Assessment of Biomass Feedstock as Auxiliary Heat Sources Required for a Solar-biomass Hybrid Cooling System

Boonrit Prasartkaew[1,]*

[1] Department of Mechanical Engineering, Faculty of Engineering, Rajamangala University of Technology Thunyaburi, Klong 6, Thunyaburi, Patumthani 12110, Thailand
*Corresponding Author: E-mail: boonrit_p@rmutt.ac.th or prasartkaew@yahoo.com,
Tel. (662) 549-3547, Fax. (662) 549-3432

Abstract

According to global warming and energy crisis problems, renewable energy based systems should be encouraged. The use of solar energy for cooling purpose is attractive because of the cooling load is roughly in phase with solar energy availability. In addition, Thailand has fruitfully biomass sources from waste and crop. This study presents the simulation study using validated mathematical model aims at assessing the performance of solar-biomass hybrid absorption cooling system and to determine the required auxiliary heat for this system. The simulation results show that the lowest and highest requirements of biomass auxiliary energy are in December and October, respectively. The annual averaged energy of solar, biomass and cooling capacity are 201.19, 86.94 and 188.56 MJ/day, respectively. The chiller coefficient of performance varies between 0.64 and 0.67 with the annual averaged value at 0.65. In addition, the yearly averaged consumption rate of some available biomass feedstock; charcoal, wood chip, rice husk, bagasse and coconut shell are 5.80, 11.44, 12.16, 12.07 and 9.71 kg/TR/day, respectively.
***Keywords**:* Solar; Absorption; Biomass; Cooling; Hybrid.

Nomenclature

A	area (m^2)
C_p	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 (MJ/kg, MJ/m^3)
M	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}$	energy rate (kW)
T	temperature (K)
t	time (s)
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	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
cg	combustion gas
ev	evaporator
f	liquid state
fg	liquid-vapor mixture
g	vapor state
ge	generator
gw	gas to water
hi	high
i	inlet
lo	low
o	outlet
PG	producer gas
set	set point value
st	storage tank
tl	tank to load
u	solar useful energy
w	water
we	water to environment

1. Introduction

Thailand has a hot and humid climate and its mean ambient temperature has been increasing continuously. The peak electricity demand has also dramatically increased during the summer period [1] mainly because of the use of air-conditioning systems. This also requires new plants or higher generation rate, and this consequently contributes to an increase of CO_2 emissions leading to global warming. The higher the ambient temperature, the more will be the use of air conditioner and electricity. In the past few years, due to the environmental concern originated by CFCs and increase of global temperature and electricity demand, the interest in absorption air-conditioner has increased [2].

To address the global warming and energy crisis issues, renewable energy based environmental friendly systems could be used. The use of solar energy for cooling purpose is attractive because the cooling load is roughly in phase with solar energy availability. To cool with solar energy using vapor absorption system, LiBr-water absorption systems is one option and they are commercially available. Absorption chillers are well-known as cooling systems which can be driven by solar or low grade heat. Most absorption cooling and solar cooling systems are currently equipped with fossil auxiliary fuels, e.g.: LPG, CNG and oil, thus these systems are fossil energy based cooling systems.

Biomass, such as wood, has many advantages: it can be obtained from the wastes, can be cropped within a short period, can capture CO_2 and generate O_2 [3]. Compared to fossil based systems, biomass utilization processes will not emit more CO_2 to the atmosphere [4]. In this sense, solar and low grade heat especially biomass, as CO_2 neutral energy source, can contribute also to the reduction of CO_2 emission [5]. This research purposes a solar-biomass hybrid air conditioning system which developing to be a fully renewable energy system. This paper presents the prediction results aims at assessing the performance of solar-biomass hybrid absorption cooling system and to determine the required auxiliary heat for this system under Thailand conditions.

2. System Description

The proposed solar-biomass hybrid absorption cooling system, as shown in Fig. 1 consists of three sub-systems: solar water heating and storage tank, automatic biomass gasifier-boiler, and absorption chiller with cooling tower.

