



Application of computer aided design for palm kernel shell steam boiler

Kamoru Olufemi OLADOSU^{1,3*}, Buliaminu KAREEM², Basil Olufemi AKINNULI² and
Tesleem Babatunde ASAFA^{1,3}

¹*Department of Mechanical Engineering, Ladoke Akintola University of Technology
Ogbomoso, Oyo State, Nigeria*

²*Department of Mechanical Engineering, Federal University of Technology Akure, Ondo
State, Nigeria*

³*Science and Engineering Research Group (SEARCH) LAUTECH, Oyo State, Nigeria*
E-mails: kooladosu@lautech.edu.ng, karbil2002@yahoo.com, ifembola@yahoo.com and
tbasfa@lautech.edu.ng

*Corresponding Author: +234 803 595 9872

Received: December 21, 2016 / Accepted: June 13, 2017 / Published: June 30, 2017

Abstract

Steam boiler is an integral and important component of steam turbine used for electricity generation. Its design is however complex, time consuming and prone to errors if done manually. This study aimed at using computer based approach to design palm kernel shell combustling furnace for generating a desired amount of electricity. By way of backward calculation approach, standard design equations were used to size furnace and its components. The equations were coded and solved using C-Sharp programming language. The results showed that to generate 5 kW of electricity from palm kernel shell; 5.5 kW turbine, 3.1 m super heater, 3.8 m riser, furnace of 1.432 m height and 0.45 m³ volume were required having considered power loss due to friction and others. While these results are in good agreement with those calculated manually, human errors are virtually eliminated. In addition, calculations and drafting time were reduced from 5 hrs. 47 minutes when done manually to about 4 minutes when the developed code was used. This code can be used to size boiler for any desired power output.

Keywords

Steam boiler; Palm Kernel Shell; Design; Computer aided; Power output

Introduction

In palm oil processing industry, biomass residues can be converted from being potential environmental pollutants to useful fuel for steam and electricity generation which are largely needed for industrial use [1]. Nigeria, being the fifth largest producer of palm oil, accounts for about 1.5% (930,000 metric tonnes) of the global output. However, a huge quantity of oil palm residues which could otherwise be used for energy generation are being wasted [2]. In [3] is reported that about 30 tonnes of fresh fruit bunches /hr. produce from a few palm oil mills can be used to generate up to 20 - 35 MW of electricity. This can significantly reduce greenhouse gases and increase employment for local population [1].

There are several technologies that enable oil palm mill to generate enough energy for its consumption and sometimes for export. Among them are fixed (1 kW- 50 MW), fluidized (5 MW- 100 MW) and dust technology (10 MW- 500 MW). Efficiencies of these technologies are dependent on fuel properties and the mixing quality between flue gas and combustion air [4]. Pitchet and Vladmir [5] recorded high combustion efficiency and low emission performance in a fluidized bed combustion of palm kernel shell using optimized particle size, although the start up and running cost of operation associated with this technique make it difficult to be operated by small scale business. Remarkable improvement has also been recorded on design of large scale grate furnaces (fixed bed), yet additional work need to be done in small scale businesses in term of poor mixing especially when co-firing different fuel and high moisture fuel content for improve combustion and reduction of ash deposition on components of grate furnace [6].

The unique features of grate furnace are the tolerance of fuel type; positive movement of fuel down grates reduces blockages and well controlled air distribution lead to high combustion efficiency [7]. In addition, the use of additive mixed with solid wastes can significantly reduce alkaline metals deposition on the surface of riser tubes [8] and [9]. These will increase combustion process and decrease ash deposition. Boiler design is a complex and time consuming procedure. It is also prone to errors if done manually. Previously, emphasis was laid on primitive and probabilistic design processes which resulted in high cost of production. Dimensions of boiler for power generation often depend on fuel and vaporization efficiency; the mass balance, heat balance and heat transfer which has to be specified through empirical results and experiences. The aim of this study is to use computer based approach to

design palm kernel shell combusting furnace for generating a desired amount of electricity, thereby providing substantial savings in terms of time and cost of production.

Nomenclature

A	Area (m ²)
H	Height (m)
V	Volume (m ³)
m	Mass (kg)
h	Heat transfer coefficient (kJ/kg)
k	Thermal conductivity (W/m ⁰ C)
d	Diameter (m)
L	Specific latent heat of vaporization (2256kJ/kg)
C	Specific heat capacity (kJ/kgK)
q	Heat duty required (J/s)
T	Temperature (⁰ C)
M	Molecular mass of flue gas (kg/kMol)
U	Overall heat transfer coefficient
t	Wall thickness
R	Characteristic gas constant (J/kg ⁰ C)
P	Power (W)
P _s	Pressures at steam drum (N/m ²)
C _{pm}	Specific heat capacity of steam (kJ/mol ⁰ C)
m _s	Mass flow rate of steam (kg/s)
c	Size of bottom PKS fired furnace (m)

Greek symbols

η	Generator efficiency
γ	Index number
x_g	Dryness fraction of steam/water mixture
μ_k	Dynamic viscosity of the flue gas (kg/ms)
α	Exterior angle of truncated cone (Degree)

Subscripts

<i>in</i>	Inlet	<i>S</i>	Saturation
<i>out</i>	Outlet	<i>g</i>	Flue gas
<i>p</i>	Constant pressure	<i>w</i>	Stainless steel
<i>v</i>	Constant volume	<i>b¹</i>	Water boiling
<i>sp</i>	Superheater	<i>sp₁</i>	Superheated
<i>b</i>	Risers	<i>k</i>	Fraction
<i>f</i>	Fuel	<i>turb</i>	Turbine

Material and method

Palm Kernel Shell (PKS) were collected from a local palm oil processing mill in Ogbomoso, Southwestern Nigeria. The shells were crushed into smaller pieces by using a granulator (SG-16 Series) and further reduced with a blender. They were subsequently sieved to 5.0 mm particle size according to [5]. The proximate and ultimate analyses of the PKS were done following [10]. Higher Heating Value (HHV) of the mixtures was determined using GallenKamp Bomb Calorimeter according to [11].

