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THERMAL PERFORMANCE EVALUATION OF RENEWABLE HYBRID ENERGY SOURCES FOR HEATING GREENHOUSES

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ABSTRACT

The interest in alternative or renewable heat energy sources for greenhouses heating is currently high, due to the large burden of heating and the relatively high price of fossil fuels. The objective of this study is to analyse the thermal performance of hybrid system; solar energy and biomass greenhouse heating system (SBGHS). This study experimentally investigates the total burden of heating required for commercial greenhouse (1010.4 m^3) heating by solar and biomass heat energy under the climatic conditions of eastern region of costal Delta, Egypt (latitude angle of 31º 02' 41'' N, longitude angle of 31º 21' 55'' E, and mean altitude above sea level of 6 m) during winter season of 2012-2013. The thermal performance analysis was experimentally determined, by measuring the temperature increase at various water inlet temperatures and intensity of solar radiation, under clear sky conditions. A complete solar heating system (six solar collectors and storage tank) was utilised for heating 1500 litres of solution (water and antifreeze). The daily average overall thermal efficiencies of the solar heating system and the storage system during the heating period, respectively, were 83.19% and 95.51%. Over 147 days heating season the solar heating system collected 9 222 kWh of which 8 830 kWh (31.788 GJ) of solar power was stored in the storage tank. It provided 27.47% of the total power required by the greenhouse. The biomass heating system added 145.136 kWh (522.490 MJ) of heat energy into the greenhouse which provided 66.37% of total power required for greenhouse heating. The hybrid heating system provided 205.206 kWh (93.84%) of the daily total heat energy required. The economics of such a system remains marginal at present power prices in Egypt, although changes in power costs may drastically alter the situation.

INTRODUCTION

Nowadays, the impact of the fuel price crisis coupled with the awareness of global problem has brought changes in the structure of energy usage all over the world. Greater importance has now been given to research, development, and operation of clean (renewable) energy, for a greater energy security. Renewable energy is accepted as a key source for the future, not only for Egypt but also for the whole-world. Renewable energy technologies produce marketable energy by converting natural phenomena into useful forms of energy. These technologies use the sun's energy and its direct and indirect effects on the earth (solar radiation, wind, falling water, and various plants residual, i.e. biomass), gravitational forces (tides), and the heat of the earth's core (geothermal) as the resources from which energy is produced. Egypt has a considerably high level of renewable energy sources particularly solar energy that can be contributed a large part of the total heat energy required in the country. The benefits arising from the installation and operation of renewable energy system can be distinguished into three categories; energy saving, generation of job opportunities, and the decrease of environmental pollution.

Carbon dioxide is one of the commonly accepted techniques to enhance photosynthesis process resulting in improved yields and income. Operators typically increase levels from 800 to 1000 ppm from an atmospheric level of 380 ppm. Enrichment is commonly practiced with pure carbon dioxide in bulk or from combustion of hydrocarbon fuel (natural gas or propane). Usually, these fuels are employed in dedicated burners to provide carbon dioxide (CO_2) while a separate heating system provides most of the heat to the greenhouse. Carbon dioxide enrichment from the exhaust of a natural gas or propane heating system has proven to be feasible, but using renewable energy could have further benefits. In terms of enrichment applications, combustion of dry and clean wood biomass can produce two times more useful CO_2 than natural gas for the same energy unit (Chalabi, *et al.,* 2002 ; Jaffrin, *et al.,* 2003).

The efficiency of biomass conversion into energy depends on the chosen thermo-chemical reaction as well as the system's design. Internal modifications are important for reducing fuel consumption, pollution emissions and the costs of external modification. They may even alleviate the necessity of implementing an external modification (McKendry, 2002). Among the main thermo-chemical processes of biomass, combustion is the most well known and widely applied reaction for heating the greenhouses. Efficiency of combustion heating systems, by implementation of proper residence time, temperature and turbulence, varies depending on their design, manufacturing and operation. Various combustion technologies exist such as fixed bed, fluidizer bed and pulverized bed combustion. The system may be built for the flame to be directed counter-current, co-current or cross-flow to the fuel. Biomass moisture content is recommended to be lower than 50% since it impacts the system's efficiency (Raymar, 2006).

Biomass heating systems differ from conventional wood-burning stoves and fireplaces in that they typically control the mix of air and fuel in order to maximize efficiency and minimize emission. They also include a heat distribution system to transport heat from the site of combustion (biomass burner) to the heat load (greenhouse). Many biomass heating systems incorporated a sophisticated automatic fuel handling system. Biomass heating systems consist of a number of elements, including a heating plant, which typically includes an automated biomass combustion system and a peak load and back-up heating system, a heat distribution system, and a biomass fuel supply operation. The system can also include a waste heat recovery system from a process or electricity generation unit (NRCan, 2002 ; NRCan, 2005 ; Gousgouriotis, *et al.,* 2007). The biomass combustion system, biomass fuel or feedstock moves through the system in a number of stages, many of which are used.

Using a wood biomass boiler could reduce over 3000 ton of carbon dioxide (CO_2) equivalents of greenhouse gases annually. Wood biomass boilers generate a higher volume of particulate matters (PM) and ash emissions than natural gas. An installed electrostatic precipitator (ESP) can efficiently reduce the particulate matters (PM) emission from the wood biomass combustion flue gas resulting in the similar level as natural gas. The positive net present value (NPV) results indicated that the installation of electrostatic precipitator (ESP) did not affect the feasibility of a wood biomass boiler at a discount rate of 10% (Chau, *et al.,* 2009).

