Combined Optimization of Solar Thermal and Biomass Heat Sources with Thermal Storage for the Design of an Ejector Refrigeration System

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***Abstract*— The study focuses on the optimization and design of a dual heat-source-management system utilizing parallel connection of solar thermal and biomass gasifier with a thermal storage facility. The main objective is to supply constant and ample heat energy to the ejector refrigeration system. Since solar thermal energy is intermittent, it is best to store thermal energy in a storage facility to have flexible energy source to operate a heat driven system. Biomass gasifier provides additional heat energy at night or during stormy weather or cloudy sky condition.**

Majority of studies related to this technology connects an auxiliary heat source, directly to the vapor generator. Then, the pair is connected in series with the thermal storage. The drawback for this type of connection is the underutilized heat from the gasifier to the vapor generator, since much of it needed by the latter is just as much of that with the amount of heat given by the solar thermal source. The excess heat will only be exhausted as a waste heat by the refrigeration system.

An average annual solar fraction of 60% was set corresponding to using an evacuated tube solar collector in the calculation. Ambient temperature of 28°C was. The condenser temperature was set at 33°C, which is 5°C above the ambient. The thermal capacity of the gasifier was at 10 kW, while the area of the solar collector was 9.21 m^2 and the volume of the thermal storage was set at 1 m^3. The evaporator temperature was set at 15°C intended for comfort cooling.

Using an algorithm, the solar collector efficiency was calculated at 0.67. The simulation resulted to an average system thermal ratio at around 0.265 and an ejector refrigeration subsystem with COP of 0.30. The higher thermal ratio and COP of the proposed setup with parallel connections of biomass gasifier-boiler and solar thermal collector indicated that this alternative system could be used to attain higher energy efficiency in utilizing the sustainable heat sources. This research also necessitates further investigation of the function and role of the gasifier in using 3D CFD modelling and experimental validation of the system.

***Index Terms*— evacuated tube, gasifier, heat-driven refrigeration system, overall heat loss coefficient, thermal energy storage**

I. Introduction

THE Philippines is a tropical country and generally throughout the year, the weather is humid and hot specially during summer. Thus, space cooling for homes, offices and even commercial and industrial work places are given priority to be safe, conducive for work and comfortable to the occupants. Heat driven refrigeration system is the best logical solution to these cooling requirements. Thus, harnessing heat driven refrigeration technologies that would operate on low grade energy source and at the same time mitigate harmful effects to the environment brought about by the refrigerants used in conventional vapor compression systems, would suffice for the requirement. Ejector refrigeration system is a heat-driven refrigeration cycle that has the advantage of providing cooling effect even at low grade thermal heat sources like solar, low-enthalpy geothermal [1], and waste heat. Furthermore, the ejector refrigeration system has minimal cost compared with other heat driven systems like absorption refrigeration system [2]. The only drawback is the low coefficient of performance or COP.

The Philippines has an abundance of natural resources with the potential to provide energy to heat driven cooling systems. It is located 1614 km above the equator [3] which has ample solar energy available.

Annual solar irradiance in Metro Manila, the capital Metropolis of the country, is on the average of about 4.34 kWh/m^2-day [4]. This could provide ample heat to a medium capacity cooling system, given appropriate considerations to solar collector area, fluid flow rate, and collector's tilt angle.

Also, being an agricultural country, the country has plenty of biomass resources which include but not limited to agricultural crop residues, forestry residues, agro-industrial wastes, municipal solid wastes and aquatic biomass. Majority of which are rice hull, bagasse, coconut shell/husks and coconut coir. Biomass energy plays a vital role in the nation's energy supply [5].

The study investigates the effect of installing the biomass heater as connected to the thermal storage in parallel with the solar collector (Fig. 1), instead of installing the usual auxiliary heater that is normally connected in series with thermal storage and vapor generator [6].

