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Mathematical Modeling of an Absorption Chiller System Energized by a Hybrid Thermal System: Model Validation

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Abstract

Nowadays, global warming and energy crisis problems become serious issues which affect on all creatures on our earth. One of the best ways to simultaneously address or mitigate these problems is more utilizing the renewable energy sources instead of the fossil fuel. A solar-biomass hybrid cooling system is one of the technologies for the climate change and green-house-gas mitigation. The mathematical model of this system was developed and used in the theoretical prediction of its performance and system design. To assess the accuracy of the developed mathematical model, the obtained experimental data is then compared with the simulation results, with the same operating parameters and weather conditions. This paper presents the validation of the developed model. The validation results show that the simulation results are in good agreement with the experimental results from both qualitative and quantitative points of view.

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Keywords: Solar-biomass hybrid; Cooling system; Simulation model; Validation

1. Introduction

Nowadays, the global warming and energy crisis problems become the most serious issues. These problems, mainly attributed to the combustion of fossil fuel [1], substantially affect on all life on the earth. To address these problems, the renewable energy based should be encouraged. Among the energy utilization systems, cooling applications demand higher energy for their functioning [2]. Solar cooling technology is environmentally friendly and contributes to a significant decrease of the CO₂ emissions which cause the green house effect [3]. From this point of view, solar powered cooling systems as a green

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แบบฟอร์มขอรับการสนับสนุนการตีพิมพ์บทความวิจัย มหาวิทยาลัยเทคโนโลยีราชมงคลธัญบุรี

ส่วนที่ 1 ประวัติผู้ขอรับการสนับสนุน

- 1.1. ชื่อ - สกุล..... ศพ.ดร. มนอภุทธิ์ มิ่งลาทแก้ว
- 1.2. ตำแหน่งทางวิชาการ..... ศพ.
- 1.3. สังกัด..... คณะวิศวกรรมศาสตร์
- 1.4. โทรศัพท์..... 02-549-3430 โทรสาร..... 02-549-3432

ส่วนที่ 2 รายละเอียดผลงาน (กรุณาขีดเครื่องหมาย ✓ หน้าข้อความที่ตรงกับคุณสมบัติ)

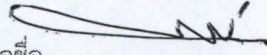
1. คุณสมบัติของผู้ขอรับการสนับสนุนการตีพิมพ์บทความวิจัยมหาวิทยาลัยเทคโนโลยีราชมงคลธัญบุรี
- 1.1 เป็นนักวิจัยของมหาวิทยาลัย และปฏิบัติราชการเต็มเวลามาแล้วไม่น้อยกว่า 9 เดือน โดยนับตั้งแต่วันรายงานตัว จนถึงวันที่เสนอขอรับเงินอุดหนุน
- 1.2 เป็นผู้เขียนชื่อแรก (First author) หรือผู้เขียนที่เป็นชื่อหลัก (Corresponding author) ที่ระบุชื่อหน่วยงานของมหาวิทยาลัยเทคโนโลยีราชมงคลธัญบุรีไว้ที่ตำแหน่งที่อยู่ของผู้เขียน ปรากฏในบทความอย่างชัดเจน (เอกสารแนบที่.....)
- 1.3 ยังไม่เคยได้รับทุนเพื่อตีพิมพ์เอกสารจากแหล่งทุนอื่น
2. คุณสมบัติของบทความวิจัยที่ขอรับการสนับสนุนการตีพิมพ์
- 2.1 เป็นบทความวิจัยที่ไม่เคยตีพิมพ์ในวารสารใดมาก่อน
- 2.2 ผลงานวิจัยที่ขอรับการสนับสนุนรางวัลไม่ใช่อะไหล่หนึ่งของการศึกษาระดับปริญญาโท หรือปริญญาเอกของผู้ขอรับการสนับสนุน
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- 5.6 เอกสารแสดงค่า Impact factor (ถ้ามี)

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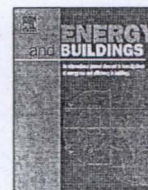
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Design of a renewable energy based air-conditioning system

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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-H₂O 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.

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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 CO₂ 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

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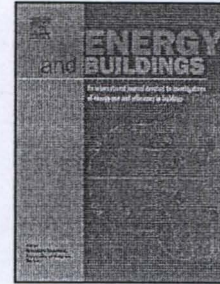
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Abstract

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-H₂O 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

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2		
3		
4	β	tank load control function
5	γ	temperature differential control function
6	φ	boiler load control function
7	ε	heat exchanger effectiveness
8	ρ	density (kg/m^3)
9		

Subscripts

10		
11		
12	0	initial condition
13	a	ambient
14	ab	absorber
15	aux	auxiliary
16	b	boiler
17	BM	biomass
18	bl	boiler to load
19	c	collector
20	co	condenser
21	ev	evaporator
22	f	liquid state
23	fg	liquid-vapor mixture
24	flu	flue gas
25	g	vapor state
26	ge	generator
27	gw	gas to water
28	hi	high
29	i	inlet
30	lo	low
31	o	outlet
32	PG	producer gas
33	ref	refrigerant
34	set	set point value
35	sol	solution of refrigerant and absorbent
36	st	storage tank
37	tl	tank to load
38	u	solar useful energy
39	w	water
40	we	water to environment
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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 CO₂ 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-

