building applications" [19] investigated a Solar water heating (SWH) system consists of mainly three parts, namely a flat plate solar collector, a heat exchanger (storage tank) and a circulating pump. The system performance is evaluated based on the experimental data of the Izmir province, in Turkey. In this paper, were possible the modelling of SWH systems using exergy analysis (second law) method, the investigation of the effect of varying water inlet temperature to the collector on the exergy efficiencies of the SWH system components, the study of some thermodynamic parameters (fuel depletion ratio, relative irreversibility, productivity lack and exergetic factor) and exergetic improvement potential and the presenting of an exergy efficiency curve similar to the thermal efficiency curve for solar collectors. Exergy destructions (or irreversibilities) as well as exergy efficiency relations are determined for each of the SWH system components and the whole system. Exergy efficiency values on a product/fuel basis are found to range between from 2.02 to 3.37%, and 3.27 to 4.39% at a dead (reference) state temperature of 32.77 8C, which is an average of ambient temperatures at eight test runs from 1.10 to 3.35

p.m., for the solar collector and entire SWH system, respectively. An exergy efficiency correlation for the solar collector studied was also presented to determine its exergetic performance.

M. Clausse, K.C.A. Alam and F. Meunier (2008) in “Residential air conditioning and heating by means of enhanced solar collectors coupled to an adsorption system” [13] and A. El Fadar, A. Mimet and M. Perez-Garcıa (2008) in “Modelling and performance study of a continuous adsorption refrigeration system driven by parabolic trough solar collector” [14], worked with the modelling and optimizing of the performance of solar systems in which solar parabolic trough collector has been introduced to the adsorption refrigeration cycles, with really interesting results concerning the structure and the performance of the system considered.

Giampaolo Manfrida and Vincent Gerard (2008) in “Maximum Exergy Control of a Solar Thermal Plant Equipped with Direct Steam Collectors” [15] studied the performance of solar thermal power plants via the radiation intensity. The advantage of introducing direct-steam solar collectors with respect to the use of a separate heat transfer fluid in the primary circuit was demonstrated by the model simulation, which predicted a performance improvement - compared to traditional control laws - ranging from 10 to 20% (depending on the reference month).

H. Zhai, Y.J. Dai, J.Y. Wu and R.Z. Wang (2008) in “Energy and exergy analyses on a novel hybrid solar heating, cooling and power generation system for remote areas” [16] proposed and investigated a small scale hybrid solar heating, chilling and power generation system, including parabolic trough solar collector with cavity receiver, a helical screw expander and silica gel–water adsorption chiller etc, located in north western region of China. The system has the merits of effecting the power generation cycle at lower temperature level with solar energy more efficiently and can provide both thermal energy and power for remote off-grid regions. It is found that both the main energy and exergy loss take place at the parabolic trough collector, amount to 36.2% and 70.4%, respectively. Also found is that the studied system can have a higher solar energy conversion efficiency than the conventional solar thermal power generation system alone. The energy efficiency can be increased to 58.0% from 10.2%, and the exergy efficiency can be increased to 15.2% from 12.5%. Moreover, an economical analysis in terms of cost and payback period (PP) has been carried out. The study reveals that the PP of the proposed system is about 18 years under present energy price conditions.

Tiago Mateus and Armando C. Oliveira (2008) in “Energy and economic analysis of an integrated solar absorption cooling and heating system in different building types and climates” [17] researched the potential of integrated solar absorption cooling and heating systems for building applications (by the use of TRNSYS software). Different building types were considered: residential, office and hotel. The TRNSYS models are able to run for a whole year (365 days), according to control rules (self-deciding whether to operate in heating or cooling modes), and with the possibility of combining cooling, heating and DHW applications. Three different locations and climates were considered: Berlin (Germany), Lisbon (Portugal), and Rome (Italy). The different local costs for energy (gas, electricity and water) were taken into account. Savings in $CO_2$ emissions were also assessed. An optimization of solar collector size and other system parameters was also analysed. Both energy and economic results were presented for all cases.

J.V.C. Vargas, J.C. Ordonez, E. Dilay and J.A.R. Parise in their research “Modeling, simulation and optimization of a solar collector driven water heating and absorption cooling plant” (2009), [10] analysing a cogeneration system to simultaneously produce heating (hot water heat exchanger) and cooling (absorption refrigerator system), managed to develop an interesting mathematical model, which combines fundamental and empirical correlations, and principles of classical thermodynamics, mass and heat transfer. Performing a numerically simulation of the system transient and steady state response under different operating and design conditions and a system optimization for maximum performance (or minimum energy destruction), their results are useful in evolving of quantitative criteria which will assist designers in the process of minimizing plant size (in particular the matching of the solar energy converter (concentrator dish/collector) and the hot side heat exchanger).

A. Baldini, G. Manfrida and D. Tempesti (2009) in “Model of a Solar Collector/Storage System for Industrial Thermal Applications” [11], developed a model for the thermodynamic analysis of a non-stationary solar thermal system for dairy products factory’s needs. The main system components were a parabolic tube collector and a steam accumulator, which provides heat for industrial processes. Exergy analysis leads to the identification of potentials for performance improvements and the rate of exergy destruction was possible to be calculated over a daily operation cycle. The relatively simple thermodynamic model is a useful tool for sizing the solar field and the steam accumulator for a given industrial application using medium/high temperature solar heat. The exergy analysis indicated that the largest exergy destructions took place in the solar collector, while the contribution of the steam accumulator was relatively small. The main parameter determining the system's exergetic efficiency is the collector temperature difference, which is directly linked to the specific flow rate across the collector.

T. Tsoutsos, E. Aloumpi, Z. Gkouskos and M. Karagiorgas (2009), in their research “Design of a solar absorption cooling system in a Greek hospital” [12], studied the performance and economic evaluation of a solar heating and cooling system of a hospital in Crete using the transient simulation program (TRNSYS). The objective of this study is to simulate a complete system comprised of a solar collector, a storage tank, a backup heat source, a water cooling tower and a LiBr-H2O absorption chiller. The exploitation of the results of the simulation provided the optimum sizing of the system, considering the financial and environmental benefits. The object of this paper is similar to that of the present thesis, so the methodology, economic relations and the results of this paper were considered with care.

## 2.3 Tools – Software

The present thesis was carried out by the use of the following available software tools for the receipt of the required official statistical data, all necessary evaluations and the design, simulation, modelling and optimization of the system.

The basic tools were:

- Engineering Equations Solver (EES) v.6, was used to simulate the hybrid solar plant, to evaluate the dairy products factory's energy consumption, to carry out financial analysis and system-optimization. In Appendix A (at the end of this study) is given a general introduction about this software. Analytical guide of the specific software is given in [34].
- Autocad-2005 was used to make all the necessary drawings of the plant.
- METEONORM 6 online, (provided free in web-site) was another useful tool for the receipt of necessary for the system simulation, meteorological, statistical data (such as ambient temperature, relative humidity, solar radiation etc – in mean monthly values) regarding the location of the dairy products factory.

# CHAPTER 3:

# DESIGN OF HYBRID SOLAR PLANT

The basic concept of the sustainable System is a solar plant which in combination with the existing energy sources should be able to cover all energy requirements of the application factory.

Beginning by the coverage of total cooling load, a solar assisted cooling technology can be adopted in combination with the existing electrical driven, mechanical compressor cooling system. A solar thermal plant (comprised by solar collectors, an exchanger and a hot water storage tank) should be installed with the existing conventional (crude-oil) thermal system, for covering the total thermal load and for driving the solar cooling plant. Finally, electricity supply for covering the electricity requirements and for driving mechanical compressor cooling system is provided by an on-Grid Photovoltaic plant.

## 3.1 Selection of the applicable technology

Designing the above described system, it should be selected the solar technology which is expected that can be applicable with good performance in as possible lower initial cost. The considerations for this decision are the exceptional solar climate conditions of the region of Volos, the available solar technologies in nowadays Greek market, study of above mentioned literature and all the needed factory's particularities as they are the limitation in available surface of solar panels installation and its energy requirements.

A thermally driven one-stage absorber chiller is selected as a promissory technology, considering the simplified decision scheme for solar assisted air conditioning, nowadays market availability, financial factors (initial cost) and the better performance of other solar systems for cooling (COP). In Figure 44 is presented, a simplified decision diagram for solar assisted air conditioning technologies.

The type of solar collectors is induced by the selected solar cooling technology. A good decision might be the flat plate collectors with selectable surface considering that they have lower installation cost relative to other types and that in a region with high solar radiation, they can drive efficiently the one-stage absorber chiller.

Electricity generation can be applied by an on-Grid Photovoltaic system comprised by polycrystalline silicon panels.

## 3.2 Layout of hybrid solar plant

The designed plant is then comprised of a flat-plate with selectable surface solar collector, a heat exchanger, a storage tank, a backup heat source (existing crude-oil burner), a LiBr-H2O absorption chiller in combination with the existing mechanical one, a water cooling tower and an on-grid connected PV-system. One possible of previously decided components layout is presented in the Hybrid Solar Plant's Flow Diagram shown in Figure 15.

The Flow Diagram of the system studied is shown in Figure 16 and the general layout of the corresponding Hybrid Solar Plant in Figure 15.

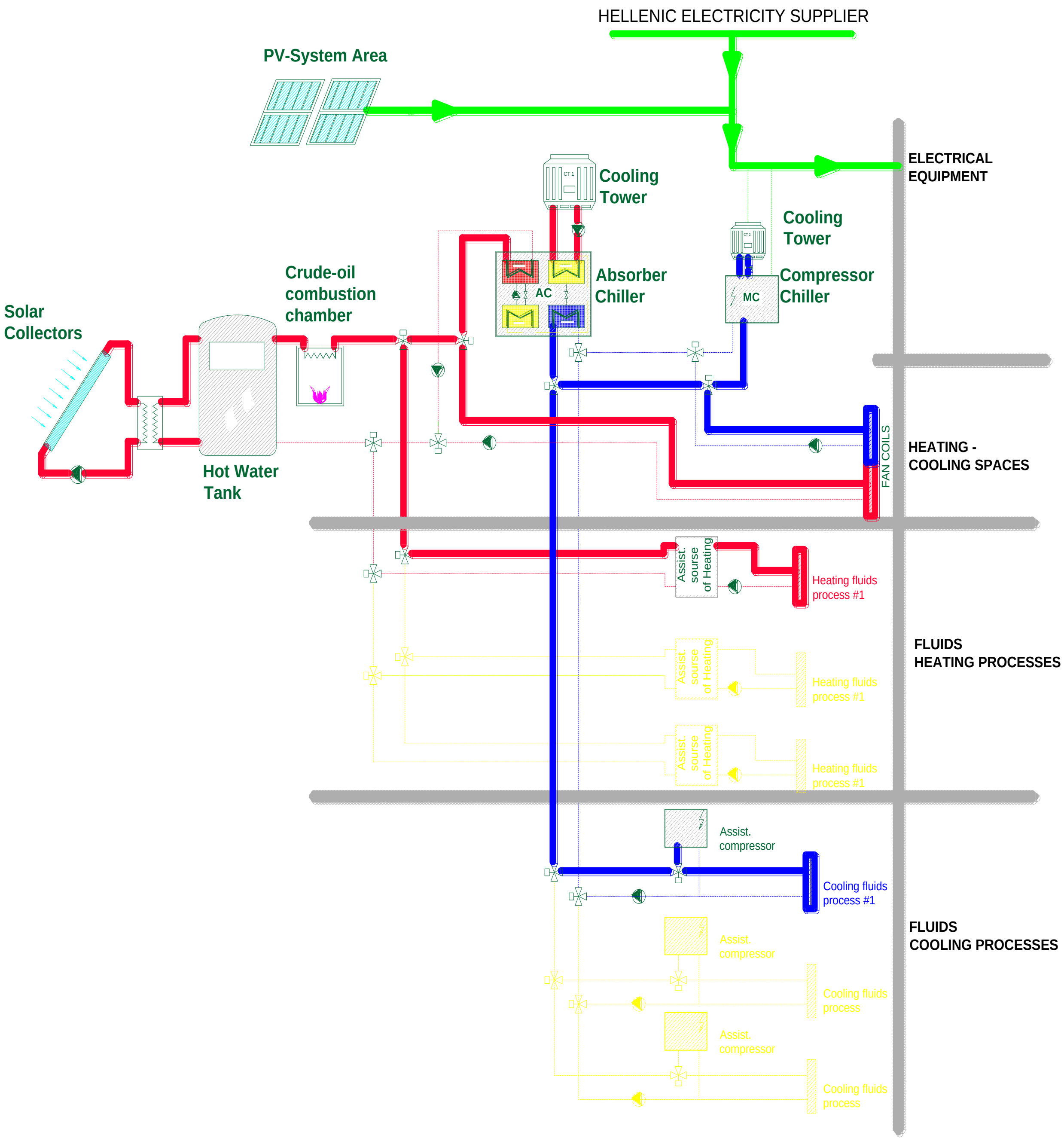



**Figure 15: Hybrid solar plant layout**

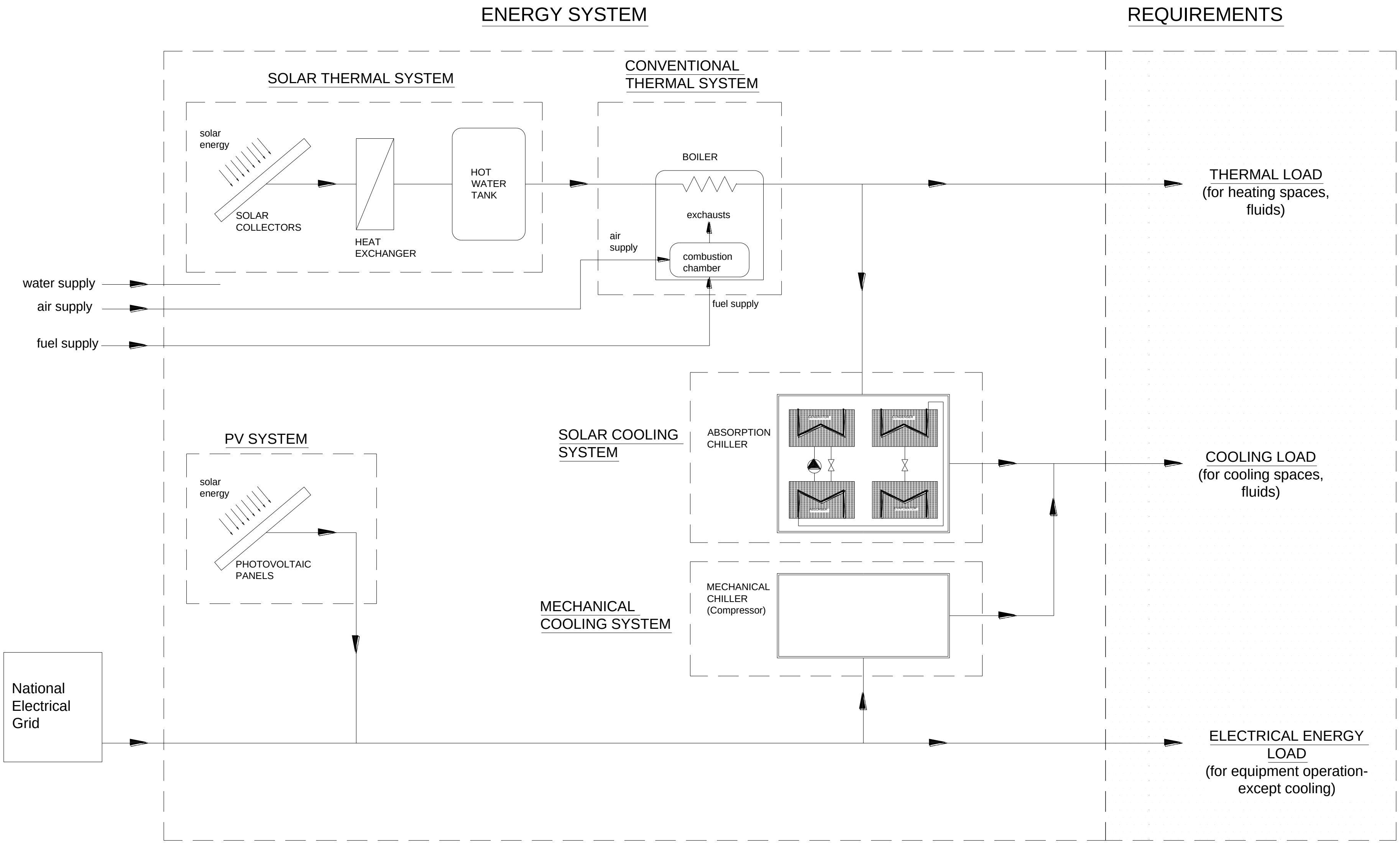



**Figure 16: Flow Diagram of the Hybrid Solar Plant**

# CHAPTER 4:

# MODELING AND SIMULATION

Solar plant performance was modelled in mean average values (for a typical month's day), considering steady state operation of the system, in order to overcome the objections of its strong dependency by the variability of meteorological data and the time.

## 4.1 Meteorological data

Greece is grouped in six climatologic zones (Figure 17) and all regions of the same zone has similar data of average, monthly, ambient temperatures, relative humidity or solar radiation which falls on a horizontal surface.


**Figure 17: Climatologic zones of Greece**

The application factory is located in an area of the 4th climatologic zone of Greece, and one can see these average values applying for temperatures in Figure 18.

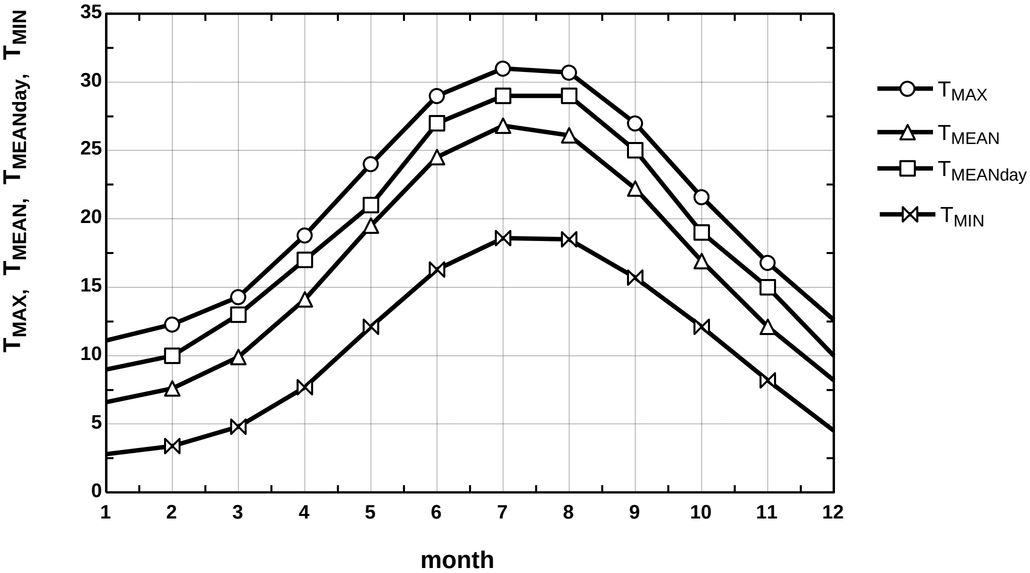



**Figure 18: Temperature (monthly means of minimum, maximum, mean and mean-day) of dairy products factory's region**

The total mean monthly irradiation H [MJ/m²-mo], which falls to a horizontal surface, is given by tables and it depends to different climatic zone (Table 4 - for Greece).

| | ZONE 1 | ZONE 2 | ZONE 3 | ZONE 4 | ZONE 5 | ZONE 6 |
|---|---|---|---|---|---|---|
| **JAN** | 230 | 230 | 220 | 194 | 169 | 169 |
| **FEB** | 277 | 274 | 259 | 234 | 223 | 216 |
| **MAR** | 439 | 418 | 400 | 371 | 360 | 349 |
| **APR** | 558 | 493 | 493 | 493 | 493 | 468 |
| **MAY** | 706 | 691 | 684 | 644 | 644 | 612 |
| **JUN** | 770 | 752 | 745 | 724 | 680 | 666 |
| **JUL** | 817 | 781 | 781 | 781 | 727 | 706 |
| **AUG** | 760 | 713 | 713 | 695 | 670 | 641 |
| **SEP** | 598 | 536 | 526 | 504 | 486 | 464 |
| **OCT** | 421 | 382 | 367 | 349 | 328 | 313 |
| **NOV** | 284 | 270 | 241 | 220 | 220 | 202 |
| **DEC** | 220 | 198 | 187 | 173 | 162 | 162 |
| **SUM** | **6080** | **5738** | **5616** | **5382** | **5162** | **4968** |

**Table 4: Total solar irradiation to a horizontal surface per zone and month – Greece (in MJ/m2-mo)**

Hours of sunshine of a specific day of the year, in a region, are evaluated by the equation:

$$N = \frac{2}{15^{o}/h}\cos^{-1}(-\tan\phi\cdot\tan\delta)$$

where,

$\phi$ : the Latitude of the region (here 39.374°) and

$\delta$ : the solar deviation of a specific day, given by $\delta = 23.45^o \cdot \sin(360\frac{284+n}{365})$,

n is the increasing number of the day year.

The mean hours of sunshine of a month's day is evaluated by the above relation, assuming the 15th as the typical day of each month. The results are plotted in Figure 19.

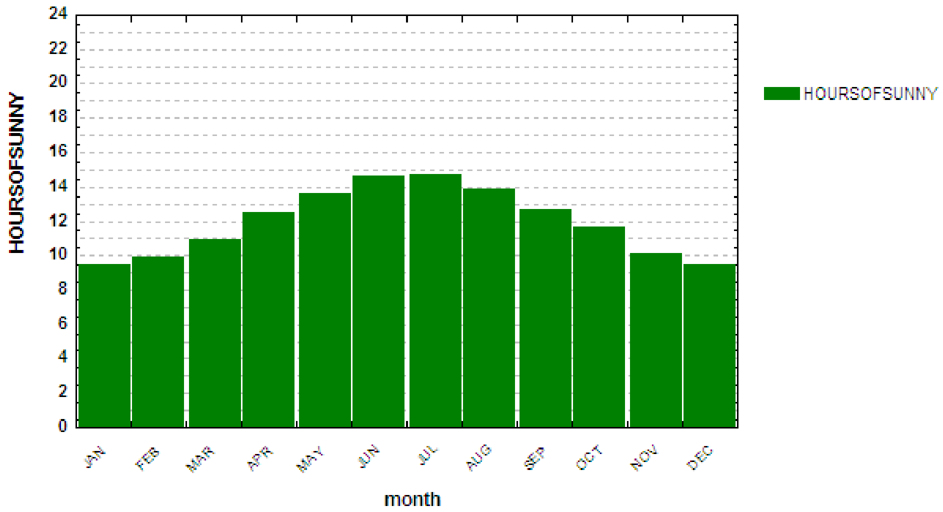



**Figure 19: Estimated mean monthly hours of sunshine per month**

## 4.2 Energy Loads

The evaluation of all energy requirements of the dairy products factory is necessary, in order the hybrid solar plant to be simulated.

