LOW GRADE WASTE HEAT RECOVERY IN AN INTERCOOLED RECUPERATED CLOSED CYCLE GAS TURBINE

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ABSTRACT

The context of the work is the study of a 180 bar nitrogen recuperated and intercooled turbo-compressor of 300 MWe. The net efficiency of this closed cycle gas turbine is assessed between 37 and 37.5%, but which could be increased by considering waste heat recovery techniques.

Among waste heat recovery technologies which are currently used in conventional gas turbines we find organic Rankine cycles (ORC). This kind of machine is similar to a conventional steam cycle energy conversion system, but uses an organic fluid instead of water. A main challenge is the low level of temperature of waste (90°C).

Preliminary studies are carried out on the design of ORC using three different fluids: traditional and mature R245fa technology, innovative R125 cycle and high pressure low temperature transcritical CO_2 machine. Different architectures are compared and advantages of using transcritical cycles have been underlined. From a thermodynamic point of view the best solution would be to use R125 cycle at a supercritical pressure at precooler secondary side and at a subcritical pressure at intercooler secondary side. A supplementary power of more than 15 MWe could be achieved. The resulting combined cycle could reach a net efficiency of 39%.

INTRODUCTION

The project concerns the study of a heat-to-power cycle at a maximal temperature of 515°C generating 600 MWe. The specificity of the present work is to consider an alternative cycle to traditional Rankine machine in order to avoid high pressure steam. From a theoretical point of view numerous heat-to-power technologies can be proposed. Among them, gas turbine based cycles or Stirling engines seem the most interesting processes at high temperature, depending on the power plant size. A recent review of existing plants for the various technologies has been published (Figure 1).

Figure 1, which can be used at a pre-design phase, shows that the most interesting technology could be of Brayton type, even if no existing product is identified (present project is too large or not sufficiently hot). More deeply in the design process, an innovative gas power conversion system is under investigation. It concerns 180 bar nitrogen turbo-compressor (with operating temperatures between 330°C and 515°C) in two recuperated and intercooled Brayton cycles operating in parallel to lower the unit power. This power conversion system takes advantage of past German experience and studies carried out in the years 2000 for helium closed cycle projects, as well as current studies on conventional open gas turbines. Thus, this type of cycle has been chosen as a reference for preliminary studies, considering that this is a major industrial innovation for a 300 MWe machine.

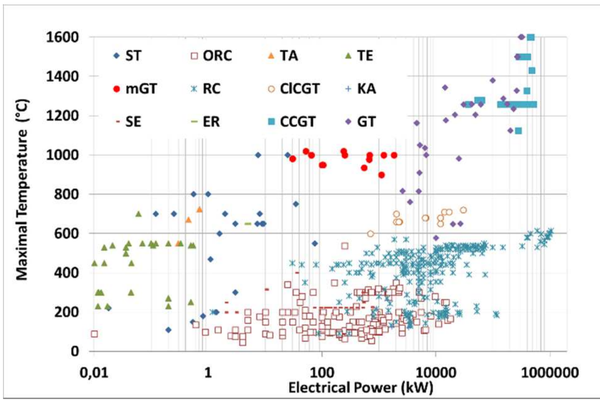

Figure 1 Map of Maximum temperature versus Power technologies [1]

(Corresponding nomenclature: Rankine cycle plant (RC), Steam engine (SE), Organic Rankine Cycle plant (ORC), Kalina cycle plant (KA), Gas turbine plant (GT) and Micro gas turbine (MGT), Closed cycle gas turbine plant (ClCGT), Ericsson engine (ER), Stirling engine (ST), Thermogenerator (TE), Thermoacoustic generator (TA), Combined cycle plant (CCGT))

CONFIGURATION

The gas turbine configuration is presented in Figure 2. Figure 3 shows the thermodynamic diagram and main parameters of the cycle.

