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Thermodynamical and Economical Valuation of Low-temperature ORC with Alternative Working Fluids

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ABSTRACT

In order to use the comparatively low German thermal water temperatures of 120 to 160°C effectively by deep geothermal power stations, the Organic-Rankine-Cycle (ORC) is deployed. Due to the substantial global warming potential of the working fluids currently in use and the lawfully compulsion of their replacement in the near future, alternative working fluids to operate power plants using the ORC-process need to be found. For that purpose, this thesis evaluates a selection of organic fluids with a power plant simulation. The most productive candidates are compared with conventional working fluids with regard to power output, cost and material compatibility. Furthermore, necessary technical and economical effort for the substitution of the working fluids will be examined.

1. INTRODUCTION

The reduction of climate active gases is one of the main aim of the German government. With the intensive installation of renewable energies the target fo 35% up to 2020 and 80% until 2050 should be reached BMU (2012).

The integration of volatile renewables into the electricity grid is an enormous challenge. While their contribution is increasing, energy supply and demand are increasingly diverging. In order to compensate for the fluctuating nature of renewables, base load-capable power plants, which play the role of coal-fired and nuclear power plants, are needed due to the absence of storage capacities.

This is where geothermal energy can make a contribution. It is a technology with a high proportion of full load hours because it is not subject to daily or annual fluctuations.

The thermal water temperatures between 100°C and 200°C found in Germany are rather low for electricity generation. Binary processes are used in Germany to enable the efficient geothermal use of sites with lower temperature levels, so-called low-enthalpy reservoirs Vetter (2013).

The Clausius-Rankine cycle in conventional steam power plants is based on water as a working fluid at high temperatures. The use of organic fluids based on carbon compounds in the low-temperature process of binary geothermal plants gives it the name Organic Rankine Cycle (ORC).

Up to now, isopentane, R134a and R245fa have been primarily used for this purpose Heberle (2017). The last two are refrigerants and have a very high greenhouse gas potential or global warming potential (GWP), of several thousand CO_2 equivalents. New laws Becken (2015) require plant operators to replace these substances with more climate-friendly substances in the next years. Potential candidates as substitutes are hydroolefins (HFO-1234yf , HFO-1234ze-E).

The present study compare the standard working fluids from the NIST database NIST (2010) for the operation in low temperature ORC with more climate-friendly fluids like propane or hydroolefins. For this purpose, simulations with an Matlab in house code GESI Vetter (2014) are used to determine which fluids would be suitable due to their thermodynamic properties and which effects the exchange of the working fluid would have on the power plant performance, necessary changes to power plant components and thus on the specific costs for the plant operators.

2. MODELING

2.1 ORC

The Rankine cycle, using water as a working fluid, is state-of the-art in coal, gas and nuclear power plants. Water, however, is not a suitable working fluid for converting low-temperature heat, due to high evaporation temperature at ambient pressure; therefore, in Organic Rankine Cycles (ORCs) organic fluids with lower vaporization temperatures are used. Fig. 1 shows the ORC using propane as a working fluid in the Temperature entropy (Ts)-diagram, as well as process flow. The working fluid is compressed in the feed pump, heated up and evaporated in the heat exchanger before it is expanded in the turbine. In the last step, which closes the cycle, the remaining heat is removed in the condenser. Changes in state of an ideal process are as follows:

- 1-2: isentropic compression, supply of work to the cycle

- 2-3: isobaric supply of heat (heat exchanger)

- 3-4: isentropic expansion, submission of work out of the cycle

- 4-1: isobaric removal of heat (condenser).

