G Model
ENB 4506 1–9

Energy and Buildings xxx (2013) xxx–xxx

Contents lists available at ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

Design of a renewable energy based air-conditioning system

Boonrit Prasartkaew[a,*], S. Kumar[b]

[a] Department of Mechanical Engineering, Faculty of Engineering, Rajamangala University of Technology Thunyaburi, Klong 6, Thunyaburi, Patumthani 12110, Thailand
[b] Energy Field of Study, Asian Institute of Technology, P.O. Box 4, Klong Luang, Pathumthani 12120, Thailand

ARTICLE INFO

Article history:
Received 10 August 2012
Received in revised form 11 July 2013
Accepted 1 September 2013

Keywords:
Solar
Absorption
Biomass
Cooling
Gasifier
Simulation
Hybrid

ABSTRACT

This paper presents the design of a solar-biomass hybrid system for air-conditioning. Three possible overall system configurations were first considered, based on which the most suitable configuration considered for the design. The principles of component selection of a flat plate solar collector with storage, biomass gasifier with boiler, and a LiBr–H_2O absorption system has been described, and the design criteria for the proposed system elaborated. The design was to satisfy a cooling load of 4.5 kW, and solar to auxiliary heat ratio of at least 0.7. Simulation at various solar collector size and storage tank volume for the weather conditions at Bangkok indicate the most suitable design and specifications for the size/capacity of the solar collector, storage tank, set point temperature of the absorption generator and biomass-gasifier boiler.

1. Introduction

To address the global warming and energy crisis issues, using renewable energies appear as an interesting alternative. As the ambient temperature increases, the need for air conditioning will dramatically increase. To reduce the electricity consumption and CO_2 emissions, solar air conditioning systems seems appropriate and vital [1].

The two main solar air conditioning options are electrical and thermal driven systems. The electrical systems require the use of photovoltaic panels; these are still expensive and have low efficiencies. The thermal driven systems are divided into heat transformation systems and thermo-mechanical processes. Most thermo-mechanical processes are in laboratory scale and they use engines to drive a compressor instead of an electrical motor. The heat transformation processes are divided into close and open cycles, and they may use liquid or solid sorbents. Almost all solar cooling systems are closed cycle liquid sorbent systems.

Solar driven system offers a good model of a clean process for sustainable technology [2]. The two driving energy sources for the conventional solar cooling systems are without auxiliary heat source and with fossil fuel auxiliary heater. As solar energy is intermittent, the solar cooling system without auxiliary heat source cannot be continuously used, especially at night, and so its reliability is low. Almost all solar cooling systems use gas (LPG/CNG) as an auxiliary heater [3]. Thus, a conventional solar cooling system has three major sub-systems – absorption chiller, solar water heating system and backup/auxiliary heating sub-systems (renewable or fossil fuel based).

A solar-biomass hybrid cooling system has been proposed, which is a completely renewable energy based system [3,4]. This paper first identifies the optimal system configuration (among three options), and then presents the design procedure of a solar-biomass hybrid cooling system for the chosen option. Section 2 describes the possible system configurations of solar-biomass hybrid absorption cooling system (SBAC). Section 3 presents the mathematical model and simulation inputs used for the design of the SBAC system. Section 4 presents the selection of the best system configuration. Section 5 presents the design of a solar-biomass hybrid air-conditioning system. Finally, the conclusions are given in Section 6.

2. Possible system configurations

As demonstrated in Fig. 1, most conventional solar cooling systems use fossil fuel as an auxiliary heat source, and so their operating cost and emissions are high. This research proposes biomass energy as a cheaper (or free if it is the waste material) auxiliary heat source to address the above issue. Thus, the main difference of the proposed system as compared to the conventional system is

* Corresponding author. Tel.: +66 897 672 533; fax: +66 2 549 3432.
E-mail addresses: prasartkaew@yahoo.com, boonrit_p@rmutt.ac.th (B. Prasartkaew).

0378-7788/$ – see front matter © 2013 Published by Elsevier B.V.
http://dx.doi.org/10.1016/j.enbuild.2013.09.001

Please cite this article in press as: B. Prasartkaew, S. Kumar, Design of a renewable energy based air-conditioning system, Energy Buildings (2013), http://dx.doi.org/10.1016/j.enbuild.2013.09.001

Design of a Renewable Energy Based Air-conditioning System

Boonrit Prasartkaew[1,*] and S. Kumar[2]

[1] Department of Mechanical Engineering, Faculty of Engineering, Rajamangala University of Technology Thunyaburi, Klong 6, Thunyaburi, Patumthani 12110, Thailand
[2] Energy Field of Study, Asian Institute of Technology, P.O. Box 4, Klong Luang, Pathumthani 12120, Thailand

*Corresponding Author: prasartkaew@yahoo.com, +66-897-672-533, +66-2-549-3432

Abstract

This paper presents the design of a solar-biomass hybrid system for air-conditioning. Three possible overall system configurations were first considered, based on which the most suitable configuration considered for the design. The principles of component selection of a flat plate solar collector with storage, biomass gasifier with boiler, and a $LiBr-H_20$ absorption system has been described, and the design criteria for the proposed system elaborated. The design was to satisfy a cooling load of 4.5 kW, and solar to auxiliary heat ratio of at least 0.7. Simulation at various solar collector size and storage tank volume for the weather conditions at Bangkok indicate the most suitable design and specifications for the size/capacity of the solar collector, storage tank, set point temperature of the absorption generator and biomass-gasifier boiler.

Keywords: Solar; Absorption; Biomass; Cooling; Gasifier; Simulation; Hybrid

Nomenclature

A	area (m^2)
C	specific heat capacity, (kJ/kg.K)
F_R	heat removal factor
G_T	solar insolation on tilted surface (kW/m^2)
h	specific enthalpy (kJ/kg)
LHV	low heating value (kJ/kg)
M	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}$	energy rate (kW)
t	time (sec)
T	temperature (K)
U	heat transfer coefficient (W/m^2 K)
U_L	overall heat transfer coefficient (W/m^2 K)
$\dot{V}$	volumetric flow rate (m^3/s)
v	specific volume (m^3/kg)
x	solution mass concentration (%)

Greek Symbols

η	efficiency
τ	transmittance
α	absorptance

β	tank load control function
γ	temperature differential control function
φ	boiler load control function
ε	heat exchanger effectiveness
ρ	density (kg/m^3)

Subscripts

0	initial condition
a	ambient
ab	absorber
aux	auxiliary
b	boiler
BM	biomass
bl	boiler to load
c	collector
co	condenser
ev	evaporator
f	liquid state
fg	liquid-vapor mixture
flu	flue gas
g	vapor state
ge	generator
gw	gas to water
hi	high
i	inlet
lo	low
o	outlet
PG	producer gas
ref	refrigerant
set	set point value
sol	solution of refrigerant and absorbent
st	storage tank
tl	tank to load
u	solar useful energy
w	water
we	water to environment

1. Introduction

To address the global warming and energy crisis issues, using renewable energies appear as an interesting alternative. As the ambient temperature increases, the need for air conditioning will dramatically increase. To reduce the electricity consumption and CO_2 emissions, solar air conditioning systems seems appropriate and vital [1].

The two main solar air conditioning options are electrical and thermal driven systems. The electrical systems require the use of photovoltaic panels; these are still expensive and have low efficiencies. The thermal driven systems are divided into heat transformation systems and thermo-mechanical processes. Most thermo-mechanical processes are in laboratory scale and they use engines to drive a compressor instead of an electrical motor. The heat transformation processes are divided into close and open cycles, and they may use liquid or solid sorbents. Almost all solar cooling systems are closed cycle liquid sorbent systems.