Development of Grate Furnace and its Components

The furnace under consideration was based on principle of water tube natural circulation. The main components of this furnace are steam drum, down comer, riser tubes which represents the complete fluid flow loop. Water flows to the steam drum through down comer riser loop.

The riser tubes were situated inside furnace where heat of flue gases vaporizes the water into steam and back to the steam drum through steam header collection (Fig. 1). Because steam water mixture inside riser tubes is less dense than the saturated water at inlet tube, fluid flows upwards in the riser tubes and back to the drum. The density difference between water at the inlet tube and steam-water mixture produces enough force to overcome friction and gravitational resistance to flow, therefore maintain a steam flow system [12]. The steam drum is partitioned into two zones.

The lower section allows water intake to the drum while the upper section produces steam which flows from the top of the drum into the super heater tube. The superheated steam is expected to turn turbine to generate electricity.

The design analysis follows backward calculation approach of sizing the power plant component to generate steam for 5 kW of electricity (Generator-Turbine-Super heater -Riser tubes- Furnace dimensions). The design approach to each component is described in the following sections.

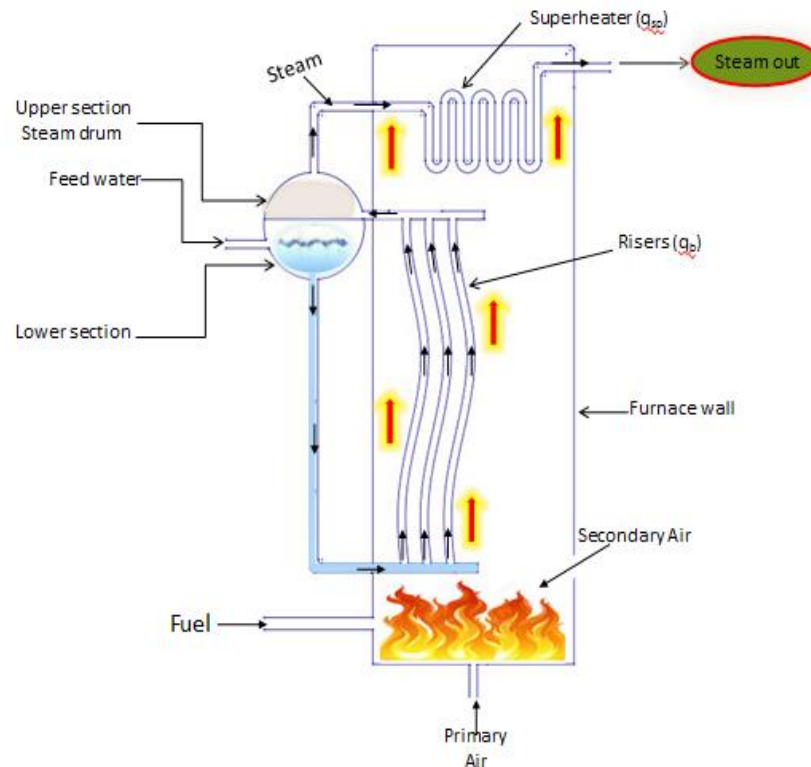


Figure 1. Schematic diagram describing water to steam circulation loop

Turbine

Turbine is a rotary engine that converts the energy of the steam, water or gas into mechanical energy. The mechanical energy is then transferred through a driven shaft to power electric generator. The power input from turbine P_{turb} (kW) can be related to the power output of generator P_{out} (kW) by Eq. (1).

$$P_{turb} = P_{out} + P_{loss} \quad (1)$$

The generator efficiency η is calculated from Eq. (2).

$$\eta = \frac{P_{out}}{P_{out} + P_{loss}} \quad (2)$$

Where: $P_{loss} = P_{mechanical} + P_{I^2R_{loss}}$

Mass flow rate of steam m_s (kg/s) from super heater entering turbine was estimated using energy equation for adiabatic expansion which relates the power output to steam energy declining by passing through the turbine [13], Eq. (3):

$$m_s = \frac{P_{turb}}{C_{pm} \eta_t (T_{in} - T_{out})} \quad (3)$$

For steam, $C_p = 1.8723 \text{ kJ/kgK}$ and $C_v = 1.4108 \text{ kJ/kgK}$ [14]

$$\text{But, } \gamma = \frac{C_p}{C_v}, \text{ therefore: } T_{out} = T_{in} \left(\frac{P_{out}}{P_{in}} \right)^{\frac{\gamma-1}{\gamma}} \quad (4)$$

Where: T_{in} - inlet steam temperature ($^{\circ}\text{C}$), T_{out} - outlet steam temperature ($^{\circ}\text{C}$), γ - index number, C_p - specific heat capacity of steam at constant pressure (kJ/kgK) and C_v - specific heat capacity of steam at constant volume, C_{pm} - mean specific heat capacity of steam ($\text{kJ/mol}^{\circ}\text{C}$), η_t - efficiency of steam turbine; For 5 kW power rating, T_{in} , P_{in} , P_{out} , and η_t are 400°C , 0.1 MPa, 0.45 MPa, and 80% respectively. These were the state properties of superheated steam obtained from [15].

Coulson and Richardson [16] defined the mean specific heat capacity C_{pm} over the temperature range T_1 to T_2 as follow, Eq. (5):

$$C_{pm} = \frac{\int_{T_1}^{T_2} C_p dT}{\int_{T_1}^{T_2} C_p dT} \quad (5)$$

The specific heat capacity (C_p), as a function of temperature, is given by Eq. (6):

$$C_p = a + bT + cT^2 + dT^3 \quad (6)$$

Super heater

Super heater is heat exchanger that transfers heat energy from a heating medium to a heated medium. The heating medium is usually flue gas while the heated medium is steam. The energy balance equation of super heater is:

$$q_{sp} = m_s C_{pm} (T_{sp1} - T_s) \quad (7)$$

Where: q_{sp} - heat duty required by the super heater (J/s), T_{sp1} ($^{\circ}\text{C}$) and T_s ($^{\circ}\text{C}$) are the temperature of superheated and saturated steam from super heater and boiler respectively.