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5 engines to drive a compressor instead of an electrical motor. The heat transformation processes
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8

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10 Solar driven system offers a good model of a clean process for sustainable technology [2].
11 The two driving energy sources for the conventional solar cooling systems are without auxiliary
12 heat source and with fossil fuel auxiliary heater. As solar energy is intermittent, the solar cooling
13 system without auxiliary heat source cannot be continuously used, especially at night, and so its
14 reliability is low. Almost all solar cooling systems use gas (LPG/CNG) as an auxiliary heater [3].
15 Thus, a conventional solar cooling system has three major sub-systems -absorption chiller, solar
16 water heating system and backup/auxiliary heating sub-systems (renewable or fossil fuel based).
17

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19 A solar-biomass hybrid cooling system has been proposed, which is a completely renewable
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22 for the chosen option. Section 2 describes the possible system configurations of solar-biomass
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24 simulation inputs used for the design of the SBAC system. Section 4 presents the selection of the
25 best system configuration. Section 5 presents the design of a solar-biomass hybrid air-
26 conditioning system. Finally, the conclusions are given in Section 6.
27

28 29 30 31 **2. Possible System Configurations**

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33 As demonstrated in Figure 1, most conventional solar cooling systems use fossil fuel as an
34 auxiliary heat source, and so their operating cost and emissions are high. This research proposes
35 biomass energy as a cheaper (or free if it is the waste material) auxiliary heat source to address
36 the above issue. Thus, the main difference of the proposed system as compared to the
37 conventional system is that the fossil auxiliary heat source is replaced by biomass based gasifier
38 boiler.
39

40
41 Figure 2 show three possible system configurations for solar-biomass hybrid cooling system:
42 (Case 1) an auxiliary heater is connected in series with the solar water heating system, (Case 2) an
43 auxiliary heater is connected in parallel with solar collector, and (Case 3) an auxiliary heater is
44 connected in parallel with the solar water heating system.
45

46
47 The common feature in all the three cases is that the insulated biomass gasifier boiler has two
48 functions: it works as an auxiliary energy source when solar energy is not enough and works as
49 the main heat source when the solar radiation is not available. In case 1 and 3, the gasifier boiler
50 is controlled by a controller and supplies hot water to the generator of the absorption chiller. The
51 solar energy heats the water at the collector field which is then pumped to the hot water storage
52 tank. This pump will be activated by controller. Usually, it remains off until the difference
53 between collector outlet and inlet water temperature is above the upper dead band value. The
54 controller will switch the pump off when this difference reaches the lower dead band.
55

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57 The cooling is provided by a single-effect LiBr-H₂O absorption chiller. Heat from solar
58 collector or from biomass-boiler evaporates the water (strong solution of water and LiBr) in the
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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:

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

where,

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

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

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

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

$$(MC_p)_{st} (dT_{st}/dt) = \gamma \dot{Q}_{u} - \dot{Q}_{tl} - (UA)_{st} (T_{st} - T_a) \quad (5)$$

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

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

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

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

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:

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

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:

$$(MC_p)_b (dT_b/dt) = \dot{Q}_{gw} - \dot{Q}_{bl} - (UA)_{w\alpha} C_{p,\alpha} (T_b - T_a) \quad (10)$$

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:

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

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

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

where,

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

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

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

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

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

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

The consumption rate of biomass feed stock is obtained from:

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

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:

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

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

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

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

where,

$$T_{gs,i} = \beta T_{st} + \varphi T_b \quad (22)$$

$$\dot{Q}_{gsn} = \dot{Q}_{tl} + \dot{Q}_{bl} \quad (23)$$

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.

$$\log P = C + D/(T_{rsf} + 273) + E/(T_{rsf} + 273)^2 \quad (24)$$

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

$$T_{sol} = \Sigma B + T_{rsf} \Sigma A \quad (26)$$

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

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

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

where,

$$T = T_{gs} - T_{rsf} \quad (29)$$

$$H_{SH1} = 32.508 \ln P + 2513.2 \quad (30)$$

$$H_{SH2} = 0.00001 P^2 - 0.1193 P + 2689 \quad (31)$$

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

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

The energy balance on the generator is given by,

$$\dot{Q}_{gs} = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3 \quad (33)$$

$P_8 = P_4$, 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 $h_9 = h_8$.

$$h_g = -0.00125397T^2 + 1.88060937T + 2500.559 \quad (34)$$

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

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

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

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

$$P = 2 \times 10^{-12} T^6 - 3 \times 10^{-9} T^5 + 2 \times 10^{-7} T^4 + 3 \times 10^{-5} T^3 + 0.0014 T^2 + 0.0444 T + 0.6108 \quad (37)$$

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

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

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:

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

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:

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

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

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

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]).