Depending on the pressure at which the heat is supplied, the process is either subcritical, with the fluid evaporating as it passesthrough the two-phase region (bold line, Fig. 1), or supercritical (dotted line, Fig. 1). Location of the critical point (CP) depends on the fluid. From the specific enthalpy differences Δh between the individual state points, the specific energy contribution of each component can be calculated:

- Work supplied in the feed pump: w_{pump} = h2 - h1

- Heat supplied in the heat exchanger: q_{in} = h3 - h2

- Specific work of the turbine: $w_{turbine}$ = h3 - h4

- Heat removed in the condenser: q_{out} = h4 - h1

By this the thermal efficiency of the cycle η_{ORC} can be calculated

$$\eta_{ORC} = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} \quad (1)$$

The previously described process is an ideal case, that in reality is affected by losses. Pressure losses in the pipes, heat exchanger and condenser cannot be avoided. In addition, there are losses during compression in the pump and expansion in the turbine. These losses result in an increase in entropy during compression and expansion. This can be described with the isentropic pump η_{pump} and turbine efficiency $\eta_{turb.}$

$$\eta_{pump} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (2)$$

$$\eta_{turb} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad (3)$$

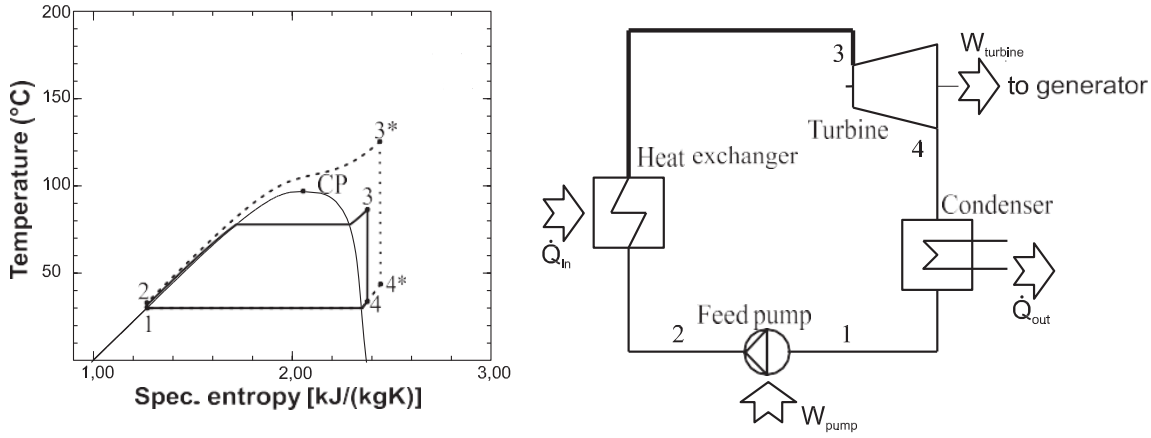


Figure 1: Sub- and supercritical ORC in a Temperature entropy-diagram (left side, fluid data from Vetter (2014)) and process flow diagram (right side).

2.2 Heat exchanger

Heat input can be calculated using an energy balance over the heat exchanger.

The left side of the formula is the heat extracted from the geothermal fluid. This can be calculated from the temperature difference of the geothermal fluid between entry and exit from the heat exchanger, and the mean specific isobaric heat capacity c_p of the geothermal fluid. This heat leads to increased enthalpy and, therefore, to evaporation of the organic fluid on the secondary side.

In this study, the use of a counter current heat exchanger is assumed. The finite heat exchange area leads to temperature differences between the two fluids at all points of the heat exchanger. An important design parameter for the heat exchanger is a minimal temperature difference (MTD) between the two fluids.

The point where this MTD occurs, the Pinch Point, depends on two factors:

1. The pressure and temperature of the organic fluid exiting the heat exchanger, and
2. the temperature of the geothermal fluid at the entry point.