Solar driven system offers a good model of a clean process for sustainable technology [2]. The two driving energy sources for the conventional solar cooling systems are without auxiliary heat source and with fossil fuel auxiliary heater. As solar energy is intermittent, the solar cooling system without auxiliary heat source cannot be continuously used, especially at night, and so its reliability is low. Almost all solar cooling systems use gas (LPG/CNG) as an auxiliary heater [3]. Thus, a conventional solar cooling system has three major sub-systems -absorption chiller, solar water heating system and backup/auxiliary heating sub-systems (renewable or fossil fuel based).

A solar-biomass hybrid cooling system has been proposed, which is a completely renewable energy based system [3,4]. This paper first identifies the optimal system configuration (among three options), and then presents the design procedure of a solar-biomass hybrid cooling system for the chosen option. Section 2 describes the possible system configurations of solar-biomass hybrid absorption cooling system (SBAC). Section 3 presents the mathematical model and simulation inputs used for the design of the SBAC system. Section 4 presents the selection of the best system configuration. Section 5 presents the design of a solar-biomass hybrid air-conditioning system. Finally, the conclusions are given in Section 6.

2. Possible System Configurations

As demonstrated in Figure 1, most conventional solar cooling systems use fossil fuel as an auxiliary heat source, and so their operating cost and emissions are high. This research proposes biomass energy as a cheaper (or free if it is the waste material) auxiliary heat source to address the above issue. Thus, the main difference of the proposed system as compared to the conventional system is that the fossil auxiliary heat source is replaced by biomass based gasifier boiler.

Figure 2 show three possible system configurations for solar-biomass hybrid cooling system: (Case 1) an auxiliary heater is connected in series with the solar water heating system, (Case 2) an auxiliary heater is connected in parallel with solar collector, and (Case 3) an auxiliary heater is connected in parallel with the solar water heating system.

The common feature in all the three cases is that the insulated biomass gasifier boiler has two functions: it works as an auxiliary energy source when solar energy is not enough and works as the main heat source when the solar radiation is not available. In case 1 and 3, the gasifier boiler is controlled by a controller and supplies hot water to the generator of the absorption chiller. The solar energy heats the water at the collector field which is then pumped to the hot water storage tank. This pump will be activated by controller. Usually, it remains off until the difference between collector outlet and inlet water temperature is above the upper dead band value. The controller will switch the pump off when this difference reaches the lower dead band.

The cooling is provided by a single-effect LiBr-H_2O absorption chiller. Heat from solar collector or from biomass-boiler evaporates the water (strong solution of water and LiBr) in the generator of the absorption chiller. This is led to the condenser, where it rejects heat to the ambient and condenses. This is taken to the evaporator through the expansion valve, where it receives heat from the space to be cooled and evaporates. The evaporated refrigerant (water) is absorbed by the weak solution in the absorber (from the generator). The absorption process also releases heat to the ambient, and the solution, now rich in water (strong solution) is taken to the generator (by a pump) to complete the cycle. The heat required for its generator is drawn from hot water pumped from a hot water storage tank fed by the solar collectors and/or sometimes boosted/fed by biomass boiler. The condenser and absorber of absorption chiller are cooled by water pumped through a cooling tower. The chilled water produced from evaporator is pumped for cooling proposes.

3. Mathematical Model and Input Parameters Used for the Design

The mathematical model of the SBAC system and its validation presented in [3] was used in the system configuration selection and system design. The procedure used the governing equations for each sub system, and that the inputs and outputs between each sub-system were according to the configuration in Figure 2. The model equations were constructed using the following assumptions:

1) The model considered the energy and mass balances at each component, and of the overall system.
2) The system is considered to be at steady state
3) The specific heat and density of the working fluids are constant.
4) The loss of the water vapor and moisture (at the hot water storage tank and solar collector vents) is not taken into account.
5) There is no pressure loss and no heat loss/gain in the lines (pipes) connecting the system components.
6) The fluid temperatures increasing due to the friction in plumbing and valves, blowers and pumps are negligible.
7) The energy considered are solar and biomass energy, while the power consumed by other equipment (e.g. pumps, blower, fans and controllers) is excluded.

3.1 Solar water heating system

The expression (Equation (1)) for collector efficiency given by the Hottel-Whillier Bliss equation was used:

$$\dot{Q}_u = \dot{m}c_p\left(T_{c,o} - T_{c,i}\right) = \eta_c A_c G_T \tag{1}$$

where,

$$\eta_c = F_R(\tau\alpha) - F_R U_L\left(T_{c,i} - T_a\right)/G_T \tag{2}$$

$$T_{c,i} = \beta T_{st} + (1-\beta)T_a \tag{3}$$

$$T_{c,o} = \beta\left(T_{c,i} + \dot{Q}_u/\left(\dot{m}_c C_p\right)\right) \tag{4}$$

The temperature distribution in the hot water storage tank is obtained from the energy balance expressed as:

$$\left(MC_p\right)_{st}(dT_{st}/dt) = \gamma \dot{Q}_u - \dot{Q}_{tl} - (UA)_{st}(T_{st} - T_a) \tag{5}$$

where, the extracted heat $\dot{Q}_{tl}$ and control function used for the collector and load energy terms, are defined as:

$$\dot{Q}_{tl} = \beta \dot{m}_{gs} C_p\left(T_{st} - T_{gs,o}\right) \tag{6}$$

$$\beta = \begin{cases} 1 & if T_{st} > T_{set}, \\ 0 & otherwise. \end{cases} \tag{7}$$

$$\gamma = \begin{cases} 1 & if T_{c,o} > T_{st}, \\ 0 & otherwise. \end{cases} \tag{8}$$

Whenever the temperature of hot water supplied to the chiller machine is lower than set point temperature, the auxiliary heat ($\dot{Q}_{aux}$) is needed and this required heat will be supplied by the biomass gasifier-boiler. This required heat can be determined as follows:

$$\dot{Q}_{aux} = (1 - \beta)\dot{m}_{gs} C_p\left(T_{set} - T_{gs,o}\right) \tag{9}$$

3.2 Biomass gasifier-boiler

The gas-fired boiler is modeled as a heat exchanger, where heat is transferred between combustion products and water. The transient temperature of water inside the boiler can be determined by:

$$\left(MC_p\right)_b(dT_b/dt) = \dot{Q}_{gw} - \dot{Q}_{bl} - (UA)_{ws} C_{p,a}(T_b - T_a) \tag{10}$$

where, the added heat into the water heater boiler $\dot{Q}_{gw}$ and its effectiveness relations for the heat exchangers between combustion gases to water can be calculated from:

$$\dot{Q}_{gw} = \varphi \varepsilon_{gw}(\dot{Q}_{PG} - \dot{m}_{flu} C_{p,flu} T_{flu}) \tag{11}$$

where, $$\varphi = \begin{cases} 1 & if \dot{Q}_{aux} > 0, \\ 0 & otherwise. \end{cases} \tag{12}$$

$$\varepsilon_{gw} = [1 - exp(-NTU(1 + R))]/(1 + R) \tag{13}$$

where,

$$R = (\dot{m}C)_{min}/(\dot{m}C)_{max} \tag{14}$$

$$NTU = (UA)_{gw}/(\dot{m}C)_{min} \tag{15}$$

Heat losses from flue gas and from boiler surface to ambient surrounding have been considered in the overall energy supplied by the gasifier, and is given by

$$\dot{Q}_{PG} = \varphi[(\dot{Q}_{aux}/\varepsilon_{gw}) + \dot{m}_{flu}C_{p,flu}T_{flu} + (UA)_{ws}C_{p,a}(T_b - T_a)] \tag{16}$$

where, the extracted heat $\dot{Q}_{bl}$ from the boiler to meet the load can be calculated from:

$$\dot{Q}_{bl} = \varphi\dot{m}_{ge}C_p(T_b - T_{ge,o}) \tag{17}$$

The consumption rate of biomass feed stock is obtained from:

$$\dot{m}_{BM} = (LHV_{PG}\dot{V}_{PG})/(\eta_G LHV_{BM}) = \dot{Q}_{PG}/(\eta_G LHV_{BM}) \tag{18}$$