The energy balance equation of the riser tube is given by Eq. (8):

$$q_b = m_s C_p (T_s - T_d + x_s m_s L) \quad (8)$$

Assuming the water from down comer is saturated $T_d = 100^{\circ}\text{C}$, and $T_s = 100^{\circ}\text{C}$; steam is in equilibrium with water in the riser. $T_s = T_d = 100^{\circ}\text{C}$, Therefore Eq. (9):

$$q_b = x_s m_s L \quad (9)$$

Where: q_b - heat duty required by riser tube (J/s), L - specific latent heat of vaporization (2256 kJ/kg), x_g - dryness fraction of steam/water mixture.

$$q_f = m_{fuel} \times LHV \quad (10)$$

Where: m_{fuel} - mass of the fuel (kg/hr), q_f - heat liberated by the fuel (J/s).

$$m_{air} = Air - fuel\ ratio \times m_{fuel} \quad (11)$$

Furnace volume

Chungen [17] documented typical value of volumetric heat release rate q_v , for biomass as 0.176 MW/m³. Similarly, Sebastian (2002) reported furnace strain level largely depends on different fuels and if the electric power of the plant is known, strain levels for volume can be chosen. A large number of package boilers have a cylindrical furnace with truncated cone as shown in Figure 2. The exterior angle of the truncated cone is within 50 to 55°C. The furnace volume V (m³) grate area and furnace height h (m) can be obtained from Eq. (12-14), respectively.

$$V = \frac{q_f}{q_v} \quad (12)$$

$$A = \frac{\pi d^2}{4} \quad (13)$$

$$h = \frac{V}{A} \quad (14)$$

Sizing of riser and super heater

The heat duty required in the riser is given by Eq. (15):

$$q_b = UA\Delta T_{LM} \quad (15)$$

And the overall heat transfer (U) based on the outside area (A) of the riser tube can be estimated as;

$$\frac{1}{U} = \frac{1}{h_g} + \frac{t_r}{k_w} + \frac{1}{h_{b1}} \left[\frac{d_o}{d_i} \right] \quad (16)$$

Where: h_g - heat transfer coefficient of flue gas (kJ/kg), t_r - wall thickness of riser tube (m), k_w - thermal conductivities of stainless steel (304), (W/m°C), d_o (m) and d_i (m) are internal and external pipe diameter, h_{b1} - heat transfer coefficient of water boiling (W/m°C).

The analysis of heat transfer associated with flow past the exterior surface of a solid is a complicated situation due to boundary layer separation [21]. Nusset number can also be used to calculate heat transfer coefficient of flue gas (h_g).

Specific heat capacity, dynamic viscosity and thermal conductivity of flue gas

Verbanck [18] determined specific heat capacity of flue gas (C_{pg}) as the summation of the product of the mass fraction of each component of flue gas m_k (kg) by its respective specific heat at the relevant temperatures c_k (kJ/kg⁰C) as:

$$C_{pg} = \sum m_k c_k \quad (17)$$

$$c_k = A + BT + CT^2 \quad (18)$$

Hassan and Ibrahim [19] stated that heat losses through casings must be accounted for if accurate computation of flame temperature is to be made. This was done by setting up heat balance equation for fuel gas as follows, Eq. (19-21):

$$Q_{combustion} - Q_{loses} = \sum C_{pg} \times m_g \times (T - T_{air}) \quad (19)$$

Where:

$$Q_{combustion} = m_{fuel} \times LHV \quad (20)$$

According to [19],

$$Q_{loses} = 5\% \times Q_{combustion} \quad (21)$$

The dynamic viscosity μ_g (kg/ms) of flue gas is obtained from Eq. (22) by [18]:

$$\mu_g = \sum \frac{\mu_k m_k}{\sqrt{M_k}} \div \sum \frac{m_k}{\sqrt{M_k}} \quad (22)$$

Thermal conductivity of the flue gas k_g (W/m⁰C) is determined from Eq. (23) by [20]:

$$\frac{k_g}{\mu_g c_v} = 1.32 + \frac{1.77 R_g}{c_v} \quad (23)$$

$$C_v = C_{pg} - R_g \quad (24)$$

From Eq. (23) and Eq. (24), we have;

$$k_g = \mu (C_{pg} - R_g) \left[1.32 + \frac{1.77 R_g}{C_{pg} - R_g} \right] \quad (25)$$

Where: R_g - characteristic gas constant of flue gas;

$$R_{CO_2} = 188, R_{O_2} = 260, R_{H_2O} = 462, R_{N_2} = 297, R_{SO_2} = 130,$$

Cv - specific heat capacity at constant volume (kJ/kgK)

Prandtl and Reynolds numbers of the flue gas

The Prandtl number P_r and Reynolds number Re_D of flue gas are given by Eq. (26) and Eq. (27), respectively.

$$P_r = \frac{\mu_g C_{pg}}{k_g} \quad (26)$$

$$Re_D = \frac{m_g d_b}{\mu_g A_{cr}} \quad (27)$$

Where: A_{cr} - cross sectional area of flow of flue gas; D_b - diameter of riser tube (m). For an external cross flow to a cylindrical pipe, the Reynolds number range 40-4000 and $P_r \geq 0.7$, the average corresponding Nusselt number according to [21] is given by:

$$\overline{NU_D} = 0.683 Re_D^{0.466} P_r^{1/3} \quad (28)$$

$$\overline{NU_D} = \frac{h_g D_b}{k_g} \quad (29)$$

The empirical equation proposed by [22] for the calculation of heat transfer coefficient of water boiling (h_g) is as follow:

$$h_g = 2.8 P^{0.176} q^{0.7} \quad (30)$$

Valid at $0.2 \text{ bar} \leq P \leq 98 \text{ bar}$.

