Northumbria University
NEWCASTLE

Journal of Energy and Power Engineering 5 (2011) 392-402

An Evaluation of R134a and R245fa as the Working Fluid in an Organic Rankine Cycle Energized from a Low Temperature Geothermal Energy Source

S. Masheiti[1], B. Agnew[2] and S. Walker[2]

1. School of Mechanical and Systems Engineering, University of Newcastle upon Tyne, NE1 7RU, UK

2. School of the Built and Natural Environment, Northumbria University, Newcastle upon Tyne, NE1 8ST, UK

Received: September 23, 2010 / Accepted: February 21, 2011 / Published: May 31, 2011.

Abstract: The characteristics of an organic Rankin cycle designed to operate with a low temperature geothermal source and constant temperature cooling water supplied from freshwater ponds typical to those found near Waddan City in the Al Jufrah region of Libya were examined. Two working fluids were examined and it was concluded that the most suitable for this application was R-245fa. The off design performance of the organic Rankine cycle was examined and it was shown that the cycle is controlled by the performance of the condenser which is cooling water side temperature limited.

Key words: Low-temperature resources, IPSEpro, organic Rankine cycle (ORC), R-245fa, R-134a.

Nomenclature

UA	Overall heat transfer coefficient area (kW/k)
ε	Heat exchanger effectiveness
η	Efficiency
h	Specific Enthalpy (kJ/kg)
Q	Heat transferred (kW)
W	Work done (kW)
Subscripts	
1, 2, 3, 4	State points
m	mechanical
s	Isentropic
wf	Working fluid
p	Pump
Evap	Evaporator

1. Introduction

This paper examines the possibility of utilizing a low temperature geothermal resource as part of a sustainable energy supply for Waddan City in the Al Jufrah region of Libya. The city is located at global coordinates of X = 16º 09' 46", Y = 29º 07' 06" and has a population of 27,590. The region around Waddan City is well supplied with fresh water lakes that contain water at a constant year round temperature of 25 ℃ and the area has numerous low temperature artesian wells that act as a geothermal source for this study. The particular well studied, shown in Fig. 1, is 291 meters above sea level and surrounded by several shallow cold water reservoirs. It is proposed to use the artesian well as an energy source to an organic Rankin cycle (ORC) designed to provide electrical power to meet some of the demand of the city and to use both the ORC cooling water and the waste water from the ORC plant as an energy source for domestic use. The fresh water lakes supplied the cooling water to the ORC. The energy system was modelled using the thermal system simulation software IPSEpro [1] which is a highly flexible tool used for heat balance analysis of power plants, component design, acceptance test calculation and on-line optimization.

Organic Rankine cycles are now well established as

Corresponding author: S. Masheiti, Ph.D., M.Sc., research fields: low powered thermal cycles (thermodynamics and fluid mechanics). E-mail: s.a.a.masheiti@ncl.ac.uk.

Fig. 1 Artesian well heat source.

means of achieving high energy conversion efficiencies when used in combined cycle power plants to utilize the thermal energy available in the exhaust of the primary or topping cycle. In applications where the exhaust temperature of the primary cycle is above 400 ℃ there is little to choose between an air bottoming cycle and an organic Rankin cycle in both power generation and combined heat and power (CHP) modes [2]. In other applications however that exhibit low exhaust gas temperatures, such as recuperative gas turbines in which the topping cycle is optimized for high efficiency, the lower exhaust temperatures make ORCs the bottoming cycle of choice [3].

ORC binary cycle power plants are the most common technology for utilizing medium and low temperature sources for producing electrical power between 0.5-5 MW but, as has been shown by Leibowitz et al. [4], it is possible to produce ORC units for power recovery with outputs of as little as 20 kW at an economic and viable cost and with source temperatures between 50 ℃ and 350 ℃. Geothermal power systems fall in this lower temperature range and are considered as one of the most reliable continuous sources of renewable energy, in comparison to solar and wind which generate energy intermittently. The temperature of the geothermal heat sources is the key factor in determining the most suitable type of technology that can be successfully applied.

The selection of the working fluid plays a significant role in the design of the ORC process [5]. Badr et al. [6, 7] produced a significant piece of research work in evaluating fluids suitable for use in ORCs covering mainly the halocarbon compounds R-11, R-113 and R-114. This has now been overtaken by events in that the fluids examined have now been banned because of concern about their Ozone Depletion and Global Warming Potentials (ODP and GWP). Recently Saleh et al. [8] showed that 31 pure component working fluids are suitable for use in ORCs. In these studies the critical temperature, normal boiling temperature, and critical pressure for the working fluids were arranged in specific order to give an indication of their suitability as working fluids for ORCs. Fluids with high critical temperatures allow on the one hand higher boiling temperatures but on the other hand lower pressures and thereby lower cycle pressure differences are reached. The selected fluids covered a wide range of critical temperatures from 101.3 ℃ for R-134a up to 280.5 ℃ in the case of cyclo-hexane. The researchers further compared the overall performance of two fluids, cyclo-hexane and R-245fa at isobars of 1, 10 and 25 bar with that of water. The comparison shows that the enthalpy differences for organic fluids were significantly lower than that of water which leads to higher mass flows for the same output power. Other workers in this field [9] have also studied the performance of different fluids for low temperature Rankin cycles. This study examined 29 working fluids for low temperature geothermal power conversion applications and indicated that the final choice of the working fluid was mainly a compromise based on several conflicting factors. Schuster et al. [10] used IPSEpro to investigate the economics of a stand-alone solar desalination system, for producing drinking water and a waste heat recovery from biogas digestion plants for domestic electrical power generation. Their results pointed out that the ORC process can work with saturated vapour or with a constant low degree of superheating depending on the fluid. Higher superheating, in order to avoid liquid in the exhaust vapour, is not necessary because in ORC cycles the expansion ends in the superheated region. This avoids droplet erosion in the expander, increases reliability and allows a fast start up of the cycle. Higher superheating of the vapour is favourable for higher efficiencies, but because of low heat exchange coefficients this would lead to very large and expensive heat exchangers. Lee et al. [11] carried out a parametric analysis on an organic Rankine cycle energy recovery system. He pointed out that the ORC system efficiency only correlates with the normal boiling point, the critical pressure and the molecular mass of the working fluid. Brasz et al. [12] described a unique development work of an R134a ORC cycle to obtain electrical power from a low-temperature (74 ℃) geothermal source at Chena Hot springs in Alaska. The system thermal efficiency obtained was 8% when the cooling water temperature, pumped from the nearby river, was 3 ℃.

