EXTENSIVE TECHNO-ECONOMIC ASSESSMENT OF COMBINED INVERTED BRAYTON – ORGANIC RANKINE CYCLE FOR HIGH-TEMPERATURE WASTE RECOVERY

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ABSTRACT

The general trend to the decarbonisation stimulates the continuous development of waste heat recovery technologies. Nowadays, the most deployed solution in this field up to certain installed capacity is the organic Rankine cycle (ORC). However, for high-temperature cases, above 400 ºC, this technology has essential limitations on exploited temperature because of the properties of available working and intermediate heat-transfer fluids. As a sequence, waste heat recovery based on the ORC technology has lower efficiency than it could have from the thermodynamic viewpoint. This work continues the study of a combined scheme based on the coupling of the ORC with the inverted Brayton cycle (combined IBC-ORC) which enables to use the high-temperature waste heat potential more effectively. In this scheme, hot gas from a heat source expands in the IBC turbine to the subatmospheric pressure created by a compressor which follows in the downstream of the gas duct. Before the compressor, gas transmits heat via heat exchangers to regenerative ORC and partially to the atmosphere. The scheme may be employed for waste heat recovery from high-temperature heat sources such as internal combustion engines and some technological processes such as cement kilns or heat treating furnaces. In the paper, the analysis of the scheme performance under different ambient temperatures is presented, showing the negative effect of high ambient temperature. Specificities of the optimisation results are explained with the analysis of IBC performance, considering water condensation issues. Pareto fronts for system electric efficiency and levelized cost of energy for several temperatures of the primary heat source provide valuable insight for techno-economic assessment, recommending the most suitable sets of parameters in the trade-off between electrical power and investment results.

1. INTRODUCTION

Recent decades, many works have been performed in the field of waste heat recovery (WHR) and, in particular, internal combustion engine (ICE) exhaust gas heat recovery. Elson *et al.* (2015) and Jouhara *et al.* (2018) have thoroughly reviewed the major sources of waste heat and technologies applied for recovery. Organic Rankine cycle (ORC) nowadays resulted in being the most deployed solution both in terms of the volume of scientific publications and industrial applications (Campana et al., 2013). The broad scope of different configurations of this technology has been studied, showing their pros and cons and giving a room for further improvements (Lecompte *et al.*, 2015).

The heat which is suitable for the recovery and conversion to electricity may have different temperature level. Among the wide variety of heat sources in the industry and residential sector, there is a number of technologies with the temperature of waste heat flows higher than 400 ºC such as internal combustion engines, heat treating furnace, cement kilns and other (Elson *et al.*, 2015). For such a high-temperature application, ORC as a WHR technology has an essential disadvantage associated with the maximum possible temperature of the working fluid and thermal-oil used in the intermediate thermal-oil loop (TOL). That imposes a limit on the reachable efficiency level. Other technologies usually applied for higher temperatures are not suitable for small scale application. For instance, the steam Rankine cycle has the unfitting size of the expander and heat exchangers due to the high volumetric ratio in different phases and significant latent heat value. Also, operational challenges for the steam cycle follow out from the type of the water phase diagram, which is the "wet" type (Pethurajan *et al.*, 2018). Another example is a conventional gas turbine with external heating. However, usually, the temperature of exhaust gases does not exceed 600-650 ºC that is not high enough for this technology, especially considering the need in the heat exchanger, which lowers thermal potential of the exhaust, and relatively low efficiency of small-scale turbomachinery. Accordingly, the sources of exhaust gas with temperature in the range of approximately 400-600 ºC do not yet have a technology, which could exploit all the thermodynamic potential of this waste heat.

This work studies an alternative solution which may utilise the high-temperature exhaust with higher efficiency, smaller layout and better economic indices of investments in comparison with the conventional ORC. Combined inverted Brayton – organic Rankine cycle (IBC-ORC) demonstrates all these advantages. IBC usage enables to exploit the high-temperature more efficiently as the heat is converted to power by turbine immediately after being exhausted. It reduces the size of all subsequent heat exchangers what decreases layout and keeps capital cost on a similar level with ORC-based facility. Considering higher efficiency, and hence higher capacity, the combined scheme (CS) has better expected investment results as well, what was demonstrated by Abrosimov *et al.* (2019).

