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# Design of a Biomass Burner/Gas-to-gas Heat Exchanger for Thermal Backup of a Solar Dryer

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**Abstract:** The solar drying process is interrupted during the cloudy or rainy days and also at night. Thus, a combination of mixed mode dryer using solar as main input and biomass burner as auxiliary source of thermal energy will compensate for the absence of the solar radiation. This study describes a detailed design procedure of a thermal back up unit which is flexible to supply hot flue to dry solid waste or to supply warm dry air to dry food and fish. The unit consists of biomass burner and gas to gas heat exchanger topping the burner. The design calculations are based on requirements to dry 2.5 kg of palm fiber (EFB). A mathematical model was developed to calculate the geometries of the heat exchanger. The model is based on the energy balance, convection heat transfer correlations and suitable initial assumptions. The material selection criterion is presented since it is crucial for the success of the design. The fabricated unit is tested experimentally and the measurement results are in good agreement with the basic design assumptions.

**Key words:** Solar dryer, Gas-to-gas heat exchanger, biomass burner, thermal backup

## INTRODUCTION

Solar and biomass are the two main renewable energy sources extremely suitable for drying application. Since the availability of solar is limited only during the sunny days, the uses of biomass burner as a backup heater is highly preferable. Many agriculture producers nowadays are applying a solar dryer assisted by electricity or biomass fuel to extend the period of drying after sunset and during the cloudy days, as well as drying large capacity of products. Conventional fuel operated driers are more efficient but it is beyond the reach of rural people with limited product (Prasad *et al*., 2006). High prices and shortages of fossil fuels have increased the emphasis on using alternative renewable energy resources (Muhlbauer, 1986).

Biomass fuel is suitable to be applied in burner as a backup heater in drying application. Biomass can be described as "stored solar energy" (www.aesenergy.net). Biomass sources include food crops, grassy and woody plants, residues from agriculture or forestry, organic components of municipal and industrial wastes and animal waste. The combustion products from complete combustion of biomass generally contain nitrogen, water vapor, carbon dioxide and surplus of oxygen. If there is a surplus of solid fuel due to incomplete combustion, the products of combustion are combustible gases like Carbon monoxide, CO, Hydrogen, $H_2$ and traces of Methane and non-useful products like tar and dust (Rajvanshi, 1986). These combustible gases are not suitable for drying food application. So the usage of the gas to gas heat exchanger (G-to-G HEX) is crucial to make sure that the gas that will be used for food drying application is clean, free from smoke, soot and ash in order to protect the food from being contaminated. The biomass fuel combustion heated the wall of the G-to-G HEX and allowed the transmission of the heat to the ambient air, resulting in temperature raise. This process produces a heated clean air for food drying application.

A compact G-to-G HEX needs large heat transfer areas on both fluid sides which can be done by adding fins. The fins extend the heat transfer surfaces and promote turbulence (Wang *et al*., 1999). The higher the thermal conductivity of the material used, the higher the heat transferred across the heat exchanger. A lot of studies which focused on the design of burner with heat exchanger have been done by previous researchers. Fang (2000) studied on a clamshell heat exchanger in residential gas furnace. The clamshell is created by joining two steel panels containing half channel in a manner similar to joining the two halve of a sandwich. The half channels are stamped or molded into the panels forming complete gas channels at assembly to guide the flow of combustion gases inside the channel. The furnace heat

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exchanger separates the high temperature flue gas stream from the low temperature circulating air stream while at the same time transferring thermal energy from the former to the latter. The heat exchanger is made of aluminized steel. The density increases and its volumetric flow rate decreases as the flue gas is cooled while travel downstream.

Tomimura *et al.* (2004) studied on a multi layered type of gas to gas heat exchanger using porous media. The porous metal plate used in this process is extremely high porosity material and give a large temperature drop and larger amount of converted radiant energy. This type of heat exchanger had higher overall heat transfer coefficients then the conventional heat exchanger and also exhibit excellent reaction characteristics as a steam reformer. The enthalpy of the high temperature gas is effectively transferred to the porous metal plate via extremely high heat transfer coefficient between the flowing gas and the porous plate with fine mean pore size, at the same time a substantially large surface area of the porous for heat transfer.