This paper takes the evaluation of two working fluids further than that of Saleh et al. [8] and Tao Guo et al. [9] by taking the system analysis further in simulating a complete ORC using the software IPSEpro energized from a low temperature energy source at 73 ℃ coupled with a cooling water supply at 25 ℃. The fluids examined were R-134a and R-245fa. For validation purposes, the models developed were also simulated and checked by the leading European ORC manufacturer Turboden® using the Aspen Plus software package [13], and the results were very similar to those obtained by Brasz et al. [12].

2. General Characteristics of ORC Working Fluids for Low Temperature Sources

The properties of the chosen working fluid have a significant impact on the performance of the ORC cycle. Ultimately appropriate thermodynamic properties can result in higher cycle performance and low costs. In order to achieve a successful ORC process, the ideal organic working fluid should have the following general characteristics:

(1) High molecular weight.

(2) Small heat content (low enthalpy).

(3) High critical pressure and temperature to allow engine operating temperature to absorb all the heat available up to that temperature.

(4) Low operating pressure to avoid danger of explosion or rupture and avoid negative impact on the reliability of the cycle.

(5) Small specific volume of fluid in its gaseous state to avoid the need of large and costly turbines, evaporators, and condensers.

(6) Has higher pressure inside condenser to prevent air inflow into the system.

(7) Inexpensive to avoid high overall system cost.

(8) Low heat latency.

(9) Non flammable, corrosive or toxic.

(10) Low environmental impact.

The least toxic of fluids are the refrigerants and they also exhibit good material compatibility and stability limits. The two fluids chosen as the subject of this work belong to the new class of refrigerants developed to have no Ozone Depletion Potential (ODP) and small Global Warming Potential (GWP). Also they have thermodynamic properties that make them suitable for use with a low temperature heat source. The important parameters of these fluids are shown in Table 1.

3. Thermodynamic Analysis of the ORC Cycle

An ORC can be depicted schematically in Fig. 2. The cycle is entirely in the sub-critical region of the T-S chart utilizing phase change heat transfer processes for both energy addition and rejection. The cycle consists of an evaporator (4-1) in which the energy from the geothermal source is transferred to the ORC working fluid. The fluid leaves the evaporator in the dry saturated condition and enters an expansion device (1-2). The expansion process drives an electrical generator. On leaving the expander the fluid is fully condensed (2-3) leaving the condenser as a low temperature, low pressure liquid. It is then compressed to the evaporator pressure and the cycle is repeated.

The cycle can also be shown on a T-S diagram, Fig. 3.

Table 1 Properties of working fluids used in the simulation from Ref. [9].

Property	R-134a	R-245fa
Molecular mass, (kg/kmol)	102	134.05
Critical points	101 ℃-40.6 bar	154 ℃-36.4 bar
Latent heat of vaporization heat at 1 atm., (kJ/kg)	217.2	197.5
Boiling temperature at 1 atm., (℃)	-26.4	14.9
Safety	Non-flammable	Non-flammable
Atmospheric lifetime years	14	7.6
ASHRAE level of safety	A1	B1
Ozone depleting potential ODP	≈ 0	≈ 0
Net greenhouse warming potential (GWP) 100 year	1430	1030

