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METHODOLOGY FOR TECHNO–ECONOMIC OPTIMIZATION OF SOLAR ASSISTED HEATING SYSTEMS

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Abstract: This paper has objective, to estimate the thermal performance of solar assisted heating systems in regard of solar fraction and perform life cycle cost analysis to assess the feasibility of their implementation in residential sector. In general it is known that the resource for solar thermal systems i.e. the solar irradiation is free, but the equipment to collect it and convert in to useful form (heat or electricity) has a cost. Solar thermal systems are characterized by high investment and low operational cost. It is presented methodology for obtaining the right size of a solar thermal assisted heating system that gives the lowest combination of solar and auxiliary energy costs. Thermal performance of the solar thermal systems are estimated using numerical methods and software since the solar processes are transient in nature been driven by time dependent forcing functions and loads. The system components are defined with mathematical relationships that describe how components function. They are based on first principles (energy balances, mass balances, rate equations and equilibrium relationships) at one extreme or empirical curve fits to operating data from specific machines. As a result of the analysis specific indicators are derived in order to facilitate the techno–economic analysis and design of solar assisted heating systems.

Keywords: solar assisted heating systems, thermal performance, specific indicators

INTRODUCTION

Reduction of fossil fuel consumption and harmful emissions to the environment could be reduced by implementing the solar energy in heating and cooling of the buildings. It is well known that in the European Union more than 25% of the total energy consumption is due to buildings with heating and cooling representing a major percentage. In the EU–32 countries the final energy consumption in 2003 for heating and cooling the buildings represented about 3600 TWh with 93% for heating and only 7% for cooling [1]. But a tremendous increase in the market for air–conditioning can be observed worldwide especially in developing countries such as Macedonia.

On Figure 1 are presented the sales rates for room air–conditioners (RAC units) in different regions of the world (blue representing worldwide sales and green one European ones). In 2002 were sold 44 million units worldwide and more than 94 million units in 2012 (source by Japan Air–conditioning & Refrigeration News 2013). In order to limit the negative impact on the energy consumption and on the electricity network management, new environmentally sound concepts are of particular importance.

Energy consumption in Europe is expected to face an increase within the next 30 years. This is due the climate and comfort requirements, architecture and technical equipment of larger, commercial buildings require more and more cooling. Space cooling is moving quickly from luxury into necessity and represents a fast growing market. The rise in cooling demands is due to more reasons such as: greater comfort expectations, the perception that cooling contributes to higher productivity and the increase of internal loads of electronic equipment.

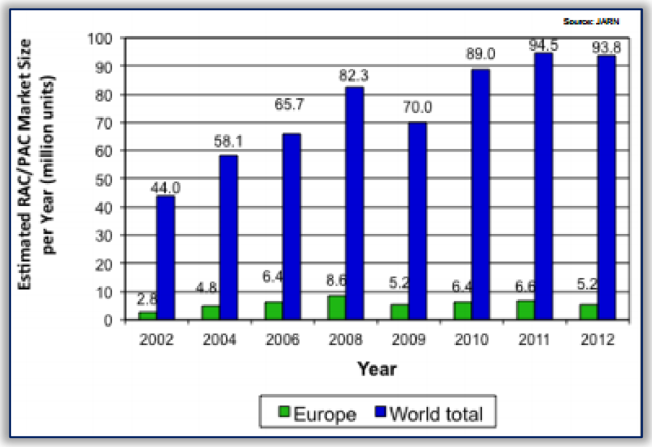


Figure 1. Evolution of air–conditioning market worldwide

By 2020 all new and refurbished buildings should be near zero energy. So the cooling demand will have to decrease. But this means as well that a massive use of renewable energy sources will have to be done.

Building air–conditioning is today based mainly on electrically driven mechanical vapour compression technologies. In [2], R. Ciconkov performed survey on refrigerating and air–conditioning systems regarding the use of CFS fluids in Macedonia Although for new developed predominately large capacity scale developments it is reported about high efficiencies in the compression cycle, for the standard air–conditioning in existing buildings can be assumed that on an average less than 3 kWh ‘cold" are produced with the electricity input of 1 kWh_{el}. Subsequently this implies that 1 kWh primary energy is used for the provision of 1 kWh useful cooling energy. Until now mostly the electrical peak loads were occurring during the winter period, but now are shifting to the summer months and challenging capacity limits and therefore increasing the need of solar cooling technology even in Europe. In [3] is examined TEWT concept for estimating of the global warming from refrigerating and air–conditioning systems.

Solar technologies can supply the energy for all of the building's needs—heating, cooling, hot water, light and electricity—without the harmful effects of greenhouse gas emissions created by fossil fuels thus solar applications can be used almost anywhere in the world and are appropriate for all building types.

