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Research Article

Working fluid selection of low grade heat geothermal Organic Rankine Cycle (ORC)

Mohammed A. Al-Weshahi[A*], Fathi Latrash[B], Alexander Anderson[A] and Brian Agnew[B]

[A]School of Mechanical and Systems Engineering, Newcastle University, UK, Newcastle Upon Tyne
[B]School of Engineering and Environment, Northumbria University, UK, Newcastle Upon Tyne

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Abstract

The paper presents working fluid selection of the ORC unit powered by geothermal source. A validated model was used to assess 25 working fluids belonging to different chemical compositions, HFC, HCFC, HC, mixture and inorganic. The selection criteria were based on: net produced power, thermal efficiency, refrigerant pump power consumption, evaporating and condensing pressure, safety and environmental consideration. The study revealed that based on the selection criteria and studied ORC unit operating conditions, R236ea, R236fa and R227ea were the preferred refrigerants. Despite R141b, R123, R245ca, R717, R600 and R245fa showed an attractive thermal performance they were discarded because they did not meet the selection criteria.

Keywords: *ORC, Refrigerant, Geothermal, Selection criteria.*

1. Introduction

The high demand of electricity causes an increase of fossil fuels consumption that leads to global warming and environmental impacts. Most of the researcher's attention was driven to investigate the possibilities of utilizing the 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 ºC to 200 ºC (Yamamoto, T *et al* 2001). This type of energy could be used for direct heating or power generation (Basaran, A *and* Ozegener, L 2013). 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 a refrigerant as the working fluid instead of water so it can be powered by a low temperature source due to a low specific vaporization heat (Papadopoulos, A. I *et al*, 2010; Schuster, A *et al*, 2009; Quoilin, S *et al*, 2013). The proper selection of the ORC refrigerant is the key for achieving high performance and a cost-effective ORC unit (Tchanche, B. F *et al* 2009). The good refrigerant candidate owns low specific volume, high efficiency, acceptable pressure within cycle, low ODP, low GWP, low cost, low toxicity and high safety (Yamamoto, T *et al*, 2001; Tchanche, B. F *et al*, 2009).
The refrigerants are classified according to their chemical composition into chlorofluorocarbons (CFC), hydrochlorofluorocarbons (HCFC), hydrofluorocarbons (HFC), hydrocarbons (HC) and mixtures. The CFC refrigerants are non-toxic, non-flammable, non-reactive and quite safe and for these reasons have been used for years in industry (Li, J *et al*, 2012; Hundy, G. F *et al*, 2008). However, they have a high impact on the ozone layer due to the availability of the chlorine atom in the compound, therefore the Montreal protocol set out a control for their consumption and production in developed countries (Kim, H *et al*, 2011; Boot, J. L. E, 1990). Examples of the CFC refrigerants are R11, R12, R113, R114 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 (Hundy, G. F *et al*, 2008; Kim, H *et al*, 2011; Boot, J. L. E, 1990) resulting in lower ODP. These HCFC shared similar physiochemical properties similar to CFC however they are less reactive and have shorter atmospheric lifetime. Despite that, the Montreal protocol planned to totally phase out them by 2030 [Hundy, G. F *et al*, 2008]. Examples of the HCFC refrigerant are R22, R123, R124 and R131. The unavailability of the chlorine atoms in HCF made them zero ODP with low GWP and raises their importance as a replacer of the CFC and HCFC [Saleh, B *and* Ozegener, L 2006; Granryd, E, 2001]. R134a, R245fa, R152a and R227ea are examples of the HCF. The HC refrigerants contain the hydrogen and carbon atoms such as butane, isobutene and propane. They have good promising properties as refrigerants; however, the inflammability is their disadvantage. The mixture refrigerants introduced to improve certain properties from mixing different refrigerants. They can be classified into two groups; azeotrope and zoetrope. The azoetrope refrigerants can be considered as one compound because they exhibit the similar composition at liquid and vapor