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II. Review Related Literature

A study by Colle et al. [7], employing simulation of a solar-assisted ejector cooling system, investigated the validity of a design methodology. The study focused on the hourly simulation results for the computation of the solar fraction with constant cooling capacity of the ejector cycle during daily periods. The computed solar fraction is compared and showed good agreement with estimates coming from the f-chart method based on the utilization.

Refrigerant plays an important role in ejector refrigeration system as studied by Sun [8]. It traditionally operates using water but has low COP. Eleven refrigerants, including water, halocarbon compounds, cyclic organic compound and azeotrope are chosen and their performances in an ejector refrigeration system are compared.

In a study by Pollerberg et al. [9], the setup has a solar collector as its only heat source. Thus, providing heat to the vapor generator only during day time, then from the generator to the ejector. The study did not consider the continuous operation of the system since solar energy is intermittent by nature. It could have been better if there is a secondary or an auxiliary heat source to compensate for the solar energy gap.

A study of a system having two heat sources, solar thermal and biomass was done by Zhang et al. [10], but the two heat sources operate as one is act as a prerequisite of the other. The heat from the solar collector was used to produce steam needed by the gasifier during the gasification process, together with the other inputs like oxygen and the feedstock from biomass. The resulting syngas will be used as fuel by the internal combustion engine (ICE), supplying mechanical energy to an electric generator. The heat driven refrigeration system in this case is an absorption type that utilizes the exhaust waste heat from ICE for heating of the aqua-ammonia mixture. Even though the system has two heat inputs, the solar energy operates simultaneously with the gasification process. Thus, again dependent on the availability of the solar energy for its continuous operation.

The study by Vidal et al. [11], utilizing dual heat sources, solar and auxiliary for the ejector refrigeration system incorporated a thermal storage. The system description starts with the radiant energy captured by solar collector then directing it to the thermal storage. Then, heat energy flows from the thermal storage to the vapor generator that would generate vaporized fluid and provide the primary flow to the ejector. Between the vapor generator and the ejector there is an auxiliary heat source. This energy back up provides needed heat in cases energy lag from the solar collector.

The system done by Pridasawas and Lundqvist [2] added an improvement. This also has dual heat source, solar and an auxiliary heater for energy back up. Auxiliary heater is run only by electricity with rated capacity. The solar collector is the main heat source of the ejector refrigeration system connected to the thermal storage, and then to the auxiliary heat source, which provides heat during power fluctuation. The auxiliary heater is then connected to the vapor generator that vaporizes refrigerant as primary fluid to the ejector. This together with the previously mentioned studies regarding energy management configurations that have dual heat sources, are almost the same but having different locations of the auxiliary heater. The previous one is located between the generator and the ejector. In the latter, auxiliary heater is installed between the thermal storage tank and the vapor generator.

III. System Description And Modelling

The system is divided into two subsystems: the *dual heat source subsystem* and *ejector refrigeration subsystem*. The dual heat source subsystem composes of the solar collector and the biomass-gasifier-boiler that would provide heat for the refrigeration system. Both heat source components are connected in parallel to the thermal storage tank. The solar thermal collector would be the main energy source and it is backed up by the biomass heat source. The refrigeration system is composed of the vapor generator, ejector, condenser, pump, thermostatic expansion valve and evaporator. The cooling load for the refrigeration system is set as 3.517 kW at evaporator temperature of 15°C.

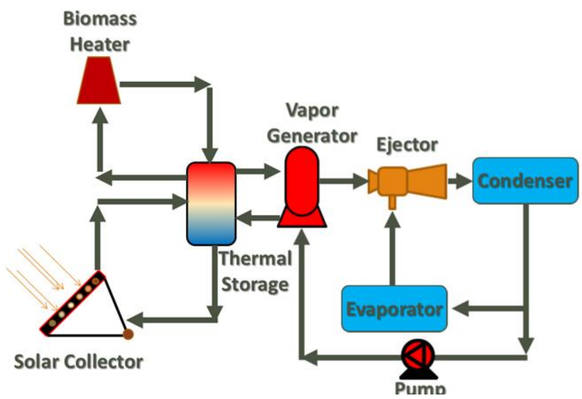

Fig. 1. System diagram for the whole dual source ejector refrigeration system.

