Design of Solar Energy based Hybrid Community Cooking System

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Abstract: Energy is considered to be an essential factor contributing in the economic and social development of any nation. Shortage of energy or energy imbalance is an impediment to the growth of a country. Energy generated from the non – renewable sources like coal, petroleum etc. harm the environment by emitting harmful green house gases. Using alternative renewable sources for generation of energy will save our earth and reduce the emission of greenhouse gases. Renewable energy sources can serve as alternatives to conventional fuels (LPG, coal, petroleum etc.). Most of the community messes use LPG as a prime source of energy for cooking purposes. This research work focuses on the design of solar based hybrid community cooking system which is a good alternative to LPG based cooking system. Heat required for boiling rice, pulses and green vegetables per day was calculated for 400 persons in community mess. A hybrid cooking system based on solar water heater, heat pipe, biogas and boiler was designed and integrated as per the heat requirements of 400 students. The integration also aimed at the reduction of wood consumption in the boiler of hybrid cooking systems, thus reducing emissions of harmful gases to a minimum.

Keywords: Hybrid cooking system, solar based cooking system, solar water heater, heat pipe set-up, cooking vessel and community cooking system.

I. INTRODUCTION

Hybrid systems are the integration of various individual renewable energy based devices. This paper explains the methods adopted in designing of devices involved in hybrid renewable energy based cooking systems. The research paper includes the designing, development and capacity estimation of solar water heater, heat pipe system, biogas system and fire tube boiler (auxiliary system) for the required heat load in cooking, keeping in consideration all losses. This system was designed to maximize solar energy conversion into heat energy for cooking purposes. Ong et al. (2012) reported that solar water heaters performance depend upon collector, storage tank design, sizing, intensity of solar radiation and surrounding temperature.

Venden et al. (2012) in their paper discussed formula and calculations involved in design of solar thermal system which generates steam at atmospheric pressure. Application of this system to generate steam which generates steam various purpose as a cooking, laundry etc. This paper deals with calculations involved in area estimation of solar collector and number of tubes require in solar thermal system.

Sivaraman et al. (2005) in their studies focused on analysis effects of solar intensity through artificial neural network (ANN), L/di (total length/ inner diam. of heat pipe), water inlet temperature, L_c/L_e (length of condenser / length of evaporator) and tilt angle of collector on solar heat pipe collector. A set of five heat pipe having different ratio of L/d_i and L_c/L_e had been developed, assembled and utilized in solar based collector absorber.

Ayompe (2013) presented thermal performance analysis of solar based water heating system with evacuated heat pipe tube type solar heat collector. For this data recorded from a field trial installation over a year in Dublin, Ireland. An advance sub-system was developed which incorporated with a control system on the hot water draw- offs and having an electric heating system with it to support operation of solar water heating systems in domestic dwellings when sunlight is not presented. The recorded maximum outlet temperature of collector fluid was 70.2°C whereas temperature of water at hot water tank bottom was 59.4°C. The average energy collected daily basis was found to be 20.5 MJ/day. The energy absorbed by coil of system was 16.7 MJ/day; loss in supply pipe was 3.5 MJ/day. Solar fraction was calculated as 33.7% and efficiency of collector was 63.4%. The efficiency of system was calculated as 52.2 %. It was concluded that reduction in losses of supply pipe was mentioned 17.8 % of energy collected, Energy delivered to hot water tank was 21.5%. Intermittent and heavily overcast cloud covered days system performance can be improvement through better pump control strategy.

Hlaing (2014) in his work investigated an analysis of heat transfer in heat pipe. Through the use of comsol multi-physics, heat transfer between the outer and inner glass tubes and heat transfer in heat pipe wall etc. were analyzed. Result showed that high solar radiations were required to obtain maximum temperature of hot water. Thus, this research provided important information for heat pipe construction for heating water. Siva Kumar et al. (2017) conducted their studies on solar collector. An evacuated tube solar collector (ETSC) was theoretically modeled and fabricated with evacuated tube and heat pipe.

