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Optimal Selection of Using Fluids (HFC, HCFC, HFC) for an Organic Rankine Cycle Utilising a Low Temperature Geothermal Energy Source

Fathi A. Latrash[1+], Brian Agnew[1], Mohammed A. Al-Weshahi[2] and Nuri M. eshoul[1]

[1]School of Mechanical and System Engineering, Newcastle University, UK
[2]Shinas College of Technology, Alqr, Oman

Abstract. The performance of ORC systems strongly depends on the working fluid properties. The work reported in this paper assessed 15 working fluids belonging to different chemical compositions groups such as HFC, HCFC, HC and mixtures. The selection criteria for these fluids were as follows: - Safety, environmental consideration, refrigerant pump power consumption, net produced power, thermal efficiency and evaporating and condensing pressure. This study revealed that, based on the selection criteria the best refrigerants were R227ea and R236ea but other refrigerants such as R142b, R124, R245ca, R600a and R717 were not selected.

Keywords: Refrigerant, Geothermal, Selection Criteria, (ORC).

1. Introduction

The population of the world is increasing dramatically, and the demand for energy is greater than ever before. Fossil fuels, the main source of energy, are not renewable resources and fuels such as coal, oil, and natural gas; provide the energy that powers our life style and the world economy. Based on current known reserves and consumption of these fuels, the quantities amount of each fossil fuel remains economically available (as of 2010) for only a further 30 years. The increased consumption of fossil fuels has led to global warming and environmental impacts that have stimulated engineers and scientists to search for alternative energy sources. Most of these studies have investigated the possibility of utilizing renewable energy (e.g. solar, wind, hydro and geothermal) or recovering the waste heat from industrial processes. The geothermal energy (thermal energy under the earth's crust) is an example of the low grade heat source whose temperature varies from 60 ^{0}C to 200 ^{0}C [1]. This type of energy could be used for direct heating or power generation [2]. The Organic Rankine Cycle (ORC) showed a promising solution for conversion of low grade heat (including the geothermal energy) to electrical energy. The ORC uses an organic fluid currently used in refrigeration applications as the working fluid so it can be powered by low temperature sources [3-5]. The proper selection of the ORC working fluid is the key for achieving high performance and a cost-effective ORC units [6].

The main objective of this paper is to determine the most suitable working fluid for an ORC energised from a low temperature geothermal source. The choice of working fluid was characterized by the following specifications: high efficiency, acceptable pressure within cycle, low ozone depletion potential (ODP), low global warming potential (GWP), low specific volume, low toxicity, high safety and low cost [1, 6].

The working fluids were divided to groups according to their chemical composition; hydro fluorocarbons (HFC), hydrocarbons (HC), chlorofluorocarbons (CFC), hydro chlorofluorocarbons (HCFC), and mixtures. These fluids have been used for many years as refrigerants in industry and domestic applications [7, 8]. The

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[+] Corresponding author. Tel.: + (447933963158)
E-mail address: F.Latrash1@NEWCASTLE.AC.UK

CFC have a high impact on the ozone layer due to the reactivity of the chlorine atom in the compound, therefore the Montreal protocol set out a control for their consumption and production in developed countries [9, 10]. Examples of the CFC refrigerants are R11, R113 and R115. To avoid the environmental impacts of the CFC, the HCFC was introduced by adding the hydrogen atoms to reduce the impact of the chlorine resulting in lower ODP [8-10]. These HCFC shared similar physiochemical properties with CFCs however they are less reactive and have shorter atmospheric lifetime. Despite that, the Montreal protocol planned to totally phase them out by 2030 [8]. Examples of the HCFC refrigerant are R22, R123, R124 and R131. The unavailability of the chlorine atoms in HFC made them zero ODP with low GWP and raises their importance as a replacement of the CFCs and HCFC [11, 12]. R125, R245fa and R227ea are examples of the HCF. The HC such as butane, isobutene and propane contain only hydrogen and carbon atoms. They have good promising properties as refrigerants; however, they are flammable which a disadvantage is. The refrigerant mixtures were introduced to improve certain properties from mixing different refrigerants but they exhibit temperature glide due to the varying composition in the phase change process. When used in ORC these fluids can be further categorised to wet, isotropic and dry according to the exit conditions from the turbine [7, 13]. Many studies have been carried out to investigate the fluid selection in the ORC cycle. A study has shown R134a was the most suitable for such application and R152a, R600a and R290 show an attractive performance with safety concerns. Another study carried out by Mago and Luck [14] found that the best performance was obtained by using the R113 and the worst when R236fa was used. In a study of an ORC energized by vehicle engine waste heat Wang et al [15] found that R11, R141b, R113 and R123 produced slightly higher performance, and putting in mind safety concerns; R245fa and R245ca were seen as the most environmental-friendly working fluids. It found that R123 was the best working fluid when the source temperature was in the range of 100-180 ℃, but R141b was the optimal when the temperature was higher than 180 °.