Where: $q = \text{heat flux} \frac{W}{m^2} = \frac{q_b}{A_s}$, P : saturated pressure (kPa), A_s - surface area of riser (m^2)

Evaluation of logarithmic mean temperature (ΔT_{LM})

Heat obtained by the riser is the heat given out by the flue gas

$$q_b = m_g C_{pg} (T_{gin} - T_{gout}) \quad (31)$$

For a cross flow heat exchanger, [23] gives the Logarithmic Mean Temperature Difference as follow

$$L_{MTD} = \frac{(T_{gin} - T_{so}) - (T_{gout} - T_{sin})}{\left[\frac{T_{gin} - T_{so}}{T_{gout} - T_{sin}} \right]} \quad (32)$$

Where: T_{gin} – temperature of flue gas in (0C); T_{gout} - temperature of flue gas out (0C); T_{so} – temperature of saturated steam out (0C), T_{sin} – temperature of saturated steam in (0C). From the equation (15) the total heat transfer surface area A (m^2), outside diameter d_o (m) and length of the tube; L (m) is given by;

$$A = \pi d_o L \quad (33)$$

The following procedural steps / algorithm were adopted to design the PKS combustng furnace using backward calculation approach.

1. Calculate the turbine power input from turbine, $P_{turb} = P_{out} + P_{loss}$
2. Mass flow rate of system from super heater, $m_s = \frac{P_{turb}}{C_{pm} \eta_t (T_{in} - T_{out})}$
3. Calculate heat duty required by the super heater, $q_{sp} = m_s C_{pm} (T_{sp1} - T_s)$
4. Calculate heat duty required by the riser tube, $q_b = m_s C_p (T_s - T_d + x_s m_s L)$

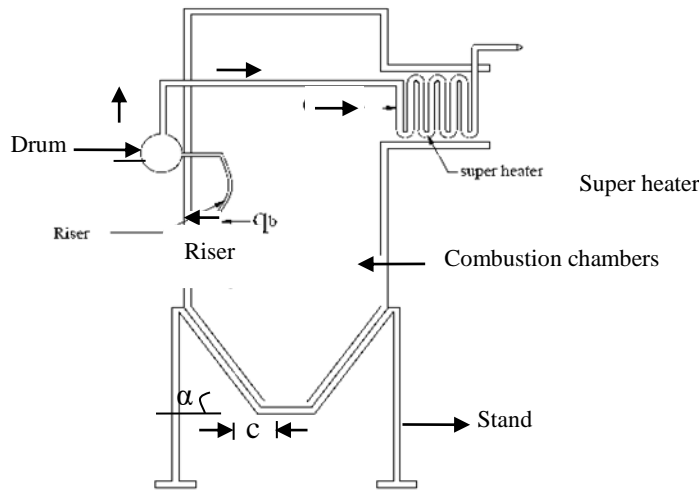


Figure 2. Schematic picture of the furnace

5. Calculate total heat duty required by the combustion chamber, $Q = (q_{sp} + q_b)$
6. Calculate heat liberated from the fuel, $q_f = m_{fuel} \times LHV$

7. Determination of heat losses through wall furnace for an accurate flame temperature.

$$Q_{combustion} - Q_{loses} = \sum C_{pg} \times m_g \times (T - T_{air}),$$

$$Q_{loses} = 5\% \times Q_{combustion}.$$

8. Calculate the volumetric flow rate, $V_{air} = \frac{M_{air}}{\text{density of air}}$

9. Determine the furnace volume, $V = \frac{q_f}{q_v}$, area, $A = \frac{\pi d^2}{4}$ and furnace height, $h = \frac{v}{A}$

10. Calculate the overall heat transfer U, based on the outside area of the riser and super

$$\text{heater, } \frac{1}{U} = \frac{1}{h_g} + \frac{t_r}{k_w} + \frac{1}{h_{b1}} \left[\frac{d_o}{d_i} \right]$$

11. Calculate Prandtl and Reynolds numbers of the flue gas.

$$P_r = \frac{\mu_g C_{pg}}{k_g}, \text{ and } Re_D = \frac{m_g d_b}{\mu_g A_{cr}}$$

12. Calculate the logarithmic mean temperature difference for a cross flow heat exchange,

$$L_{MTD} = \frac{(T_{gin} - T_{so}) - (T_{gout} - T_{sin})}{\left[\frac{T_{gin} - T_{so}}{T_{gout} - T_{sin}} \right]}$$

13. Calculate the heat transfer area of the riser and to the heat duty and overall heat

$$\text{transfer coefficient, } A_s = \frac{q_b}{[V \Delta T_{LM}]}$$

14. Calculate the length of the risers and super heater required to exchange the desire

$$\text{heat. } L = \frac{A_s}{\pi d}$$

Software development

Based on the equations (1-33) an algorithm was prepared and then translated to computer code using C# programming language and .Net framework.

The flow chart upon which the algorithm was based is shown in Figure 3.

The code receives input parameters in order to size components for 5 kW of electricity and gives dimensions of furnace, riser and super heater tube as outputs. In addition, 2D drafting of combusting furnace was done in AutoCAD and was dynamically loaded into the Visual Studio workspace of the application. It should be noted that .Net frame

work was selected because of its ease of deployment, interoperability, automatic management of resources and cross platform support.

