

Influence of Flash Temperature on Exergetical Performance of Organic Flash Cycle

Kyoung Hoon Kim and Chul Ho Han

Abstract—Organic Flash Cycle (OFC) has potential of improving efficiency for recovery of low temperature heat sources mainly due to reducing temperature mismatch in the heat exchanger. In this work exergetical performance analysis of ORC is conducted for recovery of low grade heat source. Effects of system parameters such as flash evaporation temperature or heating temperature are theoretically investigated on the exergy destructions (anergies) at various components of the system as well as exergy efficiency. Results show that exergy efficiency has a peak with respect to the flash temperature, and the optimum flash temperature increases with the heating temperature. The component where the largest exergy destruction occurs varies with the flash temperature or heating temperature.

Keywords—Organic flash cycle (OFC), low grade heat source, exergy, anergy, flash temperature.

I. INTRODUCTION

LIMITING green house gas emissions at the global level will require a major shift away from fossil fuels and a dramatic increase in the use of renewable energy, nuclear power and carbon capture and storage as well as greater energy efficiency. The International Energy Agency (IEA) has identified the development of renewable energy sources as a key element to mitigate climate change [1]. As it is impossible to converse low grade energy to electricity efficiently by conventional methods, most of the low grade energy is just discarded. Therefore the research is important how to generate electricity efficiently from low grade energy sources. In recent years the organic Rankine cycle (ORC) and the power generating system using binary mixture as a working fluid have attracted much attention as they are proven to be the most feasible methods to achieve high efficiency in converting the low-grade thermal energy to more useful forms of energy [2], [3].

Schuster et al. [4] mention numerous running applications, such as geothermal power plant, biomass fired combined heat and power plants, solar desalination plants, waste heat recovery or micro CHP. Drescher and Bruggemann [5] investigate the ORC in solid biomass power and heat plants. They propose a method to find suitable thermodynamic fluids for ORCs in biomass plants and found that the family of alkylbenzenes showed the highest efficiency. Dai et al. [6] use a generic

optimization algorithm, identified isobutane and R236ea as efficient working fluids. Heberle and Brueggemann [7] investigate the combined heat and power generation for geothermal re-sources with series and parallel circuits of an ORC. Tranche et al. [8] investigate comparatively the performance of solar organic Rankine cycle using various working fluids. Volume flow rate, mass flow rate, power ratio as well as thermal efficiency are used for comparison. Hung et al. [9] examine Rankine cycles using organic fluids which are categorized into three groups of wet, dry and isentropic fluids. They point out that dry fluids have disadvantages of reduction of net work due to superheated vapor at turbine exit, and wet fluids of the moisture content at turbine inlet, so isentropic fluids are to be preferred.

In ORC temperature matching to the source stream is important in minimizing the irreversibilities caused by heat transfer across a finite temperature difference. When using the ORC to produce electricity from a finite thermal reservoir though, temperature mismatching often is inevitable because the source stream is single-phase and possesses a near linear temperature profile along the heat exchanger. Recently, Ho et al. [10], [11] proposed a novel cycle named as organic flash cycle (OFC). In the cycle heat addition occur with the cycle working fluid remaining in the liquid state. In other words, heat is transferred to the ORC until the working fluid reaches a saturated liquid state. The fluid would then be flash evaporated to produce a two-phase mixture; the saturated vapor would be separated and then expanded to produce power.

Exergy is a measure of the departure of the state of a system from that of the environment, and the method of exergy analysis is well suited for furthering the goal of more energy resource use, for it enables the location, cause, and true magnitude of waste and loss to be determined [12]–[15]. In this work exergetical performance analysis for OFC is conducted for use of low grade heat source in the form of sensible energy by using isobutane as the working fluid. Effects of operating temperatures on the exergy destruction at each component of the system as well as the exergy efficiency were theoretically investigated.

II. SYSTEM ANALYSIS

The schematic diagram of the system is shown in Fig. 1. The system consists of pump, heat exchanger, flash evaporator, turbine, mixer and condenser. A low grade heat source is supplied to the system as sensible heat energy. The working fluids considered in this work is isobutane. In this work the thermodynamic properties of the working fluids are calculated using the Patel-Teja equation of state [16], [17]. The basic data

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of the fluid which are needed to calculate the thermodynamic properties are as follows; molar mass (M) is 58.123kg/kmol, critical temperature (T_c) 408.14K, critical pressure (P_c) 3.648 MPa, and accentric factor (ω) 0.177 [18].

