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PERFORMANCE ANALYZING OF AN ORGANIC RANKINE CYCLE UNDER DIFFERENT AMBIENT CONDITIONS

A. V. Akkaya[1,*]

ABSTRACT

The goal of this study is to develop a thermodynamic model in order to show the effect of ambient conditions on performance of an Organic Rankine Cycle based power generation system. This system is simply consisted of a turbine, a condenser, a boiler, pumps and a cooling tower. Each component in this system is modeled based on energy and mass balance equations. Then, the system model is obtained with integration of component models. After that, simulation studies are iteratively carried out to determine the performance of the considered system under variation of ambient conditions such as dry bulb temperature and relative humidity. Simulation results show that that Organic Rankine Cycle performance can be sensitive to the seasonal and daily variation of the ambient conditions.

Keywords: *Organic Rankine cycle, Cooling tower, Ambient Conditions, Performance*

INTRODUCTION

For power production from low-grade heat sources, Organic Rankine Cycles (ORCs) are considered as a promising option because of its high thermal efficiency and flexibility [1-3]. In the ORC applications, organic fluids are utilized as working fluid. Organic fluids can provide low boiling points and this skill results in improving the thermal efficiency in low temperature applications.

When compared to Steam Rankine Cycles (RC), Organic Rankine Cycle (ORC) is feasible alternative power generation system in the small/medium scale unit sizes for specific applications such as solar energy systems [4,5], waste heat recovery systems [6-10], biomass based systems[11] and geothermal energy systems [12,13].

In the literature, a number of investigations related to ORC systems were carried out for working fluid selections [14-16] and parameter optimizations [17-19]. Furthermore, many studies emphasized that an ORC performance was sensitive to the variation in the heat source temperature, the working fluid flow rate and the load [20,21]. Additionally, the cooling fluid temperature at the condenser inlet is an important parameter influencing on the ORC performance [22]. The power output and thermal efficiency increase when condenser pressure decreases. It should be noted that condenser pressure is depended on ambient condition and heat rejection method.

In the present work, an Organic Rankine Cycle based power generation system, which uses the thermal energy of exhausted gases from an industrial plant, is examined. The ORC system uses R123 as a working fluid. This study focus on the development of a thermodynamic model in order to investigate steady-state working characteristic and performance of the ORC system under ambient conditions. Dry bulb temperature and relative humidity are varied in the specific range, which simulates seasonal and daily variations. Under these conditions, the key system parameters, which mainly includes cooling water temperature, condenser pressure and thermal efficiency, are analyzed parametrically.

THE CONSIDERED ORC SYSTEM

The simplified configuration of the Organic Rankine Cycle based system taken into account in this investigation is illustrated in Figure 1. The considered system consists of six main components that are a heat recovery unit (HRU), a turbine (T), a condenser (C), two pumps (P), an electrical generator (G) and a cooling tower (CT).

The working principle of the considered system can be explained as following: The feed fluid pump pressurizes the organic fluid. Then, the pressurized fluid enters the HRU. Point 1 and 2 denote the inlet and outlet

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[1]Department of Mechanical Engineering, Yildiz Technical University, Istanbul, TURKEY
**E-mail address: aakkaya@yildiz.edu.tr*
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of the economizer, whereas point 3 and 4 represent inlet and outlet of the super-heater, respectively. The way of the waste gas through the HRU is shown with points of A, B, C and D. Particularly, A and D represent inlet and outlet of the HRU, respectively. The waste gas at the economizer of HRU firstly heats the fluid. After that, the fluid is vaporized and super-heated at the evaporator and super-heater section of HRU, respectively. The super-heated and pressurized organic vapor goes to the turbine.