Al-Omari (2006) focused on a biomass furnace design for experimental and investigation on combustion and heat transfer characteristics. A digital weighing scale is used to monitor and record the fuel mass in the bed during the combustion process. LPG combustion was found to be more convenient and effective to initiate the combustion around the 2 to 3 min which is stabilized and guided by means of conical bluff body.

The use of biomass burner as back up heater is relatively synonym in the industry of drying. A suitable design of controllable biomass burner is important so that the required heat is supplied to dryer without affecting the quality of the product to be dried. Madhlopa and Ngwalo (2007) studied on an indirect solar dryer with biomass backup heaters. The biomass burner was made of brick and consists of rock pebbles which acted as a thermal storage. Thanaraj *et al.* (2004) came out with a furnace which consists of heat exchanger using bricks, clay and cement to the rotary dryer. The same material type of burner also has been reported by Prasad (2006) Prasad and Vijay (2005), Tarigan and Tekasakul (2005) and Bena and Fuller (2002). Ocused on a solar hybrid tunnel dryer incorporated with a biomass stove-heat exchanger, consists of a cross-flow shell and tube heat exchanger. Serafica and del Mundo (2005); (Bhattacharya *et al.*, 2000) focused on a biomass gasifier stove design as a backup heater to the hybrid solar dryer for fish and fruits and vegetables, respectively. The biomass gasifier consists of shell and fin heat exchanger configuration and the heat delivery and combustion rate could be controlled using a butterfly valve at the primary air inlet. Among the biomass fuel materials that has been reported in biomass burner application are coconut shells (Serafica and del Mundo 2005), woodchips (Bhattacharya *et al.*, 2000; Madhlopa and Ngwalo, 2007), charcoal (Prasad and Vijay, 2005), paddy husk (Thanaraj *et al.*, 2004), fuel wood (Prasad *et al.*, 2006; Bena and Fuller, 2002; Tarigan and Takasakul, 2005) and briquetted rice husk.

The El Paso Solar Energy Association in 2010 provides a basic guidelines to dry food where the temperature ranges between 37 to 71°C will effectively kill bacteria and inactivate enzyme although temperatures around 43°C are recommended for solar dryers and aims to remove 80 to 90% of moisture from the food. The allowable maximum temperature of heat under solar or biomass burner supplied for most of the tropical fruits, vegetables and also fish (Serafica and del Mundo, 2005) drying is about 70°C. For safe storage, crops usually dried to a final moisture content of < 14% with equilibrium moisture content = 14% and RH of 80-90% is preferred (Ayensu, 1997).

The present study is documenting the design procedure of burner/gas-to-gas heat exchanger as a backup heater for solar drying.

## DESIGN OF THE DRYING SYSTEM

The flow diagram of biomass burner with gas-to-gas heat exchanger is shown in Fig. 1. The unit is uniquely designed to meet the drying requirements.

To dry food, a heat exchanger is needed so that the flue gas can be separated from the clean warm air to protect the food from contamination by the smoke, soot and ash. Waste drying product does not require any specific temperature limit and quality control, thus the heat from direct fuel can be used as the source of heat. The material to be dried is located inside the drying chamber of solar dryer. The maximum allowable

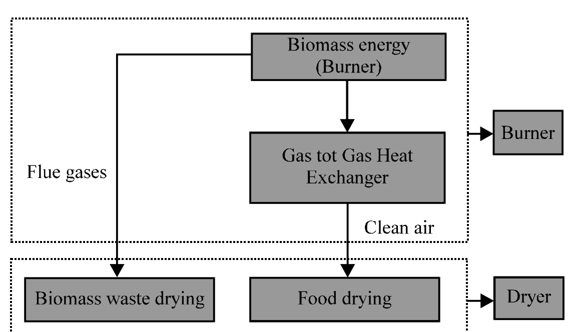


Fig. 1: Biomass burner with gas to gas heat exchanger

Table 1: Conceptual design condition

| Solar dryer system | Average recorded temperature (°C) |
|---|---|
| $T_{inlet/ambient}$ | 30 |
| $T_{absorber}$ | 76 |
| $T_{1/near\ absorber}$ | 52 |
| $T_{chamber}$ | 49 |
| $T_{2/outlet}$ | 35 |

temperature in the drying chamber either under solar or heat from burner is 65°C. This temperature is selected based on the studies of solar drying and biomass backup heater design journals and furthermore, it is the suitable drying temperature for all types of product.