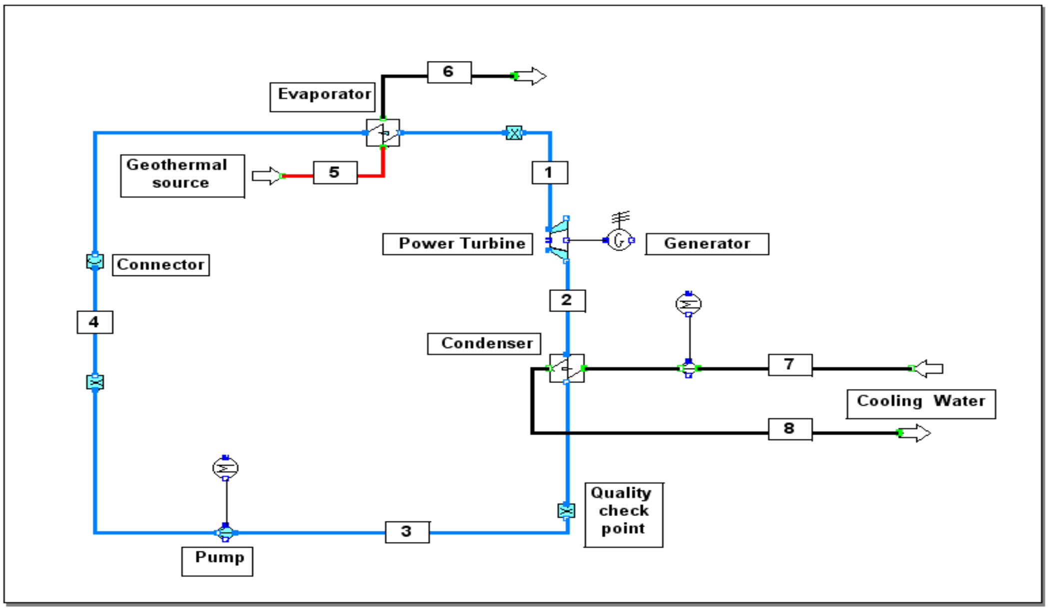

Fig. 2 Schematic of an ORC cycle.

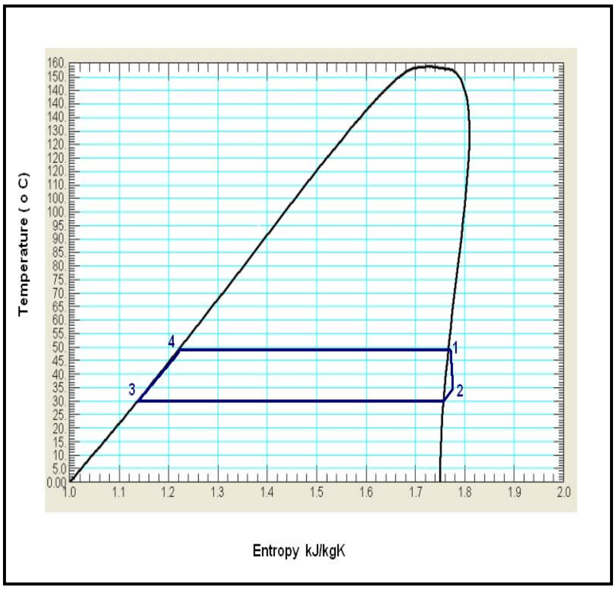

Fig. 3 T-S diagram for the R-245fa cycle.

It can be seen that in the case of R-245fa the expansion occurs in the superheated region due to the slope of the dry saturation line which makes this fluid suitable for use with radial inflow turbines as opposed to screw type expanders [14].

The analysis of the cycle consists of applying mass and energy balances to each of the processes mentioned above.

The effectiveness–NTU method has been used to model the evaporator and condenser. By choosing each component as a control volume, each process in the system cycle can be written as follows:

Process 1 → 2 is the actual expansion of the working fluid through the turbine:

$$W = (h_1 - h_2)\eta_m \cdot \eta_s \quad (1)$$

Process 2 → 3 is the condensation process which occurs within counter flow heat exchanger using cooling water at 25 ℃ so the heat transferred to the cooling water (heat out) is:

$$Q = h_2 - h_3 \tag{2}$$

Process 3 → 4 is the work done by the working fluid pump:

$$W_p = (h_3 - h_4)/\eta_p \tag{3}$$

Process 4 → 1 is the heating process in the evaporator where geothermal heat transferred to the working fluid:

$$Q = h_4 - h_1 \tag{4}$$

The net electrical power produced by the ORC unit is:

$$W_{unit} = W - W_p \tag{5}$$

The thermal efficiency of the ORC unit is:

$$\eta_{net} = W_{unit}/Q_{Evap} \tag{6}$$

4. Modeling Using IPSEpro

The ORC was modeled using the simulation software IPSEpro. The main cycle and component parameters that were set values in the simulation are shown in Table 2. These were chosen as being representative values of commercially available ORC units. The cycle efficiency was a set value as the objective of this study is to examine the impact the choice of fluid has on the cycle components for a desired level of performance. The modeling was performed in two stages. Firstly a design case, based on the set parameters shown in Table 2, was performed to identify the fluid with the best overall performance. The efficiency of 4% was chosen to reflect the temperatures of the geothermal source and the cooling water. This is less than that produced by Brasz et al. who was able to use a much lower cooling water temperature. Secondly a sensitivity analysis examining the impact of variations in the geothermal source temperature from 68 ℃ to 78 ℃ on the cycle performance was performed for this fluid.

The cycle is represented in IPSEpro by the diagram shown in Fig. 4 which is Fig. 2 repeated with the addition of state properties at specific points. The four figures in the groups represent specific enthalpy, temperature, pressure and mass flow rate in order starting at the top right hand quadrant and moving clockwise.

5. Results and Discussions from the Modeling

In this section the results of the simulations are discussed.

5.1 Comparison of the Output for the Two Working Fluids

The impact of the choice of working fluid is summarized in Table 3.