In that work (Abrosimov *et al.,* 2019), for WHR from the 1 MW natural gas fueled engine, three fluids were assessed for the better performance in high-temperature ORC under parametric limitations provided by Quoilin *et al. (*2012) for several fluids of different levels of critical temperature. So, the study was focused on R245fa, pentane, and toluene, which corresponds to the low, middle, and high critical temperature levels. Pentane has demonstrated the best results under the system efficiency optimisation; it was used for further analysis. Then, two configurations of the CS were compared with the conventional ORC also in two configurations. Pentane was used as the ORC working fluid for all schemes. Thermodynamics and economic optimisations with the system efficiency and levelized cost of energy (LCOE) as objective functions correspondingly have shown the superiority of the combined scheme with the regenerative ORC (CS Reg) for both objectives. This configuration is explained in details further in this work.

In this work, for the same case-study, a more detailed analysis is performed for the advantageous configuration of the CS. Influence of the ambient temperature on the scheme performance was assessed for the nominal exhaust temperature with particular attention on the IBC performance, which has an essential impact on overall scheme optimisation results. The Pareto fronts plotted, helping to choose an appropriate design strategy, depicts the trade-off between economics attractiveness (LCOE) and electric system efficiency for 470, 520, and 570 ºC temperatures of the heat source.

To the best of our knowledge, the WHR technology discussed in this work was not studied in the scientific literature besides Abrosimov *et al.* (2019). So, the extensive assessment of the scheme performance and its achievable techno-economic results contribute to the further development of the proposed solution and help to clarify its economic prospects.

2. SCHEME DESCRIPTION

As previously mentioned, the WHR system studied in this work is a combination of IBC, and regenerative ORC applied for the case of 1 MW natural gas fueled engine. In Figure 1, the schematic view of the system is reported. At point 1, hot gas from the ICE enters the system and expands to the subatmospheric pressure in the turbine. Further, the temperature of the gas decreases, first, in the heat exchanger (HE-1) cooled by the intermediate TOL (point 3), then, in the cooler fanned with air (point 4). On the second stage of cooling, the condensation may occur, so a gas-liquid separator was introduced before the compressor. The compressor takes back the flow to the atmospheric pressure (point 5). The work consumed by the compressor is approximately one third smaller for the nominal case than the work produced by the turbine. The difference is the useful work of the IBC part of the system. Previously mentioned intermediate loop, in turn, transmits heat to the ORC part of the scheme. The loop itself enables to operate high-temperature ORC with sufficient reliability and controllability and, thus, is compulsory from the engineering point of view (Shi *et al.*, 2018). At point 21, the working fluid of the ORC is in a vapour phase and has the maximum temperature, being heated up by thermal-oil. It expands in the expander to the pressure, which for the pentane is slightly below the atmospheric one (point 22). Then, the flow goes through the regenerator and transfers heat to the fluid from the Condenser. In the air-cooled condenser, the temperature of the working medium comes close to the ambient and the superheated vapour condensed (point 23). The ORC pump (III) increases the pressure of the working fluid and directs it through the regenerator to the evaporator (point 24). Power generators are mantled on the shafts of both the turbine-compressor unit of IBC and the expander of ORC. The equipment types assumed in this scheme (Turton *et al.*, 2008) are reported in the list below:

- Heat exchanger HE-1: finned cross-flow shell and tube with a cylindrical housing
- Evaporator: bayonet shell and tube
- Condenser: air-cooled finned tubes
- Regenerator: "multiple pipe" shell and tube
- Cooler: air-cooled finned tubes
- Turbine, compressor, expander: radial bladed machines
- Separator: venturi scrubber

Abrosimov *et al*. (2019) provide more details for understanding of the scheme operation.