The average measured temperature of the solar dryer system under solar mode for 8 h day$^{-1}$ is shown in Table 1.

Based on Table 1, the recorded drying temperature inside the drying chamber was 49°C. In order to increase the rate of drying and dryer efficiency, the use of biomass burner as a backup heater is favorable to maintain temperature inside the drying chamber within the range of 50 to 65°C.

## MATERIALS AND METHODS

**Conceptual design:** Basically, the unit consists of three zones. The lower zone is for the solid biomass burning. The upper part is the gas-to-gas heat exchanger. The upper part has a cylindrical shape compromises two zones; inner cylindrical hot zone and outer annulus cold zone. The flue gas flow up in the inner zone and the cold air flow up in the outer zone, similar to parallel double pipe HEX. The thermal process in the unit is that the burned biomass produces hot flue. Previous measurements have shown that the flue temperature, $T_f$ is around 320°C. The flue moves up within the inner cylindrical passage by natural draft assisted by a chimney. During its up flow, the hot flue exchanges heat with air. Eight holes with 25.4 mm drilled on the outer surface to permit ambient air to flow to the outer zone of the G-to-G HEX. The air enters the annulus outer zone of the heat exchanger at ambient temperature (~30°C) and moves up due to buoyancy effect assisted by a chimney. To enhance the heat transfer process, 8 extended surfaces have been added in both sides of the cylinder. The heat required, Q obtained from the dryer design is used in order to design the heat exchanger inside the burner.

**Detailed calculation procedure:** Among the steps involved are finding of thermal flue coefficient, $h_{flue}$ to be applied in the thermal balance equation, finding of thermal air coefficient, $h_{air}$, deliberating the amount of mass flow rate from the dryer into design calculation of the burner and finally, comparison of the calculated length characteristic, $L_c$ by iteration where the parameter setting

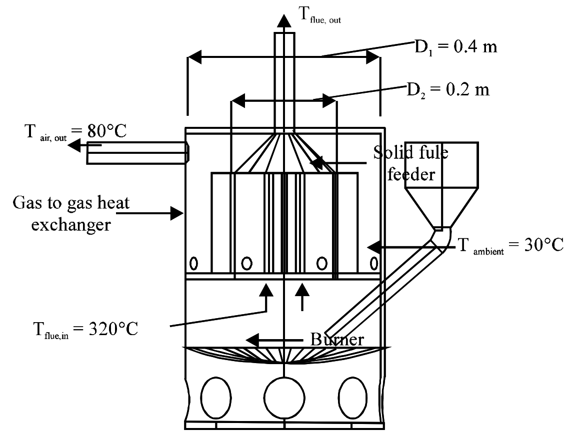


Fig. 2: The preliminary unit design and the parameter assumption

of error is 0.001. The overall design of the burner is shown in Fig. 2.

Equations 1 and 2 are used to calculate the heat supplied from the heat exchanger and the outlet temperature of flue respectively:

$$Q_{burner} = \dot{m}_a c_{pa} (T_{ao} - T_a) \tag{1}$$

where, $\dot{m}_a$ is the mass flow rate obtained from dryer (kg s$^{-1}$), $c_{pa}$ is the specific heat capacity of air (J kg$^{-1}$.K), $T_{ao}$ is the outlet air temperature (K) and $T_a$ is the ambient temperature (K).

In the flue flow side,

$$Q_{burner} = \dot{m}_f c_{pf} \Delta T_f = \rho_f A V_f c_{pf} (T_{fi} - T_{fo}) \tag{2}$$

where, $p_f$ is the density of flue (kg m$^{-3}$), A is the area of heat exchanger (m$^2$), $V_f$ is the velocity of flue (m s$^{-1}$), $c_p$ is the specific heat capacity of flue (J kg$^{-1}$.K), $T_{fi}$ is the inlet flue temperature (K) and $T_{fo}$ is the outlet flue temperature (K).

The type of flow for flue is determined by calculating Re number as in Eq. 3:

$$Re = \frac{\rho_f V_f D_2}{\mu_f} \tag{3}$$

Where, if:

- Re<2300 = Laminar flow
- Re>2300 = Turbulent flow

The Nu equation is basically depends on the mode of convection heat transfer, whether it is natural or forced. This is based on ratio of Gr/Re$^2$.