Solar thermal systems for hot water production are already mandatory in new buildings according to solar ordinances for example in Spain [4], Portugal, Italy, Greece and other European countries [5].

It is very logical to apply solar energy for cooling purposes since in many applications, such as air–conditioning cooling loads and solar gains are more or less in phase on daily time basis. Thermally driven cooling was applied within last decades in niche–markets preferably in the large capacity scale range, using waste heat or heat from combined heat and power production. A survey made on the basis of IEA Task 38 and Task 48 work has shown the estimated number of installation worldwide nearly of 600 systems in 2010 and nearly 1000 systems in 2012.

In 2013, Solar Air–Conditioning is more than ever representing a huge potential of development for solar energy but this promising technology is facing one man issue, a general lack of economic competitiveness – as it is still the case for many renewable energies unless incentives are in place.

SIMULATION SOFTWARES FOR SOLAR THERMAL SYSTEMS

In order to assess the performance of the solar air–conditioning system under weather conditions for Macedonia, simulation model is developed for solar assisted air–conditioning system applied in residential building.

Simulation in solar cooling and air–conditioning is possible at different levels. A classification may be made by sorting the tools into:

- materials level – analyzing the effect of e.g. different sorption materials on the sorption process;
- component level: detailed analysis of a system component, e.g., chillers, cooling towers, etc.
- process quality level: theoretical analysis of various processes.
- detailed system simulation for optimizing control strategies

Few simulation programs for planning support and sizing of solar assisted air–conditioning systems exist. Also some more programs used internally may exist; additionally, more commercial simulation platforms like Matlab/Simulink, Modelica, etc. can be used, but do not provide of a sufficient number of components for modeling a complete solar air–conditioning system yet.

In this paper is used TRNSYS simulation software and the TESS library for the system component numerical models.

TRNSYS is a commercial time step simulation tool worldwide available. High flexibility in the choice and arrangement of the system components, the desired system can be constructed by selecting and connecting the individual components and by defining the system control. Own written, types'(component models) may be added. Once the time step of the simulation is chosen, it is constant during the simulation run. A major advantage of the program is the availability of a building model, which can be edited in a special building editor and allows the calculation of building loads.

SOLAR SYSTEM COMPONENT DEFINITION

Up to now, existing SHC prototypes were mostly designed on an empirical basis. For small–size systems, simple layouts are generally preferred, in order to improve the reliability and reduce the capital cost of the plant. For example, fixed–volume pumps are selected, and a gas–fired heater is used as the only auxiliary device. For large–size plants, more expensive but also more efficient components can be taken into consideration, such as variable–speed pumps and auxiliary electric chillers. In any case, the main choices to be taken when designing the layout of a SHC system concern:

- the type of solar collectors;
- the thermal–driven chiller (for example, absorption or adsorption machine);
- the auxiliary system for cooling and heating have been developed.

Usually, SHC systems are based on absorption chillers, since the commercial availability of adsorption chillers is very scarce. In addition, adsorption chillers are only available for small cooling capacities, and their cost is significantly higher than for absorption chillers. Thus, most of the SHC prototypes installed all over the world are equipped with an absorption chiller. Single–effect absorption chillers are usually adopted, since double–effect devices must be supplied with an hot stream at temperature higher than 150°C, that would involve the use of concentrating solar collectors [6]. Such configuration – high temperature solar collectors and double–effect chillers – is obviously interesting from an energetic point of view, but is presently too expensive to be considered in pre–commercial applications.

Thus, the most common configuration is based on the coupling of evacuated–tube solar collectors with single–effect absorption chillers. In particular, LiBr–H2O models are commonly preferred, since H2O–NH3 chillers require higher temperatures for the inlet hot stream, and in addition handling ammonia can be somewhat dangerous. For such arrangement, three different system layouts were investigated in this paper, whose characteristics and working principles are briefly summarized in the following:

The model i.e. analyzed system, generally consists of four main subsystems:

- First subsitem is composed of solar collectors with complete hydraulic fittings and control – differential controllers, plate heat exchangers ie this system is represented the source of thermal energy for heating or thermal energy for driving the cooling the absorption machine

- Second is the subsystem for hot and cold storage which includes the storage tanks for hot / cold water that actually represents the connection between the heating system in the building ie absorption cooling machine and the source of heat.
- The heating system introduced with heating / cooling devices, hydraulic components, heat exchangers, cooling absorption machine and eventually existing conventional sources of heat and / or cooling energy.
- The fourth subsystem is the consumer of thermal energy ie the building. This system is represented by the thermal characteristics of the object, i.e its orientation in space.

In the analyses are considers vacuum tube and flat plate collectors product of Camel Solar, type: CS Full Plate 2.0–4 and Vacuum CS 10. The thermal performances of the solar collectors are given in their solar key mark certificate.