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Optimized parameter for ETSC was evaluated numerically by doing outdoor testing at Coimbatore district Tamilnadu. They found that the time required by this collector to gain the heat was less compared to other collectors. The outlet temperature varied from 50°C to 79°C while surrounding temperature lies between 24°C to 33°C. For finding the suitable model, the regression analysis was performed. For experimental data suitability, five different models were used. The best curve fitting with and lowest value of root mean square error and highest correlation coefficient was obtained using MATLAB software. The ETSC was designed and fabricated with the length and diameter of 1.8 m and 0.058 m respectively.

Prabhakant et al. (2011) wrote the energy balancing equation for the different components of hybrid photovoltaic thermal (HPVT) biogas plant for the quasi - steady state conditions to develop a thermal model. An expression for slurry temperature has been obtained as a function of design and climatic parameters. it has been observed that the optimum slurry temperature 37°C at 0.05kg/s for L = 25m is achieved for a given set of design parameters and hybrid collectors (M_S =2000) of plant. It was observed that number of hybrid PVT collector had significant effect on slurry temperature.

Bolaji et al. (2009) in their studies parameters for an improved coal stove were designed. Stove was fabricated. And testing of stove was conducted for evaluating performance related to thermal parameters. Performance of designed stove was compared with old practiced coal based and conventional kerosene based stove. Results showed that the improved coal stove handles fuel more efficiently having coal burning rate of 0.15kg/h. And it is also economically in comparison to the old type coal stove, which had 0.20 kg/h coal burning rate. The thermal efficiency of the modified coal stove was calculated as 42.6%, as compared to old practiced stoves such as old coal stoves and kerosene based stove which had an efficiency of 28.2% and 40.5% respectively.

Sharma et al. (2017) in their paper aimed to design a horizontal fire tube boiler having two pass system. The output of this boiler was about 300 kg steam per hour. It's tank was fabricated by pure mild steel. Indian coal as the fuel was used in this boiler. The boiler generated dry saturated steam at 5 bar and 150°C. This boiler was used for commercial cooking of the Indian food in restaurants, food industries, dairies, etc.

rectification in the final paper but after the final submission to the journal, rectification is not possible.

I. Design of the community cooking system

List of main components:

- ETC based solar water heater
- Solar heat pipe manifolds
  (Pressurized copper tube type)
- Intermediate storage tank
- Deenbandhu biogas plant
- Biogas or wood fired Non IBR Boiler
  (Auxiliary system for Backup)
- Steam cooking vessels

Formulas for designing the components of hybrid community cooking system

- **Initial design conditions and assumptions**

Total number of students : 400

Working fluid for heating & steaming : water

Maximum cooking temperature : 130°C

Pressure at the time of final heating in the boiler : 2.7 bar

Average heating time : 8 hours

(8:00 hours to 16:00 hours)

Average Insolation, I : 600 W/m^2

Ambient temperature, T_1 : 25°C

Heat required for cooking food per person per day: 2MJ (Amarasekara, 1994)

Food items for boiling: Pulses, green vegetables, rice, idli, boiled bati (bafla bati) and hot water for other general purpose.

- **Energy analysis for the system**

Table: 1 Heat required for cooking the below given food items (per meal per day) for 400 students.