A. *The dual heat source subsystem*

As shown in Fig. 1, the dual heat source subsystem will be installed in such a way that the solar collector would be connected in parallel with the biomass-gasifier-boiler making the thermal storage located in between them.

Solar Collectors

Solar collectors that are commonly employed for this purpose are flat plate-single glazed, flat plate-double glazed and evacuated tube because of economy and efficiency.

Previous studies supported the superiority of evacuated tube over flat plate collectors [12]. Since solar collector efficiency is a function of heat loss coefficient, the evacuated tube collector is characterized with reduced internal conduction and convection losses which results to higher thermal efficiency even at low incident angle compared with the flat plate collector.

There are several types of evacuated tubes and the following four types were studied and simulated to find the most efficient type [13]: finned tubes, U-tube welded inside a circular fin, U-tube welded on a copper plate, and U-tube welded inside a rectangular duct. Numerical method was used to find the best shape of the absorber tube to be used as solar collector. Based on the performance testing of a single tube collector, the U-tube welded inside a circular fin is the best among mentioned U-tube collectors, see Fig. 2.

Consequently, the working equation for the U-tube welded inside a circular fin are as follows,

$$q_{u\,left} = W L_u F'[S - U_L(T_{f1} - T_a)] \qquad (1)$$

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$$q_{u\,right} = WL_uF'[S - U_L(T_{f2} - T_a)] \quad (2)$$

where $q_{u,\,left}$ is the heat transfer by conduction at the point of

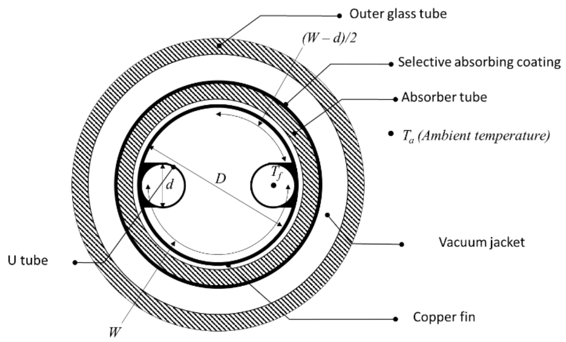

Fig. 2. Cross section of glass evacuated tube with U- tube.

contact between the fin and the tube at the left side of the U-tube where T_{in} and T_{mid} exist. $q_{u,\,right}$ is the heat transfer by conduction at the point of contact between the fin and the tube at the right side of the U-tube where T_{mid} and T_{out} exist. W is the circumferential distance between U-tubes, W= P/2 in meters and P is the overall perimeter of the fin. L_u is the length of U-tube per left or right side. F' is the collector efficiency factor. S is the solar energy absorbed by the selective absorbing coating. U_L is the overall heat loss coefficient, with a range value of 2-4 W/m^2-K, based on experimental and theoretical evaluation [14]. T_{f1} and T_{f2} are the average working fluid temperatures in the two sides of the tubes.

Also,

$$T_{f1} = \frac{(T_{in} + T_{mid})}{2} \quad (3)$$

$$T_{f2} = \frac{(T_{mid} + T_{out})}{2} \quad (4)$$

where, T_{in} and T_{out} are the inlet and outlet temperatures of the working fluid in the U-tube, respectively. And T_{mid} is the temperature of the working fluid at the tube bend. Thus, the temperature of the working fluid can be obtained by the following equations based on (1) and (2).

Consequently,

$$q_{u\,left} = m\,C_{pf}\,(T_{mid} - T_{in}) \quad (5)$$

$$q_{u\,right} = m\,C_{pf}\,(T_{out} - T_{mid}) \quad (6)$$

Then combining (5) and (6) with T_{mid} as common variable for the heat capacity of the whole single evacuated U- tube copper fin

$$q_{u\,left} + q_{u\,right} = q_{u\,total} \quad (7)$$

From (7) that represents the heat output of a single evacuated tube, the calculated capacity is 51.32 W. The system need 172 tubes. With a total solar collector aperture area of 9.21 m^2, the total energy output of the collector is 8.83 kW.

