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# A CCHP system fed by low enthalpy sources in Mediterranean area

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**ABSTRACT**

Climate change and energy consumption increasingly call for a better use of energy sources. Renewable sources are still growing in the power production of many countries, so leading to the reduction of $CO_2$ emissions. Further restrictions will be applied to the use of fossil fuels with a lot of consequences; for example, some German cities are all contemplating bans related to diesel pollution, and on the other hand some of the biggest car producers are planning to stop the diesel engines production in a few years. At the same time, some restrictions are applied also to many refrigerating fluids, in order of limiting the ODP and GWP values. This restriction leads to research new fluids and/or to enhance the use of natural refrigerants. In this scenario, Energy Designers are called to conjugate energy needs with the environment safety. Main guidelines concern energy clean production basically obtained by means of renewables and energy saving. In this article authors consider a CCHP for a typical Mediterranean end user such as a large hotel, composed by an ORC linked to a refrigerator. Several working fluids, particularly natural fluids, which operate both in the ORC and in the refrigerator, are investigated.

## 1. INTRODUCTION

The Weather Meteorological Organization (WMO) has confirmed that the years 2015, 2016 and 2017 are the three warmest years with anomalous temperature conditions, lower precipitations and lack of snowfall. Meantime, quick and extreme adverse weather conditions more frequently occurred in several world regions (e.g. hurricanes, heavy rains, exceptional copious snowfall, etc.). As it is well known, fossil fuels are among the main responsible for climate change and this led all countries to cooperate for resolving or, at least, mitigating this problem.

In this context, the energy policies of Developed Countries, following mainly the COP 21 targets, aim at reducing the impact on climate caused by the power production, aim at better using the energy sources and at increasing the use of renewable energy sources. The transition towards an increasing use of sustainable energy systems yet leads to rethink the feasible use of the current available technologies.

This work deals with Combined Cooling, Heating and Power (CCHP) applications fed by low enthalpy sources such as renewable energy or waste heat ones. Specifically, the authors suggest an Organic Rankine Cycle (ORC) combined with a Refrigeration cycle to satisfy the needs of an end-user especially a large hotel. This residential building cluster is located in the Mediterranean area where refrigeration load is considered a priority, but also power and Domestic Hot Water (DHW) are very important loads.

The proposed system has been investigated with several pairs of fluids. The simulations have been performed with parameters related to the end-user chosen.

## 2. OBJECTIVES AND LITERATURE REVIEW

Energy demand is continuously growing, due to a better quality of life for a greater number of persons, a higher industrialization and new developing markets. On the other side environmental problems urge and demand to change our way to utilize the Earth resources. Fossil fuels with their $CO_2$ emissions led to increase temperatures and consequently vary the global equilibrium, but also all oil derivatives caused harmful effects. On the other hand, even the compatibility between the use of renewable energy and the actual structure of electricity, heating and transport sectors, requires a specific attention [1].

Latest energy policies try to give a straight line in order to reduce the growth of $CO_2$ levels and mitigate the climate changes. Many common fluids utilized in the refrigerating equipment present high values of the Global Warming Potential (GWP) index, thus denoting a strong impact on the climate change. Consequently, for example, the European Regulation No 517/2014, dated 16th April 2014 [2], stated a step-by-step phase out of fluorinated greenhouse gases, particularly HFC fluids, which thus are no longer a possible choice when designing thermal plants.

Moreover, in the last years, the trend is to enhance the use of renewable energy sources and energy savings and words such as nearly Zero Energy Building (nZEB) have become even more common. Therefore, it is better to consider each building not only as an energy user, but also as a power producer so to reduce the energy inlet.

A hard work is thus demanded to engineers, which have to consider more carefully the safety design of plants and equipment, besides satisfying the customer needs.

Each application has its own peculiarities and must be regarded as an *unicum*. In the case of hotels, the cogeneration has to be considered as a first option because three different energy uses (i.e. heating, cooling and electric power) are usually needed in these types of buildings.

Cogeneration is a well-known technique utilized all over the world and CCHP, sometimes called trigeneration, is more and more investigated in order to achieve a greater energy efficiency. Lately, CCHP applications have been studied also for net zero-energy settlements, as a large hotel or a holiday village may be regarded.

Ascione et al. [3] investigated a residential village in Greece, called "Olympiad village". A detailed analysis is performed, but, in this case, the HVAC system is connected with electric heat pumps whereas power production is assured with photovoltaic fields and wind turbines.