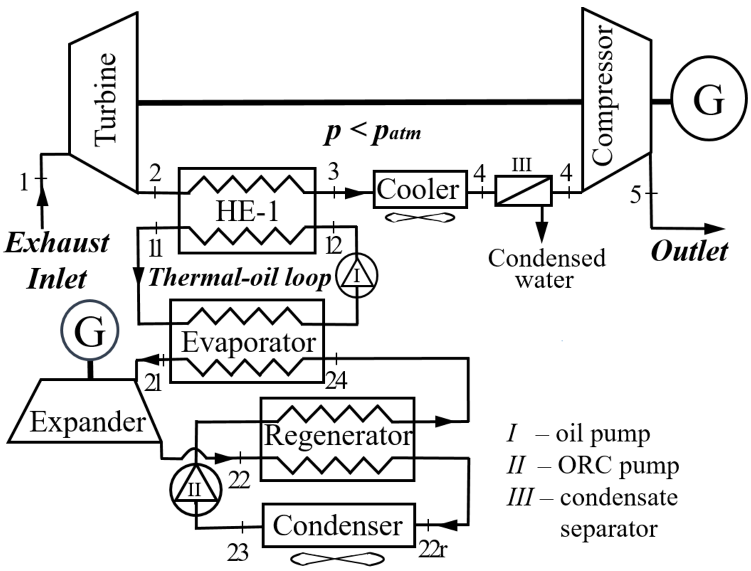

Figure 1: Combined scheme: IBC plus regenerative ORC

3. METHODOLOGY

A 1-D model of the IBC-ORC was designed in the Aspen HYSYS v.9 environment and simulated in steady-state for predesign parameter setting, meaning that design of the components was assumed to be varying. Peng-Robinson equation of state was adopted for flue gas, RefProp for organic fluids and UNIQUAC for thermal-oil (Baccioli *et al.*, 2018). Standard Aspen "building blocks" were used with desired settings corresponding to the suitable and feasible components.

Pressure drops in heat exchangers were assessed with an introduction of the relative pressure drops, so they vary and depend on the pressure of the incoming flow.

Average heat transfer coefficients were adopted from Sinnott *et al.* (2005) for the assessment of the surface area of heat exchangers. The evaporator was split into three parts corresponding to liquid preheating, evaporation, and superheating zones to make its modelling more accurate.

Air cooling of the condenser and intermediate cooler is calculated based on the assumption of a fan electric consumption of 17 W of power for 1 kW of the rejected heat according to Antonelli *et al.* (2016).

Water condensation in the flue gas was assumed was evaluated by the Peng-Robinson equation of state. The gas-liquid separator ensured the complete removal of water before the compressor (Scheme description). Possible corrosion is negligible, considering the low sulfur concentration in the natural gas (Kass *et al.*, 2005).

3.1. Efficiency of bladed machines in sub-atmospheric conditions

The efficiencies of Turbine and Compressor were assigned based on high typical values of the turbomachinery efficiency corrected on the IBC conditions of the exploitation, following an approach from Henke *et al.* (2013). The basic approach was advanced to have a possibility to correct values of the efficiency during the optimisation. In more details, this methodology is described in Abrosimov *et al*. (2019). Table 1 provides data about efficiency values for hypothetical BC conditions and corresponding IBC.

Table 1: Assumed efficiency of turbomachines in BC and IBC conditions.

	BC	**IBC**
Turbine efficiency	87 %	86 %
Compressor efficiency	85 %	83.7 %

3.2. Levelized cost of energy assessment

LCOE calculation is based on the simple assumption for the standard LCOE calculation (Pili *et al.*, 2017):

$$\mathrm{LCOE} = \frac{\mathrm{C_{tot}} + \sum_{\mathrm{i}=1}^{\mathrm{t}} \frac{\mathrm{C_{O\&M.i}}}{(1+\mathrm{r})^{\mathrm{i}}}}{\sum_{\mathrm{i}=1}^{\mathrm{t}} \frac{\Delta \mathrm{E_i}}{(1+\mathrm{r})^{\mathrm{i}}}} \tag{1}$$

Where C_{tot} is total capital investment cost, $C_{O\&M}$ is operation and maintenance cost (3 % of C_{tot}), t is run-time of the plant in years (20 years), r stands for the discount rate (7 %) (Freyman & Tran, 2018), ΔE is the net energy generated by the system over one year (considered capacity factor[1] (CF) is equal to 0.9). Single-tranche capital investments are assumed. Taxation, year-by-year growth of $C_{O\&M}$, performance reduction, as well as insurance payments, were not considered. As so, ΔE and $C_{O\&M}$ from the Equation 1 are constants.