Biomass is now considered to have a key role in building a foundation for energy generation, because it can be produced and implemented effectively. In addition, biomass can be used as an alternative source of energy in the production of hydrogen, which is an efficient and clean fuel for highly effective combustion with environmental friendliness (Paengjuntuek and Mungkalasiri, 2013). Biomass is available in large volume, which is mostly derived from plants; it is a key source of renewable energy for the world. Biomass is also classified as an alternative energy to be used instead of the energy source from fossil fuels, which are limited and may become depleted (Kirtay, 2011). Among the different technologies proposed for biomass conversion into energy, a gasification process is the most promising way as it provides a gaseous product with high hydrogen content (synthesis gas) (Castello and Fiori, 2011). Improving the efficiency of biomass conversion is an important issue to be considered.

In Egypt, there is no readily available information about the application of heat energy generated from the biomass burner assisted solar water heating system. Therefore, the objective of this study was to demonstrate some renewable energy sources (biomass energy and solar energy) which can be efficiently used for heating a commercial greenhouse during the typical winter conditions in eastern region of costal Delta of Egypt. For this purpose, a hybrid system of solar water heating and biomass have been designed and integrated into the greenhouse.

MATERIALS AND METHODS

Materials

Description of the investigated hybrid system

This study employed a biomass burning unit uses combustion technology with different types of field residues. This unit was designed and constructed in the workshop of Faculty of Agriculture, Mansoura University. Vertical biomass combustion equipment was used, at which the biomass solid fuel took place on horizontal stationary steel grate. The biomass fuel is semi-continuous supplied in the batch. The furnace design has been used as a stationary burning system with front-feed solid fuel burner and consists of two sections as shown in Fig. (1). The top section of biomass burner is cylindrical in shape, and made of 3 mm thick double layer of steel tightly fixed together along the frame elements of the walls with 5 cm between. The gape between is fulfilled by an insulation material (Thermo-clay) in order to minimize the heat energy loss from the top base of biomass burner. The gross dimensions of top base of biomass burner are 80 cm diameter, and 120 cm high. A 25.4 mm stainless steel pipe coil was used as a heat exchanger which vertically located in central line of the top section.

The bottom section of the biomass burner is trapezoidal in shape and made from double layer of iron sheets (inner layer) and stainless steel sheets (outer layer) with 5 cm space between which filled completely by an insulation material (Thermo-clay) in order to minimize the heat energy loss from the bottom base of biomass burner. The gross dimensions of the inner layer of biomass burner are 70 cm and 110 cm top and bottom base of the trapezoid, respectively, 200 cm long, and 100 cm high, with a net volume of 1.8 x 10^6 cm^3 (1.8 m^3). Another heat exchanger (similar to the first one) which horizontally located in central line of the bottom section just above the grate This heat exchanger was connected to the first heat exchanger (in the top section) in order to increase the heat transfer rate. The storage tank located inside the greenhouse (1500 litres) was connected to the two heat exchangers by two junctions in order to provide heat energy in the greenhouse. Hot water (heated by solar energy) at the end of daylight was circulated using water pump for heating the water in the storage tank and returned to the biomass burner as low-temperature water (lower than the heat energy inside the biomass burner). This system was used to heat the solution (water and antifreeze) when the solar radiation was insufficient to raise the temperature to 85ºC. To provide and maintain an adequate amount of oxygen for igniting the biomass materials, the bottom section was connected to air blower (2 hp ≈ 1.5 kW).

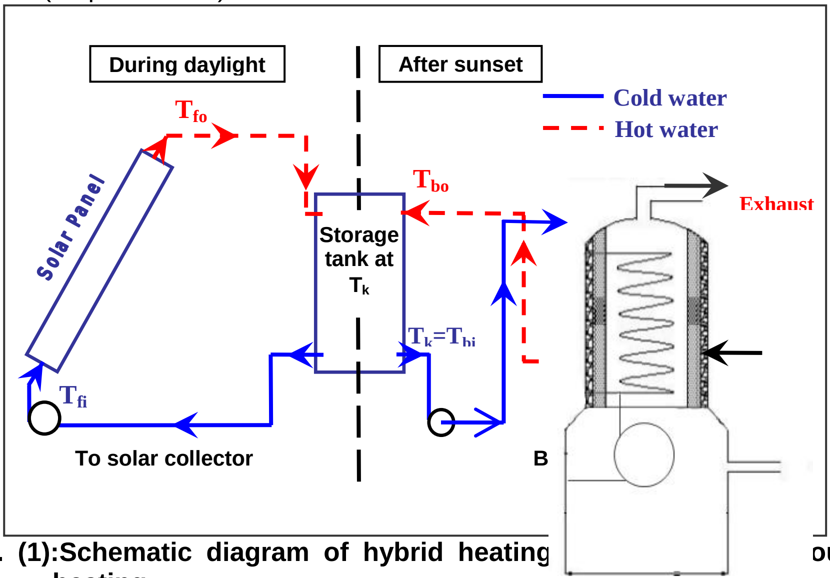

Fig. (1):Schematic diagram of hybrid heating ouse heating.

Biomass field residues are combusted in a firebox inside the biomass burner to provide heat energy approximately equal to the heating value (calorific value) of field residues. Three functional parts of the heat energy generated from the combustion of the field residues are utilized. The first functional part of the heat energy generated is absorbed by the heat exchanger coil inside the biomass unit and transfers into the water passes through the coil. The second part of the heat energy generated is absorbed by another heat exchanger coil also located inside the biomass unit and transfers into the air passes through the coil which is continuously heated and expelled into the greenhouse. The last functional part of the heat energy generated from the combustion of the biomass materials is initially contained in the exhaust, which ranges between 9.5 to 12.5 kWh (McKendry, 2002).