Ashourian et al. [4] and also Diab et al. [5] pursued the objective of a green energy management with a hybrid system that includes again photovoltaic fields and wind turbines in conjunction with diesel generators and battery packages. However, the use of diesel engines arises some concerns particularly because of the generated pollution. These engines indeed cause a high emission of $NO_x$ besides $CO_2$ and several particulates as $PM_{10}$. The diesel vehicle market is worried by likely restrictions on the circulation of diesel vehicles (as, for instance, that imposed by the decision of the Germany's federal administrative court in Leipzig ruled on 2018 February, 27th that permits to each city to limit the circulation of diesel vehicles) and by the announcement of several car makers - FCA, Volvo, Toyota, etc. - to stop the development and production of diesel engine in a few years.

As a result, it is hard to think a future for diesel engine also in other applications.

Mohammadi et al. [6] studied a CCHP composed of a Gas Turbine, an ORC and an absorption refrigerator. The combustion of a fuel permits to reach an inlet temperature value of 350 °C at the ORC turbine. Likewise, Lizarte et al. [7] have contemplated, in their parametric study, high values of the ORC turbine inlet temperature that ranged from 85 to 315 °C, and a refrigerating cascade cycle with Ammonia and Carbon Dioxide enabling to reach -55 °C. La Rocca et al. [8] have simulated an ORC system with a turbine inlet temperature of 60 °C.

## 3. THE CCHP SYSTEM

The proposed CCHP system is designed to satisfy a user that needs heating, cooling and power. In order to fix ideas, a hotel located in the Mediterranean area was analyzed [9], whose main duty is therefore the refrigeration load during the summer season. This duty was estimated at approximately 500 kW of cooling load. Figure 1 shows a sketch of the user arrangement.

The CCHP system consists of an Organic Rankine Cycle (ORC) combined with a Refrigerating Vapor Compression Cycle (REF). The ORC is able to give power, the refrigerator assures the cooling load coverage and both cycles are able to recovery the condensation heat to produce Domestic Hot Water.

The work produced by the ORC turbine is used to drag both the REF compressor and the electric generator so at the same time it is possible to get a cooling load and power.

In this study, the inlet heat to the ORC boiler is considered coming out from a low enthalpy source, such as a field of solar collectors, a geothermal source or waste heat from some industrial facility.

All the above cited assumptions characterize the simulation performed in the present study, as reported in section 4.2.

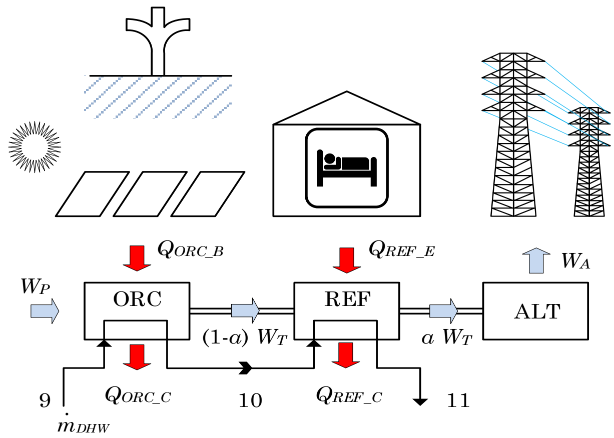



**Figure 1.** A model of a CCHP plant for the hotel

In Figure 2 the main components of the proposed system are illustrated.

Each of the two cycles must operate with a working fluid that is able to change phase.

At the ORC inlet fixed enthalpy level, only fluids with quite a low critical temperature may be considered.

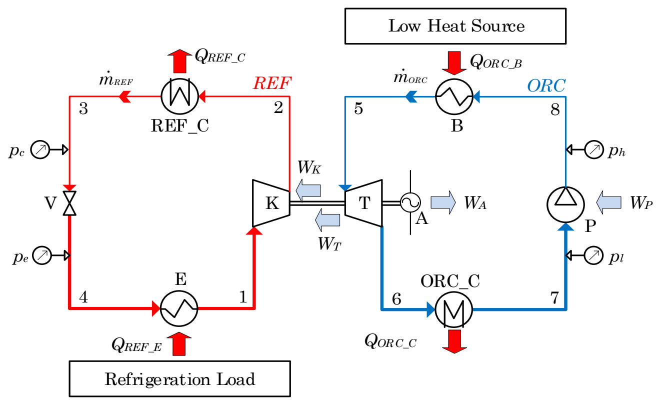



**Figure 2.** The combined ORC and refrigerating equipment

### 3.1 Fluids

Among the fluids that are suitable for refrigerating and for the ORC equipment, many cannot be considered because of the restriction imposed by Authorities.