It can be seen that the net power output of each cycle is the same but the means of achieving this are quite different. R-134a has the largest internal pumping requirement to accommodate a high refrigeration flow rate, lower latent heat and the highest cycle pressure ratio. It also requires a higher cooling water flow rate to remove a higher condenser load. This last point is reflected in higher UA values in the evaporator and condenser which will lead to higher initial and ongoing costs. These effects combine to reduce the overall cycle efficiency. A further point in favour of R-245fa can be realized by considering the pinch analysis. The pinch point analysis, shown in Figs. 5 and 6, indicates that R-245fa operates with a higher pinch temperature difference. This is a result of the evaporator temperature for R-245fa (49 ℃) being significantly

Table 2 Main fixed parameters of ORC cycle that are typical of marketed units.

Fixed parameter	R-134a	R-245fa
Plant thermal efficiency, (%)	4.0	4.0
Power turbine Isentropic efficiency, (%)	80	80
Evaporator inlet temperature from energizing source, (℃)	73	73
Evaporator inlet mass flow from energizing source, (kg/s)	114	114
Condenser inlet cooling water temperature, (℃)	25	25

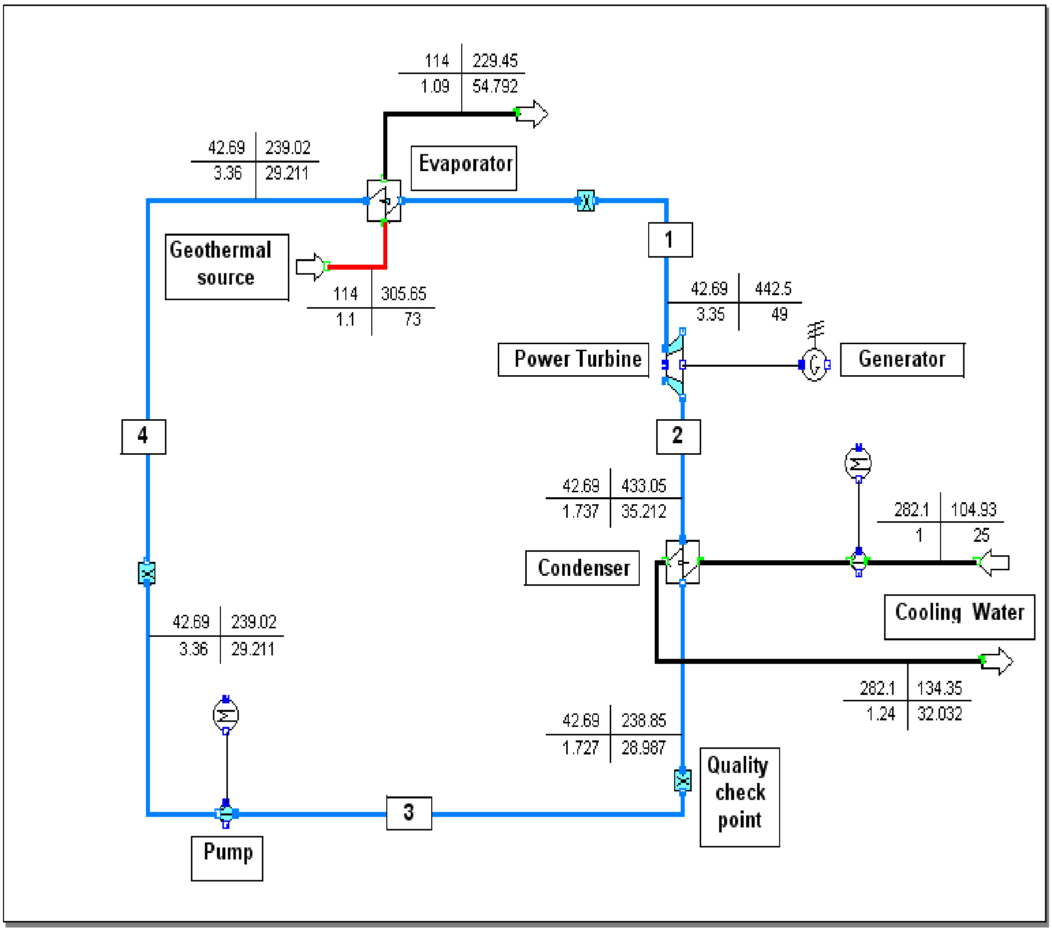

Fig. 4 Representation of the ORC cycle in IPSEpro.

Table 3 Comparison of the results from the simulation for both fluids.

No.	Description	Working fluid	
		R-134a	R-245fa
1	Gross electrical output power, (kW)	400.9	368.2
2	Refrigerant pump power consumption, (kW)	38.4	7.92
3	Plant net electrical power output, (kW)	350.1	350.1
4	Cooling water pump power consumption, (kW)	12.5	10.5
5	Refrigerant expander ΔP (P_{inlet}-P_{outlet}), (bar)	6.086	1.614
6	Refrigerant mass flow, (kg/s)	49.26	42.69
7	Cooling water mass flow, (kg/s)	335.4	282.1
8	UA condenser, (kW/k)	2745	2330.8
9	UA evaporator, (kW/k)	421.2	350.4
10	NTU condenser	1.949	1.967
11	NTU evaporator	0.88	0.732
12	Cooling water outlet temperature, (℃)	31	32
13	Source outlet temp. from evaporator, (℃)	55	55

lower than that of R-134a (55 ℃), which for fixed source temperatures results in a higher temperature difference for the evaporation process for R-245fa with resultant reduction in heat exchanger size for a given heat load. This will promote stable operation of the R-245fa cycle and reduce the evaporator UA value. It is clear that R-245fa has the best performance of the two fluids considered for the specified conditions. Therefore