To utilise this heat energy contained in the exhaust, a thin-walled tube is connected to the exhaust flue and passes horizontally through the center line of the greenhouse (longitudinal direction) at which the warm exhaust transfers heat energy to the cooler metal of the tube. This thin-walled tube is sufficiently tall so that shifting winds cannot sweep emitted gases into the greenhouse, where they can cause plant injury. All joints in the thin-walled tube are taped with fir-resistant tape to hold prevent leakage of fumes into the greenhouse atmosphere. Much of the heat energy is removed from the exhaust by the time it reaches the stack through which it leaves the greenhouse into the treatment unit of the flue gas. The cool air surrounding the hot metal tube is continuously warmed and expels it downward the floor resulting in rises the air temperature of the greenhouse.

The flue gas emission from the combustion of biomass materials inside the firebox of biomass burner is including different gases according to the type combustions (complete or incomplete combustion). When the field residues are of high purity and are thoroughly combusted, only carbon dioxide and water vapour are produced. Products of incomplete combustion, including ethylene, sulfur, and carbon monoxide gases are injurious to plants and causes environmental pollution. Therefore, the exhaust smoke after passing the greenhouse through the thin-walled tube is passed in the treatment unit. The treatment unit consists of plastic tank (60 litres) has a liquid of mixing water and lime, and active carbon in order to absorb the majority of different gases before it leaves into the atmosphere. The exhaust gases inside the biomass burner and just leaving the treatment unit are continuously measured using gas analyzer device.

Solar water heating system

The solar water heating system consists of six individual solar collector panels each having a gross dimensions of 200 cm long, 100 cm wide, and 10 cm thick with net surface area of 2.0 m^2, and constructed from copper with a selectively absorbing surface coating. The operating fluid (mixing of pure water and antifreeze) flowed through parallel waterways built into each panel. The 6 solar panels are arranged in two banks with three panels in series array in each bank. Thus all the operating fluid passes through three panels at a flow rate which is sufficient to give reasonable efficient heat transfer, while still enabling the water to reach 65-70ºC under ideal conditions. The solar panel will not operate at its peak potential unless it is tilted and orientated from the horizontal plane in such a way that it will minimize the angle of incidence and maximize the transmittance of glass cover and absorptance of the absorber plate. Consequently, it will receive and absorb the maximum amount of solar energy flux incident. Therefore, each six solar panels are mounted on a movable frame outside the greenhouse so that to track the sun's rays from sunrise to sunset. They used a quadrant and clamp as a tilt angle controller as clarified in Fig. (2).

The operating fluid has pumped to pass through the solar collector panels. After passing through the solar collector panels it is stored in a

1500 litres insulated storage tank situated inside the greenhouse. The flow rate of operating fluid through the solar water heating system (24 litres/min) is tested and adjusted each five days using the control valve and a measuring cylinder with a stopwatch. The storage tank is located inside the greenhouse in order to reduce the heat energy loss. The storage tank is connected to a direct burner using field residuals (rice strew, cotton stalks, maize stalks, and wood of trees) to utilize the heating value of residuals burning for heating water when the solar radiation is insufficient to raise the water tank temperature to 85ºC.

Fig. (2): Solar collector panel array, with a total surface area of 12.0 m^2 and mounted on a movable frame.

Measurements and Data Acquisition Unit

Instrumentation

Meteorological station (Vantage Pro 2, Davis, USA) located just above the greenhouse 2 is used to measure different macroclimate variables such as, the solar radiation flux incident on a horizontal surface (pyranometer), dry-bulb, wet-bulb, and dew-point air temperatures (ventilated thermistor), wind speed and its direction (cup anemometer and wind vane), air relative humidity (hygrometer) and rainfall amounts (rain collector). The amount of heat energy added to the water in the storage tank which situated inside the greenhouse from the solar heating system (during daylight) and the biomass burning system (prior to sunset), a 12 channel data-logger (Digi-sense scanning thermometer type), was also used for taking and storing reading from different sensors (thermocouple type K) mounted at twelve different locations. A solarimeter integrated to a computer based data-logger, mounted on a surface parallel to the plane of the solar collectors, was functioned to measure the global solar radiation flux incident on the tilted surface of collector.

The following data were regularly measured and recorded during the experimental work with a time interval of 5 min.:

(a) Water-antifreeze solution temperatures entering and leaving the solar heating system (flat plate solar collectors) by copper constantan thermocouples mounted on the water-antifreeze solution inlet and outlet lines.

(b) Water-antifreeze solution temperatures entering and leaving the biomass burner heat exchanger by copper constantan thermocouples mounted on the water-antifreeze solution inlet and outlet lines.

(c) Air temperature entering and leaving the air heat exchanger coil mounted on the top section of biomass burner by copper constantan thermocouples mounted on the inlet and outlet lines.

(d) Flue gas at the beginning and end of the thin-walled tube and peripheral temperature of tube located inside the greenhouse by copper constantan thermocouples mounted on the inlet and outlet lines.

(e) Water-antifreeze solution in the storage tank by copper constantan thermocouple mounted on the centre point of tank.

(f) Solar radiation flux incident on the tilted surface of solar heating system using solarimeter device.