During factory's operation energy consumption are for:

1. Climate (heating and cooling) of spaces and
2. Carrying out all required dairy products factory's processes.

### 4.2.1 Loads for climate of spaces

Energy is needed for keeping standard, internal conditions in production spaces of the factory, in accordance with the European Prescriptions which are present for Food Industry and especially for dairy products factories. The internal temperature of the production spaces and laboratories must be kept up to 17°C, the refrigeration-chambers up to 4°C, offices and other auxiliary staff-rooms up to 24°C. This imposes the need of an air-conditioning system plant for all climate spaces-rooms in the dairy products factory.

Energy requirements for air-conditioning are fluctuant quantities, relevant to the hourly, daily and monthly fluctuations of ambient temperature, absolute ambient humidity and solar irradiation. Meteorological data for the specific region of A' Industrial Area of Volos are available only in average monthly values, so the estimation of energy loads was attainable only assuming average monthly values of ambient temperature, absolute ambient humidity and solar irradiation given for regions of $4^{th}$ zone (§4.1).

Monthly heating and cooling (air-conditioning) requirements are analytically estimated by developed models in EES, as summation of all monthly heating and cooling energy loads of climate spaces in the dairy products factory, respectively. Detailed description of the subprograms in EES is not of major interest of this study.

It should be mentioned that:

- The thermal load for the heating of the climate spaces was evaluated by a model in accordance to Degree Method (a brief presentation of the method is given in Appendix C) and
- Cooling load for air-conditioning of the climate spaces was evaluated by a model in accordance to ASHRAE – Cooling Load Temperature Difference (CLTD) Method [35].

The applicable models required data about the building's space as they are the structure and the profile of the building, technical and orientation details of different types of walls, other special prescriptions and use-particularities which are valid per each climate space, and also meteorological data of the region of EVOL, literature data related to above mentioned methods etc.

The climate spaces of the main production building of the factory and the appropriated internal temperature conditions are listed in the following table (Table 5):

| No | SPACE | A [m2] | operation hours per day [h] | $T_{in}$[space] [°C] |
|---|---|---|---|---|
| 1 | **Laboratory** | 132 | 16 | 17 |
| 2 | **Cheese Freezer** | 302 | 24 | 4 |
| 3 | **Main Dairy products factory** | 720 | 16 | 17 |
| 4 | **Subsidairy products factory** | 100 | 16 | 23 |
| 5 | **Cheese Ripening** | 98 | 16 | 17 |
| 6 | **Milk Bottling** | 329 | 16 | 17 |
| 7 | **Dairy products factory Freezer** | 115 | 24 | 4 |
| 8 | **Yoghurt dairy products factory** | 70 | 16 | 17 |

**Table 5: Climate spaces of building**

### 4.2.2 Loads for processes

In the Dairy products factory many processes and procedures are carried out. Large amounts of energy are needed for heating and cooling of fluids or for the operation of the mechanical equipment.

These requirements could be evaluated, as monthly thermal, cooling and electricity loads, assuming standard mean values (see data in Table 6) of energy consumption per 1ton of processed milk, considering the type of desired final product and the monthly potential of the factory.

The evaluation of the above mentioned loads was done, assuming:

- The total monthly quantity of processed milk is: 25tons/day X 25 days/month = 625tons/month.
- The types of final products of the factory, which are:

a) <u>pasteurized liquid products in bottles</u> – in a quantity of 20% of total monthly quantity of processed milk

b) <u>pasteurized liquid products in one-way containers</u> – in a quantity of 50% of total monthly quantity of processed milk, and

c) <u>ripened cheeses with whey processing</u> – in a quantity of 30% of total monthly quantity of processed milk.

| Type of service | Unit | Requirement-including CIP for one ton of milk processed into: | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Liquid products in bottles | | Liquid products in one-way containers | | Skim milk powder and butter | Full cream milk powder | Ripened cheeses | | Evaporated and condensed milk |
| | | pasteurized | sterilized | pasteurized | UHT | | | Without whey processing | with whey processing | |
| NET REQUIREMENT | | | | | | | | | | |
| *Steam* | kg/t | 250 | 300 | 100 | 150 | 880 | 830 | 190 | 700 | 440 |
| *Refregeration total energy equivalent* | kWh/t | 50 | 40 | 50 | 40 | 60 | 45 | 70 | 70 | 45 |
| *Refregeration electric power equivalent* | kWh/t | 20 | 16 | 20 | 16 | 24 | 18 | 28 | 28 | 18 |
| *Heating* | kWh/t | 165 | 200 | 70 | 100 | 585 | 530 | 125 | 460 | 295 |
| *Electric power [total requirement]* | kWh/t | 55 | 70 | 50 | 90 | 90 | 80 | 75 | 100 | 60 |
| *Electric power net requirement]* | kWh/t | 35 | 54 | 30 | 74 | 66 | 62 | 47 | 72 | 42 |
| ***TOTAL NET REQUIREMENT*** | **kWh/t** | **220** | **270** | **120** | **190** | **675** | **610** | **200** | **560** | **355** |
| *<u>Gross energy requirement</u>* | | | | | | | | | | |
| *For heating (furnace fuel)* | kWh/t | 205 | 250 | 90 | 125 | 730 | 660 | 155 | 575 | 370 |
| *For electric powergenerator fuel)* | kWh/t | 195 | 250 | 180 | 315 | 315 | 280 | 265 | 350 | 210 |
| ***TOTAL GROSS REQUIREMENT*** | **kWh/t** | **400** | **500** | **270** | **440** | **1045** | **940** | **420** | **925** | **580** |
| *% of energy in steam in total requirement* | % | 75 | 74 | 58 | 53 | 87 | 84 | 63 | 82 | 83 |
| *% of energy in furnace fuel in total gross* | % | 51 | 50 | 33 | 28 | 70 | 70 | 37 | 62 | 64 |

**Table 6: Standard Energy requirements per 1ton processed milk, in Diaries (source: Institute for sustainable techniques and systems [29])**

## 4.3 Solar Collectors

The relationship of total mean monthly irradiation $H_T$ [MJ/m2-mo], which falls to an oblique surface of a specific angle and the mean monthly irradiation which falls to a horizontal surface, is:

$$H_T = R \cdot H$$

where,

R: is the solar irradiation convert coefficient to oblique surface. For Greece, Table 7, gives the values of R per month and angle, for the 4th climate zone.

| angle per month | 0° | 10° | 20° | 30° | 40° | 50° | 60° | 70° | 80° | 90° |
|---|---|---|---|---|---|---|---|---|---|---|
| **JAN** | 1 | 1.18 | 1.33 | 1.46 | 1.55 | 1.61 | 1.62 | 1.6 | 1.54 | 1.44 |
| **FEB** | 1 | 1.12 | 1.22 | 1.29 | 1.34 | 1.35 | 1.33 | 1.28 | 1.21 | 1.11 |
| **MAR** | 1 | 1.07 | 1.13 | 1.15 | 1.16 | 1.14 | 1.09 | 1.02 | 0.93 | 0.82 |
| **APR** | 1 | 1.03 | 1.04 | 1.03 | 0.99 | 0.94 | 0.87 | 0.78 | 0.68 | 0.56 |
| **MAY** | 1 | 1 | 0.98 | 0.94 | 0.88 | 0.81 | 0.73 | 0.63 | 0.52 | 0.41 |
| **JUN** | 1 | 0.98 | 0.95 | 0.9 | 0.83 | 0.75 | 0.66 | 0.56 | 0.45 | 0.35 |
| **JUL** | 1 | 0.99 | 0.96 | 0.92 | 0.85 | 0.77 | 0.68 | 0.58 | 0.47 | 0.36 |
| **AUG** | 1 | 1.02 | 1.02 | 1 | 0.95 | 0.89 | 0.81 | 0.71 | 0.6 | 0.48 |
| **SEP** | 1 | 1.06 | 1.11 | 1.13 | 1.2 | 1.09 | 1.03 | 0.94 | 0.84 | 0.72 |
| **OCT** | 1 | 1.12 | 1.22 | 1.29 | 1.33 | 1.34 | 1.32 | 1.26 | 1.18 | 1.07 |
| **NOV** | 1 | 1.17 | 1.32 | 1.44 | 1.53 | 1.58 | 1.59 | 1.57 | 1.5 | 1.4 |
| **DEC** | 1 | 1.19 | 1.37 | 1.51 | 1.61 | 1.68 | 1.71 | 1.69 | 1.64 | 1.54 |

**Table 7: Coefficient R per month and angle of oblique surface, for regions in 4th climate zone of Greece.**

The efficiency of an oblique solar collector is given by Hottel-Whillier-Bliss (1959) equation, for flat plate solar collectors FPC:

$$\boxed{\eta_{sc} = F_R(\tau\alpha)_n - F_R U_L \frac{T_{f,i} - T_a}{I_T}}$$

where

$T_\alpha$: ambient temperature

$T_{f,i}$: Inlet to collector water temperature

$I_T$: Instant solar irradiation to solar collector

$F_R$: Correction factor in collector performance equation

$U_L$: Heat loss coefficient from collector to ambient

The quantities:

$F_R U_L$ is in range 5.0-21.5 ($F_R U_L$=5) and

$F_R(\tau\alpha)_n$ in range 0.45-0.86 ($F_R(\tau\alpha)_n$=0.75),

are depended on the type and the constructor of the collector (Typical values given in Table 8).

$(\tau\alpha)_n$: Mean overall transmissivity – absorptivity product of an oblique solar collector.

| Type | Collector | $F_R(\tau\alpha)_n$ | $F_R U_L$ [W/m2*°C] |
|---|---|---|---|
| I | Black colour, one glass | 0.82 | 7.5 |
| **II** | **Black colour, double glassed or selectable surface with one-glass** | **0.75** | **5.0** |
| III | Vacuum tube | 0.45 | 1.25 |
| IV | Plastic collector without glass and insulation | 0.86 | 21.5 |

**Table 8: Characteristic values of quantities $F_R(\tau\alpha)_n$ and $F_R U_L$ for different types of collectors**

## 4.4 Solar Thermal System

As it was described in the previous chapters, a solar thermal system composed by:

- solar collectors area
- an exchanger and
- a heat storage tank.

The behaviour of a solar thermal plant which covers part of mean monthly load can be adequately described by the F-chart method, which mentioned in the book Solar heating design by F-chart method (Klein et al., 1997). According to this method the percentage of monthly thermal load $L_{SC}[i]$, which is covered by solar energy (monthly fraction of coverage), is determined by Equation:

$$\boxed{f_x[i] = 1.029 \cdot Y[i] - 0.065 \cdot X[i] - 0.245 \cdot Y[i]^2 + 0.0018 \cdot X[i]^2 + 0.0215 \cdot Y[i]^3,}$$

where: i=1…12 month

$f_x[i]$: mean monthly cover fraction of load

$$\boxed{X[i] = A_{SC} \cdot F_R U_L \cdot \left(\frac{F_R'}{F_R}\right) \frac{\left[T_{SC,ref} - \overline{T}_\alpha[i]\right]}{L_{SC}[i]} \cdot \Delta t[i] \cdot K_2 \cdot K_3[i]}$$

$$\boxed{Y[i] = A_{SC} \cdot \frac{\left(\frac{F_R'}{F_R}\right) \cdot F_R(\tau\alpha)_n \dfrac{(\tau\alpha)}{(\tau\alpha)_n} \cdot H_T[i] \cdot K_4}{L_{SC}[i]}}$$

$L_{SC}[i]$: mean monthly load connected to the solar thermal plant

$A_{SC}$ :solar collectors area

$\left(\frac{F_R'}{F_R}\right)$ : the percentage of transferred heat to the tank by the exchanger (usually takes values in range of 90%-95%, in present study $\left(\frac{F_R'}{F_R}\right)$=0.95).

$T_{SC,ref}$: reference temperature (usually taken for FPC as 100°C).

$\overline{T}_{\alpha}[i]$: mean monthly ambient temperature (during daylight).

$\Delta t[i]$: the duration of each month i in sec.

$\frac{(\tau\alpha)}{(\tau\alpha)_n}$: mean monthly values depended upon the type of collector, the month and the angle of the collector to horizontal surface. It is given in tabulated data (Table 9).

| angle per month | 0° | 10° | 20° | 30° | 40° | 50° | 60° | 70° | 80° | 90° |
|---|---|---|---|---|---|---|---|---|---|---|
| **JAN** | 0.74 | 0.81 | 0.86 | 0.89 | 0.91 | 0.93 | 0.93 | 0.93 | 0.93 | 0.9 |
| **FEB** | 0.8 | 0.85 | 0.88 | 0.9 | 0.92 | 0.92 | 0.93 | 0.92 | 0.91 | 0.8 |
| **MAR** | 0.85 | 0.88 | 0.9 | 0.91 | 0.92 | 0.92 | 0.91 | 0.89 | 0.87 | 0.8 |
| **APR** | 0.89 | 0.91 | 0.92 | 0.92 | 0.91 | 0.9 | 0.88 | 0.85 | 0.8 | 0.7 |
| **MAY** | 0.92 | 0.92 | 0.92 | 0.91 | 0.9 | 0.88 | 0.85 | 0.8 | 0.73 | 0.6 |
| **JUN** | 0.92 | 0.93 | 0.92 | 0.91 | 0.89 | 0.87 | 0.83 | 0.88 | 0.7 | 0.6 |
| **JUL** | 0.92 | 0.93 | 0.92 | 0.92 | 0.9 | 0.87 | 0.84 | 0.78 | 0.71 | 0.6 |
| **AUG** | 0.91 | 0.92 | 0.92 | 0.92 | 0.91 | 0.9 | 0.87 | 0.83 | 0.77 | 0.6 |
| **SEP** | 0.88 | 0.9 | 0.91 | 0.92 | 0.92 | 0.92 | 0.9 | 0.88 | 0.85 | 0.7 |
| **OCT** | 0.82 | 0.86 | 0.89 | 0.91 | 0.92 | 0.93 | 0.93 | 0.92 | 0.9 | 0.8 |
| **NOV** | 0.75 | 0.82 | 0.87 | 0.9 | 0.92 | 0.93 | 0.94 | 0.94 | 0.93 | 0.9 |
| **DEC** | 0.72 | 0.8 | 0.85 | 0.89 | 0.91 | 0.93 | 0.94 | 0.94 | 0.93 | 0.9 |
| **average** | **0.84** | **0.88** | **0.9** | **0.91** | **0.91** | **0.91** | **0.9** | **0.88** | **0.84** | **0.74** |

**Table 9: Values of $\frac{(\tau\alpha)}{(\tau\alpha)_n}$ per month and angle of oblique collector, for collectors with double glass wall.**

$H_T[i]$: Mean monthly solar irradiation.

The correction factors $K_2$, $K_3$ and $K_4$:

$K_2$: Correction factor for the volume of storage tank. The equation of $f_x[i]$ was estimated for volumes equal to 75lt/m2 collector's area.

For different sizes of the volume: $K_2 = (V_{\tan k} / 75)^{-1/4}$,

$V_{\tan k}$ the volume of storage tank, in lt/m2 collector's area. For low volumes of storage tank the heat losses of the system are increased.

$K_3[i]$: Correction factor for hot water supply per month. It is given by the equation: $K_3[i] = \dfrac{11.6 + 1.18T_w + 3.86T_m[i] - 2.32T_\alpha[i]}{100 - T_\alpha[i]}$ where,

$T_w$: Desirable water temperature in storage tank.

$T_m[i]$: Mean water temperature by water supply net. (In Table 10 are given the mean monthly values for $T_m[i]$, per zone in Greece).

| | JAN | FEB | MAR | APR | MAY | JUN | JUL | AUG | SEP | OCT | NOV | DEC |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| zones 1,2 | 12 | 12 | 14 | 16 | 19 | 22 | 24 | 24 | 22 | 19 | 16 | 14 |
| zones 3,4 | 10 | 10 | 12 | 15 | 19 | 21 | 24 | 24 | 22 | 19 | 15 | 12 |
| zones 5,6 | 8 | 8 | 10 | 13 | 17 | 19 | 22 | 22 | 20 | 17 | 13 | 10 |

**Table 10: Water temperatures by water supplier in Greece per month and climate zone**

$K_4$: Correction factor for the magnitude of exchanger. It is given by the equation: $K_4 = 0.39 + 0.65e^{-0.139/\lambda}$ where $\lambda$ is a variable in range 1-3 for reasonable viable plants. In present study $\lambda$=1.5.

## 4.5 Absorber Chiller

An absorber chiller is composed by four basic units:

- the generator
- the condenser
- the absorber and
- the evaporator.

Mainly, they are supplied with district heat, waste heat or heat from hybrid solar system. A thermal compression of the refrigerant is achieved by using a liquid refrigerant solution and a heat source, thereby replacing the electric power consumption of a mechanical compressor. For chilled water above 0°C, as it is used in air conditioning, typically a liquid H2O/LiBr solution is applied with water as solution refrigerant.

The cooling effect is based on the evaporation of the refrigerant (water) in the evaporator at very low pressures. The vaporized refrigerant is absorbed in the absorber, thereby diluting the H2O/LiBr solution. The solution is continuously pumped into the generator, where the regeneration of the solution is achieved by applying driving heat (e.g. hot water). The refrigerant leaving the generator by this process condenses through the application of cooling water in the condenser and circulates by means of an expansion valve again into the evaporator. The absorber refrigerant cycle is shown in Figure 20.

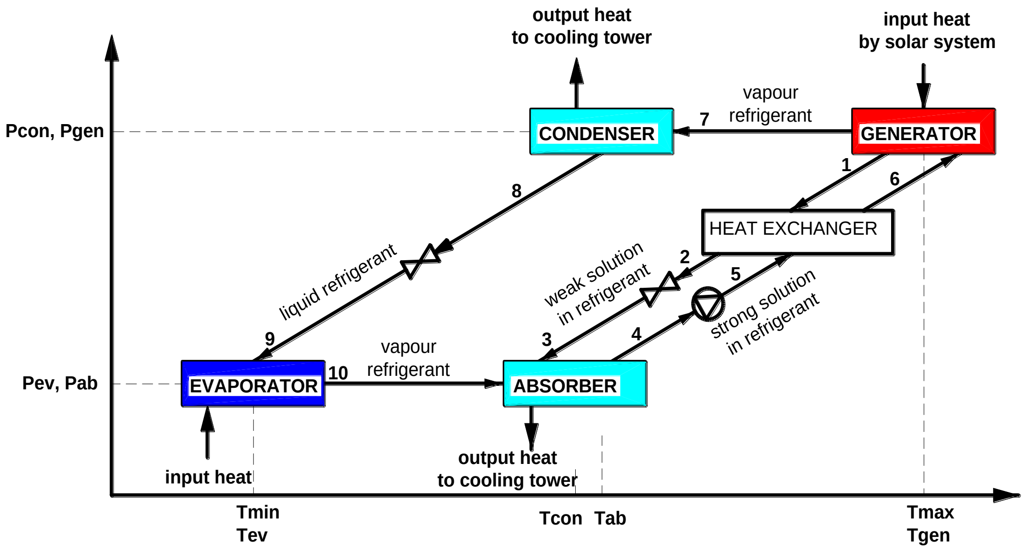



**Figure 20: Absorption Refrigeration Cycle**

Energy analysis of absorption chiller was carried out in order to define the thermodynamic attributes in each state (1-10) of cycle.

We assume that the operation is in steady state, all states are in thermodynamic equilibrium, the temperature difference between the inputs and outputs of exchanger is $\Delta T_{EXCHANGER} = 10^o C$, no pressure drop is occurring in tubes.

Also it is considered that h[2]=h[3], h[4]=h[5] and h[8]=h[9] in equal-enthalpy processes.

The operation temperatures are in the range of $T_{\min}$ - $T_{\max}$ :

$$T_{MAX} = T[1] = T[7], \quad T_{MIN} = T[9] = T[10],$$

$$T_{AB} = T[4] = T[5], \quad T_{CON} = T[8], \quad T[2] = T[3] = T[5] + \Delta T_{EXCHANGER}$$

We assume two operative pressures:

$$P_{MAX} = P[1] = P[2] = P[5] = P[6] = P[7] = P[8] \qquad \approx 8.799kPa$$

$$P_{MIN} = P[3] = P[4] = P[9] = P[10] \qquad \approx 0.834kPa$$

It is defined as $X_s[i]$, i=1..8 state the [%] mass concentration of LiBr in the solution.

It is obvious that $X_s[7] = X_s[8] = X_s[9] = X_s[10] = 0$ because refrigerant is pure water.

By expert thermodynamic functions of EES, are being evaluated the:

$X_s[1] = X_s[2] = X_s[3] = X\_LIBR(T[i], P[i]] = X_{S,AB}$, weak solution in refrigerant and

$X_S[4] = X_S[5] = X_S[6] = X\_LIBR(T[i], P[i]] = X_{S,R}$, strong solution in refrigerant.

The enthalpies in states 7,8, and 10 (water) are being calculated by steam tables or expert thermodynamic functions of EES program. It would be also h[8]=h[9].

In the same manner, they are calculated by EES functions which are for LiBr solutions:

- the enthalpies h[1], h[2] and h[4] by function H_LIBR('ENG',(T[i]),($X_S$[i])) and

- the temperature T[6], by function T_LIBR('SI',P[6],$X_S$[6])

If m is the symbol of relative mass flow of fluid, applying the mass balance equations for each component of absorber chiller, apply:

$m_r = m[7] = m[8] = m[9] = m[10]$ - mass flow or refrigerant,

$m_{ab} = m[1] = m[2] = m[3]$ - mass flow of absorber's solution and

$m_s = m[4] = m[5] = m[6]$ - mass flow of solution.

and for condenser it will be present that $m[6] \cdot X_{S,R} = m[1] \cdot X_{S,AB}$

Applying energy balance equations for the components of the chiller:

$Q_{GEN} + m[6] \cdot h[6] = m[1] \cdot h[1] + m[7] \cdot h[7]$: energy balance in generator

$Q_{AB} + m[4] \cdot h[4] = m[3] \cdot h[3] + m[10] \cdot h[10]$: energy balance in absorber

$Q_{CON} + m[8] \cdot h[8] = m[7] \cdot h[7]$: energy balance in condenser

$Q_{EV} + m[9] \cdot h[9] = m[10] \cdot h[10]$: energy balance in evaporator

Efficiency of absorber chiller: $\boxed{COP_{AC} = \frac{Q_{EV}}{Q_{GEN}}}$

Power of required cooling tower: $\boxed{Q_{CT} = Q_{AB} + Q_{CON}}$.

