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Evaporator Design With Ammonia-Water Mixture as Working Fluid for Kalina KCS34 Cycle On Electric Power Plant

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Abstract

Natural hot spring can be used as a heat source for generating electricity by using a Kalina Cycle KCS34 technology with ammonia-water mixture as the working fluid. An Evaporator is one component of the Kalina KCS34 cycle to change the phase of the working fluid. The purpose of this research was to obtain evaporator design for the Kalina KCS34 cycle. Simulation using cycle tempo was conducted to define the inlet and outlet temperature of the evaporator. These temperatures will be used in designing the evaporator was designed using Log Mean Temperature Difference and energy balance methods. The dimension of the evaporator was evaluated with the pressure drop, the effectiveness of the heat exchanger, and standards. The evaporator was using 4 pass shell and tube type with inlet and outlet temperature of the hot fluid (heat source) at 80 °C and 50 °C respectively and the inlet and outlet temperature of the cold fluid (working fluid) at 45 °C and 64 °C respectively. The dimension of the evaporator is 2,5 m in length, 34 tubes per pass with 19,15 diameters. This results in the capacity of the evaporator at 77,41 kW and 75% effectiveness.

Keywords: Kalina cycle, Heat exchanger, the working fluid, ammonia-water, LMTD

1. Introduction

Geothermal energy is one of the potential renewable energy to be used in generating electricity. Indonesia has 40% of geothermal potential in the world (Indonesia Investments, 2015). One of Geothermal potential indication is a natural hot spring source. Natural hot spring has a low temperature ranging from 60 °C to 80 °C. This lowtemperature heat source can be utilized for an electrical power plant using special technology such as the Kalina KCS34 cycle (Indonesia Investments, 2015).

Kalina cycle was invented by Dr. Alexander Kalina by using ammonia-water mixture as the working fluid. Ammonia-water mixture has low boiling temperature characteristics and suitable for low-temperature heat sources (Pramuda et al, 2017). The components in the Kalina KCS34 cycle are evaporator, condenser, separator, recuperator, and turbine (Figure 1).

The evaporator in the Kalina KCS34 cycle used to change the phase of the working fluid using the heat from natural hot spring water fluid. The working fluid exit the evaporator were two phases, ammonia vapor, and water mixture.

Simulation of Kalina KCS34 cycle using *Cycle* Tempo with 79% ammonia-water mixture fraction was conducted to obtain the inlet and outlet temperature of the evaporator. These temperatures will be used for designing the evaporator. The result expected from this research is the type of evaporator, dimensions, and effectiveness of the evaporator.

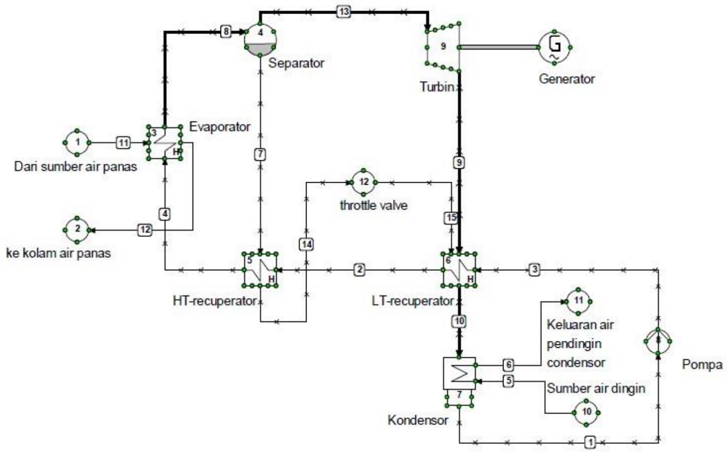

Fig. 1: Schematic Diagram of Kalina KCS34 Cycle (Pramuda et al, 2017)

2. Research methodology

The methodology conducted on this research was simulation using *cycle tempo* to obtain inlet and outlet temperature of the evaporator and designing the evaporator using Log Mean Temperature Difference (LMTD) method and evaluated using standards.