For the capital cost, the module costing technique (MCT) was applied (Toffolo *et al.*, 2014). Cost-functions from Turton *et al. (*2008) were used for all components besides generators (Lemmens & Sanne, 2016) and compressor (Calise *et al.*, 2007). The separator was assessed with an assumption from Garrett (2012). Multiplication over 1.18 was applied to receive the final total cost; it corresponds to the system mounting, including the cost of intermediate heat-transfer loop (Turton *et al.*, 2008). To bring all values to current prices in the same currency ($ 2017 year), we used the official open historical data for CEPCI ($\mathrm{CEPCI_{2017}}$=567.5) (*Chemical Engineering*, 2019) and currency exchange rate (*Free Online Exchange Rates Conversion Calculator*, 2019*)*. More details may be found in Abrosimov *et al.* (2019).

3.3. Optimisation details

Constraints of optimisation are the same both for the case of efficiency and economic optimisation. The constraints originate from the feasibility of heat exchangers (pinches), the operational stability of the ORC cycle (temperature gap to critical point), and the stability of the fluids (oil temperature, superheating). The maximum oil temperature was taken from the product documentation for the thermal-oil (Therminol VP-1). Table 2 contains a list of constraints and their values.

[1] Capacity factor – a unitless ratio of the generated power output to the maximum possible output (evaluated at 8760 hours of operation for one year period)

The standard Aspen Hysys optimiser with BOX algorithm provided the tolerance of optimisation 2e-7. "Maximum change per iteration" was varied in the range 0.3–3 for obtaining global extrema with sufficient calculation speed and accuracy.

Table 2: Constraints of the optimisation.

Constraint	Value
Minimum pinch of HE-1, ^oC	30
Minimum pinch of evaporator, ^oC	15
Minimum pinch of condenser, ^oC	5
Minimum pinch of regenerator, ^oC	15
Minimum pinch of cooler, ^oC	15
Minimum temperature gap to critical point, ^oC	15
Maximum oil temperature, ^oC	380
Maximum superheating, ^oC	40

3.4. Pareto front building algorithm

Exploiting the Pareto front technique is a quite common methodology for the assessment of energy facilities (e.g., Imran *et al.*, 2019 or Astolfi *et al.*, 2017). The Pareto fronts were constructed with the following methodology. The scheme was set on the maximum efficiency; then, with a constraint on the minimum system efficiency, the scheme was LCOE-optimised. In several steps of the minimum efficiency limit, the parameters came to the LCOE-optimal set. An additional piece of the curve on the left, not belonging to the Pareto front as such according to its definition, to say, non-Pareto part of the graph, was received, adjusting system parameters by the LCOE optimisation with a constraint on the maximum system efficiency. In that case, the initial set of system parameters for the further optimisation corresponded to the state of high LCOE and low system efficiency (lower than a value in the LCOE-optimal point). A usual weighted approach for the Pareto front formation was not productive with the standard Aspen Hysys optimiser used.

3.5. System and cycle efficiency introduction

This study orients on the final results of the system performance in the available heat exploiting, so the main parameter of efficiency assessment is net system efficiency:

$$\eta_{sys} = \frac{P_{net}}{Q_{av}} = \frac{P_{gross}*\eta_{gen} - P_{cooling} - P_{pump}}{(H_{in}-H_{amb})*\dot{m}_g} \tag{2}$$

P_{gross} corresponds to the sum of shaft powers of the turbine-compressor unit and the ORC expander. $P_{cooling}$ is a sum of electrical power spent on air cooling of the condenser and cooler (Scheme Description).

4. RESULTS AND DISCUSSION

4.1. Influence of ambient temperature on the system efficiency

Technical systems may be optimised for the different conditions of operation depending on the average climatic parameters of the territory of application. A study on the deviation of the best reachable performance under the changing reference ambient temperature was performed for the scheme investigated in this work. The temperature was varied in the range between 5 and 35 ^oC.