In consequence, it is necessary to iteratively evaluate the lowest possible temperature of the geothermal fluid at the heat exchanger's exit. This is achieved by the Matlab program GESI Vetter(2014)
The starting value for the iteration is the temperature of the organic fluid at the entry point of the heat exchanger plus the MTD. To check compliance with the minimal temperature difference at all points in the heat exchanger, the program calculates the temperature profiles of the geothermal and organic fluid, depending on the heat exchanged. If the MTD falls below the allowed value, as is shown in figure 2, the program aborts the calculation and starts with a higher geothermal fluid exit temperature. This is performed until the MTD is maintained at all points. By establishing the temperatures at this stage, the missing parameters of the cycle can be calculated. For the present study a pinch point of 5 K is assumed.

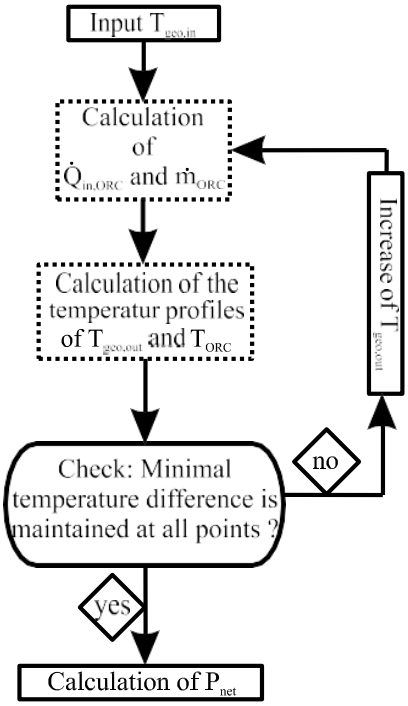


Figure 2: Iteration schema of GESI

2.3. Specific net power output

In order to compare different working fluids this paper focuses on the specific net power output of the thermodynamic cycle. This is the relevant variable for geothermal application and waste heat utilization. If the energy is extracted from a closed loop (e.g. CHP systems) or from a valuable energy source, one has to compare the efficiency of the processes. Furthermore, the net power output e considering the electricity demands of the cycle pump and condenser e is taken into account to evaluate the thermodynamic systems. The electricity demand of the geothermal production pump is not considered in the calculations because of its site-specific aspects. Economic aspects can also play an important role in gross power output; this is not addressed in the present work. The net power output of the cycle is the product of thermal efficiency and the heat supplied to the organic fluid.

$$P_{net} = \eta_{th} \cdot \dot{m} \cdot q_{in} = (w_{turb} - w_{pump}) \cdot \dot{m} \tag{4}$$

$$P_{net,spec} = \frac{P_{net,out}}{\dot{m}_{geo}} \left[\frac{kWs}{kg}\right] \tag{5}$$

In this equations the ORC mass flow $\dot{m}$, the net power output of the ORC P_{net} , the specific net power output of the ORC Pnet,spec with the geothermal water mass flow $\dot{m}_{geo}$

2.4. General Settings

Table 1: General settings for the simulation

Thermal water

pressure	9 MPa
inlet temperature	100 - 300 °C
mass flow	1 kg/s

ORC- process

Max. Steam content after the turbine	0,9
pump efficiency	0,7
turbine efficiency	0,8
min. Temperature difference capacitor	5 K
Condenser pressure loss	0,02 MPa
condenser temperature	30 °C
Cooling water Pressure	0,3 MPa
Cooling water Inlet temperature	19 °C
min. Temperature difference heat exchanger	5 K
Pressure loss heat exchanger	0,02 MPa

2.5 Power output

The GESI calculation with the settings of table 1 is shown in figure 3.