3.3 Absorption chiller

The thermodynamic model of absorption chiller was used to simulate its performance with the assumptions presented in [3]. The effectiveness of generator can be calculated as:

$$\varepsilon = 1 - exp[-(UA)/(\dot{m}_w C_p)] \tag{19}$$

At the generator, knowing the hot water inlet temperature and generator heat flow, the generator temperature can be determined by the following equations:

$$T_{ge} = T_{ge,i} - \dot{Q}_{ge}/(\varepsilon_{ge}\dot{m}_{w,ge}C_p) \tag{20}$$

$$T_{ge,o} = T_{ge,i} - \dot{Q}_{ge}/(\dot{m}_{w,ge}C_p) \tag{21}$$

where,

$$T_{ge,i} = \beta T_{st} + \varphi T_b \tag{22}$$

$$\dot{Q}_{gen} = \dot{Q}_{tl} + \dot{Q}_{bl} \tag{23}$$

For a defined solution mass fraction (range of 45<X<70%LiBr) and calculated generator temperature, the saturation pressure, P_4 and h_4 can be calculated from.

$$Log\,P = C + D/(T_{ref} + 273) + E/(T_{ref} + 273)^2 \tag{24}$$

$$T_{ref} = \left(-\frac{2E}{D} + [D^2 - 4E(C - log\,P)]^{0.5}\right) - 273 \tag{25}$$

$$T_{sol} = \Sigma B + T_{ref}\Sigma A \tag{26}$$

$$h = \Sigma A + T_{sol}\Sigma B + \Sigma C T_{sol}^2 \tag{27}$$

where, all of above cofactors (A to E) can be calculated as shown in [3].

At point 7, the refrigerant is in superheated state and its enthalpy, h_7 can be determined from

$$h_{sh} = ((H_{SH2} - H_{SH1})/100)T + H_{SH1} \tag{28}$$

where,

$$T = T_{ge} - T_{ref} \tag{29}$$

$$H_{SH1} = 32.508 \ln P + 2513.2 \tag{30}$$

$$H_{SH2} = 0.00001\, P^2 - 0.1193\, P + 2689 \tag{31}$$

The mass flow rate of dilute solution in the generator can be determined using the energy and mass balances on the generator, as:

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \text{and} x_3\dot{m}_3 = x_4\dot{m}_4 \tag{32}$$

The energy balance on the generator is given by,

$$\dot{Q}_{ge} = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3 \tag{33}$$

P_8 = P_4, T_8 can be calculated from Equation (25) and $h_8$can then be obtained by $h_f = h_g - h_{fg}$, where h_g and h_{fg} are given by the curve fit equations. The enthalpy at point 9 can be calculated by considering the throttling process, as $h_9 = h_8$.

$$h_g = -0.00125397T^2 + 1.88060937T + 2500.559 \tag{34}$$

$$h_{fg} = -0.00132635T^2 - 2.29983657T + 2500.43063 \tag{35}$$

The heat rejected at condenser, where$\dot{m}_7 = \dot{m}_8$, can be determined by writing the heat balance at condenser as:

$$\dot{Q}_{co} = \dot{m}_7(h_7 - h_8) \tag{36}$$

At the evaporator, P_{10} and h_{10} can be calculated from the curve fit of Equations (37) and (34).

$$P = 2\times10^{-12}T^6 - 3\times10^{-9}T^5 + 2\times10^{-7}T^4 + 3\times10^{-5}T^3 + 0.0014T^2 + 0.0444T + 0.6108 \tag{37}$$

From the energy balance at the evaporator, the cooling capacity $\dot{Q}_{ev}$ at the evaporator can be calculated from:

$$\dot{Q}_{ev} = \dot{m}_9(h_{10} - h_9) \tag{38}$$

The enthalpy at point 1 can be calculated by Equation (27) with absorber temperature (calculated by Equations (25) and (26)) at the same pressure as the evaporator.

The solution pump is modeled as an isenthalpic process, then $h_2 = h_1$. At the heat exchanger, the enthalpy h_5 can then be calculated as:

$$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5 \tag{39}$$

At the throttling process, $h_6 = h_5$. The enthalpy h_3 can be determined at the solution heat exchanger, and the absorber heat rejection can be calculated as:

$$\dot{Q}_{ab} = \dot{m}_{10} h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1 \tag{40}$$

All four heat quantities must be satisfy the chiller energy balance equation, expressed as

$$\dot{Q}_{ge} + \dot{Q}_{ev} - \dot{Q}_{co} - \dot{Q}_{ab} = 0 \tag{41}$$

Figure 3 summarizes the input and output parameters of the simulation. In the system configuration selection and system design, the input parameters were varied to obtain the maximum system performance.

4. Selection of the Best System Configuration

To select the best system configuration, the COP_{sys} of the three configurations listed in Section 2 were determined using Equation (42), where $\dot{Q}_{ev}$ is the heat absorption rate at evaporator, G_T is solar radiation, A_c is collector area, $\dot{m}_{BM}$ is biomass consumption rate and LHV_{BM} is biomass heating value.This performance index was determined for four sky conditions using weather data obtained from the meteorological station at the Asian Institute of Technology, Bangkok, Thailand (latitude of 14.08 °N) as shown in Figure 4. The simulation results of COP_{sys} of all system configurations for each sky condition are shown in Figure 5(a) to 5(d), respectively (using the input parameters given in Table 1 of [3]).

$$COP_{sys} = \dot{Q}_{ev} / (G_T A_c + \dot{m}_{BM} LHV_{BM}) \tag{42}$$

Figure 5 shows the simulation results of COP_{sys} for the three system configurations considered in figure 2, namely, cases 1 to 3. The results show that with the same amount of solar energy input, case 3 has the highest COP_{sys} both during night time (energized by auxiliary heater) and day time (energized by solar and auxiliary heater), and it yields the highest performance for all sky conditions. On a clear sky day, the monthly average daily COP_{sys} of case 1, case 2 and case 3 are 0.24, 0.29 and 0.48, respectively. During the period when the system is solely energized by the auxiliary heater, case 3 has significantly higher COP_{sys} (0.64) than the others. In addition, at the end of the day, the COP_{sys} of case 3 is very higher than the others as the stored heat can serve the chiller heat requirement without external heat source. Therefore, for these assumptions and conditions, case 3 is chosen as the best system configuration. In the following sections, case 3 will be the only configuration that will be considered.