Heat losses from a greenhouse

The heating system should be properly sized to meet the needs of the greenhouse under extreme weather conditions. The total heat losses from inside to outside of the greenhouse can be computed from the following equation (ANSI/ASAE, 2003 ; ASHRAE, 2005 ; Nelson, 2006 ; Esen and Yuksel, 2013):

$$Q_{loss} = Q_{CL} + Q_{inf} \quad (1)$$

Where, Q_{CL}, is the combination heat losses by conduction, convection, and radiation through the concrete blocks and the glazing materials of the greenhouse. It can be estimated by the following equation:

$$Q_{CL} = U\ A_C\ (T_{ai} - T_{ao}) \quad (2)$$

Where, U, is the overall heat transfer coefficient in W m^{-2} ºC^{-1}, A_C, is the total surface area of covering material in m^2, Tai, is the indoor air temperature, and, T_{ao}, is the outdoor air temperature in ºC. The heat loss due to infiltration of cold into the greenhouse is computed in terms of the mass flow rate of cold air (m_{ac}) in kg s^{-1}, latent heat of evaporation of water (h_{fg}) in kJ kg^{-1}, and moisture content difference between indoor (W_{ai}) and outdoor (W_{ao}) in $kg_w\ kg_a^{-1}$ as follows:

$$Q_{inf} = m_{ac}\ h_{fg}\ (W_{ai} - W_{ao}) \quad (3)$$

The indoor air temperature (T_{ai}) of 16ºC generally meets the needs of most protected cropping. Using the above three equations the total heat losses (or burden of heating) of the greenhouse was found to be 49.792 kWh.

Useful solar energy

The instantaneous useful heat energy gained by solar heating system (Q_U) is computed by the following equation (Duffie and Beckman, 2007):

$$Q_u = F_R\ A_C\ [R\ (\tau\alpha) - U_O\ (T_{fi} - T_{ao})] = m\ C_P\ (T_{fo} - T_{fi}) \quad (4)$$

Where, F_R, is the heat removal factor, A_C, is the solar collectors surface area in m^2, R, is the solar radiation flux incident on the tilted surface of collectors in W m^{-2}, τα, is the optical efficiency, U_O, is the overall heat transfer coefficient in W m^{-2} ºC, T_{fi}, inlet temperature of the operating fluid in ºC, T_{ao}, is the outdoor air temperature in ºC, m, is the mass flow rate of operating fluid in kg s^{-1}, C_P, is the specific heat of operating fluid in J kg^{-1} ºC^{-1}, and, T_{fo}, is the outlet temperature of the operating fluid in ºC.

The instantaneous overall thermal efficiency of the solar heating system is calculated as follows **(Duffie and Beckman, 2006)**:

$$\eta_o = \frac{m\, C_P\, (T_{fo} - T_{fi})}{R\, A_C} \qquad \textbf{(5)}$$

Biomass system

A mathematical model describes the system of a biomass burner unit is set up with an active condensation unit located inside the greenhouse. Furthermore, formula for the energy balance on the biomass burner unit are presented and discussed as follows:

$$NHV = H_{wa} + H_{aa} + H_{con} - H_{loss} \qquad \textbf{(6)}$$

Where, NHV, is the net heating value in kWh, H_{wa}, is the heat energy absorbed by the operating solution (pure water and antifreeze) passes through the heat exchanger in kWh, H_{aa}, is the heat energy absorbed by the air passes through the heat exchanger in kWh, H_{con}, is the sum of sensible heat and latent heat of condensing water in the heat exchanger located inside the greenhouse in kWh, and, H_{loss}, is the sum of heat energy loss from flue gas and outer surface of the biomass burner unit in kWh. The NHV can be computed in terms of the lower heating value (LHV) by the following equation (Khor, *et al.,* 2007 ; Musil-Schlaeffer, *et al.,* 2011):

$$NHV = 3.6\ LHV, \quad kWh/kg \qquad \textbf{(7)}$$

$$LHV = GHV - 2.453\,(9\,H_2 + MC), \quad MJ/kg \text{ of residue} \qquad \textbf{(8)}$$

Where, GHV, is the gross heating value (higher heating value), H_2, is the percentage contain of Hydrogen element in the field residues, and, MC, is the moisture content in the field residues. The GHV can be calculated using the following formula:

$$GHV=130.225(H_2)+35.160(C)+10.4653(S)+6.28(N_2)-11.09(O_2)\ MJ/kg \text{ of residue} \qquad \textbf{(9)}$$

Where, C, is the percentage of Organic carbon, S, is the percentage of Sulfur, N_2, is the percentage of Nitrogen, and, O_2, is the percentage of Oxygen. Four field residues (biomass materials) are collected from different fields (rice straw, cotton stalks, corn stalks, and wood of trees). Samples of these four field residues were chemically analysed.The chemical analysis of the four samples is summarised and listed in **Table (1).** These percentages of different elements contain in the field residues are functioned to determine the GHV and NTH using the equations (8) and (9). The gross and net heating values (higher and lower heating values, HHV and LHV) were computed using the above two equations according to the percentage of different chemical elements contained in the field residues and listed in **Table (2)**.

The heat energy absorbed by the operating fluid passes through the heat exchanger coil inside the burner (H_{wa}) can be computed in terms of the mass flow rate of operating fluid (m_w) in kg s^{-1}, specific heat of fluid (C_P) in kJ $kg^{-1}\,{}^oC^{-1}$, and temperature difference between outlet (T_{wo}) and inlet (T_{wi}) of operating fluid in ºC as follows:

$$H_{wa} = m_w\ C_P\ (T_{wo} - T_{wi}) \qquad \textbf{(10)}$$

Table(1):Chemical analysis for some field residues uses in this research work as a source of renewable energy

Field residue	H_2 %	C %	S %	N_2 %	O_2 %	MC %
Rice straw	3.93	34.60	0.16	0.93	35.38	4.50
Cotton stalks	5.56	43.65	0.01	0.16	43.31	10.70
Corn stalks	5.07	39.47	0.02	1.20	39.14	8.85
Wood	5.98	49.00	0.01	0.05	44.75	12.65

Table (2): Gross heating value (Higher Heating Value, HHV) and net heating value (lower heating value, LHV) for four different field residues in MJ/kg of residues

