Thermal design of 5 kg capacity coffee bean dryer simulator using Geothermal energy

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Abstract. A coffee bean dryer simulator that would be able to fully simulate the whole process of coffee bean drying using heat recovered from geothermal energy source has been thermally designed. This simulator is planned to educate people living near the geothermal resources or power plants about the direct use of geothermal energy, especially in coffee drying. The maximum capacity of this simulator is 5 kilogram of fresh coffee bean that is dried using hot air at 45°C and mass flow rate of 0.23 kg/s. The duration of drying is about 3000 seconds which should be adequate to represent the drying process. The heat exchanger proposed for this thesis is a compact heat exchanger with staggered pipe arrangement. The total number of pipes is 10 pipes at 36 cm in length and 65 flat plate aluminium fins measured at 0.6m x 0.16m x 0.005m. The fin efficiency value and the overall surface efficiency value are 72% and 73% respectively.

1. Introduction
Geothermal energy is one of Indonesia a potential energy source to be fully utilized. By 2016, Indonesia has 12386 MW geothermal resource available with around 16,000 MW worth of geothermal energy reserves. Despite that, only around 1,344 MW worth of energy has been utilized. Minimal amount of research on geothermal energy is due to lack of knowledge and awareness on the potential utilization of geothermal energy. Indonesian people are not fully aware many benefits of geothermal energy utilization, either as an indirect energy source or as a direct energy source.

Coffee is one of the largest plantation commodities in Indonesia. Data from the Directorate of Plantation Indonesia recorded coffee production in Indonesia in 2015 reached 750,000 tons. With 100,000 tons of that number is robusta coffee [1] drying by coffee planters mostly still uses a very conventional way of drying technique which takes a very long time. Another type of drying that can cut the drying time of the coffee beans is mechanical drying. With mechanical drying, the coffee beans can be removed by a few days. Mechanical drying is still less favorable due to the very expensive drying costs.

The quality of the coffee beans may change due to changes in drying parameters such as temperature difference, drying time and remaining water content in coffee beans. The processing of coffee beans from cherry shape to dry coffee beans can generally be divided into three types of process: wet, dry, and semi-wet process. The purpose of drying coffee beans is to reduce the water content contained in coffee beans from the initial moisture content of about 65% to reach the standard water content of about 11% -12%. The rate of drying of coffee beans is strongly influenced by seed temperature, water content in seeds, drying fluid temperature, fluid velocity and relative humidity. Drying coffee beans is
approximately 55% - 60% to drop to a maximum of 12% as stated in the Indonesian National Standard (SNI) [1].

The purpose of this study is to conduct a thermal design for a prototype of coffee bean dryer simulator using geothermal heat energy. This coffee bean dryer simulator is expected to be able to educate people living near the geothermal source about direct use of geothermal energy.

2. Design Requirements and Objectives

Design Requirement and Objective (DR&O) in the thermal design of this coffee drying simulator, have been developed; that the simulator must:

1. Be able to simulate actual drying process Coffee dryers can simulate the process of coffee bean drying. The drying process in the field includes the presence of hot air supplied to the drying chamber and the drying of coffee beans as an object.

2. Have drying temperature of about 45 °C and not exceed 50 °C. Critical temperature drying coffee beans is 50 °C. This is because the drying temperature above 50 °C can lead to case hardening. Case hardening phenomenon caused damage to coffee beans. The recommended temperature for drying is about 45 °C.

3. Have the overall small dimensions. Referring to the purpose of making simulator is to educate, then the dimension of the simulator should be small and portable. Simulator must have the proper dimensions simulator in general to gain entry to the exhibition hall. These simulator dimensions should not exceed 1.5m x 3m for length and width and height not exceeding 1.5m simulator for easy observation.

4. Have mass reduction observable in the past 10 minutes, Coffee dryer simulator aims to educate and demonstrate how the coffee drying process occurs while providing adequate description of the use of renewable energy without having to be converted into electrical energy in advance. Therefore, a mass reduction due to drying coffee must be observed mainly in 5 minutes - 10 minutes early. It also refers to the drying curve where the most drastic mass reduction in the period of the initial period of drying.