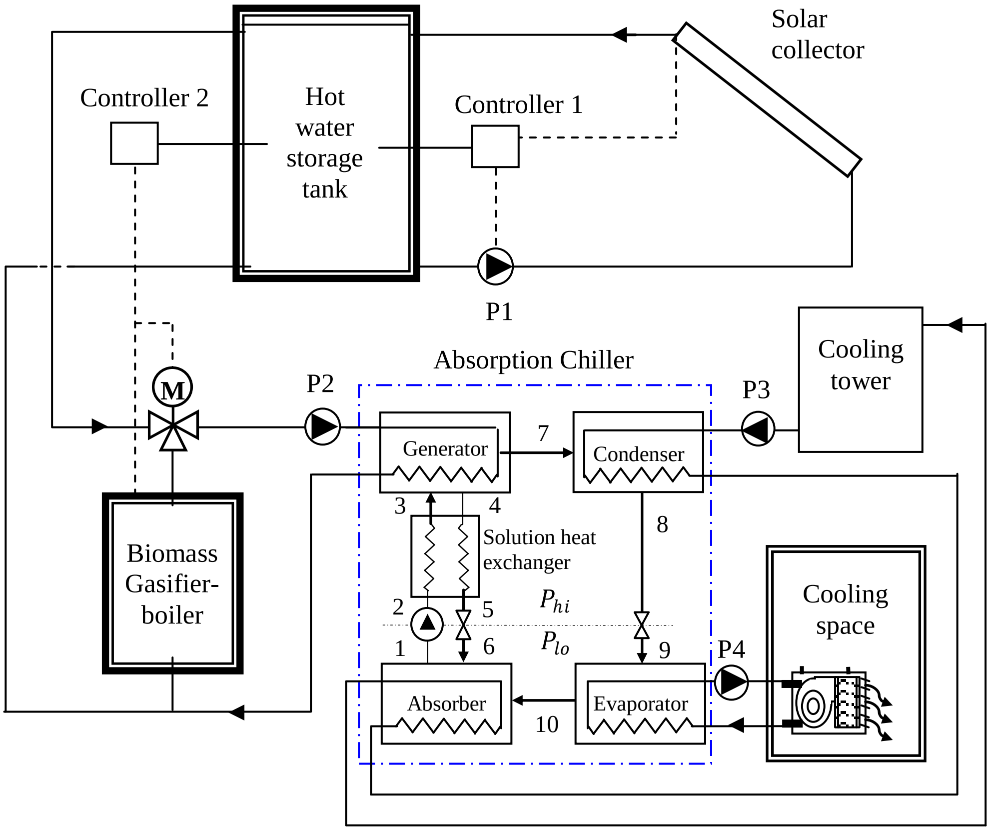

Fig. 1 Schematic diagram of the solar-biomass hybrid absorption cooling system

The solar energy is absorbed for heating the water as a working fluid at the collector field and pumped to the tank by the forced pump 1. This pump will be activated by temperature differential controller. Usually, it remains off until the difference between collector outlet and average tank temperature is above the upper dead band value. The controller will switch the pump off when this difference reaches the lower dead band. The second part is automatic on/off biomass gasifier boiler located between hot water storage tank and absorption chiller. This insulated boiler has two functions: works as auxiliary boiler when solar energy is not enough and works as main heat source when the solar radiation is not available. Because of its intermittent working conditions, this kind of gasifier boiler is proposed to utilize the available biomass resources. The automatic gasifier boiler is controlled by another controller and supplies hot water for absorption chiller. In this study, a small refrigeration system proposed is a commercial single-effect lithium bromide (LiBr)-water absorption chiller. The heat required for its generator is drawn from hot water pumped (pump 2) from a storage tank fed by the solar collectors and sometimes boosted/fed by biomass boiler when the tank temperature is not high enough. The condenser and absorber of chiller are cooled by cooling water pumped through a cooling tower. The chilled water produced from evaporator is pumped by pump 4 for cooling proposes.

3. Mathematical Model and Input Data

The proposed system was analyzed based on the validated model described in [5,6]. The system model was developed with the following assumptions:

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

3.1 Solar water heating system

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

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

where,

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

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

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

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

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

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

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

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

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

Whenever the temperature of hot water supplied to the chiller machine is lower than SPT, 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:

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

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:

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

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

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

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

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

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 following assumptions.

1) This model is a steady state model and the steady state refrigerant is pure water.
2) There are no pressure changes except through the flow restrictors and the pump.
3) At points 1, 4 and 8, there is only saturated liquid.
4) At point 10, there is only saturated vapour.
5) There is no liquid carryover from evaporator to absorber.
6) Flow restrictors are adiabatic.
7) The pump is isentropic.
8) There are no jacket heat losses.
9) Heat flow at condenser and absorber can be fully removed by cooling water supplied by a cooling tower. Therefore, the temperature and mass flow rate of the cooling water circuit are not taken into account.

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:

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

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

where,

$$T_{gei} = \beta T_{st} + \varphi T_b \quad (22)$$

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

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.

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

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

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

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

where, all of above cofactors can be calculated as:

T_{sol} is solution temperature (°C)
T_{ref} is refrigerant saturated temperature (°C)
P is saturation pressure (kPa)
A_0 = -2.00755
A_1 = 0.16976

A_2 = -0.003133362
A_3 = 0.0000197668
B_0 = 124.937
B_1 = -7.71649
B_2 = 0.152286
B_3 = -0.0007959
C = 7.05
D = -1596.49
E = -104095.5
ΣA = $A_0X^0 + A_1X^1 + A_2X^2 + A_3X^3$
ΣB = $B_0X^0 + B_1X^1 + B_2X^2 + B_3X^3$

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,

$$T = T_{ge} - T_{ref} \quad (29)$$

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

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

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

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

The energy balance on the generator is given by,

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

$P_8 = P_4$, T_8 can be calculated from Eq. (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 $h_9 = h_8$.