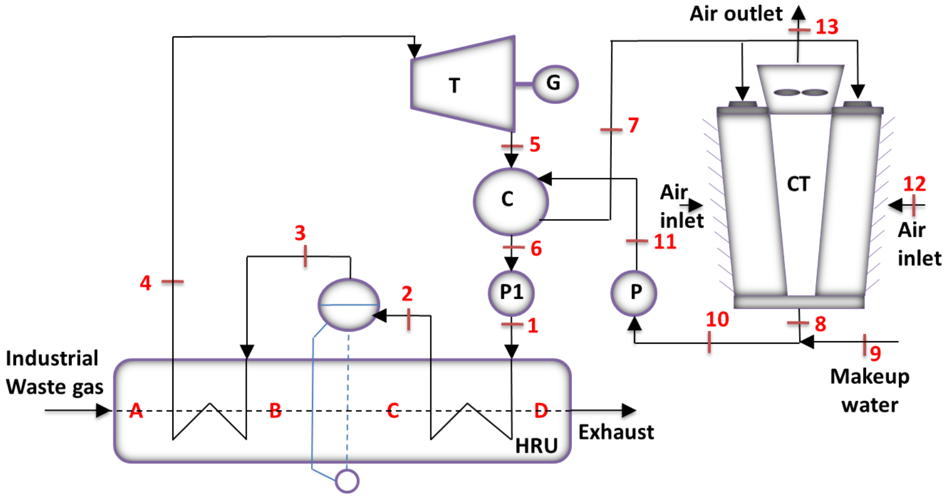

Figure 1. The layout of Organic Rankine Cycle based system

In order to obtain mechanical power, the vapor expands in the turbine (T) and the mechanical power is obtained. The generator (G) shifts the mechanical power to electrical power. At the turbine outlet, the pressure and temperature of the working vapor are decreased. In the condenser component (CND), the vapor is condensed at the specific pressure and is send to pump. Then, the ORC cycle starts again.

At the condenser, cooling water takes heat of vapor. Therefore, cooling water temperature is increased. This state is represented by 7 point. Then, it goes to cooling tower (CT). In the cooling tower, hot cooling water is cooled by ambient air entering cooling tower (12). Cooling degree is limited by ambient air conditions. During cooling process, some part of cooling water is vaporized and they exit the cooling tower together as a high degree humidified air flow (13). The makeup water (9) is added to the cooled cooling water (8) and they are sent to circulating pump (10). Then, the pressurized cooling water enters the condenser (11). Condenser pressure is determined by the cooling water. Finally, it cools the organic vapor coming from turbine.

THERMODYNAMIC MODEL

Since the aim of this study is to analyze the ORC performance under variation of ambient condition, a thermodynamic model development of the ORC and cooling tower system is important. Some assumptions are made in order to develop the thermodynamic model of the ORC system. These assumptions are [23]:

a) All cycle components are supposed to work in adiabatic and steady-state conditions,

b) Pressure drops in the pipe networks are neglected,

c) The effects of potential and kinetic energy changes are neglected in the energy balance equations,

d) The turbines and pumps have isentropic efficiencies,

e) It is assumed that the waste gas behaves as an ideal gas mixture.

Under these assumptions, mass and energy balance equations for any control volume can be expressed as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \quad (2)$$

where, $\dot{Q}$ is heat transfer rate between the control volume and its surroundings, $\dot{W}$ is electrical power output, $\dot{m}$ is mass flow rate and h is the specific enthalpy .

Accordingly, thermodynamic model for the ORC based system can be formed by the related the component equations based on mass and energy balances mentioned above. When this model is constituted, thermodynamic properties such as temperature, pressure and enthalpy at each point of the considered ORC based system can be calculated. In order to compute the turbine power output, the enthalpy value at the turbine exit should be known which depends on condenser pressure. Condenser pressure is determined by cooling water temperature at the condenser inlet. This relation is given by an empirical equation as following:

$$P_{cnd} = 1.6732 + 0.0009 T_{w,in} + 0.0009 T_{w,in}^2 \tag{3}$$

Ambient conditions such as dry bulb temperature and relative humidity are not constant. These conditions may change daily and seasonally. Therefore, cooling water temperature value at the condenser inlet varies depending on the ambient conditions. In this study, to calculate cooling water temperature, a cooling tower is modeled by considering ε-NTU method is. The detail of model can be found in the Ref. [24]. When condenser heat rate, air flow rate, dry bulb temperature and relative humidity are given, the cooling water temperature at the cooling tower outlet can be determined by this model. Then, the cooling water temperature can be computed from:

$$\dot{Q}_{cnd} = \dot{m}_{w,in}\, c_{p,w} T_{w,in} - \dot{m}_{w,out}\, c_{p,w} T_{w,out} \tag{4}$$