The solar collector efficiency is calculated using the formula,

$$\eta = \frac{(qu{,}Total)\,(N_{tubes})}{Io\,Ap} \quad (8)$$

where η is the solar collector efficiency. N_{tubes} for the number of evacuated tubes needed to supply required heat energy to the system. Io is for the solar radiation intensity. Ap is the diffuse reflection area of solar collector, $Ap = 2DL$. D is the outer diameter of the absorber tube, L is the length of the absorber tube.

The solar fraction (f) is the ratio between heat energy supplied by the solar collector and the total energy requirement of the cooling system

$$f = \frac{Q_u}{Q_{storage}} \quad (9)$$

Given in Table I are the parameters used in the calculation for the optimal operation of the evacuated tube solar collector.

TABLE I
PARAMETERS OF EVACUATED TUBE SOLAR COLLECTOR [15]

Material	Parameters	Quantity	Unit
Absorbing coating	Absorptivity (α)	0.92	
	Emmissivity (ε_p)	0.08	
Outer glass tube	Outer diameter	47	mm
	Thickness	1.2	mm
	Conductivity	1.2	W/m-K
Absorber tube	Outer diameter	37	mm
	Thickness	1.2	mm
	Conductivity	1.2	W/m-K
Copper fin	Thickness	0.6	mm
	Conductivity	307	W/m-K
Air layer	Thickness	1	mm
	Conductivity	0.03	W/m-K
U-tube	Outer diameter	8	mm
	$h_{f,i}$	700	W/m^2-K
Length of the tubular collector		1200	mm
Bond conductance		30	W/m-K

Thermal Energy Storage

The common working fluid used for storing thermal energy is water because of its high specific heat which is 4.187 kJ/kg-K. Water is used both as energy storing and transporting medium. This is done to prevent heat transfer losses between the storing medium and transport medium as compared to using different fluids [16].

One among the major objectives of the study is to investigate for the improvement of the system and its correlation with the physical arrangement of the solar collector and the biomass-gasifier-boiler connected in parallel to the thermal energy storage tank. Simple sensible thermal storage type is adopted in the study due to its simplicity and established principles compared with the other two types which are phase change and chemical reaction type of thermal storage materials. Given that water or any kind of working fluid has a uniform temperature, fully mixed or

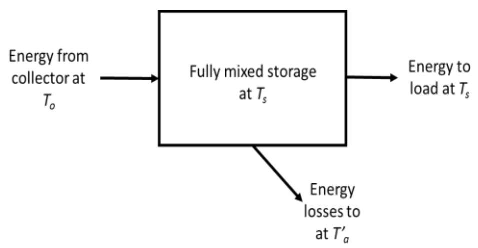

Fig. 3. Unstratified storage of mass m operating at time-dependent temperature T_s in ambient temperature T'_a [16]

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unstratified then operating at a definite temperature difference is given by

$$Q_{storage} = \left(mC_p\right)_s \ \Delta T_s \qquad (10)$$

where Q_s is the total heat capacity for the cycle operating through temperature range ΔT_s, m is the mass of water in the unit and C_p is the specific heat of the working fluid. The operating temperature range where the unit operate is determined and limited at the lower extreme by the requirement of the load or process to be served. The upper limit is determined also by the process, which is the vapor pressure of the working liquid and heat loss in the collector.

The energy balance of the unstratified tank shown in Fig. 3 is

$$\left(mC_p\right)_s \frac{dTs}{dt} = Q_u - \dot{L_s} - (UA)_s\,(T_s - T'_a) \qquad (11)$$

Where Qu is the rate of energy addition from the solar collector to the storage. $\dot{L}s$ is the rate of energy extraction of the load, $(UA)_s$ is the heat loss coefficient. $T'a$ is the ambient temperature in the location of the storage tank, which is different from the ambient air of the collector.