This paper aims to use a previously validated model of an ORC unit powered by low temperature geothermal to assess another 15 fluids from different categories (HCFC, HCF, HC and mixtures) for their suitability for use in an ORC energised from a low temperature heat source.

2. Methodology

The ORC in this study consists of four main components: evaporator, turbine, condenser and refrigerant pump. It is necessary to emphasize that in this study the evaporator (energized by hot water from geothermal source) represents both a preheater and evaporator (where the isothermal process occurs by changing the refrigerant from saturated liquid point to saturated vapour) thermodynamically, the system is analysed assuming steady state with negligible potential and kinetic energy effects. Consequently the mass and energy balance are obtained:

$$\sum \dot{m}_i = \sum \dot{m}_e \qquad (1)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \qquad (2)$$

ORC unit net power ($\dot{W}_N$) is the gross power ($\dot{W}_G$) after subtracting the refrigerant pump power consumption

$$(\dot{W}_{pp}): \quad \dot{W}_N = \dot{W}_G - \dot{W}_{PP} \qquad (3)$$

ORC thermal efficiency (energetic efficiency) (η_I) is defined as net power divided by the heat input to the evaporator

$$\dot{Q}_E : \qquad \eta_I = \frac{\dot{W}_N}{\dot{Q}_E} \qquad (4)$$

The ORC heat exchangers (evaporator and condenser) are analyzed using effectiveness (ε) and Number of Transfer Units (NTU).

Where UA and are heat exchanger overall conductance and smaller heat capacity rate of the fluid that pass through the heat exchanger.

Evaporator and condenser effectiveness were defined as [16]:

$$NTU = \frac{UA}{C_{\min}} \tag{5}$$

$$\varepsilon_E = \frac{T_A - T_1}{T_A - T_B} \quad \text{and} \quad \varepsilon_C = \frac{T_3 - T_D}{T_D - T_C} \tag{6}$$

3. ORC Validation

A previously developed IPSEpro model for an existing 250 kW ORC unit utilizing the heat from an underground hot spring in Chena [17], Alaska was used to assess the studied refrigerants. The operational details of the ORC are shown in Table 1 [18]. The schematic of the IPSEpro model is shown in Fig. 1. During the validation procedure the same heat input to the evaporator (Q_E) at the same original existing equipment design data (e.g. evaporator and condenser effectiveness and turbine mechanical efficiency). It should be noted in this case evaporator pressure (p1) and condenser pressure (p3) will be the corresponding pressure for evaporator and condenser at fixed design temperature. That the turbine differential pressure (p1-p2) and condenser differential temperature (T2-T3) differed from those with R134a. The simulation was carried at the T_A=70 ℃, m_A =33.3 kg/s. The properties of the fluids tested is shown in Table 2. And provides a comparison between the results of the model and the data unit in Table 3 [17]. Form a reasonable estimate of the ORC and reflects the performance of the actual unit.

