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Optimizing Binary Cycles Thanks to Radial Inflow Turbines

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ABSTRACT

Historically expansion turbines used for binary cycles were axial type turbines adapted from steam turbine design. They expand high-molecular-weight working fluid like isopentane which condenses at atmospheric pressure. Thus pressure ratio is limited across the turbine if a single stage design is considered.

Process data for binary cycles are ideal for radial inflow turbines: pressure ratios, flows and temperatures ensure an operation very close to the maximum achievable isentropic efficiency. An extensive survey of different working fluids has been conducted: not only to determine what is the best thermal efficiency achievable but also to estimate which fluid is the most suitable for radial turbine operation. Fluids are ranked by a performance factor which enables one to select the fluid giving the best compromise between efficiency and turbine size. Then advantages of operating the binary cycle as close as possible to the critical pressure or above and with lighter organic fluid than usual are explained: It increases the recovered electrical power, whilst decreasing the expander frame size and its price.

In most of the cases there is a large benefit to optimize the binary cycle process data together with the turbine design to offer the best net cycle efficiency. Another benefit in using radial inflow turbine in standard execution with variable inlet nozzles is the ability to smooth seasonal variations inherent to geothermal process. This device can be used to control the flow widely through the expander without wasteful throttling. All the expansion energy in the nozzles and wheel is recovered almost at constant isentropic efficiency throughout the year leading to very important additional income over the plant lifetime.

1. INTRODUCTION

The Organic Rankine Cycle (ORC) is a Clausius Rankine cycle in which an organic fluid is used instead water. This process is widely recognized for its ability to transform low grade heat sources, including waste heat from industrial plants and geothermal brines, into useful and higher value electrical energy. In this latter case: the heat sources are either (1):

-hot water from liquid-producing geothermal wells;

-hot water obtained as the liquid fraction in steam separators of flash plants;

-low-pressure steam, even with a large content of non condensable gases;

-low-pressure steam downstream of traditional counter -pressure steam turbines.

Concerning geothermal power generation, the latest study in terms of applied technology for the capacity and number of units is dated 2008 (2). Considering Binary, Binary Kalina and Ormat Combined Cycle units in a single category, it accounts for 30% of the capacity and 50% of the units, whereas Single Flash, Double Flash and Back Pressure reach 60% of the capacity and 40% of the units.

As already mentioned Organic Rankine cycles use organic fluids instead of water as working fluid. These fluids can be used at temperature as low as 74°C (3). For some organic compounds, high superheating is not necessary as for conventional steam-water Rankine cycles. This results in higher efficiency of the cycle and a reduced cost. The organic working fluid (confined in a closed loop) is vaporised using the heat of the hot source in the evaporator. The organic fluid vapour expands in the turbine and is condensed using cold source (most of the time: air, cooling water or a hybrid system). The condensate is then pumped back to the evaporator, thus closing the thermodynamic cycle.

The challenge is the choice of the organic working fluid and the process conditions. The goal is of course to maximize the thermal efficiency defined by the net power production divided by the total heat available. But it is often forgotten that the utilization of the heat source should be also maximized. Moreover environmental, safety and economical criteria are also required fulfilments for designing and building a modern Power Plant based on ORC.

The published literature on fluid selection for ORC is very rich: a research on publications with the key words “Organic Rankine Cycles” and “Fluids” gives more than 1000 references (4). Unfortunately it is very difficult to find information which takes into account all the in- and-outs of ORC for geothermal power plants based on radial inflow turbines: unrealistic turbine sizes and pressure ratios, too optimistic values of cold sources, limitations on high evaporative pressure, forbidden working fluids…

For all these reasons, our own survey was conducted for realistic conditions of turbine operation. This is the first part of the present paper.

The second part focuses on the limitations of the current ORC technology. It is always surprising that the big number of innovative ORC technologies presented in literature is hardly applied in real life. Most ORC power plants today utilize a single working fluid which evaporates at a constant temperature in normal conditions (subcritical) and condenses at atmospheric pressure.

There is room for ORC efficiency improvement whilst keeping the constrains of a realistic “buildable” power plant. Therefore cycles are proposed using binary mixtures or supercritical pressures to achieve better fit and utilization with the heat source.

2. WORKING FLUIDS SURVEY FOR LOW-TEMPERATURE ORC

2.1 System Description and Modeling

For the survey study, only the configuration without regenerator (or internal heat exchanger) was considered. This additional equipment is a way of increasing the thermal efficiency of the ORC but does not change fundamentally the comparison between different working fluids.