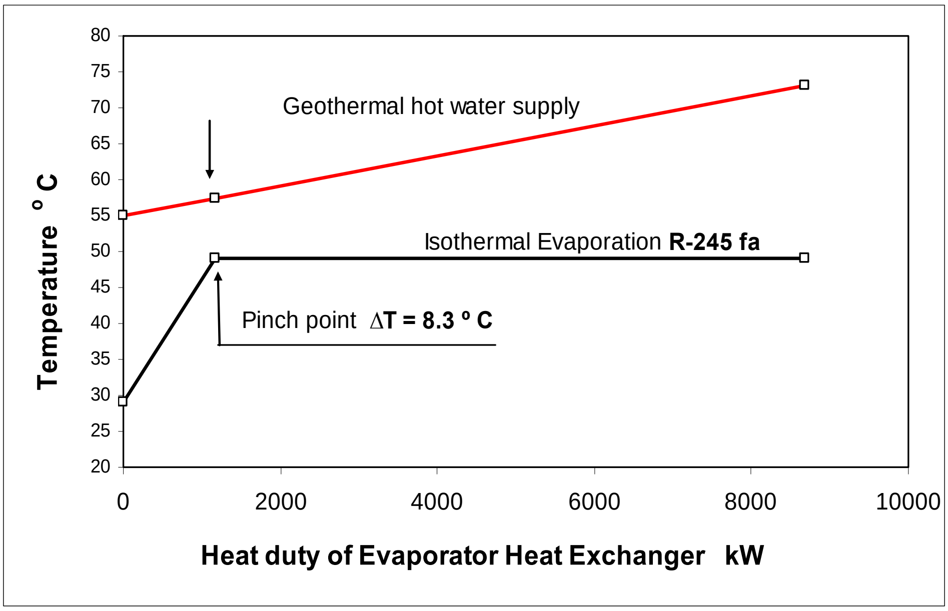

Fig. 5 Pinch point of R-245fa.

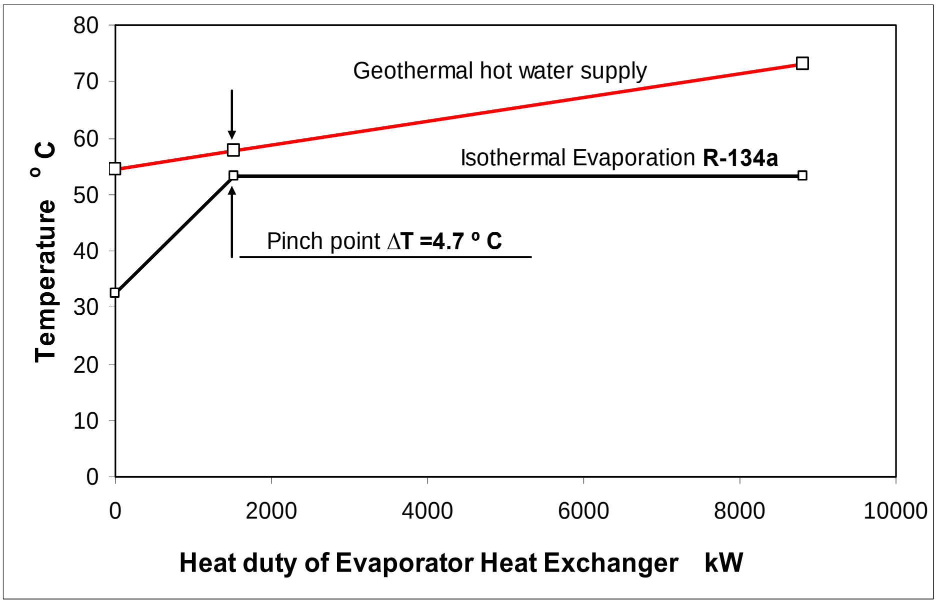

Fig. 6 Pinch point of R-134a.

the following analysis and discussion will focus only on the performance of this refrigerant.

5.2 Detailed Performance of R-245fa

The sensitivity analysis considered the impact of changing the geothermal source temperature from 68 ℃ to 78 ℃ on the cycle performance with the main components having the performance parameters as determined by the design case IPSEpro analysis.

It can be seen from Figs. 7 and 8 that the net power, efficiency and turbine pressure drop all increase with increased source temperature and this is accompanied by a decrease in refrigerant flow rate. The reduction in flow rate is a result of the improved cycle efficiency with a high source temperature. This is the dominant effect that exceeds that resulting from a reduction in the pumping load for both the refrigerant and the reduced cooling water requirement. The performance of the evaporator is