5. Design of a Solar-biomass Hybrid Air Conditioning System

Figure 6 shows the schematic of the SBAC system. The first part (in the left) is a solar water heating system, which consists of a field of solar collectors, hot water storage tank and circulating pump. The second part (in the middle) is a biomass gasifier-boiler which consists of an automatic up-draft gasifier and gas-fired boiler. The part on the right hand side is an absorption air-conditioner and consists of an absorption chiller, fan coil unit (FCU), cooling tower and three pumps (cooling, hot and chilled water pumps).

5.1 Cooling load

The first step in the design of a cooling system is the estimation of cooling load of the conditioned space. The room is considered as a (residential) space for two persons and complying with an ambient ventilation rate of 1.12 m^3/min (0.00945 m^3/s/person) as the indoor air quality requirements. The ambient air condition of 31.9°C and 72.8 %RH is from the cooling and dehumidification design conditions (2% wet-bulb/mean coincident dry-bulb temperature) of Bangkok, Thailand. The room air condition of 26°C and 50 %RH is consistent with the ASHRAE's summer comfort condition. The cooling load comprises of latent heat load (LHL) and sensible heat load (SHL), and the total heat load (THL) were determined using the procedure presented in [5]. It was calculated using the equivalent temperature differential (ETD or ΔT_E) or cooling load temperature difference (CLTD) method. The results show that the maximum cooling load is about 4.5 kW or 1.3 tons of refrigeration (TR) and occurs at around 16:00-17:00.

5.2 Component selection

The component selection and design considerations for this system are discussed in this section. Solar collectors are available as flat plate and concentrating collectors. Collector efficiency, size (available area for installation), temperature requirements at load and cost of solar collector must be considered before final selection. For this system, flat plate solar collector was chosen since though the efficiency of evacuated tube solar collector is higher than flat plate solar collector by about 20% (at output temperature of 80-90 °C), its cost is about 40% more than the flat plate collector. Compared to evacuated tube collector, without the limitation of installation area, the flat plate solar collectors are cost effective, more architecturally adaptive, easy to fabricate and install, and requires less operating and maintenance costs [6]. With reduced costs and improved performances, concentrating collectors may be attractive in the future.

The storage tank to collector area ratio is one of the important parameters for designing the flat plate solar water heating system. This ratio, depends on heat demand (or chiller/load power), solar fraction [7] (or local solar energy potential) and requirement of solar energizing period (for solar cooling system).Usually, it is in the range of 25 to 100 l/m^2 and a 25 l/m^2 offers the best performance.

Gasification is one method for waste-to-energy conversion. It is a clean and offers high conversion efficiencies, and the possibility to convert different feedstock [8]. Since it can produce higher useful heat with lower environmental effects [9], it was chosen for this study. It works as an auxiliary boiler when solar energy is not enough and works as main heat source when the solar radiation is not available. The gasifier boiler is controlled by controller and supplies hot water to the chiller. In addition, by mean of gasification, this system can be possibly adapted for the other available feed stocks, such as refuse derived fuel (RDF) and municipal solid waste (MSW). Municipal solid waste can be utilized via gasification as a waste-to-energy conversion process, which is more environmental friendly compared to the conventional incineration systems and currently very attractive. So, such (SBAC) systems are interesting also options for urban areas. Among the gasifiers fixed bed reactor was selected. Among up-draft, down-draft, cross-draft, and open-core, down-draft and up-draft gasifiers outperforms others [9]. Up-draft gasifiers are suitable for the gasification of biomass containing higher ash and moisture content with higher thermal efficiency. The up-draft gasifier is essentially suitable for thermal applications. Up-draft gasifiers have been used for gasification of a variety of biomass e.g.: straw, bark, wood blocks, maize cobs, chips and pellets, waste pellets, and refuse derived fuel [10].

Among the refrigeration cycles driven by thermal energy, absorption cycle was chosen for this study because of the following reasons: First, with regard to the chiller dimension, absorption chiller has the smallest dimension compared to adsorption and desiccant systems. Second, regarding the coefficient of performance (COP), as the cycle mass is related to the refrigerant flow rate, the COP of the absorption chiller is higher. So, a smaller solar collector area is sufficient and will lead to lower (initial) cost of system than adsorption and desiccant systems [11]. Other systems, such as ejector and metal hybrid systems are unsuitable for air conditioning purpose, while vapor with thermal engines requires very high driving temperatures [12]. Thus, compared to the others, an absorption chiller is more feasible, reliable, and quiet [13]. Even though the COP of double- and triple-effect absorption chiller are higher than the single effect one, they must be driven with high temperature heat (higher than 100°C for double effect and higher than 150°C for triple effect). To date, most solar cooling systems have utilized single-effect chillers with low-temperature solar collectors [14].

Absorption chillers can be classified as direct- or indirect-fired type. In direct-fired units, the heat source, e.g.: gas or some other fuel, is burned in the chiller generator. But, the fuel for this chiller type should be clean and strictly operated at clean combustion and suitable temperature. Indirect-fired units use water, steam, or some other transfer fluid which brings heat from another source, such as solar water heating system, boiler or heat recovered from an industrial process. Indirect fired type was chosen for this study as the hot water from solar water heating system can be supplied directly to the chiller and the producer gas from gasifier may not be clean enough to combust inside the direct fired chiller.

The two (pairs of) working fluids available for absorption chillers, H_2O-NH_3 and LiBr-H_2O, have their inherent advantages and characteristics For applications above 0°C (primarily for air conditioning), LiBr-H_2O pair is preferred, while H_2O-NH_3 pair is usually employed for temperature needs below 0°C [15]. For the same operating temperature, the COP of H_2O-NH_3 pair is lower than that of LiBr-H_2O pair. This is in part due to rectification losses [16]. NH_3–H_2O absorption cycles have lower thermal efficiency (average COP = 0.6) and so requires higher initial cost and greater heat transfer areas [17]. H_2O is not toxic or flammable (unlike ammonia) and has a higher enthalpy of vaporization. This led to the choice of LiBr-H_2O absorption chiller system as the working fluid for this study.

5.3 Components sizing

For the prescribed cooling load and the input parameters of each component (specification data provided by the manufactures), the model (presented in [3]) was used for component or sub-system sizing which will thus form the complete system.

The size of solar water heating sub-system and auxiliary heating sub-system depends on the prescribed solar fraction (or solar to auxiliary heat ratio, SAR). If the required SAR is high (this value can be varied in the range 0<SAR<1), the solar water heating system must be bigger as compared to a system of lower SAR value. This section presents the design procedure to determine the appropriate sizes of the water heating system components e.g.: collector effective area (A_c), hot water storage tank size (V_{tank}) and minimum auxiliary heat supply rate ($\dot{Q}_{aux}$), for any prescribed SAR using the design flow chart as presented in Figure 7.

The solar fraction criterion of 0.7 (it means that more than 70% (yearly average) of heat energy supply from solar energy and less than 30% supply from the backup energy) was used as the basis for sizing the components. To choose the size of collector and storage tank, the model equations was used for the simulation. As the maximum and minimum monthly average daily solar insolations are in February and October, respectively, data for October was taken for the design as the minimum requirement of component size. This will ensure that this cooling system can be used throughout the year.

The mathematical models used for the simulation are based on energy and mass balance equations. The components of the system size are varied, and the optimal size that is able to provide the expected cooling capacity thus gives the specifications for the design.

