

Design of 10 kW Low Temperature Power Cycle Using Amine—CO₂ Fluid

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Abstract Our thermodynamic study indicated that low temperature power cycle using amine-CO₂ fluid can obtain the performance equal to or higher than that of the current organic Rankine cycle. We designed a 10 kW test equipment. We set high temperature heat source is hot water at the temperature of 90°C and the flow rate of 8,200kg/h, which is a coolant from a gas engine. The heat and mass balance of the equipment was calculated thermodynamically. The result showed the power of 10.5kW and the system efficiency of 7.3 per cent at the the amine – CO₂ fluid flow rate of 1,000kg/h, and the turbine expansion ratio of 4.8. The preliminary estimation shows as follows. The diameter and rotational speed of the turbine blade are 0.115m and 54,400min⁻¹, respectively. The number, width, and length of the recuperator plates are 20, 117mm, and 835mm, respectively. The height and cross sectional area of the absorption bed are 0.35m and 0.027m², respectively.

1 Introduction

Low temperature heat below 200°C is produced massively (Shindo *et al.*, 2008). Kalina cycle using NH₃ and organic Rankine cycle using HFC-245fa (CHF₂CH₂CF₃) can generate electricity from the low temperature heat. However, the former is toxic and corrosive and the latter has a high GWP value, which is required to reduce by the Kigali amendment to the Montreal Protocol and the Regulation (EU) No. 517/2014. As a process that utilizes less toxic and low GWP working fluid, the present authors have proposed to use Amine-CO₂ absorption/desorption system for power generation. The thermodynamic simulation shows the maximum power of the low temperature cycle using 30 mass% methyl diethanol amine (MDEA)-H₂O solution (CO₂/MDEA mole ratio: 0.15) is equal to that of the Organic Rankine cycle using HFC245fa. The experimental results suggest the low temperature cycle using the MDEA-based solution containing carbon dioxide is available as a low temperature cycle (Ogawa *et al.*, 2019). This paper shows mass and heat balance, preliminary designs of a recuperator, a turbine, and an absorption bed in a 10 kW low temperature power cycle using MDEA-based solution.

2 Mass and Heat Balance

Figure 1 shows a low temperature cycle using amine-CO₂ fluid. An amine-CO₂-H₂O solution(S8) is pressurized in a pump(P1) and heated in a recuperator(Hx3). The preheated fluid(S12) is heated at a

boiler (Hx1) by hot water (S1), whose temperature and flow rate are 90°C and 8,200 kg/h, respectively. A two phase flow (S2) from the boiler (Hx1) is separated into a H₂O-CO₂ gas mixture (S11) and a residue (S13) by a gas-liquid separator (Sep1). The H₂O-CO₂ gas mixture (S11) drives a turbine (Ex1) and expands adiabatically. The residue (S13) heats a pressurized amine-CO₂-H₂O solution (S3). The expanded H₂O-CO₂ gas mixture (S5) and the residue (S13) cooled in the recuperator (Hx3) are mixed in a mixer (M1) and cooled to 40°C in an CO₂ absorber/cooler (Hx2) where CO₂ is absorbed in the residue and water mixture.

It is assumed that the values of the polytropic efficiency of P1, the isentropic efficiency of EX1, the pressure losses of Hx1, Hx2, Hx3, and Sep1 are 75%, 80%, 0kPa, respectively. The pinch temperatures of Hx1, Hx2, and Hx3 are assumed to be 4°C, 5°C, and 5°C, respectively.