The critical variables of the chiller's model are:

- the power of absorber chiller $Q_{EV}$ ,
- the efficiency $COP_{AC}$
- the driven temperature $T_{MAX}$ and
- the power of required cooling tower $Q_{CT}$ .

## 4.6 Hybrid Solar Plant

The simulation of the installation of the hybrid solar plant in the dairy products factory, was based on a simplified model (Figure 21) in which the basic systems of the plant (eg. the solar thermal system, the solar cooling system, the conventional thermal system etc) were grouped into distinct energy units. Each unit was simulated by separate subprograms considering the form and the technology of its equipment, according the models presented before, in §4.2, §4.3, §4.4 and §4.5. Dimensionless critical coefficients f, y, z, w, which are characteristic to the size of the equipment of units, were defined in order to simplify the simulation and mainly to control the degrees of freedom of the applied model during optimization analysis (for properly sizing of equipment). Consecutively, the interfacing relations between the groups were formed, according to the model (Figure 21) which is studied.

The system as it was designed is comprised by the following main groups:

1. THERMAL LOAD of the factory.
2. COOLING LOAD of the factory.
3. ELECTRICITY (LOAD) of the factory.
4. CONVENTIONAL HEATING SYSTEM: crude-oil burner, boiler – existing energy system for providing heating.
5. SOLAR THERMAL SYSTEM: solar collectors, storage tank, exchanger.
6. SOLAR COOLING SYSTEM: absorber chiller, cooling tower.
7. MECHANICAL COMPRESSOR COOLING SYSTEM: mechanical compressor -– existing energy system for providing cooling.
8. ELECTRICITY BY GRID
9. PHOTOVOLTAIC SYSTEM

The main object is the evaluation of the annual energy of the system in any stage (inputs - outputs) of the above mentioned groups. The annual values are evaluated as the sum of the respective monthly rates.

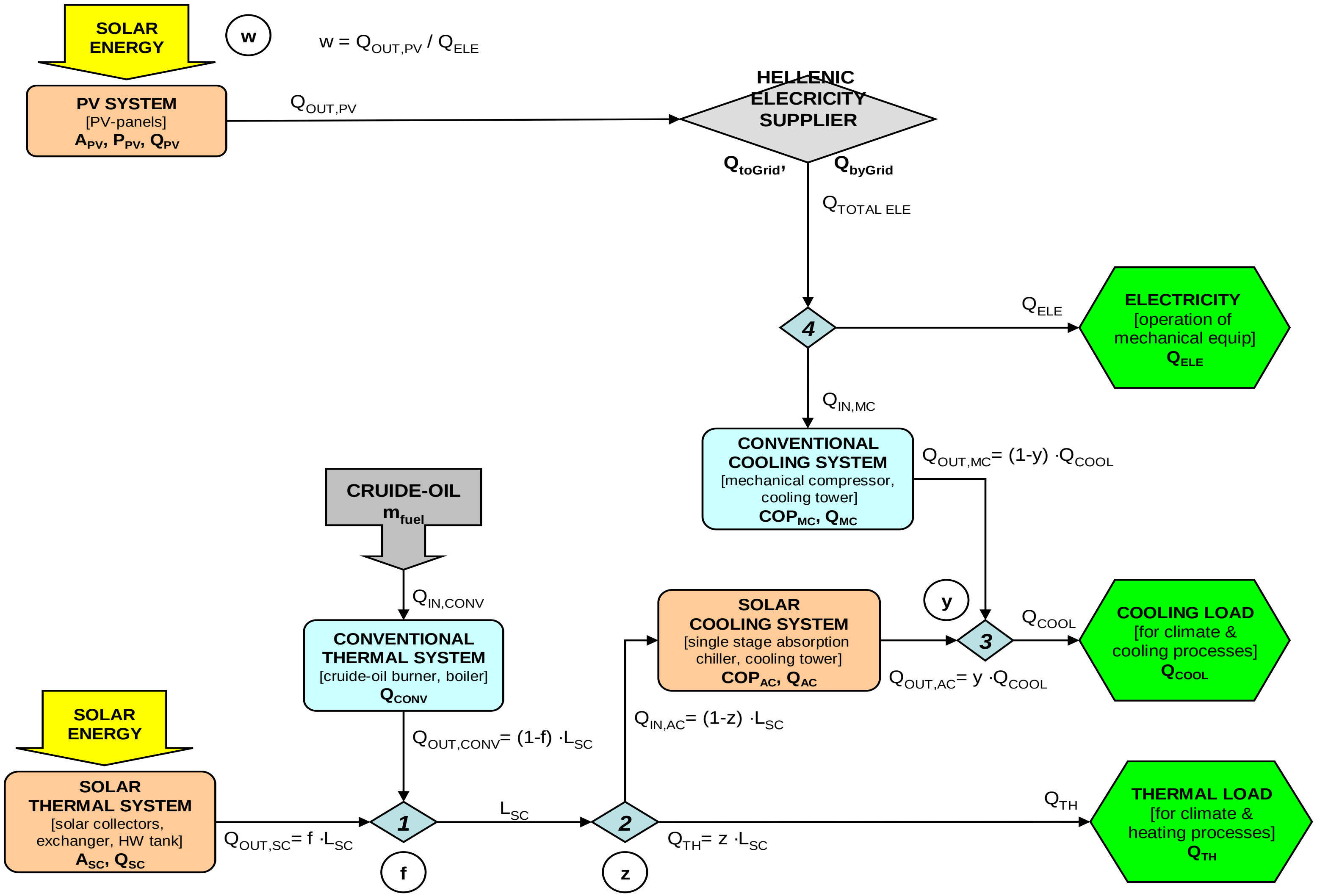



**Figure 21: Model of the Hybrid Solar Plant**

The critical **critical coefficients f, z, y, w and x** (annual rates) which are defined in the model, so as technical analysis and operation's optimisation could be applied, are:

- Mean annual load coverage fraction: $\boxed{f = f[13] = \frac{Q_{SC}^{out}[13]}{L_{sc}[13]}}$.

- Ratio of mean annual thermal load to total annual load: $z = \frac{Q_{TH}[13] / \eta_{EXCH}}{L_{sc}[13]}$.

- Ratio of mean annual cooling energy by AC to total cooling load or cooling load coverage fraction by AC: $\boxed{y = y[13] = \frac{Q_{AC}^{out}[13]}{Q_{COOL}[13]}}$.

- Ratio of mean annual electricity by PV-SYSTEM to total annual electrical consumption (existed conventional plant) or electricity coverage fraction by PV-SYSTEM: $\boxed{w = \frac{Q_{PV}^{out}[13]}{Q_{GRID}^{0}[13]}}$.

- Fuel reduction fraction by hybrid solar system: $\boxed{x = \frac{m_{FUEL}[13]}{m_{FUEL}^{0}[13]}}$.

The modelling variables of the studied system are:

- The estimated monthly and annual thermal $Q_{TH}[i]$, cooling $Q_{COOL}[i]$ and electrical $Q_{ELE}[i]$, dairy products factory's loads.
- The area for the installation of solar collectors $A_{SC}$ .
- The area for the installation of photovoltaic panels $A_{PV}$.
- The total covered by solar panels (solar collectors and photovoltaic panels) area ($A_{TOTAL}$= $A_{SC}$ + $A_{PV}$ – should be less than 4.000m$^2$).
- The operation temperatures of AC components (initial design values: $T_{MAX}$=88.8°C, $T_{MIN}$=4.5°C, $T_{AB}$=33°C, $T_{CON}$=43°C).
- The desire temperature of hot water needed by the load (initial design value: $T_W$=88.8°C).

It is noticed that the numbers included inside the parentheses corresponds to each month for i=1..12 and the i=13 for the annual values.

According to the above defined critical coefficients, the equations of the model are written as (i=1…12 for each month and i=13 for the annual values):

- output by AC: $Q_{AC}^{out}[i] = y[i] \cdot Q_{COOL}[i]$ (at point 3),

- output by Solar Thermal System: $Q_{SC}^{out}[i] = f[i] \cdot L_{SC}[i]$ (at point 1), where $f[i]$ is estimated by the Solar thermal system model, as function to the designing parameters and loads of factory (f[i]=$f_x$(zone,angle, i, $A_{SC}$, $L_{SC}$[i], d_exh, $T_W$)).

• operation of AC: $Q_{AC}^{out}[i] = COP_{AC} \cdot Q_{AC}^{in}[i]$ - the $COP_{AC}$ is estimated by the AC model, as function to the AC operation temperatures.

• operation of MC: $Q_{MC}^{out}[i] = COP_{MC} \cdot Q_{MC}^{in}[i]$,

• operation of conventional thermal system: $Q_{CONV}^{out}[i] = \eta_{COMBUSTION} \cdot \eta_{BOILER} \cdot Q_{CONV}^{in}[i]$

• operation of combustion chamber (conventional thermal system): $LHV \cdot m_{FUEL}[i] = Q_{CONV}^{in}[i]$

• operation of PV-SYSTEM: $Q_{PV}^{out} = P_{PV} \cdot Q_{PV}^{out0}[i]$, the energy which is buying by Electricity Supplier $Q_{ELE}^{byGrid} = Q_{GRID}[i]$ if $Q_{GRID}[i] \geq 0$, the energy which is sold to Electricity Supplier $Q_{ELE}^{toGrid} = -Q_{GRID}[i]$ if $Q_{GRID}[i] \leq 0$,

and applying energy balance equations for each hub-point, in average monthly consideration, for i=1…12:

at point 1: $Q_{CONV}^{out}[i] = L_{SC}[i] - Q_{SC}^{out}[i]$,

at point 2: $L_{SC}[i] = Q_{AC}^{in}[i] + \dfrac{Q_{TH}[i]}{\eta_{exch}}$,

at point 3: $Q_{MC}^{out}[i] = (1 - y[i]) \cdot Q_{COOL}[i]$,

at point 4: $Q_{MC}^{in}[i] + Q_{ELE}[i] = Q_{GRID}[i] + Q_{PV}^{out}[i]$.

The consumption of existing conventional dairy products factory's plant are:

• for crude-oil: $m_{fuel}^{0}[i] = \dfrac{Q_{TH}[i]}{\eta_{exch} \cdot \eta_{combustion} \cdot \eta_{boiler} \cdot LHV}$, i=1..12 month

• for electricity: $Q_{GRID}^{0}[i] = Q_{ELE}[i] + \dfrac{Q_{COOL}[i]}{COP_{MC}}$, i=1..12 month

and:

the annual fuel consumption is $m_{fuel}^{0}[13] = \sum_{i=1}^{12} m_{fuel}^{0}[i]$

where $Q_{GRID}^{0}[13] = \sum_{i=1}^{12} Q_{GRID}^{0}[i]$.

The area of solar collectors $A_{SC}$ is calculated, as a function of the energy should be provided by the solar thermal plant.

The area of PV panels $A_{PV}$ is calculated, as function of the installed PV-System power.

The total solar plant's required area is: $\boxed{A_{TOTAL} = A_{SC} + A_{PV}}$.

The available output surface (on the roof of buildings A and B), is 4.000m[2]. This value should be not exceeded by the covered area of the solar panels (constraint in the optimization analysis).

The <u>modelling parameters</u> of the studied system are:

- climatologic zone of the dairy products factory's region [zone=4],
- the Latitude of of the dairy products factory's region [φ=39.374°],
- assuming angle for solar collectors and photovoltaic panels equal to 30° to south,
- type of solar collector [TC=2], corresponding to flat plate collectors with selectable surface,
- dairy products factory's operation hours per day [OHRS=16h/day],
- dairy products factory's operation days per month [ODAYS=25d/m],
- the quantity of total processed milk per day [TOTALMILK_d=25tons/d],
- efficiency of exchangers [$\eta_{exch}$=0.9],
- efficiency of combustion chamber [$\eta_{combustion}$=0.7],
- efficiency of conventional boiler [$\eta_{boiler}$=0.85],
- low Heating value for crude-oil [LHV= 10.97kWh/kg],
- efficiency of mechanical cooling system – MC [$COP_{MC}$=2.5],
- output electricity by PV system, for the dairy products factory's region and applied PV technology per KW of installed power. (Table 11 - $Q_{PV}^{out0}[i]$ i=1..12),
- the required area of PV panels per KW of installed power of PV-SYSTEM [≈16m2/KW].

| month | JAN | FEB | MAR | APR | MAY | JUN | JUL | AUG | SEP | OCT | NOV | DEC | SUM |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| energy by PV-SYSTEM per kW of installed power [kWh/KW] | 59.8 | 67.1 | 95.9 | 121 | 134 | 138 | 142 | 133 | 113 | 94.4 | 64.9 | 47.7 | **1211** |

**Table 11: PV-SYSTEM output per KW of installed PV power.**

## <u>4.7 Financial Model</u>

The evaluation of crucial economic parameters and the optimization of an energy plant and the feasibility of an investment are attained by the financial model.

First step is the determination of all required model's parameters:

- Financial life of system [BL=20 y],
- Nominal market discount rate [$r_n$ =10%],
- Inflation [$r_i$ =3%],
- Initial Purchase costs for each required equipment [€/unit of its size] – Table 12.
- Cost of electricity bought by the Grid [€/KWh] – Table 12.
- Cost of electricity sold to the Grid [€/KWh] – Table 12.
- Fuel cost [€/ton of fuel] – Table 12.

- Maintenance cost of solar plant factor [% of TCI] – Table 12.
- Plant depreciation S, at the end of financial life.

| **Equipment Costs** | **Symbol** | | **[unit]** | |
|---|---|---|---|---|
| Absorption Chiller (one-stage): | $c_{AC}$ | 400 | [€/kW] | - price in 2010 |
| Solar Collectors (FPC): | $c_{SC}$ | 180 | [€/m$^2$] | - price in 2010 |
| Preheater-Exchanger: | $c_{EXCH}$ | 50 | [€/kW] | - price in 2004 |
| Cooling Tower: | $c_{CT}$ | 50 | [€/kW] | - price in 2004 |
| Storage Tank: | $c_{TANK}$ | 600 | [€/m$^3$] | - price in 2004 |
| TCI-PV System: | $c_{PV}$ | 4500 | [€/kW] | - price in 2010 |
| **Energy prices** | | | | |
| Fuel (crude-oil) cost: | $c_{FUEL}$ | 0.34 | [€/kg] | - price in 2010 |
| Electricity cost bought by the Grid: | $c_{ELE}^{byGrid}$ | 0.09412 | [€/kWh] | - price in 2010 |
| Electricity cost sold to the Grid: | $c_{ELE}^{toGrid}$ | 0.45 | [€/kWh] | - price in 2010 |
| **Operation & Maintenance Costs:** | | | | |
| Operation Cost of PV-system: | | 0.01 | (0.5%-1.5%) annual of TCI | |
| Equipment Installation: | | 0.12 | 12% of PEC | |

**Table 12: Economic parameters of present study**

All financial data in the Table 12, should be readjusted to reference year (2010) by the compound amount factor [ $CAP = \frac{f}{p} = (1+i)^{2010-yearofprice}$ , where i is the rate of interest].

Several economic criteria have been proposed, according to which, a feasibility analysis can be applied. These are the critical variables of the model:

- The net present value of an investment (NPV).
- The internal investment's refund rate (IRR).
- The Pay-Back Period of an investment (PP).

Afterwards the applying model is formed as it is described below.

The total analysis will be done in constant values consideration, in order to overcome hyper-costing of total system.

The rate of interest $i$ (in constant values) is calculated by the relationship: $(1+r_n) = (1+i)\cdot(1+r_i)$ .

The ***Total Investment Cost (TCI)*** is a cost which is paid in a priori and as it was designed previously, it is estimated as the sum of TCI of solar thermal and cooling plant and the TCI of PV-system: $TCI = TCI_{SOLAR} + TCI_{PV}$ , where:

$TCI_{PV} = P_{PV} \cdot c_{PV}$ and

$TCI_{SOLAR}$ is empirically evaluated as a function of the total new equipment cost (PEC) of the solar thermal and cooling plant. In present study, the solar plant represents an expansion installation of an existing dairy products factory in operation therefore it was assumed that:

$$\boxed{TCI_{SOLAR} = 1.5 \cdot PEC_{SOLAR}}$$

The total new equipments cost is the summation of the equipment's cost of solar thermal plant and solar cooling plant: $PEC_{SOLAR} = PEC_{SOLARTH} + PEC_{COOL}$.

Each term of previous equation is defining, as follows:

- $PEC_{SOLARTH} = PEC_{SC} + PEC_{EXCHANGER} + PEC_{TANK}$, where

$PEC_{SC} = A_{SC} \cdot c_{sc}$ , $PEC_{EXCHANGER} = P_{EXCHANGER} \cdot c_{EXCH}$, $PEC_{TANK} = V_{TANK} \cdot c_{TANK}$.

- $PEC_{COOL} = PEC_{ABSORBER} + PEC_{COOLINGTOWER}$, where

$PEC_{ABSORBER} = P_{AC} \cdot c_{AC}$, $PEC_{CT} = P_{CT} \cdot c_{CT}$.

The running costs which are paid every year are for the operation and maintenance of proposed plant are determined:

$Z_K = Z_{FUEL} + Z_{ELE} + Z_{SOLAR\_ELE} + Z_{OM}$, where:

$Z_{FUEL} = m_{FUEL}[13] \cdot c_{fuel}$: the annual money which are spend for the fuel (crude-oil) consumption of conventional part of hybrid plant.

$Z_{ELE} = Q_{ELE}^{byGrid}[13] \cdot c_{ELE}^{byGrid}$: the annual money which are spend for electricity to official national Supplier for the operation of hybrid plant.

$Z_{SOLAR\_ELE} = 0.93 \cdot A_{SC} \cdot c_{ELE}^{byGrid}$: money which are spend annually for electricity to official national Supplier for the operation of solar plant (additional operation of absorber chiller, pumps etc).

$Z_{OM} = 0.1\% \cdot TCI$ : amounts which are spend annually for the maintenance of solar part of hybrid plant.

The running operation's costs for the existing conventional plant are for the fuel and electricity annually consumption:

$Z_K^0 = Z_{FUEL}^0 + Z_{ELE}^0 = m_{fuel}^0[13] \cdot c_{FUEL} + Q_{GRID}^0 \cdot c_{ELE}^{byGrid}$

The running annual benefits of the hybrid solar plant, can be evaluated as the summation of the benefits of the reduction of operation costs and of the incoming money by electricity sale to electricity Supplier:

$ES = ES_{SC} + ES_{PV} = (Z_K^0 - Z_K) + Q_{ELE}^{toGrid}[13] \cdot c_{ELE}^{toGrid}$

Finally, the critical variables of NPV, IRR and PP can be calculated:

Net Present Value of an investment:

$$\boxed{NPV = ES \cdot USCAF(i, BL) - \left[TCI - S \cdot PWF(i, BL)\right]}\ [€], \text{ where}$$

$USCAF(i, BL) = \dfrac{(1+i)^{BL} - 1}{i}$: the uniform series compound-amount factor and

$PWF(i, BL) = \dfrac{1}{(1+i)^{BL}}$: the present worth factor.

Ending the payback period of the investment [PP] is defined as:

$$\boxed{PP = \frac{\log 10\left[\dfrac{TCI}{ES}\left(\dfrac{i}{100} + 1\right)\right]}{\log 10\left(1 + \dfrac{i}{100}\right)}}$$

# CHAPTER 5:

# RESULTS AND DISCUSSION

## 5.1 Energy Loads

The energy requirements of the dairy products factory are:

1. Thermal load for heating climate spaces.
2. Cooling load for cooling climate spaces.
3. Thermal load for heating fluids processes.
4. Cooling load for heating fluids processes.
5. Electrical load for the operation of remaining electromechanical machinery.

All the energy needs of the factory for heating, cooling and electricity were estimated by runs of the simulation model, developed in EES, according to details given previously in §4.2.

In Table 13 are tabulated the total monthly, energy requirements [kWh] per type of load: thermal (as sum of thermal load for heating the climate spaces and all heating procedures of the factory), cooling (as sum of cooling load for air-conditioning spaces and for cooling processes in the factory) and electricity for the operation of the mechanical equipment (except those for cooling – mechanical compressors). In the final row of the table, are written the annual energy requirements of the present application.

| month | $Q_{THERMAL}$ [kWh] | $Q_{COOL}$ [kWh] | $Q_{ELE}$ [kWh] |
|---|---|---|---|
| **JAN** | 198,535 | 114,996 | 35,425 |
| **FEB** | 194,314 | 113,928 | 35,425 |
| **MAR** | 182,162 | 122,880 | 35,425 |
| **APR** | 168,222 | 144,305 | 35,425 |
| **MAY** | 162,364 | 163,721 | 35,425 |
| **JUN** | 161,563 | 189,104 | 35,425 |
| **JUL** | 161,563 | 195,681 | 35,425 |
| **AUG** | 161,563 | 197,465 | 35,425 |
| **SEP** | 161,563 | 166,239 | 35,425 |
| **OCT** | 162,871 | 132,164 | 35,425 |
| **NOV** | 174,770 | 102,320 | 35,425 |
| **DEC** | 194,314 | 108,245 | 35,425 |
| **SUM** | **2,083,804** | **1,751,048** | **425,100** |

**Table 13: Thermal, Cooling and Electrical Load per month**

Resuming we report that the annually requirements of the factory are approximately 2.085.000kWh in thermal energy, 1.750.000 kWh as cooling load and 425.000 kWh in electrical energy.

Also, the installed cooling capacity was calculated 567.8kW.