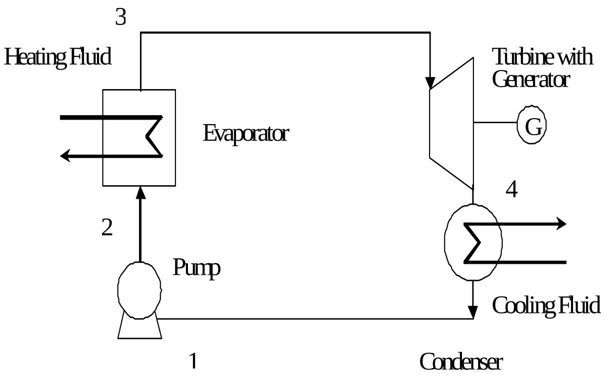

Figure 1: The ORC system used for calculation

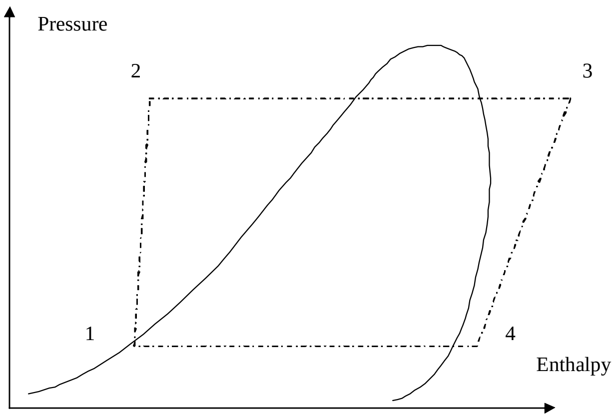

Figure 2: The corresponding pressure – enthalpy diagram

For the cycle performance simulation, it was assumed that the system reached a steady state, and pipe pressure drop and heat losses to the environment in the evaporator, condenser, turbine and pump were neglected. The turbine isentropic efficiency was set to 85% and the pump efficiency to 75%. Only saturated liquid was considered at condenser outlet.

A temperature difference of 5 K was applied between the inlet of the hot source and the inlet of the turbine. This difference was increased to 10 K between the inlet of the cold source and the inlet of the pump to be more realistic (compatible with performances of dry air coolers)

The analysis was performed using Microsoft Excel solver and thermodynamical properties coming from equation of states from REFPROP-NIST (5).

The energy analysis is based on the first law of thermodynamics. The thermal efficiency is the final result. With the assumptions previously stated, it does not depend on the working fluid mass flow rate. The pressure ratio P_3/P_4 across the turbine is fixed to 4. This value is generally achievable by single stage radial inflow turbine. Higher pressure ratio will lead to special design or double stage machines which are less economically viable. All these hypotheses mean that the calculation of the net efficiency depends only on the temperatures of the hot and the cold source : T_{hot} and T_{cold}.

As a matter of fact, for a given working fluid the bubble pressure at the temperature

(T_{Cold} + 10) gives the pressure $P_1 = P_4 = P_{bubble}$.

The pressure at the inlet of the evaporator and turbine inlet is then $P_2 = P_3 = 4.P_1$

Temperature at the inlet of the evaporator is $T_3 = T_{hot} - 5$

The equations for the different components are then the following:

For the pump:

$$h_2 = \frac{(h_{2,is} - h_1)}{\eta_p} + h_1 \quad (1)$$

h_i are the enthalpies at points 1 to 4

$$W_p = \dot{m}_{ORC}.(h_2 - h_1) \quad (2)$$

where W_p is the pump power and m_{ORC} the mass flow of working fluid.

For the turbine:

$$h_3 = \frac{(h_3 - h_4)}{\eta_t} + h_{4,is} \quad (3)$$

$$W_t = \dot{m}_{ORC}.(h_3 - h_4) \quad (4)$$

In all these cycles the heat Q_{23} is added to the working fluid during the evaporation and the heat Q_{41} is removed from it during condensation. The work Wt is taken from the turbine during process expansion, and an amount of work W_p is required to pump the liquid during the process (1–2). Then the thermal efficiency of the cycle is given as:

$$\eta_{th} = \frac{W_t - W_p}{\dot{Q}_{23}} \quad (5)$$

The heat coming from the given heat source to heat and vaporize the working fluid:

$$Q_{hot} = m_{ORC}.(h_3 - h_2) \quad (6)$$

Then the thermal efficiency becomes independent of the ORC working fluid mass flow:

$$\eta_{th} \approx 1 - \frac{(h_4 - h_1)}{(h_3 - h_2)} \quad (7)$$

Beyond the best theoretical thermal efficiency, it is also interesting to have an idea of the feasibility of the ORC.

For the static equipment like evaporator and condenser, changing the working fluids and operating pressures means mainly changing the heat transfer coefficient by varying the heat exchange surface which in practice means more or less units of shell-and-tubes heat exchangers and bays of air condensers. It will have more an impact on the cost than on the feasibility of the ORC.

Axial or radial-inflow turbines can be used for generating power through ORC. Axial turbines are commonly used but the radial type presents a lot of advantages (6) like good "off-design" efficiencies and high-pressure operating range.

Axial turbines are more used for processes involving large flow and low pressure, whereas it is the opposite for radial turbines. Therefore beside the thermal efficiency we have also estimated the radial turbine frame size. This is only achievable if the mass flow of working fluid is known, this can be calculated from equation (4).

The ideal turbine diameter D_{ideal} is then calculated thanks to Cryostar's proprietary formula.

$$D_{ideal} = f\left(\left(h_3 - h_{4,is}\right)V_4\right) \tag{8}$$

Where V_4 is the volumetric flow

<u>2.1.1 Thermal Efficiency</u>

Most of the fluids present in NIST-REFROP database were studied.

The figures 3 and 4 give the values of net thermal efficiency for (respectively) hydrocarbons HC working fluids and hydrofluorocarbons HFC (including hydrochlorofluorocarbons HCFC).

Calculations were made for hot source of 450K and cold source of 293 K.

Both figures show that the maximum net thermal efficiency is in the range 0.12-0.14.