The nature of convection heat transfer can be determined:

- $0.1<Gr/Re^2<10$ = Natural convection+Forced convection, i.e. combined
- $Gr/Re^2<0.1$ = Forced convection
- $Gr/Re^2>10$ = Natural convection

$Gr_L$ is determined between the difference of mean flue temperature, $T_f$ and inside wall temperature, $T_{wi}$ of heat exchanger using Eq. 4.

$$Gr_L = \frac{g\beta\cdot(\bar{T}_f - T_{wi})D_2^3}{v^2} \tag{4}$$

The combine Nu is evaluated by Eq. 5.

$$Nu_{combine} = \left(Nu_{forced}^3 + Nu_{natural}^3\right)^{1/3} \tag{5}$$

For natural convection (6) is used, as:

$$Nu = CRa_L^n \tag{6}$$

For laminar, C = 0.59, n = ¼; and for turbulent, C = 0.10, and n = 1/3. Rayleigh number, $Ra_L$ is

$$Ra_L = Gr_L Pr \tag{7}$$

where, Pr is the Prandtl number.

For forced convection and laminar flow, the Nu is evaluated by two assumed cases. First assuming fully developed flow of the flue; then Nu = 3.66. For the case of considering the flow at entrance conditions, the suitable correlation for Nu number for entry length is obtained from Eq. 8 and 9. For turbulent, fully developed forced convection, the well known Dittus-Boelter equation has been used:

$$Nu = 1.86\left(\frac{RePr}{L_t}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \tag{8}$$

$$L_t = 4.4DRe^{1/6} \tag{9}$$

The convection heat transfer coefficient for flue, $h_{flue}$ is then calculated using (10):

$$h_{flue} = \frac{Nu_{combine}k_f}{D_2} \tag{10}$$

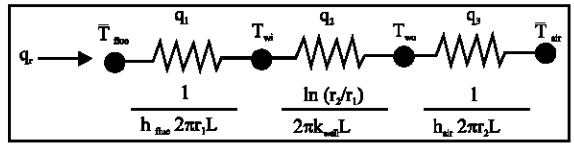


Fig. 3: Heat transfer network from the flue to the air

The illustration of heat transfer across the wall of heat exchanger is shown in Fig. 3.

The characteristic length, $L_c$ or height of the heat exchanger is obtained by deliberating the same amount of the calculated heat required from dryer.

$$Q_{dryer} = Q_{system} = q_1 = q_2 = q_3 \tag{11}$$

The heat transfer from the flue to the inside wall:

$$q_1 = h_{flue}A_f(\bar{T}_f - T_{wi}) \tag{12}$$

Across the wall of heat exchanger:

$$q_2 = \frac{2\pi Lk_{wall}\left(T_{wi} - T_{wo}\right)}{\ln\frac{r_o}{r_i}} \tag{13}$$

And from the outside wall of the cylinder to air:

$$q_3 = h_{air}A\left(T_{wo} - \bar{T}_{air}\right) \tag{14}$$

where, $A_f$ is the flue side thermal area ($m^2$), A is the air side thermal area ($m^2$), $h_{air}$ is the air convection coefficient ($Wm^{-2}.K$), $h_{flue}$ is the flue convection coefficient ($Wm^{-2}.K$), $r_o$ is the outer radius of cylinder (m), $r_i$ is the inner radius of cylinder (m), $T_{air}$ is the mean air temperature (K), $T_f$ is the mean flue temperature (K) and $T_{wo}$ is the wall out temperature (K).

The steps of procedure are repeated again using the previous calculations across the wall of heat exchanger. The type of flow for air is determined by calculating Re number as shown in Eq. 15 using properties at air temperature.

$$Re = \frac{\rho VD_h}{\mu} \tag{15}$$

$$V = \frac{Q_{air}}{A} \tag{16}$$

where, V is the mean air velocity ($m\ s^{-1}$) and $D_h$ is the hydraulic diameter (m).