*Corresponding author: **Mohammed A. Al-Weshahi**

phase composition; however, the zoetrope shows different composition at the two phases (Basaran, A *and* Ozegener, L 2013). The azoetrope refrigerants are indicated by R5xx while the zoetrope refrigerants are named as R4xx. In addition some researchers divided the ORC refrigerants based on their phase state when leaving the ORC turbine: wet, isotropic and dry. The wet refrigerants have a negative slope of saturated vapor curve and in general they have a low molecular mass such as ammonia (R717). The isentropic fluids have a vertical saturated vapor line with a moderate molecular mass such as R123, R141b and R600. The dry fluids own a positive slope of saturated vapour line with high molecular mass such as R601a and R601 (Li, T *et al*, 2012; Qiu, G, 2012).

Many studies have been carried out to investigate the fluid selection in the ORC cycle. Tchanche *et al.* studied 20 different refrigerants in the ORC cycle powered by hot water gained from solar energy. The study revealed that R134a was the most suitable for such application and R152a, R600, R600a and R290 show an attractive performance with safety concerns. Nine pure refrigerants were evaluated as the working fluid in the ORC energized by vehicle engine waste heat by Wang *et al.* The study found that R11, R141b, R113 and R123 produced slightly higher performance; however, putting in mind safety concerns; R245fa and R245ca were seen as the most environmental-friendly working fluids for such application. Mago and Luck examined the heat recovery from micro turbine exhaust using the ORC with four different dry refrigerants; R113, R123, R245fa and R236fa. The study concluded the best performance was achieved using the R113 and the worst when R236fa was used (Mago, P.J *and* Luck, R 2012). Another study used a multi-function objective optimization method to select the best ORC refrigerant among 13 working fluids. It found that R123 is the best working fluid when the source temperature is in the range of 100-180°C; however, R141b was the optimal when the temperature was higher than 180°C (Wang, E. H *et al* 2012). Aljundi also compared different ORC working fluids and the result revealed that n-hexane was the best working fluid whereas the R227ea was the worst. The study claimed the hydrocarbons could be the future working fluids for geothermal and waste heat applications.

This study aims to use the previously validated model of ORC unit powered by low temperature geothermal to assess another 25 refrigerants from different categories (HCFC, HCF, HC and mixtures) to select the best candidates. Throughout this comparison the modelled existing ORC unit equipment design data were maintained constant.

2. Methodology

The studied ORC unit in this study consists of four main components: evaporator, turbine, condenser and refrigerant pump. It necessary to emphasize in this study the evaporator (energized by hot water from geothermal source) represents both preheater (moves the refrigerant state from sub-cooled to saturated liquid) and evaporator (where the isothermal process occurs by changing the refrigerant from saturated liquid point to saturated vapor) Thermodynamically, the system is analyzed with assumption: as control volume at a steady state with negligible potential and kinetic energy effects. Consequently the mass and energy balance are obtained by:

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{1}$$

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

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

$$\dot{W}_N = \dot{W}_G - \dot{W}_{PP} \tag{3}$$

ORC thermal efficiency (energetic efficiency) (η_I) is defined as net power divided by the heat input to the evaporator $\dot{Q}_E$:

$$\eta_I = \frac{\dot{W}_N}{\dot{Q}_E} \tag{4}$$

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

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

where UA and C_{min} 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 (Herold, K. E *et al* 1996):

$$\varepsilon_E = \frac{T_A - T_1}{T_A - T_B}$$

$$\varepsilon_C = \frac{T_3 - T_D}{T_D - T_C} \tag{6}$$

3. ORC modelling and validation

The previously developed model (Al-Weshahi, M. A et al, 2014) for an existing 250 kW ORC unit utilizing the heat from an underground hot spring in Chena, Alaska (Table 1) (Alaska Energy Authority Chena Power, 2007) using the IPSEpro refrigeration library (SimTec, 2005) was used to assess the studied refrigerants. Fig. 2 represents the original ORC unit state points on the R134a refrigerant p-h diagram. The process from 1 to 2 represents the expansion process in the turbine of the slightly superheated vapor to the vapor-liquid state at the turbine exit. The point 2 to 3 describes the constant-pressure heat rejection process in the condenser while 3-4 shows the refrigerant pump compressing the refrigerant to the operating pressure of the preheater. The process 4 to 1 shows the constant pressure heat addition in the preheater and evaporator. The comparison between the model results and existing unit data is presented in Table 2 (Al-Weshahi, M. A et al, 2014). The model reflects a reasonable estimation of the actual ORC unit performance.