Sr. no	Food articles	Quantity required (kg)	Specific heat Kwh/kg	Heat required KJ/kg	Total heat required (MJ)
1	Rice	65	0.08	288	18.72
2	Pulses	40	0.1	360	14.4
3	Green vegetables	65	0.1	360	23.4
4	Hot water for tea	25	0.0116	4.18	7.8
Heat required (Q_{ideal})					64.3 MJ

Total heating load for boiling of food items : Q_T

Efficiency of cooking vessel due to Heat losses (η_{vessel}): 50%

Average heat required : 25 % of total heat (for breakfast and other snacks)

Actual heat required for cooking, Q_{actual}

$$Q_{actual} = \frac{\text{Total Heat required}}{\text{Efficiency of vessels}} \quad (1)$$

$$Q_{actual} = \frac{Q_T}{\eta_{vessel}} \quad (2)$$

= 64.3/0.5

= 128.6 MJ/meal

Heat energy required for 2 meal/day (Q_{meal}) = 2 × Q_{actual} (3)

= 257.2 MJ /day

$Q_{breakfast}$ = 257.2 ×25 /100

= 64.3 MJ/day

Total heat required /day = Q_{meal} + $Q_{breakfast}$ (4)

= 257.2+64.3

= 321.5 MJ /day

- **Calculations for the requirement of water:**

Actual heat required for cooking

Q_{actual} = $m_w C_p \Delta T$ (5)

Mass flow of water required (Kg/day)

$$m_w = \frac{321.5 \times 10^3}{4.18 \times (403 - 298)}$$

= 732 kg/day

Volume of water required per day , V_w

$$V_w = M_w/\rho \quad (6)$$

$$= \frac{732}{1000} = 732\times10^{-3}\ m^3/day$$

Volume of water required per day (V_w) = 732 liters/day

Total hours of running = 8 hours

Therefore, capacity of solar water heater = 732/9

= 81.5 liters/h

Considered standard Capacity of solar water heater

= 100 liters/h

Heat gain per hour from hybrid system , Q_G

For cooking, temperature of water will be raised from ambient temperature (T_1=25°C) to T_8=130°C at working pressure of 2.7 bar. Total heat gain by water in hybrid system at the end of process was given by -

$$Q_G = m_w C_p \Delta T \quad (7)$$

Where

$$\Delta T = T_8 - T_1 \quad (8)$$

m_w = mass flow of water required (kg/h)

T_1 = Initial temperature of water or temperature in overhead water tank

T_8 = Final temperature of pressurized water (2.7 bar) before entering into the cooking vessel

(from steam table data T_{sat}.= 130C at 2.7 bar)

C_p = Specific heat (KJ/kg K)

= 4.18 KJ/kg K (at 1 bar pressure)

From observations

T_1 = 25 °C + 273 =298 K

T_8 =130 °C + 273 = 403 K

Q_G = 100 ×4.18× (403-298) = 43890 KJ/h

= 43.890 MJ/h

Required operating hours of system

= Actual heat required / heat gain per hour

= Q_{actual} / Q_G (9)

= 321.5 /43.89

= 7.33 hrs/day

Consider

≈ 8 hrs/day

Design parameters of various components

- **Design of Solar water heater**

The designing of solar water heater included consideration of various parameters like mass flow rate per hour, area of the collector, number of evacuated tubes required etc. Following terminologies were used for the calculation of various design parameters.

I = Intensity of solar radiation (insolation)	: 600 W/m^2
Angle of inclination of solar collector	: 45˚
τ = Transmissivity of the Glass	: 0.8
α = Absorptivity of the collector	: 0.9

(Glass tubes coated with high absorptivity material inside)

Q_{loss} = Loss of heat through convection and radiation

U_L = Overall heat loss coefficient

T_1 = Ambient temperature of water

T_c = Collector temperature

T_1 & T_2 = Inlet and Outlet temperature of water at solar water collector

F_R = Collector heat removal factor

A_c = Area of solar collector

Q_i = Incident energy

The thermal analysis of evacuated tube collector incorporates the energy balance of collector which is based on thermodynamics first law. Solar radiation are converted into heat which falling on the collector. That heat is absorbed by the working fluid for useful applications (Tyagi et al., 2012)

a) Total mass of water required = 800 liters/day

Operational hours = 8 hours

Capacity of solar water heater required = 800/8

=100litres /h.