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

$$h_{fg} = -0.001326635T^2 - 2.29983657T + 2500.43063 \quad (35)$$

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 Eqs. (37) and (34).

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

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 Eq. (27) with absorber temperature (calculated by Eqs. (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:

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

To determine the solution pump power, it is assumed that the specific volume at point 1 and 2 is constant. This value can then be determined using the density equation of solution [6]. The minimum pump power can be determined by

$$w = \dot{m}_1 v_1 (P_2 - P_1) \quad (41)$$

$$\rho_x = 1145.36 + 470.84X_0 + 1374.79X_0^2 - (0.333393 + 0.571749X_0)(273 + T) \quad (42)$$

where,

$$X_0 = X_1/100 \quad (43)$$

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

$$\dot{Q}_{ge} + \dot{Q}_{ev} - \dot{Q}_{co} - \dot{Q}_{ab} = 0 \quad (44)$$

The simulations were performed using weather data obtained from the meteorological station at the Asian Institute of Technology (AIT), Bangkok, Thailand (latitude of 14.08 ◦N).

Fig. 2 shows the total solar irradiation and ambient temperature through 8,760 hours in 2010. The yearly average temperature was nearly 30°C and the difference between maximum and minimum was about 10°C. The maximum temperature is close to 40°C. Almost all ambient temperature during daytime are higher than 30°C. Only a few ambient temperatures during nighttime, at the beginning of December and the middle of January are lower than 20°C but not lower than 16°C. This suggests that, for human body comfort, there needs 24-hour cooling air conditioning for residential room where there has many heat gains from solar energy, human body and household apparatuses. The maximum solar was received during the clear sky day. On the other hand, the minimum solar was received in the cloudy day at the beginning of rainy season.

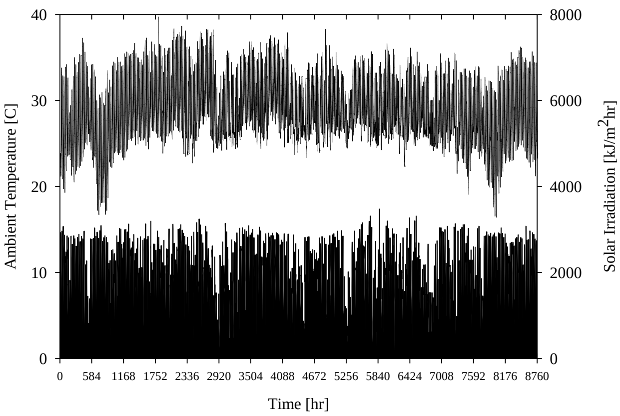

Fig. 2 The hourly variation of ambient temperatures and total solar irradiations in 2010

4. Simulation Results

4.1 Annual average energy analysis

To estimate over the year for a 3.5-kW chiller (1 TR), the simulation results through 7:00-17:00 period of working time were integrated into the total energy per year of energy inputs; supplied by collector and heater and cooling capacity. The simulation results show that the lowest and highest requirements of auxiliary energy are in December and October, respectively as shown in Fig. 3. The annual averaged energy of solar, auxiliary and cooling capacity are 201.19, 86.94 and 188.56 MJ/day, respectively. The system coefficient of performance varies between 0.64 and 0.67 with the annual averaged value at 0.65. Finally, the annual average energy fraction of solar and auxiliary energy are 70% and 30% of total input energy, both values are equal to the criterion base requirements.

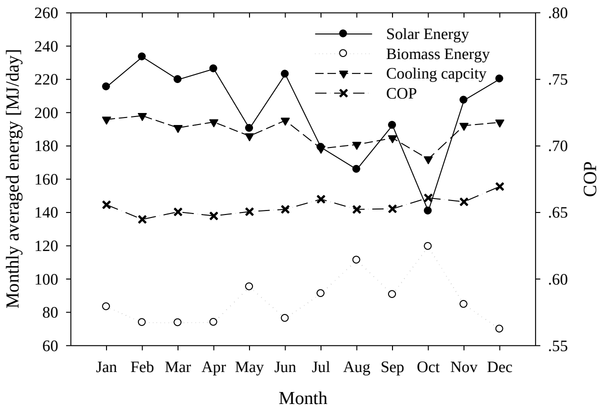

Fig. 3 Monthly average solar and auxiliary energy, cooling capacity and COP of system (7:00-17:00 operation)

4.2 Biomass feedstock consumption

The biomass gasifier-boiler efficiency is determined by using Eq. (45).