3. Configuration of System

The closed configuration for coffee bean drying is presented in Figure 1. The geothermal heat source energy in water or steam (brine) is transferred to fresh water. The heat of fresh water is then transferred through a heat exchanger (HE) to air which is blown to coffee bean using a fan.

![Figure 1. Drying configuration using Brine](image_url)

Data properties of air used for this study are listed in Table 1. This data shows the properties of the air at the beginning, after heating, and after evaporating the moisture from the coffee beans. Air nature of the data in Table 1 are used to determine the mass of water that needs to be evaporated so that the
water content in coffee beans to 12% wet basis. Based on the data presented in Table 1, the following information in Table 2 can be calculated.

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**Table 1. Data properties according to the diagram psychometric air dryer**

| Symbol          | Unit                  | Value  |
|-----------------|-----------------------|--------|
| NATURE OF AIR IN THE BEGINNING |                        |        |
| air temperature | °C                    | 25     |
| relative humidity | RHA %                | 65     |
| absolute humidity | ωa kg water / kg dry air | 0.01298 |
| enthalpy        | Ha kJ / kg dry air    | 58.002 |
| NATURE AIR AFTER HEATING |                        |        |
| air temperature | °C                    | 45     |
| relative humidity | RHA %                | 21.47  |
| absolute humidity | ωa kg water / kg dry air | 0.01298 |
| enthalpy        | Ha kJ / kg dry air    | 78.5   |
| NATURE THROUGH THE AIR AFTER COFFEE BEANS |                        |        |
| air temperature | °C                    | 30.1   |
| relative humidity | RHA %                | 70     |
| absolute humidity | ωa kg water / kg dry air | 0.0189  |
| enthalpy        | Ha kJ / kg dry air    | 78.5   |

**Table 2. Calculation of water and air mass.**

| Equation | Calculation |
|----------|-------------|
| Mass of water | \( m_w = \frac{m_{c}(M_{C_1} - M_{C_f})}{100 - M_{C_f}} \) | 2.737 kg |
| the amount of air that needs to be channeled to evaporate water | \( m_{udara} = \frac{m_w}{(H_{R_f} - H_{R_i})} \) | 462.45 kg Dry Air |
| Mass of dry air | \( \dot{m}_a = \frac{m_{a - F}}{t} \) | 0.23 kg/s |
| Mass flowrate and final air temperature | \( Q = \dot{m} \times c_p x (T_f - T_i) \) | 4.6092 kJ/s |
|            | \( T_f = 350.25 \degree K \) |  |
|            | \( = 77.25 \degree C \) |  |
4. Heat Exchanger Calculation Process

In order to calculate the thermal conditions for the heat exchanger, the operating conditions of heat exchangers are:

1. Hot working fluid is hot
2. Cold working fluid is air dried
3. The heat from the working fluid temperature range of 70 °C - 80 °C
4. Cold fluid temperature ranges of 25 °C - 45 °C
5. The fluid pressure in the pipeline has a range of 1 bar - 2 bar
6. The dimensions of the heat exchanger is not too large, below 100 cm x 100 cm

4.1. Analysis Method Using Log Mean Temperature Difference (LMTD)

Based on some consideration, the selected heat exchanger is a compact heat exchanger with a finned surface of the square. Analysis LMTD method (Log Mean Temperature Difference) for this kind of heat exchanger is tabulated at Table 3. Of the value of the heat transfer rate can be determined the value obtained by multiplying the surface area by an average heat transfer coefficient by first knowing the value of the temperature difference logarithmic (ΔTlm). For heat cross flow exchangers, the value of the logarithmic temperature difference needs to be corrected. Correction values obtained using the graph illustrated in Figure 2. To get the value of the correction factor, it should be calculated in advance the value of P and Z as shown in Table 3.