Radiation received by solar collector, Q_r

The amount of radiation received by the solar collector can be given as follows-

$$Q_r = I\,A_c \quad (10)$$

Actual Incident Energy, Q_i

A part of total radiation energy received by the collector is reflected back and some of solar radiation is absorbed by the transparent absorber. So actual incident energy on collector was given by-

$$Q_i = I\,(\tau\alpha).A_c \quad (11)$$

Heat loss from the collector, Q_{loss}

The heat loss (Q_{loss}) rate depends on overall heat transfer coefficient (U_L) of the collector and the temperature of collector was given by-

$$Q_{loss} = U_L A_c (T_c - T_a) \quad (12)$$

Rate of useful energy extracted by collector, Q_u

Useful heat delivered through collector is equal to the difference between energy absorbed by the absorber and heat losses from the surface directly or indirectly;

$$Q_U = Q_i - Q_{loss}$$

$$= I\,(\tau\alpha).\,A_c - U_L A_c (T_c - T_a) \quad (13)$$

Heat absorbed by the fluid (Q),

Heat absorbed by fluid is the heat gain by water from the solar water heater

$$Q = m\,C_p\,(T_o - T_i) \quad (14)$$

Collector heat removal factor, F_R

$$= \frac{\text{Heat absorbed by fluid } (Q)}{\text{Useful heat delivered by collector } (Q_u)}$$

Heat balance equation;

$$M_w\,C_p(T_2-T_1) = A_c\,F_R\,\{I\tau\alpha - U_L(T_i-T_a)\} \quad (15)$$

$$Q = A_c\,F_R\,\{I\tau\alpha - U_L(T_i-T_a)\} \quad (16)$$

This equation is known as *Hottle- whillier-Bliss* equation.

Solar collector efficiency, η

Performance of collector or efficiency stated as the ratio of the useful energy gain (Q_u) to the incident solar energy over a particular time period

$$\eta = \frac{Q_u}{A_c It}$$

$$\eta = \frac{F_R\ A_c\{\tau\alpha - U_L(T_1 - T_a)\}}{A_c It} \quad (17)$$

Total collector area (A_c)

$$= \frac{Total\ heat\ requirenemnt\ per\ hour}{Thermal efficiency\,(\eta) \times Global\ radiation\ (I) \times Time}$$

$$= \frac{M_w\ C_p\ \Delta T}{\eta\, I\, t} \quad (18)$$

T_1= 25°C=298K

T_2= 50°C

(Assume maximum possible temperature at out let) =323K

Average solar intensity = 600 W/m^2

The efficiency of the solar collector ranges between 0.4 to 0.6 (duffle et al., 1991).

For the present work an efficiency of 0.4 was considered.

$$\frac{100 \times 4180 \times (333 - 298)}{600 \times 0.4 \times 8 \times 3600} = 2.11 m^2$$

Number of Tubes, N

Number of tubes in the solar water heater was determined as

$$Number\ of\ tubes\ = \frac{required\ area\ of\ collector}{projected\ area\ of\ standard\ tube} \quad (19)$$

A_{tube} = area of each evacuated tube in m^2 (L×D) = 0.106 m^2

$$\text{number of Evacuated tubes required}\,(N_t) = \frac{2.11}{0.106}$$

N_t = 20 tubes

Solar fraction, SF

Solar fraction is defined as the ratio of solar heat yield to the total energy requirement for solar water heating. It is given by

$$SF = \frac{Q_s}{(Q_s + Q_{aux})} \quad (20)$$

Where

Q_s = Solar heat yield

Q_{aux} = Energy from other sources

- **Design of heat pipe manifold system**

Working fluid selection:

Points to be kept in mind while selecting fluid-

- Operating pressure and temperature range
- Operating temperature range and Operating Pressure
- Liquid transport factor
- Vapour phase properties
- Thermal conductivity
- Fluid compatibility and stability