Heating value, MJ/kg	Biomass materials (solid fuel)			
	Rice straw	Cotton stalks	Corn stalks	Wood of trees
Gross (HHV)	13.435	17.796	16.217	20.057
Net (LHV)	12.464	16.317	14.890	18.439

The heat energy absorbed by the cold air passes through the heat exchanger coil which situated inside the burner unit (H_{aa}) can be calculated in terms of the mass flow rate of air (m_a) in kg s^{-1}, specific heat of air (C_{Pa}) in kJ kg^{-1} $^oC^{-1}$, and temperature difference between outlet (T_{ha}) and inlet (T_{ca}) of air in ^oC as follows:

$$H_{aa} = m_a \, C_{Pa} \, (T_{ha} - T_{ca}) \quad (11)$$

Depending on the temperature of flue gas emission from the biomass burner, the sensible heat energy added to the indoor air of the greenhouse through the peripheral of the thin-walled tube. It can be computed in terms of the mass flow rate of flue gas (m_{fg}) in kg s^{-1}, specific heat of flue gas (C_{Pfg}) in kJ kg^{-1} $^oC^{-1}$, and temperature difference between the inlet (T_{fgi}), and outlet flue gas (T_{fgo}) in ^oC. The latent heat due to condensation of water in the moist flue gas, can be calculated by the latent heat of condensation of water (h_{fg}) in kJ kg^{-1}, and change in moisture content of flue gas (W_{fgi} - W_{fgo}) in kg_w kg_a^{-1} as follows:

$$H_{con} = m_{fg} \, C_{Pfg} \, (T_{fgi} - T_{fgo}) + m_{fg} \, h_{fg} \, (W_{fgi} - W_{fgo}) \quad (12)$$

The heat energy losses from the biomass burner (H_{loss}) are the sum of heat energy loss from the unit due to radiation and convection loss which dependent upon the actual output and the air cooled wall factor **(DOE, 2004)** and the heat energy loss during the quench of flue gas in the treatment unit. Heat losses could also be due incomplete combustion, high moisture content in the field residues (biomass), high ash content in the field residues, and the inefficient burner design. Therefore, the heat losses from the biomass burner can be estimated in terms of the overall heat transfer coefficient (U_{Ob}) in W m^{-2} $^oC^{-1}$, surface area of biomass burner (A_b) in m^2, temperature difference between inside (T_{hai}) and outside (T_{ao}) of air in ^oC, mass flow rate of flue gas, specific heat of flue gas, and the temperature of the outlet flue gas (T_{fgo}) in ^oC as follows:

$$H_{loss} = U_{Ob} \, A_b \, (T_{hai} - T_{ao}) + m_{fg} \, C_{Pfg} \, T_{fgo} \quad (13)$$

Thermal efficiency of biomass burner

The biomass burner thermal efficiency is computed as the ratio of heat energy output (heat energy absorbed by the operating fluid and air, and heat energy gained by thin-walled tube from the flue gas) to the heat energy input (net heating value of biomass). The input-output method is used to determine the burner efficiency (Ganapathy, 1997 ; Barroso, *et al.,* 2003 ; Covarrubias and Romero, 2007):

$$\text{Burner efficiency} = \frac{\text{Heat energy gained (outlet)}}{\text{Net heating value of biomass}} \qquad (14)$$

The model has implemented as a stand-alone program running on IBM compatible microcomputer. The developed mathematical model has solved with the help of computer program based on MATLAB (MATrix LABoratory). The program requires two input files: one contains the simulation parameters and the other contains the input data.

This experimental research work is designed to heat the large-scale greenhouse (1010.4 m^3). The solar energy and biomass greenhouse heating system (SBGHS) is an important economic alternative over the other conventional heating methods such as fuel-oil, LPG, and electric in Egypt. Also, the SBGHS can considerably reduce primary heat energy use for greenhouse heating.

RESULTS AND DISCUSSION

Owing to large burden of heating is essential requirement for greenhouse heating and relatively high prices of fossil fuels, alternative heat energy sources for greenhouse heating has been gained utmost interest. For heating and cooling of the greenhouse, it is of primary importance to choose a correct and alternative source for improving its efficacy in sustainable production and productivity of crops. Some of the important alternative sources of heat energy are the solar collectors and thermal energy storage systems (STES), and the thermal energy applications in space heating and hot water of the modern biomass combustion system (BCS).

The obtained results from the experimental work over the heating period from 6^{th} of December 2012 to 30^{th} of April 2013 were evaluated to determine the thermal performance characteristics of the hybrid system (SBGHS). The two solar heating systems (each one having six solar collectors, storage tank, heat distributing system, and control board) have been operating satisfactorily for approximately five months without malfunction. Water temperatures have been monitored for five months beginning in December 2012, and the monthly average solar energy contribution is demonstrated in Fig. (3). During the heating period, there were 979 hours of bright sunshine of which 857 hours (87.54%) were recorded and used in the thermal performance analysis and applications, slightly lower than average due to clouds. Although on day to day figures the correlation between sunshine hours and solar energy collected was lower, nevertheless the agreement was good on a monthly average basis (Fig. 3). The discrepancies between months arise due to number of bright sunshine hours, solar altitude angles, water temperature in the storage tank at the beginning of each day, and number of operating hours.