Figure 8 shows the simulation results, which gives the monthly average daily cooling capacity close to the design criterion at 16,200 kJ/hr. It can be seen that, in October, the variation of cooling capacity is close to this design value when the tank size of about 0.5 to 2.0 m^3 and collector area of about 44-54 m^2 are used. For 10-hour period (7:00-17:00), the hot water should be heated and supplied to the generator of absorption chiller. A maximum COP_{sys} of 0.62, given by 1 m^3 tank size and 54 m^2 was chosen for this small size chiller.

The size of auxiliary heater relates to the size of biomass gasifier-boiler. To choose the size of an auxiliary heater, the maximum energy supplied by the auxiliary source in October was calculated by using the simulation results. To determine the size of auxiliary heat supply, the maximum heat supply rate (as the ability of each heater size to raise the water temperature, (i.e.) big heater can heat water faster than small heater) was varied with the limitation of hot water temperature (at the set point temperature) at the inlet of generator. The results are shown in Figure 9. The cooling capacity can be increased by supplying (higher) heat rate by the auxiliary heater. Based on the minimum cooling capacity of 16,200 kJ/hr during 7 – 17 hrs, the capacity of auxiliary heater is estimated at 6.5 kW (23,400 kJ/hr). The heater size depends on the energy conversion efficiency [3] and the heat loss through the evaporated water and hot water pipe surface. Considering the losses and the conversion efficiency, the auxiliary heater size has been taken as 15 kW. The cooling capacity during the highest insolation month was simulated, and the results shown in Figure 9. The results show that the influence of heater size is more prominent during the early morning and late evening hours, as expected.

Table 1 shows the system specifications obtained from the design. To satisfy the design criterion in this case, the main components, e.g.: solar collector area, tank size, set point temperature and biomass-gasifier boiler size, should be about 54 m^2, 1 m^3, 84 °C and 15 kW, respectively.

6. Conclusions and recommendations

The model based design of a renewable energy based solar-biomass hybrid air-conditioning system usingLiBr-H_2O absorption chiller has been presented. The design was done for the criterion, namely, cooling load, solar fraction to be greater than 4.5 kW and 0.7, respectively, and set point temperature of 84 °C. The model results indicate that for these needs and constraints, solar collector area, tank size, set point temperature and biomass-gasifier boiler size should be about 54 m^2, 1 m^2, 84 °C and 15 kW, respectively.

Acknowledgments

Boonrit Prasartkaew wishes to thank Rajamangala University of Technology Thunyaburi (RMUTT) for providing financial support for this research.

References

[1] M.P. Vinaspre, M. Bourouis, A. Coronas, A. Garcia, V. Soto, J.M. Pinazo, Monitoring and Analysys of an Absorption Air-conditioning System, Energy and Buildings 36 (2004), 933-943.

[2] A. Elsafty, A.J. Al-Daini, Economical comparison between a solar-powered vapour absorption air-conditioning system and a vapour compression system in the Middle East, Renewable Energy 25 (2002) 569-583.

[3] B. Prasartkaew, S. Kumar, A low carbon cooling system using renewable energy resources and technologies, Energy and Buildings 42 (2010) 1453–1462.

[4] B. Prasartkaew, S. Kumar, Experimental Study on the Performance of a Solar-Biomass Hybrid Air-Conditioning System, Renewable Energy 57 (2013) 86-93.

[5] C.P. Arora, Refrigeration and Air Conditioning, second ed., McGraw-Hill International Editions, 2001.

[6] A. Hobbi, K. Siddiqui, Optimal design of a forced circulation solar water heating system for a residential unit in cold climate using TRNSYS, Solar Energy 83 (2009) 700–714.

[7] D. Lindenberger, T. Bruckner, H.M. Groscurth, R. Kummel, Optimization of solar district heating systems: seasonal storage, heat pumps, and co-generation, Energy 25 (2000) 591–608.

[8] A.F. Kirkels, G.P.J. Verbong, Biomass gasification: Still promising? A 30-year global overview, Renewable and Sustainable Energy Reviews 15 (2011) 471–481.

[9] M. Puig-Arnavat, J.C. Bruno, A. Coronas. Review and analysis of biomass gasification models, Renewable and Sustainable Energy Reviews 14 (2010) 2841–2851.

[10] S. Chopra, A. Jain, A Review of Fixed Bed Gasification Systems for Biomass, Agricultural Engineering International: the CIGR Ejournal 5 (9) (2007).

[11] R.Z. Wang, T.S. Ge, C.J. Chen, Q. Ma, Z.Q. Xiong, Solar sorption cooling systems for residential applications: Options and guidelines, International Journal of Refrigeration 32 (2009) 638–660.

[12] C. Afonso, Recent advances in building air conditioning systems, Applied Thermal Engineering 26 (2006) 1961–1971.

[13] M. Qu, H. Yin, D.H. Archer, A solar thermal cooling and heating system for a building: experimental and model based performance analysis and design, Solar Energy 84 (2010) 166–182.

[14] C.A. Balaras, G. Grossman, H.M. Henning, C. Infante-Ferreira, E. Podesser, L. Wang, E. Wiemken, Solar air conditioning in Europe-an overview, Renewable and Sustainable Energy Reviews 11 (2007) 299–314.
[15] N. Rona, Solar Air-Conditioning Systems Focus on components and their working principles, (eBook-edition), Building Services Engineering (2004).
[16] K. Gommed, G. Grossman, Performance analysis of staged absorption heat pumps: water-lithium bromide systems, ASHRAE Transactions 96 (1990) 1590-1598.
[17] X.G. Casals, Solar Absorption Cooling in Spain: Perspectives and Outcomes from the Simulation of Recent Installations, Renewable Energy 31 (2006) 1371-1389.

Table 1: The design system specifications

Component	Inputs	Value	Unit
Solar Water Heating System			
Collector	Area, A_c	54	m^2
	Intercept efficiency,	0.789	-
	Loss efficiency coefficient,	5.829	W/m^2K
	Collector Flow rate,	1,200	kg/hr
Tank	Volume,	1	m^3
	Mass of fluid in tank,	1,000	kg
	Tank heat loss,	4.068	W/m^2K
	Hot water set temperature,	84	°C
Auxiliary Boiler			
	Maximum heating power	15	kW
Absorption Chiller (absorption chiller)			
	Absorption chiller size,	7	kW
	Generator water flow rate,	1,500	kg/hr