Table 1 Specifications of the Chena Alaska ORC unit (Alaska Energy Authority Chena Power, 2007).

Parameter		Value
Refrigerant	-	R134a
Heat source type	-	Hot spring water
Heat source temperature inlet	°C	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	°C	4.44/10.0

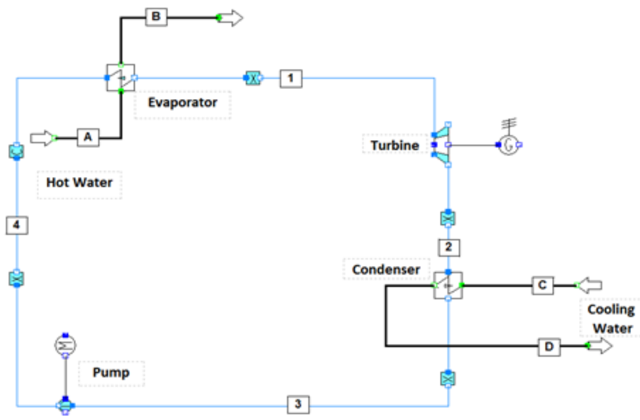

Fig. 1 IPSEpro schematic of ORC (Al-Weshahi, M. A et al, 2014)

4. Results and discussion

To assess the role of refrigerant selection, a comparison between different refrigerants was performed for the same heat input to the evaporator ($\dot{Q}_E$) 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 (p_1) and condenser pressure (p_3) will be the corresponding pressure for evaporator and condenser at fixed design temperature. Therefore, the turbine differential pressure (p_1-p_2) and condenser differential temperature (T_2-T_3) differed from those with R134a. The simulation was carried at the T_A=70 °C, $\dot{m}_A$ =33.3 kg/s, with saturated vapour at turbine inlet and similar cooling water parameters at Chena. The studied refrigerants properties are illustrated in Table 3.

The simulation results for the 25 refrigerant net produced powers are shown in Fig. 3. The R141b was found producing the highest power followed by the R123. On other hand, R125 showed the lowest production. HFCs, such as R134a, R236ea, R236fa, R245ca, and R245fa, were noted as attractive and not far from best. Similarly the R717 (ammonia), R600 (Butane) and R600a (Isobutene) was seen with good production. In general, all mixture refrigerants studied were observed producing less than 200kW net power.

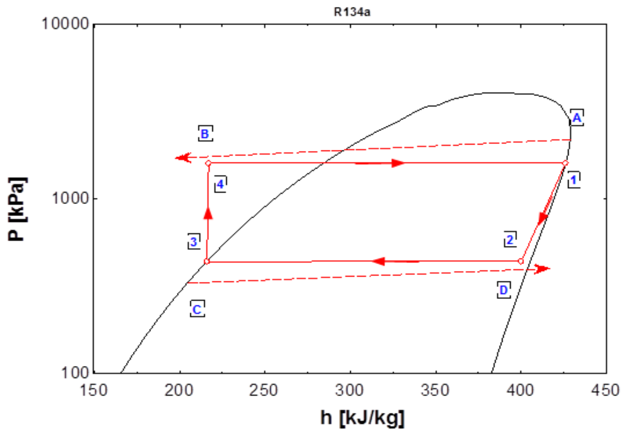

Fig. 2 p-h diagram of the Chena ORC unit using R134a refrigerant (Al-Weshahi, M. A et al, 2014).

