MODELING THERMODYNAMIC ANALYSIS AND SIMULATION OF ORGANIC RANKINE CYCLE USING GEOTHERMAL ENERGY AS HEAT SOURCE

Colak L.* and Bahadir T.
*Author for correspondence
Department of Mechanical Engineering,
Baskent University,
Ankara, 06810,
Turkey,
E-mail: lcolak@baskent.edu.tr

ABSTRACT

In this study, modeling and thermodynamic analysis of an Organic Rankine Cycle using geothermal heat source in Aydın, Turkey has been made and optimum operation conditions were determined by developing simulation software. The model has been validated using an existing study. In consequence, the differences between existing study data and the model seems reasonable close, so the model was verified. Simulation of model has been made by using EES software. In simulation the effect of minimum and maximum pressure and working fluid changes to the system performance has been investigated. ORC using working fluids R141b and R123 and Isopentane has been simulated for selected input values separately. Because of using Isopentane caused geothermal re-injection temperature decrease to inconvenient level, it has not been used as working fluid in simulation. For R141b and R123, maximum and minimum pressure values have been changed from 500 kPa to 3500 kPa and from 100 kPa to 300 kPa, respectively. 2331 kW of net work obtained at 3184 kPa maximum pressure and 2749 kW of net work obtained at 100 kPa minimum pressures for R141b. For R123, 1798 kW of net work obtained at 3184 kPa maximum pressure and 2119 kW of net work obtained at 100 kPa minimum pressures. It has been seen that reducing minimum pressure effects net work more than increasing maximum pressure and R141b has a better performance than R123, especially in the way of net work production.

NOMENCLATURE

$\dot{Q}$	[kW]	Heat transfer rate
$\dot{W}$	[kW]	Power
h	[kJ/kg]	Specific enthalpy
$\dot{m}$	[kg/s]	Mass flowrate
$\dot{E}x$	[kW]	Flow exergy rate
ex	[kJ/kg]	Specific flow exergy
$\dot{E}_D$	[kW]	Exergy destruction rate
T	[°C]	Temperature
P	[kPa]	Pressure
s	[kJ/kg.K]	Specific entropy
$\dot{I}$	[kW]	Irreversibility rate
X	[-]	Quality
EES		Engineering Equation Solver
Special characters		
ε	[%]	Exergetic efficiency
η	[%]	Energy efficiency
Subscripts		
cw		Cooling water
geo		Geothermal
p		Pump
t		Turbine
e		Evaporator
c		Condenser
wf		Working fluid
th		Thermal
i		Inlet
max		Maximum
min		Minimum

INTRODUCTION

Organic Rankine Cycle (ORC) is a vapour power cycle that use low temperature heat sources, such as waste heat, solar energy, geothermal energy, biomass etc. These cycles have low operational cost and low Carbon emission since necessary heat is obtained without combustion process. Also fossile fuel consumption can be decreased by the use of ORC.

In literature, Quoilin et.al. developed Organic Rankine Cycle applications for geothermal energy, solar energy and for biomass and for waste heat sources and made technoeconomic analysis of Organic Rankine Cycles [1]. Huijuan et.al. analyzed different working fluids that can be used in Organic Rankine Cycles for determining the effectiveness of organic working fluids for different conditions and applications [2]. Dipippo studied on ideal thermal efficiencies of geothermal sourced Organic Rankine Cycles and obtained a thermal efficiency range of % 8 -16 for different heat source temperatures [3]. Quoilin developped a dynamic model for solar and waste heat sourced Organic Rankine Cycles and analyzed the time rate of change of this system in his PhD thesis [4]. Bundela and Chawla pointed out that Organic Rankine Cycle was feasible for low and medium temperature waste heat recovery in a cement factory in Japan [5]. Li et.al. performed energy and exergy analysis of Organic Rankine Cycles for different heat source temperatures in between 70 °C and 100 °C [6]. Wu et.al. realized an optimization study for optimal compression point temperature difference of evaporator in heat recovery applications. In this study the effect of flow exergy losses on compression point temperature difference were analyzed [7]. Lui et.al. analyzed the performance analysis of ORC systems for different hydrocarbon working fluids, where the geothermal inlet temperature was in between 100 °C and 150 °C and the exit temperature was 70 °C [8].