They are big variations of this efficiency for HC (figure 3) whereas HFC exhibit quite constant performances (figure 4). This is attributed to the fact that HC have a range of chemical formulas and thermodynamical properties compared to groups of HFC which are mostly mixtures of components.

These figures can be compared to the Carnot efficiency of 0.275 defined by:

$$\eta_{Carnot} = 1 - \frac{T_{cold}}{T_{hot}} \tag{9}$$

A heuristic would be that at most half of the Carnot efficiency can be reached by an efficient ORC.

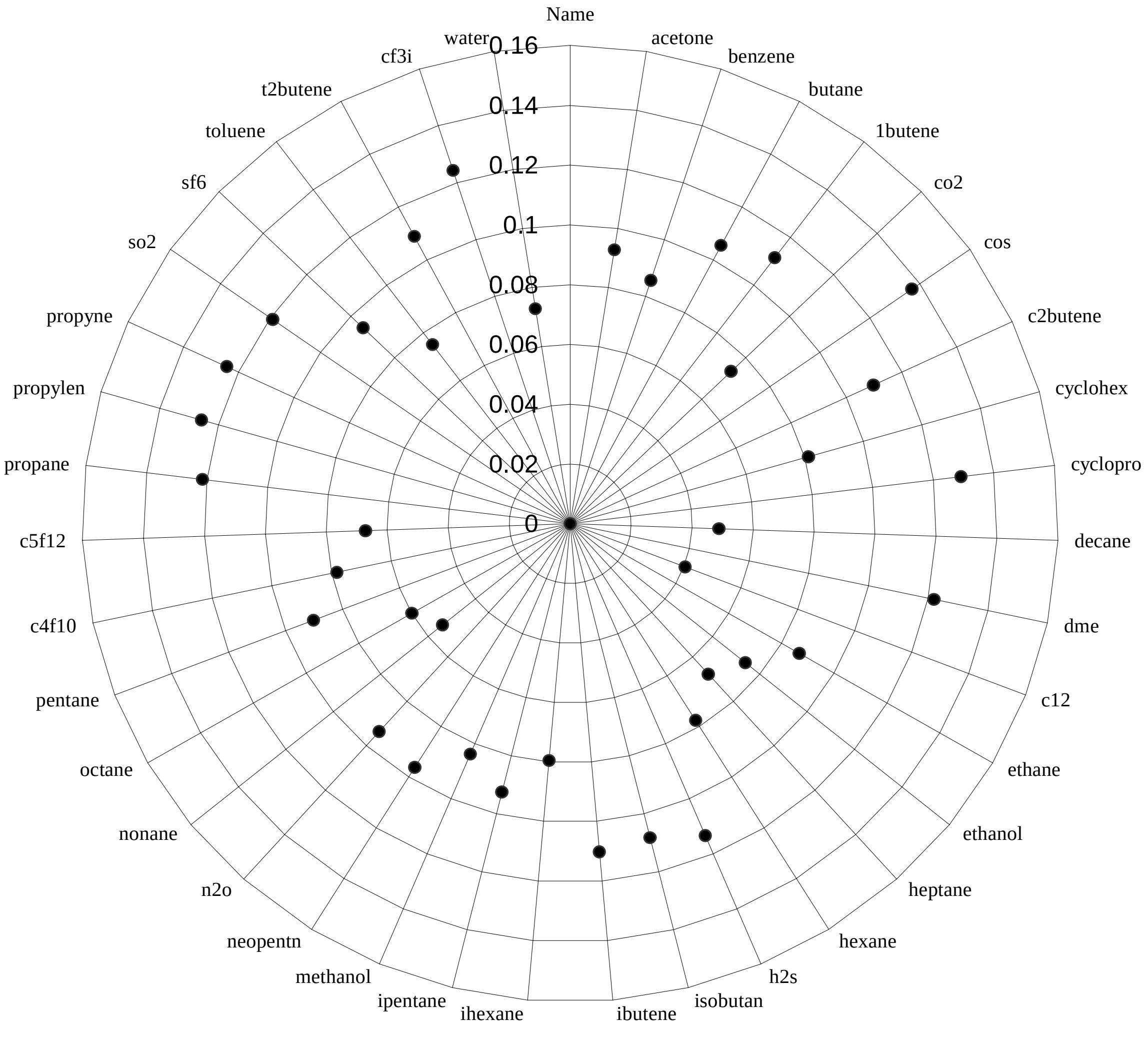

Figure 3: Net efficiency for HC working fluids: T_{hot} = 450 K, T_{cold} = 293 K, P_4/P_1 =4, η_t = 0.85, η_p = 0.75