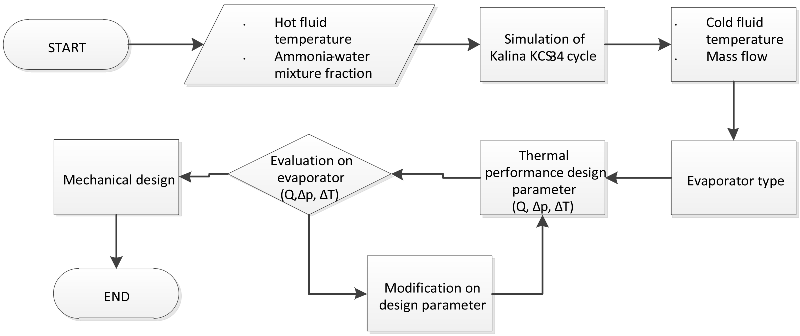

Fig. 2: Research Flow Chart

3. Simulation

A Simulation conducted on this research based on the field measurement conducted by Sirodz et al (2016). The temperature of natural hot spring water was 80°C. The evaporator outlet (hot side) was maintained at 50°C due to hot spring tourism requirements. With the 79% mixture fraction of ammonia-water, the simulation shows the inlet and outlet cold fluid (ammonia-water mixture) temperature ($T_{c,i}$ & $T_{c,o}$) were 45°C and 64°C (Figure 3). The operating pressure of the heat exchanger was 12 bar.

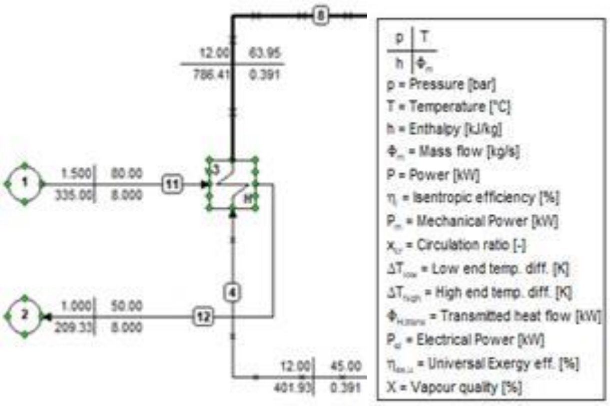

Fig. 3: Simulation results

Based on the operating pressure and operating temperature, shell and tube heat exchanger type C will be used in this research. Shell and tube heat exchanger also has another benefit which is easy for maintenance. To minimize the length of the evaporator, 2 shell passes and 4 tube passes will be used. Considering that ammonia is corrosive, therefore the working fluid will be assigned to the tube side.

4. Evaporator design

4.1 Shell and tube design

To design the heat exchanger performance, the total heat transfer rate must be related to inlet and outlet fluid temperature, overall heat transfer coefficient, and total heat transfer area. This relation can be calculated using the energy balance. In this research, energy balance was calculated to obtain the heat received by the working fluid from the heat source. The heat released by the hot spring water can be calculated with:

$$Q_h = \dot{m} \times (h_{w\,in} - h_{w\,out}) \tag{1}$$

With hot water inlet temperature ($T_{h,i}$) 80 °C, outlet temperature ($T_{h,o}$) maintained at 50°C, and mass flow of 8 kg/s, the heat released by the hot water was 150,54 kW.

To determine the evaporator heat transfer area, Log Mean Temperature Difference (ΔLMTD) method was used. The value of ΔLMTD that has been calculated will be evaluated by a correction factor because the evaporator has multipass flow.

$$LMTD = \frac{(Th,i-Tc,o)-(Th,o-Tc,i)}{\ln\frac{(Th,i-Tc,o)}{(Th,o-Tc,i)}} \tag{2}$$

From the calculation value of 9,45 K was obtained. The correction factor was determined by using LMTD correction factor graph with known temperature efficiency (P) and the ratio of the hot fluid temperature difference and cold fluid temperature difference (R).

$$P = \frac{T_{c,o}-T_{c,i}}{T_{h,i}-T_{c,i}} \quad (3) \qquad R = \frac{T_{h,i}-T_{h,o}}{T_{c,o}-T_{c,i}} \quad (4)$$

From Figure 4 with P = 0,54 and R = 1,54, the LMTD correction factor is 0,75. With this correction, the LMTD now has value of 7,09 K.