Natural refrigerants seem to be a good choice to satisfy all current requirements. In this field, many fluids belong to the hydrocarbon category (HC). They are obviously flammable and therefore it must be carefully utilized. Nevertheless, they are generally used in domestic refrigerators (propane is one of the most common fluids) because little amounts of fluid are contained in them; the currently imposed limit is in fact 150 g.

However, because of the current restrictive trend on refrigerating fluids and the necessity of ensuring heat and power, a new point of view should be adopted. Technically there is no problem, in fact, a lot of fuel is daily manipulated and, according to new regulations HC refrigerants may be adopted. A great interest is shown in this direction, and some countries are thinking to enlarge the safe use of HC refrigerants to refrigeration, HVAC and – with ORC – power generation [10].

In the study presented here, six refrigerant fluids have been selected and investigated both for the REF cycle and for the ORC cycle: Ammonia, R717, is the typical industrial inorganic refrigerating fluid, classified as B2 for toxicity and flammability, according to the classification scheme (summarized in Table 1) adopted by standards ISO 817 and EN 378; the hydrocarbons Isobutane, R600a, and Propane, R290, are the most widely used fluids in non-industrial applications; Propylene, R1270, is used in many applications, sometimes in blend with other HCs; Toluene is an hydrocarbon widely considered in ORC applications [10]; R245fa is one of the former fluids well considered in the ORC research, but it is a HFC and consequently it will come to phase out in the next year; R134, is a common fluid widely utilized in refrigerating and ORC equipment, but it is also in phase out.

Table A1, in the Appendix, shows the main parameters of these fluids. All the previous fluids were investigated and R134a was considered as a reference.

**Table 1.** Refrigerant safety classification scheme

| Classification | | | Toxicity | |
|---|---|---|---|---|
| | | | Class A | Class B |
| | | | Lower chronic toxicity | Higher chronic toxicity |
| Flammability | Class 1 | No flame propagation | A1 | B1 |
| | Class 2 | Lower flammability | A2 | B2 |
| | Class 3 | Higher flammability | A3 | B3 |

Notes: extracted by [9]

Figure 3 shows the *T-s* diagram for all the selected fluids, in order to compare their properties at a glance. For each fluid, the isothermal-isobaric line at 1 bar is reported as a dashed line (the limit is only in the saturated region for clarity). As it can be observed, only Toluene presents a value that exceeds the 100 °C, consequently all the related cycles lie in the sub-atmospheric region.

## 4. THERMODYNAMIC ANALYSIS

The calculations were performed based on the following assumptions:

- thermodynamic equilibrium exists throughout the system and the cycle is operated under steady-state conditions;
- in tubing and components, pressure losses are negligible, and they are adiabatic;
- through the valve, flow has the same enthalpy at inlet and outlet;
- the ORC pump, for the sake of simplicity, is considered isentropic, while for the compressor and the turbine an isentropic efficiency is assumed.

The thermodynamic models of the proposed plants have been developed using the well-known Engineering Equation Solver (EES) software [11].

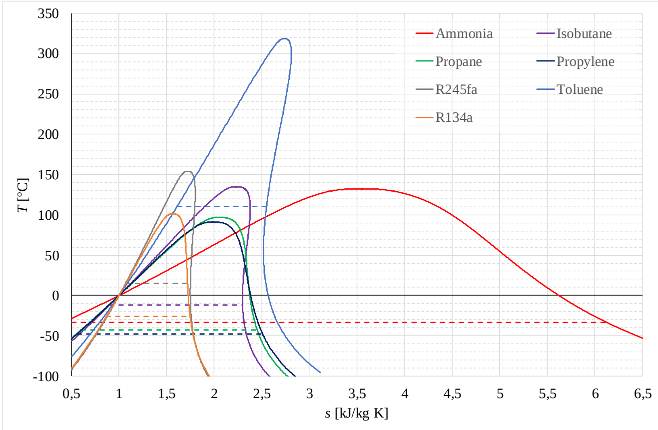



**Figure 3.** The *T- s* diagram for the investigated fluids

### 4.1 Model

Bearing in mind the plant scheme, the following equations define the model.