After the calculation using simple Euler integration by rewriting the temperature derivative as $(T^+_s - T_s)/\ \Delta t$, the resulting energy balance, hence follows as

$$\left(mC_p\right)_s \frac{(T'_s - T_s)}{\Delta t} = Q_u - L_s - (UA)_s\,(T_s - T'_a) \qquad (12)$$

Where, Ts' is the temperature of the storage after the time increment Δt usually per hour. Then the following is obtained upon solving for the tank temperature at the end of each time increment

$$T'_s = T_s + \frac{\Delta t}{\left(mC_p\right)_s}\left[Q_u - L_s - (UA)_s\,(T_s - T'_a)\right] \qquad (13)$$

B. The ejector refrigeration subsystem

Basic refrigeration cycle for an ejector technology as shown, in Fig. 4 for the system diagram and Fig. 5 for the T-s diagram, starts at the vapor generator where the saturated liquid refrigerant is supplied with heat for its vaporization at specified pressure and temperature and the amount of which is Q_g, expressed in (14). This vapor at high pressure is described as the primary fluid. The primary fluid will now pass through the ejector as it expands through the converging and diverging nozzle. The secondary fluid, coming from the evaporator, is now entrained due to low pressure created by the expansion. The primary and secondary fluid will mix and flow through the condenser, where heat is being rejected to the surrounding, amount of which is described as Q_c expressed in (15). The fluid mixture will now flow and supply liquid refrigerant to the generator and evaporator. The evaporator will now absorb heat of which also called cooling load or Q_e and as described in (16). Considering pump work in supplying fluid to the generator, the work W is as described in (17). These equations came from the conservation of energy for each component under steady state conditions.

$$Q_g = \dot{m}_p\,(h_2 - h_1) \qquad (14)$$
$$Q_c = \dot{m}_m\,(h_4 - h_5) \qquad (15)$$
$$Q_e = \dot{m}_s\,(h_3 - h_6) \qquad (16)$$

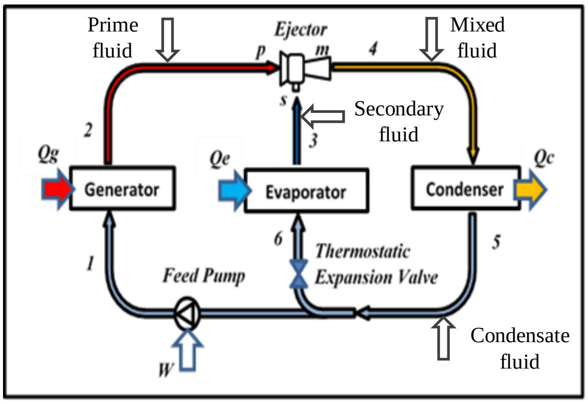

Fig. 4. Ejector refrigeration system schematic diagram

$$W = \dot{m}_p\,(h_1 - h_5) \qquad (17)$$

Relating the equations above, the cycle performance can be measured by the coefficient of performance *(COP)* as described in (18) and entrainment ratio *(ω)*, as described in (19).

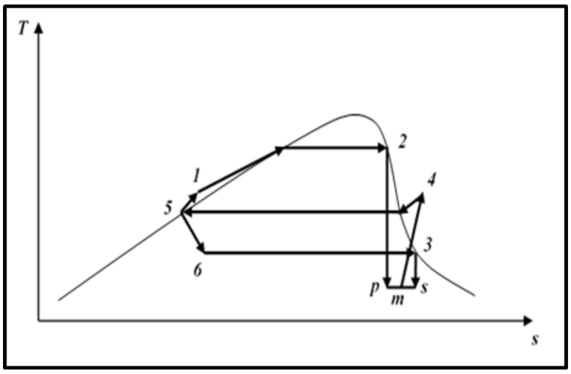

Fig. 5. Ejector refrigeration system T-s diagram.

$$COP = \frac{Q_e}{Q_g} \qquad (18)$$
$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \qquad (19)$$