At the simulated building internal heat gains are consider by the lighting power density 5 W/m2 and home appliances with specific power of 2 W/m2. The absorption chiller condenser is connected to the wet cooling tower product of Baltimore AirCoil type PF2–0406AA–31–3. Numerical modeling of the cooling tower is provided by the TRNSYS Type510 model from Tess library, a closed circuit cooling tower which cools the liquid stream by evaporating water from the outside of coils containing the working fluid. The working fluid is completely isolated from the air and water in this type of system.

The cooling system in the building is represented with ventilation air distribution system. The heat exchange between the chilled water from the absorption chiller and the ventilation air is provided with heat exchanger water–air modeled Type 508a which is a cooling coil modeled using a bypass approach in which the user specifies a fraction of the air stream that bypasses the coil. The remainder of the air stream is assumed to exit the coil at the average temperature of the fluid in the coil and at saturated conditions. The two air streams are remixed after the coil. Chilled water flow from the absorption chiller to the cooling coil is set to 2900 kg/h and the air flow rate to the building is 4000 kg/h. The auxiliary heater power is modeled 12 kW and the outlet temperature is 80°C, which is the absorption machine driving temperature.

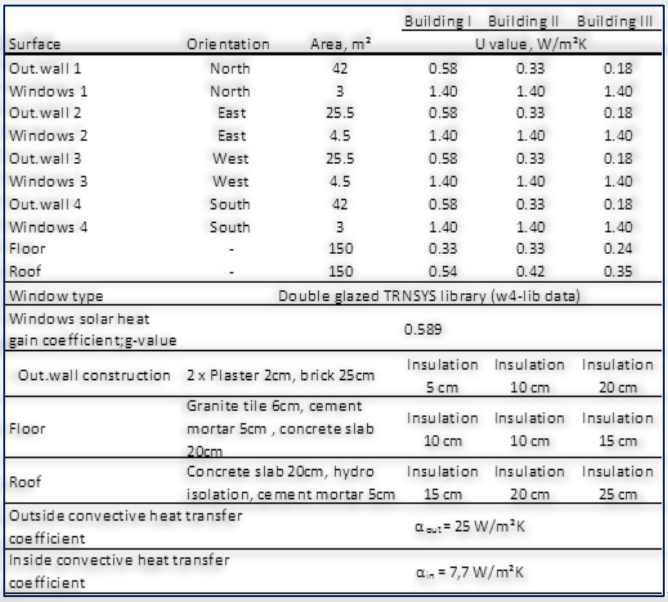
SYSTEM COMPONENT MODELS DEFINITION AND VALIDATION

Validation is performed for the basic solar thermal system components: solar collector, storage tank and differential controller. The experimental system consist of: flat plate solar collector with area of 2 m^2, connected with the internal heat exchanger of the storage tank. Control is provided by differential controller which is set to turn the circulation pump on, when the temperature difference between the collector outlet temperature and the tank temperature is greater than five. There is no consumption of hot water from the storage tank i.e. the only heat transfer is with the surroundings. The circulating pump is set to maintain fluid (water) flow rate set to 7.5 lit/min.

Measurements are made on at an hour interval for the fluid inlet T1 and outlet T2 temperatures from the solar collector, tank fluid temperature T3 and the solar radiation is measured with the pyrometer S as presented on Figure 3. The experimental setup of the analyzed solar thermal system is located in Skopje, R. Macedonia, northern latitude of 42° and 21.43° east longitude. Temperature measurements are performed with temperature data logger thermocouple probes type K.

Solar collector type is evacuated tubular direct flow, product of Camel Solar type Vacuum CS 15 Solar KeyMark certified. During the measurements was placed under tilt of 45°, south orientated i.e. azimuth angle of 0°. The collector thermal performance test results according EN 12975 are presented in Table 1.

Table 1. Reference building physical and thermal performance data



			Building I	Building II	Building III
Surface	Orientation	Area, m^2	U value, W/m^2K		
Out.wall 1	North	42	0.58	0.33	0.18
Windows 1	North	3	1.40	1.40	1.40
Out.wall 2	East	25.5	0.58	0.33	0.18
Windows 2	East	4.5	1.40	1.40	1.40
Out.wall 3	West	25.5	0.58	0.33	0.18
Windows 3	West	4.5	1.40	1.40	1.40
Out.wall 4	South	42	0.58	0.33	0.18
Windows 4	South	3	1.40	1.40	1.40
Floor	-	150	0.33	0.33	0.24
Roof	-	150	0.54	0.42	0.35
Window type	Double glazed TRNSYS library (w4-lib data)				
Windows solar heat gain coefficient;g-value			0.589		
Out.wall construction	2 x Plaster 2cm, brick 25cm		Insulation 5 cm	Insulation 10 cm	Insulation 20 cm
Floor	Granite tile 6cm, cement mortar 5cm , concrete slab 20cm		Insulation 10 cm	Insulation 10 cm	Insulation 15 cm
Roof	Concrete slab 20cm, hydro isolation, cement mortar 5cm		Insulation 15 cm	Insulation 20 cm	Insulation 25 cm
Outside convective heat transfer coefficient			$\alpha_{out} = 25\ W/m^2K$		
Inside convective heat transfer coefficient			$\alpha_{in} = 7{,}7\ W/m^2K$		