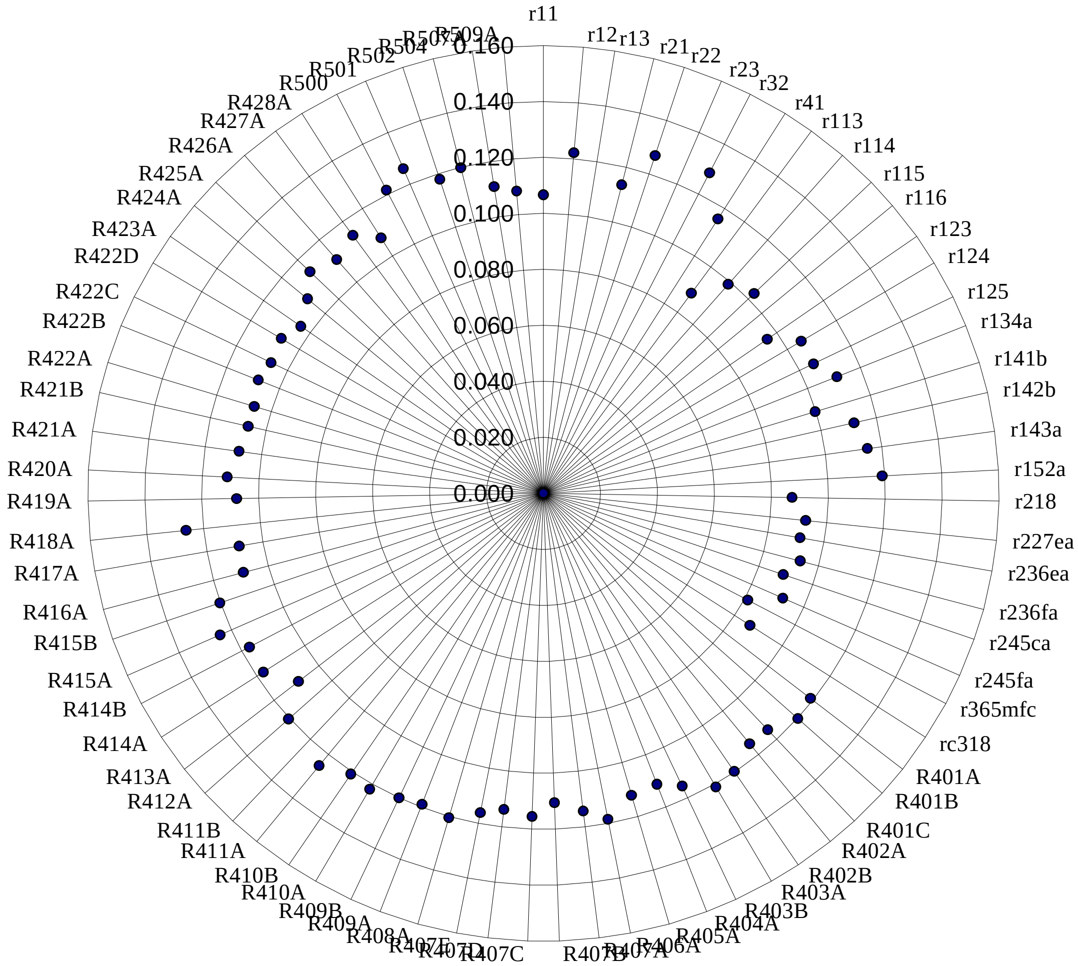

Figure 4: Net efficiency for HFC working fluids: T_{hot} = 450 K, T_{cold} = 293 K, P_4/P_1 =4, η_t = 0.85, η_p = 0.75

2.1.2 Suitable Fluids for Radial Inflow Turbine

All the fluids can not be used.in a real Power Plant Other important parameters are:

a) Working fluids legislation (Ozone depletion potential, global warming potential, toxicity, flammability)

b) Working fluids availability and costing

c) Feasibility of the cycle

d) Frame size of the radial turbine

These additional constraints will limit the numbers of fluids selected to build a real Power Plant.

a) HCFCs are excluded because they are forbidden by the Montreal Protocol, as well as fatal compounds by inhalation.

b) Fluids which can be bought on a tonnage capacity are excluded.

c) If the high pressure of the ORC is above 60 bara, then the fluid is excluded because the execution will need 900 lbs class flanges which is not realistic for Power Plant design. Also if the low pressure is below 1 bara then vacuum execution is not technically simple and may lead to fluid deterioration.

d) To calculate the frame size of the turbine, the power production by the turbine is assumed to be Wt = 5000 kW. Then the flow of working fluid is calculated by equation (3) which leads to the turbine wheel diameter using equation (8).

To select the best fluids showing the best thermal efficiency with the smallest frame size, a “Performance factor” PF was arbitrarily defined. This factor is proportional to the thermal efficiency and inversely proportional to the square of the wheel diameter.

$$PF = \eta_{th} / D^2 \qquad (10)$$

With D wheel diameter in m.

Table 1 is a summary of the results. The columns show the nature of the cycle : supercritical means that the high pressure is above the critical pressure (subcritical below). Cp/Cv shows how efficient the expansion work across the turbine is. The higher this value, the more work can be created by the expansion. Wheel diameter and thermal efficiency are self-explanatory parameters. The latest column is showing the so-called Performance Factor.

It is worth noting that for 5000 kW gross power production, the range of turbine diameter is between 400 and 600 mm for the most efficient working fluids. This frame size is standard for radial turbine manufacturers.