In this study a simulation was done for analyzing the utilization of geothermal energy in electricity production by using an ORC in Aydın, Turkey. The temperature range of geothermal source in Aydın varies in between 203 °C and 232 °C. Although the maximum reversible (Carnot) thermal efficiency was calculated as % 38,92 for 203 °C heat source temperature and 17,7 °C average ambient temperature in Aydın, there are many other parameters that affects the real thermal efficiency such that maximum and minimum cycle pressures, working fluid types, besides the heat source and ambient temperature.

MATHEMATICAL MODEL

The ORC model in this study was shown in Figure 1. It use geothermal energy as heat source and has four main components such as evaporator, turbine, condenser and pump.

The assumptions, that were done for the thermodynamic solution of ORC model, were given as follows;

- Cycle operates in steady state steady flow condition,
- Heat losses in pump, evaporator and condenser were neglected [9],
- The heat loss from turbine was taken as %5 due to a similar study which was done by Li et al. [10],

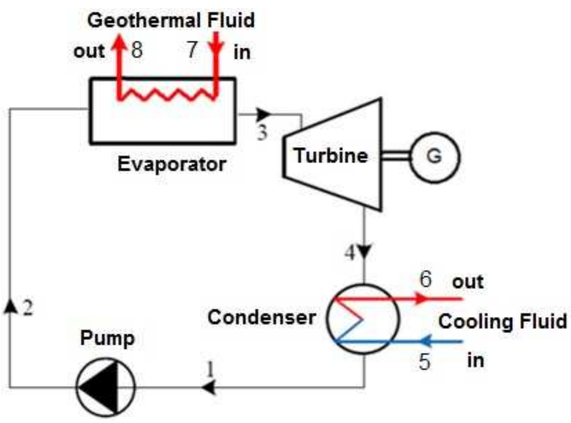

Figure 1 Organic Rankine Cycle

- Isentropic and dry fluids was taken as working fluids in analysis,
- Since superheat was not necessary for isentropic and dry fluids, the state at turbine inlet was taken as saturated vapour,
- Working fluid was taken as saturated liquid at pump inlet,
- The isentropic efficiencies of turbine and pump was taken as % 85 and % 80 respectively,
- Geothermal source was hot water,
- Thermophysical properties of working fluids and geothermal source was assumed as constant,
- The effect of pressure reduction does not affect evaporation temperature,
- Evaporator and condenser were shell and tube type heat exchangers,
- Dead state was taken as 17,7 °C and 101,3 kPa.

The thermodynamic equations that will be used in simulation were given in Table 1. By using these governing equations energy and exergy analysis of each component of ORC was done.

Table 1 Mathematical model governing equations

Sub System	Energy Equations	Exergy Equations
Evaporator	$\dot{Q}_e = \dot{m}_{wf}(h_3 - h_2)$	$\varepsilon_e = \frac{\dot{Ex}_3 - \dot{Ex}_2}{\dot{Ex}_7 - \dot{Ex}_8}$
		$\dot{Ex}_{D,e} = (\dot{Ex}_7 - \dot{Ex}_8) - (\dot{Ex}_3 - \dot{Ex}_2)$
Turbine	$\dot{W}_t = 0{,}95\dot{m}_{wf}(h_3 - h_4)$	$\varepsilon_T = \frac{\dot{W}_t}{\dot{Ex}_3 - \dot{Ex}_4}$
		$\dot{Ex}_{D,t} = (\dot{Ex}_3 - \dot{Ex}_4) - \dot{W}_t$
Pump	$\dot{W}_p = \dot{m}_{wf}(h_2 - h_1)$	$\varepsilon_p = \frac{\dot{Ex}_2 - \dot{Ex}_1}{\dot{W}_p}$
		$\dot{Ex}_{D,p} = \dot{W}_p - (\dot{Ex}_2 - \dot{Ex}_1)$
Condenser	$\dot{Q}_c = \dot{m}_{wf}(h_2 - h_1)$	$\varepsilon_y = \frac{\dot{Ex}_6 - \dot{Ex}_5}{\dot{Ex}_4 - \dot{Ex}_1}$
		$\dot{Ex}_{D,c} = (\dot{Ex}_4 - \dot{Ex}_1) - (\dot{Ex}_6 - \dot{Ex}_5)$

SIMULATION

The simulation was done by using EES® software which was developed by F-Chart Software Corporation. It was efficacious software for thermodynamic analysis, since thermodynamic properties of various substances were inside.

For the simulation of ORC, necessary codes were written in EES by taking the governing equations and assumptions into consideration. In this simulation, the effects of different working fluids and minimum and maximum cycle pressures on the net work and thermal and exergetic efficiencies of cycle were analyzed.