In the TRNSYS model solar collector is model with the Type 538 from the Tess library with commercial solar collector performances used from Solar Key Mark certificate for producer CamelSolar. The storage tank is modeled with the Type 60d including the internal heat exchanger for which are supplied technical data from the producer "Sun System". Type 2b–2 is used to simulate the differential controller set with upper dead band of 5 and lower dead band 2, the high limit cut–off temperature set to 100°C. Between the solar collector and storage tank is connected pipe Type 31 modeled with internal diameter 0.0025 m, length of 10 m and loss coefficient of 0.3 W/m^2K to account for the heat losses.

The pipe network modeled with Type 31 is used to increase the thermal capacity of the system and thus increase the simulation stability. Circulating pump is represented with the component Type 3d with mass flow rate 450 kg/h i.e. 7.5 l/min same as in the experimental setup 2.

Measurements are performed starting from date 18.09.2013 until 28.03.2014 and in parallel are measured two systems with same capacity storage tank of 150l but different type of collector's i.e. flat plate and vacuum tube solar collectors. Validation process use data for the vacuum tube collector and the results are presented only for one day period (18.09.2013) with collection time interval ranging between 20min and 45min interval, from 10:40 up to 16:05 h.

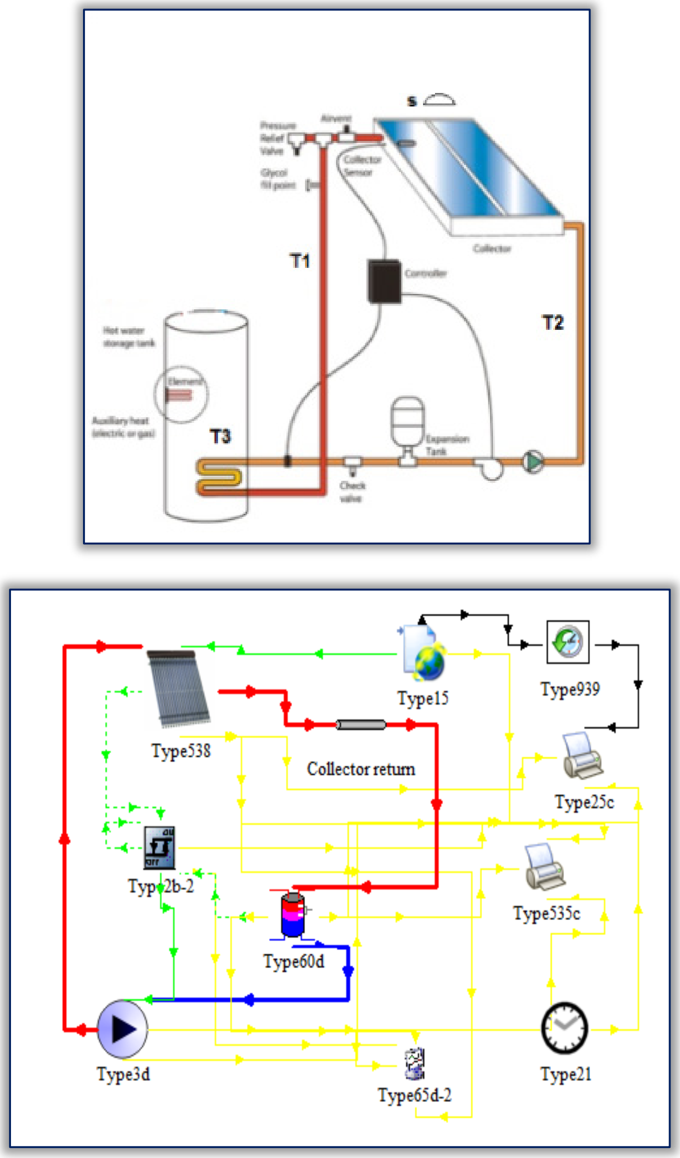


Figure 2. Solar system experimental and simulation scheme

According above presented data i.e. diagrams can be concluded that there is acceptable match between the measured and simulated results Figure 3, and Figure 4. The discrepancies between the measured and simulated results are expected since the solar radiation has different values i.e. simulated values are taken from the Meteonorm database for the selected location while the measure are obtained directly for the specific location as given on Figure 6.