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

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 $1.12 \text{ m}^3/\text{min}$ ($0.00945 \text{ m}^3/\text{s}/\text{person}$) 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 $80\text{-}90^\circ\text{C}$), 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,

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4 which is more environmental friendly compared to the conventional incineration systems and
5 currently very attractive. So, such (SBAC) systems are interesting also options for urban areas.
6 Among the gasifiers fixed bed reactor was selected. Among up-draft, down-draft, cross-draft, and
7 open-core, down-draft and up-draft gasifiers outperforms others [9]. Up-draft gasifiers are suitable
8 for the gasification of biomass containing higher ash and moisture content with higher thermal
9 efficiency. The up-draft gasifier is essentially suitable for thermal applications. Up-draft gasifiers
10 have been used for gasification of a variety of biomass e.g.: straw, bark, wood blocks, maize cobs,
11 chips and pellets, waste pellets, and refuse derived fuel [10].
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15 Among the refrigeration cycles driven by thermal energy, absorption cycle was chosen for
16 this study because of the following reasons: First, with regard to the chiller dimension, absorption
17 chiller has the smallest dimension compared to adsorption and desiccant systems. Second,
18 regarding the coefficient of performance (COP), as the cycle mass is related to the refrigerant flow
19 rate, the COP of the absorption chiller is higher. So, a smaller solar collector area is sufficient and
20 will lead to lower (initial) cost of system than adsorption and desiccant systems [11]. Other
21 systems, such as ejector and metal hybrid systems are unsuitable for air conditioning purpose,
22 while vapor with thermal engines requires very high driving temperatures [12]. Thus, compared to
23 the others, an absorption chiller is more feasible, reliable, and quiet [13]. Even though the COP of
24 double- and triple-effect absorption chiller are higher than the single effect one, they must be
25 driven with high temperature heat (higher than 100°C for double effect and higher than 150°C for
26 triple effect). To date, most solar cooling systems have utilized single-effect chillers with low-
27 temperature solar collectors [14].
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32 Absorption chillers can be classified as direct- or indirect-fired type. In direct-fired units, the
33 heat source, e.g.: gas or some other fuel, is burned in the chiller generator. But, the fuel for this
34 chiller type should be clean and strictly operated at clean combustion and suitable temperature.
35 Indirect-fired units use water, steam, or some other transfer fluid which brings heat from another
36 source, such as solar water heating system, boiler or heat recovered from an industrial process.
37 Indirect fired type was chosen for this study as the hot water from solar water heating system can
38 be supplied directly to the chiller and the producer gas from gasifier may not be clean enough to
39 combust inside the direct fired chiller.
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44 The two (pairs of) working fluids available for absorption chillers, $\text{H}_2\text{O-NH}_3$ and $\text{LiBr-H}_2\text{O}$,
45 have their inherent advantages and characteristics For applications above 0°C (primarily for air
46 conditioning), $\text{LiBr-H}_2\text{O}$ pair is preferred, while $\text{H}_2\text{O-NH}_3$ pair is usually employed for
47 temperature needs below 0°C [15]. For the same operating temperature, the COP of $\text{H}_2\text{O-NH}_3$ pair
48 is lower than that of $\text{LiBr-H}_2\text{O}$ pair. This is in part due to rectification losses [16]. $\text{NH}_3\text{-H}_2\text{O}$
49 absorption cycles have lower thermal efficiency (average COP = 0.6) and so requires higher initial
50 cost and greater heat transfer areas [17]. H_2O is not toxic or flammable (unlike ammonia) and has
51 a higher enthalpy of vaporization. This led to the choice of $\text{LiBr-H}_2\text{O}$ absorption chiller system as
52 the working fluid for this study.
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56 5.3 Components sizing

57 For the prescribed cooling load and the input parameters of each component (specification
58 data provided by the manufactures), the model (presented in [3]) was used for component or sub-
59 system sizing which will thus form the complete system.
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5 The size of solar water heating sub-system and auxiliary heating sub-system depends on the
6 prescribed solar fraction (or solar to auxiliary heat ratio, SAR). If the required SAR is high (this
7 value can be varied in the range $0 < \text{SAR} < 1$), the solar water heating system must be bigger as
8 compared to a system of lower SAR value. This section presents the design procedure to
9 determine the appropriate sizes of the water heating system components e.g.: collector effective
10 area (A_c), hot water storage tank size (V_{tank}) and minimum auxiliary heat supply rate (\dot{Q}_{aux}), for
11 any prescribed SAR using the design flow chart as presented in Figure 7.
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15 The solar fraction criterion of 0.7 (it means that more than 70% (yearly average) of heat
16 energy supply from solar energy and less than 30% supply from the backup energy) was used as
17 the basis for sizing the components. To choose the size of collector and storage tank, the model
18 equations was used for the simulation. As the maximum and minimum monthly average daily
19 solar insulations are in February and October, respectively, data for October was taken for the
20 design as the minimum requirement of component size. This will ensure that this cooling system
21 can be used throughout the year.
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24 The mathematical models used for the simulation are based on energy and mass balance
25 equations. The components of the system size are varied, and the optimal size that is able to
26 provide the expected cooling capacity thus gives the specifications for the design.
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29 Figure 8 shows the simulation results, which gives the monthly average daily cooling
30 capacity close to the design criterion at 16,200 kJ/hr. It can be seen that, in October, the variation
31 of cooling capacity is close to this design value when the tank size of about 0.5 to 2.0 m³ and
32 collector area of about 44-54 m² are used. For 10-hour period (7:00-17:00), the hot water should
33 be heated and supplied to the generator of absorption chiller. A maximum COP_{sys} of 0.62, given
34 by 1 m³ tank size and 54 m² was chosen for this small size chiller.
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38 The size of auxiliary heater relates to the size of biomass gasifier-boiler. To choose the size
39 of an auxiliary heater, the maximum energy supplied by the auxiliary source in October was
40 calculated by using the simulation results. To determine the size of auxiliary heat supply, the
41 maximum heat supply rate (as the ability of each heater size to raise the water temperature, (i.e.)
42 big heater can heat water faster than small heater) was varied with the limitation of hot water
43 temperature (at the set point temperature) at the inlet of generator. The results are shown in Figure
44 9. The cooling capacity can be increased by supplying (higher) heat rate by the auxiliary heater.
45 Based on the minimum cooling capacity of 16,200 kJ/hr during 7 – 17 hrs, the capacity of
46 auxiliary heater is estimated at 6.5 kW (23,400 kJ/hr). The heater size depends on the energy
47 conversion efficiency [3] and the heat loss through the evaporated water and hot water pipe
48 surface. Considering the losses and the conversion efficiency, the auxiliary heater size has been
49 taken as 15 kW. The cooling capacity during the highest insolation month was simulated, and the
50 results shown in Figure 9. The results show that the influence of heater size is more prominent
51 during the early morning and late evening hours, as expected.
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56 Table 1 shows the system specifications obtained from the design. To satisfy the design
57 criterion in this case, the main components, e.g.: solar collector area, tank size, set point
58 temperature and biomass-gasifier boiler size, should be about 54 m², 1 m³, 84 °C and 15 kW,
59 respectively.
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6. Conclusions and recommendations