Validation of Mathematical Model

The mathematical model used in simulation was validated by using the input and output data of a previously done study for exergy analysis of ORC by Kalinci et.al. [11]. The schematic view of the ORC was given in Figure 2 and the geothermal fluid inlet temperature and pressure were 140 °C and 1600 kPa respectively. The mass flowrate of geothermal fluid was 64,87 kg/s and the return temperature and pressure of geothermal fluid were 80 °C and 1600 kPa. The isentropic efficiencies of turbine and pump were selected as % 85 and % 80.

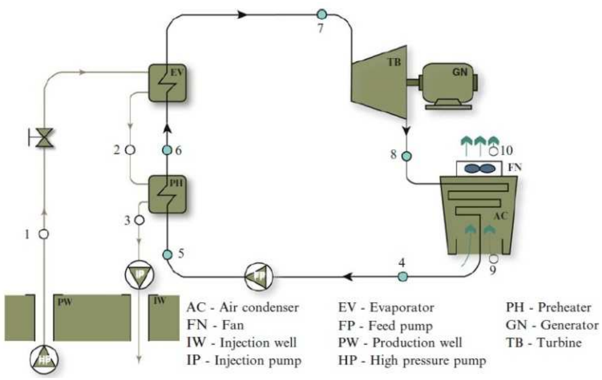

Figure 2 Reference ORC that was used in validation [11]

Isopentane was used as working fluid with a mass flowrate of 36,04 kg/s. The maximum and minimum pressures of cycle were 700 kPa and 95 kPa respectively. In air cooled condenser, the inlet and exit temperatures of air, which has a mass flowrate of 1414,12 kg/s, were 15 °C and 25 °C.

Pressure, temperature, enthalpy, entropy and flow exergy values for each state of reference study were given in Table 2.

Table 2 Reference values for selected application [11]

State no.	Fluid	$\dot{m}$ (kg/s)	P (kPa)	T (°C)	h (kJ/kg)	s (kJ/kg)	ex (kJ/kg)	$\dot{E}x$ (kW)
0	Geothermal		101.325	15	63.01	0.2242		
0′	Isopentane		101.325	15	−372.4	−1.766		
1	Geothermal	64.87	1,600	140	590	1.738	91.05	5,906
2	Geothermal	64.87	1,600	104	437.2	1.351	49.66	3,221
3	Geothermal	64.87	1,600	80	336.6	1.074	28.33	1,838
4	Isopentane	36.04	95	25	−350	−1.69	0.382	13.77
5	Isopentane	36.04	700	25.32	−348.8	−1.689	1.375	49.55
6	Isopentane	36.04	700	98.62	−167	−1.146	26.86	968.1
7	Isopentane	36.04	700	98.62	108.1	−0.406	88.85	3,202
8	Isopentane	36.04	95	54.33	46.3	−0.373	17.33	624.7
9	Air	1414.12	101.325	15	288.5	5.661		0
10	Air	1414.12	101.325	25	298.6	5.695	0.308	435.55

In reference study, the exergetic analysis of each component was done by using above values. In this analysis, flow exergies at inlet and exit, exergy destructions and exergetic efficiencies of each component were calculated and given in Table 3.

Table 3 Calculated exergy values for selected application [11]

Component	$\dot{E}x_{in}$ (kW)	$\dot{E}x_{out}$ (kW)	$\dot{I}$ (kW)	ε (%)
Feed pump	44.34	35.78	8.56	80.69
Preheater	1,383	918.55	464.45	66.42
Evaporator	2,685	2223.90	461.10	82.83
Turbine	2577.30	2229.00	348.30	86.49
Condenser	610.93	435.55	610.93	71.29
ε_{cycle} (%)				44.78
ε_{sys} (%)				30.84

In reference paper, the thermal and exergetic efficiencies of ORC were given as % 11,06 and % 5,34 respectively.

For the validation of the mathematical model, input data of the reference paper were given in simulation software that were developed in this study. The results obtained by the simulation software were given in Table 4.