$$W_T = \dot{m}_{ORC}\left(h_5 - h_6\right) = \dot{m}_{ORC}\,\eta_{ORC\_T}\left(h_5 - h_{6s}\right) \quad (1)$$

$$W_K = \dot{m}_{REF}\left(h_2 - h_1\right) = \dot{m}_{REF}\frac{h_1 - h_{2s}}{\eta_{REF\_K}} \quad (2)$$

$$\dot{m}_{REF} = \frac{Q_{REF\_E}}{h_1 - h_4} \quad (3)$$

$$\dot{m}_{ORC} = \dot{m}_{REF}\frac{h_2 - h_1}{\left(1-\alpha\right)\left(h_5 - h_6\right)} \quad (4)$$

where the parameter $\alpha$ is expressed in terms of turbine power fraction needed for feeding the electric alternator. In the following this parameter is assumed equal to 0.5, i.e. half turbine power feeds the compressor and the rest feeds the alternator.

$$W_A \equiv \alpha W_T = W_T - W_K \quad (5)$$

$$Q_{REF\_C} = \dot{m}_{REF}\left(h_2 - h_3\right) \quad (6)$$

$$Q_{ORC\_C} = \dot{m}_{ORC}\left(h_6 - h_7\right) \quad (7)$$

$$Q_{DHW} = Q_{REF\_C} + Q_{ORC\_C} \quad (8)$$

$$Q_{ORC\_B} = \dot{m}_{ORC}\left(h_5 - h_8\right) \quad (9)$$

$$W_P = \dot{m}_{ORC}\left(h_8 - h_7\right) \quad (10)$$

### 4.2 Parametric study

The simulation parameters were set up considering the specific application, that is a hotel sited in the Mediterranean area. The maximum boiler temperature, achieved thanks to the low enthalpy sources considered, may not exceed 90 °C and the typical HVAC load ranges between 7 and 12 °C at the cold exchanger, so the evaporating temperature may be set to 5 °C.

DHW must be at least at 50 °C; since the heat is recovered by the two condensers of ORC and REF, it is reasonable to set the REF condensing temperature to 60 °C, while in order to maximize as possible the turbine's work, the ORC condensing temperature was set to 20 °C. The fresh supply water was considered at 15 °C.

**Table 2.** The values of parameters used for simulations

| | | | |
|---|---|---|---|
| $Q_{REF\_E}$ | 1 | kW | REF: Cooling load |
| $T_e$ | -5 | °C | REF: Evaporating temperature |
| $T_c$ | 60 | °C | REF: Condensing temperature |
| $T_h$ | 60 | °C | ORC: $T_{sat}$ at high pressure, $T_{sat}\,(p_h)$ |
| $T_l$ | 20 | °C | ORC: $T_{sat}$ at low pressure, $T_{sat}\,(p_l)$ |
| $T_{max}$ | 90 | °C | ORC: Maximum temperature |
| $T_9$ | 15 | °C | DHW: Inlet temperature |
| $T_{11}$ | 50 | °C | DHW: Outlet temperature |
| $\eta_{REF\_K}$ | 0.80 | | REF: Compressor isentropic efficiency |
| $\eta_{ORC\_T}$ | 0.80 | | ORC: Turbine isentropic efficiency |
| $x_1$ | 1 | | REF: Quality at the compressor suction |
| $x_3$ | 0 | | REF: Quality at the condenser outlet |
| $x_7$ | 0 | | ORC: Quality at the condenser outlet |
| $\alpha$ | 0.5 | | ORC: Power ratio Alternator/Turbine |

Neither sub-cooling nor superheating were considered in the study.

In the simulation, the refrigeration load is assumed equal to $Q_{REF\_E}$ = 1 kW. For our case study characterized by a refrigeration load equal to 500 kW, it is possible to scale all extensive parameters.

The simulation has calculated the following parameters: mass flows, work and heat fluxs.

Figure 4 shows the phase diagram for the cycle R134a-R134a. For the remaining 36 cycles the diagrams are quite similar.

Please take note that in the remaining cycles, the plant combination is indicated as "REF_fluid-ORC_fluid", so the first term refers to the fluid that circulates in the REF cycle while the latter refers to the fluid that is used in the ORC cycle.

