Power Cycles Using ORC Technology:
A Comparative Analysis wrt Conventional WRC

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Introduction

In the last years, the production of electricity from renewable energies as well as from waste heat sources has been led by Organic Rankine Cycle (ORC) technology that uses an organic working fluid instead of water. The ORC is not a new concept and many investigations have been carried out [1], [2], [4]. The selection of the working fluid is critical to achieve high-thermal efficiencies as well as optimum utilization of the available heat source. Also, the organic working fluid must be carefully selected based on safety and technical feasibility. Our aim here, is to concentrate on the production of electricity from a bottoming organic Rankine cycle which recovers the thermal power of the exhaust gases of a micro-gas turbine typically available in the range of 250–300 C.

Objectives

1. Perform a comparative analysis among different fluids in Organic Rankine Cycles (ORCs) as well as between ORCs and Water Rankine Cycle (WRC) technologies.
2. Establish the best fluid choice for micro-cogenerative power plant fed by the output gases of a micro-gas turbine at a temperature of 300 C where the **net electrical power level requested is 30kWe**.
3. Demonstrate the advantages of ORC technology wrt WRC in terms of heat recovery efficiency and turbine feasibility when low enthalpy heat sources are available.

Methods

1. We first chose three different organic fluids: N-Pentane, Cyclohexane and Toluene, as well as water vapor for comparison purposes.

These fluids have different critical temperatures and were classified considering their molecular complexity which is a function of the heat capacity of the vapor and, as a consequence, is directly related to the molecular structure of the fluid [2], [6].

	N-Pentane	Cyclohexane	Toluene	Water
Molecular complexity	6,95	9,2	9,32	-10,34
Molar Mass [kg/kmol]	72.149	84.161	92.138	18,015
Critical Temperature [°C]	196,55	280,49	318,6	373,95
Critical Pressure [MPa]	3,370	4,075	4,1263	22,064

Tab.1.Parameters of the used fluids

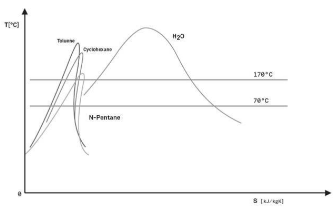


Fig.1 Representation on a T-S diagram of the used fluids

2. Next, we established [2] the evaporation and condensation temperatures, respectively 170°C and 70°C, and the corresponding pressure values were [cfr. Tab.2]:

Then we performed an evaluation of the cycle thermal efficiency and, at this purpose, in accordance with, literature [4] we fixed different values of isentropic turbine efficiencies for water and organic fluids, respectively 0,35 and 0,7. Finally we investigated the heat exchange process with the aim of assessing the total recovery efficiency of the system.
All values of thermodynamic quantities of fluids were calculated using EES [3].

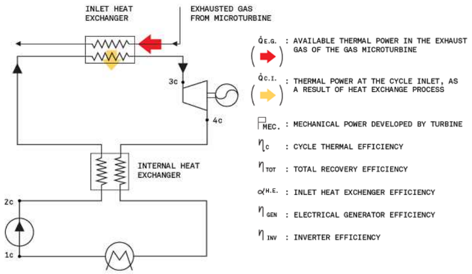
T_{Evap} = 170°C T_{Cond} = 70°C	N-Pentane	Cyclohexane	Toluene	Water
P_{Evap} [kPa]	2237	808,2	423,6	792,2
P_{Cond} [kPa]	283,2	72,5	27,35	31,2
$(P_{Evap}) / (P_{Cond})$	7,899	11,15	15,488	25,39

Tab.2.Pressure values for evaporation and condensation temperatures

3. In the second part of the work, for each fluid, a preliminary sizing of a single-stage inflow radial turbine was carried out using fundamental turbine design criteria [7] based on the following assumptions [cfr. Fig.7]:
- At the rotor inlet the relative flow is radial
- At the rotor outlet the absolute flow is axial

We calculated the main dimensions of the rotor, its rotational speed and its functional parameters as a function of the width of the rotor inlet blade, starting from an initial value of 2 mm and then gradually increasing it. Great attention has been also given to the behavior of the flow field at the exit of the turbine nozzle.

Analysis



$$\eta_C = \frac{P_{MEC.}}{\dot{Q}_{C.I.}} \quad \text{(Without internal heat exchanger)}$$

$$\eta_{TOT} = \frac{P_{MEC.}}{\dot{Q}_{E.G.}}$$

$$P_{MEC.} = \dot{Q}_{C.I.} \cdot \eta_C = \dot{Q}_{E.G.}\left(\alpha_{H.E.} \cdot \eta_C\right) = \eta_{TOT} \cdot \dot{Q}_{E.G.}$$

$$P_{EL} = P_{MEC.} \cdot \eta_{GEN} \cdot \eta_{INV}$$

Fig. 2. Schematic view of the system

<u>The total recovery efficiency depends on two parameters:</u>
<u>Efficiency of Inlet Heat Exchanger and Cycle Thermal efficiency (which depends also on the turbine efficiency)</u>

MAIN DIFFERENCES BETWEEN ORGANIC FLUIDS AND WATER

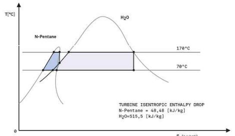


Fig. 3. Turbine isentropic enthalpy drop

- Values of Specific Isentropic Enthalpy drops in turbine [cfr. fig.3]
- Slope of a saturated vapor line [cfr. Fig.4]

Starting from saturated conditions [1] of the fluid, and in accordance with literature [4] turbine efficiency for single stage inflow radial turbine was 0,35 for Water and 0,7 for Organic Fluids [cfr. Tab.3].