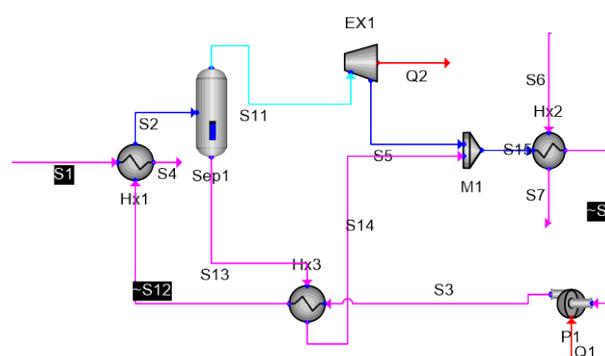


Figure 1. Low temperature cycle using amine-CO₂ fluid

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Table 1. Mass and heat balance

Stream	Temperature (°C)	Pressure (kPa)	Flow Rate (kg/h)
S1	90.0	103.30	8200.0
S2	76.8	33.81	1000.0
S3	40.0	33.81	1000.0
S4	75.0	103.30	8200.0
S5	39.2	7.00	216.70
S6	30.0	103.30	14451.0
S7	37.9	103.30	14451.0
S8	40.0	7.00	1000.0
S11	76.8	33.81	216.7
S12	63.8	33.81	1000.0
S13	76.8	33.81	783.3
S14	45.0	33.81	783.3
S15	42.9	7.00	1000.0

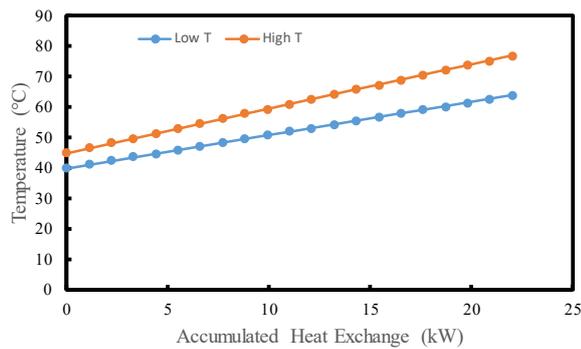


Figure 2. Temperatures in recuperator (Hx3) as a function of accumulated heat exchange

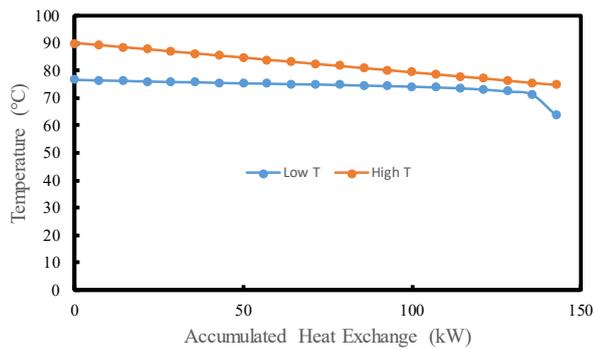


Figure 3. Temperatures in boiler (Hx1) as a function of accumulated heat exchange

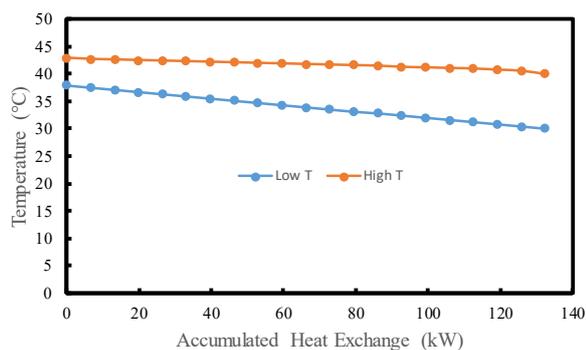


Figure 4. Temperatures in cooler (Hx2) as a function of accumulated heat exchange

The vapor phase fraction at the EX1 outlet is more than 0.88 (Nishikawa, 1965). The pressure at the pump inlet is the minimum value when the vapor phase fraction is zero. The process simulator VMGSim™ v10.0 (Thermodynamic model: Amines) of the Virtual Materials Group Inc. simulated the model.

Its power of 10.5 kW and efficiency of 7.3 per cent are achieved at the fluid flow rate of 1,000 kg/h and expansion ratio of 4.8. Table 1 shows the mass and heat balance.

Figures 2, 3, and 4 show the temperatures of the both side fluids as a function of the accumulated heat exchange in the recuperator (Hx3), the boiler (Hx1), and the cooler (Hx2), respectively.