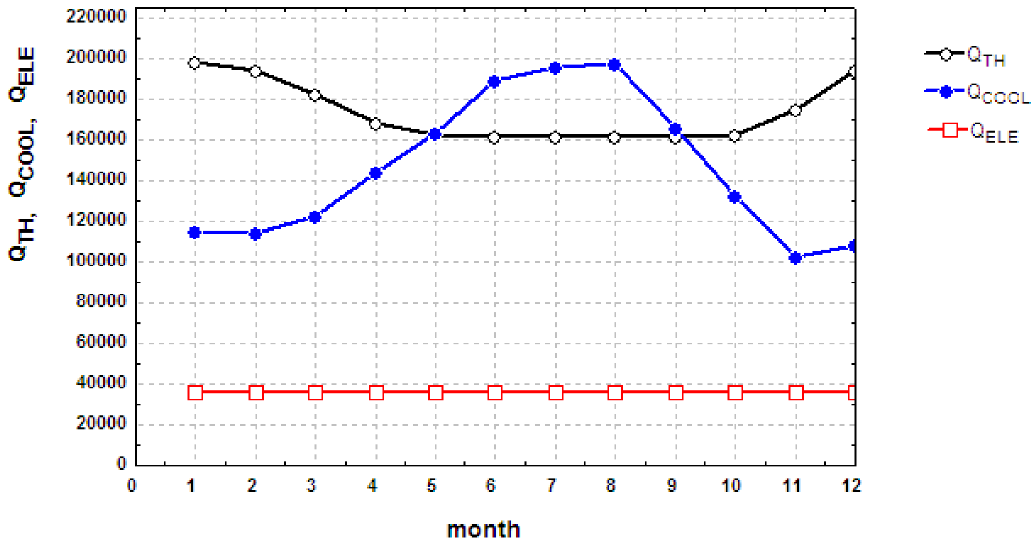



**Figure 22: Thermal, Cooling and Electrical Load per month**

In Figure 22, the estimated loads per month are plotted. Large amounts of energy are for thermal loads, due to many heating processes which are carrying out in dairy products factory. At summer periods the total thermal load is decreasing due to the fact of needless heating spaces. Total cooling load, is high at summer months, because of air-conditioning.

By the present energy state, thermal loads are covered by the operation of conventional thermal system of boilers driven by crude-oil burner. Cooling loads are covered mainly by the operation of electrical driven mechanical compressor chillers. Finally, electricity is provided by the Hellenic Electricity Supplier (Public Power Corporation).

## 5.2 Simulation results

### 5.2.1 Effect of the volume of heat storage tank in plant profitability

The volume of the solar storage tank affects the performance and the cost of the solar system.

If we consider $V_0$ as the initial volume in $m^3$ which corresponds to each $m^2$ of solar collector area, the graph of the cover load fraction [f] and the variation of the dimensionless cost of tank [$V_0$], for defined solar collector area [$A_{SC}$], can give useful connotations about the appropriate magnitude of storage tank.

In figure 23 is given the graph for defined $A_{SC}$ = 2500$m^2$. The cross point of the graph, which is for $V_0$ = 0.045 m3 per m2 of solar collector area [$A_{SC}$], is a good value for the defined case.

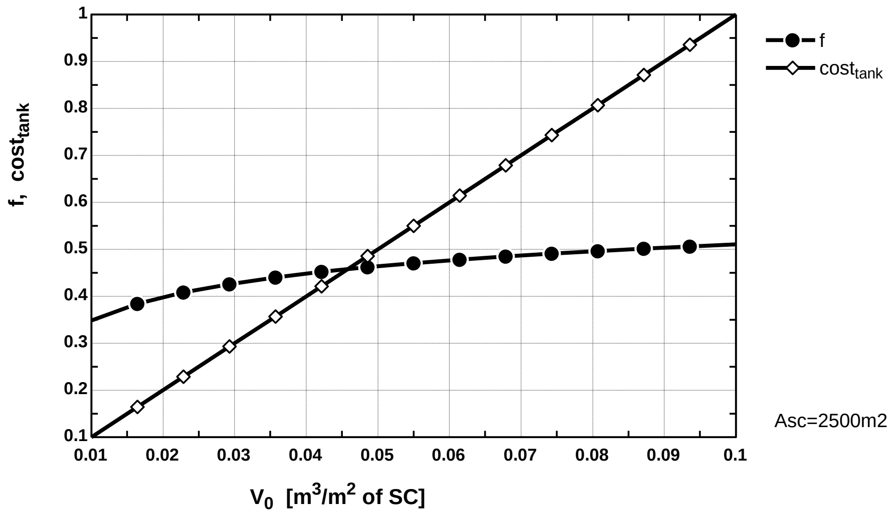



**Figure 23: Variation of load cover fraction f & unit cost of the storage tank per its volume (for the case of $A_{SC}$ = 2500m²)**

Correspondingly, in figure 24 is given the graph for $A_{SC}$ = 4000m². The cross point of the graph, was moved in $V_0$ = 0.063 m3 per m2 of solar collector area [$A_{SC}$], for the special case.

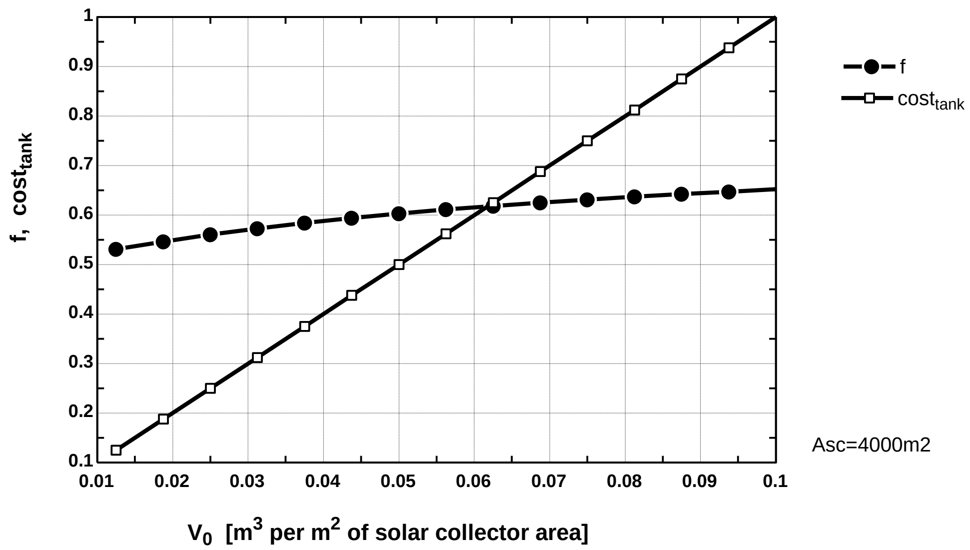



**Figure 24: Variation of load cover fraction f & unit cost of the storage tank per its volume (for the case of $A_{SC}$ = 4000m²)**

As the solar collector area becomes larger, the load cover fraction as expected becomes higher. If the area of solar collectors is small, the load is supplied almost directly, so the storage tank has no influence to the operation of the thermal system. Anyway, a good selection of the magnitude of storage tank volume would be (as it is suggested by several authors) around the value of $V_0$=0.05$m^3$ per $m^2$ of solar collector area [$A_{SC}$].

### 5.2.2 Effect of the desirable water temperature of load [$T_w$].

Another parameter which is critical for the performance of the hybrid solar system and it strongly affects the size and the initial cost of the system studied, is the desired temperature of the hot water [$T_w$], in which the hot water should be supplied to the load. This upper level of water temperature is defined by the type of energy needs which should be covered by solar system. In the case of cooling solar system this level should be higher (≥80°C) than in the case of the needs for heating in a dairy products factory. As the driving temperature becomes higher (increased energy requirements), the load cover fraction by solar collectors is reduced or the additional required heating is covered by the conventional crude-oil thermal system.

In figure 25 is given the graph of the variation of load coverage fraction f by the water temperature in storage tank for $A_{SC}$=4000m2.

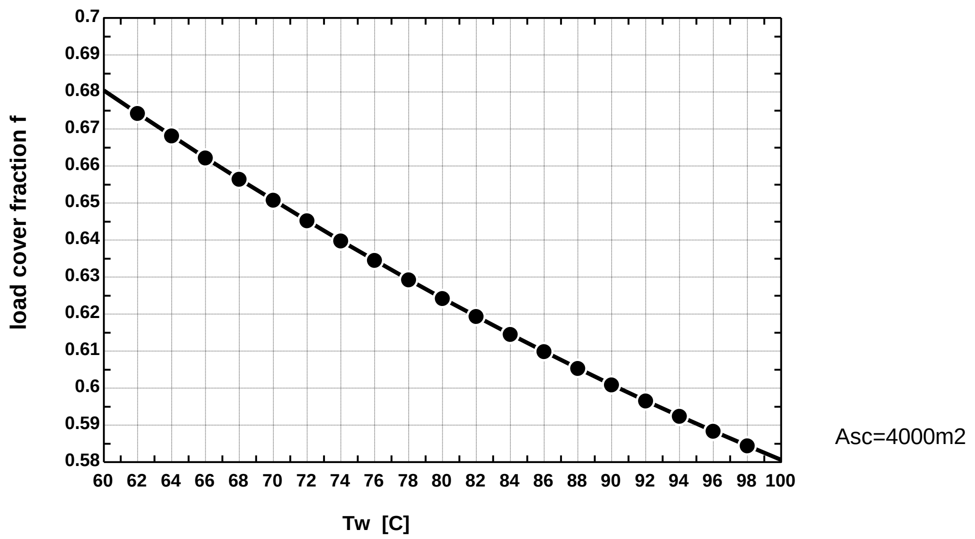



**Figure 25: Variation of load coverage fraction f by the hot water temperature ($A_{SC}$=4000m2)**

If no solar cooling system is examined, the desired temperature of water can be chosen in lower value (for example Tw=80°C) for higher system performance, or for less needs of solar collector's area (reduced size and capital cost of hybrid system).

This is a realistic scenario if we consider the heating processes in the dairy products factory. The most of them require large amounts of heating energy provided by hot water supply in lower than 80°C. Processes in dairy products factory (Figures 26, 27) which are operating under these water temperature conditions are:

- Bottles cleaning: 60°C.
- Pasteurization: 70-80°C.
- Cheese Ripening: 40-45°C.
- CIP (Cleaning-in-Place): 70-80°C.

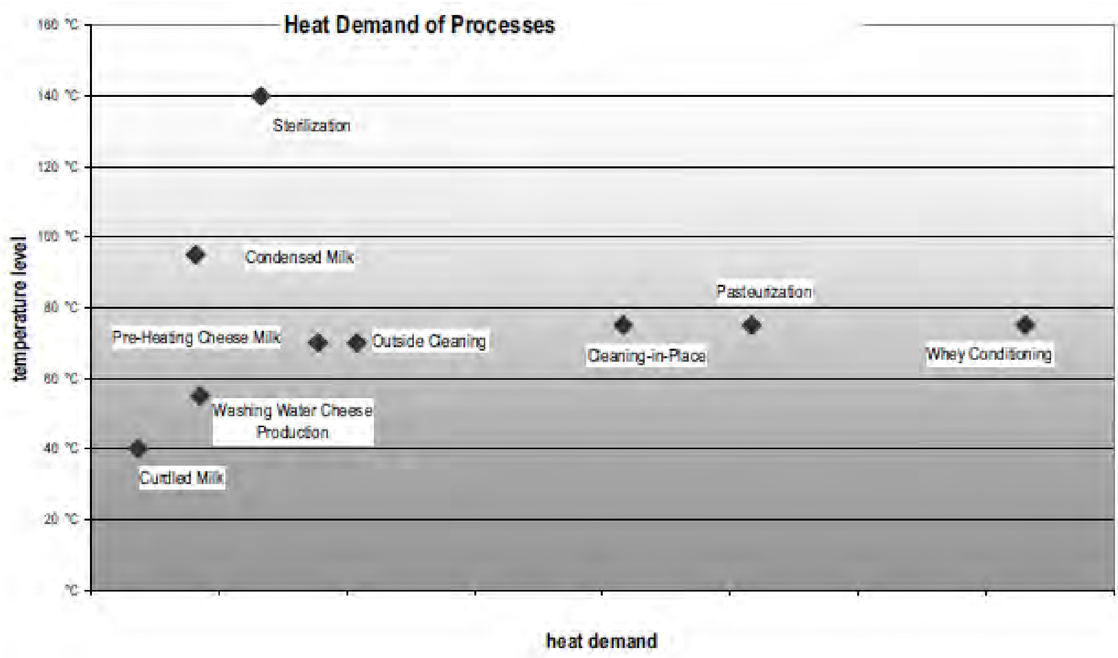



**Figure 26: Classification diagram of dairy products factory's processes relative to temperature level and energy requirements**

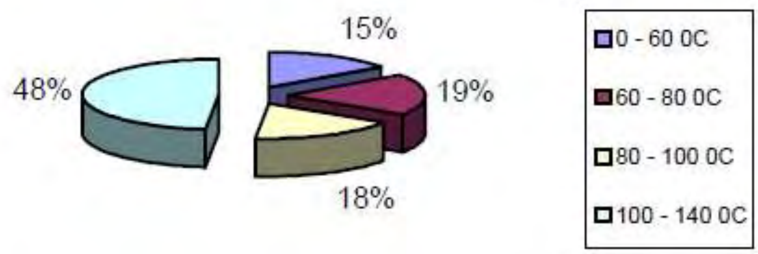



**Figure 27: Distribution of thermal energy requirements relative to the process temperature in diaries**

Considering the case of a hybrid solar system of $A_{SC}$ = 4000m$^2$ with the operation of a 50kW absorption solar chiller, in Figure 28 is plotted the variation of both load coverage fraction f and $COP_{AC}$ by the desired operational water temperature [$T_w$].

The variation of the coefficient of performance [COP] of absorption chiller is precipitate up to $T_w$ =82°C. On the contrary, the variation of load coverage fraction f by the water temperature shows higher gradient for temperatures greater than 80°C. Solar cooling systems (absorption chiller) are claiming higher water driven temperatures for its operation than the solar systems for heating. This means that the cover load function f by the solar system becomes lower i.e it is needed more energy supply by the conventional part of system in order to access an efficient operation of the solar chiller. An acceptable selection band of water temperatures, in the case of installation of cooling solar system assisted by flat plate collectors, is between 84°C - 90°C. For the system studied, this parameter was defined $T_w$=86°C with a performance of Absorption Chiller about $\underline{COP_{AC} \approx 0.739}$.

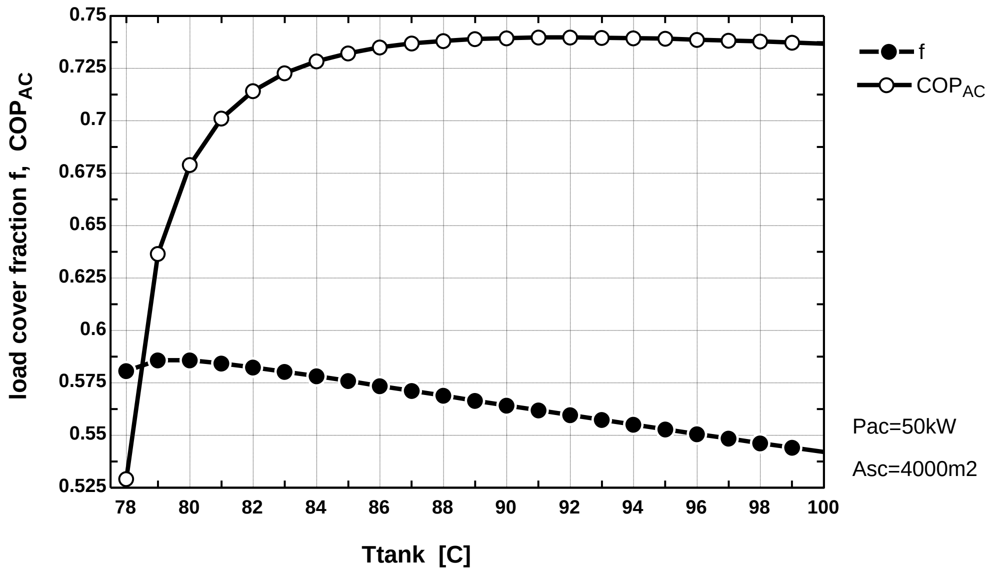



**Figure 28: Variation of the coefficient of performance [COP] of absorption chiller and load cover fraction by the desired hot water temperature**

### 5.2.3 Variation of the NPV by the size of the solar collector's area

The size of the applied solar systems effects to the profitability of the corresponding investment.

In Table 14 are listed simulation results of different sized hybrids plants, where only thermal solar system is considered (y=w=0). The profitability of the system is better when the total thermal load is covering by solar collectors in about f=57% which is achieving with an area of $A_{SC}$=3500m$^2$ (Figure 29).

| $A_{TOTAL}$ | $A_{PV}$ | $A_{SC}$ | f | w | y | x | $P_{AC}$ | $P_{PV}$ | NPV | PP |
|---|---|---|---|---|---|---|---|---|---|---|
| [m$^2$] | [m$^2$] | [m$^2$] | | | | | [kW] | [kW] | [€] | [years] |
| 10 | 0 | 10 | 0.003 | 0 | 0 | 0.997 | 0 | 0 | 6083 | 10.5 |
| 50 | 0 | 50 | 0.014 | 0 | 0 | 0.986 | 0 | 0 | 30142 | 10.6 |
| 100 | 0 | 100 | 0.027 | 0 | 0 | 0.973 | 0 | 0 | 59606 | 10.7 |
| 250 | 0 | 250 | 0.066 | 0 | 0 | 0.934 | 0 | 0 | 144004 | 10.9 |
| 500 | 0 | 500 | 0.128 | 0 | 0 | 0.872 | 0 | 0 | 271763 | 11.3 |
| 1000 | 0 | 1000 | 0.237 | 0 | 0 | 0.763 | 0 | 0 | 481956 | 12.1 |
| 1500 | 0 | 1500 | 0.329 | 0 | 0 | 0.671 | 0 | 0 | 637405 | 13.1 |
| 2000 | 0 | 2000 | 0.406 | 0 | 0 | 0.594 | 0 | 0 | 744936 | 14.0 |
| 2500 | 0 | 2500 | 0.471 | 0 | 0 | 0.529 | 0 | 0 | 811376 | 15.1 |
| 3000 | 0 | 3000 | 0.525 | 0 | 0 | 0.475 | 0 | 0 | 843551 | 16.2 |
| 3500 | 0 | 3500 | 0.571 | 0 | 0 | 0.429 | 0 | 0 | 848287 | 17.3 |
| 4000 | 0 | 4000 | 0.610 | 0 | 0 | 0.390 | 0 | 0 | 832411 | 18.5 |
| 4500 | 0 | 4500 | 0.645 | 0 | 0 | 0.355 | 0 | 0 | 802748 | 19.6 |

**Table 14: Variation of NPV by the size of thermal system (without solar cooling & PV-system)**

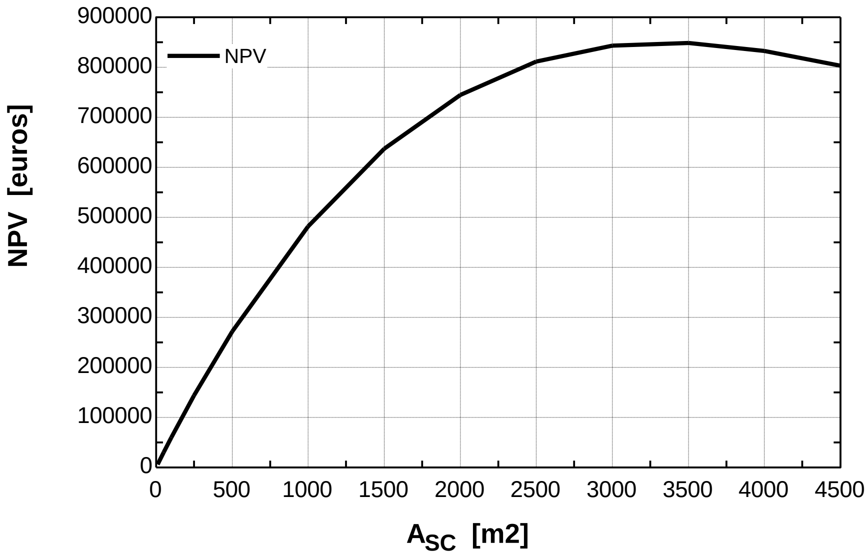



**Figure 29: Variation of NPV by the size of thermal system (without solar cooling & PV-system)**

### 5.2.4 Variation of the NPV by the size of the solar thermal and cooling system

In Table 15 are listed simulation results of different sized hybrids plants, where thermal and cooling solar system is regarding (no PV-system, w=0)). The profitability of the system is decreased increased relevant to the size of the absorption chiller. (Figure 30)

| $A_{TOTAL}$ | $A_{PV}$ | $A_{SC}$ | f | y | z | x | $P_{AC}$ | $P_{PV}$ | NPV | PP |
|---|---|---|---|---|---|---|---|---|---|---|
| [m²] | [m²] | [m²] | | | | | **[kW]** | [kW] | [€] | [years] |
| 3897 | 0 | 3897 | 0.570 | 0.088 | 0.891 | 0.483 | 50 | 0 | 691941 | 19.9 |
| 4100 | 0 | 4100 | 0.570 | 0.132 | 0.845 | 0.509 | 75 | 0 | 613658 | 21.2 |
| 4302 | 0 | 4302 | 0.570 | 0.176 | 0.803 | 0.535 | 100 | 0 | 535342 | 22.6 |
| 4505 | 0 | 4505 | 0.570 | 0.220 | 0.765 | 0.562 | 125 | 0 | 456997 | 23.9 |
| 4708 | 0 | 4708 | 0.570 | 0.264 | 0.731 | 0.588 | 150 | 0 | 378628 | 25.2 |
| 4911 | 0 | 4911 | 0.570 | 0.308 | 0.700 | 0.615 | 175 | 0 | 300238 | 26.5 |
| 5113 | 0 | 5113 | 0.570 | 0.352 | 0.671 | 0.641 | 200 | 0 | 221830 | 27.8 |
| 5519 | 0 | 5519 | 0.570 | 0.440 | 0.620 | 0.694 | 250 | 0 | 64968 | 30.5 |
| 5722 | 0 | 5722 | 0.570 | 0.484 | 0.597 | 0.720 | 275 | 0 | -13483 | 31.8 |
| 5918 | 0 | 5918 | 0.570 | 0.528 | 0.577 | 0.746 | 300 | 0 | -89849 | 33.1 |
| 6160 | 0 | 6160 | 0.570 | 0.616 | 0.546 | 0.787 | 350 | 0 | -198840 | 35.0 |
| 6268 | 0 | 6268 | 0.570 | 0.704 | 0.527 | 0.816 | 400 | 0 | -268539 | 36.2 |
| 6339 | 0 | 6339 | 0.570 | 0.793 | 0.512 | 0.840 | 450 | 0 | -327740 | 37.2 |