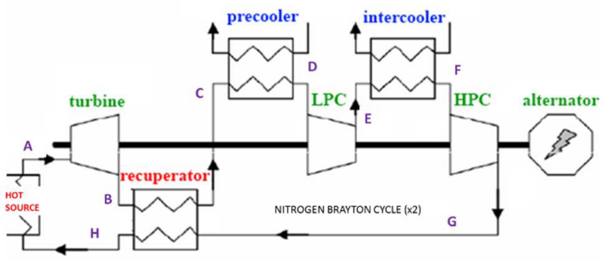

Figure 2 Recuperated and intercooled Brayton cycle configuration

The net efficiency of the power station is assessed between 37 and 37.5%, which remains under the efficiency level of a Rankine cycle (≥40 %), but which could be increased by considering waste heat recovery techniques, as nitrogen reaches a temperature of 94°C and 86°C before being precooled and intercooled (Figure 3). Waste heat recovery analysis selection [2] is based on a similar approach as Figure 1 has resulted in the selection of an Organic Rankine Cycle (ORC) system.

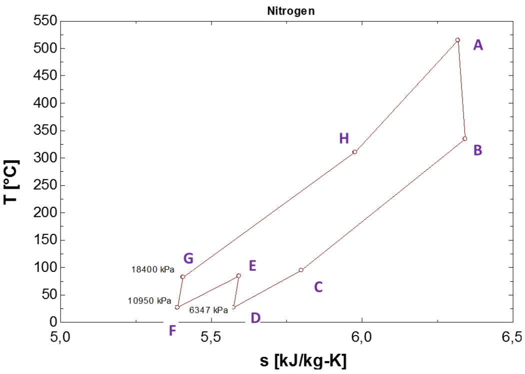

Figure 3 Recuperated and intercooled Brayton cycle T-s diagram

ORGANIC RANKINE CYCLES

An organic Rankine cycle (ORC) machine is similar to a conventional steam cycle energy conversion system, but uses an organic fluid, characterized by higher molecular mass and lower ebullition/critical temperature than water. An Organic Rankine Cycle (ORC) has several advantages over conventional steam power plant:

• The evaporation process can take place at lower pressure and temperature.

• The condensation pressure is often higher than one atmosphere avoiding air intake and reducing the size of the low pressure components (especially last turbine stages)

• Depending on the fluid choice, with dry organic ones, the expansion process ends in the vapor region and hence the superheating is not required and the risk of blades erosion is avoided.

• The smaller temperature difference between evaporation and condensation also means that the pressure drop/ratio will be much smaller and thus simpler turbines can be used.

Thus, the thermodynamic optimization of the cycle is achieved through the choice of the most suitable organic fluid, in regard to the temperature heat source. Potential substances identified for ORC are: Hydrocarbons (HC), Hydrofluorocarbons (HFC), Hydrochlorofluorocarbons (HCFC), Chlorofluorocarbons (CFC), Perfluorocarbons (PFC), Siloxanes, Alcohols, Aldehydes, Ethers, Hydrofluoroethers (HFE), Amines, Fluids mixtures (zeotropic and azeotropic), Inorganic fluids (CO_2, NH_3).

ORC have been used in a wide range of applications, including geothermal, biomass or solar power plants, waste heat recovery from industrial processes or combustion engines, ocean thermal energy conversion and a wide range of power outputs from a few kW to tens of MW [3]. Moreover ORC is investigated in different projects as the bottoming cycle of a combined cycle gas turbine with an opened [4] [5] or closed [6][7] gas circuit configuration.

FLUID SELECTION

The first issue to deal with when working with ORC is to select the appropriate fluid. At this early stage of the project, we can consider four criteria: thermodynamic performance, environmental impact, technological maturity and economic performance. In this paper, we will not expose economic results and we mainly concentrate on thermodynamic performance.

For environmental considerations, we generally consider toxicity, flammability, ozone depletion potential, and global warming potential. The last parameters are defined with a quantitative criterion (ODP, GWP).

At a first stage of a project, it is convenient to use a simple approach to make a first selection of the working fluid. We have used a simple methodology based on the fluid critical temperature T_{crit} developed in [8]:

$$(T_{crit})_{opt} = (T_{cycle})_{max} - 33\text{ K}$$

Other studies report optimal critical temperature values as a function of hot source temperature:

$$(T_{hot\ source}) - (T_{crit})_{opt} = [35\text{ K};55\text{ K}] \text{ such as in [9]}$$

or

$$(T_{crit})_{opt} / (T_{hot\ source}) = [0.8\text{-}0.9] \text{ (as in [10], [11] or [9]).}$$

Thus a rough estimation of working fluid optimal critical temperature should be in the range of [20°C; 70°C].