In the boiler the low temperature fluid is the heated amine-CO₂-H₂O solution (S12), which desorbs the CO₂-H₂O gas mixture. Its equilibrium temperature between liquid and gas phases increases with CO₂ desorption and water vaporization.

3 Recuperator

The total heat load of a plate heat exchanger (PHE) is as follows:

$$Q = n \sum Q_i = n \sum U_i A_i \Delta T_{LMTD,i} \quad (1)$$

where Q is total heat load of a PHE (W), n is number of the plates for heat exchange, Q_i is local heat load (W), U_i is local overall heat transfer coefficient (W/(m²·K)), A_i is local heat transfer area (m²), and $\Delta T_{LMTD,i}$ is local logarithmic average temperature difference (°C).

The local overall heat transfer coefficient U_i is calculated as follows :

$$U_i = 1/(1/\alpha_{H,i} + 1/\alpha_{L,i} + \delta/\lambda) \quad (2)$$

where $\alpha_{H,i}$ and $\alpha_{L,i}$ are local film heat transfer coefficients (W/(m²·K)) for hot and cold streams, respectively, δ is plate thickness (m), λ is thermal conductivity (W/(m·K)) of the plate material.

The local logarithmic average temperature difference $\Delta T_{LMTD,i}$ is defined as follows :

$$\Delta T_{LMTD,i} = (\Delta T_{A,i} - \Delta T_{B,i}) / \ln (\Delta T_{A,i} / \Delta T_{B,i}) \quad (3)$$

where $\Delta T_{A,i}$ and $\Delta T_{B,i}$ are local temperature differences at both ends of local area, respectively.

The local film heat transfer coefficients are calculated by empirical correlations:

$$Nu_i = A Re_i^m Pr_i^{0.4} (\mu_i / \mu_{w,i})^{0.14} \quad (4)$$

where Nu_i is local Nusselt number, Re_i is local Reynolds number, Pr_i is local Prandtl number, μ_i and $\mu_{w,i}$ are local viscosity (Pa·s) at stream and at wall temperatures, respectively (Arsenyeva *et al.*, 2010).

Re_i , Pr_i , and Nu_i are defined as follows :

$$Re_i = \rho_i v_i d_{eq,i} / \mu_i \quad (5)$$

$$Pr_i = \mu_i C_{p,i} / k_i \quad (6)$$

$$Nu_i = \alpha_i d_{eq,i} / k_i \quad (7)$$

where ρ_i is local density (kg/m³), v_i is local velocity (m/s), $d_{eq,i}$ is local hydraulic equivalent diameter (m), $C_{p,i}$ is local specific heat at constant pressure (J/(kg·K)), and k_i is local thermal conductivity (W/(m·K)).

Equations (1)–(7) calculate the plate length as a function of the accumulated heat exchange in the recuperator, whose plate width, distance between plates, plate thickness, and number of plates are 117 mm, 2.35 mm, 0.5 mm, and 20 mm, respectively. The plate length results in 835 mm. The reference of thermophysical properties for working fluid (Amine-CO₂-H₂O) is the thermodynamic model: Amines of the Virtual Materials Group Inc.

Figure 5 shows the plate length as a function of the accumulated heat exchange in the recuperator.