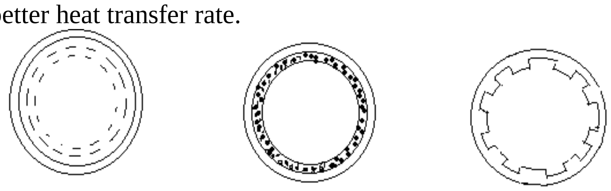
Table: 1 Range of temperature for working fluids used in the heat pipes

Medium	Boiling point at atmos. Pressure (°C)	Melting point (°C)	Useful range (°C)
Helium	-261	-271	-271 to -269
Ammonia	-33	-78	-60 to -100
Acetone	57	-95	0-120
Methanol	64	-98	10-130
Ethanol	78	-112	0-130
Water	100	0	50-200
Toluene	110	-95	50-200
Mercury	361	-39	250-650
Sodium	892	98	600-1200
Silver	2212	960	1800-2300

Water was selected for the required temperature range 130°C, because even if there was any leakage during the cooking process, no harmful toxins were released in the food.

Selection of wick design

For the return of liquid from the condenser to evaporator wick provides the necessary flow area. And it also provides pores for capillary pumping. Wick properties are characterized by the effective pumping radius and high permeability. These cross sectional areas and properties determine the ability of the heat pipe to overcome hydrodynamic losses. Grooved wick structre was selected for better heat transfer rate.



(A) Wrapped screen (B) Sintered metal (C) Axial groove

Figure :1 Types of Wick Structure in heat pipe

Container Design:

Container was shell or tube type to accommodate the working fluid and wick structure within itself. Generally copper was used for higher thermal conductivity (K=400 W/m °C) and low cost as suggested by Hada (2015). Other Consideration were

- Leak tight containment of working fluid.
- Compatibility with working fluid
- Compatibility with external environment.
- External interfacing with the heat source and sink.
- Internal size and geometry suitable for liquid and vapor flow requirement.
- Conduction through the container wall

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- Heat transfer in evaporation and condensation process within the pipe

Heat Pipes Limits

Capillary Limit

The capillary limit is the phenomenon of developing differences in capillary pressure across the liquid-vapor interfaces in the condenser and evaporator of heat pipe. When the capillary pressure is inadequate to provide liquid flow between condenser and evaporator, wick of evaporator will dry-out. The capillary pressure difference acts as driving potential for the circulation of the working fluid. The sum of all pressure losses inside the heat pipe must be less than the maximum capillary pressure. (Nemec et al, 2013)

$$(\Delta P)_c \geq \Delta P_{total}$$

The drop in total pressure was calculated as the sum of the liquid, vapor, normal hydrostatic, and axial hydrostatic pressure drops.

$$\Delta P_T = \Delta P_v + \Delta P_l + \Delta P_b + \Delta P_{ph} \quad (22)$$

Where

ΔP_v = *frictional pressure drop along vapor path*

ΔP_l = *frictional pressure drop along liquid path*

ΔP_b = *pressure drop due to body forces*

ΔP_{ph} = *pressure drop due to phase Transition*

Viscous limit

When the operations of heat pipe takes place at low temperatures, the available vapor (saturation) pressure in the evaporator region may be very small and of the same magnitude as the required pressure gradient to drive the vapor from the evaporator to the condenser. In this case, opposing viscous forces will be balanced by the total vapor pressure in the vapor channel. Therefore, within the vapor region total vapor pressure may be inadequate to sustain an increased flow. This condition in the vapor region is known as the viscous limit. Maximum heat transport capacity under viscous limitation is Q_v (Nemec et al., 2013).

Sonic Limit

The sonic limit works as an upper limit to capacity of the axial heat transport and may not always result in total heat pipe failure or dry-out of the evaporator wick. Crossing the sonic limit will result in increasing both the axial temperature gradient and evaporator temperature along the heat pipe. This will further reduce the isothermal characteristics typically found in the region of vapor flow. It is concern with the cross-sectional area through which the vapor flows. Maximum capability of heat transfer as a function of thermo-physical and geometric properties are Q_s as limited by sonic limit.