Another key point in this evaluation is the fluid type; the wet type (similar as water); the isentropic type, which has an isentropic saturation curve; and the dry type. As suggested before a superheater is theoretically only needed for the wet type fluids. Dry type fluids most often possess a great amount of heat after expansion and thus a recuperator is generally beneficial to the overall cycle efficiency, whereas for wet and isentropic fluids, recuperation is most often not possible because the fluid is already at a saturated state.

Thus we have chosen to consider three different fluids: R245fa, R125 and Carbon dioxide.

R245fa (1,1,1,3,3 pentafluoropropane) is a hydro-fluorocarbon (HFC) nearly non-toxic, non-flammable refrigerant permitted under the Montreal Protocol (ODP=0) and with a moderate global warming potential of 1000 (1000 times the global warming effect of CO_2), with a boiling point at atmospheric pressure of 15 °C. This dry fluid is a current fluid used in subcritical ORC systems. Its critical temperature is equal to 154 °C.

R125 (pentafluoroethane) is a hydrofluorocarbon (HFC) nearly non-toxic, non-flammable refrigerant permitted under the Montreal Protocol (ODP = 0) and with a high global warming potential of 3500. This wet, but quasi-isentropic fluid has a critical temperature is equal to 66 °C, which could be a very good candidate for a transcritical ORC.

Carbon dioxide (R744 in ASHRAE nomenclature) is a natural refrigerant with many advantages. It is a low-cost fluid that is non-toxic, non-flammable, chemically stable, and readily available. In addition, the high fluid density of supercritical CO_2 leads to very compact systems. Its critical temperature is equal to 31 °C. Many studies have been published to evaluate the potential of supercritical CO_2 as working fluid in power cycles and heat pumps [12] [13]. Cayer carried out an analysis [14] and optimization [15] of transcritical CO_2 cycle with a low-temperature heat source. More recently, the use of CO_2 for multi-megawatt power cycles has reached a commercial step with the American company Echogen [17]. Such systems must be distinguished from complex high-temperature ($(T_{cycle})_{max}$ > 450°C) supercritical CO_2 systems as those described in [18].

PROBLEM DEFINITION

The recuperated Rankine cycle is characterized by 6 main steps: the working fluid leaves the cold heat exchanger as a subcooled liquid at $T_1 = T_{sat}(LP) - \Delta T_{sc}$, (necessary to avoid cavitation in the pump) and is adiabatically compressed (1 → 2) in a feed pump with isentropic efficiency $\eta_{s,p}$. At the outlet of the pump, the fluid at a high pressure P_2 = HP is first preheated through the regenerator (2 → 3), then heated further through the hot heat exchanger (3 → 4) transferring heat from the gas turbine cycle. At the entrance of the turbine, the fluid at a defined temperature T_4 (=$(T_{cycle})_{max}$) is adiabatically expanded (4 → 5) to the subcritical low pressure LP delivering a mechanical work with isentropic efficiency $\eta_{s,t}$. Finally, the fluid is cooled in the regenerator (5 → 6) before being condensed through the cold exchanger (6 → 1). The thermodynamic states can be obtained from the energy balances of each components:

$$\dot{W}_p + \dot{m}(h_1 - h_2) = 0 \quad (1)$$

$$(h_2 - h_3) + (h_5 - h_6) = 0 \quad (2)$$

$$\dot{Q}_{hot} + \dot{m}(h_3 - h_4) = 0 \quad (3)$$

$$\dot{W}_t + \dot{m}(h_4 - h_5) = 0 \quad (4)$$

$$\dot{Q}_{cold} + \dot{m}(h_6 - h_1) = 0 \quad (5)$$

h_i (J/kgK) and $\dot{m}$ (kg/s) being respectively the specific enthalpy at state i and the mass flow rate relating to the discharging cycle. $\dot{W}_p(W) = \dot{m}(h_{2s} - h_1)/\eta_{s,p} > 0$, $\dot{W}_t(W) = \dot{m}(h_{5s} - h_4)\eta_{s,t} < 0$, $\dot{Q}_{hot}(W) > 0$ and $\dot{Q}_{cold}(W) < 0$ are respectively the pump power, the turbine power, the heat flux transferred from the hot source and the heat flux transferred to the cold source.