Table 4 Simulation results for selected reference study

State	P (kPa)	T (°C)	h (kJ/kg)	s (kJ/kgK)	ex (kJ/kg)	Ex (kW)
1	1600,0	140,00	590,0	1,7380	90,92	5898,00
2	1600,0	104,30	438,4	1,3540	49,88	3236,00
3	1600,0	80,00	336,2	1,0740	28,31	1836,00
4	95,0	26,01	-347,8	-1,6820	0,45	16,25
5	700,0	26,33	-346,5	-1,6820	1,45	52,11
6	700,0	98,62	-167,0	-1,1460	26,77	964,70
7	700,0	98,62	108,1	-0,4063	88,64	3195,00
8	95,0	54,33	46,3	-0,3727	17,12	617,10
9	101,3	15,00	288,5	5,6610	0	0
10	101,3	25,00	298,6	5,6950	0,17	240,90

The simulation software gave results for thermal and exergetic efficiencies as % 13,3 and % 5,7 respectively. The comparison and the deviations of these values were given in Table 5.

Table 5 Comparison of reference and calculated values

	Real Values	Calculated Values	Deviation
Energy Efficiency (%)	11,06	13,30	2,24
Exergetic Efficiency (%)	5,34	5,70	0,36

Since the deviations in thermal efficiency and exergetic efficiency were very small, the mathematical model was validated.

TRENDS AND RESULTS

By using the validated simulation software, the analysis of ORC with geothermal heat source in Aydın Turkey was done by the data given in Table 6. By this analysis the change in net work, in thermal efficiency and in exergetic efficiency with respect to maximum and minimum cycle pressures for two different working fluids of R141b and R123 were determined.

Table 6 Data for geothermal sourced ORC in Aydın Turkey

$T_{geo,i}$= 203 °C	$\dot{m}_{geo}$= 92 kg/s	P_7=P_8=101,3 kPa
$T_{cw,i}$=15 °C	$\dot{m}_{cw}$= 100 kg/s	P_5=P_6=101,3 kPa
X_3= 1	$\dot{m}_{wf}$= 50 kg/s	P_2=P_3=2000 kPa
X_1= 0	η_p= 0,80	P_1=P_4=200 kPa
η_T= 0,85		

By using the developed simulation software, net power, thermal and exergetic efficiencies of ORC were calculated for turbine inlet pressure of 2000 kPa and turbine exit pressure of 200 kPa due to data given in Table 6. The results were obtained for two different working fluids R141b and R123 separately. For R141b net power, thermal and exergetic efficiencies were calculated as 2103 kW, % 15,52 and % 47,78 respectively. On the other hand for R123 net power, thermal and exergetic efficiencies were calculated as 1614 kW, % 15,18 and % 45,78.

Results for Working Fluid R141b

At the first stage, in which the isentropic working fluid of R141b was used, the net power of ORC were calculated by simulation software, for maximum cycle pressure values between 500 kPa and 3500 kPa, while keeping minimum cycle pressure constant at 200 kPa. As it can be seen from Figure 3, the maximum net power of 2331 kW was achieved at maximum pressure of 3184 kPa.

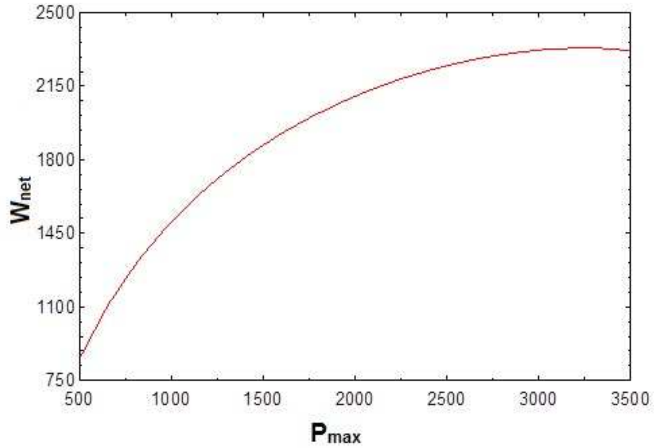

Figure 3 The change in net power with respect to maximum cycle pressure for R141b

The change in thermal and exergetic efficiencies with respect to maximum cycle pressure, which varies between 500 kPa and 3500 kPa, while keeping minimum cycle pressure constant at 200 kPa, was given in Figure 4. Maximum thermal and exergetic efficiencies was achieved at 3500 kPa as % 17,10 and % 54,19 respectively. The increase in maximum cycle pressure, which raises the turbine inlet temperature, increase both thermal and exergetic efficiencies of ORC. As it can be seen in Figure 4, the effect of maximum cycle pressure increase in exergetic efficiency was much higher than that was in thermal efficiency due to the quality rise of working fluid energy with respect to temperature.