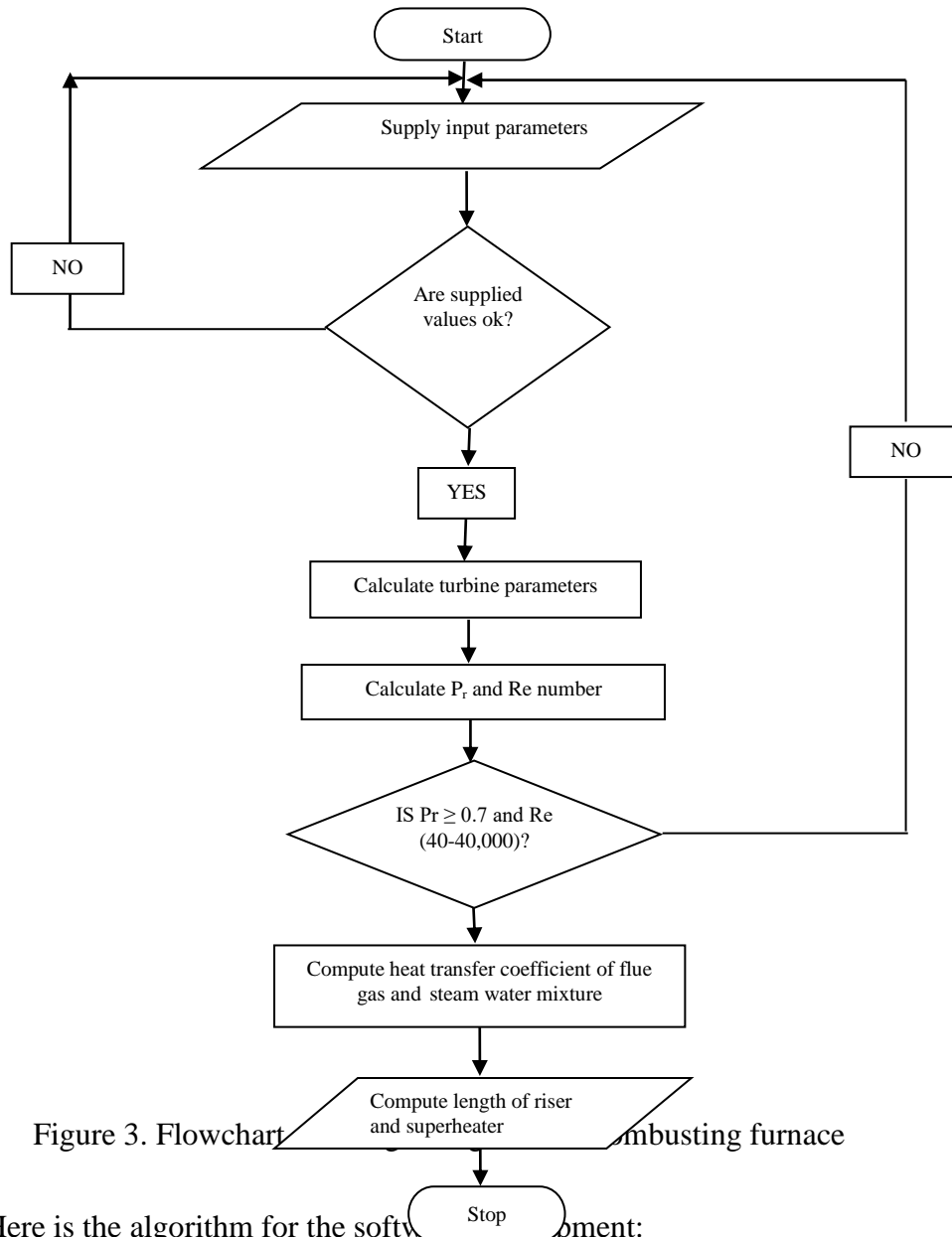


Figure 3. Flowchart

Here is the algorithm for the software development:

- Enter the electrical output power in kW and efficiency of synchronous generator in (%) from design requirement;
- The properties of the PKS based on its ultimate analysis (i.e. carbon, hydrogen, oxygen nitrogen and sulphur) were supplied;
- The mean fraction of the flue gas component m_k and the mean specific heat of the flue gas component c_k available were stored in the software database;
- Submit to compute for turbine parameters;

- (e) Click on further calculations on furnace, heat transfer coefficient of flue gas and heat transferred on super heater and riser tube;
- (f) Submit to compute furnace parameters, convection coefficient and length of super heater and riser tube; When all conditions required for sizing furnace components have been adequately satisfied with respect to the calculation to the six steps above, the design parameters are then used to draft the 2D of the furnace. This is done through the drafting modules wherein the geometries had been mathematically represented within the developed software.

Fabrication and assembly of steam boiler

The components of the boiler developed are super heater, riser, water tank, drum, and furnace chamber. Figure 4 and Table 1 showed the exploded view of PKS combustion unit and their part list. Each of the component's fabrication process, material selection and cost analysis were not reported in this paper.

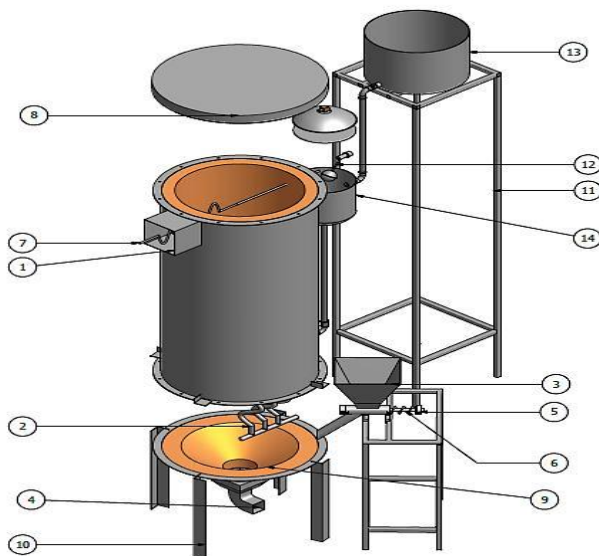


Table 1. Part list of PKS furnace

Item	Qty	Part list
1	1	Superheater exit port
2	3	Riser
3	1	Hopper
4	1	Primary air inlet
5	2	Ball bearings
6	1	Auger
7	1	Superheater
8	1	Furnace cover
9	1	Bricks
10	1	Furnace stand
11	1	Water tank stand
12	1	Water control valve
13	1	Water tank
14	1	Drum

Figure 4. Exploded view of PKS combustive furnace

Results and Discussion

Proximate and ultimate analysis

The proximate analysis of the sample of PKS collected from a local oil palm mill in Iresapa Ogbomoso Southwestern, Nigeria (Table 2) showed moisture content, volatile matter, fixed carbon, and ash of 2.70%, 44.20%, 52.79% and 0.31%, respectively.