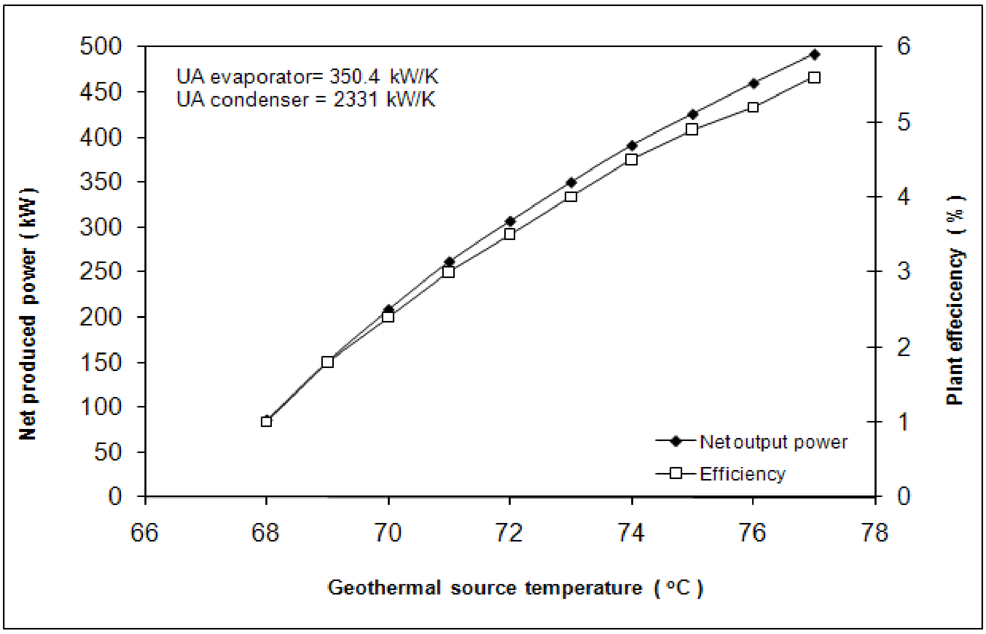

Fig. 7 Variation of net power and efficiency with variation in geothermal source temperature.

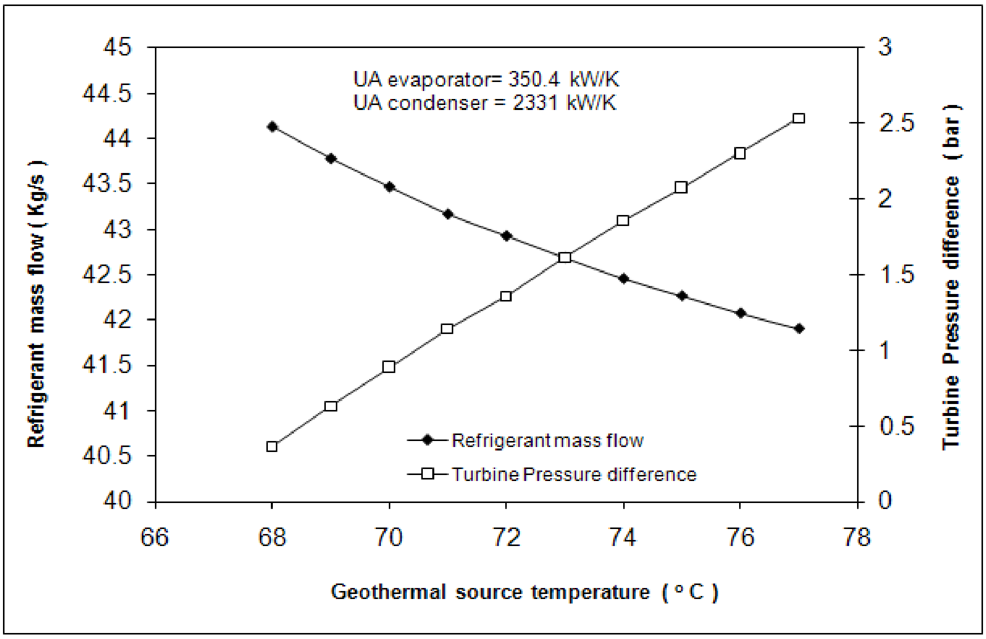

Fig. 8 Variation of refrigerant flow rate and turbine pressure difference with variation in geothermal source temperature.

hardly influenced by the variation in the source temperature except towards the lower range, Fig. 9. The evaporation temperature is not fixed at this condition but is allowed to float to accommodate the variation in the heat transfer rate and the refrigerant flow rate, Fig. 8. As the source temperature decreases, however, the pinch temperature difference sets a limit on the lowest value of the source temperature.

Fig. 9 shows the gradual increase of the evaporator heat transfer rate and the power turbine inlet temperature with increased source temperature. As the source temperature is raised, both the heat transfer rate and the turbine inlet temperature increase improving both the cycle efficiency and power output as shown in Fig. 7.

The evaporator is the main source of irreversibilities

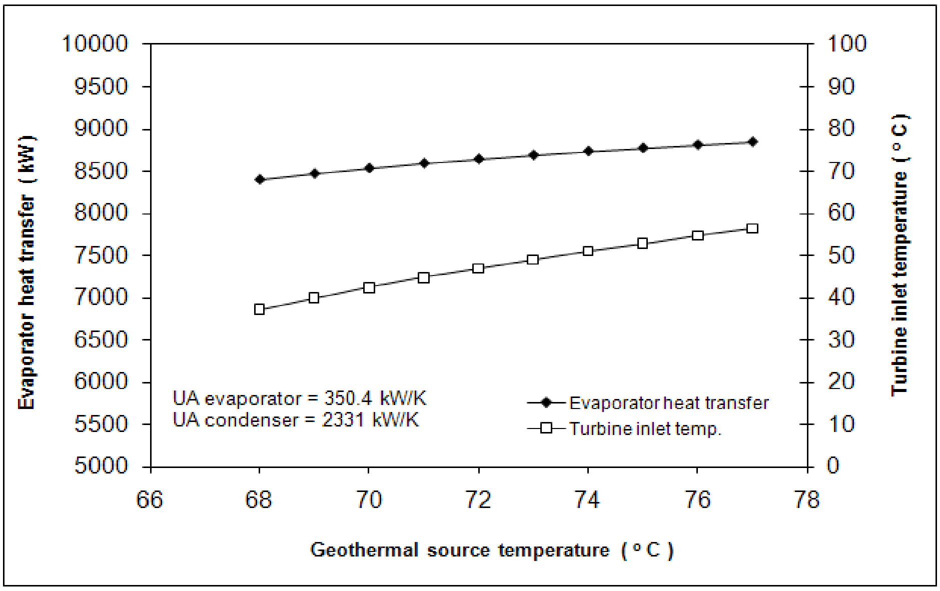

Fig. 9 Variation of evaporator load and turbine inlet temperature with geothermal source temperature.