Table 1: Working fluids performances, T_{hot} = 423 K, T_{cold} = 293 K, P_4/P_1 =4, η_t = 0.85, η_p = 0.75

Fluid	Cycle type	Cp/Cv [-]	ηth [-]	D [m]	PF [m^{-2}]
8,5	supercritical	1.427	0.126	0.410	0.748
1butene	subcritical	1.144	0.111	0.405	0.680
propane	supercritical	1.331	0.121	0.450	0.600
butane	subcritical	1.118	0.105	0.450	0.521
r143a	supercritical	1.420	0.115	0.470	0.520
R407A	supercritical	1.412	0.114	0.470	0.518
R407C	supercritical	1.408	0.116	0.475	0.512
R418A	subcritical	1.448	0.126	0.500	0.505
R407E	supercritical	1.404	0.116	0.480	0.504
R407B	supercritical	1.394	0.111	0.475	0.490
R507A	supercritical	1.389	0.111	0.480	0.481
c2butene	subcritical	1.132	0.110	0.480	0.476
R427A	supercritical	1.386	0.114	0.490	0.474
R404A	supercritical	1.387	0.111	0.485	0.473
propyne	subcritical	1.248	0.124	0.535	0.434
R422A	supercritical	1.357	0.106	0.500	0.425
R407D	supercritical	1.350	0.114	0.520	0.421
R421B	supercritical	1.353	0.106	0.505	0.417
R422C	supercritical	1.354	0.106	0.505	0.417
R425A	subcritical	1.342	0.114	0.530	0.406
R422D	supercritical	1.336	0.107	0.520	0.397
dme	subcritical	1.247	0.122	0.555	0.396
R419A	supercritical	1.340	0.108	0.525	0.391
R422B	supercritical	1.323	0.108	0.535	0.377
R401B	subcritical	1.343	0.120	0.570	0.370
R421A	supercritical	1.321	0.108	0.540	0.370
R424A	supercritical	1.313	0.108	0.545	0.364
R417A	supercritical	1.308	0.108	0.550	0.359
R401A	subcritical	1.322	0.119	0.590	0.342
R500	subcritical	1.328	0.122	0.600	0.338
r152a	subcritical	1.261	0.119	0.605	0.325
R413A	subcritical	1.260	0.109	0.605	0.298
R401C	subcritical	1.270	0.116	0.645	0.278
r134a	subcritical	1.232	0.111	0.640	0.271
R420A	subcritical	1.227	0.111	0.655	0.259
r115	supercritical	1.280	0.103	0.635	0.255
so2	subcritical	1.347	0.119	0.705	0.240
R416A	subcritical	1.214	0.109	0.690	0.229
R423A	subcritical	1.197	0.104	0.705	0.209
isobutane	subcritical	1.140	0.108	0.755	0.190
r218	supercritical	1.219	0.087	0.680	0.189
isobutene	subcritical	1.142	0.110	0.795	0.174
cf3i	subcritical	1.289	0.124	0.870	0.164
r142b	subcritical	1.166	0.112	0.870	0.148
t2butene	subcritical	1.131	0.109	0.895	0.136
r124	subcritical	1.155	0.106	0.890	0.133
r227ea	subcritical	1.137	0.093	0.870	0.122
r236fa	subcritical	1.109	0.093	1.070	0.082
neopentane	subcritical	1.085	0.096	1.110	0.078
rc318	subcritical	1.098	0.086	1.080	0.074
r114	subcritical	1.108	0.099	1.235	0.065
r236ea	subcritical	1.094	0.092	1.210	0.063
c4f10	subcritical	1.082	0.078	1.215	0.053
r245fa	subcritical	1.091	0.092	1.365	0.049
isopentane	subcritical	1.073	0.093	1.485	0.042
r245ca	subcritical	1.080	0.089	1.640	0.033
r123	subcritical	1.096	0.096	1.785	0.030
c5f12	subcritical	1.044	0.067	2.180	0.014

A sensitivity study was performed by changing the temperature of the hot source by ± 20 K. Performance factor was calculated for the first ten components (see table 2).

Table 2: Performance factor for T_{hot} = 403,423 & 443 K, T_{cold} = 293 K, P_4/P_1 =4, η_t = 0.85, η_p = 0.75

Component	T_{hot} = 403 K	T_{hot} = 423 K	T_{hot} = 443 K
propylene	0.652	0.748	0.820
1butene	0.667	0.680	0.690
propane	0.539	0.600	0.630
R403A	0.451	0.525	0.574
butane	0.517	0.521	0.524
r143a	0.449	0.520	0.578
R402B	0.456	0.520	0.580
R407A	0.446	0.518	0.564
R408A	0.444	0.513	0.557
R407C	0.442	0.512	0.558

This shows that the ranking of the fluid is only slightly affected by the temperature of the hot source. Of course, the higher this temperature, the better the performance factor.

Compromise between availability of the working fluid, its price and its performances lead to the choice of propane for the next step of the study.

3. OPTIMISATION OF ORC

3.1. Limitation of ORCs

One important limitation of the ORC is the constant temperature of evaporation. Conventional ORC are sometimes not suitable to recover energy from heat sources with large temperature differences.

This is shown in figure 5. To avoid any cross of the heat curves of both hot and cold sources (negative pinch point), the outlet temperature of the hot water must be increased from $T_{hot,out1}$ to $T_{hot,out2}$. The ultimate goal in the design of a power plant is not reaching the highest thermal efficiency of the ORC process but rather a maximum output for a given heat source. It is then obvious that the $T_{hot,out}$ should be kept as low as possible.

One must find process data which allows reasonable pinch point (between 3-10°C) and lowest outlet temperature of the heat source. Such cycles use binary mixtures like Kalina (7) cycles, dual-pressures or supercritical pressures, to achieve variable heat addition to the working fluid for a better fit with the hot source (8).