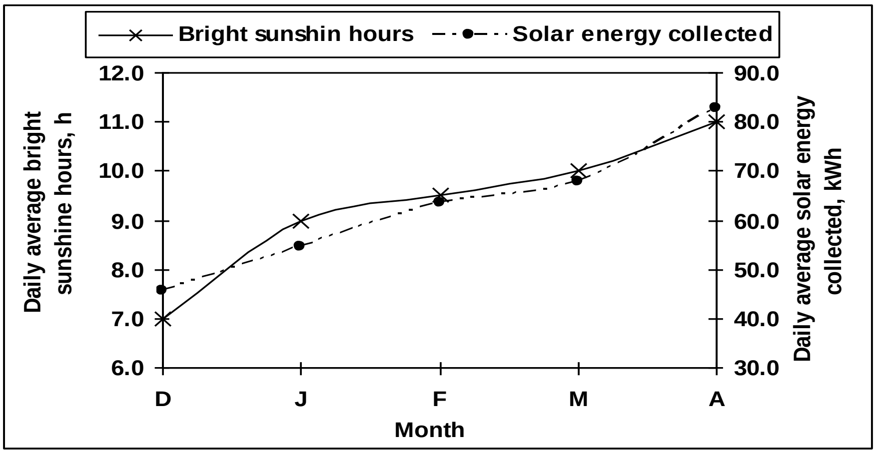

Fig. (3): Daily average solar energy collected by solar collectors and daily average sunshine hours during the experimental period.

The actual solar radiation recorded on the tilted surface of solar heating system was always higher than that on the horizontal surface. For the duration of December, January, February, March, and April the daily averages solar radiation measured from sunrise to sunset on the horizontal surface, respectively, was 2.332, 2.830, 3.711, 4.636, and 5.728 kWh/m^2 day. Whereas, the actual solar radiation measured on the tilted surface of solar collectors at that period was 4.696, 5.569, 6.646, 6.668 and 8.024 kWh/m^2 day, respectively, consequently, the solar collectors orientated and tilted from the horizontal plan increased the actual received solar radiation during that period by 201.4%, 196.8%, 171.0%, 143.8%, and 140.1%,, respectively.

The thermal performance analysis of the solar collectors is mainly determined by its overall thermal efficiency in converting solar energy into stored heat energy. A comparison between the daily average total solar radiation and total solar energy collected was executed and plotted in Fig. (4). The correlation between the solar energy collected (62.734 kWh) and the available solar radiation (75.127 kWh) was high (99.49%) except that the solar collectors appear to be more efficient in March and April than in other months because the heat energy stored from the solar heating system during daylight was consumed at nighttimes (biomass heating system did not operated during these months). Accordingly the water temperatures in the storage tank at the beginning of each day throughout the two months were lower than the indoor air temperature and at the same time the intensity of solar radiation was high during these two months. As the temperature difference between the absorber surface and the water passing through the solar collectors are increased, the heat transfer rate between the absorber surface and the water is increased.

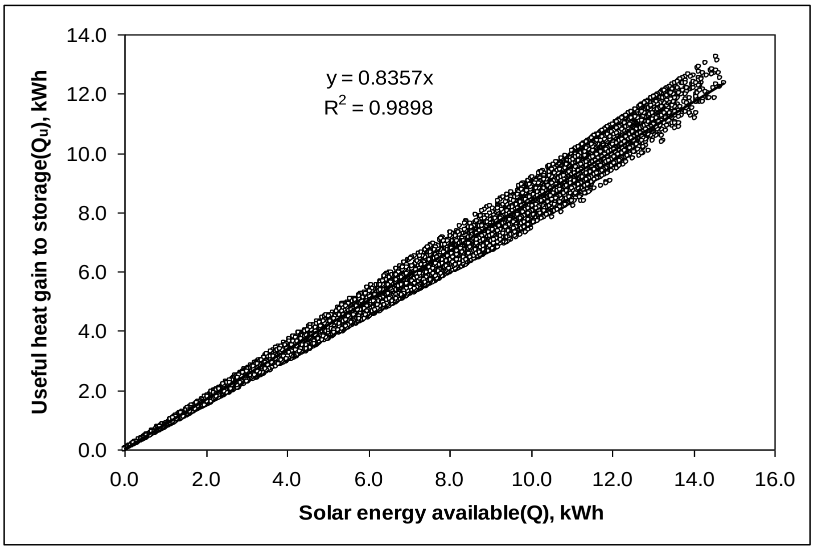

Fig. (4): Solar energy collected (useful to storage) versus solar energy available during the experimental period.

The overall thermal efficiency is the ratio of the solar energy collected by the solar collectors to the solar energy available. The daily average overall thermal efficiency of the solar collectors during the experimental period was 83.19%, consequently, 16.81% of the solar energy available was lost. The overall thermal efficiency (η_o) was correlated with the normalized temperature rise (D_T) as shown in Fig. (5). It reveals a highly coefficient of determination (R^2 = 0.9691 ; $p > 0.001$) between these parameters. The regression analysis also showed that the Y-intercept is equal to the product of the heat removal factor (F_R), and optical efficiency ($\tau\alpha$). The slope is equal to the product of the heat removal factor and overall heat transfer coefficient (U_o). The plot of overall thermal efficiency (η_o) versus normalized temperature rise (D_T) was straight line with Y-intercept F_R ($\tau\alpha$) and slope (- F_R U_o). It is clear that (U_o) is a function of temperatures difference between absorber plate and ambient air surrounding the solar collectors and wind speed. Due to the solar collectors have selectively absorber plates and covered with thermal glass, its mean value of overall heat transfer coefficient during the heating period was 5.815 W/m^2 ºC. Some variations of the relative proportions of direct, diffuse, and ground-reflected components of solar radiation occurred. Thus scatter in the data were to be expected, because of temperature dependence and wind effects. In addition, the heat removal factor (F_R) is a weak function of U_o. Therefore, the heat removal factor (F_R) during the heating period (tests) was 0.9653.