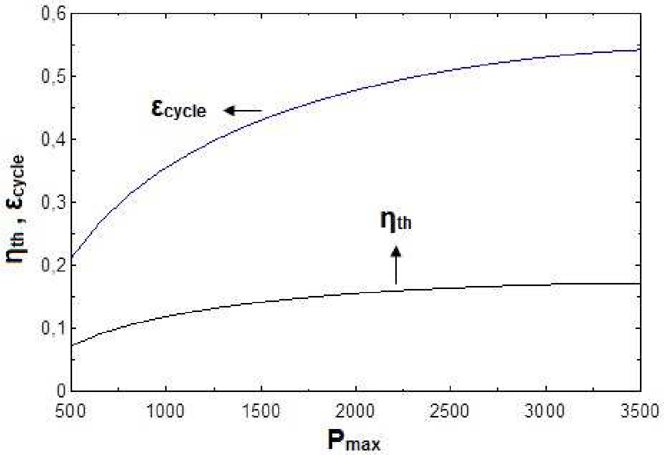

Figure 4 The change in thermal and exergetic efficiencies with respect to maximum cycle pressure for R141b

Secondly, for the isentropic working fluid of R141b, the net power of ORC were calculated for minimum cycle pressure values between 100 kPa and 300 kPa, while keeping maximum cycle pressure constant at 2000 kPa. As it can be seen from Figure 5, the maximum net power of 2749 kW was achieved at minimum pressure of 100 kPa.

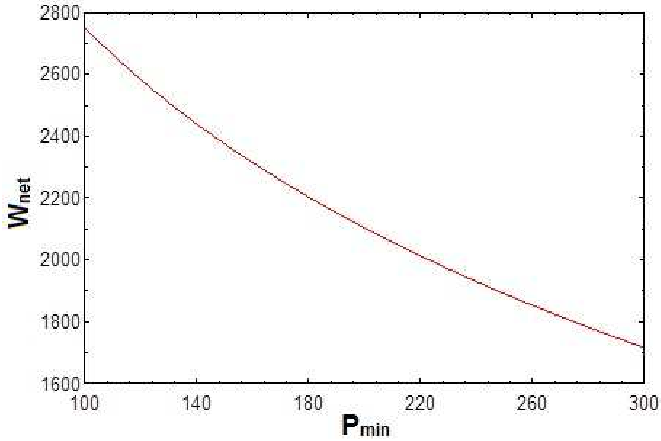

Figure 5 The change in net power with respect to minimum cycle pressure for R141b

The change in thermal and exergetic efficiencies with respect to minimum cycle pressure, which varies between 100 kPa and 300 kPa, while keeping maximum cycle pressure constant at 2000 kPa, was given in Figure 6. Maximum thermal and exergetic efficiencies was achieved at 100 kPa as % 18,55 and % 59,01 respectively.

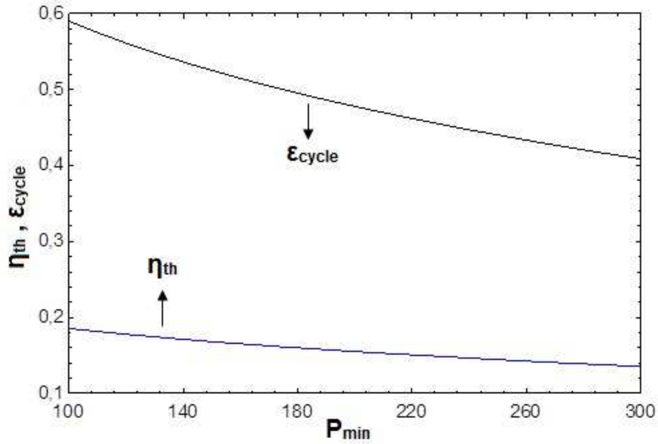

Figure 6 The change in thermal and exergetic efficiencies with respect to minimum cycle pressure for R141b

Results for Working Fluid R123

At the first stage, in which the dry working fluid of R123 was used, the net power of ORC were calculated by simulation software, for maximum cycle pressure values between 500 kPa and 3500 kPa, while keeping minimum cycle pressure constant at 200 kPa. As it can be seen from Figure 7, the maximum net power of 1798 kW was achieved at maximum pressure of 3184 kPa.