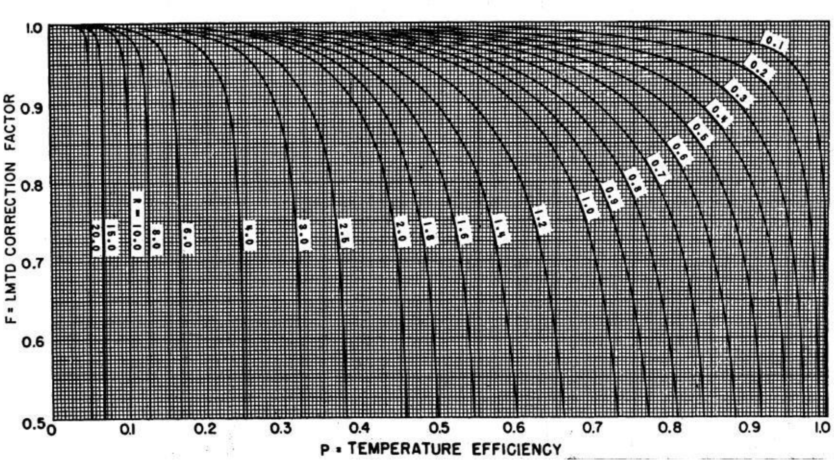

Fig. 4: LMTD Correction Factor (TEMA 8th Edition, 1999)

To determine the heat transfer area, the overall heat transfer coefficient of the ammonia-water mixture has to be known. With a 79% mixture fraction, the overall heat transfer coefficient is 952,77 W/m2K. The heat transfer area was calculated using the formula:

$$A = \frac{Q_h}{U_{mixture} \times \Delta LMTD_{corrected}} \tag{5}$$

The heat transfer area calculated is 22,28 m^2. In determining the dimension of the heat exchanger, Standard of The Tubular Exchanger Manufacturer Association (TEMA) is used. The tube length is assumed to be 2,5 m for 1 pass. Copper was selected as the tube material due to high thermal conductivity. Iteration was conducted to find the optimum tube diameter with pressure drop and tube area as the parameters. From the iteration, 19,1 mm tube diameter is optimum. With this diameter, the area of the single tube (A^t) is 0,149 m^2 therefor the number of tubes (N) required for the evaporator is calculated using:

$$N = \frac{A}{A'} \tag{6}$$

The number of tubes required for the evaporator is 48 tubes. To facilitate maintenance activity, the square tube pitch was selected. The pitch distance (pt) is 1,25 times the outside tube diameter. With this pitch, the bundle diameter will be 393,49 mm. The bundle diameter will determine the shell inside diameter.

The shell type of the evaporator is assumed to be fixed and U tube type. From the bundle diameter value and type of the shell, using the graph in Figure 5 the clearance between shell and bundle was 13 mm. With the minimum shell thickness of 9,5 mm and shell diameter (DS) of 406,49 mm, the outside shell diameter will be 424,98 mm.