$$\dot{m}_{biomass} = \frac{\dot{Q}_{aux}}{\eta_{boiler} \times LHV_{biomass}} \qquad (45)$$

Assume that the efficiency of the biomass gasifier-boiler is 0.5. And the low heating values (LHV) of some available biomass feedstock such as charcoal, wood chip, rice husk, bagasse and coconut shell are 30, 15.2, 14.3, 14.4 and 17.9 MJ/kg respectively. The biomass consumption rates of above feedstock are shown in Fig. 4. The lowest and highest consumption rates of all biomass are in March and October, respectively. The yearly averaged of consumption rate are 5.80, 11.44, 12.16, 12.07 and 9.71 kg/day, respectively.

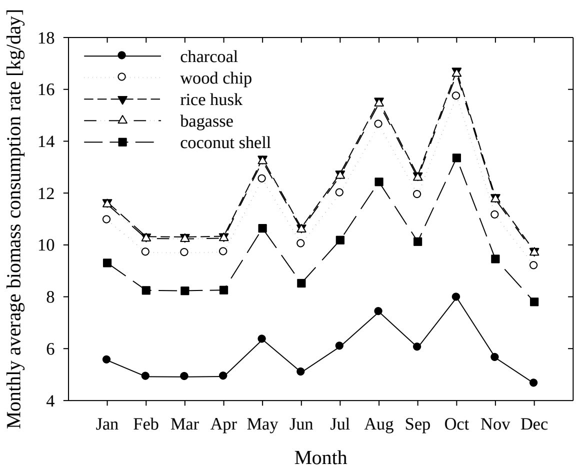

Fig. 4 Monthly average consumption rate of available biomass (7:00-17:00 operation)

For 24-hr operation, the biomass consumption rates of above feedstock are shown in Fig. 5. The lowest and highest consumption rates of all biomass are in February and October, respectively. The yearly averaged of consumption rate are 26.05, 51.41, 54.65, 54.27 and 43.66 kg/day, respectively. In addition, the 24-hr simulation results show that the annual average energy fraction of solar and auxiliary energy is 34 and 66% of total input energy.

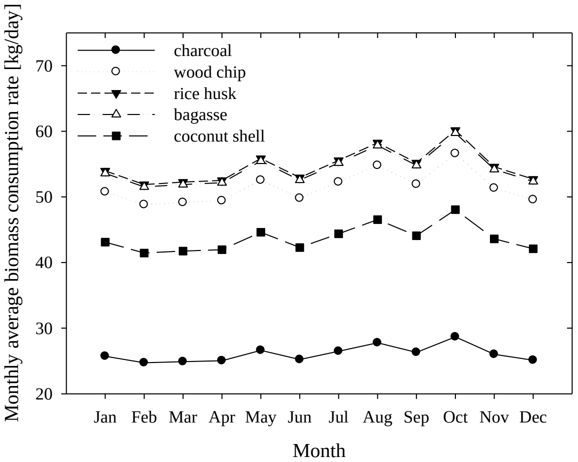

Fig. 5 Monthly average consumption rate of available biomass (24-hr operation)

5. Conclusions and Recommendations

This study presents the simulation study using validated mathematical model aims at assessing the performance of solar-biomass hybrid absorption cooling system and to determine the required auxiliary heat for this system under Thailand conditions. The simulation results show that the lowest and highest requirements of biomass auxiliary energy are in December and October, respectively. The annual averaged energy of solar, biomass and cooling capacity are 201.19, 86.94 and 188.56 MJ/day, respectively. The chiller coefficient of performance varies between 0.64 and 0.67 with the annual averaged value at 0.65. In addition, the yearly averaged consumption rate of some available biomass feedstock; charcoal, wood chip, rice husk, bagasse and coconut shell are 5.80, 11.44, 12.16, 12.07 and 9.71 kg/TR/day, respectively. The proposed system can be a sustainable energy air conditioning system for the future, and can be used with municipal solid waste (MSW) utilization and waste heat recovery. Moreover, it can also be developed to be an isolated tri-generation system. This would thus help in our efforts to address global warming and greenhouse gas emissions

6. Acknowledgement

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

7. References

[1] Koroneos, C., Nanaki, E. andXydis, G. (2009). Solar Air-conditioning Systems Impact on the Built Environment – A Thermodynamic Approach. *45th ISOCARP Congress*.

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

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

[4] Abbasi, T. and Abbasi., S.A. (2010). Biomass energy and the environmental impacts associated with its production and utilization, *Renewable and Sustainable Energy Reviews*.vol. 14, pp. 919–937.

[5] Prasartkaew, B. (2011). A Solar-biomass Hybrid Air Conditioning System, Unpublished AIT doctoral dissertation, Asian Institute of Technology, Thailand.

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