![Figure 2](image.png)

**Figure 2.** The correction factor for the logarithmic temperature difference crossflow type heat exchanger and the fluid does not mix [12]

Using in Figure 2, the value of the correction factor $F = 0.98$. After getting the F value and the value of the logarithmic temperature difference, then use equation 2 to get the value obtained by multiplying the overall heat transfer coefficients with the cross-sectional area of heat transfer.
## Table 3. Result calculation for heat exchanger

| Equation                                      | Results |
|-----------------------------------------------|---------|
| Heat Transfer inside Heat Exchanger            | $Q = U \times A \times \Delta T_{lm} \times F$ (3.3) | 0.1092 kW/K |
| LMTD                                          | $\Delta T_{lm} = \frac{(T_{a,f} - T_{u,i}) - (T_{a,i} - T_{u,f})}{\ln(\frac{T_{a,f} - T_{u,i}}{T_{a,i} - T_{u,f}})}$ | 43.05 °K |
| Constant used to calculation correction factor HE | $P = \frac{t_{a,i} - t_{a,f}}{T_{u,i} - T_{u,f}}$ (3.5) | 0.1375 |
| Constant used to calculation correction factor HE | $Z = \frac{t_{a,f} - T_{u,i}}{T_{a,i} - T_{u,f}}$ (3.6) | 0.570 |
| Diameter Equivalent                           | $d_f = 2.54 X_T \frac{X_D}{X_T} - 0.3$ | 0.099 m |
| Fin Height                                    | $h_f = \frac{1}{2} (d_f - d_r)$ | 0.02951 m |
| Heat transfer surface area                    | $A_f = \frac{N_t L_1 \pi}{s + t_f} \left[ \frac{1}{2} (d^2_f - d^2_r) + d_f t_f \right]$ | 8.53 m² |
| Pipe heat transfer surface area which is not covered by the fin | $A_b = \frac{N_t L_1 \pi}{s + t_f} (d_r \times s)$ (4.6) | 0.411 m² |
| Total Area                                    | $A_{tot} = A_f + A_b$ | 8.946 m² |
| minimum area of free flow                     | $A_{ff} = N_{tt} L_1 (P_t - d_r - \frac{2t_f h_f}{s + t_f})$ | 0.134 m² |
| Size perpendicular flow                       | $A_{fr} = L_2 \times L_3$ | 0.216 m² |
| Fin Ratio                                     | $\frac{h_f}{d_r} = \frac{0.0295}{0.042} = 0.67$ | 0.67 |
|                                               | $0.2 < \frac{h_f}{d_r} < 0.7$ | 0.2 < $h_f/d_r$ < 0.7 |

### 4.2. Process Heat Exchanger Design
The heat exchanger built was a compact heat exchanger with the former type of flat plate finned pipe. The arrangement of pipes that will be used is pipe with staggered arrangement. Illustration of the arrangement of heat exchangers and piping on dimension nomenclature to be used through Figure 3.
Figure 3. The composition of the heat exchanger finned pipe file type flat plate [2] (A) the overall composition (b) a cell block.

Trial-and-Error iteration analysis is used to determine the dimensions of the heat exchanger. The iteration process will not be shown overall but the end result of the iteration process conducted outlined in Table 4.

| Table 4. The dimensions of the heat exchanger |
|-----------------------------------------------|
| Symbol | Unit | Value |
| diameter In | In | m | 0.0254 |
| outside diameter | do | m | 0.027 |
| thick pipe | Tp | m | 0.0008 |
| The thickness of the fin | Tf | m | 0.0005 |
| diameter Root | dr | m | 0.04 |
| Spacing fins | s | m | 0.005 |
| Number of Pipe Per Column | Ntc | 5 |
| Amount Per Line Pipe | Ntr | 2 |
| The total number of pipes | nt | 10 |
| the number of fins | F | 65 |
| Longitudinal spacing between pipes | Pl | m | 0.08 |
| Transversal spacing between pipes | Pt | m | 0.12 |
| Long pipe | L1 | m | 0.36 |
| The length of the longitudinal fins | L2 | m | 0.16 |
| Long transverse fin | L3 | m | 0.6 |

Materials to be used for pipe is stainless steel AISI 304 while using aluminum alloy fin 6069. The air flow will flow cross through the pipe arrangement arranged staggered, Installation of fin in Figure 4 were not made by welding but by installing the pipe segment segment. Holes will be cut diameter adapted to the outer pipe diater and will be heated to corrected fitting and there is contact between the pipes and fins.