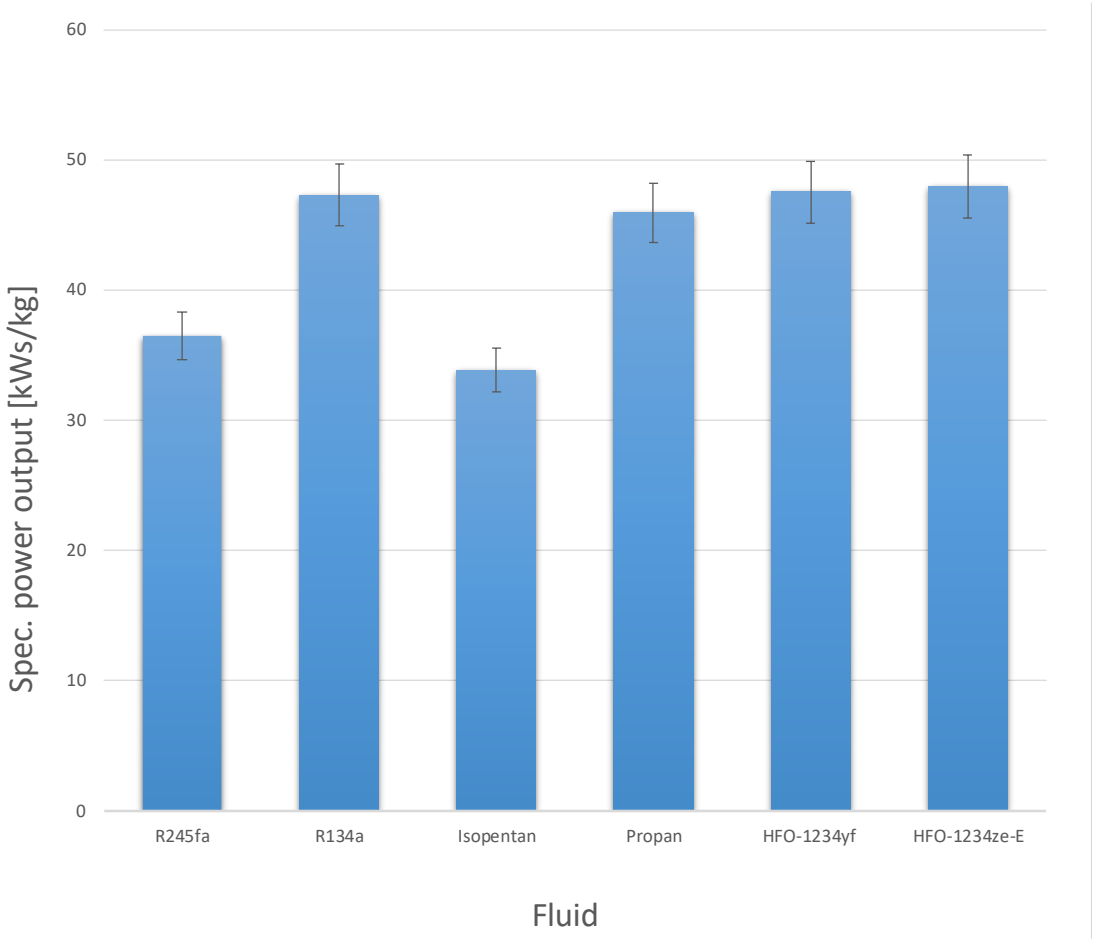


Figure 3: Specific power output of the examine fluids ($P_{net,spec}$).

2.6. Cost calculation

The investment and operating costs for each power plant component are calculated in the following section.

2.6.1 Feed pump

The feed pump compresses the working fluid after the condenser to the required pressure. The power required for this P_{pump} is calculated using the correlation in equation 6 from Tester (1976).

$$P_{pump} = 0{,}028 \cdot Z_c \cdot \frac{P_{net}}{\eta_P} \cdot \frac{1-\eta_{ORC}}{\eta_{ORC}} \cdot \frac{\Delta p_r}{(1-T_r)^{\frac{2}{3}}} \quad (6)$$

The values for Z_C are assumed to be 0.27 in the work of Tester (1976).