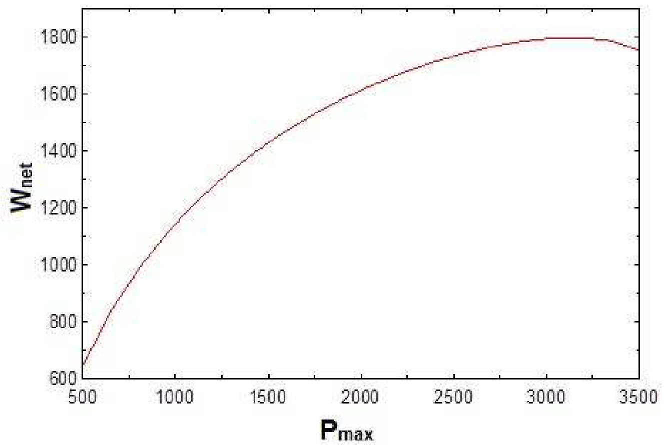

Figure 7 The change in net power with respect to maximum cycle pressure for R123

The change in thermal and exergetic efficiencies with respect to maximum cycle pressure, which varies between 500 kPa and 3500 kPa, while keeping minimum cycle pressure constant at 200 kPa, was given in Figure 8. Maximum thermal and exergetic efficiencies was achieved at 3500 kPa as % 16,88 and % 56,29 respectively.

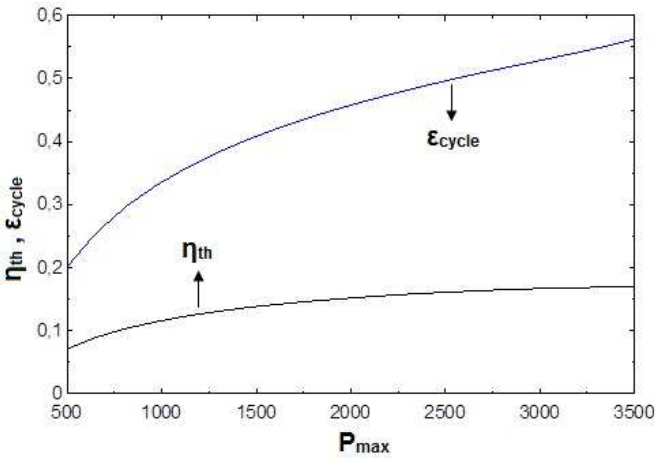

Figure 8 The change in thermal and exergetic efficiencies with respect to maximum cycle pressure for R123

Secondly, for the dry working fluid of R123, the net power of ORC were calculated for minimum cycle pressure values between 100 kPa and 300 kPa, while keeping maximum cycle pressure constant at 2000 kPa. As it can be seen from Figure 9, the maximum net power of 2119 kW was achieved at minimum pressure of 100 kPa.

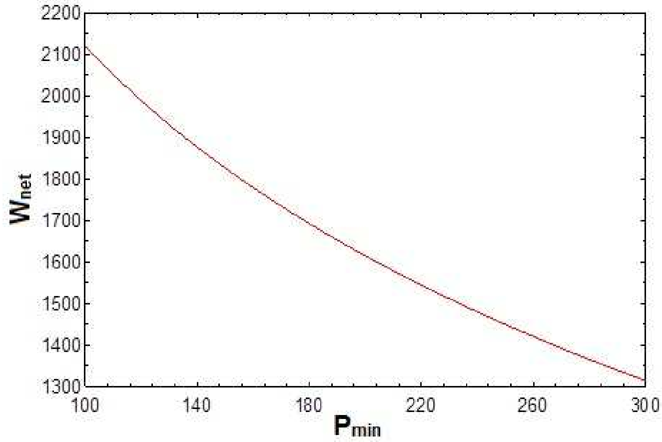

Figure 9 The change in net power with respect to minimum cycle pressure for R123

The change in thermal and exergetic efficiencies with respect to minimum cycle pressure, which varies between 100 kPa and 300 kPa, while keeping maximum cycle pressure constant at 2000 kPa, was given in Figure 10. Maximum thermal and exergetic efficiencies was achieved at 100 kPa as % 18,09 and % 55,63 respectively.

Comparison of Simulation Results for the Working Fluids

In previous section the change in net power and efficiencies with respect to maximum and minimum cycle pressure changes were analyzed separately for two different working fluids R141b and R123. For making a comparison between these two different working fluids, the graphs that show the change in net power with respect to maximum and minimum cycle pressure changes were combined.