Fig. 4 illustrates the refrigerants thermal efficiency at the studied parameters. In general, the thermal efficiency of the ORC for all the refrigerants was low (less than 10%). The trend of thermal efficiency was similar to the net power. The reduction of the net produced power, and consequently thermal efficiency, was strongly linked with refrigerant pump power consumption. The higher the pump power consumption is the lower the net produced power and thermal efficiency.

The refrigerants' pump power consumption is shown in Fig. 5. The pump power is proportional to the required saturated vapor pressure at turbine inlet, the higher turbine inlet pressure resulting in higher pump power consumption. As can be seen, the pump with low consumption achieved better thermal efficiency.

Fig. 6 illustrates the different ORC refrigerants' turbine inlet and outlet pressure. The turbine inlet pressure is obtained as saturated vapour pressure required evaporator to maintain 82% effectiveness; whereas the outlet pressure is corresponding to the condensing temperature (T_3) results in 30% condenser effectiveness. The acceptable evaporating pressure (turbine inlet pressure) and condensing pressure (turbine outlet pressure) are lower than 25 bar and higher than 1 bar, respectively (Tchanche, B. F et al 2011). For current operating parameters of cooling water and geothermal source, refrigerants that could be rejected for low condensing pressure are R141b, R123, R245ca and R245fa even though they showed a good thermal efficiency previously. Moreover, R125, R143a and R410A are discarded due to high evaporating pressure. It is worth mentioning that condensing pressure was increasing as cooling water temperature rose to maintain constant condenser effectiveness; therefore fluid with low condensing pressure could be accepted in hotter cooling water temperature (e.g. hot countries).

Table 2: Comparison between model results and existing unit data (Al-Weshahi, M. A et al, 2014)

Parameter		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	°C	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	-

Table 3: Studied refrigerant properties (Wang, E. H *et al* 2012; Bitzer International; BOC GASES; ASHRAE, 2009; http://webbook.nist.gov)

Refrigerant		Chemical class	Physical properties			Safety class	Environmental properties		
			T_{bp} (ºC)	T_c	P_c (bar)		Atmospheric life	ODP[a]	GWP[b]
1	R123	HCFC	27	184	36.6	B1	1.3	0.02	77
2	R124	HCFC	-12	122	36.2	A1	5.8	0.022	609
3	R125	HFC	-79	66.0	36.2	A1	29	0	3500
4	R134a	HCF	-26	101	40.6	A1	14	0	1430
5	R141b	HCFC	32	204	42.1	n.a	9.3	0.11	725
6	R142b	HCFC	-10	137	40.6	A2	17.9	0.065	2310
7	R143a	HCF	-47	72.7	37.6	A2	52	0	4470
8	R152a	HCF	-25	113	45.2	A2	1.4	0	124
9	R227ea	HCF	-16	103	30.0	A1	34.2	0	3220
10	R236ea	HCF	6.2	139	35.0	n.a	8	0	1200
11	R236fa	HCF	-1.4	124	32.0	A1	240	0	9810
12	R245ca	HCF	25	174	39.3	n.a	6.2	0	693
13	R245fa	HCF	15	154	36.4	B1	7.6	0	1030
14	R407A*	Mixture	-46	83	45.2	A1	c	0	2100
15	R407B	Mixture	-48	76	40.8	A1	c	0	2800
16	R407C	Mixture	-44	87	46	A1	c	0	1800
17	R407D	Mixture	-39	91	45	A1	c	0	1600
18	R407E	Mixture	n.a	n.a	n.a	A1	c	0	1600
19	R410A	Mixture	-52	72.0	49.0	A1	c	0	2100
20	R411A	Mixture	-40	99.1	49.5	A2	c	0.044	1600
21	R411B	Mixture	-42	96.0	49.5	A2	c	0.047	1700
22	R501	Mixture	-41	96.2	47.6	A1	c	n.a	n.a
23	R600	HC	0.0	152	38.0	A3	0.018	0	≈20
24	R600a	HC	-12	135	36.3	A3	0.019	0	≈20
25	R717	inorganic	-33	132	113	B2	0.01	0	<1

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 (Calm, J. M *and* Domanski, P. A, 2004)

*: Mixtures composition illustrated in appendix A.