The model based design of a renewable energy based solar-biomass hybrid air-conditioning system using LiBr-H₂O 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 m², 1 m², 84 °C and 15 kW, respectively.

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- Figure 1:** Conventional and proposed solar cooling system.
- 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.
- 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.
- Table 1:** The design system specifications.

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
Tank	Collector Flow rate,	1,200	kg/hr
	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	$^{\circ}C$
Auxiliary Boiler			
	Maximum heating power	15	kW
Absorption Chiller (absorption chiller)			
	Absorption chiller size,	7	kW
	Generator water flow rate,	1,500	kg/hr

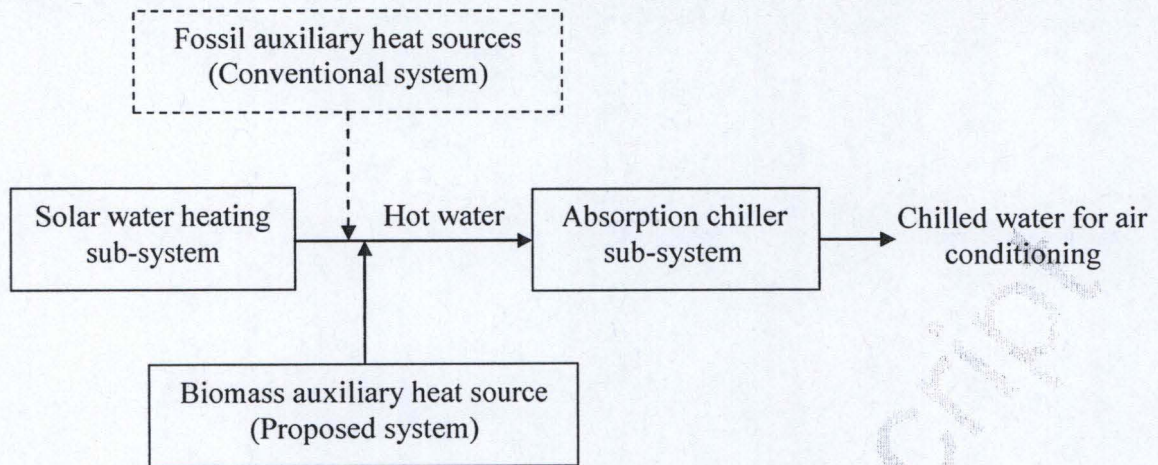
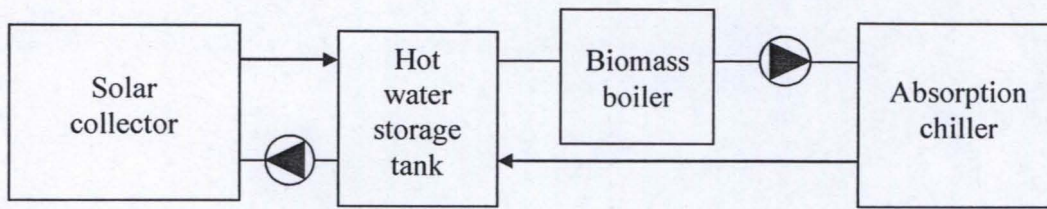
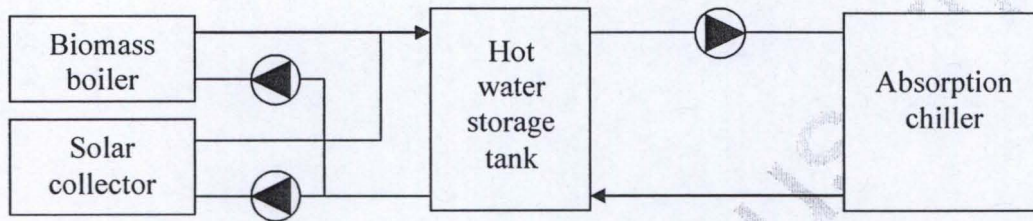