Table 2. Proximate Analysis (% by weight on dry basis)

Property	This study	[25]*
Moisture	2.70	5.40
Volatile matter	44.20	71.10
Fixed Carbon	52.79	18.80
Ash	0.31	4.70

Table 3. Ultimate Analysis (% by weight on dry basis)

Property	This study	[25]*
Carbon	45.12	48.06
Hydrogen	10.67	6.38
Nitrogen	0.27	1.27
Oxygen	40.11	34.10
Sulphur	0.62	0.09
LHV (MJ/kg)	17.12	16.30

These results are largely different from those of [25]. It can be seen that this biomass contains low moisture and ash content which resulted to substantial higher heating value of the shell while compared to [25]. Similarly, the ultimate analysis (Table 3) shows that percentage weight of oxygen and hydrogen content in this study are higher while carbon, sulphur and nitrogen content are lower compared to those of [25]. This might be due to the variations in the species, location, soil type, climatic condition of the palm kernel shell sourced. The graphical user interface for sizing turbine parameters is shown in Figure 5.

Figure 5. Template for the Turbine Parameters

The input parameters, which are the output power, efficiency of generator and location of PKS used, are provided.

To obtain furnace parameters, length of riser and super heater tube, 'further calculation' bottom is clicked. For example, to size turbine components for 5 kW of electricity generation, the efficiency of a synchronous generator (90%) is provided and Ogbomoso is selected as the location of PKS used.

Component dimensions based on the developed software

By clicking the 'submit' button, we obtained 5.56 kW of turbine, 0.0275 kg/s of steam entering turbine, 276.090C of outlet steam temperature and 0.03571 kJ/mol0C of mean specific capacity of the steam (Figure 6).

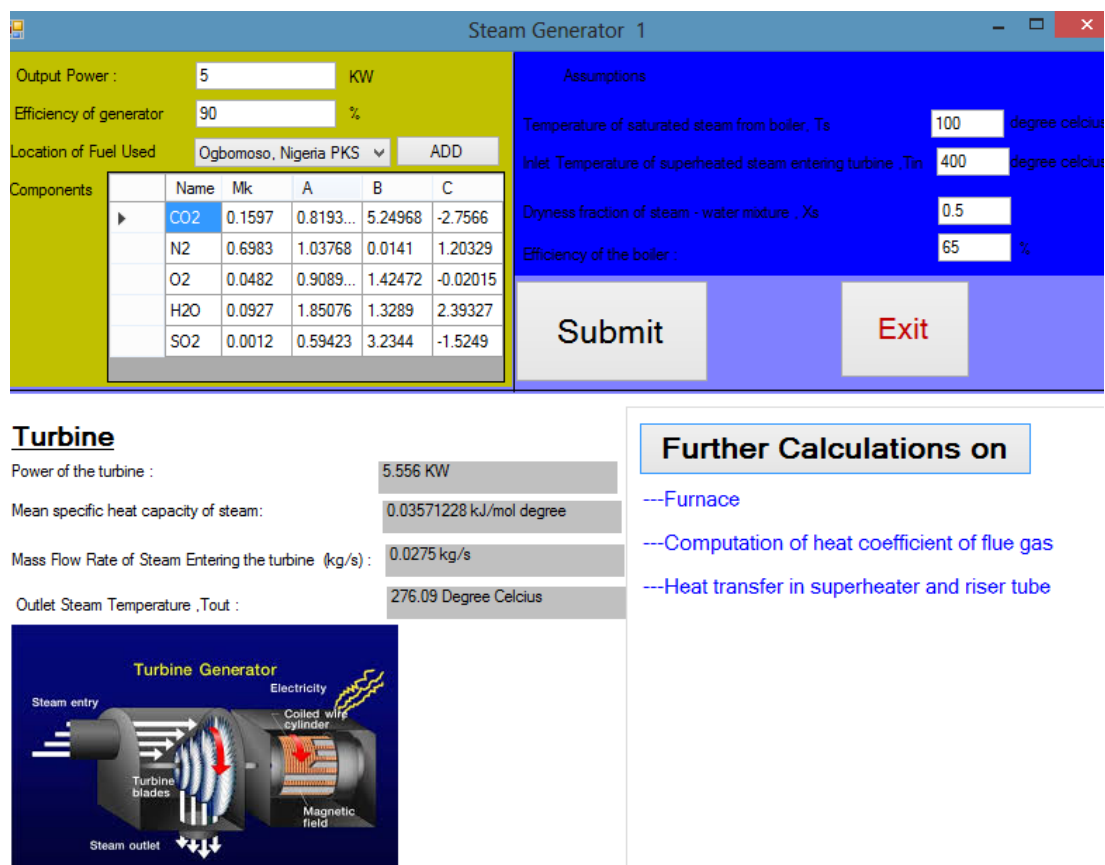


Figure 6. Output Screen for the Turbine Parameters

These values are required to generate 5 kW of electricity. By clicking 'further calculations on furnace' heat transfer coefficient of flue gas and heat transferred on super heater and riser tube, the furnace can be appropriately sized.

The screenshot displays the 'Steam Generator 2' software interface. It is divided into several sections for input and calculation.