$Gr_L$ number for air is calculated using (17) between $T_{wo}$ and $T_{air}$. The calculation is followed by finding of the nature of convection heat transfer based on $Gr/Re^2$.

$$Gr_L = \frac{g\beta\left(T_{wo} - \bar{T}_{air}\right)L_c^3}{v^2} \quad (17)$$

where, $h_{air}$ is obtained by substituting the values Nu which depending on the type of flow into Eq. 10 by using properties of air annulus hydraulic diameter. The Rayleigh number, $Ra_L$ is obtained from Eq. 7 which involves $Gr_L$ from Eq. 4 and Pr of air.

By adapting the same criteria of $0.1 < Gr/Re^2 < 10$, the convection heat transfer mode is checked and the relevant Nu equation is used to calculate $h_{air}$. For forced convection heat transfer, Dattus-Boelter relation is used as shown in Eq. 18. For the case of natural convection, the relation in Eq. 6 is used (Incropera, 2007).

$$Nu = 0.032Re^{0.8}\,Pr^n \quad (18)$$

where, n = 0.3 for cooling and n = 0.4 for heating. If the case is combined, thus the Nu is evaluated as in Eq. 5.

The total area, A of heat exchanger is calculated using Eq.19. Since the temperature distribution along the length of outer fins, $L_f$ is not constant, thus it is divided by 2.

$$A = A_{cylinder} + A_{fins} \quad (19)$$

where: $A_{cylinder} = L_c*\pi*D_2$ , and $A_{fins} = n*L_c*(L_f/2)*2$

Finally, the new length characteristic, $L'_c$ is calculated by using the same Q calculated from dryer shown in Eq. 20 followed by iteration until it reached the parameter setting of error, 0.001. The purpose of iteration is to find the corrected value based on the value of initial assumption.

$$L'_c = \frac{Q}{h_{air} \times A \times \left(T_{wo} - \bar{T}_{air}\right)} \quad (20)$$

**Material selection of the biomass burner:** The burner consists of cylinder which is the main part of the gas heat exchanger. It has a function as a major heat transfer surface, and also as a barrier to prevent flue gas to mix with the clean ambient air. Since the burner will experience a high combustion temperature, the selected material must have a good characteristic in the required criteria, which are melting point, corrosive, assembly easiness, thermal conductivity and price.

Table 2: Gas to gas heat exchanger matrix decision table

| | A | B | C | D | E | Total |
|---|---|---|---|---|---|---|
| Pure copper | 5 | 3 | 2 | 5 | 3 | 18 |
| Low carbon steel (AISI 1010) | 4 | 3 | 5 | 2 | 4 | 18 |
| Aluminum | 4 | 5 | 3 | 3 | 2 | 17 |
| Stainless alloy (AISI 304) | 4 | 5 | 5 | 1 | 4 | 19 |
| Titanium | 5 | 5 | 3 | 2 | 1 | 16 |

Criteria : A: Melting point; B: Corrosive; easiness; D: Thermal conductivity; E: Price choice : 1: Very bad; 2: Medium; 3: Good; 4: Very good; 5: Perfect

For the design, five types of material are consider reliable for the process. Table 2 represents the decision matrix leading to the final choice. All the materials are rating between 0 and 5 to each cell of the table, bound to the influence of the corresponding.

According to Table 2, material that has been selected for the burner is stainless alloy, AISI 304 with thermal conductivity, k of 14.9 W $m^{-1}$. K. Even though the thermal conductivity is low, it the most suitable type of material which can extend high combustion temperature, easy to fabricate and excellent for joining (welding).

## RESULTS AND DISCUSSION

**The final design of the unit:** The final boundary conditions of the unit design are shown in Table 3. The mathematical modeling for the governing equations for the unit designed has been converted into programming language which is MATLAB. The programming is divided into two parts, the flue side is to calculate $h_{flue}$ and the air side across the heat exchanger is to calculate the correct height, $L'_c$. The execution of the program involved many iteration processes.

The initial characteristic length, $L_c$ is assumed to be 0.4 m and the finalize length is obtained from the iteration until the difference of error between the new and old length reaching approximately to 0. The result of iteration is shown in Fig. 4. The calculated $L_c$ that obtained at final iteration was 0.33 m.

The calculated heat required, Q from dryer design was 154 Watt and this value is used by deliberating with burner design calculation.