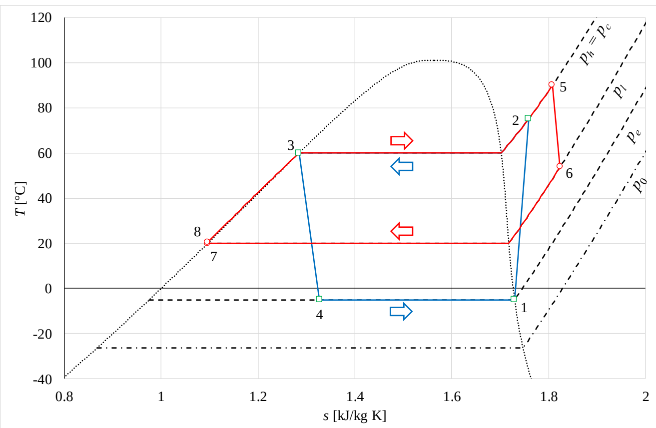



**Figure 4.** The *T-s* diagram for the "R134a-R134a" CCHP

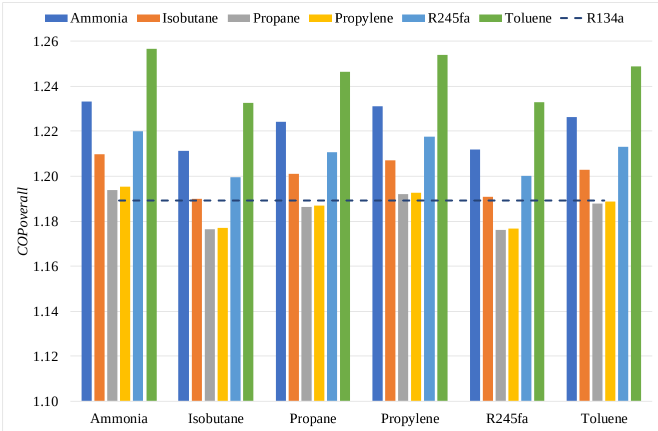
## 5. RESULTS AND DISCUSSION

The results of the simulation are reported as graphs in the following discussion.

We have to consider that, both in the REF cycle and in ORC cycle, for the fixed parameters, the state points of a working fluid are the same in all the combinations. When a parameter depends on the linkage between the two cycles, we have a lot of data, i.e. 36 different values, but when it refers to a single cycle, we have only 6 values. As stated earlier, R134a was considered as a reference condition.

Figure 5 shows the overall COP calculated as reported in [12] for all plant combinations:

$$COP_{overall} = \frac{W_A - W_{ORC\_P} + Q_{REF\_E} + Q_{DHW}}{Q_{ORC\_B}} \qquad (11)$$




Notes: Data are collected by ORC_fluid, indicated below. Columns report values for each REF_fluid, as stated in the legend above. The dashed line represents the value for the cycle "R134a-R134a", assumed as reference.

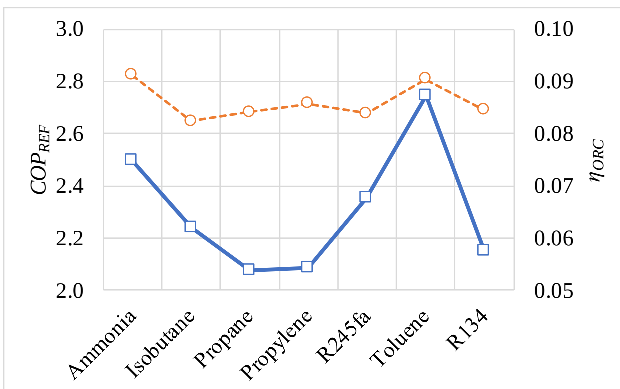
**Figure 5.** Overall COP

Generally, each plant combination presents a COP value that exceeds 1.1, because all the energies are utilized by an end user.

It is possible to observe that each cycle of the type "Toluene-any_Fluid" presents the highest values, but, as noted in the previous section and as we can see in the following, the Toluene works in vacuum conditions and this surely is not an optimal condition.

It has resulted that Toluene is followed by Ammonia that reveals a good performance in all plant combinations. Next there are R245fa, then Isobutane and last Propylene and Propane.

Moreover, only the Ammonia, as ORC_fluid, presents in all combinations values higher than those of the reference plant "R134a-R134a".