The coolant enters the condenser at temperature of T_c and the working fluid leaves the condenser as saturated liquid at T_L (point 1) where the corresponding saturation pressure P_L is the low pressure of the system. The fluid is compressed with the pump to P_H which is the high pressure of the system (point 2). The heat source is supplied at temperature of T_s and the fluid is heated with the heat exchanger and leaves the heat exchanger as saturated liquid at temperature, $T_3 = T_H$ (point 3). It is assumed that the minimum temperature difference between hot and cold streams in the heat exchanger is the prescribed value of pinch point, ΔT_{pp} . Then the fluid is flashed at temperature of FET in the flash evaporator and separated into saturated vapor (point 4) and saturated liquid (point 6) where the corresponding pressure is the intermediate pressure of the system, P_1 . The vapor is expanded in the turbine to the pressure of P_L (point 5). The liquid from the flash evaporator is throttled to pressure of P_L , mixed with the fluid from the turbine and enters the condenser (point 7). The thermodynamic properties at points 2 and 5 can be obtained in terms of the isentropic efficiencies of pump and turbine, η_p and η_t , respectively.

The ratio of mass flow rate of working fluid to source, r_s and that of coolant to working fluid, r_c can be determined as

$$r_s = \frac{\dot{m}_{wf}}{\dot{m}_s} = \frac{c_{ps}(T_s - T_{s,out})}{h_3 - h_2} \quad (1)$$

$$r_c = \frac{\dot{m}_c}{\dot{m}_{wf}} = \frac{h_7 - h_1}{c_{pc}(T_{c,out} - T_c)} \quad (2)$$

where subscripts wf, s and c denote the working fluid, the source fluid and outlet, respectively, and m the mass flow rate, T the temperature, h the specific enthalpy, and c_{ps} and c_{pc} the constant pressure specific heat of source and coolant, respectively.

Then the rate of heat addition \dot{Q}_{in} , heat rejection \dot{Q}_{out} , and net work \dot{W}_{net} are obtained as

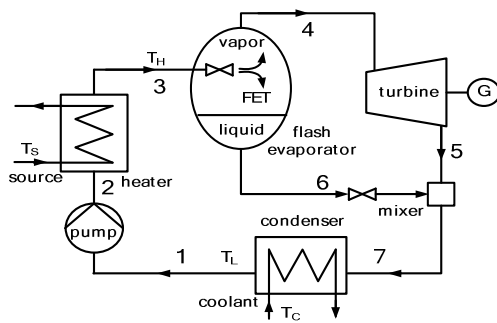


Fig. 1 Schematic diagram of the system

$$\dot{Q}_{in} = \dot{m}_{wf}(h_3 - h_2) \quad (3)$$

$$\dot{Q}_{out} = \dot{m}_{wf}(h_7 - h_1) \quad (4)$$

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p = \dot{m}_{wf}[y_{fe}(h_4 - h_5) - (h_2 - h_1)] \quad (5)$$

where y_{fe} denotes the separation ratio which is the quality of working fluid at flash evaporator, and subscripts t and p turbine and pump, respectively.

The specific exergy e and the rate of exergy input \dot{E}_{in} to the system by source fluid can be calculated as

$$e = h - h_0 - T_0(s - s_0) \quad (6)$$

$$\dot{E}_{in} = \dot{m}_s c_{ps} \{T_s - T_0 - T_0 \ln(T_s / T_0)\} \quad (7)$$

where s is the specific entropy and subscript 0 refers the dead state. Then, the exergy efficiency of the system η_{ex} , which is defined as the ratio of net work to exergy input, can be written as follows.

$$\eta_{ex} = \dot{W}_{net} / \dot{E}_{in} \quad (8)$$

The exergy destruction or anergy of the adiabatic system is calculated as the difference of exergy input and output. The anergy ratio at a system component is defined as the ratio of anergy there to the exergy input by source fluid. Then summation of all anergy ratios of the system and the exergy efficiency becomes unity.

$$\eta_{ex} + D_{he} + D_{ho} + D_{cd} + D_{co} + D_{wnet} + D_{fe} + D_{mix} = 1 \quad (9)$$

where D_{he} , D_{ho} , D_{cd} , D_{co} , D_{wnet} , D_{fe} , and D_{mix} are anergy ratio of the heat exchanger, source exhaust, condenser, coolant exhaust, net work (turbine and pump), flash evaporator and mixer, respectively.