4.1.1 Effect of water condensation on the IBC performance

Being presented before the discussion of the overall scheme results, an overview of the IBC part performance may help to explain some of their specificities. In Figure 2, net power generated by IBC part of the scheme is shown together with water condensation rate in the cooler before the compressor as functions of the turbine pressure ratio for different ambient temperatures. The ambient temperature defines the temperature of the gas before the compressor as the ambient air is used as the coolant for the gas flow in the Cooler (see Figure 1). In studied ranges of minimal pressure and temperature in the gas duct (point 4 of Figure 1), the system is balancing on the edge of water condensation, which starts to occur with the ambient temperature below 30 °C (temperature of the gas in point 4 equals to 45 °C). For the ambient temperature below 15 °C, condensation takes place at all pressures of the studied range. Condensation causes changes in the shape of optimality curve for the IBC, which for the intermediate zone (see lines for 15, 20 and 25 °C) has a convex-concave profile and even has a local minimum for 20 °C. In the previous work (Abrosimov *et al.,* 2019), a similar graph was presented but for all three temperatures of the exhaust: 470, 520, and 570 °C with the ambient temperature being equal to 20 °C. These graphs have a similar shape as the one obtained for 20 °C ambient temperature in Figure 2 but with different ratio between local maximums. For 570 °C, the right one in the zone of the high pressure ratios overcomes the left one, while for other curves, it is the opposite. That complex behaviour of the IBC part of the scheme is the reason for the complex curvatures of the graphs presented in the next paragraphs.

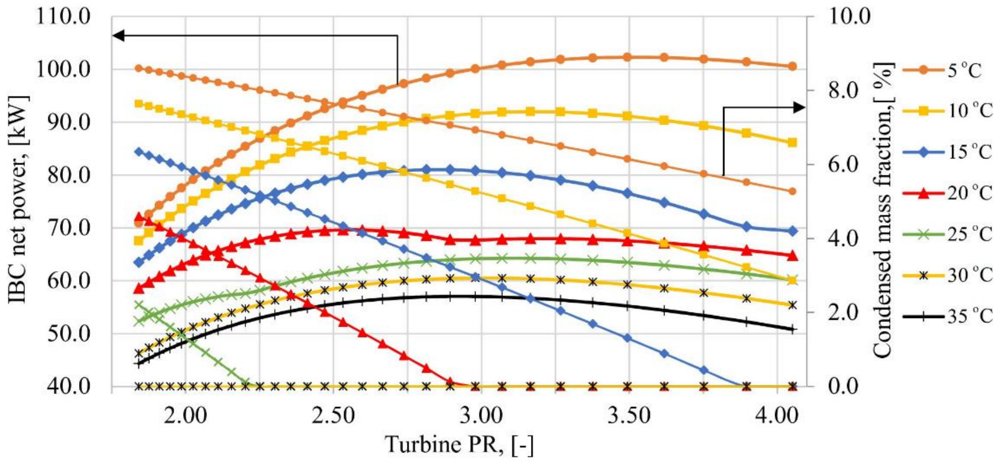

Figure 2: IBC power and water condensation rate before the compressor for different T_{amb}

4.1.2. Power and system efficiency

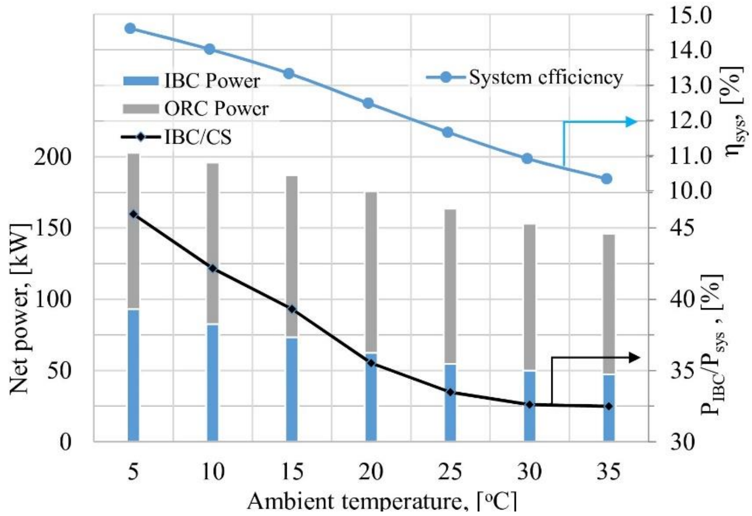

Figure 3: Net power and electric efficiency of the system for different of the ambient temperatures