Another influencing factor is the uncertainty of the measurements error and last but not the least it should not be neglected the transition nature of the solar thermal systems.

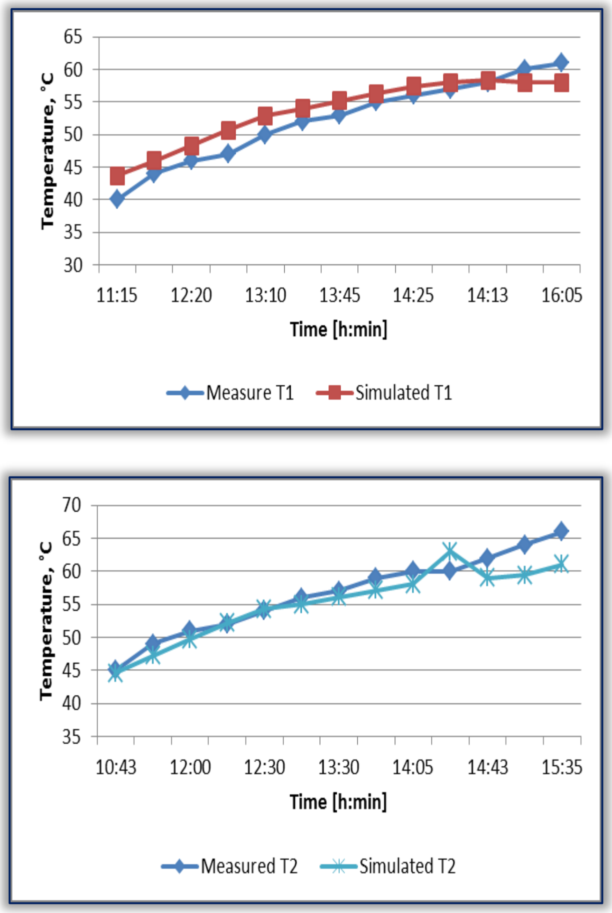


Figure 3. Measured and simulated results for the collector inlet and outlet temperature

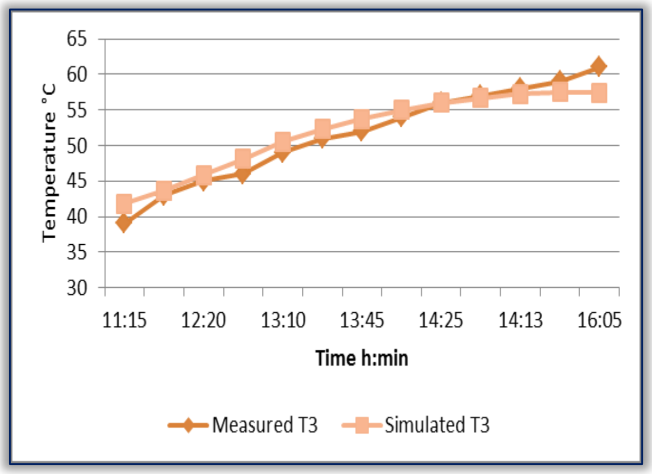


Figure 4. Measured and simulated temperatures inside storage tank

The resulting simulations reveal the individual thermal behaviour of the solar collector, storage tank, differential controller and circulating pump as well as their assembled thermal behaviour. These results coincidence with the respective experimental data, thus this fact validates these models for future application in the heating/cooling system. Validation for the absorption chiller is performed for the TRNSYS component Type 177.

This component type offers four numerical modes of absorption chiller. In this simulation is used mode "a" i.e. Type

177a which is standard mode using user supplied characteristic parameters. Since in this paper are considered only solar air–conditioning for residential buildings. In the simulation is modelled the absorption chiller H_2O/LiBr produced by Sonnenklima type Suninverse 10. Simulation of the absorbtion chiller is done with the component Type 177a, whereas input parameters are used the values for Suninverse provided in Table 3. As output for the absorption chiller cooling power is obtained 10,1 kW, which corresponds with the factory value. Validation exists for the Type 177 mode "d" performed by Albers and Ziegler [7], using the measurement results from Kühn [8] . According to this, final conclusion is that this numerical model of absorption chiller provides reliable results, thus it is suitable to be used as model for further simulations.

ASSESMENT OF THE SOLAR AIR–CONDITIONING SYSTEM

Because of the interactions of components, optimal system performance occurs under conditions different from those for optimal behaviour of each component. For example, optimal collection efficiency would not necessarily be coupled with least auxiliary energy.