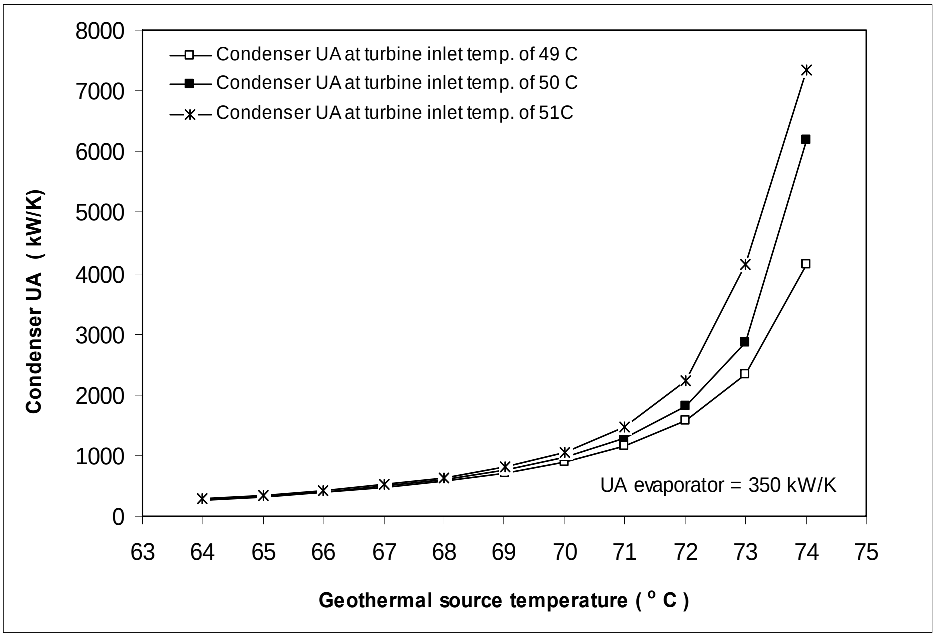

Fig. 10 Influence of geothermal source temperature on condenser UA.

in the cycle [9] but it is clear from Figs. 10 and 11 that the condenser is controlling the cycle, as at high source temperatures the condenser UA value must be increased significantly, when the evaporator operation is fixed. This then places additional constraints on the effectiveness values of these components as shown in Fig. 11. In contrast the effectiveness and NTU of the evaporator is little changed as the source temperature changes with fixed condenser UA as shown in Figs. 12 and 13.

6. Conclusions

It has been shown in this paper that it is possible to use an organic Rankine cycle to produce electrical power from a low temperature geothermal source. A comparison between the simulated results of two working fluids, R-134a and R-245fa, revealed that the refrigerant R-245fa had a better performance. This was reflected in the higher pinch temperature difference in

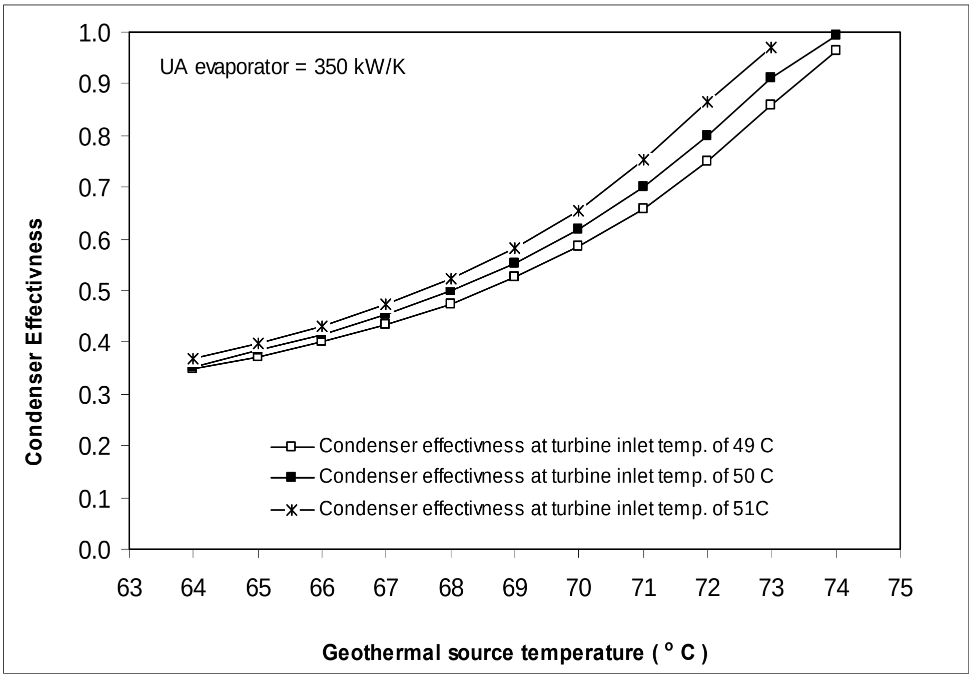

Fig. 11 Influence of geothermal source temperature on condenser effectiveness.