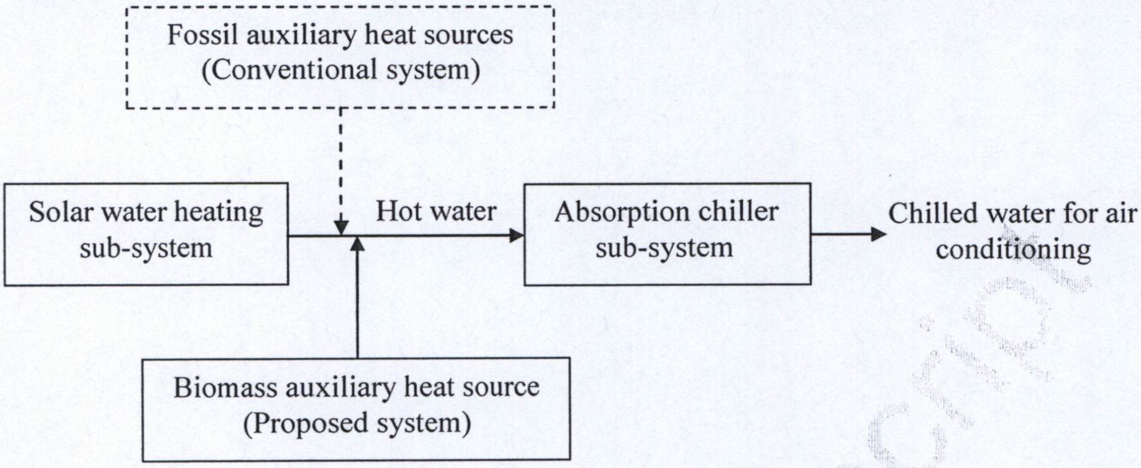
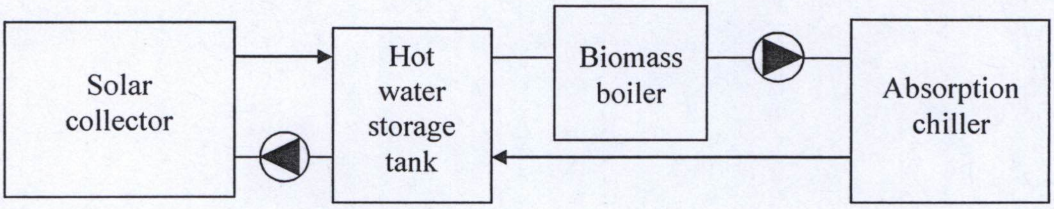
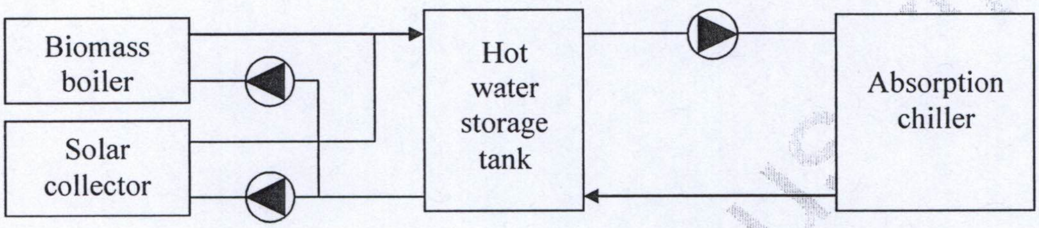


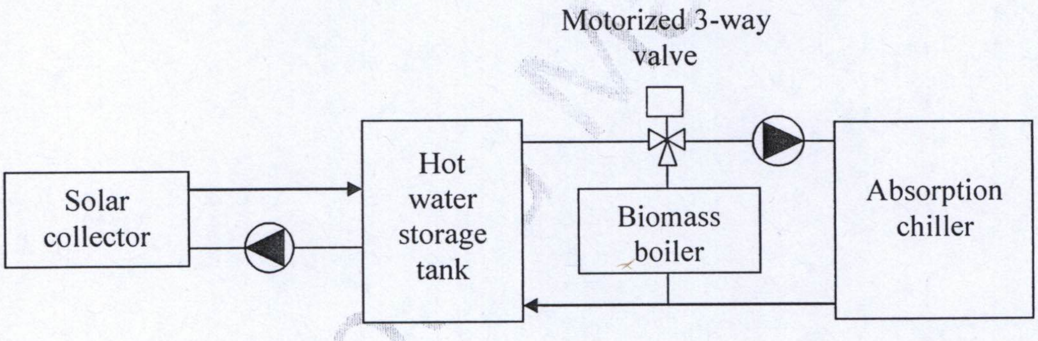
Figure 1: Conventional and proposed solar cooling system.



a) An auxiliary heater is connected in series with the solar water heating system (Case 1)



b) An auxiliary heater is connected in parallel with the solar collector (Case 2)



c) An auxiliary heater is connected in parallel with the solar water heating system (Case 3)

Figure 2: Three possible system configurations for supplying hot water to the absorption system.

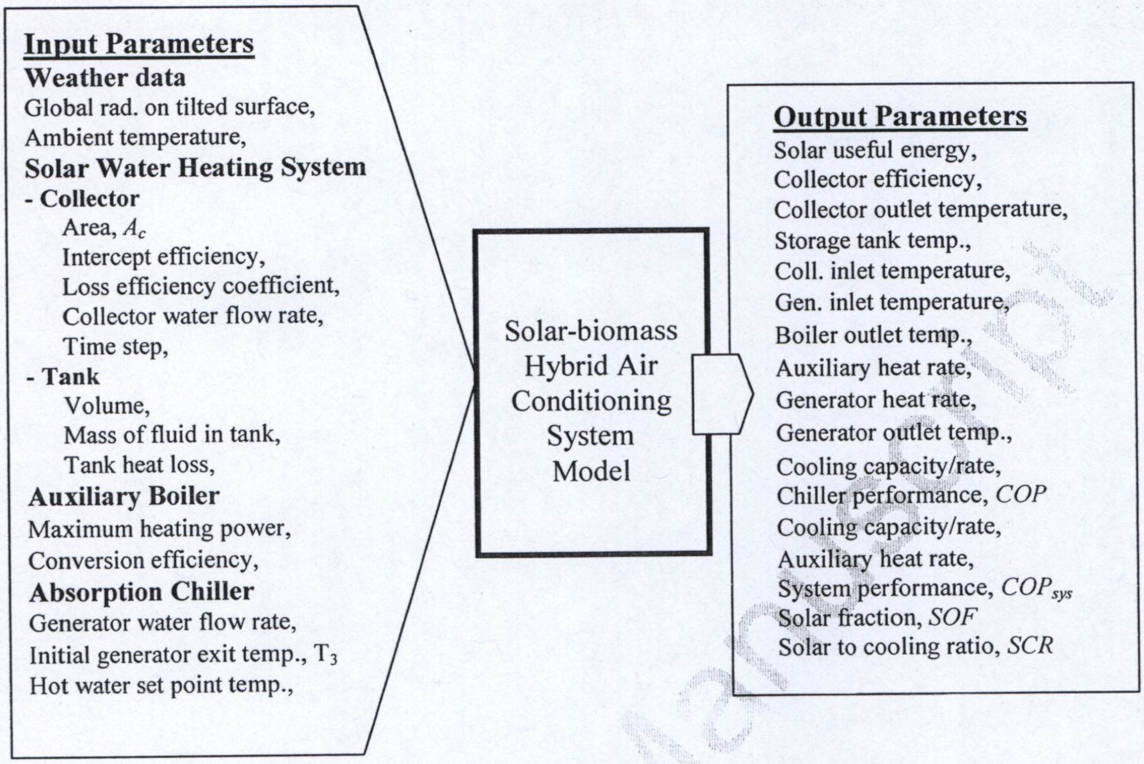


Figure 3: The inputs and outputs of the mathematical model.