The performance of the whole system is called the system thermal ratio, (STR), which is equal to

$$STR = \frac{Q_{evaporator}}{A_{sc}\,G + Q_{gasifier}} \qquad (20)$$

IV. Results And Discussion

A. Weather Data

The simulation of the combined heat source to an ejector refrigeration system starts with the weather data in a laboratory facility at the University of the Philippines, Diliman, Quezon City with location at 14.66° latitude and 121.07° longitude. The weather data provides the average horizontal solar radiation and ambient temperature

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throughout the year, illustrated in Table II. From this data, the simulation software calculates the solar energy that is available for the system in a daily and monthly basis.

The hottest months are from April to May having 29.1°C to 29.5°C and the highest radiation which are 5.65 kWh/m^2-day to 5.17 kWh/m^2-day, respectively

As a reference, we can use to calculate the kW/m^2 by using the average peak-sun hours in Quezon City with a value of 5.5 hours [17].

TABLE II
AVERAGE ANNUAL RADIATION IN THE PHILIPPINES [4]

Month	Ambient Temperature	Daily Solar Radiation Horizontal, G	Relative Humidity
	°C	kWh/sqm-d	%
January	25.4	3.73	77.20
February	26.4	4.40	74.80
March	27.5	5.17	71.00
April	29.1	5.65	69.40
May	29.5	5.17	72.80
June	28.5	4.70	80.20
July	27.4	4.23	84.40
August	27.3	3.79	84.90
September	27.2	4.05	85.40
October	27.2	3.83	83.40
November	26.7	3.79	81.70
December	25.8	3.55	80.80
Average	27.33	4.34	78.83

B. *Solar Collector Area*

The month of April has the highest intensity of solar energy of 5.65 kWh/m^2-day and ambient temperature of 29.1°C, as shown in Table II. Mentioned month qualifies for the reference period for the proper design of the area for the evacuated tube solar collector. Considering flow rate of 108 kg/h, the collector area of 9.21 m^2 has provided a satisfactory outcome. The outlet temperature range resulting from the collector is 110°C to 120°C. The design temperature difference from the collector to the thermal storage is 10°C and another 10°C from the storage to the vapor generator. Consequently, this range is suitable for a cooling cycle that is utilizing R290 or propane refrigerant, since the critical temperature of R290 is about 96.7°C.

C. *Solar Collector Tilt Angle*

The optimal tilt angle for the solar collector to maximize the available solar radiance is shown in degrees, Table III. The solar collector is installed facing south about the vertical for the month of November with the value of 59° [18].

TABLE III
OPTIMUM TILT OF SOLAR PANELS IN THE PHILIPPINES [18]

Optimum Tilt Solar Panels by Month in the Philippines					
Jan	Feb	Mar	Apr	May	Jun
59°	67°	75°	83°	91°	98°
Jul	Aug	Sep	Oct	Nov	Dec
91°	83°	75°	67°	59°	52°
angle from the vertical					

D. *Solar Fraction*

Fig. 6 shows the portion of participation of solar energy in providing heat energy side by side and in parallel with the gasifier for the ejector cooling system. With an average solar fraction of 0.6 which means that 60% of energy is supplied by the solar collector and 40% by the auxiliary heater. This is more environmentally safe because solar heat source is much safer to use compared with the gasifier since the process of latter still involves burning of materials.