Inside the shell, there are baffles to direct the fluid stream across the tubes and improve the heat transfer rate. The evaporator was designed with segmental baffles. The distance between baffles (Bs) is 81,29 mm were calculated using formula (7) and the number of baffles (Nb) is 30 pieces were calculated using formula (8).

$$Bs = \frac{1}{5} \times Ds \tag{7}$$

$$Nb = \frac{l}{Bs} \tag{8}$$

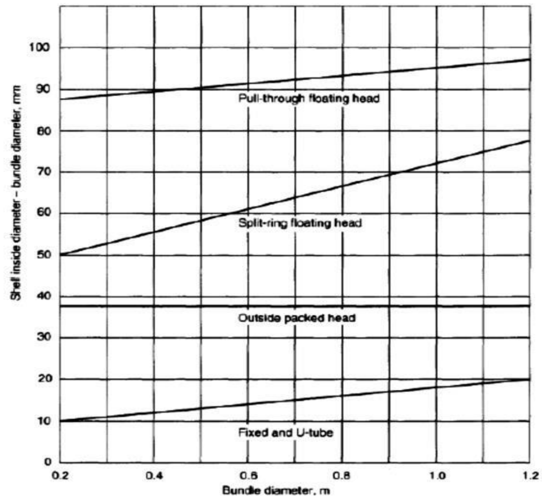

Fig. 5: Shell clearance (Sinnot,2005)

Typical baffle clearance and tolerance from Colson & Richardson (2005) were used to determine the baffle clearance. Due to the shell diameter of 406,49 mm, the baffle diameter (Dbv) will be 404,89 mm. To allow the fluid flow between the baffles, the baffles needs to be cut. The optimum baffles cut are 25 % of the baffle diameter. This cut will give a good heat transfer rate without excessive pressure drop. From equation (9) the flow area (As) was 0,0064 m^2 and give shell side mass velocity (Gs) 1250,39 kg/m^{2}s using equation (10). The shell side equivalent diameter (de) is calculated using the flow area between the tubes taken in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes (equation (11)). For the square pitch arrangement, the equivalent diameter was 18,85 mm.

$$As = \frac{(pt - do)Db.Bs}{pt} \tag{9}$$

$$Gs = \frac{Shell\ Flow\ rate(\frac{kg}{s})}{As} \tag{10}$$

$$de = \frac{1,27}{do}(pt^2 - 0,785\ do^2) \tag{11}$$

4.2 Pressure drop

Kern methods used to determine the pressure drop in the shell side due to the complex flow pattern. The Kern methods doesn't take account of the bypass and leakage streams. The pressure drop can be calculated using the formula :

$$\Delta p_s = 8 \times Jf \times \left(\frac{Ds}{d_e}\right) \times \frac{l}{Bs} \times \frac{\rho^2 u}{2} \times \left(\frac{\mu}{\mu_w}\right)^{0,14} \quad (12)$$

Where Jf is the friction factor, l is the tube length, and Bs is the baffle spacing. Arise from Reynold number of 54168,46 and using the shell side friction factors graph (Figure 6) with a 25 % baffle cut gives the friction factor of $4x10^{-1}$. Therefore the pressure drop on the shell side is 7,033 Pa.

The pressure drop inside the tube was related to the working fluid and thermal coefficient of the tube. The number of the tube for each pass can be calculated using equation (13) with 4 number of passes. The number of

tube per pass are 37 tubes. With this number of tubes, the total heat transfer area per pass was 0,00637 m^2. Therefore the flow of the working fluid was 0,00055 m^3/s. This comes with the Reynold number of 2333,3 to calculate the friction factor. The friction factor (Jf) was 0,5 with considering copper as the tube material and the tube length of 2,5 m. The pressure drop calculation using equation (14) and resulting in 19378,86 Pa of pressure drop. The pressure drop in the heat exchanger is relatively low about 1,6% compared to the operating pressure.

$$Nttp = \frac{Nt}{Pass} \quad (13)$$

$$\Delta p_s = \left[Nt\left[2,5 \times \left(\frac{8Jf \times L}{d_i}\right) \times \left(\frac{\mu}{\mu_w}\right)^{-m}\right]\right]\frac{\rho.u^2}{2} \quad (14)$$