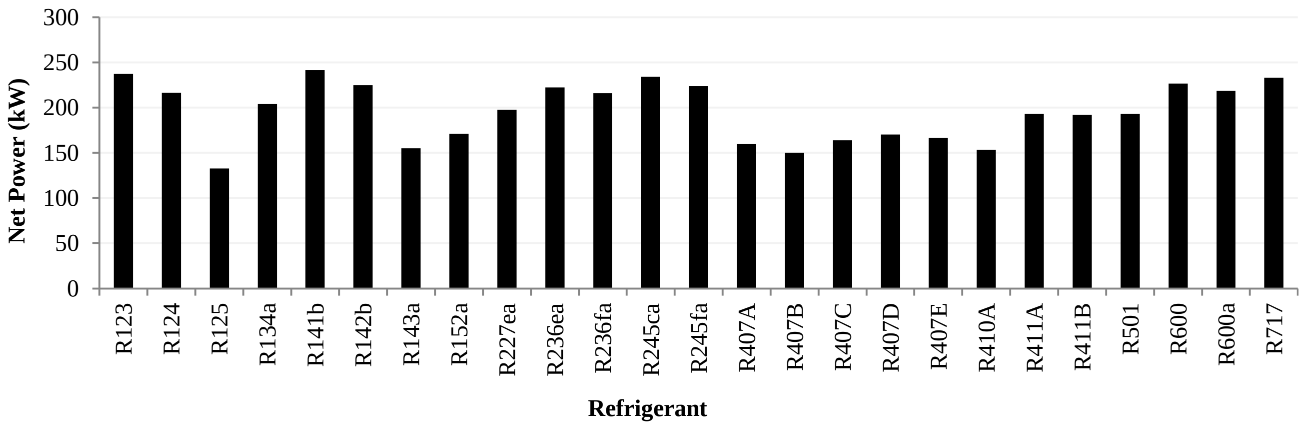

Fig. 3: ORC net power of the 25 refrigerants studied.

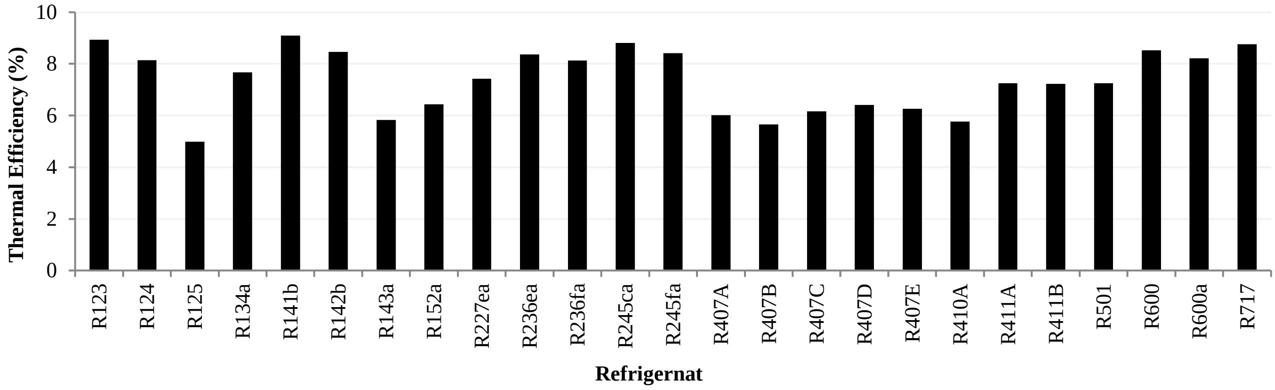

Fig. 4: ORC thermal efficiency of the 25 refrigerants studied.