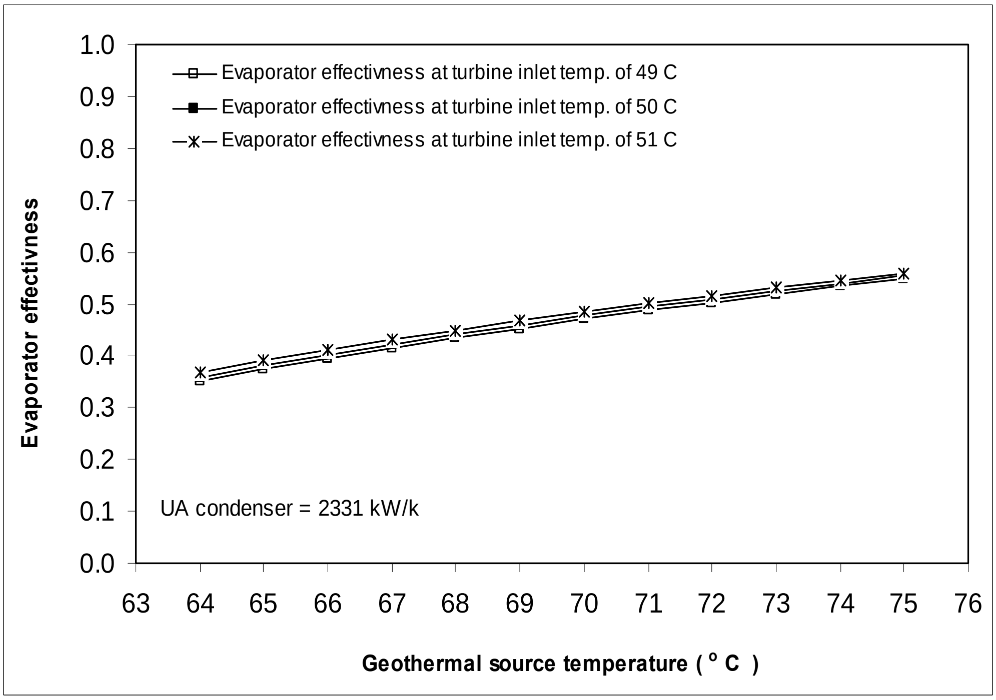

Fig. 12 Influence of geothermal source temperature on evaporator effectiveness.

the evaporator for this fluid. It also exhibited the lowest temperature and pressure at inlet to the expander and hence lower values of UA and NTU values for the condenser. This will result in a more economically efficient design. These results are in agreement with the findings of T. Guo et al. [9] and A.B. Gozdur et al. [5]. The condenser was the component that controlled the performance of the cycle, as this operated with the smallest temperature difference and was most sensitive to changes to demand.

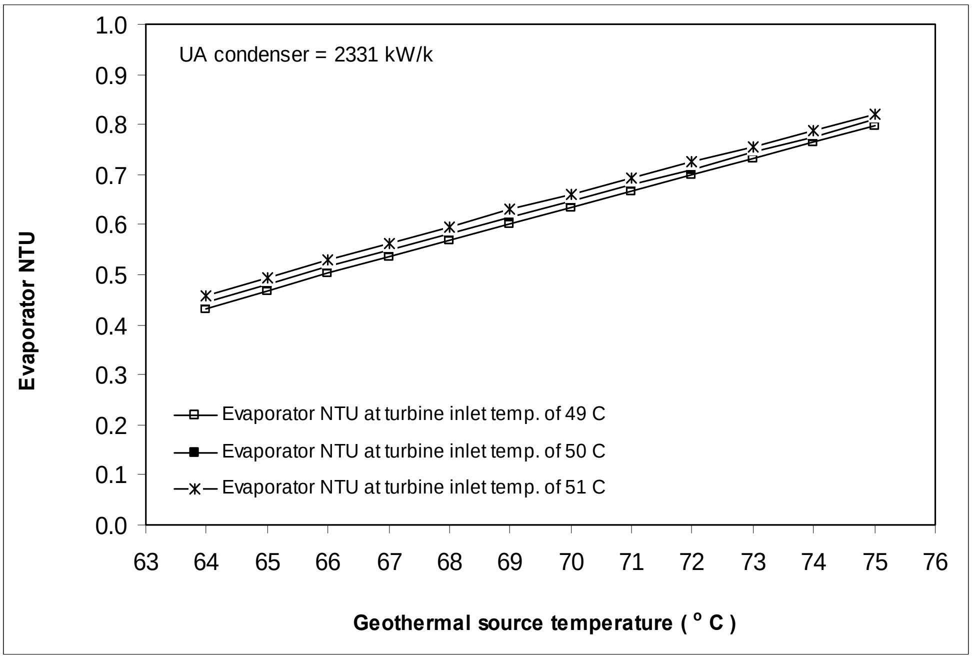

Fig. 13 Influence of geothermal source temperature on the evaporator NTU.

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