**Table 15: Variation of NPV by the size of solar thermal and cooling system (without PV-system)**

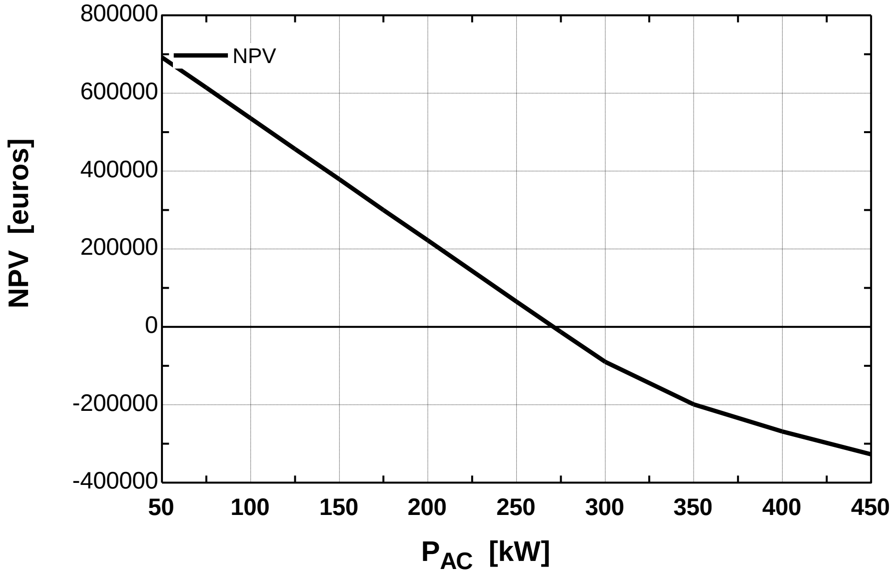



**Figure 30: Variation of NPV by the size of solar thermal and cooling system (without PV-system)**

### 5.2.5 Variation of the NPV by the size of PV area

In Table 17 are listed the simulation results of different sized hybrids plants, where only PV-system is considered (f=y=0). The profitability of the system is increased relevant to the size of PV-system (Figure 31).

| $A_{TOTAL}$ | $A_{PV}$ | $A_{SC}$ | f | w | y | x | $P_{AC}$ | $P_{PV}$ | NPV | PP |
|---|---|---|---|---|---|---|---|---|---|---|
| [m²] | [m²] | [m²] | | | | | **[kW]** | [kW] | [€] | [years] |
| 15 | 15 | 0 | 0.000 | 0.001 | 0.000 | 1.000 | 0 | 1 | 1026 | 17.8 |
| 51 | 51 | 0 | 0.000 | 0.003 | 0.000 | 1.000 | 0 | 3 | 3506 | 17.8 |
| 87 | 87 | 0 | 0.000 | 0.006 | 0.000 | 1.000 | 0 | 5 | 5985 | 17.8 |
| 123 | 123 | 0 | 0.000 | 0.008 | 0.000 | 1.000 | 0 | 8 | 8465 | 17.8 |
| 159 | 159 | 0 | 0.000 | 0.011 | 0.000 | 1.000 | 0 | 10 | 10944 | 17.8 |
| 195 | 195 | 0 | 0.000 | 0.013 | 0.000 | 1.000 | 0 | 12 | 13424 | 17.8 |
| 231 | 231 | 0 | 0.000 | 0.016 | 0.000 | 1.000 | 0 | 14 | 15903 | 17.8 |
| 266 | 266 | 0 | 0.000 | 0.018 | 0.000 | 1.000 | 0 | 17 | 18383 | 17.8 |
| 302 | 302 | 0 | 0.000 | 0.020 | 0.000 | 1.000 | 0 | 19 | 20862 | 17.8 |
| 338 | 338 | 0 | 0.000 | 0.023 | 0.000 | 1.000 | 0 | 21 | 23342 | 17.8 |
| 374 | 374 | 0 | 0.000 | 0.025 | 0.000 | 1.000 | 0 | 23 | 25821 | 17.8 |
| 410 | 410 | 0 | 0.000 | 0.028 | 0.000 | 1.000 | 0 | 26 | 28301 | 17.8 |
| 446 | 446 | 0 | 0.000 | 0.030 | 0.000 | 1.000 | 0 | 28 | 30780 | 17.8 |

**Table 16: Variation of NPV by the size of PV-system (without solar thermal and cooling system)**

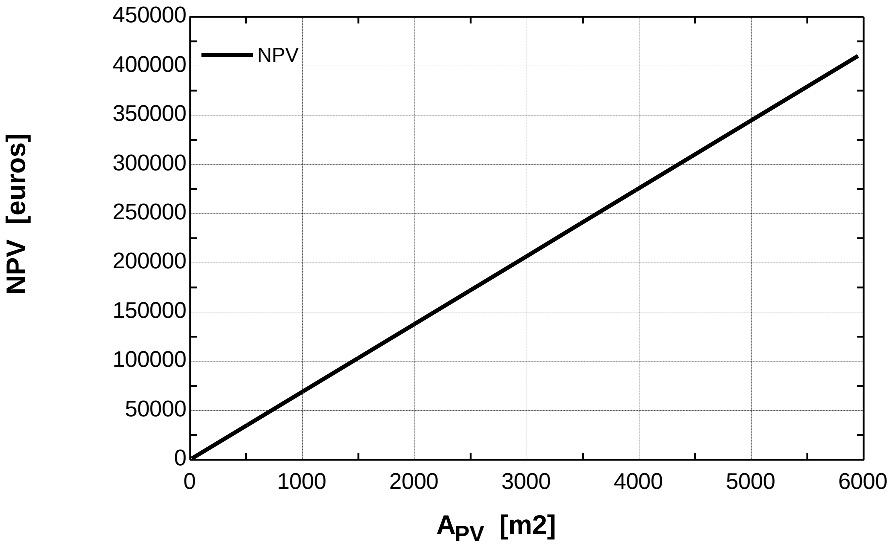



**Figure 31: Variation of NPV by the size of PV-system (without solar thermal and cooling system)**

## 5.3 Optimization Analysis

### 5.3.1 Optimization of total hybrid plant

The optimum size of total plant for different cases of solar cooling system (specific sizes of absorption chiller) is given by maximizing the Net Present Value (financial criterion of an investment) by the optimum variation of the solar collectors and PV-panels surface. The results of the optimization are listed in the Table 18 (the optimum size of the total plant is given in each row of the Table for different sizes of absorption chller).

| $A_{TOTAL}$ [m²] | $A_{SC}$ [m²] | $A_{PV}$ [m²] | f | w | y | $P_{AC}$ [kW] | $P_{PV}$ [kW] | NPV [€] | PP [years] |
|---|---|---|---|---|---|---|---|---|---|
| 18214 | 3341 | 14873 | 0.56 | 1.00 | 0.00 | **0** | 930 | 2859729 | 14.1 |
| 18653 | 3780 | 14873 | 0.56 | 1.00 | 0.09 | **50** | 930 | 3271437 | 13.6 |
| 18871 | 3998 | 14873 | 0.56 | 1.00 | 0.13 | **75** | 930 | 3480070 | 13.4 |
| 19088 | 4215 | 14873 | 0.56 | 1.00 | 0.18 | **100** | 930 | 3688701 | 13.2 |
| 19523 | 4650 | 14873 | 0.57 | 1.00 | 0.26 | **150** | 930 | 4179075 | 12.7 |
| 19958 | 5085 | 14873 | 0.57 | 1.00 | 0.35 | **200** | 930 | 4760356 | 12.1 |
| 20393 | 5520 | 14873 | 0.57 | 1.00 | 0.44 | **250** | 930 | 5396975 | 11.6 |
| 20828 | 5955 | 14873 | 0.57 | 1.00 | 0.53 | **300** | 930 | 6095092 | 11.1 |
| 21266 | 6393 | 14873 | 0.58 | 1.00 | 0.62 | **350** | 930 | 6661285 | 10.8 |
| 21666 | 6793 | 14873 | 0.60 | 1.00 | 0.70 | **400** | 930 | 7075211 | 10.7 |
| 22014 | 7141 | 14873 | 0.61 | 1.00 | 0.79 | **450** | 930 | 7404401 | 10.6 |
| 22286 | 7413 | 14873 | 0.62 | 1.00 | 0.88 | **500** | 930 | 7645102 | 10.6 |
| 22492 | 7619 | 14873 | 0.63 | 1.00 | 0.97 | **550** | 930 | 7810709 | 10.6 |

**Table 17: Optimized sizing parameters of hybrid plant**

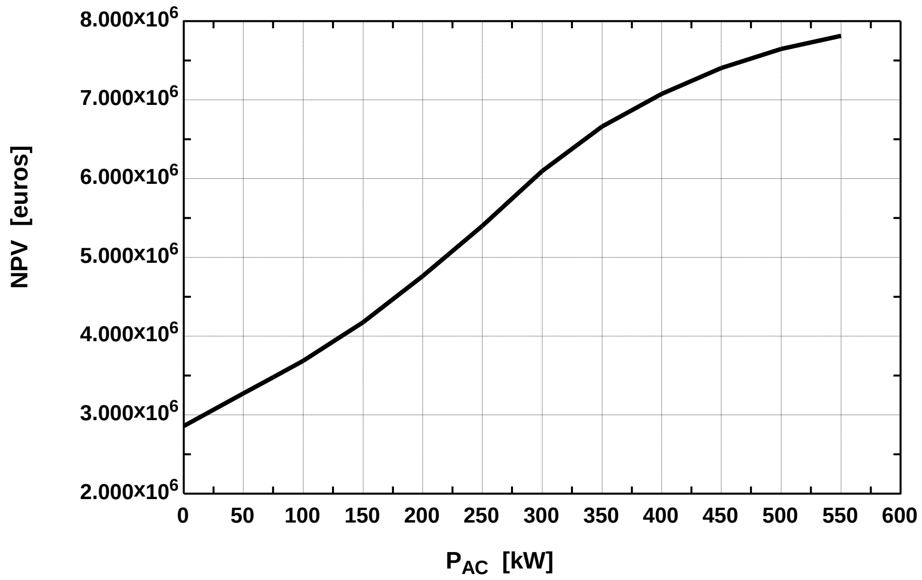



**Figure 32: Optimized NPV relative to the size of cooling part of hybrid plant**

The figure 32 shows how the NPV is maximized for different scenarios of the total power of one stage absorber chiller.

The above results of optimization, postulates large surfaces for solar collectors and PV-system ($A_{TOTAL}$ = 22.492m$^2$). That is not a reasonable proposal since the total outer available surface for installation of solar panels is less than 4.000m$^2$, considering as available for collectors and PV-panels installation surface the roofs of the existing buildings.

### 5.3.2 Optimization of total hybrid plant with total available outer surface of $A_{TOTAL}$= 4000m$^2$

#### ***5.3.2.1 Hybrid solar plant***

In the following Table 19, are presented the optimized values of the system studied, by the maximization of the NPV of investment, through the combined variation of $A_{PV}$ and $A_{SC}$ for discrete values of $P_{AC}$, in order the total solar panels area [$A_{TOTAL}$ = $A_{PV}$ + $A_{SC}$] be less than 4.000m$^2$. Similar to the optimization presented in §5.2.1 each row of table represents an optimization run for the program.

| $A_{TOTAL}$ | $A_{SC}$ | $A_{PV}$ | f | w | y | x | $P_{AC}$ | $P_{PV}$ | NPV | PP |
|---|---|---|---|---|---|---|---|---|---|---|
| [m$^2$] | [m$^2$] | [m$^2$] | | | | | **[kW]** | [kW] | [€] | [years] |
| 4000 | 2081 | 1919 | 0.42 | 0.13 | 0.00 | 0.58 | **0** | 120 | 890725 | 14.8 |
| 4000 | 3060 | 940 | 0.50 | 0.06 | 0.09 | 0.56 | **50** | 59 | 733805 | 18.4 |
| 4000 | 3236 | 764 | 0.50 | 0.05 | 0.13 | 0.59 | **75** | 48 | 641874 | 19.8 |
| 4000 | 3412 | 588 | 0.50 | 0.04 | 0.18 | 0.62 | **100** | 37 | 549941 | 21.3 |
| 4000 | 3765 | 235 | 0.50 | 0.02 | 0.26 | 0.68 | **150** | 15 | 366068 | 24.4 |
| 4000 | 4000 | 0 | 0.50 | 0.00 | 0.35 | 0.75 | **200** | 0 | 181622 | 27.8 |
| 4000 | 4000 | 0 | 0.47 | 0.00 | 0.44 | 0.85 | **250** | 0 | -10398 | 31.8 |
| 4000 | 4000 | 0 | 0.45 | 0.00 | 0.53 | 0.95 | **300** | 0 | -208788 | 37.1 |
| 4000 | 4000 | 0 | 0.44 | 0.00 | 0.62 | 1.03 | **350** | 0 | -367173 | 42.4 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 4000 | 4000 | 0 | 0.43 | 0.00 | 0.70 | 1.08 | **400** | 0 | -483442 | 47.0 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.79 | 1.13 | **450** | 0 | -583145 | 51.5 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.88 | 1.16 | **500** | 0 | -667867 | 55.6 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.97 | 1.18 | **550** | 0 | -739708 | 59.3 |

**Table 18: Optimized sizing parameters of hybrid plant (4.000m²)**

The highest NPV with the total area constraint of 4.000m², it appears when no solar cooling part is installed. The case with the great profit produces a Pay-Back Period equal to 14.8 years. The crude-oil fuel consumption can be limited up to 58% of the present installation. Hybrid solar systems with more than 250kW one-stage, absorption chillers, gives negative NPV and they are not viable ones (for mean cooling load fraction: y>0.4). As the power of absorber chiller becomes higher, larger areas of solar collectors are needed, so the PV-system is limited.

Anyway, the total coverage of the available surface by solar panels (combination of solar collectors and photovoltaic) is suggested.

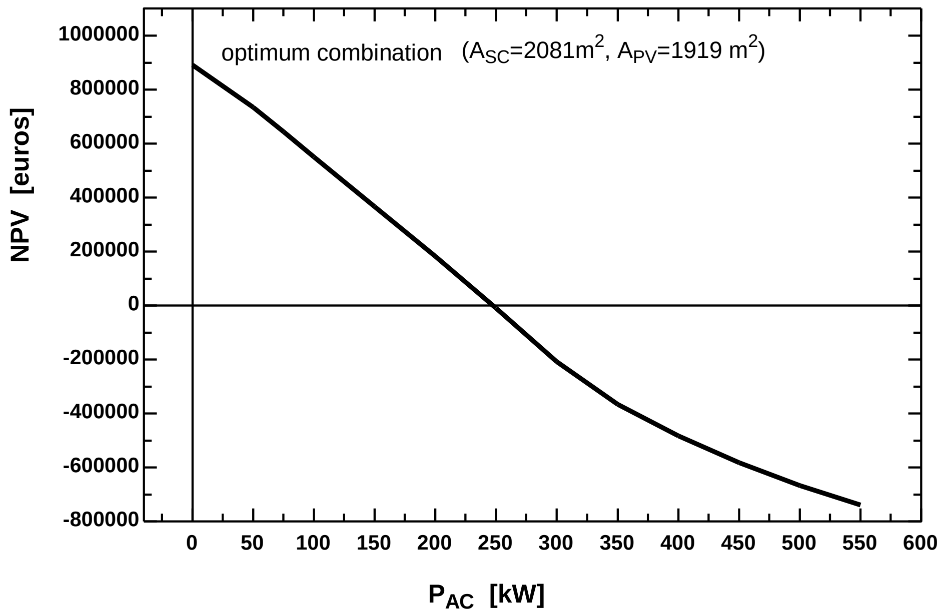



**Figure 33: Optimized NPV relative to the size of absorber of hybrid plant (4.000m²)**

In Figure 33 is drawn the optimized NPV for different sizes of the solar cooling system (size of the absorption chiller).

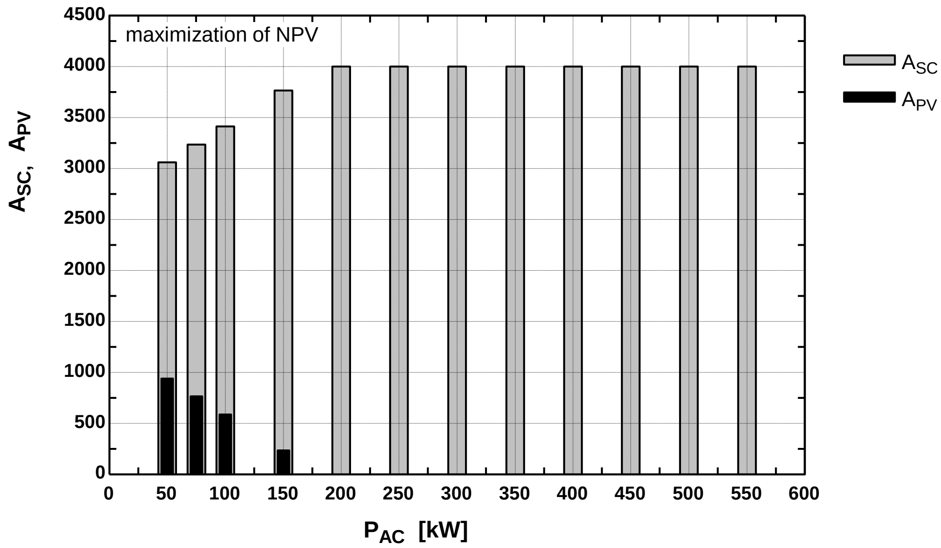



**Figure 34: Optimized occupation of the total available area (4.000m²)**

Figure 34, represents the optimum occupation of the total available area by solar collectors ($A_{SC}$) and PV-panels ($A_{PV}$), maximizing the NPV for different sizes of the absorber chiller ($P_{AC}$).

### *5.3.2.2 Hybrid solar plant – without a PV – system*

Special case of study would be the circumstance of an investment of a hybrid solar plant without the installation of a PV-system. In Table 20 are listed the optimized values of solar collector surface which maximizing the NPV.

| w=0: $P_{PV}$=0, $A_{PV}$=0 | | | | | | | |
|---|---|---|---|---|---|---|---|
| $A_{TOTAL}$ | $A_{SC}$ | f | y | x | $P_{AC}$ | NPV | PP |
| [m²] | [m²] | | | | [kW] | [€] | [years] |
| 3345 | 3345 | 0.56 | 0.00 | 0.44 | 0 | 849342 | 17.0 |
| 3780 | 3780 | 0.56 | 0.09 | 0.49 | 50 | 692483 | 19.7 |
| 3998 | 3998 | 0.56 | 0.13 | 0.52 | 75 | 614049 | 21.1 |
| 4000 | 4000 | 0.55 | 0.18 | 0.56 | 100 | 533886 | 22.1 |
| 4000 | 4000 | 0.52 | 0.26 | 0.66 | 150 | 363622 | 24.6 |
| 4000 | 4000 | 0.50 | 0.35 | 0.75 | 200 | 181622 | 27.8 |
| 4000 | 4000 | 0.47 | 0.44 | 0.85 | 250 | -10399 | 31.8 |
| 4000 | 4000 | 0.45 | 0.53 | 0.95 | 300 | -208788 | 37.1 |
| 4000 | 4000 | 0.44 | 0.62 | 1.03 | 350 | -367173 | 42.4 |
| 4000 | 4000 | 0.43 | 0.70 | 1.08 | 400 | -483442 | 47.0 |
| 4000 | 4000 | 0.42 | 0.79 | 1.13 | 450 | -583145 | 51.5 |
| 4000 | 4000 | 0.42 | 0.88 | 1.16 | 500 | -667867 | 55.6 |
| 4000 | 4000 | 0.42 | 0.97 | 1.18 | 550 | -739708 | 59.3 |

**Table 19: Optimized hybrid solar plant: ($A_{TOTAL}$≤4.000m²), without PV-system**

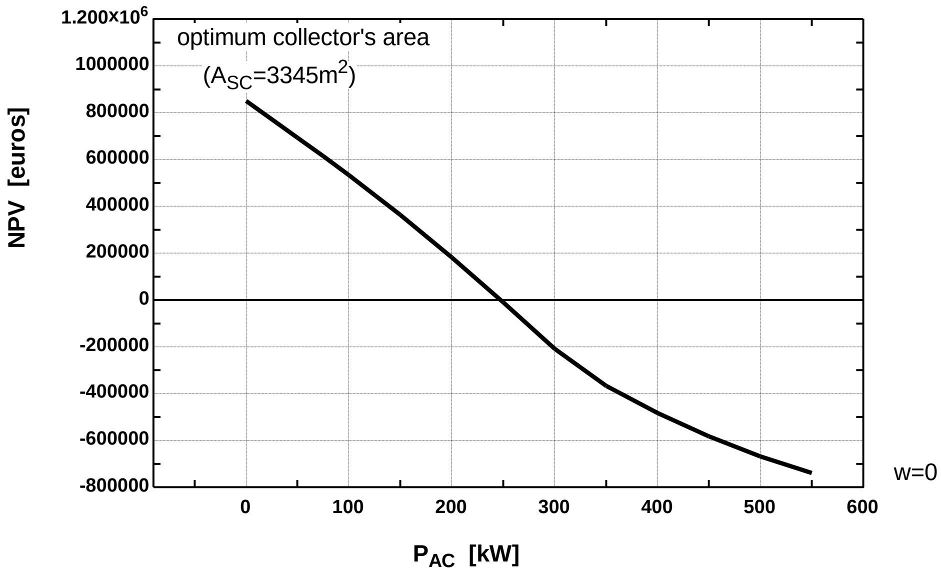



**Figure 35: Optimized NPV relative to the size of cooling part of hybrid plant (4.000m²), without PV-system**

The highest NPV with the constraint of the limitation of the total available area for the solar panels (≤4.000m²) appears again when no solar cooling part is installed. The crude-oil fuel consumption can be limited up to 44% in that case.

Hybrid solar plants with higher than 250kW one-stage, absorption chiller, show negatives NPV and they are not viable. This happens for mean cooling load fraction: y>0.4. As the power of absorber chiller becomes higher, the requirements of solar energy become higher and more area of solar collectors is needed, which is not available, (Figure 35).

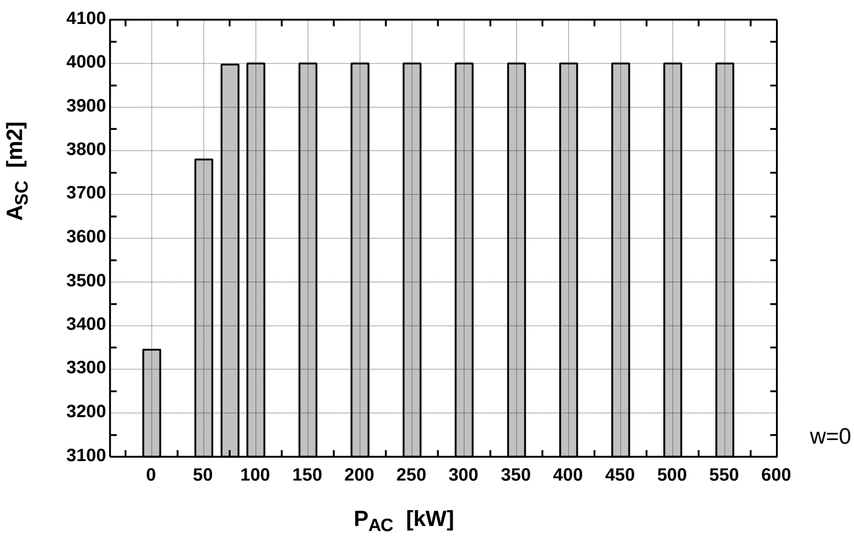



**Figure 36: Optimized occupation of the total available area (4.000m²), without PV-system**

In each case of a cooling installed plant it is suggested the optimum total coverage of the available surface by solar collectors (Figure 36).