III. RESULTS AND DISCUSSIONS

The system parameters used in this work are summarized in Table I. In this work the basic data for analysis are $T_s = 300^\circ\text{C}$ and $T_0 = 25^\circ\text{C}$. The ratio of exergy destruction or anergy at flash evaporator is plotted against the flash evaporation temperature (FET) in Fig. 2 for various heating temperatures (T_H 's). The heating temperature varies from 100°C to the critical temperature of the fluid of 134.99°C . The anergy ratio increases with decreasing FET or increasing heating temperature, since the decrease in FET or increase in heating temperature leads to greater pressure difference during the flash process and consequently larger irreversibility. Thus the anergy ratio is the greatest when the heating temperature reaches the critical temperature because of the largest temperature difference. However, it can be seen from the figure that there exists a low limit of FET for the case of $T_H = T_c$. The reason is as follows. The separation ratio can be obtained as $y_{fe} = (h_3 - h_6) / (h_4 - h_6)$. Therefore, in order that the cycle is operable, the specific enthalpy of working fluid at the heating temperature should be greater than saturated liquid and smaller than saturated vapor, namely, there is a restriction of $h_6 < h_3 < h_4$ for proper operation.

TABLE I
BASIC CALCULATION CONDITIONS

symbol	Parameter	data	unit
T_s	source temperature	300	°C
T_c	coolant temperature	25	°C
T_l	condensing temperature	40	°C
ΔT_{pp}	pinch point	10	°C
η_p	isentropic efficiency of pump	0.80	
η_t	isentropic efficiency of turbine	0.80	
	source	water	

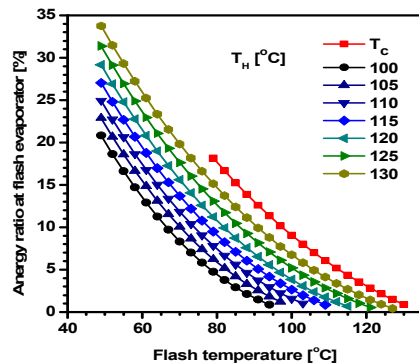


Fig. 2 Anergy ratio at flash evaporator

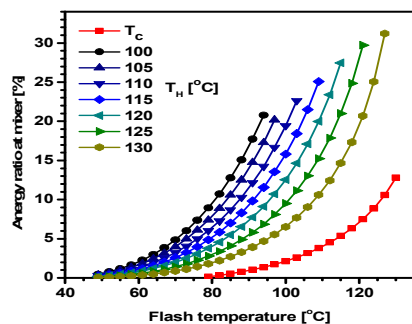


Fig. 3 Anergy ratio at mixer

Thus the separation ratio increases as FET decreases, and there may be the lower limit of FET for larger heating temperatures. The anergy ratio at mixer is shown in Fig. 3. The anergy ratio decreases with decreasing FET or increasing heating temperature. It is worthy to note that the phenomenon is the reversal to the case of the anergy ratio at flash evaporator. This can be explained as follows. As the heating temperature for a fixed FET, the difference between thermodynamic properties of saturated liquid and vapor decreases (the difference is reduced to zero at the critical point), which causes smaller irreversibility and anergy ratio during the mixing process. When FET decreases for a fixed heating temperature, the pressure ratio decreases difference, which leads to smaller difference between thermodynamic properties of the working fluid at inlet and outlet of the turbine, thus irreversibility decreases during the mixing process.

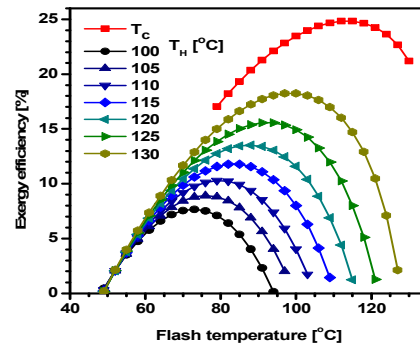


Fig. 4 Exergy efficiency as a function of FET

Exergy efficiency is defined as the ratio of net work production to the exergy supplied to the system, and its dependency on FET is shown in Fig. 4 for various heating temperatures. For a specified value of heating temperature, exergy efficiency has a peak with respect to FET. This is because as FET increases, there are both increasing and decreasing effects together.