Table 1: Specifications of the Chena Alaska ORC unit [17]

Parameter	Unit	Value
Refrigerant	_	R134a
Heat source type	_	Hot spring water
Heat source temperature inlet	℃	73.3
Hot water mass flow rate	kg/s	33.3
Gross power	KW	250
Pump power	KW	40.0
Turbine inlet pressure	Bar	16.0
Turbine outlet pressure	Bar	4.39
Turbine mechanical efficiency	%	80
Cooling water inlet/outlet temperature	℃	4.44/10.0

Table 2: Properties of fluids studied

	Physical properties				Safety	Environmental properties		
Refrigerant	Chemical class	Tbp (℃)	Tc	Pc (bar)	class	Atmospheric life	ODPa	GWPb
1- R124	HCFC	-12	122	36.2	A1	5.8	0.022	609
2- R134a	HCF	-26	101	40.6	A1	14	0	1430
3- R142b	HCFC	-10	137	40.6	A2	17.9	0.065	2310
4-R152a	HCF	-25	113	45.2	A2	1.4	0	124
5- R227ea	HCF	-16	103	30.0	A1	34.2	0	3220
6- R236ea	HCF	6.2	139	35.0	n.a	8	0	1200
7- R245ca	HCF	25	174	39.3	n.a	6.2	0	693
8- R407D	Mixture	-39	91	45	A1	c	0	1600
9- R410A	Mixture	-52	72.0	49.0	A1	c	0	2100
10-R411A	Mixture	-40	99.1	49.5	A2	c	0.044	1600
11- R501	Mixture	-41	96.2	47.6	A1	c	n.a	n.a
12- R600a	HC	-12	135	36.3	A3	0.019	0	≈20
13- R717	inorganic	-33	132	113	B2	0.01	0	<1
14- RC318	PFC	-	114	27.8	A1	3,200	0	10,3
15- R32	HFC	-	78	57.4	A2L	4.9	0	675

a: ODP: Ozone depletion potential relative to R11, b: GWP: Global Warming Potential relative to CO_2 // c: Atmospheric life times are not given for mixtures since the components separate in the atmosphere [19].

Table 3: Comparison between model results and existing unit data [17].

Parameter	Unit	Existing unit	Model result	Difference (%)
Gross power	kW	250	250	0.0
Net power	kW	210	209	0.47
Pump power consumption	kW	40.0	40.7	1.8
ORC efficiency	%	8.20	8.04	2.0
Cooling water flow	Kg/s	101	97.7	3.3
Refrigerant flow	Kg/s	12.2	12.5	2.5
Evaporator outlet temperature	℃	54.4	54.7	0.55
Evaporator heat transfer	kW	2580	2602	0.85
Condenser heat transfer	kW	2360	2297	2.7
Evaporator heat conductance	kW/K	-	98.0	-
Condenser heat conductance	kW/K	-	594	-
Evaporator effectiveness	%	-	82	-
Condenser effectiveness	%	-	30	-
Evaporator NTU	-	-	1.71	-
Condenser NTU	-	-	1.45	-