#### 5.3.2.3 Case of a subsidy

Although the subsidies are nowadays limited and it is expected that they almost become cut down in the next years, it might be useful a consideration of a case of a subsidy. So it is regarded a case of a subsidy, in which the 40% of the total investment cost is submitted by European Commission and the rest by dairy products factory's own Capitals. A similar subsidy program was in progress till last year.

According the approach of the subsidy for a building with available surface for installation of solar panels (collectors and photovoltaic) equal to 4000m², in the Table 21 are listed the optimum combined values of $A_{PV}$, $A_{SC}$ for certain sizes of solar cooling system (absorber installed power $P_{AC}$) as results of the $NPV_{SUBSIDY}$ maximization.

| $A_{TOTAL}$ | $A_{SC}$ | $A_{PV}$ | f | w | y | x | $P_{AC}$ | $P_{PV}$ | $NPV_{SUBSIDY}$ | $PP_{SUBSIDY}$ |
|---|---|---|---|---|---|---|---|---|---|---|
| [m²] | [m²] | [m²] | | | | | [kW] | [kW] | [€] | [years] |
| 4000 | 3453 | 547 | 0.57 | 0.04 | 0.00 | 0.43 | 0 | 34 | 1371929 | 10.4 |
| 4000 | 4000 | 0 | 0.58 | 0.00 | 0.09 | 0.47 | 50 | 0 | 1243091 | 12.1 |
| 4000 | 4000 | 0 | 0.56 | 0.00 | 0.13 | 0.52 | 75 | 0 | 1175379 | 12.7 |
| 4000 | 4000 | 0 | 0.55 | 0.00 | 0.18 | 0.56 | 100 | 0 | 1104028 | 13.3 |
| 4000 | 4000 | 0 | 0.52 | 0.00 | 0.26 | 0.66 | 150 | 0 | 951266 | 14.8 |
| 4000 | 4000 | 0 | 0.50 | 0.00 | 0.35 | 0.75 | 200 | 0 | 786618 | 16.7 |
| 4000 | 4000 | 0 | 0.47 | 0.00 | 0.44 | 0.85 | 250 | 0 | 611822 | 19.2 |
| 4000 | 4000 | 0 | 0.45 | 0.00 | 0.53 | 0.95 | 300 | 0 | 430546 | 22.4 |
| 4000 | 4000 | 0 | 0.44 | 0.00 | 0.62 | 1.03 | 350 | 0 | 289180 | 25.6 |
| 4000 | 4000 | 0 | 0.43 | 0.00 | 0.70 | 1.08 | 400 | 0 | 189849 | 28.4 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.79 | 1.13 | 450 | 0 | 107013 | 31.1 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.88 | 1.16 | 500 | 0 | 39098 | 33.6 |
| 4000 | 4000 | 0 | 0.42 | 0.00 | 0.97 | 1.18 | 550 | 0 | -15989 | 35.8 |

**Table 20: Optimized sizing parameters of hybrid plant (4.000m²) with investments subsidy**

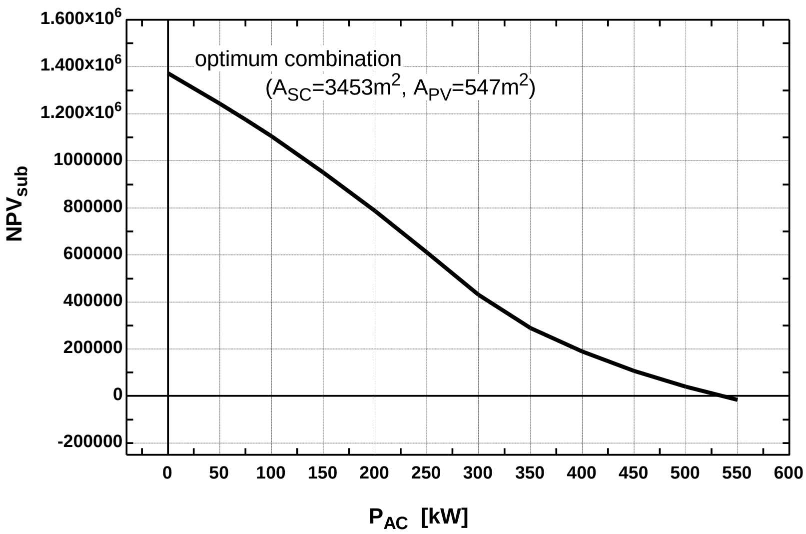



**Figure 37: Optimized NPV relative to the size of cooling part of hybrid plant (4.000m²), with subsidy**

Even in this case, the proposal scenario is a hybrid solar system with no cooling system. The total coverage of the available surface by solar panels (combination of solar collectors and photovoltaic) gives optimum values of the $NPV_{SUBSIDY}$. In Figure 37 is shown the variation of the optimum value of $NPV_{SUBSIDY}$, by the the combined variation of $A_{PV}$ $A_{SC}$ and $P_{AC}$.

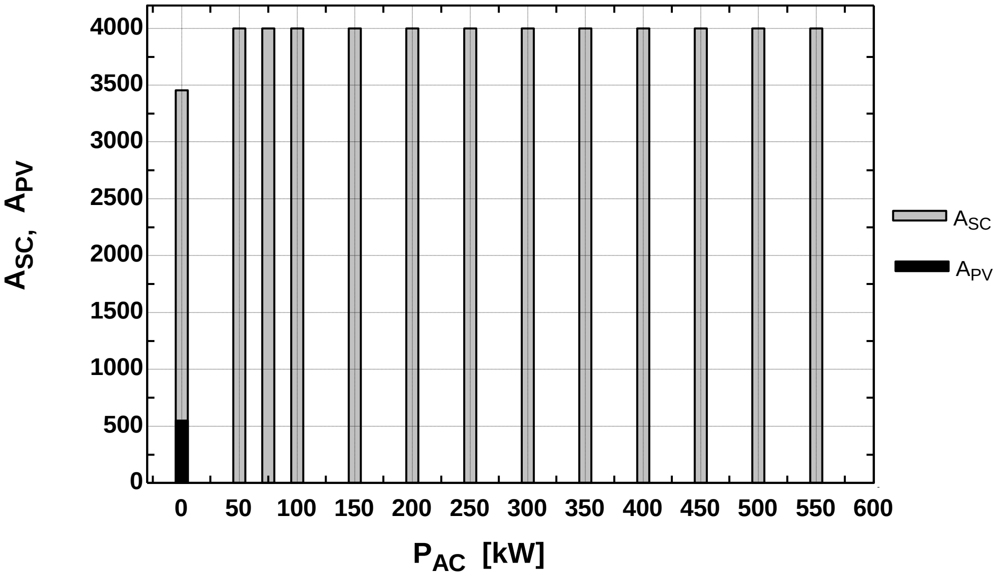



**Figure 38: Optimized occupation of the total available area (4.000m²), with subsidy**

Figure 38, represents the optimum occupation of $A_{PV}$ and $A_{SC}$, for maximum $NPV_{SUBSIDY}$ per different sizes of $P_{AC}$.

## 5.4 Study of different hybrid plants

The four hybrid solar plant scenarios, listed in Table 22, are considered of great interest for further study in simulation:

| SCENARIO | DESCRIPTION |
|---|---|
| **1** | Present plant. |
| **2** | Hybrid solar plant with solar collectors area $A_{SC}$=3345m$^2$, no hybrid solar cooling part (y=0) and with no photovoltaic system (optimum case) $A_{PV}$=0m$^2$. |
| **3** | Hybrid solar plant with solar collectors area $A_{SC}$=2081m$^2$, no hybrid solar cooling part (y=0) and with photovoltaic system (optimum case) $A_{PV}$=1919m$^2$. |
| **4** | Hybrid solar plant with solar collectors area $A_{SC}$=3060m$^2$, hybrid solar cooling part ($P_{AC}$=50kW) and with photovoltaic system (optimum case) $A_{PV}$=940m$^2$. |

**Table 21: Different scenarios in parametric analysis**

The above scenarios are specific cases of the optimization results given previously, except the first which is the present energy state of the dairy products factory.

The solution results for the above different sized cases of study are listed in Table 23.

| | | **SCENARIO** | **SCENARIO** | **SCENARIO** | **SCENARIO** |
|---|---|---|---|---|---|
| **VARIABLE** | **unit** | **1** | **2** | **3** | **4** |
| **$A_{PV}$** | [m$^2$] | 0 | 0 | 1919 | 940 |
| **$A_{SC}$** | [m$^2$] | 0 | 3345 | 2081 | 3060 |
| **$A_{TOTAL}$** | [m$^2$] | 0 | 3345 | 4000 | 4000 |
| **$P_{AC}$** | [kW] | 0 | 0 | 0 | 50 |
| **$P_{PV}$** | [kW] | 0 | 0 | 120 | 59 |
| **$V_{TANK}$** | [m$^3$] | 0 | 167 | 104 | 153 |
| **f** | [%] | 0 | 55.7 | 41.8 | 49.8 |
| **w** | [%] | 0 | 0 | 12.9 | 6.3 |
| **y** | [%] | 0 | 0 | 0 | 8.8 |
| **x** | [%] | 100 | 44.3 | 58.2 | 56.4 |
| **z** | [%] | 100 | 100 | 100 | 89.1 |
| **NPV** | [€] | 0 | 849342 | 890689 | 733805 |
| **PP** | [years] | 0 | 17.0 | 14.8 | 18.4 |

| | | | | | |
|---|---|---|---|---|---|
| **$NPV^{SUBSIDY}$** | [€] | 0 | 1298382 | 1250901 | 1201029 |
| **$PP^{SUBSIDY}$** | [years] | 0 | 10.2 | 8.9 | 11.1 |
| **$C_K$** | [€] | 0 | 1122600 | 900529 | 1168059 |
| **$C_K^{SUBSIDY}$** | [€] | 0 | 673560 | 540317 | 700836 |
| **$GAIN_{annual}$** | [€] | 0 | 65842 | 60381 | 63091 |
| **PEC** | [€] | 0 | 748400 | 600352 | 778706 |
| **$PEC_{AC}$** | [€] | 0 | 0 | 0 | 20000 |
| **$PEC_{CTOWER}$** | [€] | 0 | 0 | 0 | 7046 |
| **$PEC_{EXCANGER}$** | [€] | 0 | 26477 | 21488 | 27694 |
| **$PEC_{TANK}$** | [€] | 0 | 119823 | 74545 | 109614 |
| **$PEC_{SC}$** | [€] | 0 | 602100 | 374580 | 550800 |
| **$PEC_{PV}$** | [€] | 0 | 0 | 129740 | 63552 |
| **$Z_K$** | [€] | 207485 | 141644 | 147104 | 144395 |
| **$Z_K^{FUEL}$** | [€] | 120606 | 53402 | 70247 | 67994 |
| **$Z_K^{ELE}$** | [€] | 86876 | 86879 | 75669 | 74945 |

**Table 22: Comparative parametric solution results per different sized plants**

By comparative plots of the Net Present Value [NPV] (Figure 39) and the Pay-back Period [PP] (Figure 40), it is obvious that the third scenario of the different sized plants is the most profitable investment.

.

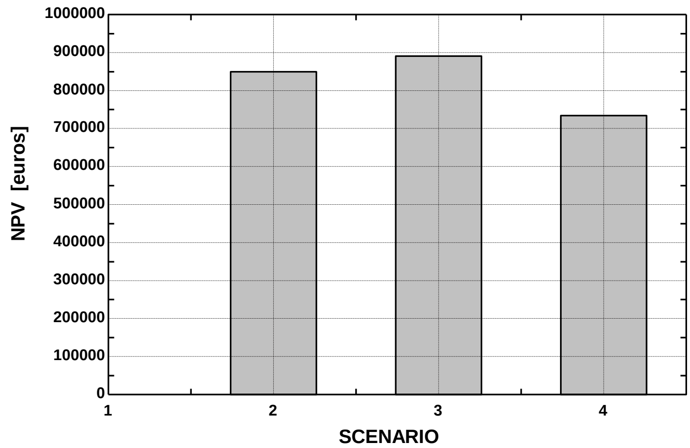



**Figure 39: Comparative NPV plots per different sized plants**

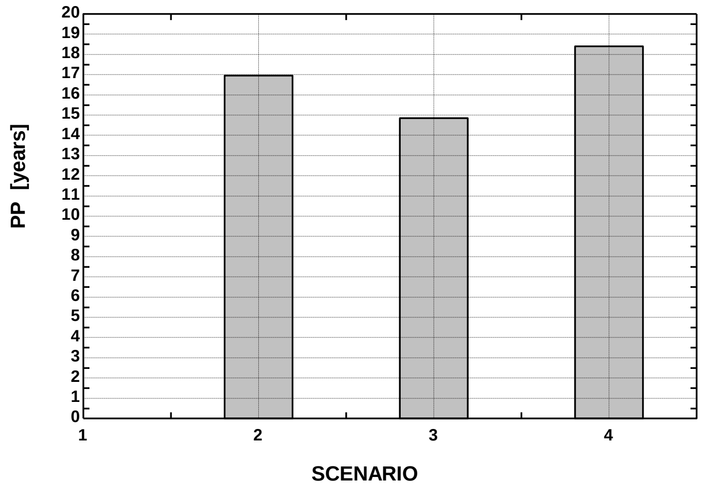



**Figure 40: Comparative Pay-back Period [PP] plots per different sized plants**

Plots of characterized costs and incomes of a 20-years financial life per each scenario are given in the following figure 41.

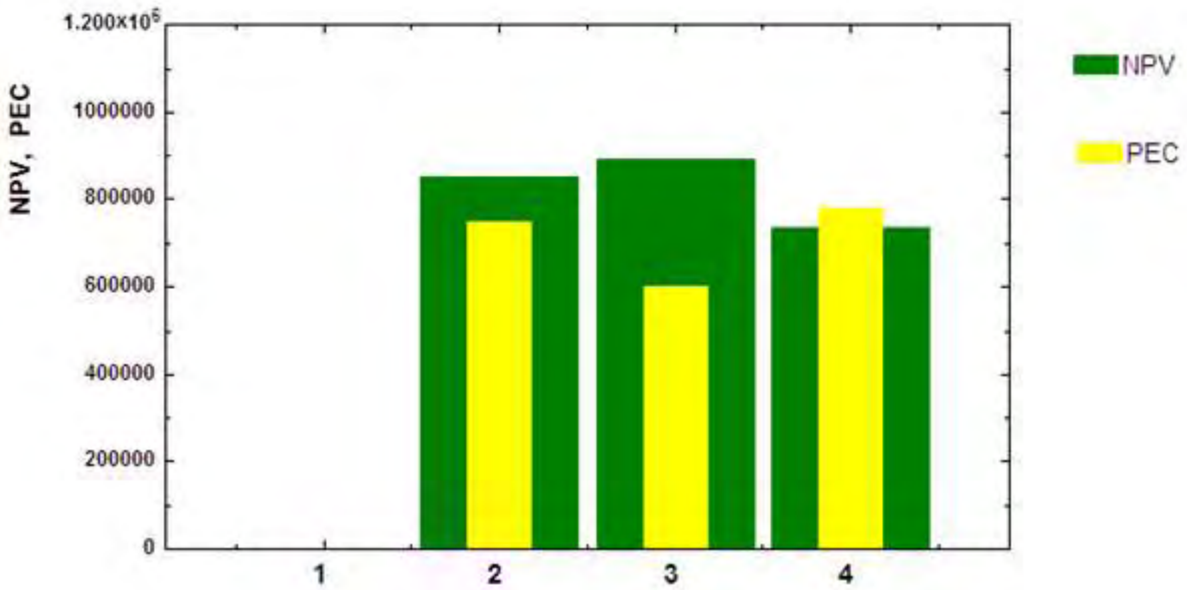



**Figure 41: Characterized costs and incomes per each scenario**

| SCENARIO 1: | **Present plant (no solar plant)** | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| MONTH | JAN | FEB | MAR | APR | MAY | JUN | JUL | AUG | SEP | OCT | NOV | DEC | SUM |
| $Q_{TH,i}$ | 198535 | 194314 | 182162 | 168222 | 162364 | 161563 | 161563 | 161563 | 161563 | 162871 | 174770 | 194314 | 2.084E+06 |
| $Q_{COOL,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1.751E+06 |
| $Q_{ELE,i}$ | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 425100 |
| $m^{0}_{fuel,i}$ | 33796 | 33078 | 31009 | 28636 | 27639 | 27503 | 27503 | 27503 | 27503 | 27725 | 29751 | 33078 | 354724 |
| $Q^{0}_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.126E+06 |
| HOURS OF SUNSHINE | 9.5 | 10.0 | 10.9 | 12.5 | 13.6 | 14.6 | 14.8 | 13.9 | 12.7 | 11.7 | 10.1 | 9.5 | 143.9 |
| $L_{SC,i}$ | 220594 | 215904 | 202402 | 186913 | 180404 | 179514 | 179514 | 179514 | 179514 | 180968 | 194189 | 215904 | 2315338 |
| $f_i$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{out}_{SC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{in}_{CONV,i}$ | 370747 | 362865 | 340172 | 314140 | 303201 | 301705 | 301705 | 301705 | 301705 | 304148 | 326368 | 362865 | 3891324 |
| $Q^{out}_{CONV,i}$ | 220594 | 215904 | 202402 | 186913 | 180404 | 179514 | 179514 | 179514 | 179514 | 180968 | 194189 | 215904 | 2315338 |
| $m_{fuel,i}$ | 33796 | 33078 | 31009 | 28636 | 27639 | 27503 | 27503 | 27503 | 27503 | 27725 | 29751 | 33078 | 354724 |
| $Q^{in}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{out}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{in}_{MC,i}$ | 45998 | 45571 | 49152 | 57722 | 65488 | 75642 | 78272 | 78986 | 66496 | 52866 | 40928 | 43298 | 700419 |
| $Q^{out}_{MC,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1751048 |
| $Q^{out}_{PV,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1125519 |
| $Q^{byGRID}_{ELE,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.13E+06 |
| $Q^{toGRID}_{ELE,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**Table 23: Parametric simulation results in monthly base for Scenario 1**

| **SCENARIO 2:** | Hybrid solar plant with solar collectors area ASC=3345m2, no hybrid solar cooling part (y=0) and with no photovoltaic system (optimum case) APV=0m2. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **MONTH** | **JAN** | **FEB** | **MAR** | **APR** | **MAY** | **JUN** | **JUL** | **AUG** | **SEP** | **OCT** | **NOV** | **DEC** | **SUM** |
| $Q_{TH,i}$ | 198535 | 194314 | 182162 | 168222 | 162364 | 161563 | 161563 | 161563 | 161563 | 162871 | 174770 | 194314 | 2.084E+06 |
| $Q_{COOL,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1.751E+06 |
| $Q_{ELE,i}$ | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 425100 |
| $m^{0}_{fuel,i}$ | 33796 | 33078 | 31009 | 28636 | 27639 | 27503 | 27503 | 27503 | 27503 | 27725 | 29751 | 33078 | 354724 |
| $Q^{0}_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.126E+06 |
| **HOURS OF SUNSHINE** | 9.5 | 10.0 | 10.9 | 12.5 | 13.6 | 14.6 | 14.8 | 13.9 | 12.7 | 11.7 | 10.1 | 9.5 | 143.9 |
| $L_{SC,i}$ | 220594 | 215904 | 202402 | 186913 | 180404 | 179514 | 179514 | 179514 | 179514 | 180968 | 194189 | 215904 | 2315338 |
| $f_{i}$ | 0.24 | 0.29 | 0.52 | 0.67 | 0.78 | 0.83 | 0.88 | 0.86 | 0.74 | 0.57 | 0.33 | 0.19 | 0.56 |
| $Q^{out}_{SC,i}$ | 52908 | 62642 | 105372 | 124639 | 140429 | 149438 | 158146 | 154289 | 132514 | 103897 | 63814 | 42066 | 1290153 |
| $Q^{in}_{CONV,i}$ | 281825 | 257584 | 163076 | 104664 | 67185 | 50549 | 35914 | 42396 | 78992 | 129531 | 219117 | 292166 | 1722999 |
| $Q^{out}_{CONV,i}$ | 167686 | 153262 | 97030 | 62275 | 39975 | 30077 | 21369 | 25226 | 47000 | 77071 | 130374 | 173839 | 1025184 |
| $m_{fuel,i}$ | 25691 | 23481 | 14866 | 9541 | 6124 | 4608 | 3274 | 3865 | 7201 | 11808 | 19974 | 26633 | 157065 |
| $Q^{in}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{out}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{in}_{MC,i}$ | 45998 | 45571 | 49152 | 57722 | 65488 | 75642 | 78272 | 78986 | 66496 | 52866 | 40928 | 43298 | 700419 |
| $Q^{out}_{MC,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1751048 |
| $Q^{out}_{PV,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1125519 |
| $Q^{byGRID}_{ELE,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.13E+06 |
| $Q^{toGRID}_{ELE,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**Table 24: Parametric simulation results in monthly base for Scenario 2**