The next step in the calculation is by letting fin geometry into a round shape. This is to facilitate the calculation of the efficiency of the fin. The fins will be divided into parts of the cell. Cell cell division
is illustrated in Figure 5. The first thing to be done is to determine the equivalent diameter of the fins. The XD and XT values are tabulated at Table 3. 

Figure 4. Fin type of L foot [2]

Figure 5. The division of cells in the pipeline with a staggered arrangement [2]

From the calculations, the total value obtained air flow heat transfer is of 8.946 m². Furthermore, it should be calculated minimum area of free flow. Spacious minimum free flow, Aff, contained in the pipe between the transverse plane. The minimum value of the free flow of this area will use to determine the value of the maximum speed of the air to flow freely.

4.3. Convection Coefficient Calculation
To get the value of the overall heat transfer coefficient, required convection coefficient of fluid flow. Assumption of thermal contact for heat exchanger is ignored because its value is very small. for the Alluminium alloy case with stainless steel, the thermal contact resistance value of about 0.04 x 10^-4 m².K/W.

Figure 6. Thermal circuit heat exchangers
Table 5. Air Dryer Properties

| Magnitude (Air)          | Symbol | Unit   | Value |
|--------------------------|--------|--------|-------|
| Mass flow rate           | m      | kg / s | 0.23  |
| Inlet temperature        | Tin    | °C     | 25    |
| Exit temperature         | Tout   | °C     | 45    |
| T average                | Tf     | °C     | 35    |

NATURE OF DATA ON AIR TEMPERATURE 35 °C

| Specific calor          | cp     | kJ / kg.K | 1,002 |
| Density                 | ρ      | kg / m³   | 1.15  |
| Conduction coefficient  | k      | kW / mK   | 26.5 x 10⁻⁶ |
| Dynamic viscosity       | μ      | Ns / m²   | 18.4 x 10⁻⁶ |
| Prandtl                 | pr     |          | 0.706 |

To get the value of the overall heat transfer coefficient, thermal circuit need to review that applies to this heat exchanger. For this heat exchanger, some of the assumptions used are the contact thermal resistance is ignored, there is no surface fouling, and there are fins on the pipe surface.

Thermal circuit schemes used are shown in Figure 6. The scheme is illustrated in Fig 6 when sorted from outside to inside is air flow convection thermal barriers, thermal barriers due fouling the air flow, thermal conduction barrier fin, pipes conduction thermal barriers, thermal barriers due to fouling in the flow of hot water in the pipes, and thermal convection barriers to the flow of hot water in the pipeline. The air dryer properties for the calculation are presented in Table 5. The result of thermal circuit is tabulated in Table 6.

4.4. Comparison Values of UA

Based on the calculations in section 4.3, the overall value of the total resistance of heat transfer to the heat exchanger can be completed. The total resistance value of the overall heat transfer is defined as the value obtained by multiplying the overall heat transfer coefficient multiplied by the heat transfer surface area of the heat exchanger. The equations that meet the definition set forth in Equation (4:24).

After getting the value of the multiplication of the results of the iteration, the next need to review the results of the design compared with the results of the calculation method *Log Mean Temperature Difference* which has been calculated at 3.6 parts. Comparison of two methods of calculation performed may be reviewed in Table 7.

Table 6. Thermal Resistances calculation

| Equation          | Calculation |
|-------------------|-------------|
| a. Thermal Convection Air Flow Resistance | $R_{cone,u} = \frac{1}{h_{a}A_{u}\eta_{f,o}}$ |
|                   | 3.66 $\frac{kW}{K}$ |
|                   | $\eta_{f} = tanh\left(\frac{2h_{a}}{\eta_{f}K_{f}}\times\Psi\right)$ |
|                   | 0.71 |
|                   | $\Psi = \frac{d_{r}}{2}\left(\frac{d_{r}}{d_{e}} - 1\right)(1 + 0.35\ln\frac{d_{r}}{d_{e}})$ |
|                   | 0.0388 |
|                   | $\eta_{f,o} = 1 - \left[\frac{R_{f,u}A_{u}}{A_{u}\eta_{f,o}}\right]$ |
|                   | 0.73 |
| b. Fouling thermal Resistance | $R_{f,u} = \frac{R_{f,u}}{A_{u}\eta_{f,o}}$ |
|                   | 0.015 $\frac{kW}{K}$ |
c. Thermal Conduction Resistance in Fins
\[ R_{\text{cond},f} = \frac{\ln\left(\frac{d_r}{d_o}\right)}{2\pi k f L_f} \]