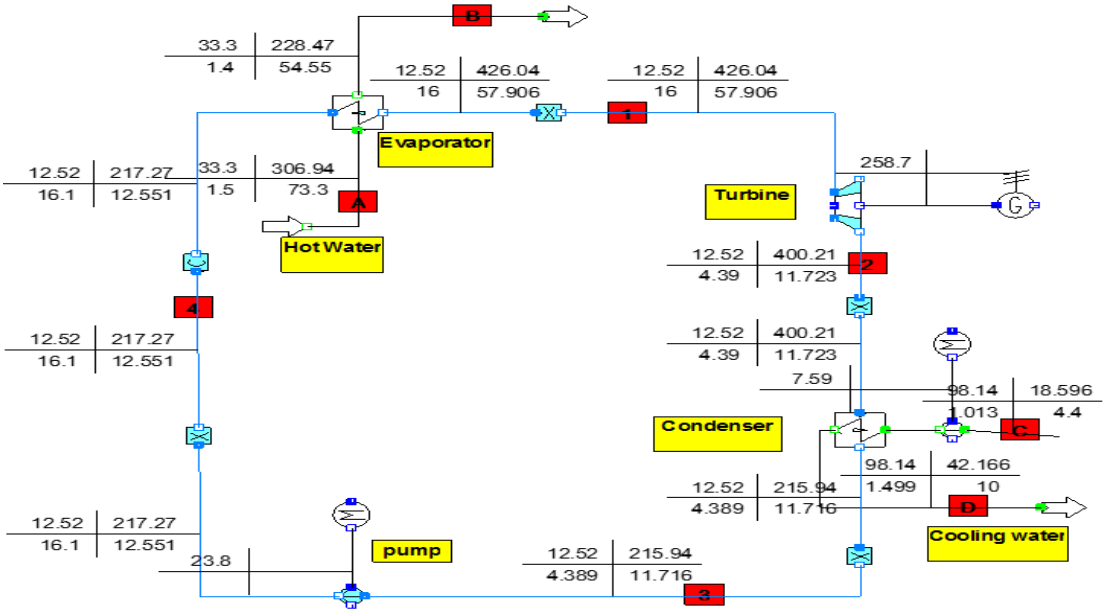

Fig. 1: IPSEpro schematic of ORC

4. Results and Discussion

The simulation results for the net power output for the different working fluids are shown in Fig. 2. R245ca was found to produce the highest power followed by the R717 and R410A showed the lowest output. R134a, R236ea and R245ca, are also attractive as they perform close to the best as does R717 and R600a. The pump power is proportional to the saturated vapour pressure at turbine inlet as shown in Fig. 3. Thus fluids with high saturation pressures have a lower net output and overall efficiency. R410A illustrates this point in Fig.s 2, 3, 4 and 5. It can be seen in Fig. 4 that R245ca, R236ea and R717 return the highest efficiency but R245ca and R236ea exhibit low condensing pressures and R410A and R717 and R32 have high evaporating pressure. These facts place these fluids low on the list of preferred fluids. Other criteria to consider are safety and environmental consideration. For example, R600a showed good performance but as it is classified as high in flammability it can be used only under special circumstances. Fluid R717 showed good thermal efficiency but it is classified as B2, low in flammability and high in toxicity. Furthermore, despite HCFC components being of low ODP the environmental preference was always given to the zero ODP fluids.

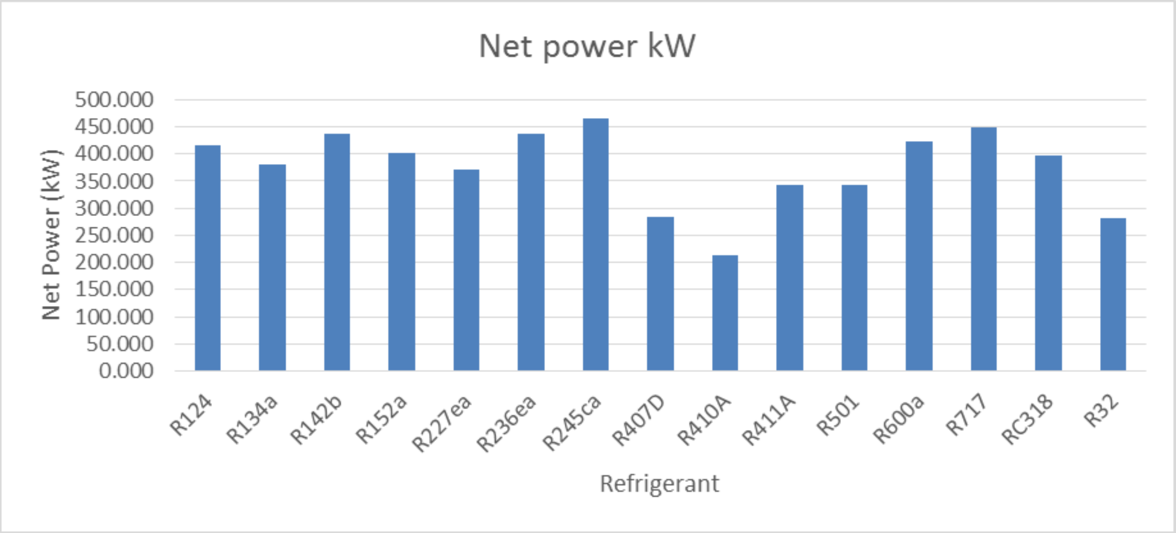

Fig. 2: ORC net power of the 15 refrigerants studied.