| SCENARIO 3: | **Hybrid solar plant with solar collectors area ASC=2081m2, no hybrid solar cooling part (y=0) and with photovoltaic system (optimum case) APV=1919m2.** | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| MONTH | JAN | FEB | MAR | APR | MAY | JUN | JUL | AUG | SEP | OCT | NOV | DEC | SUM |
| $Q_{TH,i}$ | 198535 | 194314 | 182162 | 168222 | 162364 | 161563 | 161563 | 161563 | 161563 | 162871 | 174770 | 194314 | 2.084E+06 |
| $Q_{COOL,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1.751E+06 |
| $Q_{ELE,i}$ | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 425100 |
| $m^{0}_{fuel,i}$ | 33796 | 33078 | 31009 | 28636 | 27639 | 27503 | 27503 | 27503 | 27503 | 27725 | 29751 | 33078 | 354724 |
| $Q^{0}_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.126E+06 |
| HOURS OF SUNSHINE | 9.5 | 10.0 | 10.9 | 12.5 | 13.6 | 14.6 | 14.8 | 13.9 | 12.7 | 11.7 | 10.1 | 9.5 | 143.9 |
| $L_{SC,i}$ | 220594 | 215904 | 202402 | 186913 | 180404 | 179514 | 179514 | 179514 | 179514 | 180968 | 194189 | 215904 | 2315338 |
| $f_{i}$ | 0.15 | 0.19 | 0.37 | 0.49 | 0.61 | 0.66 | 0.72 | 0.7 | 0.57 | 0.41 | 0.21 | 0.12 | 0.42 |
| $Q^{out}_{SC,i}$ | 33474 | 40685 | 74241 | 92470 | 109270 | 118975 | 129691 | 125217 | 101525 | 74377 | 41381 | 25472 | 966778 |
| $Q^{in}_{CONV,i}$ | 314488 | 294486 | 215397 | 158728 | 119554 | 101747 | 83736 | 91257 | 131075 | 179144 | 256820 | 320055 | 2266486 |
| $Q^{out}_{CONV,i}$ | 187120 | 175219 | 128161 | 94443 | 71134 | 60540 | 49823 | 54298 | 77989 | 106591 | 152808 | 190433 | 1348559 |
| $m_{fuel,i}$ | 28668 | 26845 | 19635 | 14469 | 10898 | 9275 | 7633 | 8319 | 11948 | 16330 | 23411 | 29175 | 206608 |
| $Q^{in}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{out}_{AC,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| $Q^{in}_{MC,i}$ | 45998 | 45571 | 49152 | 57722 | 65488 | 75642 | 78272 | 78986 | 66496 | 52866 | 40928 | 43298 | 700419 |
| $Q^{out}_{MC,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1751048 |
| $Q^{out}_{PV,i}$ | 7172 | 8048 | 11502 | 14512 | 16072 | 16551 | 17031 | 15952 | 13553 | 11322 | 7784 | 5721 | 145220 |
| $Q_{GRID,i}$ | 74251 | 72948 | 73075 | 78635 | 84842 | 94515 | 96666 | 98459 | 88368 | 76969 | 68569 | 73002 | 980299 |
| $Q^{byGRID}_{ELE,i}$ | 74251 | 72948 | 73075 | 78635 | 84842 | 94515 | 96666 | 98459 | 88368 | 76969 | 68569 | 73002 | 980299 |
| $Q^{toGRID}_{ELE,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**Table 25: Parametric simulation results in monthly base for Scenario 3**

| SCENARIO 4: | **Hybrid solar plant with solar collectors area ASC=3060m2, hybrid solar cooling part (PAC=50kW) and with photovoltaic system (optimum case) APV=940m2.** | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **MONTH** | **JAN** | **FEB** | **MAR** | **APR** | **MAY** | **JUN** | **JUL** | **AUG** | **SEP** | **OCT** | **NOV** | **DEC** | **SUM** |
| $Q_{TH,i}$ | 198535 | 194314 | 182162 | 168222 | 162364 | 161563 | 161563 | 161563 | 161563 | 162871 | 174770 | 194314 | 2.084E+06 |
| $Q_{COOL,i}$ | 114996 | 113928 | 122880 | 144305 | 163721 | 189104 | 195681 | 197465 | 166239 | 132164 | 102320 | 108245 | 1.751E+06 |
| $Q_{ELE,i}$ | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 35425 | 425100 |
| $m^{0}_{fuel,i}$ | 33796 | 33078 | 31009 | 28636 | 27639 | 27503 | 27503 | 27503 | 27503 | 27725 | 29751 | 33078 | 354724 |
| $Q^{0}_{GRID,i}$ | 81423 | 80996 | 84577 | 93147 | 100913 | 111067 | 113697 | 114411 | 101921 | 88291 | 76353 | 78723 | 1.126E+06 |
| **HOURS OF SUNSHINE** | 9.5 | 10.0 | 10.9 | 12.5 | 13.6 | 14.6 | 14.8 | 13.9 | 12.7 | 11.7 | 10.1 | 9.5 | 143.9 |
| $L_{SC,i}$ | 244254 | 239564 | 226061 | 210572 | 204064 | 203174 | 203174 | 203174 | 203174 | 204627 | 217848 | 239564 | 2599247 |
| $f_i$ | 0.2 | 0.24 | 0.45 | 0.59 | 0.7 | 0.76 | 0.82 | 0.79 | 0.66 | 0.5 | 0.27 | 0.16 | 0.5 |
| $Q^{out}_{SC,i}$ | 48715 | 58393 | 102235 | 124377 | 143776 | 154858 | 166296 | 161384 | 134600 | 101878 | 59466 | 37953 | 1293931 |
| $Q^{in}_{CONV,i}$ | 328636 | 304488 | 208112 | 144867 | 101324 | 81202 | 61979 | 70234 | 115249 | 172688 | 266189 | 338841 | 2193808 |
| $Q^{out}_{CONV,i}$ | 195538 | 181170 | 123826 | 86196 | 60288 | 48315 | 36877 | 41789 | 68573 | 102749 | 158382 | 201611 | 1305316 |
| $m_{fuel,i}$ | 29958 | 27756 | 18971 | 13206 | 9236 | 7402 | 5650 | 6402 | 10506 | 15742 | 24265 | 30888 | 199983 |
| $Q^{in}_{AC,i}$ | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 23659 | 283909 |
| $Q^{out}_{AC,i}$ | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 17389 | 154196 |
| $Q^{in}_{MC,i}$ | 39043 | 38616 | 42197 | 50767 | 58533 | 68686 | 71317 | 72031 | 59540 | 45910 | 33973 | 36343 | 616954 |
| $Q^{out}_{MC,i}$ | 97607 | 96539 | 105491 | 126916 | 146332 | 171715 | 178292 | 180076 | 148850 | 114775 | 84931 | 90856 | 1542385 |
| $Q^{out}_{PV,i}$ | 3513 | 3942 | 5634 | 7109 | 7873 | 8108 | 8343 | 7814 | 6639 | 5546 | 3813 | 2802 | 71135 |
| $Q_{GRID,i}$ | 70955 | 70099 | 71987 | 79083 | 86085 | 96004 | 98399 | 99642 | 88326 | 75789 | 65585 | 68965 | 970919 |
| $Q^{byGRID}_{ELE,i}$ | 70955 | 70099 | 71987 | 79083 | 86085 | 96004 | 98399 | 99642 | 88326 | 75789 | 65585 | 68965 | 970919 |
| $Q^{toGRID}_{ELE,i}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**Table 26: Parametric simulation results in monthly base for Scenario 4**

# CHAPTER 6:

# CONCLUDING REMARKS

This study demonstrates a general methodology for the modelling of the installation of a hybrid solar energy plant in a dairy products Factory, in order to cover its energy requirements (for heating, cooling and electricity providing). A simplified model was developed in which the basic systems of the plant (e.g. the solar thermal system, the solar cooling system, the conventional thermal system etc) were grouped into distinct energy units. Each of the unit was modelled and simulated by separate subprograms considering the form and the technology of its equipment. Dimensionless critical coefficients, characteristic to the size of the units were defined in order to simplify the simulation and mainly to control the degrees of freedom of the applied model during optimization analysis (appropriate sizing of equipment). Consecutively it was possible the formulation of the basic equations for the interfacing between the individual groups.

The model applied to an existing dairy products factory [Agricultural Corporations Union of Volos (EVOL)], which lies in the A' industrial Area of Volos in Greece. Its main feature is that it can be adapted to similar feasibility analysis of hybrids for different types of buildings, climate conditions or selective technologies considering the particularities of each case. By this point of view, it is expected that, the results and discussion remarks of this study could be a useful when similar researches are elaborated in the future.

The solar hybrid plant was designed, modelled and simulated, considering the type of energy requirements should be covered, meteorological data of specific region, the availability and the penetration of solar systems in nowadays market and the deliberated study of relative literature.

The solar technology which was selected in the application is: flat-plate with selectable surface solar collectors, a heat exchanger, a heat storage tank, a backup heat source (existing crude-oil burner), a LiBr-H2O one-stage absorption chiller in combination with the existing mechanical one, a water cooling tower and an on-grid connected PV-system.

A detailed simulation model for the behaviour of the complete hybrid plant was developed in EES – software environment, taking into account the thermodynamic and financial principles and acknowledged scientific methods of analysis (e.g. f-Chart method for solar collectors). The model evaluates the dairy products factory's energy requirements, calculates the efficiency, exploitation values like the Coefficient of Performance of the absorber chiller, the solar fraction of load covering by the solar thermal system etc., along with financial key figures of energy savings, investment and annual running costs, NPV of investment etc. The simulation was performed in mean monthly average values in order to overcome the objections of the strong dependency of complete system by the fluctuations of loads, solar radiation and other climate data with time.

An analytical discussion about the affection of the size and special operation conditions of main components (solar collectors, water storage tank, absorber chiller etc) of the system designed to its performance and to the relative investment's profitability was carried out. The optimized size of the system studied, was approached through simulation results and optimization analysis.

It was proven that the designed solar hybrid plant can be viable for providing heating, cooling and electricity. Optimization shows that the maximization of investment's NPV (criterion that maximizes the financial profits) occurs when the load cover fraction of solar [f] is 63%, the PV-system covers the electricity load with a fraction [w] equal to 100% and the cooling load coverage [y] by solar is of 97%, annually. However, the size of the hybrid plant given by this optimization is not reliable since such systems require extremely expanded outer surfaces available for the installation of the collectors and PV-panels. The optimum size of the system postulates 14.873m$^2$ for PV-panels, 7.619m$^2$ for solar collectors and in total is required an outer surface of 22.492m$^2$, which is impracticable for the application.

Regarding the maximum available surface for the installation of solar collectors and PV-panels, which for the specific dairy products factory was taken as 4.000m$^2$, the optimization shows that the appropriated size of the designed system occurs when the load cover fraction of solar [f] is 42%, the PV-system covers the electricity load with a fraction [w] equal to 13% and no solar cooling system is mentioned. The optimum size of the system, with the constraint of 4.000m$^2$ as total available surface, gives 1.919m$^2$ for PV-panels, 2.081m$^2$ for solar collectors and in total the available surface is oversubscribed. These values can changed if a more efficient, solar collector type is used instead of flat-plate with selectable surface.

A hybrid solar system which contains an absorber chiller can be viable for certain sizes of solar collectors and PV-panels field, but this scenario is not the one, which maximizes the NPV. Detailed optimization results for different sizes of solar cooling were demonstrated. Anyway, it was found that the size of the absorber chiller should be less than 250KW.

Other parameters which affect the efficiency and furthermore the profitability of the system are the appropriate size (volume) of hot water storage tank and the operation temperature of hot water in which it is supplied to the load. The optimum relative size of the volume of the heat storage tank per m$^2$ of solar collector area is determined by the variation of the energy load coverage, for constant total solar collector surface. In case of an absorber chiller installation the drive temperature of water should be greater than 80°C. This means that the solar thermal plant is operating with lower f [fraction of load coverage] or that the conventional part of hybrid should provide the additional needs for water heating. If no solar cooling part is examined, considering the driving temperatures of heating processes in the factory, then the total hybrid system could operates under lower temperature conditions with higher load coverage fraction [f].

Finally, the simulation results in annual and monthly base about four different scenarios of energy system for the dairy products factory were adduced. The first of them was the current existing state of the application study available for further comparison discussion. The optimum case is a Hybrid solar plant with solar collectors area of $A_{SC}$=2081m$^2$, no hybrid solar cooling part (y=0) and with a 120kW photovoltaic system ($A_{PV}$=1919m$^2$) – NPV=890.689€ and PP=14.8years.

It is obvious that the above mentioned results would been changed if other solar technologies of solar collectors (eg. vacuum tube or parabolic collectors) or solar chiller (two-stages absorption chiller, adsorption chiller) were selected to be modelled. The comparison of different solar energy technologies or the usage of different structure of the hybrid system would be some very interesting tasks for future work concerning the deep study and analysis of Solars.

Concluding, this study was focalized in the financial profitability and better performance of the hybrid solar system plant. However, these systems are characterized by benefits such as the lowering of harmful emissions or environmentally friendlier operation, which are more respectable than the evident economic profits. The main reasons are that the financial profitability is still unfortunately the strongest criterion for an investment applying and that the evaluation of these benefits is not sufficiently possible. In any case it would be worthy an environmentally driven evaluation for the hybrid solar plant installations.