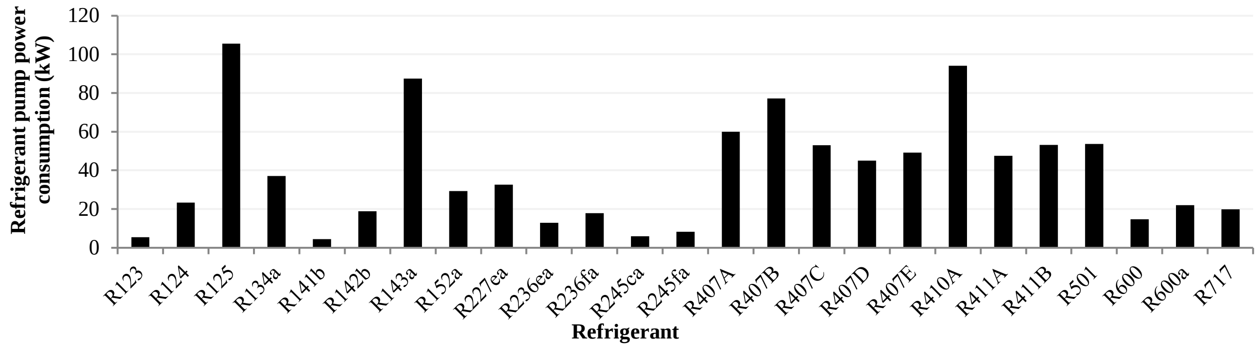

Fig. 5: ORC refrigerant pump power consumption.

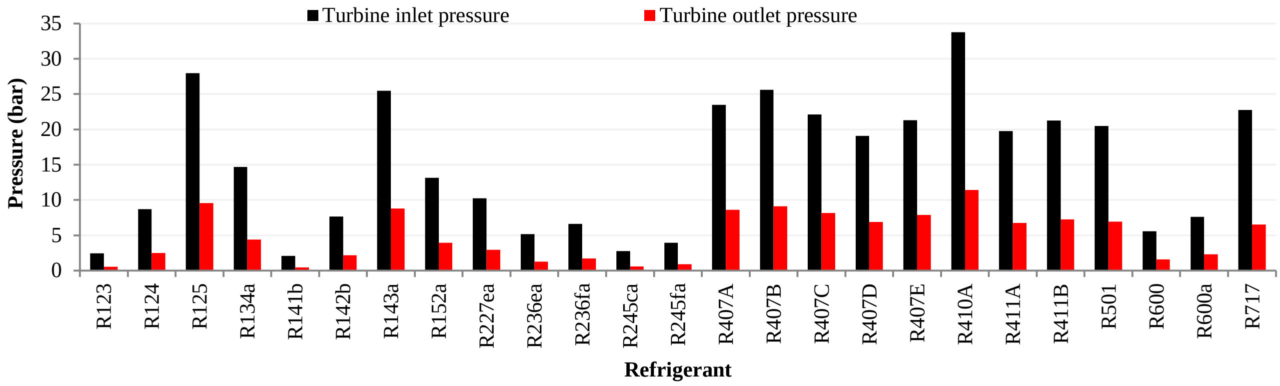

Fig. 6: ORC turbine inlet and outlet pressure for refrigerants studied.