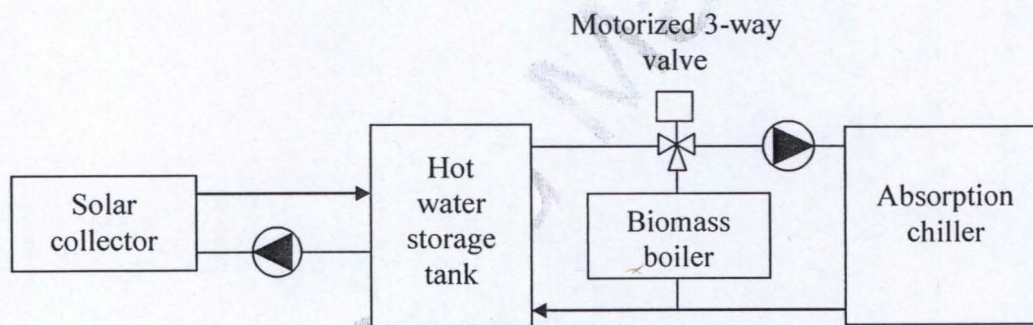
Figure 1: Conventional and proposed solar cooling system.



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.

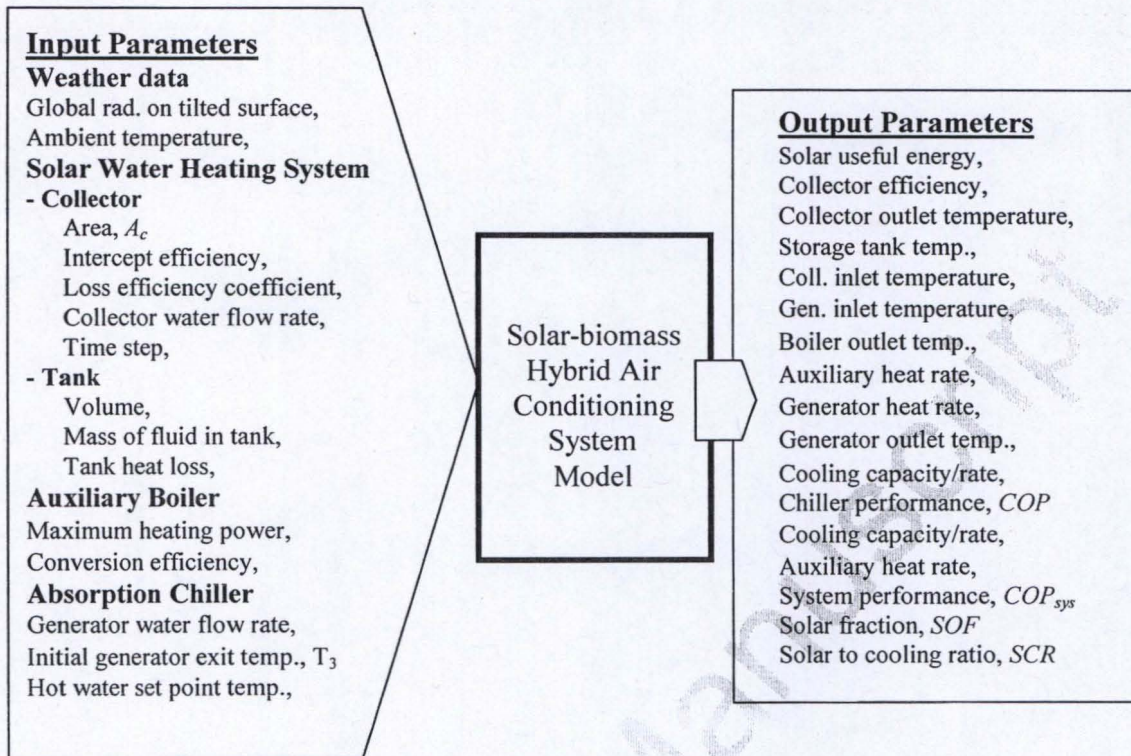


Figure 3: The inputs and outputs of the mathematical model.

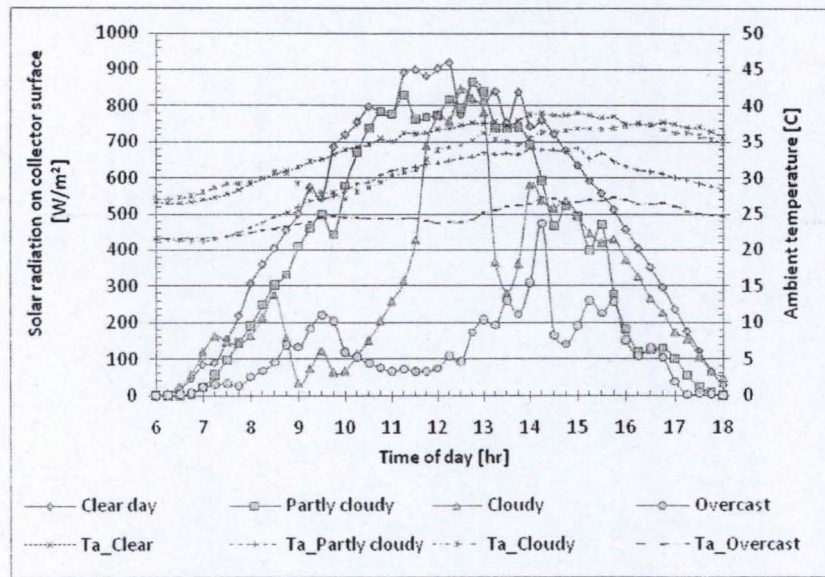


Figure 4: Solar radiation and ambient temperature of four sky conditions.