Notes: The continuous line report the $COP_{REF}$, values in the axis on the left, the dashed line the $\eta_{ORC}$, values in the axis on the right

**Figure 6.** Efficiencies of the cycles

Figure 6 reports the efficiencies of the ORC and refrigerating cycle considered as a stand-alone, with the same parameters of the previous simulations.

They are computed as:

$$COP_{REF} = \frac{Q_{REF\_C}}{W_{REF\_K}} \tag{12}$$

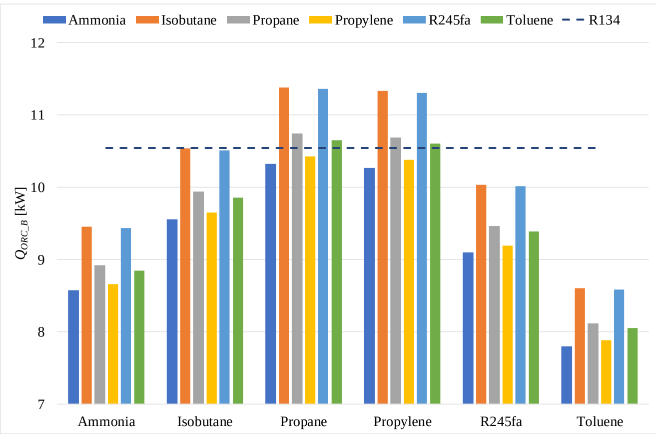
$$\eta_{ORC} = \frac{W_A}{W_{ORC\_T}} \tag{13}$$

It is possible to observe that each ORC efficiency is approximately 8-9 %, which are low values compared to the literature ones

It is likely that these low values are caused both by an unsuitable optimization of the parameter-setting process and also by quite small operational working temperatures.

The $COP_{REF}$ values range between 2 and 2.8, with the highest values in case of Toluene and Ammonia.

Figure 7 reports the heat flux needed at the ORC boiler. In same way the diagram is related to Figure 5, denoting that Toluene and Ammonia, as REF_fluid, in the fixed conditions need the lowest heat supply from the low enthalpy source. The reported values indicate kilowatts per kW of refrigerating load.




Notes: Same as Figure 5.

**Figure 7.** Heat supply to the ORC boiler

Figure 8 and Table 3 report the pressure values for each cycle. As it can be observed, Toluene is always below the atmospheric pressure and R245fa shows the same behavior only in the REF evaporator.

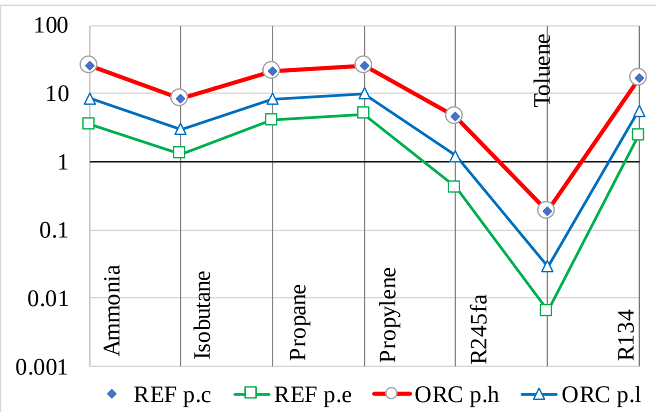



**Figure 8.** Pressure trends for the cycles

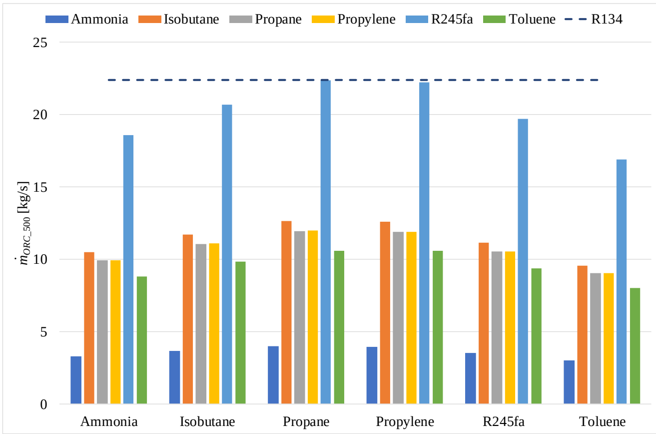
**Table 3.** Working pressure for all the fluids [bar]