The reduced pressure difference Δp_r is calculated from the difference between the maximum pressure, the live steam pressure p_3, and the minimum pressure p_4, which the fluid reaches after the condenser by losses, divided by the fluid-specific critical pressure p_C from Table 3:

$$\Delta p_r = \frac{p_{max} - p_{min}}{p_c} \quad (7)$$

The reduced temperature T_r is the ratio of the temperature after the capacitor T_0 to the critical temperature T_c

$$T_r = \frac{T_0}{T_c} \quad (8)$$

The efficiency of the pump η_P is set to 0.7 analogous to the calculations carried out with GESI. The efficiency of the ORC process η_{ORC} , the net plant power output P_{net} as well as the maximum and minimum pressures p_{max}, p_{min} in the ORC process were determined fluid-specifically with GESI and are summarized in Table 2. The efficiency η_{ORC} results from the ratio of P_{net} to Q_{zu} . The pressures are calculated from the given live steam pressure and the pressure losses.

Table 2: ORC circuit pressures and efficiency

Fluid	$\boldsymbol{p}$min [MPa]	$\boldsymbol{p}$max [MPa]	$\boldsymbol{\eta}_{ORC}$ [%]
R134a	0,77	5,02	10,95
R245fa	0,18	1,02	10,04
Propan	1,08	6,02	11,29
HFO-1234yf	0,78	6,02	10,36
HFO-1234ze-E	0,57	5,02	11,23
Isopentan	0,11	0,62	9,88

Using these data in equation 4.1, the necessary pumping power P_{pump} results to:

Table 3: Pump power

Fluid	$\Delta \boldsymbol{p}_r$	$\boldsymbol{P}_{pump}$**[kW]**
R134a	1,05	13,11
R245fa	0,03	0,26
Propan	1,16	14,55
HFO-1234yf	1,55	22,80
HFO-1234ze-E	1,29	15,24
Isopentan	0,15	1,03

The purchased equipment costs, PEC, can be calculated using a correlation from F. Marty (2017):

$$log_{10}PEC = K_1 + K_2 \cdot log_{10}X + K_3(log_{10}X)^2 \quad (9)$$

X stands here for the performance of the pump (P_{pump}) in kW. Thus, the investment costs in this correlation depend only on the performance.

The constants for pumps have the following values F. Marty, (2017): $K1$=3,3892; $K2$=0,0536; $K3$=0,1538

The result of the calculation is stated in US dollars.

Table 4: Investmentcost feed pump

Fluid	Investment cost [$]	Investment cost [€]
R134a	3162,3	2687,7
R245fa	2449,1	2081,6
Propan	3981,1	3383,8
HFO-1234yf	4570,8	3884,5
HFO-1234ze-E	4073,8	3462,0
Isopentan	2454,7	2085,9

2.6.2 Turbine

In the turbine, the live steam is expanded while work is released. The rotating shaft drives a generator which transforms the mechanical rotational energy into electrical energy.

The turbine pitch (average wheel) diameter, D_{pitch}, was derived from a universal functional relationship, for optimum stage efficiency Balje (1962) as:

$$D = \frac{\dot{V}}{0{,}177 \cdot \sqrt{\Delta h}} \tag{10}$$

The enthalpy difference Δh refers to the difference before and after decompression in the turbine h_3-h_4 and comes from the simulations with GESI:

Table 5: Working fluids, enthalpy difference and molar mass

Working fluid	Enthalpy difference [kJ/kg]	Molar mass [g/mol]
R134a	26,24	102,03
R245fa	23,86	134,05
Propan	58,5	44,1
HFO-1234yf	30,08	114
HFO-1234ze-E	26,13	114,04
Isopentan	46,13	72,15

The volume flow $\dot{V}$ is the ratio between mass flow $\dot{m}$ and density ρ of the working fluid. The mass flow was determined with GESI.

Live steam density of the fluid is considered at the turbine inlet using the specific gas constant R_S of the working fluid and the pressure p_3 and temperature T_3. The required turbine diameter then results from equation 10.