After integrating the ORC model and cooling tower model, the integrated system model needs to be iteratively solved. Then, the net power output and thermal efficiency under different ambient conditions can be determined respectively as following equations:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \tag{5}$$

$$\eta_{th} = \dot{W}_{net} / (\dot{Q}_A - \dot{Q}_D) \tag{6}$$

SIMULATION RESULTS AND DISCUSSION

In this section, the effect of ambient conditions such as air dry bulb temperature and relative humidity on performance of Organic Rankine Cycle (ORC) based system recovering waste gas heat is analyzed by the developed thermodynamic model. Table 1 presents the model input parameters. Engineering Equation Solver (EES) environment [25] is used to code the thermodynamic model.

Waste gas composition includes CO_2, N_2, O_2 and H_2O. The working organic fluid is R123. Simulation of the coded thermodynamic model is carried out by using model input parameters. System net power output is found as 303.5 kW while thermal efficiency is computed as 10.8 %. Besides, thermal energy discharged from the condenser was 2514 kW. Finally, thermodynamic properties at each state, calculated by the simulation, are given in Table 2.

Table 1. Main model input parameters

Input Parameters	Symbol	Value	Unit
Waste gas inlet temperature	T_A	177	^oC
Waste gas mass flow rate	$\dot{m}_g$	38.5	kg/s
Organic vapour pressure	P	10	bar
Turbine isentropic efficiency	η_{tis}	80	%
Pump isentropic efficiency	η_{pis}	65	%
Dry bulb temperature	T_{db}	25	^oC
Relative humidity	ϕ	50	%
Air inlet mass flow rate	$\dot{m}_a$	144	kg/s
Mass ratio of cooling water/air	r	1.5	-

Table 2. Thermodynamic properties at each state point

State Point	$\dot{m}$ (kg/s)	P (bar)	T (°C)	h (kJ/kg)
1	12.73	10.00	48.90	251.30
2	12.73	10.00	111.10	321.80
3	12.73	10.00	111.10	448.40
4	12.73	10.00	136.90	472.70
5	12.73	2.00	94.30	448.00
6	12.73	2.00	48.00	250.40
7	216.20	1.30	21.30	89.50
8	215.20	1.10	18.61	78.20
9	216.20	1.10	18.62	84.00
10	1.00	1.10	18.62	78.23
11	216.20	1.30	18.62	78.26
12	144.10	1.00	20.00	38.51
13	145.10	1.00	19.84	55.95
A	38.50	1.10	177.00	187.90
B	38.50	1.10	169.50	179.90
C	38.50	1.05	131.10	138.00
D	38.50	1.05	109.40	114.70

Then, in order to analyze the effect of the air conditions on the considered ORC system performance, parametric analysis was performed. In the parametric analysis, air dry bulb temperature is varied between -10 °C and 45 °C while all other input parameters are considered as constant. Additionally, relative humidity is changed from 10% to 90%. By this way, the variation of model output parameters is computed and the important results are plotted in the Figure 2, Figure 3 and Figure 4.

Figure 2 shows the variation of cooling water temperature at the condenser inlet in terms of dry bulb temperature and relative humidity. It is clearly seen from this figure that increasing dry bulb temperature increases cooling water temperature. For the investigated dry bulb temperature range and 50% relative humidity, cooling water temperature changes from 0 °C to 36 °C.