Table 3: Boundary conditions of the unit

| Items | Value |
|---|---|
| Mean air temperature ($T_{air}$) | 55 °C |
| Heat required, Q (obtained from dryer) | 154 Watt |
| Volumetric flow rate (from dryer) with SF 1.2 | 10 $m^3$ $h^{-1}$ |
| Velocity of flue inside burner ($V_f$) | 0.5 m $s^{-1}$ |
| Burner diameter ($D_1$) | 0.4 m |
| Heat exchanger diameter | 0.2 m |
| Number of fins (n) | 8 |
| Length of fins ($L_f$) | 0.08 m |
| AISI 304 thermal conductivity (k) | 14.9 W $m^{-1}$.K |
| Assumed initial length characteristic ($L_c$) | 0.4 m |

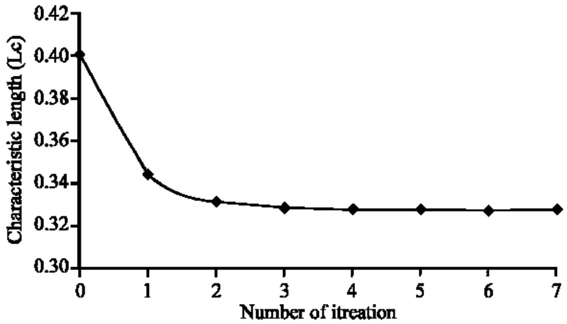


Fig. 4: Characteristic length, $L_c$ versus number of iteration

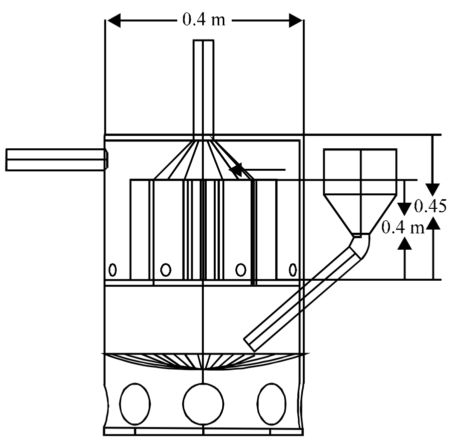


Fig. 5: The designed length of the heat exchanger inside the burner

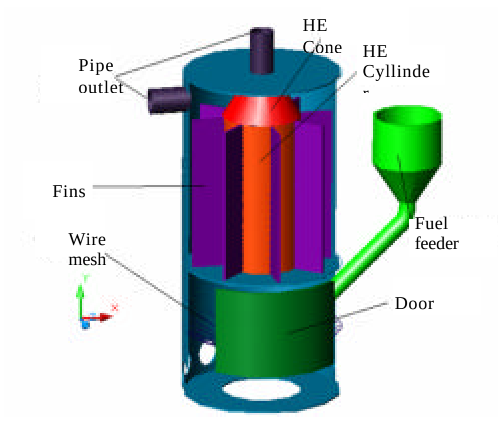


Fig. 6: The components of the biomass burner

The convective flue heat transfer coefficient, $h_{flue}$ calculated from the first part was 3.41 W $m^{-2}$.K. In the second part, the calculated Re was 319.17 which indicate that the flow is laminar with air convective heat transfer coefficient, $h_{air}$ of 4.78 W $m^{-2}$.K.

The result of design calculation of the burner at final iteration is shown in Table 4. The value of $h_{flue}$ obtained from calculation was compared with the typical values of

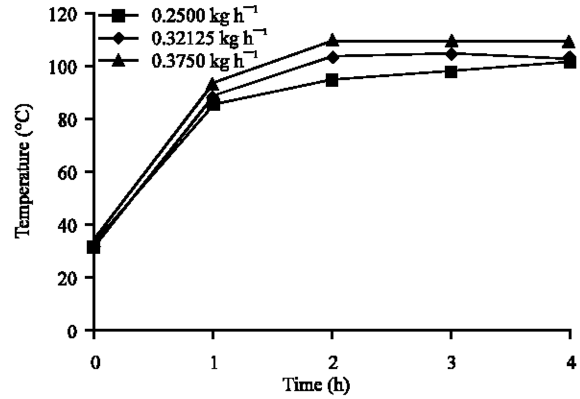


Fig. 7: The measured air temperature obtained from different burning rate of wood chip