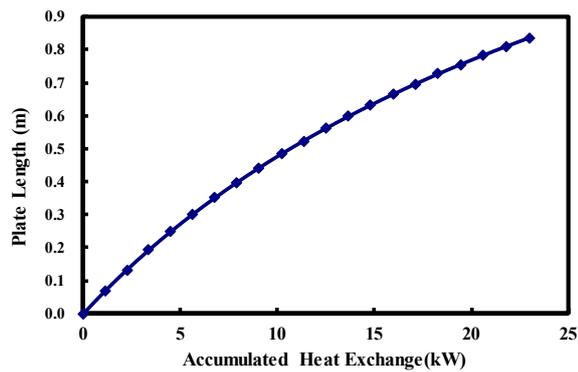


Figure 5. Temperatures in recuperator as a function of accumulated heat exchange

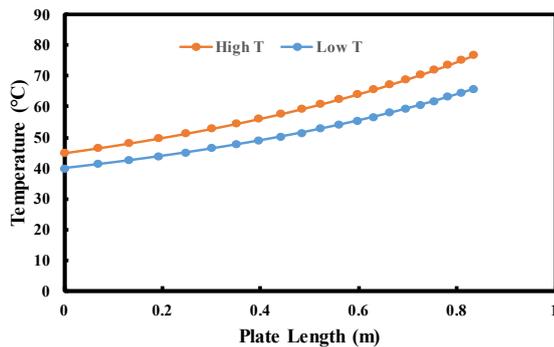


Figure 6. Temperatures in recuperator as a function of plate length

Figure 6 shows the temperatures of high and low temperature streams as a function of the plate length in the recuperator. The calculated exit temperature of the low temperature stream of 66°C is nearly equal to that in the mass and heat balance of 64°C.

4 Turbine

The expansion ratio of the turbine is 4.8. We adopt a single stage radial flow turbine. The max efficiency of a radial flow turbine should reach at the specific speed of 0.6 and the rotor speed/thermal insulation speed ratio of

0.7 (Inoue *et al.*, 2006). The specific speed, rotor speed, and thermal insulation speed are related in Eqs. (8) – (12).

$$N_s = \omega V_{T2}^{0.5} / dh_T^{0.75} = 0.6 \quad (8)$$

$$U / C_{th} = 0.7 \quad (9)$$

$$U = \omega D / 2 \quad (10)$$

$$\omega = \pi N / 30 \quad (11)$$

$$C_{th} = dh_T^{0.5} \quad (12)$$

where N_s is specific speed, ω is rotational angular velocity (rad/s), V_{T2} is outlet volumetric flow (m³/s), dh_T is adiabatic heat drop (J/kg), U is rotor speed (m/s), C_{th} is thermal insulation head speed (m/s), D is blade diameter (m), and N is number of rotations (min⁻¹).

N and D are derived from Eqs. (8) – (12) as follows :

$$N = (18/\pi) V_{T2}^{-0.5} dh_T^{0.75} \quad (13)$$

$$D = (42/\pi) N^{-1} dh_T^{0.5} \quad (14)$$

V_{T2} of 1.13 m³/s and dh_T of 217 kJ/kg resulted from the mass and heat balance are substituted in Eqs. (13) – (14). N and D result in 5.44×10^4 min⁻¹ and 0.115 m, respectively.

5 Absorption Bed

The mass transfer rate of CO₂ through the liquid layer on the absorption bed is expressed as follows:

$$N_{CO_2} = K_{CO_2} a (x^*_{CO_2} - x_{CO_2})_{av} S Z \quad (15)$$

where N_{CO_2} is mass transfer rate (mol/s) of CO₂, K_{CO_2} is mass transfer coefficient in the liquid film (mol/(m²·s)) of CO₂, a is gas-liquid contact area per unit volume (m²/m³), x_{CO_2} is mole fraction of CO₂ in liquid phase, $*$ is at equilibrium to the bulk gas, subscript *av* is arithmetic mean, S is cross sectional area (m²) of absorption bed, Z is height (m) of absorption bed.

The overall material balance of the absorption bed is expressed as follows :

$$N_{CO_2} = L S (x_{CO_2,1} - x_{CO_2,2}) S Z \quad (16)$$

where L is unit cross section liquid flow rate (mol/(m²·s)), and 1,2 are inlet and exit of the absorption bed, respectively.

Equations (15) and (16) are rewritten as follows :

$$ka(2L_{CO_2}^* - L_{CO_2,1} - L_{CO_2,2})SZ/2 = V(L_{CO_2,2} - L_{CO_2,1}) \quad (17)$$

where k is absorption rate constant(m/s), L_{CO_2} is CO₂ loading(mol/m³), and V is fluid flow (m³/s).

The CO₂ absorber has the random packed bed whose diameter, height, porosity, specific surface area, and material are 53.6mm, 250mm, 92.6%, 3160m²/m³, and SUS316, respectively.