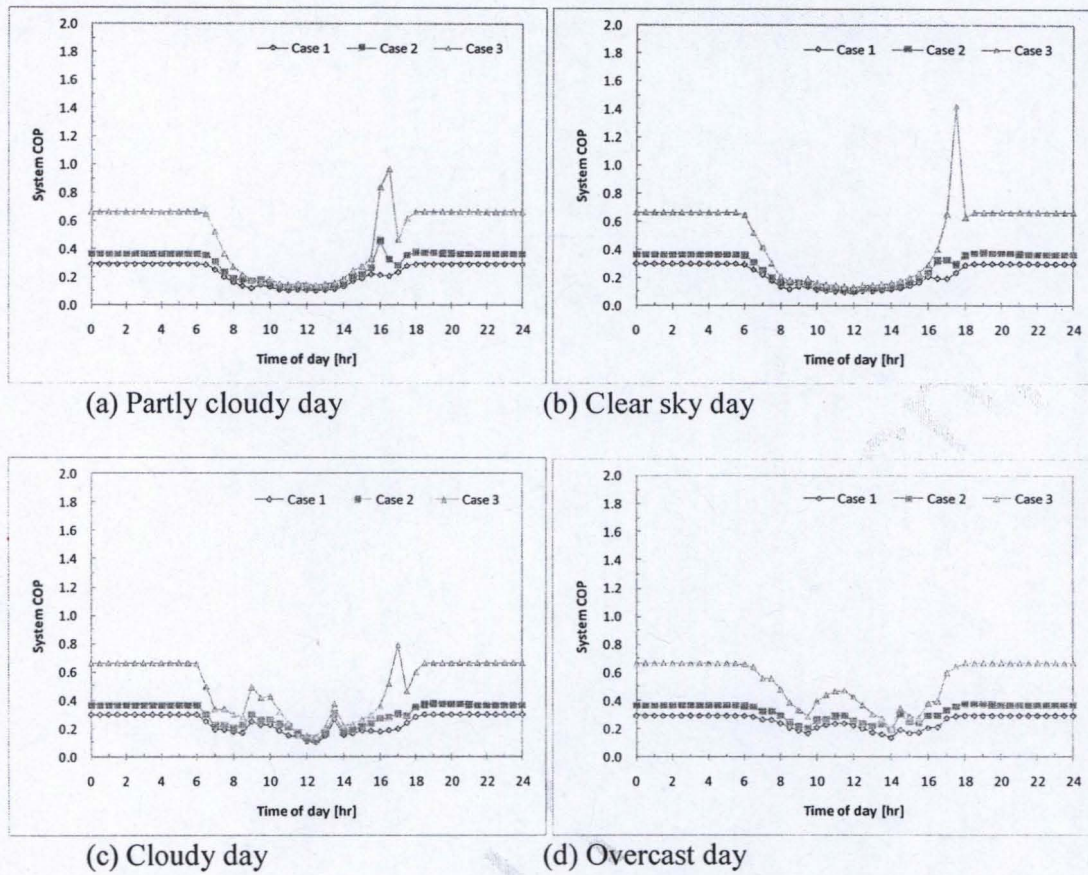


Figure 5: The COP_{sys} for the three system configurations (cases) (Case 1: auxiliary heater is connected in series with the solar water heating. Case 2: auxiliary heater is connected in parallel to the solar collector. Case 3: auxiliary heater is connected in parallel with the solar water heating system).

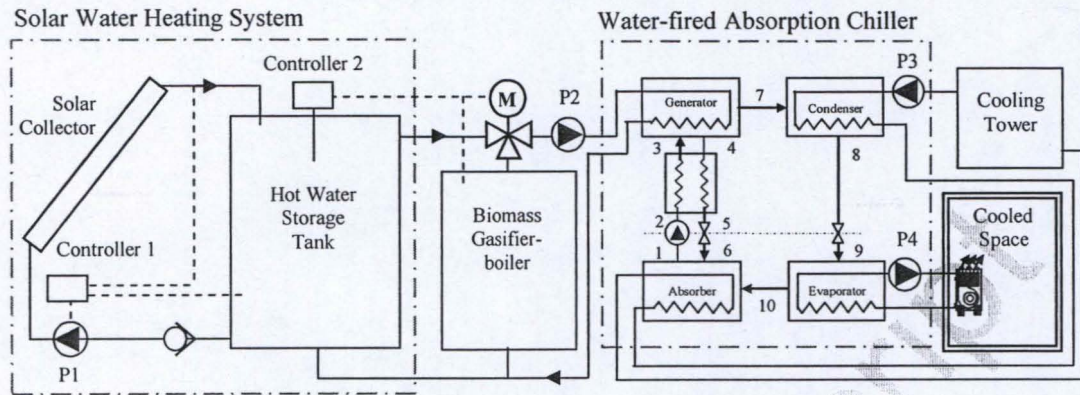


Figure 6: Schematic diagram of the solar-biomass hybrid cooling system.