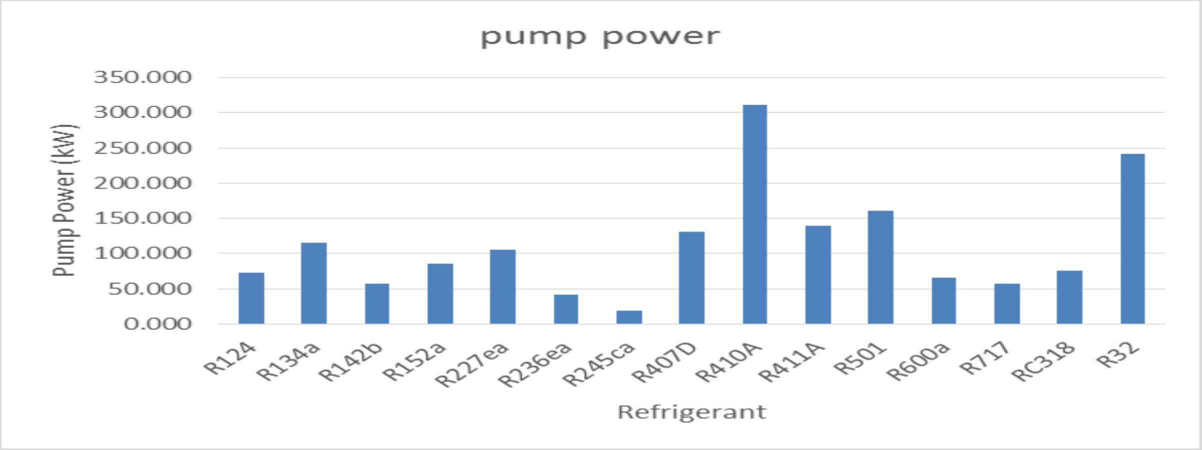

Fig. 3: ORC refrigerant pump power consumption.