By adding equations 1 to 5, it appears the energy balance of the discharging cycle:

$$\dot{W}_p + \dot{Q}_{hot} + \dot{W}_t + \dot{Q}_{cold} = 0 \quad (6)$$

The supplementary net power output of the cycle is: $\dot{W}_{el} = \eta_g \dot{W}_t$

Regenerator Efficiency is defined as:

$$\varepsilon_{rec} = \dot{m}(h_3 - h_2)/\dot{Q}_{rec,\max} = \dot{m}(h_5 - h_6)/\dot{Q}_{rec,\max} \quad (7)$$

With $\dot{Q}_{rec,\max} = Max\left[\dot{m}(h_5 - h_6)|T_6 = T_2 ; \dot{m}(h_3 - h_2)|T_3 = T_5\right]$ (8)

When the cycle is non regenerated, we have $\varepsilon_{rec} = 0$

$\dot{Q}_{hot}(W) = \sum_1^N \dot{Q}[i]$ is the real value transmitted through the hot heat exchanger, calculated with a pinch-value method:

$$\dot{Q}[i] = AU[i]\,LMTD[i] = \dot{m}(h[i+1] - h[i]) = \dot{m}_{N2}(h_{ev}[i+1] - h_{ev}[i]) \quad (9)$$

For each elementary segment, the log mean temperature difference is given by:

$$LMTD[i] = \frac{\Delta T[i] - \Delta T[i+1]}{\ln\left(\frac{\Delta T[i]}{\Delta T[i+1]}\right)} \quad (10)$$

with $\Delta T[i] = T_{ev}[i] - T[i]$ (11)

The pinch value is defined as: $Min(\Delta T[i])$.

As a preliminary work, pressure losses in the thermodynamic cycles are neglected. Simulation of the hot heat exchanger system will enable to estimate the head losses in that component and adjust the cycle parameters.

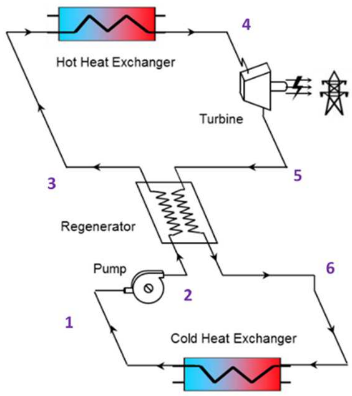

Figure 4 ORC process layout

NUMERICAL MODEL

Presentation

The thermodynamic model is implemented in the Engineering Equation Solver (EES) software [19]. The analysis is performed with EES (engineering equation solver). This software has the advantage of including fluid properties and ready-to-use optimization tools. It uses the same equation of state as REFPROP-NIST [15].

Table 1 System constant settings

Condensing temperature	30°C
Pump isentropic efficiency $\eta_{s,p}$	0.80
Turbine isentropic efficiency $\eta_{s,t}$	0.85

Generator efficiency η_g	0.985
Fluid subcooling at pump inlet ΔT_{sc}	3K
Hot heat exchanger pinch	3K
Regenerator efficiency	0.95

Validation

We have compared our results with numerical data detailed in [20] (and partially published in [14] and [15]). Differences are of 1-2% on main parameters on cases treated, especially due to fluid properties differences. We have also compared cycle analysis to [16].

RESULTS: PERFORMANCE DISCUSSION

Based on the previous modelling, it is possible to carry out a parameter analysis of the system. Used values are written in Table 1. A typical R125 cycle is shown in Figure 5. We clearly find that the working fluid is wet, nearly isentropic. A small superheating is necessary to avoid two-phase expansion. The illustrated cycle is supercritical in some parts of the circuit as pressure and temperature are larger than 36.2 bar and 66°C respectively.