Entrainment limit

This limit is the situation when high shear forces are developed as the vapor passes over the liquid saturated wick in the counter flow direction. The liquid may be entrained by the vapor and returned to the condenser. This results in inadequate flow of liquid in the wick. Maximum heat transport capacity based on entrainment limit is Q_e (Nemec et al., 2013).

Boiling Limit

This limit is attained when the evaporator heat flux applied is adequate to cause nucleate boiling in the wick of evaporator. This creates bubbles of vapor which block the return of liquid partially and results in dry-out of the evaporator wick. It is also known as the heat flux limits which found making use of a thermal impedance model for the wall and capillary system, and the evaporator area. This is a serves as a tool for choosing wall material-fluid combination. Maximum heat flux under boiling limit is Q_b.

It can be concluded that all equations need to be satisfied simultaneously for a range of operating conditions for the heat pipe to work properly according to design specifications.

- **Designing of firewood based boiler**

In the present system, Non IBR (Indian boiler regulation) horizontal water tube boiler was selected for heating water to a temperature of 130°C at a pressure of 2.7 bars. According to the Indian boiler regulation (IBR), Non IBR boilers are of capacity less than 25litres.

Thermal efficiency of the boiler, η_{Boiler}

It is the ratio of the total heat gain in the boiler to the actual heat supplied to the boiler.

$$\eta_{Boiler} = \frac{\textit{Total heat gain in the Boiler}}{\textit{Actual heat supplied to the boiler}} \quad (23)$$

Indirect efficiency of a boiler was calculated by finding out the individual losses taking place in a boiler and then subtracting the sum from 100.

η_{Boiler} = 100 – (Sum of individual losses in the boiler) (24)

Consumption of wood in the boiler, M_{wood}

It can be defined as the ratio of total thermal energy required to the actual energy input in the boiler.

$$M_{wood} = \frac{\textit{Heat energy required}}{\textit{actual Energy input}} \quad (25)$$

Actual energy input was calculated by considering efficiency of the boiler and the calorific value of the firewood.

$$M_{wood} = \frac{M_w \times C_p \times \Delta T}{\eta_{boiler} \times CV\ of\ fire\ wood} \quad (26)$$

❖ **Design of biogas system**

Energy available in Bio gas

Calorific value of pure methane (CH_4) = 55 KJ/kg
Density of methane at NTP =0.668 kg/m^3
Therefore gross CV of methane in m^3 = 39.8 MJ/m^3
Methane is the only prime source of heat, biogas contain about 50%-65% of methane. So CV of biogas = (0.50* 39.8) to (0.65*39.8) = 20 to 25.9 MJ/m^3
CV taken for calculation = 22.4 MJ/m^3.
Density of biogas =1.15 kg/m^3.
(*A plant of 1m^3 capacity produces 0.7m^3 gas per day)
Energy required per day is =321 MJ.
Energy obtained per cubic meter of bio gas in 24 hours = 22.4 MJ
Taking gas burner efficiency same as that of boiler efficiency, the requirement of gas for 1 day is = 321.5/ 22.4 = 14.35 m^3

The required capacity of biogas plant for gas production per day was 14.35 m^3. It required large area of land as well as huge amount of cow dung and kitchen waste. So, as per the quantity of kitchen waste available a standard size bio gas plant was considered.
Considering bio gas plant of capacity = 6 m^3
= 6000 liters
Additional energy requirement was fulfilled by fire wood boiler. Heat gained from a biogas digester is given by (Q_{bio})
= η (C.V) * V_d
= 0.5 * 22.4 * 6 = 68 MJ

II. RESULTS & DISCUSSION

All designed parameters were calculated according to the requirement of heat load for solar energy based community cooking system, along with their standard specification and are presented below-