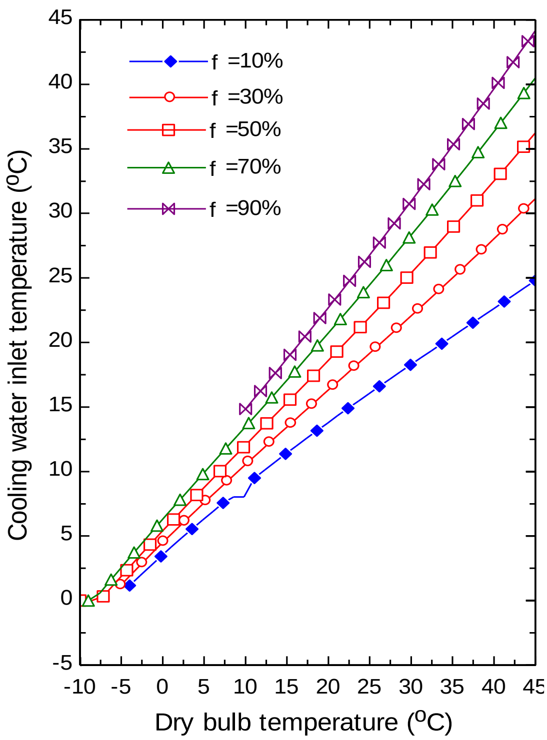

Figure 2. Variation of cooling water temperature at the condenser inlet with regard to air dry bulb temperature and relative humidity

In addition, we can observe from Figure 2 that at lower dry bulb temperatures, effect of varying relative humidity on cooling water temperature becomes lower. In other words, relative humidity can not influence considerably in such cases. On the other hand, at the higher dry bulb temperatures, the change of relative humidity can influence significantly on the cooling water temperature. For example, when dry bulb temperature is 45 ºC, cooling water temperature becomes 25 ºC at the 10% relative humidity. When the relative humidity is 90% at the same dry bulb temperature, cooling water temperature is equal to 44 ºC. It is seen from that the temperature difference is 19 ºC. Moreover, at the same high dry bulb temperatures (for instance 45 ºC), the temperature difference of cooling water is not the same with the same relative humidity increase amount. For example, when relative humidity changes from 10% to 30%, the temperature difference of cooling water is 6 ºC (31-25). However, the temperature difference of cooling water is 3ºC (44-41) when relative humidity changes from 70% to 90%.

The effect of dry bulb temperature and relative humidity on the condenser pressure is illustrated in the Figure 3. It can be understood from this figure that elevation of relative humidity value rises condenser pressure at the same dry bulb temperatures. Pressure increase rate becomes bigger when dry bulb temperature reaches higher degree. Furthermore, at the lower dry bulb temperatures condenser pressure does not change so much. For example, condenser pressure value is 1.6 bar averagely at the 0 ºC dry bulb temperature condition for all relative humidity values.

Figure 4 demonstrate the variation of system thermal efficiency with respect to dry bulb temperature and relative humidity. It is obviously seen from this figure that the thermal efficiency decreases when dry bulb temperature increases. For example, when relative humidity is considered as 50%, thermal efficiency is equal to 11.6 % at the -10 ºC dry bulb temperature. For the same relative humidity value, thermal efficiency becomes 8.8% where dry bulb temperature is 45 ºC. In the considered dry bulb temperature range, the difference of thermal efficiency is about 2.8% point.

In addition, at the same higher dry bulb temperatures, increasing relative humidity value results in decreasing thermal efficiency as shown in the Figure 4. For instance, in the range of 10%-90% relative humidity at the 45 ºC of dry bulb temperature, thermal efficiency drops from 10.2% to 7.75%. On the contrary, it is shown that at the lower dry bulb temperatures, changing relative humidity value does not affect significantly the thermal efficiency.