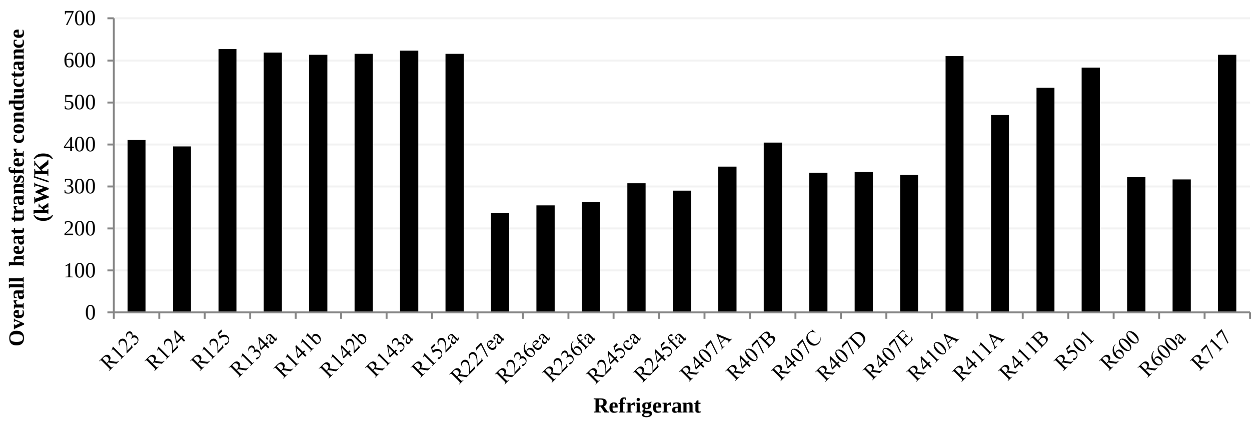

Fig. 7: Overall heat transfer conductance of the condenser for refrigerants studied.

The size of the heat exchangers is playing a role in selecting the optimal fluid for the ORC. Fig. 7 describes the overall heat transfer conductance (UA), the smaller the product the smaller the heat exchanger size to change the refrigerant ORC turbine outlet state phase to saturated liquid phase. Promising fluids such as R134a, R152a, R501 and R717 could be excluded from the shortlisted optimal fluids due to higher UA heat exchange value. Interestingly, R227ea was found with the lowest heat exchanger UA value and, again, R125 with the highest. It is necessary to pinpoint the higher the UA resulting from high absorbed heat from the refrigerants which causes a higher cooling water flow to remove the heat allowing phase changing to the saturated liquid.

Although the previous discussed criteria are important the safety and environmental consideration should not be ignored. For example, R600a and R600 showed positive performance and design indicators; however, being under A3 safety categories (higher inflammability) reduced its credit. In addition, R717 showed good thermal efficiency but it was under B2 categories (lower inflammability and higher toxicity) and making the decision to use them, it requires extra precautions. Furthermore, despite HCFC components being of low ODP the environmental preference was always given to the zero ODP fluids.

5. Conclusion

The study used validated model of an existing 250 kW ORC energized by hot geothermal source. Maintaining similar design and operating parameters; 25 refrigerants were assessed based on net output power, thermal efficiency, refrigerant pump power consumption, condenser UA, evaporating and condensing pressure, safety concern and environmental concerns. From the results the following conclusions could be inferred:

- From high net output power, more thermal efficiency and low refrigerant pump power consumption aspects: R141b and R123 were found the best candidates and R125 was the worst. However, refrigerants such as R134a, R236ea, R245ca, R245fa, R717 and R600 were not far from the best working fluids. All of the mixtures were producing lower power output, less than 200kW.
- Maintaining the acceptable evaporating and condensing pressure lower than 25 bar and higher than 1 bar: promising refrigerants such as R141b, R123, R245ca, R245fa 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 refrigerants could be accepted if the cooling temperature increases.
- Based on heat exchange UA selection criteria, R134a, R152a, R717 are not preferred due to higher value. It was noted that R227ea achieved the lowest UA.
- Putting beside previous selection criteria: safety and environmental concerns, the study revealed the R236ea, R236fa and R227ea are the best working fluids for low temperature geothermal application at the specific operating conditions studied.

Appendix A: (ASHRAE, 2009)

Refrigerant	Composition
R407A	R32/R125/R134a (20%,40%,40%)
R407B	R32/R125/R134a (10%,70%,20%)
R407C	R32/R125/R134a (23%,25%,52%)
R407D	R32/R125/R134a (15%,15%,70%)
R407E	R32/R125/R134a (25%,15%,60%)
R410A	R32/R125 (50%,50%)
R411A	R1270/R22/R152a (1.5%,87.5%,11%)
R411B	R1270/R22/R152a (3%,94%,3%)
R501	R22/R12 (75%,25%)

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