3.2. Supercritical Orcs: Literature Survey

The present paper focuses one the supercritical cycle because of its high potential and the limited references existing in the literature. In the supercritical organic Rankine cycle the working fluid receives thermal energy at a pressure exceeding its critical pressure but the condensation process operates at a pressure below the critical pressure which is different from the entirely supercritical cycle e.g. with carbon dioxide as proposed in the 60s by Feher (9).

One of the first Geothermal Binary Power Plants built in United States came on line in 1979. The “Magmamax” was located in California’s Imperial valley and used two ORCs with different working fluids : propane and isobutane at supercritical pressure (10). After severe earthquake, the plant was rebuilt using a conventional sub-critical isobutane cycle.

In the 80s, the Heat Cycle Research Program, conducted by the United States Department of Energy tested and operated different supercritical binary cycles using single component and non adjacent hydrocarbon mixtures as working fluids (11). Later it also compared the supercritical cycle with new concepts from that time like Kalina’s water-ammonia working fluid. They concluded with a graph showing that in the temperature range 360-450°F the supercritical cycles gave similar or better results than other advanced cycles (12).

Since the 90’s, to our knowledge, no binary power plant was built using supercritical ORC. With the current boom of geothermal power plants in USA and Germany and for biomass heat recovery some universities are again interested in this field and new articles are published. Gu (13) and Sato studied supercritical cycles for geothermal power generation and concluded that propane and R134a are appropriate working fluids but it is better to operate at pressure higher enough than the critical pressure to improve the fluid stability. Karellas and Schuster (14) used R227ea and R134a at supercritical pressures and showed that they can maximize the efficiency of the system and proposed to apply it in modern applications like thermal desalination or micro CHP. Saleh et al. conducted an exhaustive screening of 31 components for low temperature ORC (15) including interesting pinch analysis of supercritical cycles. For them “the crucial question, which working fluid should be used in an ORC cycle, does not only depend on its thermal efficiency but also on the supply and the further processing of the heat carrier fluid”. Very recently Cayer et al. (16) studied a carbon dioxide supercritical power cycle using an industrial low-grade stream of process gas as heat source. They did not limit their study to the first law of thermodynamics but also performed detailed exergy analysis and calculation of the surface of all heat exchangers.

Supercritical power plants are now a mature technology for water steam generators. It increases on average the overall efficiency from 45% to about 50% which is among the best performing heat engines (15) (16).

3.3. Application of Supercritical ORC

Different configurations of ORCs were compared for the same hypothesis

Geothermal brine : 300 m3/h at 135°C

Condensing temperature = 38°C (which corresponds to air cooler at 25°C ambient temperature with 13°C minimum approach).

Liquid preheater, vaporiser and superheater heat exchange surfaces were estimated for the subcritical cycles as well as the heater for the supercritical cycles. The heat exchange area was kept constant to 3100 m^2 for the three below cases.

A conventional subcritical ORC using isobutane as working fluid which is standard for this level of temperature gives the temperature-heat flow curve depicted in figure 5. It is very difficult to cool down the water below 80°C and keep a reasonable thermal efficiency.

It is possible to modify the "shape" of the evaporation curve by choosing a suitable fluid mixture which does not boil at constant temperature. That is what is shown in figure 6 with a mixture of propane and isobutane for the same process conditions. Then the water can be cooled down to 69°C by keeping a reasonable pinch of the heat exchangers because of the temperature "glide".

As explained in the above paragraph, the best solution when energy from heat sources with large temperature difference needs to be recovered is to use a supercritical cycle. As shown in figure 7, there is a good match between heat flow curves of warm and cold streams, which means that the exergy losses are minimized.

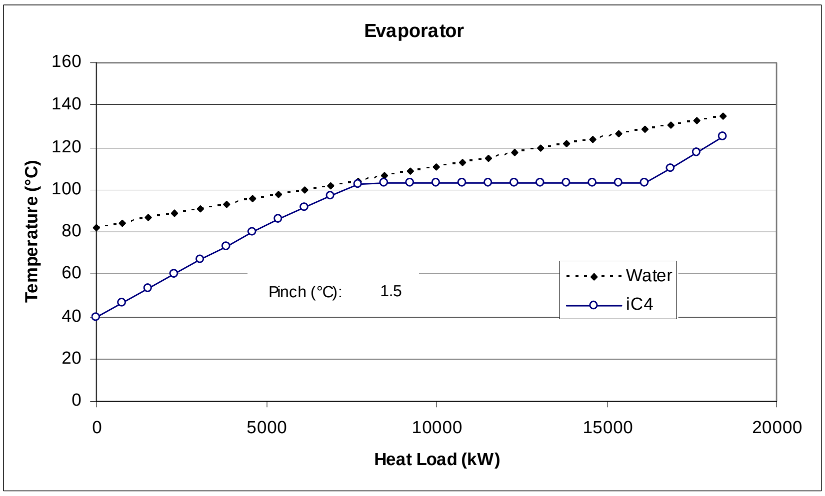

Figure 5: Temperature vs heat flow : Conventional ORC : single component, subcritical