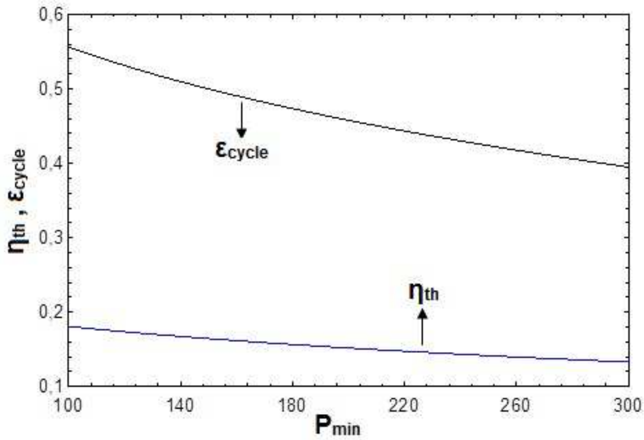

Figure 10 The change in thermal and exergetic efficiencies with respect to minimum cycle pressure for R123

Firstly, the effect of maximum cycle pressure change on net power production for two different working fluids R141b and R123 were compared. As it can be seen in Figure 11, R141b shows better performance than R123 for all maximum cycle pressure values. Moreover the slopes of curves for R141b and

R123 in graph show that, the working fluid R141b is more sensitive to increase in maximum cycle pressure than R123.

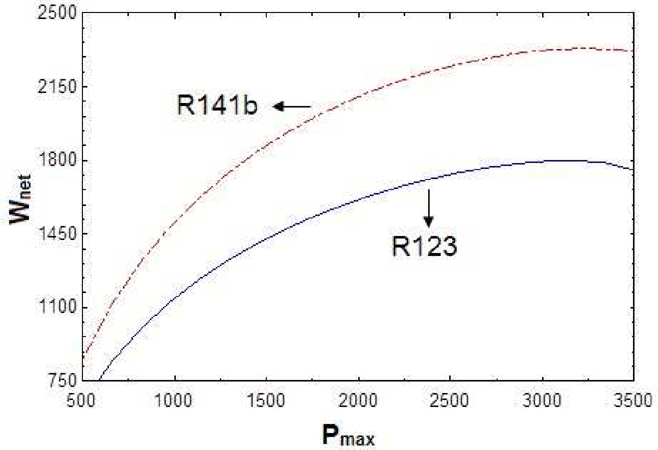

Figure 11 The change in net power with respect to maximum cycle pressure for R141b and R123

Secondly, the effect of minimum cycle pressure change on net power production for two different working fluids R141b and R123 were compared. As it can be seen in Figure 12, R141b shows better performance than R123 for all minimum cycle pressure values. Moreover the slopes of curves for R141b and R123 in graph show that, the working fluid R141b is a little bit sensitive to decrease in maximum cycle pressure than R123. So as a result, for the data of Aydın geothermal field, the working fluid R141b is a better selection for the ORC, due to its high sensitivity to changes in maximum and minimum cycle pressures.

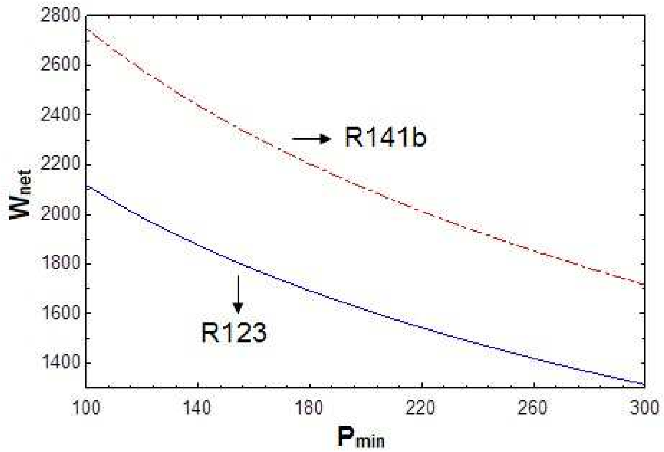

Figure 12 The change in net power with respect to minimum cycle pressure for R141b and R123

CONCLUSION

In this study, modeling and thermodynamic analysis of an Organic Rankine Cycle using geothermal heat source in Aydın, Turkey has been made and optimum operational conditions were determined by developing simulation software.

In simulation the effect of minimum and maximum pressure and working fluid changes to the system performance has been investigated. ORC using working fluids R141b and R123 has been simulated for selected input values separately.

At first, by using the developed simulation software, net power, thermal and exergetic efficiencies of ORC were calculated for turbine inlet pressure of 2000 kPa and turbine exit pressure of 200 kPa due to data given in Table 6. The results were obtained for two different working fluids R141b and R123 separately. For R141b net power, thermal and exergetic efficiencies were calculated as 2103 kW, % 15,52 and % 47,78 respectively. On the other hand for R123 net power, thermal and exergetic efficiencies were calculated as 1614 kW, % 15,18 and % 45,78.