- **Design parameters and standard specifications of various components**

Table: 1 Design parameters of solar water heater

Detail	Design data & other Specification
Type	ETC
Number of tubes	30
Area	2.5 square meter
Capacity	100 liter/h
Faces	North facing
Average heating	8 hours
Angle of inclination of tubes	45
Absorptivity	0.9
Transmissivity	0.8

Table: 2 Design parameters of heat pipe system

Details	Design & Specifications
Collector	
Material	TU1 Copper, length 1800mm with Aluminum fin
Structure of tube	Double-tube co-axial glass structure
Material of glass	Borosilicate
Absorber Plate	
Absorber coating material	AL-N/SS/CU
Absorber coating method	Three targets magnetron sputtering plating
Absorption	≥92.3%(AM1.5)
Emittance	≤3.6% (80°c±5°c)
Vacuum degree, P	$\leq 5.1\times10^{-3}$Pa
Glass tube transmittance	≥91%(AM1.5)
Stagnation parameter	>218m² ° C/kW
Heat loss (average)	ULT≤0.60W/(m²8*°c)
Hail resistance	30mm

Table: 3 Properties of heat fluid used in the heat pipe

Properties of Fluid	
Pressure-endure	0.8Mpa
Temperature-endure	250°c
Working fluid	Water
Liquid density, ρ_l	1000kg/m^3
Liquid viscosity, μ_l	0.5588*10^{-3} Ns/m^2
Surface tension, σ	0.0679 N/m
Vapour density, ρ_v	0.08 kg/m^3
Latent heat of vaporization, H	2383 KJ /kg

Table: 4 Dimensions of heat pipe

Details	Design & Specifications
Out tube diameter	58±0.7mm/47±0.7mm
Inter tube diameter	47±0.7mm/37±0.71mm
Thickness of glass	1.62 mm
Length of tube	150cm/180cm/210cm
Evaporator Length, L_e	1.36 m
Condenser length, L_c	0.4 m
Adiabatic Length, L_a	0.04m

- **Non-IBR boiler**

Material of boiler : Mild Steel
Volume of boiler : 25 liters
Type : Horizontal water tube boiler
Maximum operating temperature: 150°c
Operating pressure : 2.7 bar

Boiler Efficiency : 50% (due to thermal losses)
Fuel : Firewood and biogas
Recommended pH of boiler feed water : 6.5 to 7.5

- **Biogas plant**

Table: 4 Design parameters and specifications of bio gas plant

Details	Design data and specifications
Type	Deenbandhu
Capacity	6 m^3
Input	50% Cow dung & 50% kitchen waste was mixed with equal proportion of water

❖ **Hybrid community cooking system based on solar energy**

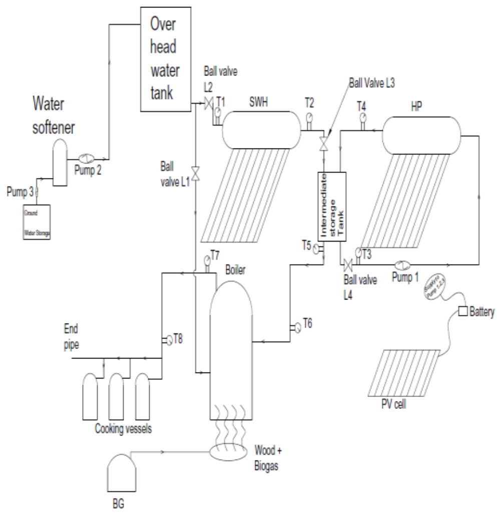

Figure: 2 Arrangements of components in hybrid community cooking system

III. CONCLUSION

This paper illustrated the step by step design process of each components of hybrid community cooking system. On the basis of design parameters hybrid community cooking system was installed for 400persons. After a number of experimental attempts the system performed efficiently which directly established the scope of renewable energy in community cooking. The overall cost of cooking through solar energy based hybrid system was $(1/3)^{rd}$ as compared to the cooking through LPG gas.

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