Table 6: average Turbine wheel diameter

Working fluid	Turbine diameter [cm]
R134a	15,8
R245fa	4,2
Propan	9,6
HFO-1234yf	13,9
HFO-1234ze-E	21,5
Isopentan	43,5

The investment costs *P*EC of the turbine can be also calculated with the help of the correlation 9 from Marty (2017). The constants are changed to adapt to the turbine.

X=$D_{turbine}$ stands for the diameter of the turbine in mm. The constants K for turbines have the values in the literature: $K1$=2,2476; $K2$=1,4965; $K3$=−0,1618

The result of the calculation is stated in US dollars.

Table 7: Investcosts for a new turbine

Working fluid l	Investment cost [$]	Investment cost [€]
R134a	53703	45647
R245fa	17139	14568
Propan	36307	30860
HFO-1234yf	91201	77520
HFO-1234ze-E	69023	58669
Isopentan	112201	9537

2.6.3 Heat exchanger

In the heat exchanger, thermal energy from the thermal water is transfered to the ORC circuit. Different working fluids provide different heat capacities, which has effects on the ideal live steam temperature $T_{ORC,out}$, as well as the transferred heat flow Q_{in}. In order to achieve the necessary live steam temperature nevertheless, the exchange surface A of the heat exchanger must be adapted.

The heat capacity of the fluid does not occur in the formula itself, it enters indirectly via the values calculated by GESI (Q_{in} transferred heat to the ORC). The following formulas apply to a countercurrent heat exchanger with U as heat transfer coefficient (W/m^2K) and $\Delta T_{m,log}$ as mean logarithmic temperature (K).

$$A = \frac{Q_{in}}{U \cdot \Delta T_{m,log}} \tag{11}$$

One simplification of the approach presented here is the assumption of a constant value of 718 W/m^2K for the heat transfer coefficient U for all working fluids. This is derived from simulations with propane as the working fluid and a fully designed Heat exchanger. Due to this simplification, the differences in the required heat exchanger surfaces are based only on the divergent heat outputs and temperature differences. For a more accurate calculation of the heat exchanger area, it would be necessary to simulate the heat transfer coefficient for each fluid individually. However, this will exceed the scope of this work.

The resulting heat exchanger are is listed in the following table:

Table 8: Heat transfer area

Working fluid	Heat transfer area [m^2]
R134a	25,75
R245fa	11,35
Propan	25,27
HFO-1234yf	39,70
HFO-1234ze-E	30,21
Isopentan	10,36

The investment cost for the heat exchanger PEC calculated with the correlation 9 from Marty (2017) with X=A, $K1$=4,3247 $K2$=−0,303 $K3$=0,1634

Table 9: Investcost for heat exchanger

Working fluid	Investment cost [$]	Investment cost [€]
R134a	15739	13375
R245fa	14789	12570
Propan	15705	13349
HFO-1234yf	16982	14434
HFO-1234ze-E	15848	13470
Isopentan	14469	12321

2.6.4 Condenser

In order to condense the working fluid after expansion in the turbine a condenser is needed to release the heat from the ORC to the ambient air. The air inlet temperature is set to 19°C, 1,1013 bar and 64% humidity.

The calculation of the condenser heat transfer area and PEC is carried out in the same way as for the heat exchanger in chapter 2.6.3. The heat transfer coefficient is set to U=168 W/m²K.

Table 10: Heat transfer area of the condenser

Working fluid	Heat transfer area [m2]
R134a	300,56
R245fa	128,04
Propan	178,89
HFO-1234yf	218,30
HFO-1234ze-E	295,62
Isopentan	89,99

The investment costs *P*EC of the condenser can be calculated using a correlation 9 analogous to the heat exchanger from 2.6.3. The constants set to X=A; $K1$=4,3247; $K2$=−0,303 $K3$=0,1634. The following table shows the results.