Many different hydraulic schemes are designed which makes difficult to compare the installations performances [9]. Methods used to determine solar heating and/or cooling energy requirements for both active and passive/hybrid systems are described by Feldman and Merriam, Hunn, Nowag and other. For thermally driven systems the scheme on Figure 5 is used to identify main components and energy flows of the system. On Figure 5 is presented small scale system for family houses, small multi dwellings, using a small size packaged ab/adsorption solar system. This configuration is an adaptation of the solar combi-system including the cooling function is also called SSC + Solar Combi.

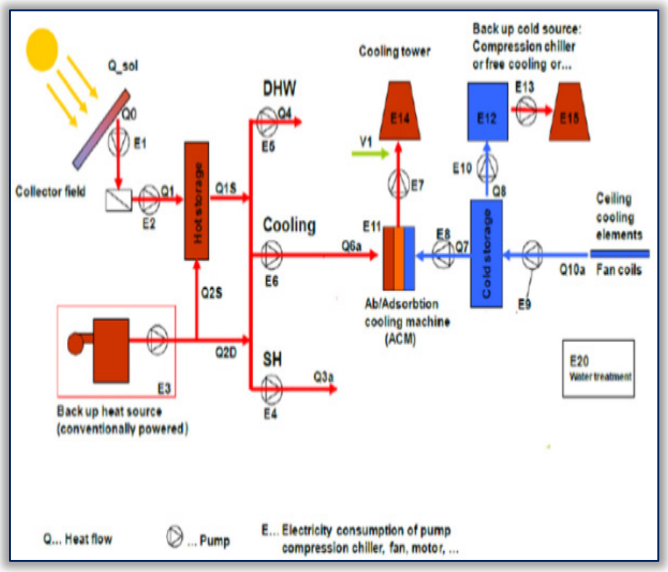


Figure 5. Components and energy flows
for solar air–conditioning system

There are four generally accepted measures of solar system performance:

- Collector efficiency applies to the performance of the solar energy collection subsystem. It is the energy collected, divided by the radiation incident upon the collectors. Influence of the collector position on the collector thermal efficiency is done in [10].
- System efficiency, or solar heating performance factor is the solar heat delivered to the load divided by the total radiation incident upon the collector.
- Solar fraction is the fraction of the total heat requirement that is met by solar energy.
- Electrical coefficient of performance is the solar heat delivered to the load, divided by the electrical energy used to operate the system.

Each solar system operates at characteristic efficiency level resulting from the interaction of the subsystems, environmental conditions and system configurations. The net savings per square meter of solar collector indicate the relative performance of each of these systems.

The five categories of system–level design parameters that limit solar system performance.

- Solar resource assessment. This category represents the solar reference weather data values used by the solar design community
- Collection subsystem. This category represents the solar collection sub–system, including devices used to capture incoming solar radiation
- Storage subsystem. This category deals with all aspects of the system effects caused by storage components.
- Controls. This category refers to equipment and methods for controlling solar components within the solar system

This category deals with the types and magnitude of the heat requirements in the buildings

MODEL OF REFERENCE BUILDING

Building as energy consumer has a major impact on the overall efficiency of the solar system i.e. can be simply said that the building itself is one of the leading parameter in sizing the system. Since the analyses are made for climatic conditions in Macedonia also the thermal performance of buildings must be in accordance with the Regulations on energy efficiency in Macedonia. Furthermore the analysis is taken into account the impact of the specific consumption of heating / cooling energy of the building kWh/m^2 a to the response and the performance of the solar collector system.

Main governing indicators according to which is based the system comparison are: thermal efficiency of solar collectors, solar fraction and power consumption for the auxiliary devices.

In Table 1 are listed three "types" of the building i.e. physically is the same building only the insulation thickness on the external walls, roof and floor are varied in order to obtain different values for specific annual heat. The main idea for this analysis is to assess the influence of the thermal performance of buildings on the economic viability of the use of solar thermal systems in air–conditioning.

Constant value of 0.3 1/h is defined for the infiltration of outdoor air, while for the summer when cooling is required in the building is envisaged/modeled mechanical ventilation defined with air mass flow and temperature entered through \the models of fan and heat exchanger air–water which is directly connected with the cooling absorption machine.

Regarding the thermal comfort, in the heating mode the inside temperature is defined to be 20°C from 05:00 – 22:00 and for the rest is defined setback temperature of 16°C, for the cooling mode is defined constant inside temperature of 26°C. Building I has 90 kWh/m^{2}a, Building II with 70 kWh/m^{2}a and Building III has 57 kWh/ m^{2}a. Comparing the energy consumption Building III has 42% lower than Building I and 19% than Building II.