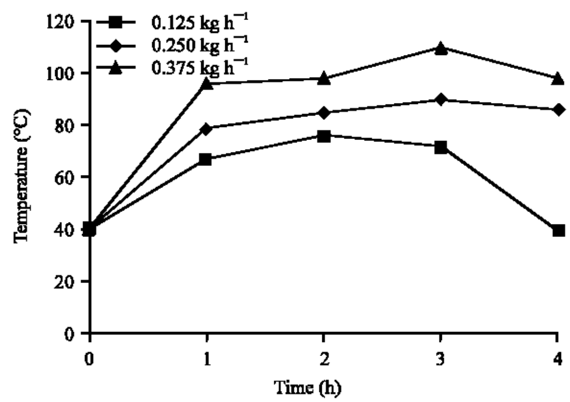


Fig. 8: The measured air temperature obtained from different burning rate of rice husk

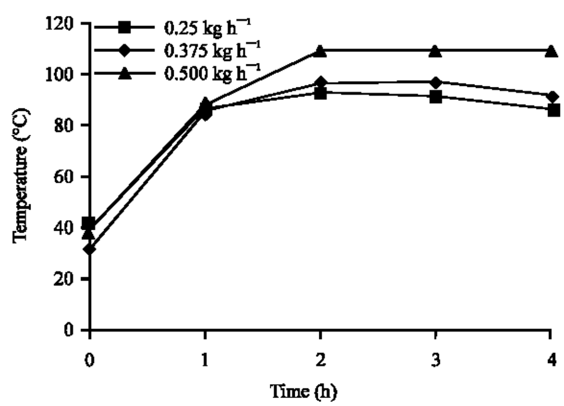


Fig. 9: The measured air temperature obtained from different burning rate of EFB

the convection heat transfer coefficient for gases (Incropera, 2007). For free convection of gases, the range is from 2 to 25 W $m^{-2}$.K which means that the calculated value of $h_{flue}$, 3.41 W $m^{-2}$.K is valid. The final $L_c$ is to be

Table 4: Result of burner design calculation

| Items | Value |
|---|---|
| Inner wall temperature, ($T_{wi}$) | 132.5 °C |
| Outer wall temperature, ($T_{wo}$) | 132.46 °C |
| Convective heat transfer coefficient, ($h_{flue}$) | 3.41 W $m^{-2}$.K |
| Heat from burner, ($Q_{burner}$) | 154 Watt |
| Hydraulic diameter for circular tubes, ($D_h$) | 0.19m |
| Reynolds number, (Re) | 319.17 (Laminar) |
| Grashof number, (Gr) | 1.295 x $10^8$ |
| Ratio ($Gr/Re^2$) | 1272 (Natural convection) |
| Nusselt number, (Nu) | 57.98 |
| Rayleigh number, (Ra) | 9.327 x $10^7$ |
| Convective heat transfer coefficient, ($h_{air}$) | 4.78 W $m^{-2}$.K |
| Characteristic length, ($L'_c$) | 0.3273 m |

multiply by safety factor of 1.2 to compensate for the losses to the exchanger surroundings. Thus, the final $L_c$ calculated for the design of burner is 0.4 m. The illustration of the final length and the components of the unit are shown in Fig. 5 and 6, respectively.

Validation test of the design procedure is carried out by experimental measurements. Various types of solid fuel have been used to compare and select the most suitable type for the drying requirements. Wood chips, rice husk and EFB of the palm oil have been fed at various rate, kg $h^{-1}$. The measurement results are shown in Fig. 7, 8 and 9 for wood chip, rice husk and EFB, respectively.

The analyses of measurement results are demonstrating that the designed and fabricated unit is capable to produce the required floe rate and temperature of the hot to conduct the drying. Rice husk shows lower performance, but still within the required range of drying air temperature. The tested feeding rates of the three solid biomass fuels produced higher than the air outlet design temperature, which is 80°C. this means, that smaller feeding rate is sufficient to provide the required drying hot air.

## CONCLUSION

A thermal unit is designed and fabricated to backup a solar dryer. The unit comprises of two main parts, biomass burner and gas-to-gas heat exchanger. The conceptual design and the boundary conditions are based on the drying requirements of 2.5 kg of EFB. The results from the adopted material selection criteria show that stainless alloy, AISI 304 is the most suitable selection for the burner and the heat exchanger unit. The mathematical calculations results show that heat exchanger of 0.4 m height and 0.4 m diameter is sufficient to produce the required flow rate and temperature of drying air.

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