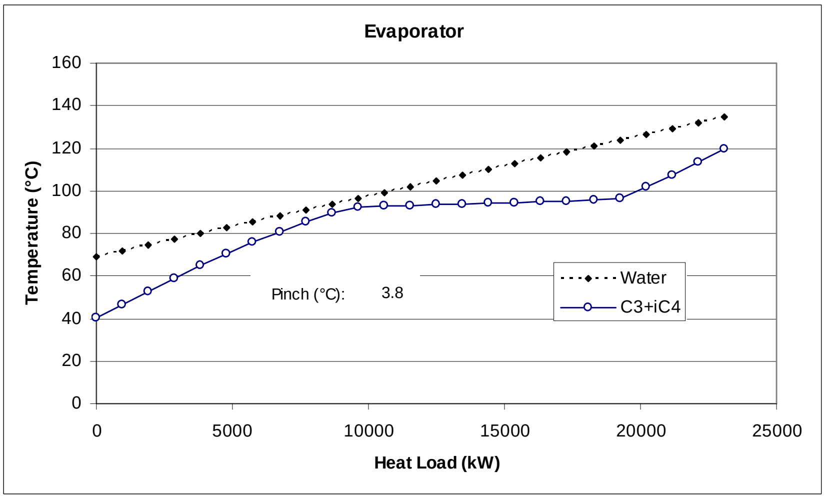

Figure 6: Temperature vs heat flow : Conventional ORC : dual components, subcritical

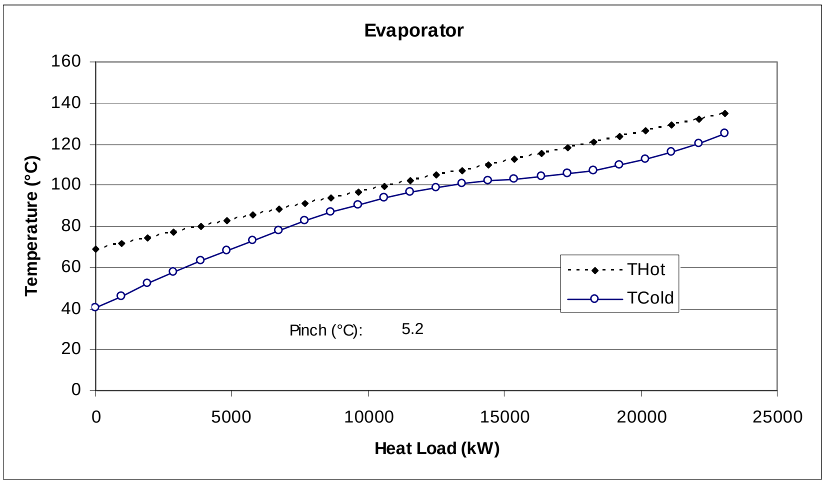

Figure 7: Temperature *vs* heat flow: Supercritical, single component ORC

Performances are summarized in table 3. The supercritical cycle gives the best net thermal efficiency with the smallest turbine wheel. The difference is even bigger for the gross power production. Indeed, with the supercritical cycle, the power needed by the working fluid pump is higher because more head is needed than with a conventional subcritical binary cycle.

Supercritical cycle is not a conventional design for ORC even if it has been widely studied. Nevertheless the process conditions are similar to standard execution for oil and gas projects, e.g. 600 lbs flanges are only needed. Sourcing of equipments of supercritical cycle is nothing special for propane heaters. Air coolers are identical to subcritical cycle ones, only the process pump requires more head than conventional ORC design but nothing really prohibiting.

The benefits of the supercritical cycles are :

- Recovering the largest temperature difference of heat with minimized energy losses.
- Smoother and easier propane evaporation process compared to the conventional design which would need liquid preheater, vaporiser and superheater.
- No risk of ingress of liquid inside the turbine.
- Standard execution for a radial turbine.
- Higher pressure operation reduces the frame size of the turbine.

3.4. Maximizing Yearly Production

Calculations and estimations of total efficiency as described above are only valid for the design point. When studying off-design process data, the performances of the expander must be recalculated. This is often the case for geothermal projects where winter and summer conditions must be taken into account. For such projects, most of the time, air cooled condensers are used, because there is very often not enough cooling water available on site. Therefore in summer, the temperature is high in the northern hemisphere, leading thus to a high condensing pressure then to a low pressure ratio over the expander and finally to a lower electrical output than in winter. There are different means to smooth the seasonal variations by using variable speed process pump on the binary cycle as well as variable speed air-coolers.

Nevertheless it is almost impossible to keep constant process conditions, mainly pressure ratio and mass flow, through the expander throughout the year. In this case there is a big benefit in using radial inflow turbines fitted as standard with variable inlet nozzles. In fact, this device can be used to control the flow widely through the expander without wasteful throttling. All the expansion energy in the nozzles and wheel is recovered almost at constant isentropic efficiency throughout the year. The goal is then to find process conditions (mainly working fluid flow and turbine inlet pressure) which match with the maximum off-design efficiencies of the turbine and the working fluid pump, the available exchange surface of the heat exchangers…This is not an easy task and should not be forgotten because the yearly revenue of a Power Plant does not come only from the design point of the ORC but from all the other operational points.

One good mean to check the off-design characteristics of a cycle is to calculate the second- law-efficiency for the off design points and to compare it with the second-law-efficiency of the design points. This efficiency is defined by the net efficiency of the cycle divided by the Carnot efficiency.

$$\eta_{II} = \eta_{th} / \eta_{Carnot} = \frac{W_t - W_p}{\dot{Q}_{23} \cdot \left(1 - \frac{T_{cold}}{T_{hot}}\right)} \tag{11}$$

The supercritical propane cycle as defined in § 3.3 (300 m3/h of geothermal brine at 135°C) was used. Complete off design calculations thanks to in-house tools were performed by taking into account:

- Available heat exchange surface of heat exchangers and air coolers;
- Off design efficiency of the radial-turbine;
- Off design performance of the working fluid pump as given by suppliers.
- The design air temperature considered was 25°C, additional points for 4°C and 35°C were chosen to represent summer and winter operations

The second law efficiencies for winter and summer off-design cases are in the same order of magnitude as the design case which is optimized showing that the process conditions are correctly defined, even if there is still room of improvement for the summer case.