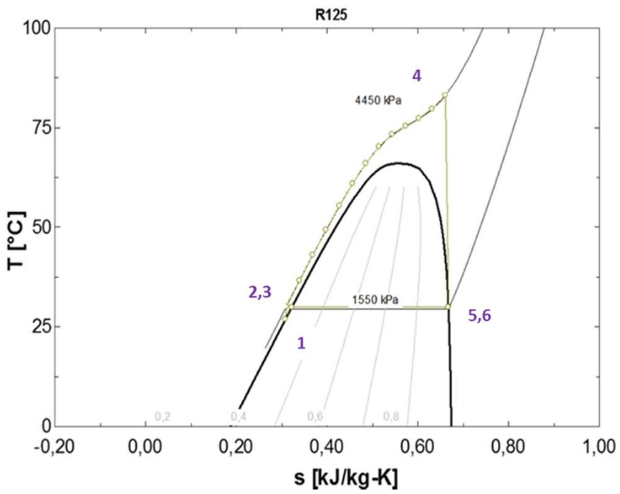

Figure 5 T-s diagram of R125 transcritical ORC

The maximal temperature ($(T_{cycle})_{max}$) is equal to 83°C with a flow rate of 1544 kg/s. At the hot heat exchanger, nitrogen is cooled down to 45°C. The transferred power is 181MWth. An additionnary precooler of 66.5 MWth is therefore needed to cool nitrogen down to 27°C. Working pressures are HP=44 bar and LP=15 bar. The gross power is 18 MWe. It is necessary to consider 4.7 MWe, which are consumed by the pump. The resulting back work ratio (defined as the ratio pump work / turbine work) is equal to 26%. The end of expansion is at 15 bar and 30 °C. As the fluid exit the pump at nearly 30 °C it does not seem necessary to add a regenerator. Thus, net power is around 13 MWe. First law efficiency is equal to 7%. This low value is due to low maximal temperature and pump power. The hot heat exchanger profile is shown in Figure 6. Pinch point location is located at the middle of the heat exchanger. The exergetic destruction is limited, due to transcritical isobaric temperature profile. A preliminary pressure losses estimation indicates that this hypothesis is justified (few tens of kilopascals) and leads to limited generated power decrease in ORC cycle (few hundreds of kilowatts). As far as nitrogen cycle is concerned, the effect of supplementary pressure losses is more important (each additional 10 kPa leads to 800 kWe electrical off). Thus the design deserves a special attention.

The turbine has a power of 18 MWe and an expansion ratio of 2.8. A 4-stage subsonic machine rotating at a frequency of 50 Hz, with a casing smaller than 1.2 m could be designed at a flow coefficient of 0.3 and a work coefficient of 0.9. Smith chart (Figure 7) indicates that such a machine could exist with a high isentropic efficiency of 94%, highly superior to 85%, which was the preliminary assumption. We have chosen to keep original value in a conservative approach.

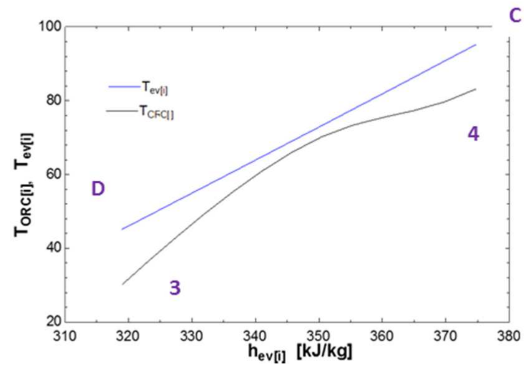

Figure 6 T-h diagram of R125 transcritical ORC hot heat exchanger

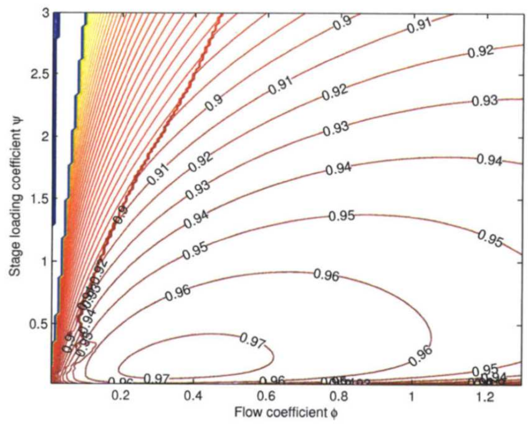

Figure 7 Smith chart for an axial turbine [21]