Operating fluid was pumped from the storage tank into the heat exchanger inside the biomass burner at which it heated and delivers its heat energy into the storage tank, and then re-circulated through the heat exchanger. Cold outdoor air was blow into the coil of air heat exchanger at which it heated up and directly delivered its heat energy into the indoor air of the greenhouse through perforated water galvanized pipe 50.8 mm diameter (2-inch). The outlet temperature of air blowing through blower-coil unit was allows higher than outlet solution temperatures particularly during the three feeding times of burner by the biomass materials, due to the lower inlet air temperatures (ambient air) and higher values of net heating. However, the outlet air temperature was drastically decreased particularly in early morning (2 hours prior to sunrise), owing to quench of fire inside the biomass burner. The flue gas which consists of the wet mass flow rate of 0.287 kg s^{-1} and an average flue gas temperature of 120ºC was also utilised to add the total amount of the sensible and latent heat energy into the indoor air of the greenhouse during it passes trough the thin-walled tube (located 2.30 m above the floor surface in the greenhouse centerline) along the longitudinal direction of the greenhouse. The total amount of the sensible and latent heat energy contain in the flue gas depends on the flue water content. Accordingly the flue gas was cooled to lower temperatures ranged between 20 to 40ºC. One can be clearly observed, the steep rise in the thermal output when the flue gas started condensing. The remainder of heat energy generated from the fuel materials was lost to the treatment unit of flue gas.

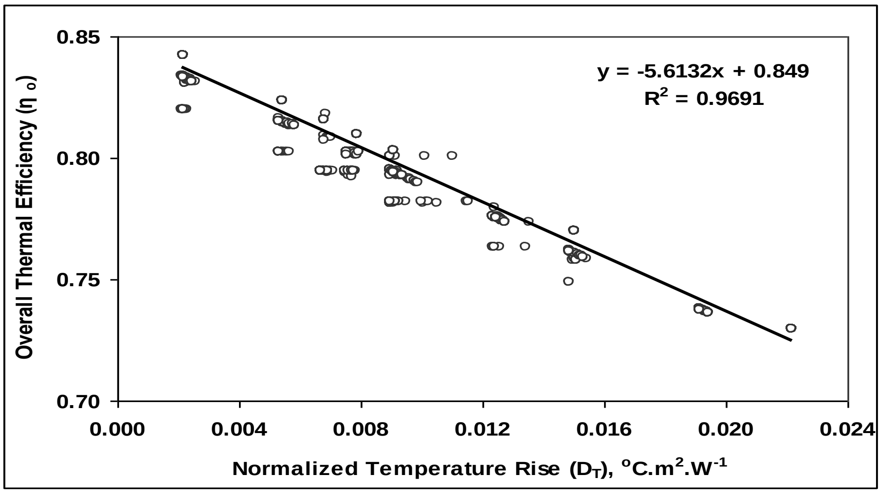

Fig. (5):Overall thermal efficiency versus normalized temperature rise during the experimental period

The greenhouse total heat energy loss from the greenhouse (sensible and latent) values (Q_{loss}), solar energy stored in the storage tank (Q_{ss}), heat energy absorbed by the operating fluid from the biomass burner (H_{wa}), heat energy absorbed by the cold air from the biomass burner (H_{aa}), heat energy added to the indoor air of the greenhouse through the peripheral of the thin-walled tube (H_{con}), heat energy losses from the biomass burner (H_{loss}) determined during the heating period from December 2012 to April 2013 are listed in Table (3). Different amount of biomass materials (wood, rice straw, cotton stalks, and corn stalks) were daily used during the heating period according the total heat energy required to provide and maintain the indoor air temperature of the greenhouse at optimal level. The biomass burner was feed by these amounts of biomass fuel in three different times per night (at 15.30, 19.30, and 23.30 hour).

Table (3): Nightly average greenhouse heat losses, solar energy stored, output heat energy, and heat energy loss during the heating period in kWh.

Month	Q_{loss}	Q_{ss}	H_{wa}	H_{aa}	H_{con}	H_{loss}
December	192.302	43.183	92.023	30.275	22.765	24.313
January	227.640	50.662	105.743	39.789	29.337	37.910
February	179.334	60.661	72.979	24.822	17.675	21.535
March	70.246	65.550	-	-	-	-
April	54.590	80.296	-	-	-	-
Total	724.112	300.352	270.745	94.886	69.777	83.758
Mean	144.822	60.070	90.248	31.629	23.259	27.919

The heat energy generated within biomass burner varied from day to day and month to another depending upon the heat energy required for heating the greenhouse, solar energy stored in the storage tank, and outdoor air temperature. The biomass burner was only operated during the first three months (December, January, and February) due to the solar energy stored in the storage tank was insufficient to meet the heat energy demanded. Therefore, the nightly average total solid fuel input in the biomass burner included wood, cotton stalks, corn stalks, and rice straw during December (22.183, 5.0, 5.0, and 5.0 kg, respectively), January (31.755, 5.0, 5.0, and 5.0 kg, respectively), and February (15.572, 5.0, 5.0, and 5.0 kg, respectively). These amount of solid fuel produced heat energy (input energy) of 174.274, 223.301, and 140.413 kWh (627.387, 803.885, and 505.487 MJ). Whereas, the output heat energy gained by the biomass system for heating the greenhouse during the same period was 145.063, 174.869, and 115.476 kWh (522.227, 629.528, and 415.714 MJ), witch provided thermal energy efficiency of 83.24%, 78.31%, and 82.24%, respectively. These data are in agreement with the data published by Hebenstreit, *et al.* (2011) and Musil-Schlaeffer, *et al.* (2011) when they reported that, the conventional small scale biomass burners reach only about 73 to 89% energy efficiency based on the net heating value. The lowest thermal energy efficiency occurred in January month due to the solid fuel materials had higher moisture content from the rainfall during this month.