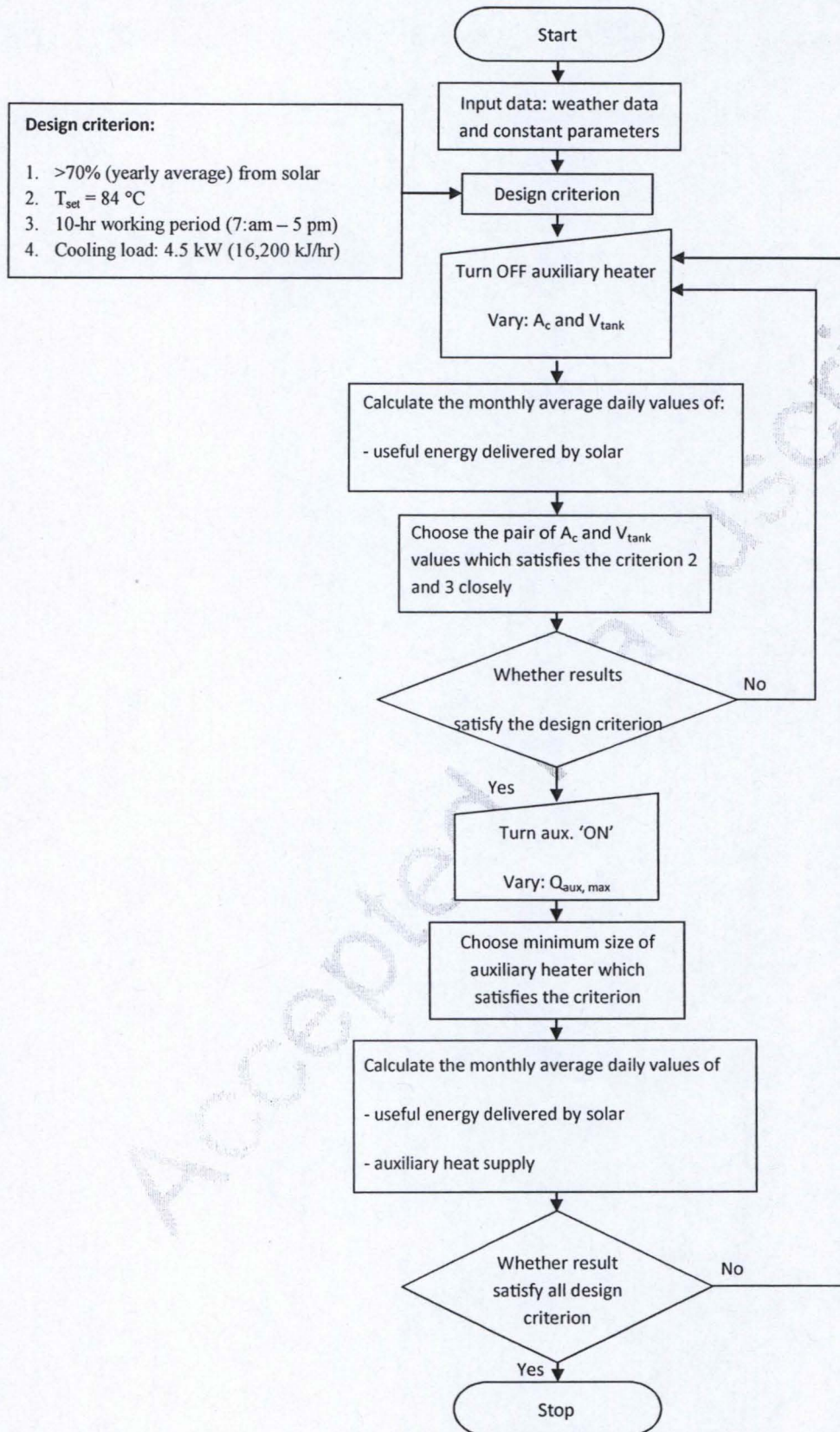


Figure 7: The flow chart of system design for the prescribed solar fraction.