Later, the maximum and minimum cycle pressures of ORC were changed in a selected range for optimizing the operation conditions. The upper limit of maximum cycle pressure was determined as 3500 kPa by considering the critical pressures of working fluids. On the other hand, the lower limit of minimum cycle pressure was determined as atmospheric pressure of 100 kPa for preventing undesired vacuum effect during operation.

At first, by keeping minimum cycle pressure constant at 200 kPa and by changing maximum cycle peressure, net power, thermal and exergetic efficiencies for working fluid R141b were determined as 2331 kW, % 17,10 and % 54,19. For R123 these values were also determined as 1798 kW, % 16,88 and % 56,29 respectively.

Secondly, by keeping maximum cycle pressure constant at 2000 kPa and by changing minimum cycle peressure, net power, thermal and exergetic efficiencies for working fluid R141b were determined as 2479 kW, % 18,55 and % 59,01. For R123 these values were also determined as 2119 kW, % 18,09 and % 55,63 respectively.

As it can be seen from above mentioned simulation results, net power, thermal and exergetic efficiencies in ORC can be increased by increasing maximum cycle pressure or by decreasing minimum cycle pressure as expected. However the decrease in minimum cycle pressure was more effective that the increase in maximum cycle pressure.

Furthermore, in this study the effect of working fluid on ORC performance was also analyzed. As compared with R123, the performance of working fluid R141b was better, especially in power production, for the selected geothermal field in Aydın Turkey. This shows that working fluid selection in ORC was very important and has to be done according to the quality of geothermal energy resource, thus temperature of it.

As a future work, numerical studies have to be supported by experimental studies, which consider the external heat losses and pressure losses from each component and piping in ORC. Besides heat transfer analysis has to be done especially for the evaporator and condenser of such Organic Rankine Cycles.

REFERENCES

[1] Quoilin S., Broek M., Declaye S., Dewallef P., Lemort V., Techno-economic survey of Organic Rankine Cycle (ORC) systems, *Renewable and Sustainable Energy Reviews*, vol. 22, s.168-186, 2013.

[2] Huijuan C., Goswami D.Y., Stefanakos E.K., A review of thermodynamic cycles and working fluids for the conversion of low-

grade heat, *Renewable and Sustainable Energy Reviews*, vol.14, Issue 9, s.3059-3067, 2010.

[3] Dipippo R., Ideal thermal efficiency for geothermal binary plants, *Geothermics*, vol.36, Issue 3, s.276-285, 2007.

[4] Quoilin S., Sustainable Energy Conversion Through the Use of Organic Rankine Cycles for Waste Heat Recovery and Solar Applications, *Ph.D. thesis*, *University of Liège*, Belgium, 183p, 2011.

[5] Bundela P. S., Chawla V., Sustainable Development through Waste Heat Recovery, *American Journal of Environmental Sciences*, vol. 6, Issue 1, s.83-89, 2010.

[6] Li J., Pei G., Li Y., Wang D., Ji J., Energetic and exergetic investigation of an organic Rankine cycle at different heat source temperatures, *Energy*, vol.38, s.85-95, 2012.

[7] Wu S., Zhou S., Xiao L., Li Y., Liu C., Xu J., Determining the optimal pinch point temperature difference of evaporator for waste heat recovery, *Journal of the Energy Institute*, vol. 87, Issue 2, s.140-151, 2014.

[8] Liu Q., Duan Y., Yang Z., Performance analyses of geothermal organic Rankine cycles with selected hydrocarbon working fluids, *Energy*, vol.63, s.123-132, 2013.

[9] Li J., Pei G., Li Y., Wang D., Ji J., Energetic and exergetic investigation of an Organic Rankine Cycle at different heat source temperatures, *Energy*, vol.38, s.85-95, 2012.

[10] Li J., Pei G., Li Y., Ji J., Evaluation of external heat loss from a small-scale expander used in Organic Rankine Cycle, *Applied Thermal Engineering*, Vol. 31, Issues 14–15, s.2694-2701, 2011.

[11] Kalinci, Y., Hepbasli, A., Dincer I., Exergetic performance assessment of a binary geothermal power plant, *Progress in Exergy, Energy, and the Environment*, Springer, London, pp. 23-49, 2014.