Input Parameters (Top):

- Internal Diameter of the furnace: 0.6 m
- External Diameter of the Pipe: 0.025 m
- Prandtl Number, Pr: 0.703
- Diameter of the riser: 0.5 m
- Internal Diameter of the Pipe: 0.023 m
- Reynold number of flue gas: 105.753
- Dimension: 25 mm
- Thickness of the riser: 0.0016
- Average Nusselt Number: 5.331
- Thermal Conductivity of stainless steel: 14
- Heat transfer coefficient of superheated steam (hs): 32
- Temperature of the flue gas: 1214.141 °C

Furnace Section (Left):

- Mass of fuel, Mfuel: 17.3011 kg/hr
- Volume: 0.405 meter cube
- Height: 1.4325 meter
- Grate Area: 0.2827 meter square
- Heat Release per volume (qv): 0.18 MW/meter cube
- Volumetric Air Flow Rate: 0.0376 meter cube/s
- Heavy duty required of superheater tube, qsp: 16.3681 kJ/s
- Heavy duty required of riser, qb: 31.02 kJ/s
- Total heat required in combustion chamber: 47.388 kJ/s

COMPUTATION OF HEAT COEFFICIENT OF FLUE GAS (Right):

- Mean Specific heat of the flue gas, Cpg: 1.289 kJ/kg degree Celcius
- Thermal Conductivity of flue gas, kg: 0.1 W/m degree celcius
- Dynamic Viscosity of the flue gas, μg: 5.43769129694058E-05
- Heat Transfer Coefficient of flue gas, hg: 21.266 W / m² degree

Heat Transferred In Riser Tube (Bottom Right):

- Overall heat transfer coefficient: 21.187
- Logarithmic mean temperature difference: 819.237 °C
- Heat Transfer Surface Area: 1.987 m²
- Lenght Of Riser: 3.891 m

Heat Transferred In Superheater Tube (Bottom Left):

- Overall heat transfer coefficient: 12.419
- Logarithmic mean temperature difference: 823.609 °C
- Heat Transfer Surface Area: 1.6 m²
- Lenght Of Superheater: 3.134 m

Navigation Buttons:

- Previous
- Submit
- Go To Design Schematic Diagram
- Exit

Figure 7. Output Screen for the Furnace parameters and Length of riser and super heater tube

For 5 kW; 17.36 kg/hr of fuel, 0.405 m³ volume, 1.432 m height and 0.0376m³ volumetric air flow rate are needed (Figure 7).

The equivalent designed PKS combusting furnace is shown in Figure 8. The software can be used to size furnace for generating specified power.

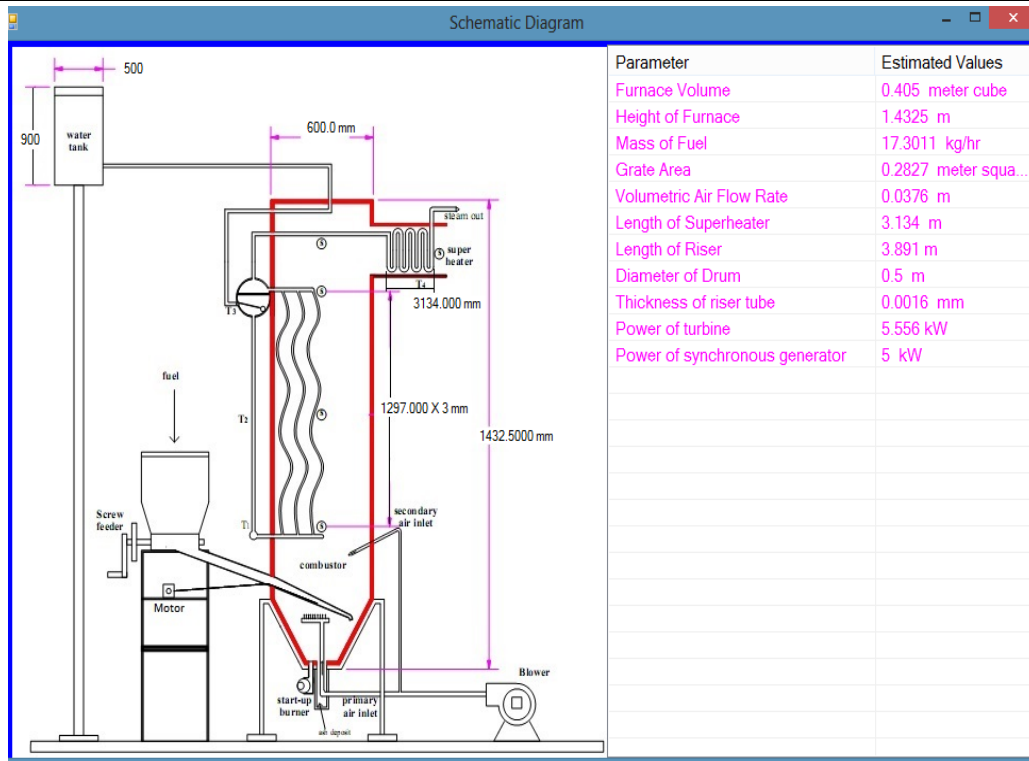


Figure 8. Output Screen Schematic Drawings and Estimated Values of (5 kW) PKS Combustor

Table 4. Comparison of Parameters Based on Algorithms Generated and Traditional Calculations

Parameters	Traditional Calculations	Based on Algorithm Generated
Power of the turbine (kW)	5.510	5.500
Mass flow rate of steam entering turbine (kg/s)	0.028	0.028
Outlet steam temperature ($^{\circ}\text{C}$)	276.10	276.05
Mean specific heat capacity of steam ($\text{kJ/mol}^{\circ}\text{C}$)	0.0362	0.0357
Mass of fuel (kg/hr)	17.300	17.301
Volume of furnace (m^3)	0.414	0.405
Grate Area (m^2)	0.283	0.287
Height of furnace (m)	1.4325	1.4325
Volumetric air flow rate (m^3/s)	0.035	0.035
Diameter of drum (m)	0.500	0.500
Thickness of riser tube (m)	1.6	1.6
Heat duty of riser tube (kJ/s)	31.01	31.02
Heatduty required of superheater tube (kJ/s)	16.461	16.368
Total heat required in combustion chamber (kJ/s)	47.349	47.388
Length of riser tube (m)	3.782	3.891
Length of superheater (m)	3.135	3.135

To validate the accuracy of the developed software, we compared the results obtained based on algorithms generated and traditional calculations. The results are found to be very similar (Table 4). In terms of time saving, traditional calculations and drafting of furnace

components details took about 5 hrs. 47 minutes while the same process was completed in 4 minutes when the algorithm generated was used. In addition, inaccuracies due to human errors are virtually eliminated.