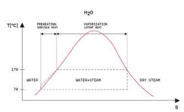


Fig. 4.a. Latent and sensible heat of water .

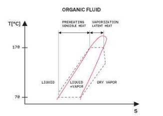


Fig. 4.b. Latent and sensible heat of organic fluids.

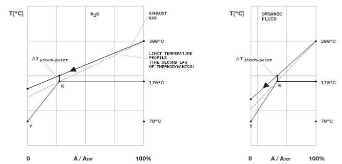


Fig. 5. Temperature profiles into primary heat exchanger.

- Organic fluids have a lower value of the ratio $(Latent/Total)_{heat}$ absorbed [cfr.Fig.4]

1. Higher ratio $(Latent/Total)_{heat}$ gives higher values of cycle thermal efficiency [cfr. Tab.3]
2. Into the primary heat exchanger higher values of $(Latent/Total)_{heat}$ implies lower capability to cool the external thermal source [cfr. Fig.5]

TURBINE

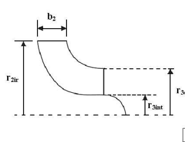


Fig. 6. Schematic view of the rotor.

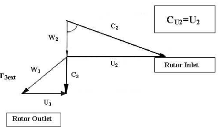


Fig. 7. Triangles of velocity at the rotor.

As a consequence of the assumptions made on the triangles of velocity:

$$\Delta h = U_2 C_{U2} - U_3 C_{U3} = U_2 C_{U2} = C_{U2}^2 \rightarrow C_{U2} = \sqrt{\Delta h}$$

The main dimension of the turbine and its rotational speed were calculated as follows:

$$r_{2ir} = (\dot{m} * v_{spec}) / 2\pi * b_2 * W_2 \qquad \omega = U_2 / r_{2ir} \ \text{[rad/s]}$$

Results

CYCLE THERMAL EFFICIENCY

Organic fluids have greater thermal efficiency than water.
Among organic fluids those with higher molecular complexity have greater cycle efficiency.
It is also important to note the value of exit temperature of turbine [cfr. Tab.3]

T_{Evap}=170°C T_{Cond}=70°C	N-Pentane	Cyclohexane	Toluene	Water
Molecular complexity	6,95	9,2	9,32	-10,34
$(Latent/Total)_{heat}$	0,3788	0,5432	0,6087	0,8281
Cycle efficiency for isentropic expansion in turbine	0,1596	0,1744	0,1805	0,208
Turbine efficiency	0,7	0,7	0,7	0,35
Cycle efficiency (without internal heat exchanger)	0,1096	0,1215	0,1251	0,07261
Temperature at the turbine exit [°C]	114,4	123,8	118,3	70

Tab. 3. Cycle efficiency of the used fluids.

TOTAL RECOVERY EFFICIENCY

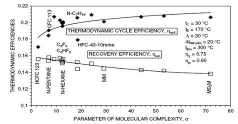


Fig. 8. Cycle efficiency and total recovery efficiency as a function of molecular complexity.

As clearly explained in [2] an increase in $(Latent/Total)_{heat}$ gives an increase on cycle efficiency which is not able to balance the decrease of inlet heat exchanger efficiency.

In this way organic fluids that lead to a higher total recovery efficiency of the system are those with lower molecular complexity [cfr.Fig.8].
This explains why, for variable temperature heat sources, it is necessary to use organic fluids instead of water to achieve higher thermodynamic performance .

TURBINE

P=30 kWe	N-Pentane	Cyclohexane	Toluene	Water
Turbine Efficiency	0,7	0,7	0,7	0,35
b_2 [m]	0,002	0,002	0,002	0,002
r_{2ir} [m]	0,1029	0,227	0,406	0,1516
n [rpm]	21740	10603	6095	26763
Nozzle Exit				
Mach number of absolute flow	1,271	1,3	1,358	0,8733

Tab. 4. Main parameters of turbines having b_2=2mm

We began our analysis assuming a value of b_2=2 mm
Water:
Single-stage solution for the rotation speed would still be acceptable but the system it self would not be competitive in terms of thermal efficiency of the cycle.

It is worth noting that for every Organic Fluid the flow field at the exit of the nozzle is supersonic.

INCREASING THE WIDTH OF ROTOR INLET BLADE

The solution for N-Pentane with, for example, b_2= 3.5 mm leads to an already over-rotating system. In conclusion, the solutions that lead to an overall satisfactory machine are [cfr. Tab.5].

	b2[mm]	n[rpm]	r2ir [cm]
Cyclohexane	5	26.000	9.08
Toluene	5.5	16.760	14.76

Tab. 5. parameters of overall satisfactory machines

Discussion

For heat sources, as output gases of micro-gas turbine, whose temperature could significantly decrease during a heat exchange process, the use of organic fluids instead of water leads to higher values of total recovery efficiency and feasibility of single stage inflow radial turbine. Among organic fluids, those with lower values of molecular complexity give higher values of total recovery efficiency. In this way N-Pentane was the best fluid considered <u>but, for a power level of 30kWe</u>, it highlights problems not solvable in terms of feasibility of the turbine and should therefore be discarded. Regarding the turbine, according to the efficiency values found in literature, it appears impossible to use single-stage turbines where the fluid is water. This because the flow through the turbine is greatly disturbed due to a typical two-phase expansion field. In contrast, using organic fluids, a single stage inflow radial turbine could be realized. These turbines have, however, in all cases, a high-supersonic flow at the exit of the nozzle, which makes its design difficult. It would seem reasonable to think of two stages turbine to prevent this. In the end, it seems worth noting that though the design of a highly supersonic nozzle would be achieved, it would imply dissipative phenomena associated with the development of shock waves and expansion waves of Prandtl-Meyer.

References

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