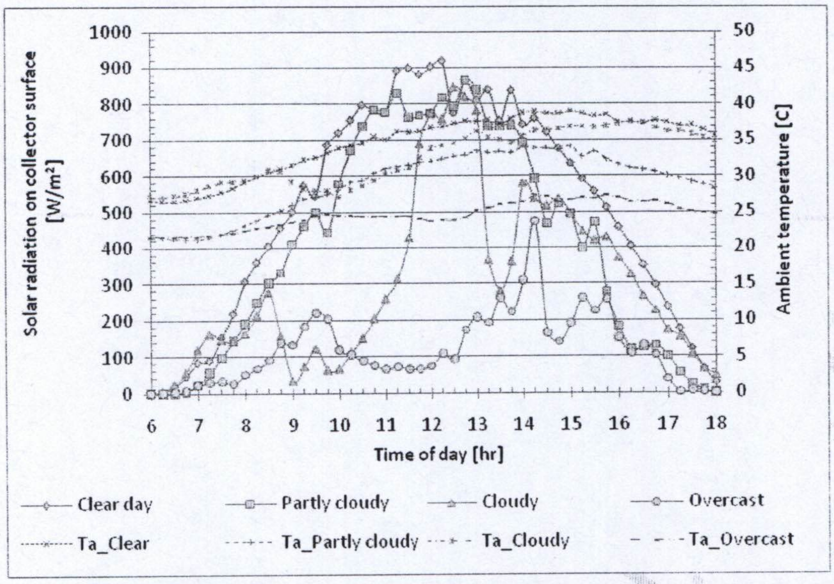
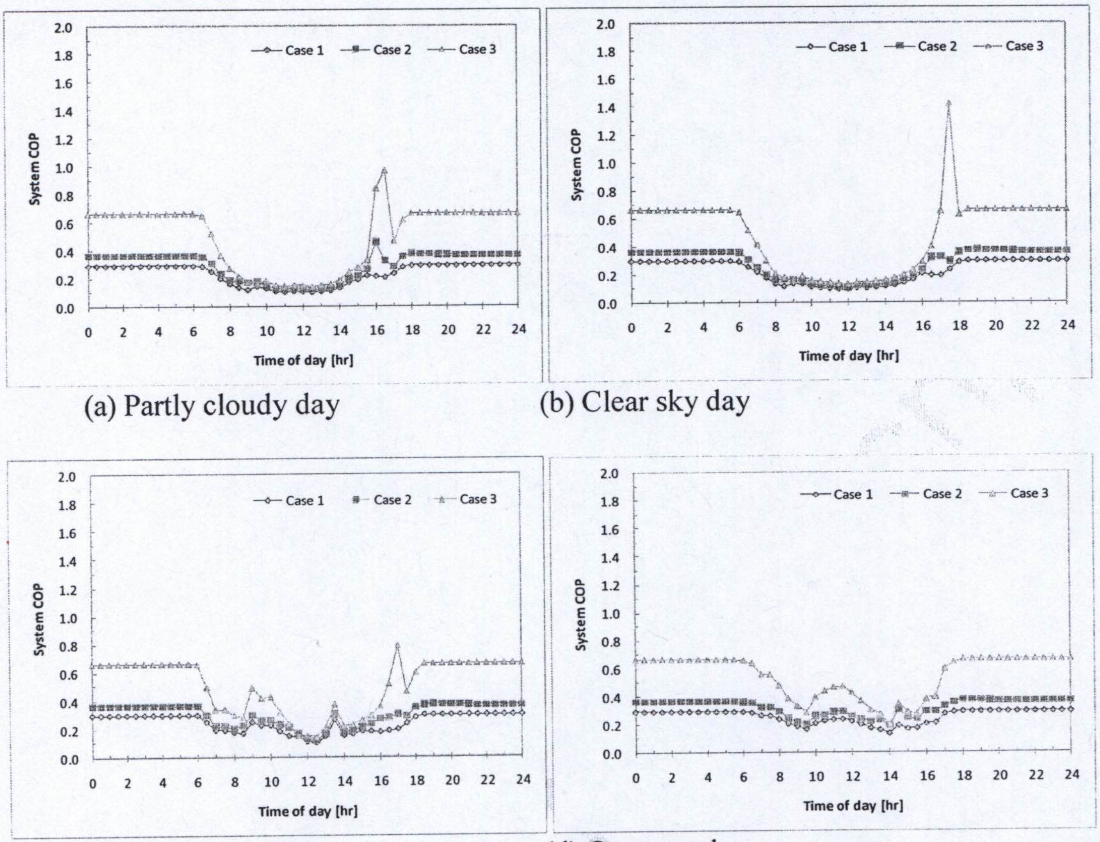


Figure 4: Solar radiation and ambient temperature of four sky conditions.



(a) Partly cloudy day (b) Clear sky day

(c) Cloudy day (d) Overcast day

Figure 5: The COP_{sys} for the three system configurations (cases) (Case 1: auxiliary heater is connected in series with the solar water heating. Case 2: auxiliary heater is connected in parallel to the solar collector. Case 3: auxiliary heater is connected in parallel with the solar water heating system).

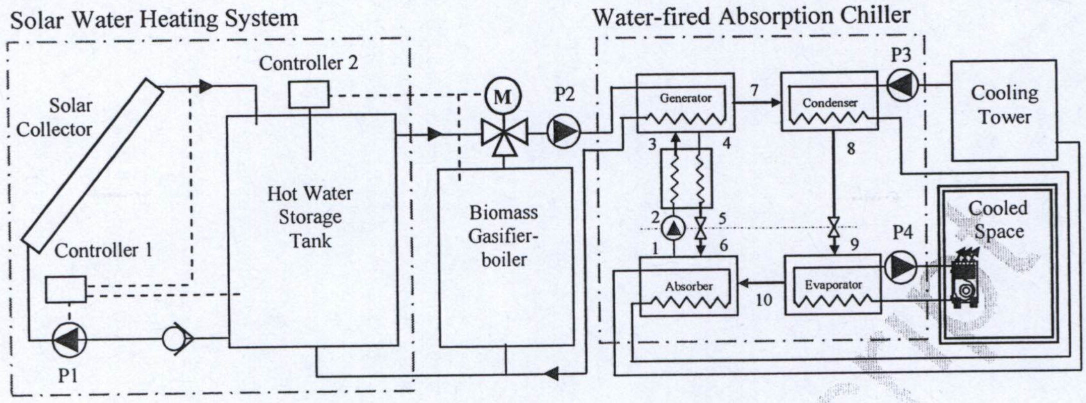


Figure 6: Schematic diagram of the solar-biomass hybrid cooling system.