Conclusion

Traditional calculations and drafting of furnace components details took about 5 hrs. 47 minutes while the same process was completed in 4 minutes when the software was used. For a fuel feed rate of 17.3 kg/hr. and volumetric air flow rate of 0.035 m³/s; the power of the turbine, volume of furnace, length of super heater, and risers tubes required for 5 kW power rating were 5.5 kW, 0.405 m³, 3.89 m and 3.13 m respectively.

Reference

1. Sulaiman N., Abdullah H., Gerhauser A. S., *An outlook of Malaysian energy, oil palm industry and its utilization of wasted as useful resources*, Biomass and Bioenergy, 2011, 35, p. 3775-3786.
2. Izah S., Ohimain E., Angaye T., *Potential Thermal Energy from Palm Oil Processing Solid Wastes in Nigeria: Mills Consumption and Surplus Quantification*, British Journal of Renewable Energy, 2016, 1, p. 39-45.
3. Muhammad A., Tjahjono H., Meta R., *Analysis of Palm Biomass as Electricity from Palm Oil Mills in North Sumatera*, Energy Procedia, 2014, 47, p. 166–172.
4. Sjaak V., Jaap K., *Handbook of biomass combustion and co-firing*, 2008, ISBN-13:978-1849711043.
5. Pichet N., Vladimir I., *Combustion of palm kernel shell in a fluidized bed: Optimization of biomass particle size and operating conditions*, Energy conversion and management, 2014, 54 (01), p. 1–9.
6. Najmi W. M., Rosil A. N., Izat M. S., *Combustion Characteristics of Palm Kernel Shells Using an Inclined Grate Combustor*, Journal of Faculty of Mechanical Engineering UiTM Malaysia, 2007, p. 15–28.
7. Thomas R., Jabouille F., Torero J. L., *Effect of excess air on grate combustion of solid wastes and on gaseous products*, International Journal of Thermal Sciences, 2009, 48, p.165–173.

8. Liang W., Lovås T., Houshfar E., *Effect of Sewage Sludge Addition on Potassium Release and Ash Transformation during Wheat Straw Combustion*. available from <http://www.aidic.it/cet>. (Accessed 10 March, 2013) 2014,37, p. 7–12.
9. Kamoru O. O., Buliaminu K., Basil O., and Abass O. A., *Optimization of Ash Yield from the Combustion of Palm Kernel Shell and Selected Additives (Al_2O_3 , CaO , MgO) Using D-Optimal Design*, Leonardo Electronic Journal of Practices and Tecchnologies, 2016, 28, p. 9-8.
10. ASTM 3174-76 *Standards method of proximate analysis of coal and coke, in gaseous fuels; coal and coke section 5*, Annual Book of ASTM Standards, 2001, 5, p. 299.
11. ASTM E711-87 *Standards test method for gross calorific value of coal and coke using GallenKamp bomb calorimeter, in gaseous fuels; coal and coke, section 5*, Annual Book of ASTM Standards 2001, 5, p. 25.
12. Astrom K., J. Bell R. D., *Drum-boiler dynamic*, Automatica 2000, 36 p. 363-378.
13. Chaibakhsh A., Ghaffari A *Simulation Modelling Practice and Theory Steam Turbine Model*, Simulation Modelling Practice and Theory, 2008, 16 p.1145–1162.
14. Kyle B.G., *Chemical and Process Thermodynamic 3IE Adapted by Permission of Pearson Education, Inc.*, Upper Saddle River, www.cen84959_ch18_ap01.qxd (Accessed 15 April, 2013), 2000.
15. Steam Turbine V-FLO Pumps and System available from www.VfLO.Com (Accessed 18 June, 2013), p. 5-8.
16. Coulson R., *Chemical Engineering Design fourth edition*, 2005, 6, p. 634-693.
17. Chungen Y., Rosen-dahl L., Kær S. K., *Grate-firing of biomass for heat and power production*, Progress in Energy and Combustion Science, 2008, 34, p. 725–754.
18. Verbanck, *Development of a mathematical model for water tube boiler heat transfer calculations*, Proc S Afr Sug Technol Ass, 1997 p. 166-167.
19. Hassan A., Ibrahim A., *Calculation of Radiant Section Temperatures in Fired Process Heaters*, Scientific Research to Knowledge, 2013,(5)1 p. 1–6.
20. Eucken M., *A model maxwell-Eucken equation for calculating the thermal conductivity of two component solutions or mixtures*, Journal of Refrigeration, 2005, 4, p. 223-225.
21. Illinois Institute of Technology (IIT), *Convection Workshop Academic Resource Centre*, Available from [http:// www.iit.edu](http://www.iit.edu). (Accessed 10 January, 2014), 2013.

22. Kandlikar, Penniger K., *A General Correlation for Saturated Two-Phase Flow Boiling Heat Transfer Inside Horizontal and Vertical Tubes*, Heat Transfer, 2006,112, p. 219–228.
23. Sebastian T., *Thermal Design of Heat Exchangers*, Energy Engineering and Environmental Protection, 2002, p. 4–9.
24. Rajkumar T., Ramaa V. M., Gobi K., *Boiler drum level control by using wide open control with three element control system*, Journal of Research in Management and Technology, 2013, 11, p. 85-96.
25. Ninduangdee P., and Kupranov V. I., *Fludised-Bed Combustion of Biomass with Elevated Alkali Content: A Comparative Study between Two Alternative Bed Materials*, International Journal of Chemical, Nuclear, Metallurgical and Materials Engineering, 2014, 8, p. 267-274.