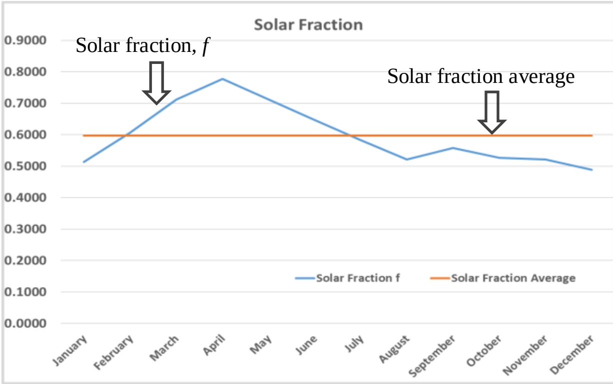

Fig. 6. Monthly solar fraction of the solar collector

E. *Thermal Storage*

In order to achieve the desired thermal storage outlet temperature of 110°C for the propane refrigerant, with vaporization temperature approximately at 100°C and a flow rate of 108 kg/h, through numerical simulation, the required storage volume is 1 m^3 for an 8-hour operation. Volume smaller than this would provide higher storage temperature output but the drawback is shorter discharging time. Larger volume can provide longer discharging time but at a storage temperature outlet lower than the system requirement.

F. *Ejector Refrigeration System*

It has been shown that the 9.21 m^2 of evacuated tube solar collector coupled with a 1 m^3 of thermal storage, with a 108 kg/h flow rate, are the prerequisites for an ample heat to run the ejector refrigeration system with a cooling load of 3.517 kW and would operate at 8 hours per day. From these data, an average annual STR of 0.265 is achieved. The COP of the refrigeration system is set to 0.30.

V. Conclusions and Recommendations for Future Work

The given data show that the installation of the gasifier-boiler heater provided appreciable thermal energy utilization to the system by allocating its energy capacity with the solar collector and engaging their energy capacities to the thermal storage tank. The thermal storage supplies ample energy to the ejector refrigeration system to absorb the 3.517-kW cooling load.

Since the Philippines receives a good solar radiance, the 9.21 m^2 apparent area for the solar collector is enough to supply energy side by side with the biomass gasifier. And the applicable angle of tilt for the solar collector is 59° from the vertical every November of the year and solar collector efficiency of 0.66 is obtained. The obtained energy share of the collector at an annual average of 60 percent and the share of the gasifier is at 40 percent.

The thermal storage is adequate for a working volume of 1 m^3 and a flow rate of 108 kg/h, this provides adequate heat to the vapor generator of the ejector refrigeration system. Lower than this rate will increase the temperature of the vapor generator which is not good to the refrigerant R290 that would reach its super-heated state. Higher than this rate, would decrease the fluid supply temperature to the vapor generator considerably.

These results are just preliminary and have shown that the heat energy produced by the parallel connection of solar

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collector and auxiliary heater were all utilized by the system through the thermal storage tank. Compared with the other connection, the one with auxiliary heater connected directly to the vapor generator some of the energy were diverted out of the system and given off as waste heat.

In the existing algorithm, the solar collector efficiency was calculated at 0.67. The simulation gave an average system thermal ratio of 0.265 and COP of an ejector refrigeration subsystem at about 0.30 using propane (R290), as refrigerant. The higher thermal ratio and COP of the proposed setup utilizing parallel connection of biomass gasifier-boiler and solar thermal collector indicated that this system provided improved energy efficiency in utilizing the sustainable heat sources for comfort cooling. Again, this research necessitates further investigation on the function and role of the gasifier in using 3D CFD modelling and system experimental validation.

NOMENCLATURE

A_p	apparent solar collector area
COP	coefficient of performance
f	solar fraction
F_R	collector heat removal factor
G	solar flux density, solar energy irradiation, W/sqm
h1	enthalpy from the pump to the vapor generator
h2	enthalpy from the vapor generator
h3	enthalpy from the evaporator
h4	enthalpy from the ejector
h5	enthalpy from the condenser
h6	enthalpy from the thermostatic expansion valve
η_{sc}	solar collector efficiency
Qc	heat rejected by the condenser
Qe	heat absorbed by the evaporator
Qg	heat absorbed0 by the vapor generator
Qs	horizontal heat radiation from the sun
Qu	usable solar energy from heat collector
STR	system thermal ratio
Ta	ambient temperature, deg. C
Ti	working fluid entering solar collector, deg. C
To	working fluid leaving the solar collector, deg. C
U_L	heat loss coefficient, W/m/K
ω	entrainment ratio
$(\tau\alpha)_e$	effective transmittance-absorbance product

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