\[ 0.864 \frac{K}{kW} \]

d. Thermal Conduction Pipe Resistance
\[ R_{\text{cond},f} = \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k f L_t} \]

\[ 1.56 \frac{K}{kW} \]

e. Fouling on Water Flow Thermal In Pipe Resistance
\[ R_{f,a} = \frac{R_{f,a}}{A_a} \]

\[ 0.63 \frac{K}{kW} \]

\[ A_a = \pi x d_i x L_a \times N_t = 0.287 \text{ m}^2 \]

f. Thermal Convection Hot water Flow in Pipe
\[ R_{\text{conv},a} = \frac{1}{h_a A_a} \]

\[ 2.37 \frac{K}{kW} \]

Comparison value \( U \times A \)
\[ \frac{1}{U A} = R_{\text{conv},a} + R_{f,a} + R_{\text{cond},f} + R_{\text{cond},t} + R_{f,a} + R_{\text{conv},a} \]

\[ 9.106 \frac{K}{kW} \]

\[ UA = 0.1098 \frac{kW}{K} \]

Table 7. Comparison of overall heat transfer coefficient multiplication with a heat transfer surface area

| Magnitude | Unit | Value |
|-----------|------|-------|
| \( U \times A \) of calculation LMTD | kW / K | 0.1092 |
| \( U \times A \) of heat exchanger design iterations | kW / K | 0.1098 |
| - gap | kW / K | 0.0006 |
| percent Difference | % | 0.54 |

From the comparison shown in Table 7 can be seen the difference value \( U \times A \) heat exchanger design results with the values obtained from the calculation using the LMTD method. Values obtained difference is equal to 0.0006 kW / K with a percentage difference of 0.54%. Based on this calculation heat exchanger design iterations are obtained as valid.

5. Analysis of Thermal Design

5.1. Heat Exchanger Performance Analysis
Parameter achievement heat exchanger can be evaluated by calculating the heat exchanger effectiveness. Effectiveness exchanger, \( \varepsilon \), heat is the ratio between heat transfer occurs in the heat exchanger with a maximum value of heat transfer occurs in the heat exchanger. The maximum heat transfer value can be determined by first determining the value of Cmin. The C as shown in Table 8 is the value obtained by multiplying the rate of mass heat capacity. The results of thermal design calculation tabulated in Table 9.

Table 8. The C values

| Flow        | The rate of mass (kg/s) | The heat capacity (kJ/kg.K) | C (kW/K) |
|-------------|-------------------------|-----------------------------|----------|
| Dry air     | 0.23                    | 1.002                       | 0.2304   |
| Hot water   | 0.4                     | 4.196                       | 1.678    |

Effectiveness value of 0.36 is very small when compared with the effectiveness of the heat exchanger in the new industry can be categorized as good, if effectiveness is above 0.70. This is due to a decrease in the temperature of the water is still small so that, when viewed from the ability of the heat exchanger to transfer heat from one fluid to another fluid, the effectiveness of this heat exchanger is small. However, LMTD correction factor value at the heat exchanger is high enough 0.98. The results of the
heat exchanger design can be categorized as good if obtained LMTD correction factor above 0.8. This is because heat transfer occurs when the fluid temperature when the heat is still high and low temperature cold fluid is better than the low temperature hot fluid and cold fluid temperature is high. Besides the efficiency of the fins on this thermal design is good enough at 72% and 73% efficiency overall surface accordingly.