Heat energy providing

During the 147 days heating period the solar heating system collected 9 222 kWh (33.199 GJ) of which 8 830 kWh (31.788 GJ) was stored in the storage tank with an average storage efficiency of 95.75%. The daily average heat energy provided by the hybrid heating system (solar collectors with biomass burner) during this period is given in Table (4), where it is compared with total heat energy requirements for providing and maintaining optimal level of indoor air temperature. During the heating period the hybrid heating system (solar and biomass energy) provided 205.206 kWh (93.84%) of the daily total heat energy required.

Table(4): Daily average total heat energy normally required (kWh) during heating season (147 days).

Energy	Heat energy, kWh per day	Providing of total, %
Solar energy		
Total useful heat energy collected	62.734	-
Total heat energy stored in the storage tank	60.070	27.47
Biomass energy		
Total output heat energy	145.136	66.37
Electrical energy		
Total energy used by water pump of solar collector	2.550	1.16
Total energy used by water pump of heat exchanger	6.355	2.91
Total energy used by water pump of heat exchanger	4.575	2.09
Total energy actually used by greenhouse	218.666	100

The potential providing from the renewable energy system not fully realized for three main reasons: firstly, little solar power was collected in the first two hours after sunrise and the last before sunset due to low solar altitude angle and water temperature in the storage tank. Secondly, throughout the heating season, the hot air from the heat exchanger of biomass burner was continuously operated and added heat energy into the indoor air in spite of its temperature was higher than the set point temperature (18ºC). This point of action resulted in extra loss of heat energy from inside to the outside atmosphere. Thirdly, during the last month of heating period (April), the solar heating system collected (82.567 kWh) and stored heat energy (80.296kWh) greater than that required for heating the greenhouse (55.090 kWh).

CONCLUSION

The primary objectives of this renewable hybrid energy sources are to increase the solar radiation converted into stored thermal energy and input heat energy of solid fuel into useful heat energy stored, and to investigate effective uses of that stored energy for heating green bens greenhouse. The hybrid system has been operated satisfactorily for over five months. The solar collectors which are continuously orientated and tilted to maintain an incident solar angle of zero from sunrise to sunset will allow maximum values of optical efficiency (0.931). The overall thermal efficiency and heat losses are mainly affected by the water inlet temperature and ambient air temperature.

Over the period December 2012 to April 2013, the solar heating system collected 9 222 kWh (33.199 GJ) of which 8 830 kWh (31.788 GJ) of solar power was stored in the storage tank with an average storage efficiency of 95.75%. During the heating period the daily average useful solar energy collected was 62.734 kWh of which 60.070 kWh was stored in the storage tank and consumed during the heating period for the greenhouse. It provided 27.47% of the total power required by the greenhouse (218.666 kWh).

Over the period December 2012 to February 2013, the biomass heating system added 145.136 kWh (522.490 MJ) of heat energy into the greenhouse which provided 66.37% of total power required for greenhouse heating. The hybrid heating system (solar and biomass energy) provided 205.206 kWh (93.84%) of the daily total heat energy required. The economics of such a system remains marginal at present power prices in Egypt, although changes in power costs may drastically alter the situation.

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تقييم الأداء الحرارى للمصادر الهجين للطاقة المتجددة لتسخين البيوت المحمية

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تهدف هذه الدراسة إلى تحليل الأداء الحرارى لنظام هجين للطاقة الشمسية وطاقة الكتل الحيوية المستخدمة فى تسخين البيوت المحمية. تم بحث الإحتياجات الحرارية الكلية عملياً لبيت محمى تجارى ($1010.4\ m^3$) يقع فى المنطقة الشرقية لدلتا النيل عند خط عرض وطول وإرتفاع عن سطح البحر على التوالى (31º 02' 41" N, 31º 21' 55" E, and 6 m) أثناء موسم الشتاء 2012-2013. تحليل الأداء الحرارى لنظام التسخين بالطاقة الشمسية تم تحديده عملياً بقياس الزيادة فى درجة الحرارة عند درجات حرارة دخول مختلفة وشدة الأشعة الشمسية تحت ظروف السماء الصافية. تم إستغلال نظام كامل للتسخين بالطاقة الشمسية مكون من 6 مجمعات شمسية وخزان تخزين للطاقة الحرارية سعته 1500 litres بلغ المتوسط اليومى للكفاءة الكلية لنظام التسخين بالطاقة الشمسية وكفاءة نظام التخزين على التوالى 83.19% and 95.51%. خلال موسم التسخين 147 days تم تجميع طاقة حرارية مقدارها 9 222 kWh والتى تم منها تخزين طاقة حرارية مقدارها 8 830 kWh إستخدام نظام التسخين بالطاقة الشمسية أدى إلى توفير نسبة مقدارها 27.47% من إجمالى الطاقة اللازمة لتسخين البيت المحمى التجارى. نظام التسخين بإستخدام طاقة الكتل الحيوية أضاف طاقة حرارية للبيت المحمى مقدارها 145.136 kWh والتى وفرت نسبة مقدارها 66.37% من إجمالى الطاقة اللازمة لتسخين البيت المحمى التجارى. النظام الهجين للتسخين وفر طاقة حرارية متوسطة مقدارها (93.84% 205.206 kWh من من إجمالى الطاقة الحرارية اللازمة لتسخين البيت المحمى التجارى. إقتصاديات تشغيل مثل هذا النظام فى مصر تبقى قريبة من الحد الأدنى لأسعار الطاقة الحالية، إلا أن أى تغير يحدث فى تكاليف الطاقة التقليدية من الممكن أن يغير الوضع بحدة.

قام بتحكيم البحث

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