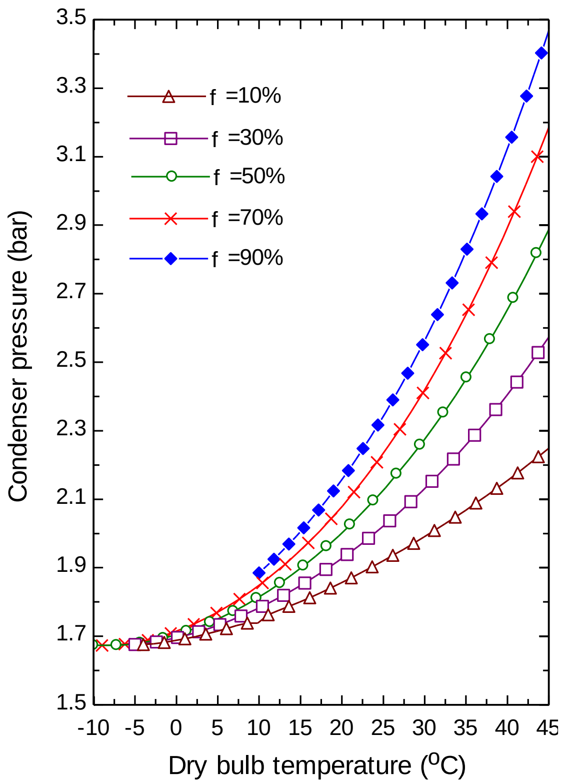

Figure 3. Variation of condenser pressure with regard to air dry bulb temperature and relative humidity

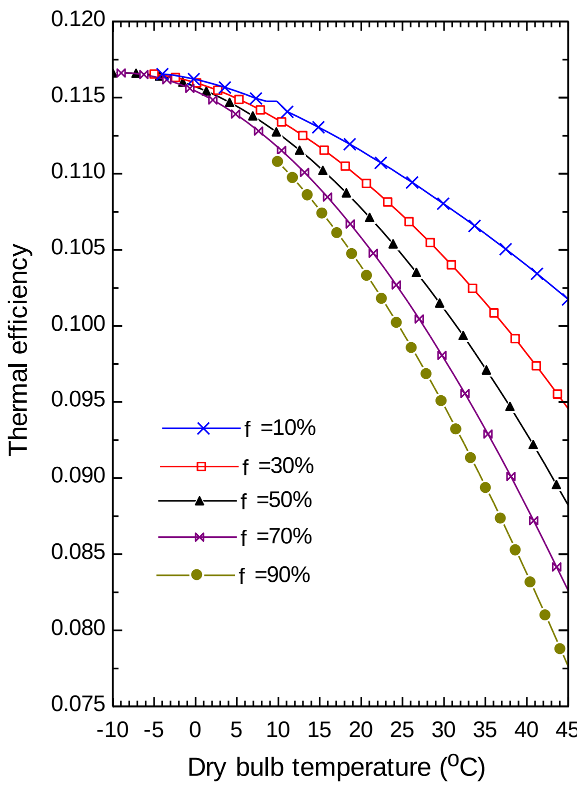

Figure 4. Variation of thermal efficiency with regard to air dry bulb temperature and relative humidity

CONCLUSION

In this study, performance of an ORC based power generation system is evaluated under the variation of air conditions. For this aim, a thermodynamic model has been developed. Based on simulation studies, it is found that varying ambient conditions can influence considerably the cooling water temperature at condenser inlet. Since cooling water temperature determines condensation pressure, power output and thermal efficiency change greatly by the variation of the pressure. This conclusion indicates that the ORC performance could be sensitive to the seasonal and daily variation of the ambient conditions. In order to avoid an excessive performance variation due to ambient condition, a proper design and operation of both ORC and cooling tower system should be implemented.

NOMENCLATURE

h	Specific enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{W}$	Power output (kW)
P	Pressure (bar)
s	Specific entropy (kJ/kg-°C)
T	Temperature (°C)
y	Mole fraction
$\dot{Q}$	Thermal energy rate (kW)

Greek Letters

η	Efficiency

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