The performance of future conventional space–conditioning systems affects the economic potential of active solar systems. The performance and cost of today's conventional heating, cooling and domestic hot water system can be readily determined, but conventional heating and cooling technology is constantly improving

In the analysis for the heating considered two reference: Buildings Type II and III (as given in Table 1), with specific heat energy consumption of 70 kWh/m^{2}a and 57 kWh/m^{2}a respectively. In the analysis for the cooling is considered only Building type III which has specific cooling energy of 12 kWh/m^{2}a. The time step used in the simulations is 7.5 min and the heating and loads are integrated on hourly basis. On Figure 6, are presented the results from the simulation of solar assisted heating with flat plate collectors varying their total area 16 m^2, 32 m^2, 64 m^2 mass flow rates are 50 kg/h m^2 and heat storage tank of 1000 l and 2000l only for the 64 m^2 collector area.. Collectors are tilted 40° toward south – azimuth 0° also is installed 200l DHW storage tank heating with the same collector array only in period when the heating storage tank is charged or the condition for the circulation pump is not satisfied.

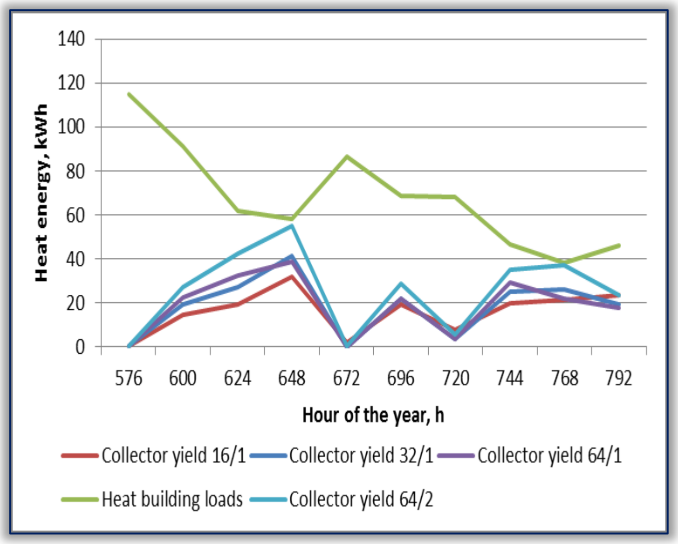


Figure 6. Hourly heating loads and collector energy yield for different areas ten day period

Common for the analysed systems is that in each of them the heat is distributed through the underfloor heating with flow rate of 2000 kg/h, solar collector array mass flow rate depending from the collector area i.e. 50 kg/h m^2, auxiliary heat energy is provided by heater located at the fluid tank outlet with capacity of 12 kW connected with the generator of the absorption cooling machine and 9 kW for the DHW installed also at the tank hot water outlet set to maintain constant temperature of 45°C.

From the obtained results it is concluded that there are no large differences between the solar collector yield of 32 m^2 and 64 m^2. This is result of the small storage tank capacity which cannot store the available heat from the 64 m^2 collector array solar/heat yield resulting in storage temperature increase thus decrease in solar fraction and collector efficiency.

On Figure 6 are also presented collector yields only for ten days.

OPTIMIZATION METHODOLOGY

In solar system energy design the collector area is considered as the primary parameter for a given load and system configuration. The collector area is also the optimization parameter i.e. the aim is to find the collector area that gives the highest life cycle savings.

A method for the economic optimization is considered in which life cycle savings are plotted against collector area A_c, to find the area that maximises the savings. The optimization procedure is simplified if life cycle savings (LCS) are expressed mathematically in terms of the collector area. Therefore the optimum is obtained when:

$$\frac{\partial LCS}{\partial A_C}=0 \qquad (1)$$

The maximum savings are obtained when the relationships between the collector area and solar fraction satisfies the following relation:

$$\frac{\partial F}{\partial A_C}=\frac{P_2 C_A}{P_1 C_{F1} L} \qquad (2)$$

where : P_1 – ratio of life cycle fuel cost savings to firs year fuel savings, P_2 – ratio of life cycle expenditure incurred from additional investment to the initial investment, L – load (GJ), C_{F1} – first year unit energy cost delivered from fuel, C_A – area independent costs

The economic evaluation of a solar application includes factors such as the capacity cost of delivering solar energy, the optimum sizing of collectors and other equipment the costs of competing technologies and financial analyses. There figures of merit that are used to accept or reject particular solar application including simple payback, cash flow, capital cost per unit of energy saved, life cycle cost, net present value and levelized energy cost.

For a range of economic assumptions, an analysis considering a 5 or 7 year simple payback is assumed to be an adequate figure of merit to establish cost goals for active solar cooling and heating technologies. For residential applications, a payback period of 5–7 years may consider as acceptable.