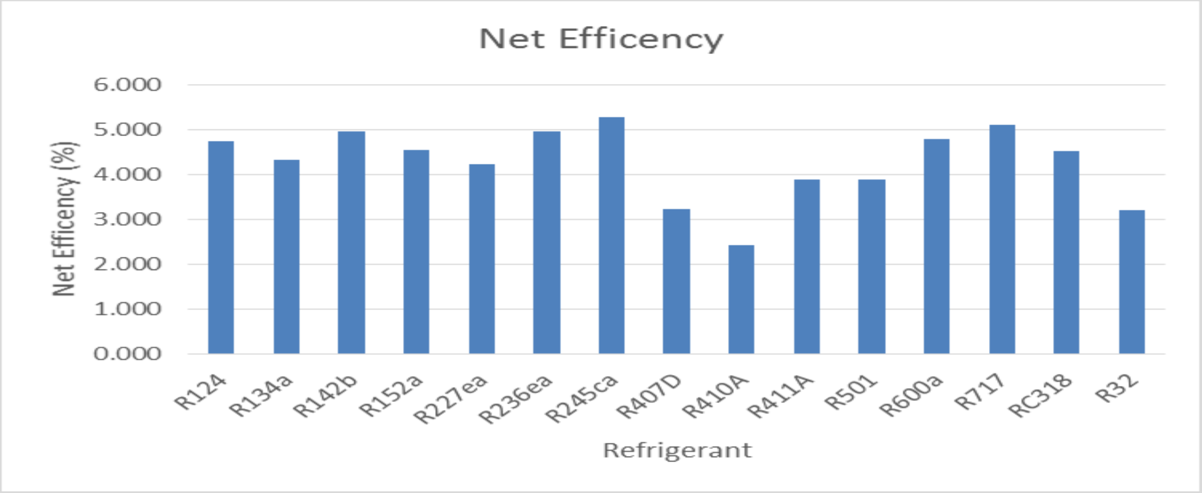

Fig. 4: ORC Net Efficiency of the 15 refrigerants studied

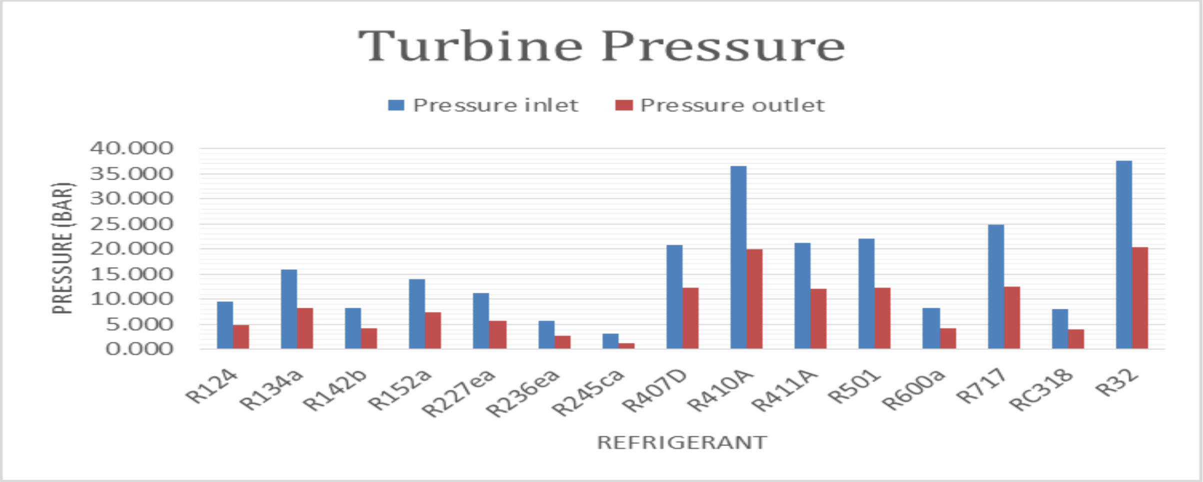

Fig. 5: ORC turbine inlet and outlet pressure for refrigerants studied.

5. Conclusions

This study used a validated model of an existing 250 kW ORC energized by a hot geothermal source to evaluate the performance of 15 working fluids. The simulation was performed maintaining the same design and operating parameters. The fluids were assessed according to net output power, refrigerant pump power consumption, evaporating and condensing pressure, safety concern and environmental concerns. The results indicated the following points:

1- From high net output power, low refrigerant pump power consumption: R142b and R124 were found the best candidates and R410A was the worst. However, refrigerants such as R134a, R236ea, R245ca, and R600a were not far from the best working fluids. All of the mixtures produced lower power output.

2- If the evaporating and condensing pressures were maintained below 25 bar and higher than 1 bar respectively the promising fluids were R142b and R245ca. The fluid R236ea could be rejected for low condensing pressure and R143a and R410A could be discarded due to high evaporating pressure. This study emphasises that at this specific operating condition, low condensing pressure fluids could be accepted if the cooling temperature increases.

3- When considering safety and environmental concerns the study revealed that R236ea and R227ea are the best working fluids for low temperature geothermal application at the specific operating conditions studied.

6. Nomenclature

ORG	Organic Rankine Cycle	-
HFC	hydro fluorocarbons	-
ODP	Ozone depletion potential	-
HCFC	hydrochlorofluorocabons	-
GWP	Global Warming Potential	-
HC	hydrocarbons	-
T	Temperature	^{o}C
CFC	chlorofluorocarbons	-
P	Pressure	Ba
UA	Heat transfer conductance	kW/m^2
NTU	Number of transfer unit	-
$\dot{m}$	Mass flow rate	kg/s
$\dot{W}$	Electrical Power	kW
Ɛ	Effectiveness	-
$\dot{Q}$	Heat transfer rate	Kw

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