5.2. Airflow Pressure Drops
The pressure drop in air flow occurs when air passes through a heat exchanger and when the air passes through a pile of coffee beans. The pressure drop that occurs when dry air passing through the heat exchanger caused by the friction between the dry air and the surface of the heat exchanger. Friction occurs on the surface of the fin and the pipe surface.

| Table 9. Thermal Design Analysis |
| Equation | Calculation |
| HE Effectiveness | $\varepsilon = \frac{Q}{C_{\text{min}}(T_h-T_i)}$ (6.1) | 0.36 |
| Air Pressure drop | $\Delta P_a = (Ka + \frac{N_f Kf}{2}) \frac{\mu a}{d_{\text{max}}}^2$ | 51.65 Pa |
| $Ka = 1 + \sigma^2 = 1.38$ | |
| $Kf = 4.567 Re^{-0.244} \left( \frac{A_a}{A_{bt}} \right)^{0.504} \left( \frac{P_l}{d_r} \right)^{-0.376} \left( \frac{P_i}{d_r} \right)^{-0.546} = 0.94$ | |
| $\sigma = \frac{A_{ff}}{A_{fr}} = 0.134$ | |
| $A_{bt} = \pi x d_r x N_t x L_1 = 0.45 \text{ m}^2$ | |
| Water pressure drop of water flow in the pipes | $\Delta P_b = 4 f_o \frac{L}{d_i} \frac{\rho v^2}{2}$ | 53.85 Pa |
| Fanning Factor | $f_o = \frac{1}{0.0074}$ | |
| $= -0.25 \left\{ \frac{1}{2} \log \left( \frac{\varepsilon}{4 \pi \mu d_{\text{max}}} \right) + \frac{5.9452}{Re} \log \left( \frac{1}{5.8506} \frac{Re^{0.8}}{Re^{0.5}} \right) \right\}$ | |
| $\varepsilon$ is friction coefficient | |
| Total Length | $L = N_t x L_1$ | 3.6 m |
| Safety Factor | $SF = \frac{23 L_2}{P(d_0 - 2t_1)}$ | 35.12 |
| Fan Pressure Requirement | $P_1 = \rho_1 g h_1 + \frac{1}{2} \rho_1 v_1^2 + P_i = \rho_2 g h_2 + \frac{1}{2} \rho_2 v_2^2 + P_2$ | 114328 Pa |
| $P_i = 114328$ Pa | |
| Fan Debit Requirement | $Aliran Udara = \frac{m}{\rho_a x 3600} \frac{720.8 \text{ m}^3}{\text{jam}}$ | |
5.3. **Hot Water Pressure Drop in Pipe**

The flow of hot water flowing in the pipe will also decrease the pressure. The pressure drop caused by friction between the water with the inner surface of the pipe.

5.4. **Safety Factor the Pipeline**

As specified in the discussion in chapter 4, ang pipe will be used for heat exchangers are stainless steel AISI 304. The fluid flowing in the heat exchanger is pumped first. Maximum pressure of the fluid in the pipeline is 2 bar. To analyze the strength of the pipe, the standards that will be used is the B31 standard ASME (American Society of Mechanical Engineers Code for Pressure Piping). The equation used to calculate the safety factor on the pipe contained in equation

5.5. **Required Fans and Pumps**

To drain fluid flow both dry air and hot water in the pipeline, needed a tool that is able to raise the pressure to be in the flow.

6. **Conclusions**

Thermal design of a coffee drying simulator using geothermal energy has been developed. Main specifications of the simulator are:

- The final dimensions of simulator are length of 2.25 m, height of 0.95 meters and a width of 0.59 m.
- The rate of dry air mass that is needed to dry 5 kg of beans is 0.23 kg/s. Initial levels of robusta coffee beans drying is 60% and the final concentration of 12%. Coffee beans can be dried significantly within a period of 50 minutes.
- Heat exchanger to be designed is a type of compact heat exchangers with finned pipe file type flat plate. U x A multiplication value obtained from the results of the design is equal to 0.1098 kW/°K with an error of 0.53%.
- The efficiency value fins on the heat exchanger designed amounted to 72% with an average efficiency level of 73%. Efktifitas heat exchanger is 36% and LMTD correction factor of 0.98. Total pressure drop of the airflow is at 61.65 Pa. The pressure drop of water flow in the pipes is equal to 53.85 Pa. Blower fan capacity is required with a capacity of 424 CFM blower.

**References**

[1] Prastowo B 2013 *Cultivation and Post Harvest Coffee*, Bogor.
[2] Kuppan T 2000 *Heat Exchanger Design Handbook 2nd Edition* Marcel Dekker Inc, New York