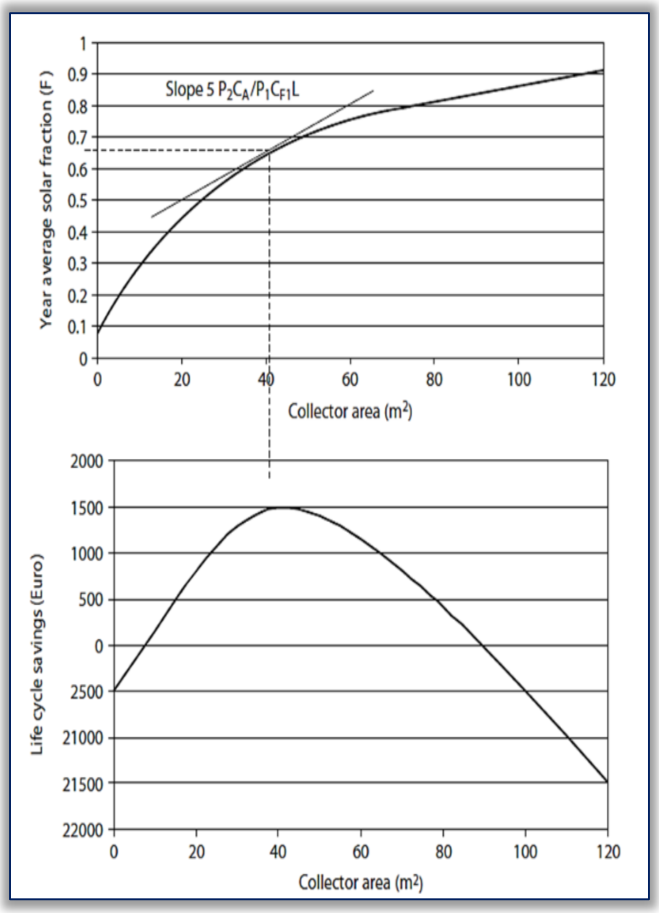


Figure 7. Optimum collector area determination from the slope of the F versus A_C curve

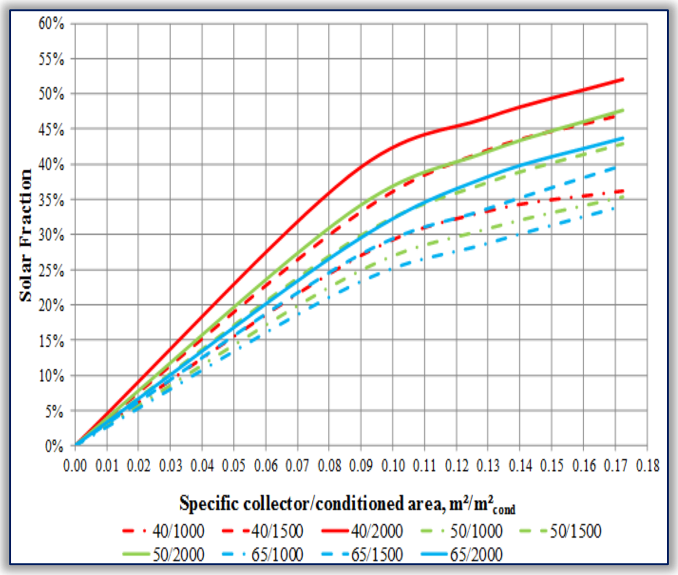


Figure 8. Function dependence between solar fraction in regard of ration solar collector area and building specific heat energy consumption

Comparison is performed between different combinations of solar thermal systems and auxiliary heating devices in regard of conventional heating system with electrical boiler. The analyzed solar systems have total solar collector area of 16 m^2, 32 m^2 and 64 m^2 combined with storage tanks of 1000 l, 1500 l and 2000 l, and auxiliary heating energy provided by electric heater or heat pump air–water with E.V.I compressors with nominal capacity of 15 kW product of Hidros model Lzti 10.

On the next Figure 8 are presented simulation results for the solar fraction savings regarding solar collector area, accumulator storage volume and type of building i.e. specific heat consumption.

CONCLUSION

In this paper were assessed the thermal performance of solar assisted air–conditioning system for residential buildings for weather conditions in Macedonia. Within the analysis are covered several solar thermal systems varying the collector area, hot water storage tank and the auxiliary heat source. It is performed verification on the model for solar assisted air conditioning system which provides reliable results i.e. can be used for further analyses.

Methodology is derived according to which can be determined optimal solar collector area and buffer tank in regard of building energy performance generating maximum life cycle cost savings. From the simulation results is generated diagram according to which can be derived the specific savings–solar fraction in regard of the solar collector area, buffer tank and building energy performance.

Note

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