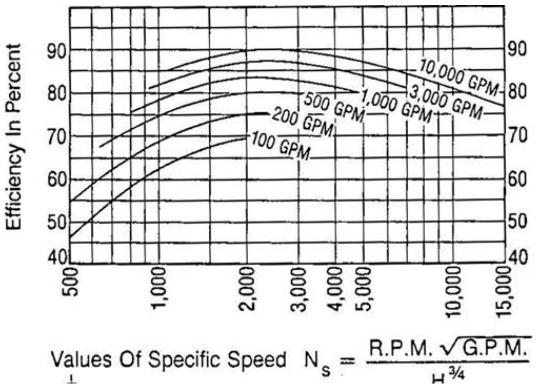

Figure 8 Ns diagram for a rotary pump [22]

The pump has a power of 4.6MW and a manometric head of 250 m. With a motor rotating at 1500 rpm, a specific speed of 1410 can be obtained. With a flow equal to 1000 gpm, an acceptable efficiency and a standard technology can be expected (Figure 8).

A parametrical study has been performed on two parameters: maximal pressure and mass flow. The results can be shown on Figure 9.

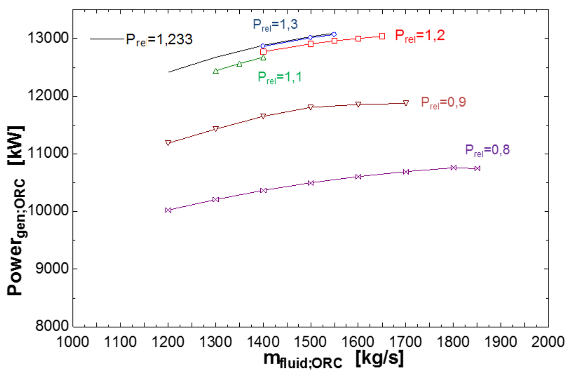

Figure 9 R125 subcritical and transcritical ORC optimization (Power versus fluid flow)

We can see that the net power increases when high pressure increases, up to a maximal value due to a high pump consumption. Concerning mass flow parameter, net production is monotonous when high pressure is imposed. However the higher is the flow rate the lower is the maximal temperature in the cycle and consequently lower is the efficiency.

RESULTS: FLUID COMPARISON

Figure 10 shows the optimal ORC cycle using R245fa as working fluid. The subcritical shape can be clearly seen. A higher exergy destruction can be expected in the hot heat exchanger (compared to transcritical cycle in Figure 5). The net power does not exceed 9.5 MWe. The main interest is the moderate level of high pressure (HP=5.2 bar). Components are expected to be simpler and cheaper.

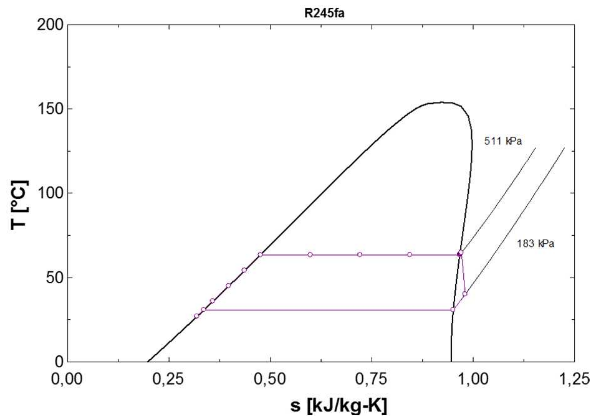

Figure 10 T-s diagram of R245fa subcritical ORC

Figure 11 shows the optimal ORC cycle using CO_2 (R744 in ASHRAE nomenclature) as working fluid. The net power (11MWe) is inferior to R125 cycle, as expected with the correlative approach. Components are expected to be complex and expensive (each pump has a 11 MWe consumption).