A multiple regression equation approximating the ka is obtained using the experimental results as follows :

$$ka = c_1 + c_2 (V/S) + c_3 L_{CO_2,1} + c_4 T + c_5 P_{CO_2} \quad (18)$$

where T is temperature ($^{\circ}C$), P_{CO_2} is partial pressure (kPa) of CO_2 . Table 2 shows the coefficients of Eq. (18).

Table 2. Coefficients of equation 18

Coefficient	Value	Unit
c1	8.39×10^{-4}	1/s
c2	6.71×10^{-1}	1/m
c3	-7.39×10^{-6}	$m^3/(mol \cdot s)$
c4	4.27×10^{-5}	$1/(^{\circ}C \cdot s)$
c5	-2.33×10^{-5}	$1/(kPa \cdot s)$

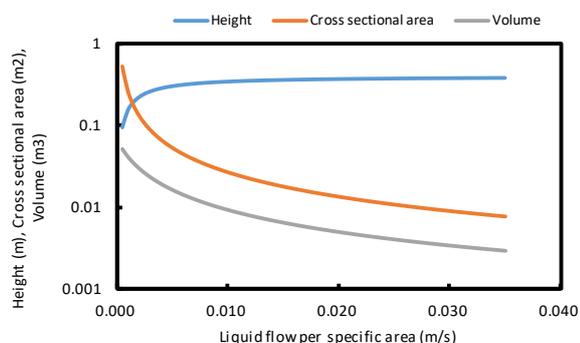


Figure 7. Height, cross section area, and volume of absorption bed as a function of liquid flow per specific area

Z is derived from Eqs. (17) and (18) and mass and heat balance of the 10kW system as follows :

$$Z = 2.69 \times 10^2 (V/S) / [1.07 + 671(V/S)] \quad (19)$$

Figure 7 shows the height, cross section area, and volume of the absorption bed as a function of liquid flow per specific area. The height of the absorption bed increases with the liquid flow per specific area. The cross sectional area and volume of the absorption bed decreases with the liquid flow per specific area. The height of the absorption bed becomes nearly constant after the liquid flow per specific area of 0.01m/s. Thus the design point results in the liquid flow per specific area of 0.01m/s. The absorption bed is the height of 0.35m and cross sectional area of 0.027m².

The pressure drop of the absorption bed is expressed using the Ergun equation :

$$\Delta p/Z = \frac{150\mu(1-\varepsilon)^2}{D_p^2 \varepsilon^3} \left(\frac{V}{S}\right) + \frac{1.75\rho(1-\varepsilon)}{D_p \varepsilon^3} \left(\frac{V}{S}\right)^2 \quad (20)$$

where Δp is pressure drop of the absorption bed (Pa), D_p is spherical equivalent particle diameter (m), ε is bed porosity (Ergun, 1952 ; Akgiary and Saatci, 2001).

If the driving force of the liquid flow through the absorption bed is only gravity, the pressure drop of the absorption bed should be less than the static pressure at the bottom of column filled by liquid.

$$\Delta p < \rho g Z \quad (21)$$

Equations 20 and 21 result in the maximum value of V/S of 0.335m/s, which is larger than the liquid flow per

specific area of 0.01m/s at the design point. Therefore the liquid is expected to flow downwards by the gravity regardless of the pressure drop.

6 Conclusion

Thermodynamic simulation of the low temperature cycle using MDEA-based solution shows the power of 10.5 kw and efficiency of 7.3% at the fluid flow rate of 1,000 kg/h and expansion ratio of 4.8 using hot water whose temperature and flow rate are 90 $^{\circ}C$ and 8,200 kg/h, respectively.

The designed recuperator is a plate heat exchanger consisting twenty plates whose width and length are 117 mm and 835 mm, respectively.

The designed turbine is a single stage radial flow turbine whose blade diameter and number of rotations are 0.115 m and $5.44 \times 10^4 \text{ min}^{-1}$, respectively.

The designed absorption bed is the height of 0.35 m and cross sectional area of 0.027 m² at the liquid flow per specific area of 0.01 m/s.

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