| | REF $p_c$ | REF $p_e$ | ORC $p_h$ | ORC $p_l$ |
|---|---|---|---|---|
| Ammonia | 26.14 | 3.549 | 26.14 | 8.578 |
| Isobutane | 8.684 | 1.307 | 8.684 | 3.02 |
| Propane | 21.17 | 4.061 | 21.17 | 8.366 |
| Propylene | 25.34 | 5.03 | 25.34 | 10.19 |
| R245fa | 4.619 | 0.4195 | 4.619 | 1.224 |
| Toluene | 0.1863 | 0.006485 | 0.1863 | 0.02911 |
| R134 | 16.83 | 2.435 | 16.83 | 5.721 |

The vacuum condition may be a worse condition, although the use of HCs requires a perfect sealing anyway.

While an external leakage may be easily monitored, the introduction of air in the tubing is more insidious in order to have a fuel mixture.

On the other side, Ammonia presents the highest values in the cycles, values that need a construction heavier in tubing and components.




Notes: Same as Figure 5.

**Figure 9.** ORC Mass flow for $Q_{REF\_E} = 500$ kW

Figures 9 and 10 report the mass flow for a CCHP refrigerating load of 500 kW, respectively for the ORC and the REF cycle.

The lowest value of ORC mass flow, 2.998 kg/s, has been found in the case of Ammonia in the “Toluene-Ammonia” combination. Ammonia has mass flows lightly higher upon this value. Except R134a, which presents the highest value of 22.41 kg/s, the highest values are for combinations that have the R245fa as REF_fluid. All HCs, used as refrigerating fluids, have values around 10 kg/s.

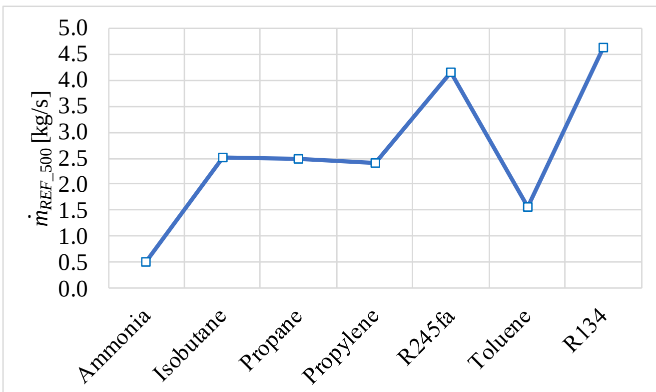



**Figure 10.** REF Mass flow for $Q_{REF\_E} = 500$ kW

Figure 10 presents values strongly lower for each fluid, starting from Ammonia with the lowest value of 0.5187 kg/s to the highest of 4.151 kg/s for R245fa (except once again R134a with 4.623 kg/s).

## 6. CONCLUSIONS

The simulations reported here show an interesting possibility to obtain trigeneration starting from low enthalpy sources and utilizing natural refrigerants, thus reducing the environment impact.

Surely, among the fluids considered, Ammonia has resulted to be one of the most attractive. It is a natural fluid, already utilized in many applications, particularly in the industrial sector. Hydrocarbons may also be a considerable alternative to common refrigerants.

Naturally, toxicity and flammability are not negligible aspects. Only with appropriate techniques and conductions these fluids will be suitable for end users, but new government directives will be necessary. Further investigations on this matter are certainly necessary.

Other questions, besides the previous, that have not been investigated in this article are still open. For example, the availability of technologies is strictly correlated to the economic feasibility, and - here - the material compatibility may be a key parameter, e.g. Ammonia is not compatible with copper in the heat exchangers, so other materials must be utilized.

## 7. ACKNOWLEDGMENT

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## NOMENCLATURE

| | |
|---|---|
| AHU | Air Heating Unit |
| CCHP | Combined Cooling, Heating and Power |
| $COP_{overall}$ | CCHP Coefficient of Performance |
| $COP_{REF}$ | REF Coefficient of Performance |
| DHW | Domestic Hot Water |
| GWP | Global Warming Potential |
| $h$ | Enthalpy, kJ kg$^{-1}$ |
| HC | Hydrocarbon |
| HFC | Hydrofluorocarbons |
| HVAC | Heating, Ventilation, and Air Conditioning |