# APPENDIX A:

## Additional FIGURES

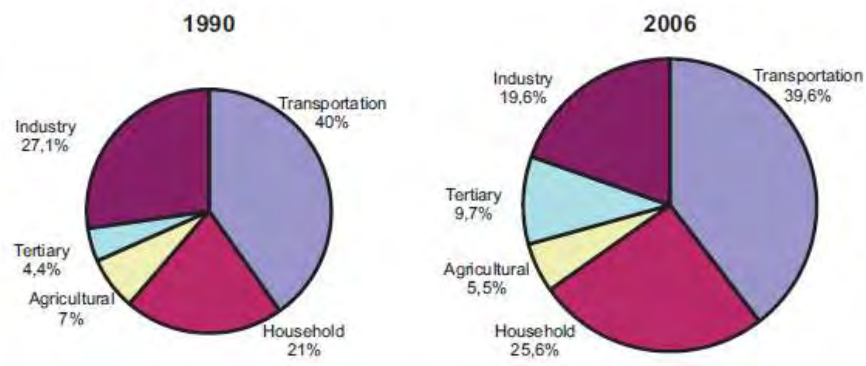



**Figure 42: Final energy consumption by sector, in Greece**

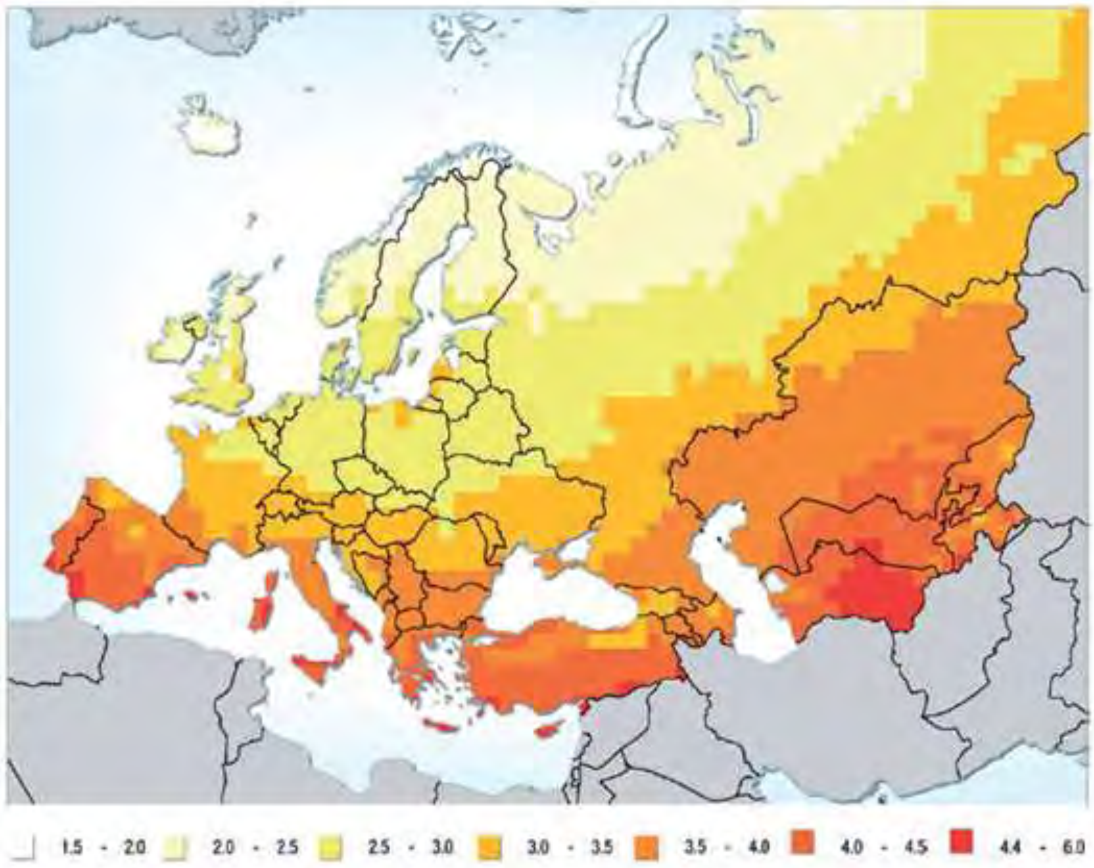



**Figure 43: Solar Radiation in Europe – Source: GRID-Arendal**

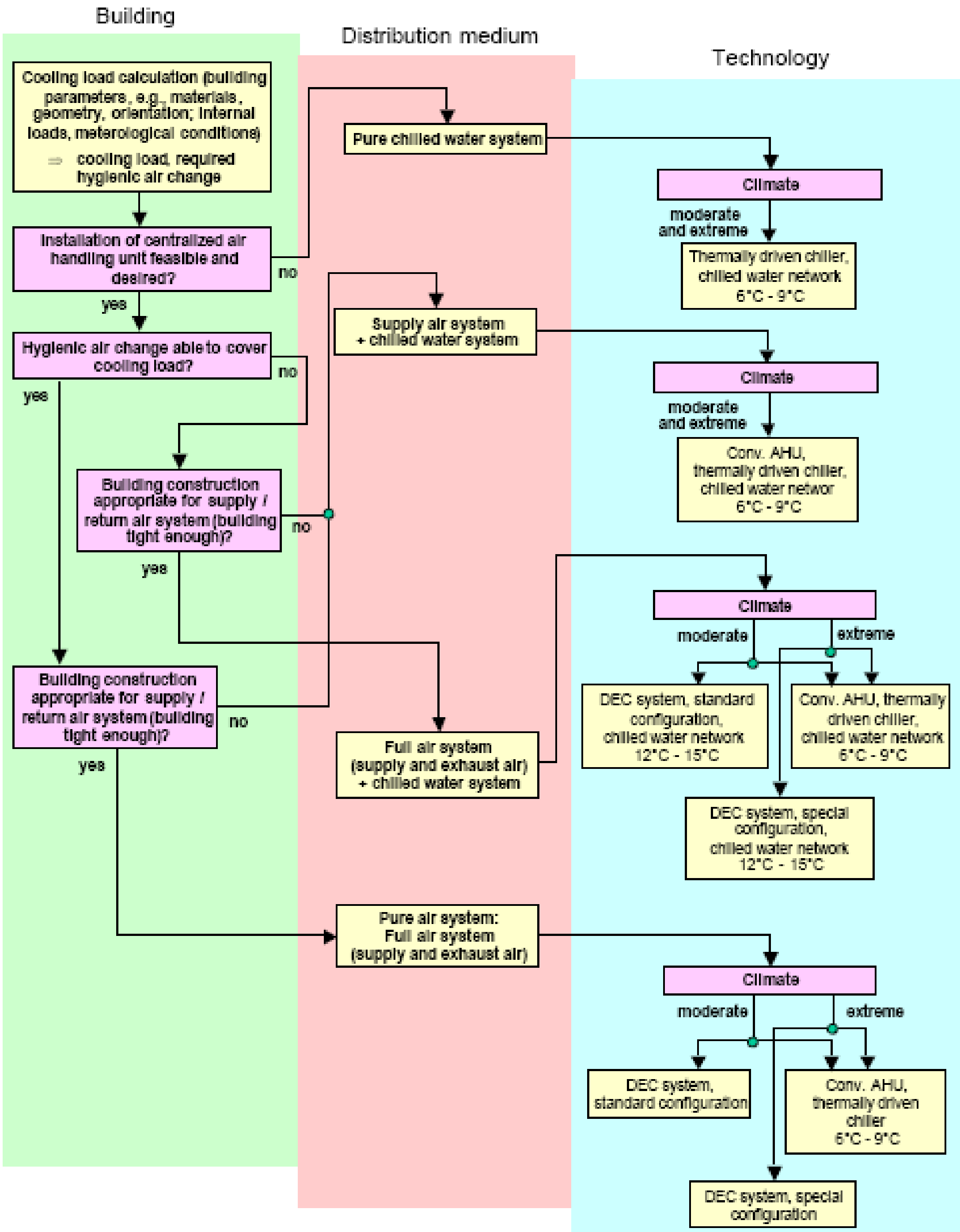



**Figure 44: Basic scheme for decision guidance [2]**

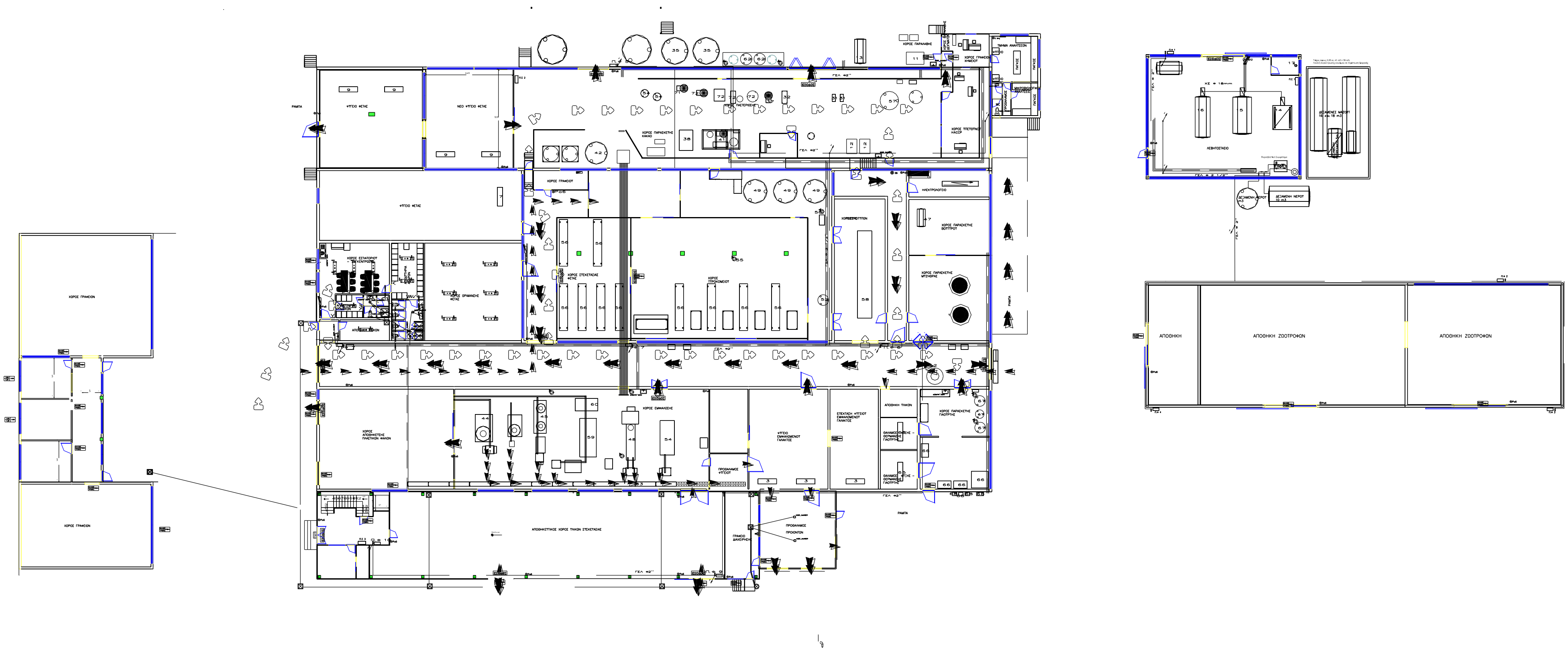


**Figure 45: Dairy Products Factory Layout**

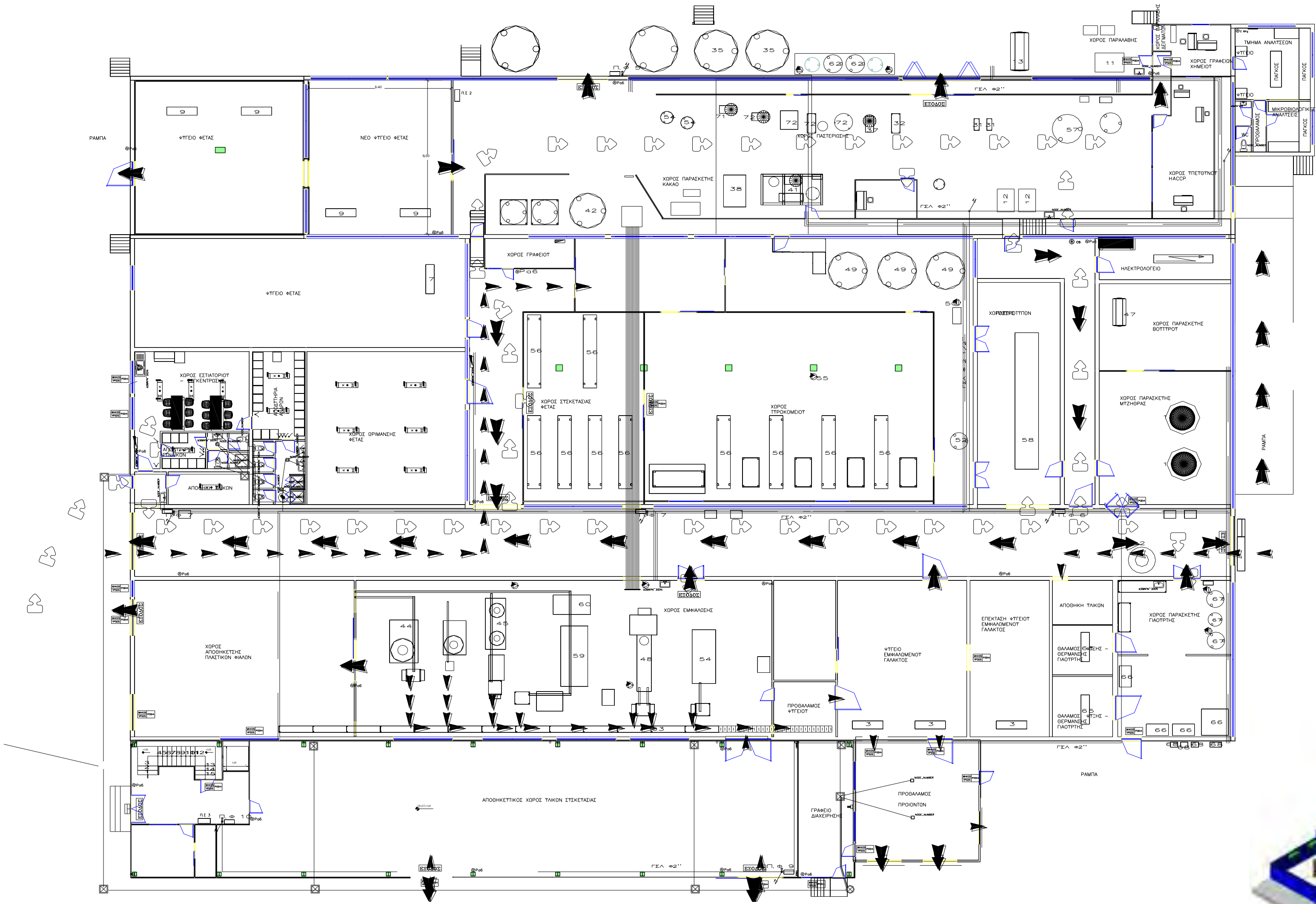


**Figure 46: Dairy products Factory Layout (Main production building)**

# APPENDIX B:

## E E S – Software Presentation

**E E S**
**Engineering Equation Solver**
for Microsoft Windows
Operating Systems
Commercial and Professional Versions

### Overview

EES is an acronym for Engineering Equation Solver.

The basic function provided by EES is the solution of a set of algebraic equations. EES can also solve differential equations, equations with complex variables, do optimization, provide linear and non-linear regression, generate publication-quality plots, simplify uncertainty analyses and provide animations.

EES has been developed to run under 32-bit Microsoft Windows operating systems, i.e., Windows 95/98/2000/XP. It can be run in Linux and on the Macintosh using emulation programs.

There are two major differences between EES and existing numerical equation-solving programs.

- First, EES automatically identifies and groups equations that must be solved simultaneously. This feature simplifies the process for the user and ensures that the solver will always operate at optimum efficiency.
- Second, EES provides many built-in mathematical and thermophysical property functions useful for engineering calculations. For example, the steam tables are implemented such that any thermodynamic property can be obtained from a built-in function call in terms of any two other properties. Similar capability is provided for most organic refrigerants (including some of the new blends), ammonia, methane, carbon dioxide and many other fluids. Air tables are built-in, as are psychometric functions and JANAF table data for many common gases. Transport properties are also provided for most of these substances. The library of mathematical and thermophysical property functions in EES is extensive, but it is not possible to anticipate every user's need.

EES allows the user to enter his or her own functional relationships in three ways.

- First, a facility for entering and interpolating tabular data is provided so that tabular data can be directly used in the solution of the equation set.
- Second, the EES language supports user-written Functions and Procedures similar to those in Pascal and FORTRAN. EES also provides support for user-written routines, which are self-contained EES programs that can be accessed by other EES

programs. The Functions, Procedures, Subprograms and Modules can be saved as library files which are automatically read in when EES is started.

- Third, external functions and procedures, written in a high-level language such as Pascal, C or FORTRAN, can be dynamically-linked into EES using the dynamic link library capability incorporated into the Windows operating system. These three methods of adding functional relationships provide very powerful means of extending the capabilities of EES.

EES allows the user to concentrate more on design by freeing him or her from mundane chores.

EES is particularly useful for design problems in which the effects of one or more parameters need to be determined. The program provides this capability with its Parametric Table, which is similar to a spreadsheet. The user identifies the variables that are independent by entering their values in the table cells. EES will calculate the values of the dependent variables in the table. The relationship of the variables in the table can then be displayed in publication-quality plots.

EES also provides capability to propagate the uncertainty of experimental data to provide uncertainty estimates of calculated variables. With EES, it is no more difficult to do design problems than it is to solve a problem for a fixed set of independent variables.

EES offers the advantages of a simple set of intuitive commands that a novice can quickly learn to use for solving any algebraic problems. However, the capabilities of this program are extensive and useful to an expert as well. The large data bank of thermodynamic and transport properties built into EES is helpful in solving problems in thermodynamics, fluid mechanics, and heat transfer.

EES can be used for many engineering applications; it is ideally suited for instruction in mechanical engineering courses and for the practicing engineer faced with the need for solving practical problems.

## General Information

The information concerning a problem is presented in a series of windows. Equations and comments are entered in the Equations window. After the equations are solved, the values of the variables are presented in the Solution and Arrays windows. The residuals of the equations and the calculation order may be viewed in the Residuals window. Additional windows are provided for the Parametric and Lookup Tables, a diagram and up to 10 plots. There is also a Debug window. A detailed explanation of the capabilities and information for each window type is provided in this section.

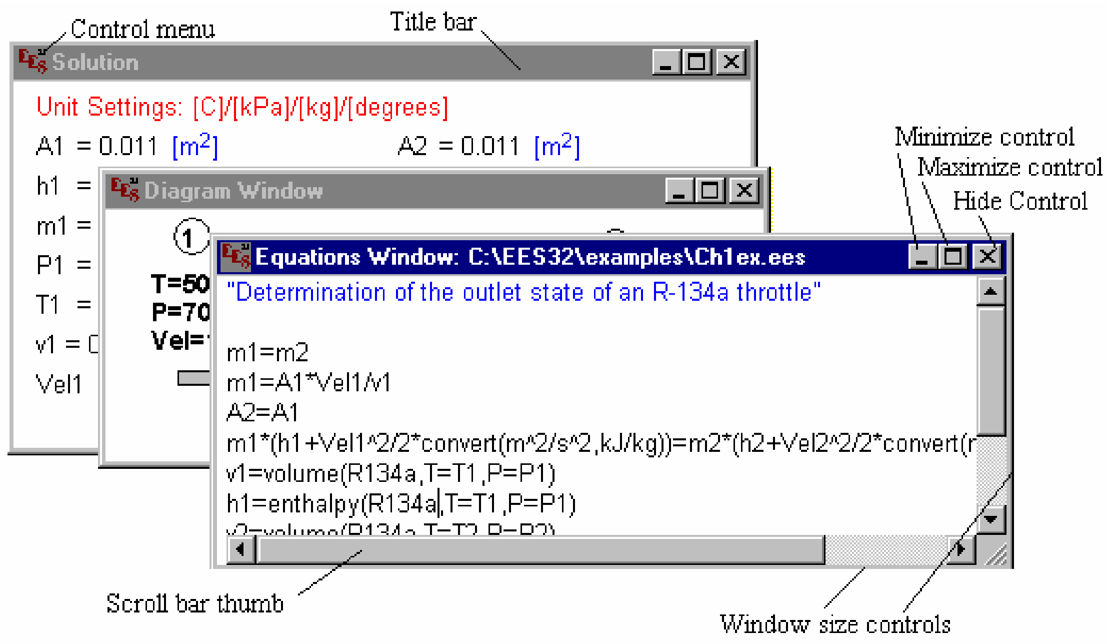



**Figure 47: General overview of EES windows**

All of the windows are:

1. Equations Window
2. Formatted Equations Window
3. Solution Window
4. Arrays Window
5. Residuals Window
6. Parametric Table Window
7. Lookup Table Window
8. Integral Table Window
9. Diagram Window

They can be open (i.e., visible) at once. The window in front is the active window and it is identified by its highlighted (black) title bar.

The figure below shows three overlapped EES windows.

The appearance may be slightly for different versions of the Windows operating system. One difference between EES and most other applications is worth mentioning. The Close control merely hides a window; it does not delete it.

Once closed, a window can be reopened (i.e., made visible) by selecting it from the Windows menu.

# APPENDIX C:

# Degree Method

The estimation of monthly energy Loads for heating of climate spaces, was calculated by the Degree Method, in EES program assuming the internal temperature of each space as the appropriate temperature base. The model for estimation was formed as follows:

For each space the monthly Heating Load is given by the equation:

$$Q_{TH,SPACE}[i,space] = OD[i] \cdot OT[space] \cdot (U \cdot A[space])_b DD[i,space]$$

[1]

for i=1…12 months and space=1…8

where:

$OD[i]$: operation days per month i (=25 for all months)

$OT[space]$: operation hours per day (for each space equal to the values of 4th column of Table 4)

$(U \cdot A[space])_b$: the product of the space area to the mean factor U which has been analytically calculated for each room, considering the characteristics of each wall surface of the specific space (details in Table 29).

$DD[i,space]$: the Degree-Days of month i, in temperature base of the internal temperature of specific space. It can be estimated according the equation:

$$DD[i,space] = \left\{ OD[i] \cdot \Delta T_b + \left(0.744 + 0.00387 \cdot D_a - 0.5*10^{-6} \cdot D_a\right) \cdot OD[i] \cdot e^{-[(\Delta T_b + 11.11)/9.02]^2} \right\}^+$$

where

$\Delta T_b$ = 20°C – $T_{in}$[space] and

$D_a$: the annual Degree-Days of base at 18°C (sum of all months). The Degree-Days of base at 18°C, are given in the Table 30, considering 4th zone (1485 DD).

| No | SPACE | A | operation hours per day | $T_{in}$[space] | U[space] |
|---|---|---|---|---|---|
| | | [m2] | [h] | [°C] | |
| 1 | Laboratory | 132 | 16 | 17 | 7.83 |
| 2 | Cheese Freezer | 302 | 24 | 4 | 7.98 |
| 3 | Main Dairy products factory | 720 | 16 | 17 | 7.04 |

| | | | | | |
|---|---|---|---|---|---|
| 4 | Subsidairy products factory | 100 | 16 | 23 | 7.56 |
| 5 | Cheese Ripening | 98 | 16 | 17 | 9.02 |
| 6 | Milk Bottling | 329 | 16 | 17 | 7.86 |
| 7 | Dairy products factory Freezer | 115 | 24 | 4 | 8.85 |
| 8 | Yoghurt dairy products factory | 70 | 16 | 17 | 8.32 |
| | | | | AVERAGE: | 7.68 |

**Table 27: Climate spaces of the factory (mean thermopenetrability factor U for each space)**

| | ZONE 1 | ZONE 2 | ZONE 3 | ZONE 4 | ZONE 5 | ZONE 6 |
|---|---|---|---|---|---|---|
| **JAN** | *127* | *264* | *281* | *310* | *396* | *405* |
| **FEB** | *147* | *224* | *225* | *263* | *313* | *349* |
| **MAR** | *131* | *196* | *205* | *251* | *268* | *300* |
| **APR** | *78* | *85* | *121* | *128* | *130* | *189* |
| **MAY** | *0* | *10* | *14* | *25* | *23* | *69* |
| **JUN** | *0* | *0* | *0* | *0* | *0* | *0* |
| **JUL** | *0* | *0* | *0* | *0* | *0* | *0* |
| **AUG** | *0* | *0* | *0* | *0* | *0* | *0* |
| **SEP** | *0* | *0* | *0* | *0* | *0* | *0* |
| **OCT** | *10* | *29* | *46* | *65* | *70* | *73* |
| **NOV** | *52* | *96* | *129* | *166* | *187* | *276* |
| **DEC** | *130* | *206* | *246* | *277* | *388* | *404* |
| **SUM** | *720* | *1110* | *1267* | *1485* | *1725* | *2065* |

**Table 28: Degree-Days at base of 18°C, for each climatic zone**

# Reference List


1. *ENERGY OUTLOOK OF GREECE. Hellenic Ministry of Development. Athens, Greece. (Feb2008).*
2. *N.Vassilakos, N.Karapanagiotis, D.Fertis, K.Tigas. Methods of financing Renewable Energy Investments in Greece. CRES (2003).*
3. *M.Karagiorgas, V.Drossou, A.Aidonis. PROTESOL II-Solar thermal plants in industrial processes - DESIGN AND MAINTENANCE GUIDELINES. European Commission - CRES. Altener 4.1030/2/02-084/2002. (2002).*
4. *CLIMASOL – SOLAR AIR CONDITIONING GUIDE. European Commission-CRES. Altener. (2004).*
5. *PROMOTING SOLAR AIR CONDITIONING, Technical overview of active techniques - ALTENER Project Number 4.1030/Z/02-121/2002*
6. *Karagiorgas M. - Technical and Economic Report on the Applications of Solar Thermal Systems in the Dairy products factory Industry - Hellenic Ministry of Development - CRES. Athens, Greece. (Aug. 2000).*
7. *CLIMASOL - GUIDELINES FOR SOLAR COOLING FEASIBILITY STUDIES & ANALYSIS OF THE FEASIBILITY STUDIES - ALTENER Project Number 4.1030/Z/02-121/2002.. (May 2005)*
8. *ENERGY AUDIT GUIDE, PART A: METHODOLOGY AND TECHNICS - Centre for Renewable Energy Sources – Athens (May 2000)*
9. *Guide for the Training of Engineers in the Electricity Production Technologies from Renewable Energy Sources - CRES. Athens, Greece. (Aug. 2001).*
10. *J.V.C. Vargas, J.C. Ordonez, E. Dilay, J.A.R. Parise. Modeling, simulation and optimization of a solar collector driven water heating and absorption cooling plant. Solar Energy 83 (2009) 1232–1244.*
11. *A.Baldini, G. Manfrida, D. Tempesti. Model of a Solar Collector/Storage System for Industrial Thermal Applications. Int. J. of Thermodynamics (IJoT) Vol. 12 (No. 2), pp. 83-88, June 2009.*
12. *T. Tsoutsos, et al., Design of a solar absorption cooling system in a Greek hospital, Energy Buildings. (2009), doi:10.1016/j.enbuild.2009.09.002.*
13. *M. Clausse, K.C.A. Alam, F. Meunier. Residential air conditioning and heating by means of enhanced solar collectors coupled to an adsorption system. Solar Energy 82 (2008) 885–892.*
14. *A.El Fadar, A. Mimet, M. Perez-Garcıa. Modelling and performance study of a continuous adsorption refrigeration system driven by parabolic trough solar collector. Solar Energy 83 (2009) 850–861.*
15. *Giampaolo Manfrida, Vincent Gerard. Maximum Exergy Control of a Solar Thermal Plant Equipped with Direct Steam Collectors. Int. J. of Thermodynamics. Vol. 11 (No. 3), pp. 143-149, (2008).*
16. *H. Zhai, Y.J. Dai, J.Y. Wu, R.Z. Wang. Energy and exergy analyses on a novel hybrid solar heating, cooling and power generation system for remote areas. Applied Energy 86 (2009) 1395–1404.*

17. *Tiago Mateus, Armando C. Oliveira. Energy and economic analysis of an integrated solar absorption cooling and heating system in different building types and climates. Applied Energy 86 (2009) 949–957.*

18. *M. Augustus Leon, S. Kumar. Mathematical modeling and thermal performance analysis of unglazed transpired solar collectors. Solar Energy 81 (2007) 62–75.*

19. *Huseyin Gunerhan, Arif Hepbasli. Exergetic modeling and performance evaluation of solar water heating systems for building applications. Energy and Buildings 39 (2007) 509–516.*

20. *G. Vokas, N. Christandonis, F. Skittides. Hybrid photovoltaic–thermal systems for domestic heating and cooling—A theoretical approach. Solar Energy 80 (2006) 607–615.*

21. *Theocharis Tsoutsos, Joanna Anagnostou, Colin Pritchard, Michalis Karagiorgas, Dimosthenis Agoris. Solar cooling technologies in Greece. An economic viability analysis. Applied Thermal Engineering 23 (2003) 1427–1439.*

22. *Jay Burch, Craig Christensen, Jim Salasovich, Jeff Thornton. Simulation of an unglazed collector system for domestic hot water and space heating and cooling. Solar Energy 77 (2004) 399–406.*

23. *Soteris Kalogirou. The potential of solar industrial process heat applications. Applied Energy 76(2003) 337–361.*

24. *E. Tasdemiroglu, M. Awad. Mathematical methods for the optimization of solar collector area in a solar heating system. Solar & Wind Technology Vol. 7, No. 2/3, pp. 341-345, (1990).*

25. *D. S. Ward, W. S. Duff, J. C. Ward, G. Lof. Integration of evaluated tubular solar collectors with lithium bromide absorption cooling systems. Solar Energy, VoL 22, pp. 335-341. Pergamon Press Ltd., (1979).*

26. *S. R. Swanson, R. F. Boehm. Calculation of long term solar collector heating system performance. Solar Energy, Vol. 19, pp. 129-138. Pergamon Press 1977.*

27. *L.W. Butz et al. Simulation of a solar heating and cooling system. Solar Energy. VoJ 16, pp 129-136. Pergamon Press 1971.*

28. *Klein, S.A., Beckman, W.A., Duffie, J.A., Solar Heating Design by F-chart Method. John Willey and Sons Inc., New York, 1997.*

29. *H.Schnitzer, G.Brunner, K.Giannakopoulou. Example of application of solar thermal plants in diaries. Joanneum Research. Graz, Austria (2004).*

30. *T.Stamatis. ΣΗΜΕΙΩΣΕΙΣ ΜΑΘΗΜΑΤΟΣ ΜΠΣ - Σχεδιασμός Ενεργειακών Συστημάτων. Τμήμα Μηχ.Μηχανικών, Πολυτεχνική Σχολή, Πανεπιστήμιο Θεσσαλίας. (2008).*

31. *Σ. Καπλάνης. Ήπιες Μορφές Ενέργειας ΙΙ – Ηλιακή Μηχανική. Εκδόσεις ΙΩΝ. Αθήνα, Ελλάδα (2004).*

32. *Σελλούντος Βάιος. Θέρμανση και Κλιματισμός: Μελέτη-Κατασκευή-Εγκαταστάσεις-Υλικά-Δίκτυα-Εξοπλισμός. Εκδόσεις ΣΕΛΚΑ. ΑΘΗΝΑ (2002)*

33. *Μ.Κροκίδα, Δ.Μαρίνος-Κουρής, Ζ.Μαρούλης. Σχεδιασμός Θερμικών Διεργασιών. Πανεπιστημιακές Εκδόσεις Ε.Μ.Π. Αθήνα (2003)*

34. *S.A. Klein. E E S Engineering Equation Solver for Microsoft Windows Operating Systems - Commercial and Professional Versions. (2002)*

35. *Spitler, J.D., F.C. McQuiston, K. Lindsey.. The CLTD/SCL/CLF Cooling Load Calculation Method, ASHRAE Transactions. 99(1): 183-192 ( 1993)*

# Links

1. http://www.hnms.gr/hnms/greek/index_html
2. http://en.wikipedia.org/wiki/Photovoltaics
3. http://www.meteonorm.com
4. http://www.fao.org/docrep/004/t0515e/T0515E00.htm#TOC
5. http://www.raee.org/climasol
6. http://www.iea-shc-task25.org/
7. http://www.ocp.tudelft.nl/ev/res/sace.htm
8. http://www.solair-project.eu/
9. http://www.elsevier.com/locate/apthermeng
10. http://www.elsevier.com/locate/enbuild
11. http://www.elsevier.com/locate/apenergy
12. http://www.icatweb.org/journal
13. http://www.highcombi.eu
14. http://www.iea-shc.org/task38
15. http://www.cres.gr/climasol
16. http://www.energycon.org/sace/sace.htm
17. http://www.solcoproject.net
18. http://www.energycon.org
19. http://www. noa.gr
20. http://www.fchart.com