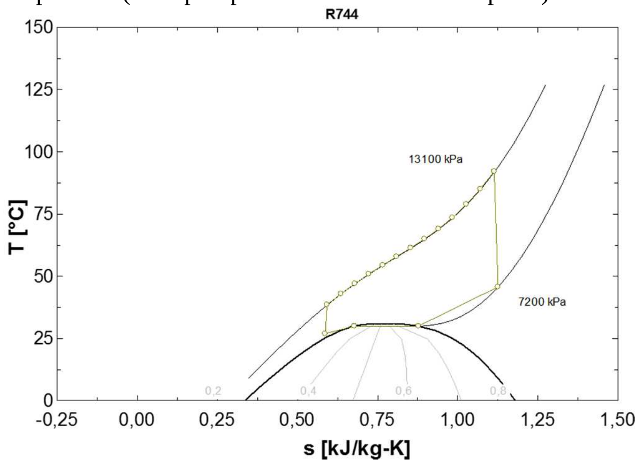

Figure 11 T-s diagram of CO_2 transcritical ORC

RESULTS: ARCHITECTURE DISCUSSION

Based on the previous modelling, it is possible to carry out a parameter analysis of different systems of distinct architecture.

<u>Regenerator</u>

A theoretical possibility to increase the power generated is to use a regenerator in the cycle. Figure 12 shows that the interest is very limited (for R125 cycle), as expected, even if the cycle thermodynamic efficiency increases.

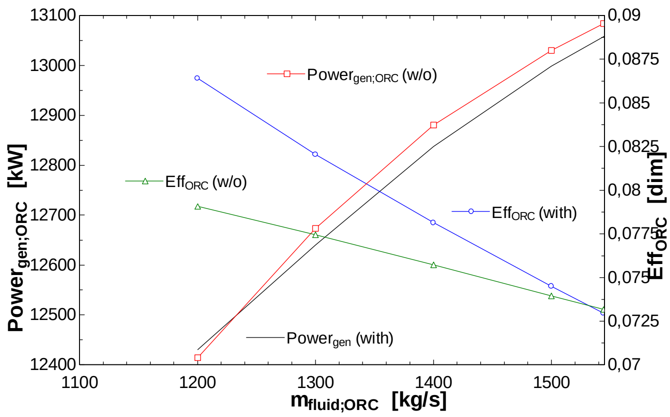

Figure 12 R125 transcritical ORC optimization with or without regenerator (Power and efficiency versus fluid flow)

Intercooler

A complementary way to increase the power generated is to recover waste heat at the intercooler. Figure 13 shows the optimal cycle with R125; a net power of 8.5MWe could be generated. We can note that this cycle is subcritical. The reason is that the intercooler nitrogen temperature is 8°C lower than precooler nitrogen temperature. This result is coherent with published studies [4-7].

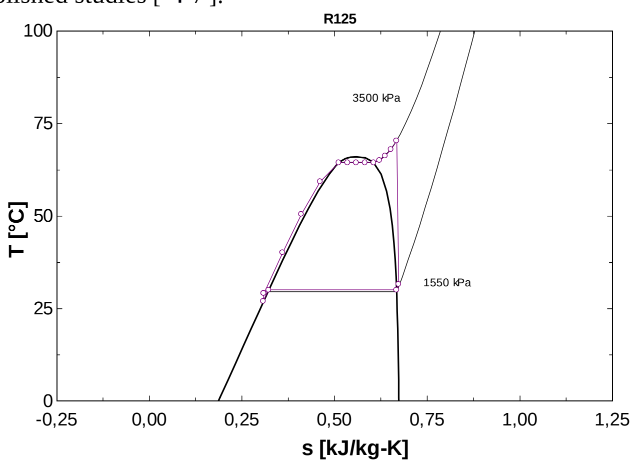

Figure 13 R125 subcritical ORC optimization, based on intercooler waste heat

CONCLUSION

After having presented the basis of the model, preliminary studies have been carried out on the design of ORC using three different fluids: traditional and mature R245fa technology, innovative R125 cycle and high pressure low temperature supercritical CO_2 machine. Different architectures have been compared and advantages of using supercritical cycles have been underlined. From a thermodynamic point of view the best solution would be to use R125 cycle at a supercritical pressure at precooler secondary side and at a subcritical pressure at intercooler secondary side. A supplementary power of more than 15 MWe could be achieved. The resulting combined cycle could reach a net efficiency of 39% on condition that pressure losses in the hot heat exchanger are limited.

Next step will consist in the economic evaluation of the different components of each system. We will also investigate alternative refrigerants to R125 with a lower GWP.

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