Maximizing a yearly production means then keeping a constant second law efficiency equal to the design one throughout the whole year.

Table 3: Performance comparison for different ORC configurations

Component	ORC Type	Turbine production [kWelec]	Net production [kWelec]	D [m]
iC4	Subcritical	1957	1779	0.460
C3+iC4	Subcritical	2229	1911	0.410
C3	Supercritical	2869	2253	0.360

Table 4: Performance comparison for off-design cases

Case	Ambient temperature [°C]	Turbine production [kWelec]	Heat recovered [kWtherm]	Second law efficiency [-]
Design	25	2847	23084	36.3
Summer	35	2417	21325	33.1
Winter	4	3631	25658	37.3

4. CONCLUSIONS

A survey of all components suitable as ORC working fluids present in NIST-REFPROP database was conducted. The goal of the survey was to select fluids which can be applied for the design of a real power plant and not only for a theoretical literature study. Therefore some additional constraints were added like inlet pressure of the turbine below 60 bara, limited pressure ratio across the turbine, no vacuum at the outlet of the turbine. HCFC and lethal fluids were also excluded.

An original "performance factor" describing the compromise between the net efficiency of the cycle versus the turbine size was defined. Usable working fluids were ranked using this performance factor. Compromise between availability of the working fluid, its price and its performances led to the choice of propane.

Supercritical cycle were extensively studied and even built in the beginning of the 80s proving its good efficiency over other process. Curiously since that time, no geothermal power plants based on this principle were built. It was shown that, compared to standard isobutane subcritical cycle, a supercritical cycle with propane can produce gross 32% more electricity. Furthermore it is the best way to recover the largest temperature difference of heat with a single turbine. Supercritical cycle is a standard execution for radial-inflow turbines and the main equipments can be found from various recognised suppliers.

Off design operation should not be forgotten. The second law thermal efficiency was considered to check the optimisation of the off-design process conditions. It was proposed that maximizing a yearly production means keeping constant second law efficiency equal to the design one throughout the whole year.

Theoretical optimization of the yearly production was never studied to our knowledge; this is probably a way of future investigation.

5. REFERENCES

Bliem CJ, GL Mines, "Advanced binary geothermal power plants – limits of performance", 1990, Report N° EGG-M-90156.

Bliem CJ, OJ Demuth, GL Mines, WD Swank, "Vaporization at supercritical pressures and counterflow condensing of pure and mixed hydrocarbon working fluids for geothermal power plants", 1986, Report N°EGG-M-09186.

Borgert J.A. and J.A.Velásquez, "Exergoeconomic optimisation of a Kalina cycle for power generation", 2004, *Int. J. Exergy, Vol. 1, No. 1.*

Cayer E., N.Galanis, M.Desqilets, H.Nesreddine P.Roy, 2009, "Analysis of a carbon dioxide transcritical power cycle using a low temperature source", 2009, *Applied Energy* 86, pp 1055-1063.

Cayer P, Galanis N, Desilets M., Nesreddine K, Roy P, "Analysis of a carbon dioxide transcritical power cycle using a low temperature source", 2009, *Applied Energy* 86, pp 1055-1063.

Cogswell F.J., "An ORC Power Plant Operating on a Low-Temperature (165°F) Geothermal Source", 2006 GRC *Transactions*, Vol. 30.

DiPippo R., "Geothermal Power Plants", 2005, Elsevier, pp 383-390.

Feher EG, "The supercritical thermodynamic power cycle", Energy Conversion, 1968, 8, pp 85-90.

Gu Z, H.Sato, "Performances of supercritical cycles for geothermal binary design", 2002, *Energy and Conversion Management*, 43, pp 961-971.

IGA News, April - June 2008 .

Karellas S., A. Schuster, "Supercritical Fluid Parameters in Organic Rankine Cycle Applications", 2008, *Int. J. of Thermodynamics*, Vol 11, pp 101-108.

Lemmon EW, Mclinden MO, Huber ML. "Reference fluid thermodynamic and transport properties (REFPROP)". NIST Standard Reference Database 23, Version 8.

Marcuccilli F., "Advantages of Using Radial Inflow Turbines for Kalina and Organic Rankine Cycles", 2008, Geothermal Resources Transactions, Vol. 32.

Marechal F., B. Kalitventzeff, "A Methodology for the Optimal Insertion of Organic Rankine Cycles in Industrial Processes", 2nd International Symposium on Process Integration Halifax, Canada.

Pioro IL, RB. Duffey, "Heat transfer and hydraulic resistance at supercritical pressures in power engineering applications", 2007, ASME Publications, pp 17-21.

Saleh B., G. Koglbauer, M. Wendland, J. Fischer, 2005, "Working fluids for low-temperature organic Rankine cycles", 2005, *Energy* 32, pp 1210-1221.

http://scholar.google.com keywords "organic Rankine cycles" & "fluids"