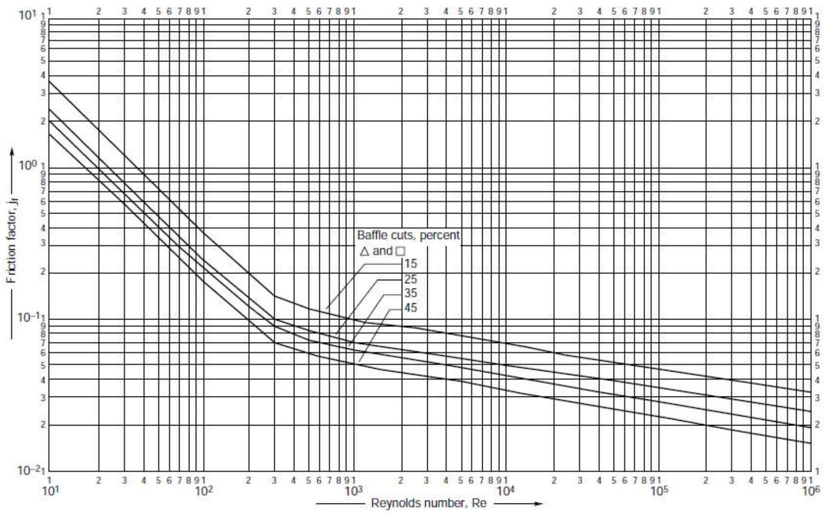

Fig. 6: Shell side friction factor (Sinnot,2005)

4.3 Fouling factor

Fouling can occur in the evaporator due to the operation and condition of the fluids. Fouling factor can be calculated with the value of the overall heat transfer coefficient during the dirtiest condition and the materials to be used. The heat transfer coefficient on the tube side (hi) can be calculated using equation (15) with the Prandtl number of 3,55. The fluid fouling factor for ammonia-water (hdi) was taken 5000 W/m2°C. The heat transfer coefficient for the shell side (ho) can use the equation (15) with the Reynolds number of 54168,46 and the Prandtl number of 2,71. The fluid fouling factor for the hot fluid (hdo) was taken 3000 W/m2°C. The overall heat transfer coefficient during the dirtiest condition can be calculated using equation (16).

$$h_i = jh\ \frac{k_f}{d_i}(Re.\mathrm{Pr})^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (15)$$

$$U_i = \frac{1}{\frac{1}{h_i}+\frac{1}{h_{di}}+\frac{d_i\,ln\left(\frac{d_i}{d_o}\right)}{2\,k_w}+\frac{d_i}{d_o h_{do}}+\frac{d_i}{d_o h_o}} \quad (16)$$

The overall heat transfer coefficient during the dirtiest condition was 595,86 W/mK. During the clean condition, the overall heat transfer coefficient was 952,77 W/mK. Therefore the fouling factor was 0,00310 using equation (17). This fouling factor is acceptable.

$$R = \frac{U_i + U}{U_i \times U} = \frac{486{,}3 + 952{,}77}{486{,}3 \times 952{,}77} \tag{17}$$

4.4 Heat transfer effectivity

Heat transfer effectivity represents heat exchanger performance. The value of heat transfer effectivity was affected by the mass flow and temperature of the cold and hot fluid. The heat transfer effectivity can be calculated using equation (18) .

$$\epsilon = \frac{Q\ act}{Q\ max} \times 100\% \tag{18}$$

The actual heat transfer (Q_{act}) was 58063,83 W and the maximum heat transfer (Q_{max}) was 77418,44 W.

Therefor the heat transfer effectivity was 75%.

5. Conclusion

The evaporator was designed with shell and tube heat exchanger type C and has the capacity of 77,41 kW and the heat transfer effectivity of 75 %. The dimension of the evaporator presented in Table 1. The general component of the evaporator presented in Figure 7.