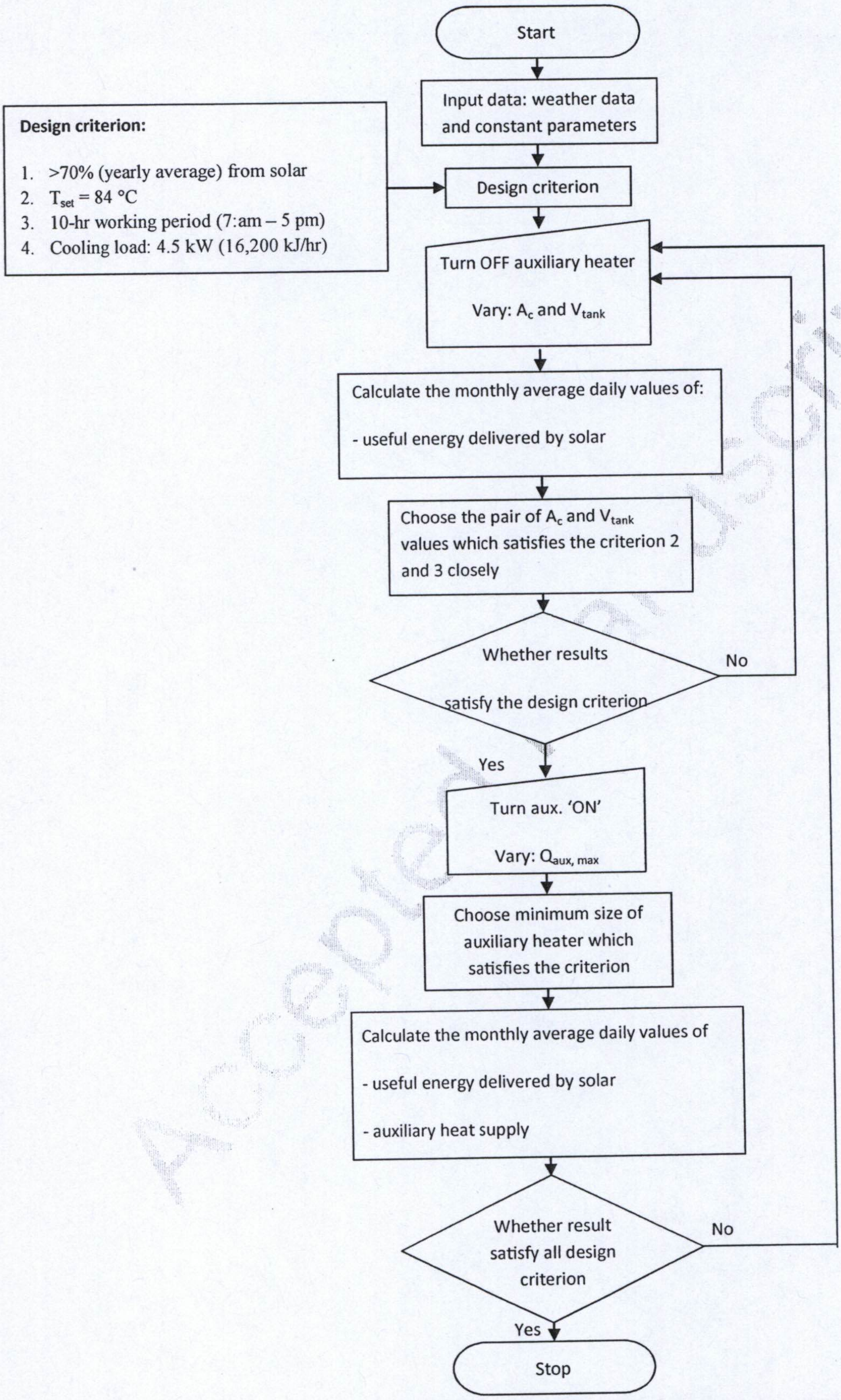


Figure 7: The flow chart of system design for the prescribed solar fraction.

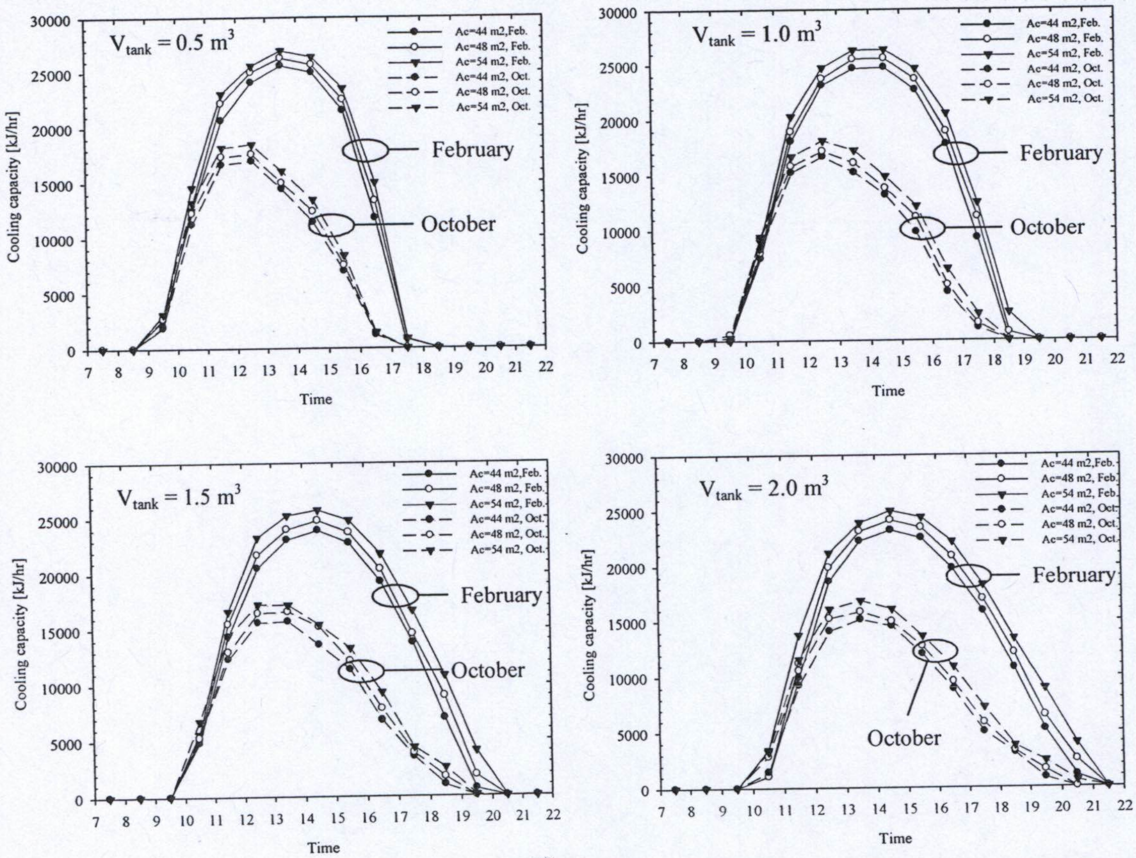


Figure 8: Monthly average daily cooling capacity as a function of tank size and collector area for the two extreme months: February and October.

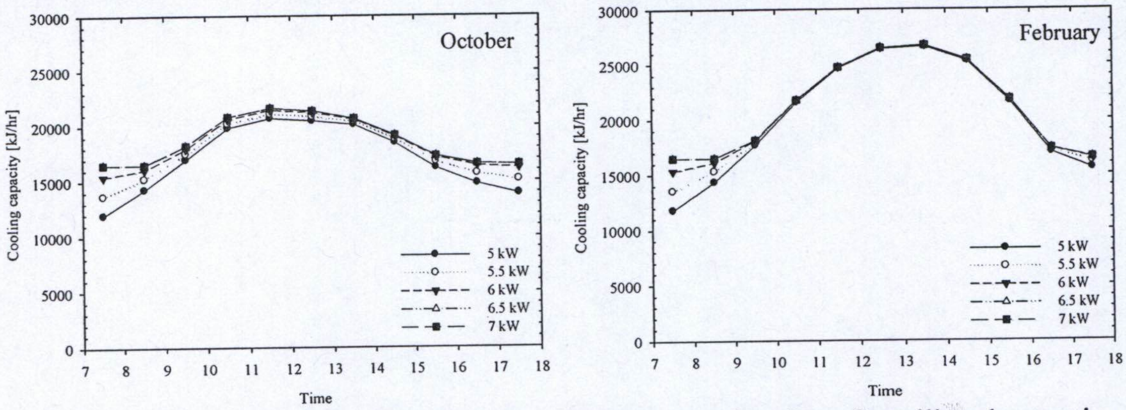


Figure 9: Monthly average hourly cooling capacity as a function of auxiliary heater size during the lowest and highest solar insolation months.

Research Highlights:

- New concept of a solar-biomass hybrid cooling system was presented.
- This paper presents the design of a solar-biomass hybrid system.
- 3-possible configurations were considered, to get the most suitable for the design.
- Design was to satisfy a cooling load of 4.5 kW, and solar to biomass ratio of 0.7.