Table 11: Invest costs of condenser

Working fluid	Invest cost [$]	Invest cost [€]
R134a	37757	32093
R245fa	25763	21898
Propan	29512	25085
HFO-1234yf	32284	27441
HFO-1234ze-E	37153	31580
Isopentan	22689	19285

3. CONCLUSIONS

3.1.Results

The evaluation matrix in Table 12 compares the three currently most common fluids R134a, isopentane and R245fa with the three most promising alternative fluids. The effects on the power plant components examined in the last chapter are considered.

After the individual power plant components have been compared, the costs that would be incurred by the exchange due to the replacement of individual parts that cannot be used with the new fluid are added.

Table 12: Results

Standard Fluids	Effects of the substitution of the conventional Fluids by the replacement fluid	R134a	R245fa	Isopentan
Propan	Change of net power	-3%	+21%	+36%
	Change of pump power	+10%	+5500%	+1312%
	Turbine diameter	-64%	+120%	-77%
	Heat exchanger area	0%	+100%	+143%
	Area of condenser	-50%	+30%	+98%
	Inflammability	Highly flammable	Highly flammable	No change
	Additional investmentcost	36.000€	84.000€	84.000€
HFO-1234yf	Change of net power	+1%	+23%	+40%
	Change of pump power	+73%	+8700%	+2113%
	Turbine diameter	-13%	+230%	-68%

	Heat exchanger area	+54%	+250%	+283%
	Area of condenser	-27%	+70%	+143%
	Inflammability	flammable	flammable	flammable
	Additional investmentcost	37.000€	143.000€	143.000€
HFO-1234ze-E	Change of net power	+1%	+24%	+41%
	Change of pump power	+16%	+5800%	+1379%
	Turbine diameter	+36%	+400%	-50%
	Heat exchanger area	+14%	+160%	+191%
	Area of condenser	-1%	+130%	+228%
	Inflammability	flammable	flammable	flammable
	Additional investmentcost	69.000€	125.000€	125.000€

Green = positive consequence / Yellow = consequence must be further investigated / Red = negative consequence

For a better understanding, an example follows:

If an operator maintains a geothermal system with R134a as a working medium and would like to convert it to a less climate-affecting medium, the following options are available: When converting to propane, a new turbine is required, since partial load operation in such dimensions leads to very poor efficiencies, which means costs of around €36,000. The pump can possibly be reused, which leads to a reduction in safety in dimensioning. The net capacity would be reduced by 3%. Alternatively, HFO-1234yf can be used, where the output increases by 1%, but both the feed pump and the heat exchanger would have to be replaced, which would cost €37,000.

When considering the investment costs, it was left out of the question that a system that is converted to a flammable fluid requires a new approval and associated conversion measures to prevent fire. Additionally the sealing material has to be compatible with the new fluid.

Another cost factor not considered in the matrix is the price of the working fluid itself. This is highest for HFO-1234yf, but this will may change in the coming years. The climate-damaging fluids are becoming more and more expensive as their quantity is artificially reduced by the F-Gases regulation and the alternative fluids should get cheaper as they are produced in larger quantities due to the increased demand.

3.1. Conclusion

At a thermal water temperature of 150°C, which is representative for German geothermal plants, HFO-1234yf, HFO-1234ze-E and propane were the most powerful fluids.

They were then compared with the three standard fluids isopentane, R134a and R245fa, which have so far been most frequently used, with regard to changes in the thermal and working capacities in the power plant components feed pump, turbine, heat exchanger and condenser. For the three alternative fluids, it was examined whether and which power plant components would have to be retained, replaced or extended. A cost analysis provided a benchmark for the order of magnitude of the additional costs.

The hydrofluoroolefin HFO-1234yf is suitable as a replacement fluid for R134a within the framework conditions and necessary simplifications of this work, since the operator would incur relatively low costs during the exchange and the performance would not suffer as a result, but would even increase slightly. Another disadvantage, apart from the cost of the conversion, is the flammability of the new fluid.

The second best replacement is propane, only the turbine has to be replaced. However, the operator must reckon with performance losses of around three percent. Here, too, systems for fire detection and fighting would have to be retrofitted.

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