| | |
|---|---|
| $\dot{m}_{ORC}$ | ORC mass flow, kg s$^{-1}$ |
| $\dot{m}_{REF}$ | REF mass flow, kg s$^{-1}$ |
| $\dot{m}_{DHW}$ | DHW mass flow, kg s$^{-1}$ |
| NOx | Nitrogen Oxides |
| nZEB | near Zero Energy Building |
| ORC | Organic Rankine Cycle |
| ORC_fluid | Generic ORC working fluid |
| $p_0$ | Atmospheric pressure, bar |
| $p_c$ | REF Condensing pressure, bar |
| $p_e$ | REF Evaporating pressure, bar |
| $p_h$ | ORC high pressure, bar |
| $p_l$ | ORC low pressure, bar |
| $Q_{DHW}$ | DHW Heat flux, kW |
| $Q_{REF\_E}$ | Refrigerating load, kW |
| $Q_{ORC\_B}$ | ORC Boiler Heat flux, kW |
| $Q_{ORC\_C}$ | ORC Condenser Heat flux, kW |
| $Q_{REF\_C}$ | REF Condenser Heat flux, kW |
| REF | Refrigerator |
| REF_fluid | Generic REF working fluid |
| $s$ | Entropy, kJ kg$^{-1}$ K$^{-1}$ |
| $T_c$ | REF: Condensing temperature, °C |
| $T_e$ | REF: Evaporating temperature, °C |
| $T_h$ | ORC: $T_{sat}$ at high pressure, °C, $T_{sat}(p_h)$ |
| $T_l$ | ORC: $T_{sat}$ at low pressure, °C, $T_{sat}(p_l)$ |
| $T_{max}$ | ORC: Maximum temperature |
| $T_{sat}(p)$ | Saturation temperature at pressure $p$, °C |
| $W_A$ | Alternator power, kW |
| $W_K$ | REF Compressor power, kW |
| $W_P$ | ORC Pump power, kW |
| $W_T$ | ORC Turbine power, kW |
| $x$ | Quality |

**Greek symbols**

| | |
|---|---|
| $\alpha$ | Turbine power fraction to feed the electric alternator |
| $\eta_{ORC}$ | ORC Efficiency |
| $\eta_{ORC\_T}$ | REF Compressor isentropic efficiency |
| $\eta_{REF\_K}$ | ORC Turbine isentropic efficiency |

**Subscripts**

| | |
|---|---|
| 1..4 | REF_fluid thermodynamic state |
| 5..8 | ORC_fluid thermodynamic state |
| 9..11 | DHW thermodynamic state |

## APPENDIX

**Table A1.** Fluid characteristics

| | | **R 717 Ammonia** | **R600a Isobutane** | **R290 Propane** | **R1270 Propylene** | **R245fa** | **Toluene** | **R134a** |
|---|---|---|---|---|---|---|---|---|
| CAS#: | | 7664-41-7 | 75-28-5 | 74-98-6 | 115-07-1 | 460-73-1 | 108-88-3 | 811-97-2 |
| Molar mass: | kg/kmol | 17.03 | 58.122 | 44.096 | 42.08 | 134.05 | 92.138 | 102.03 |
| Triple point | °C | -77.65 | -159.42 | -187.625 | -458.35 | -102.1 | -95.15 | -103.3 |
| NBT | °C | -33.33 | -11.75 | -42.11 | -320.77 | 15.14 | 110.6 | -26.07 |
| Critical temp. | °C | 132.25 | 134.66 | 96.74 | -182.09 | 154.01 | 318.6 | 101.06 |
| Critical pressure | bar | 113.33 | 36.29 | 42.512 | 455.5 | 36.51 | 41.263 | 40.593 |
| Refrigerant No. | | R717 | R600a | R290 | | R245fa | | R134a |
| Type | | Inorganic | HC | HC | HC | HFC | HC | HFC |
| Classification | | B2L | A3 | A3 | A3 | B1 | | A1 |
| ODP | | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| GWP | | 0 | 3 | 3 | 2 | 1,030 | 3 | 1,430 |
| GWP (AR5) | | 0 | 3 | 3 | 2 | 856 | 3 | 1,300 |

Notes: There are two values of GWP. The use of GWP (AR5) values is recommended. The AR5 values (IPCC Fifth Assessment Report, 2014 - AR5) are the most recent, but the previous values are also listed because they are sometimes used for inventory and reporting purposes