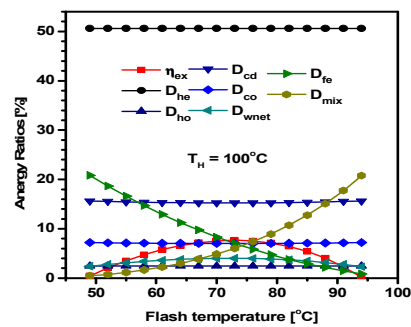


Fig. 5 Anergy ratios and exergy efficiency for $T_H = 100^\circ\text{C}$

The increasing effect is that the pressure ratio of turbine increases with increasing FET, and consequently net work production per unit mass of working fluid increases. On the other hand, the decreasing effect is that the separation ratio in the flash evaporator decreases with increasing FET, and it causes reduction of mass flow rate of working fluid relative to the source fluid. It can be seen from the figure that the optimum value of FET for maximum exergy efficiency increases with the heating temperature. For a given value of FET, exergy efficiency increases with the heating temperature. Thus, the system shows the best performance when the heating temperature reaches the critical temperature of the fluid.

Figs. 5 and 6 show the anergy ratios of the components of the system as well as exergy efficiency when the heating temperature is 100°C and T_c , respectively. When the heating temperature is 100°C , anergy ratio at heat exchanger is about 50% which is the biggest among the components of the system over the range of operating conditions. However, the overall order of magnitudes of anergy ratios and exergy efficiency varies with FET. For example, the order of the ratios in

magnitude is D_{he} , D_{fe} , D_{cd} when $FET = 50^\circ C$, D_{he} , D_{cd} , D_{fe} when $FET = 60^\circ C$, D_{he} , D_{cd} , D_{mix} when $FET = 80^\circ C$, and D_{he} , D_{mix} , D_{cd} when $FET = 90^\circ C$. When the heating temperature reaches the critical temperature of the fluid ($T_H = T_c$), the energy ratio of heat exchanger is about 34% which is also the biggest value among the components of the system, however, the magnitude is much less than that for $T_H = 100^\circ C$. The order of the ratios in magnitude varies also with FET, so D_{he} , D_{fe} , η_{ex} , D_{cd} when $FET = 80^\circ C$, D_{he} , η_{ex} , D_{cd} , D_{fe} when $FET = 100^\circ C$, D_{he} , η_{ex} , D_{cd} , D_{wnet} when $FET = 110^\circ C$, and D_{he} , D_{cd} , D_{mix} , D_{wnet} when $FET = 130^\circ C$. As is seen from the examples about order of magnitude of the ratios, therefore, the exergetical performance of the system is influenced by the operating temperatures very sensitively. However, it is worthy to note that as the exergy destruction at heat exchanger is the biggest among the components, it should be further reduced in order to improve the system performance. The improving way may be operating optimal temperature and pressure conditions, or operating modified cycles, or reducing temperature mismatch further by operation with supercritical conditions.

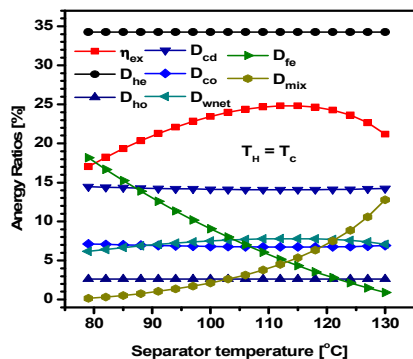


Fig. 6 Energy ratios and exergy efficiency for $T_H = T_c$

IV. CONCLUSION

In this paper, the exergetical performance analysis on Organic Flash Cycle (OFC) was conducted. The main results are as follows:

- 1) Energy ratio at flash evaporator increases with decreasing flash evaporation temperature (FET) or increasing heating temperature, T_H . On the contrary, energy ratio at mixer decreases with decreasing FET or increasing heating temperature
- 2) Exergy efficiency has a peak with respect to FET, and increases with the heating temperature. Therefore, best exergetical performance can be achieved when the heating temperature reaches the critical temperature of the fluid.
- 3) Energy ratios of the components of the system as well as exergy efficiency vary sensitively with the operating temperatures of FET and heating temperature.
- 4) In order to improve the system performance, energy at heat exchanger should be further reduced.

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