Table 1. Dimensions of the evaporator

Shell		
Shell diameter	406,87	mm
number of baffles	31	baffles
baffle thickness	45	mm
shell material		carbon steel
Tube		
number of passes	4	passes
tube length (1 pass)	2500	mm
number of tubes (1 pass)	37	tubes
tube outside diameter	19,15	mm
tube inside diameter	14,82	mm
tube layout		square pitch
tube pitch	23,8	mm
tube material		copper

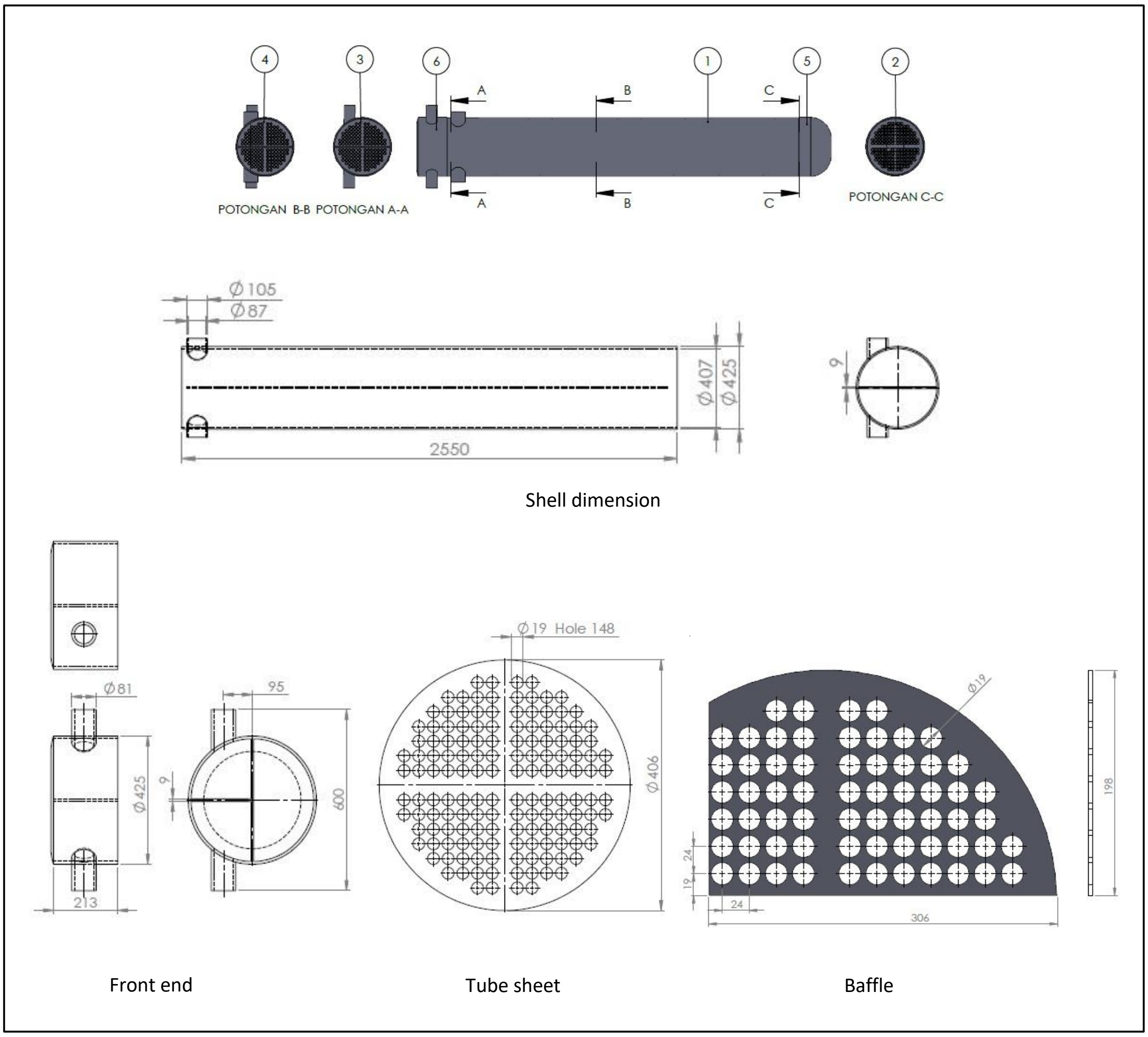

Fig. 7: Dimensions of evaporator

6. References

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