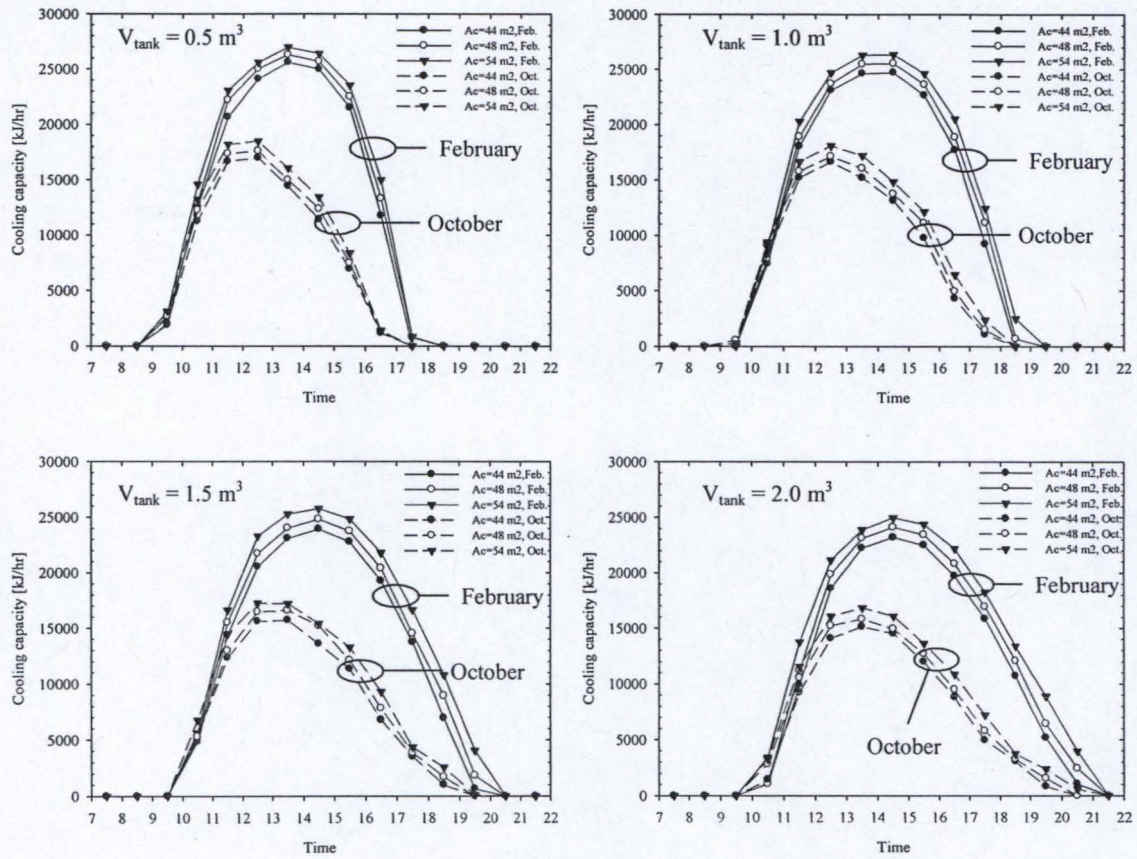


Figure 8: Monthly average daily cooling capacity as a function of tank size and collector area for the two extreme months: February and October.

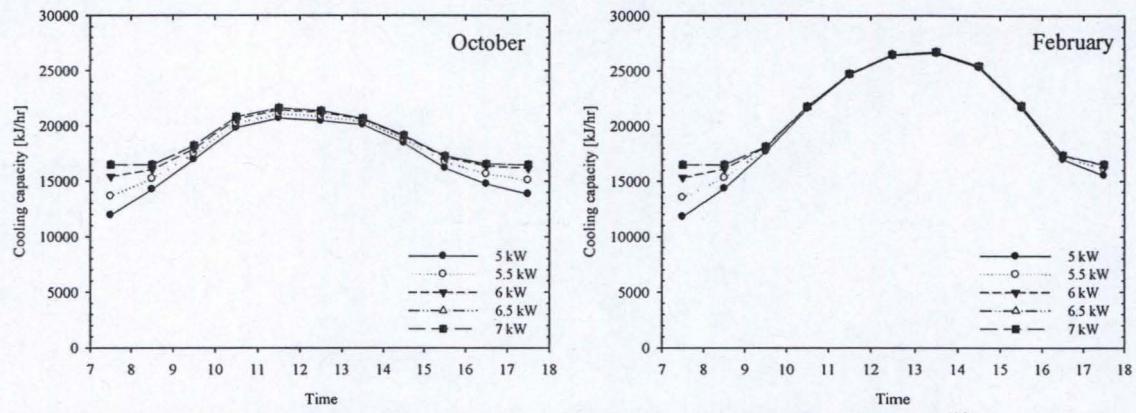
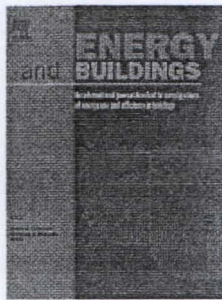


Figure 9: Monthly average hourly cooling capacity as a function of auxiliary heater size during the lowest and highest solar insolation months.

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.

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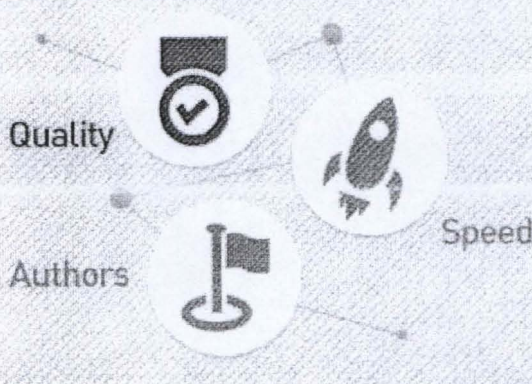
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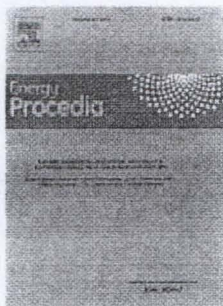
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