Figure 3 shows changes in the system performance under different ambient temperatures. The gas temperature of the heat source has a nominal value and equals to 520 °C. For each temperature, the system was optimised for the maximum system efficiency. With the increase of ambient temperature in the range from 5 to 35 °C, the efficiency of the system goes down from 14.6 % to 10.3 %, as both parts of the cycle demonstrate worse performance under higher ambient temperature. However, this decrease is not proportional; the share of IBC part is depreciating as it may be seen in Figure 3 at the black curve corresponding to the right vertical lower axis and on the bar charts of the system net power linked with the left vertical axis. From the 45 % of the total generated power with 5 °C ambient temperature, the IBC contribution decreases up to 32.5 % for the 35 °C. The changes of the curve slope are connected with condensation issues mentioned above. It should also be noticed that the decrease of system efficiency is slightly non-linear as it may be seen on the upper curve of Figure 3 due to the same cause.

4.2. Pareto fronts for system efficiency and LCOE

Separate optimisation of such contradictory objective functions as efficiency and LCOE does not provide clearness for the practical choice of system parameters. That is why, here, the Pareto front graph is presented for these two objectives at three temperatures of exhaust: 470, 520, and 570 °C, the ambient temperature being equal to 20 °C, and CF=0.9. Figure 4 shows Pareto front curves together with plots of optimal turbine pressure ratio (right vertical axis). The Pareto fronts as such are between semi-transparent marks on the curves. The left orange one corresponds to the minimum LCOE point and the green right mark – to the maximum efficiency. The lines going left from the orange mark are not on the Pareto front as such but may have interest for some practical cases mentioned below. For the 570 °C curve, this line is flat with an accuracy of ±0.06 %; hence, it may be considered as a part of the Pareto front. However, principally, it has the same meaning for the practical analysis as the conforming segments of other curves. Pareto fronts have a similar shape with one relatively sharp change of the slope. This zone can be recommended to the potential decision-maker as the most suitable in the efficiency-LCOE trade-off. The uneven shape of the optimal pressure ratio lines is connected with the trends of power generation of IBC under different pressure ratios presented in previous paragraphs. Bends of the IBC power curves (Figure 2) correspond to the curvatures on the optimal IBC pressure ratio lines of Figure 4.

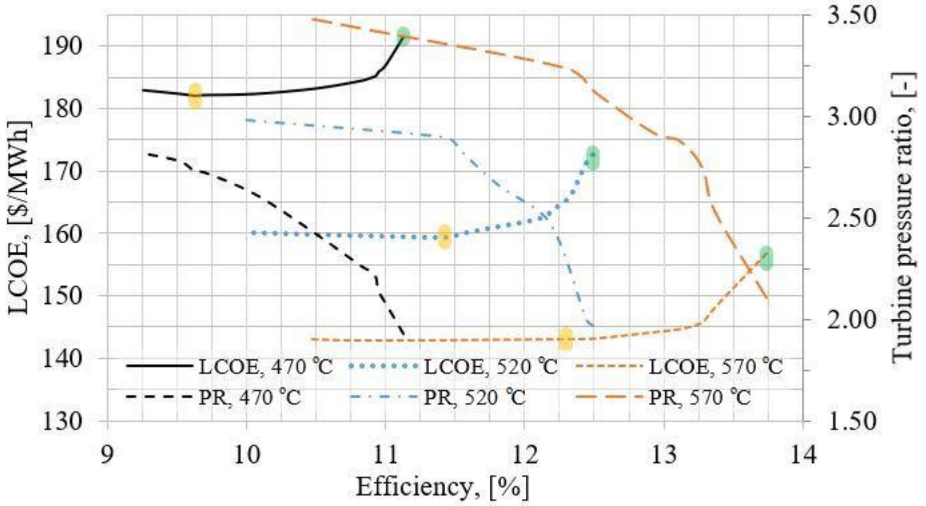

Figure 4: Pareto front for LCOE-efficiency pair and corresponding optimal pressure ratio of the IBC turbine. T_{amb}=20 °C

In Figure 5, together with Pareto fronts similar with Figure 4, the scheme capital cost corresponding to the points of the fronts is presented on the right axis instead of the turbine pressure ratio. Capital cost may provide an orienteer for the potential investors, especially in case of the investment in one tranche. If assets are limited, the prospects of the investments still may be assessed positively, and a point of the segment on the left from Pareto front may be chosen.

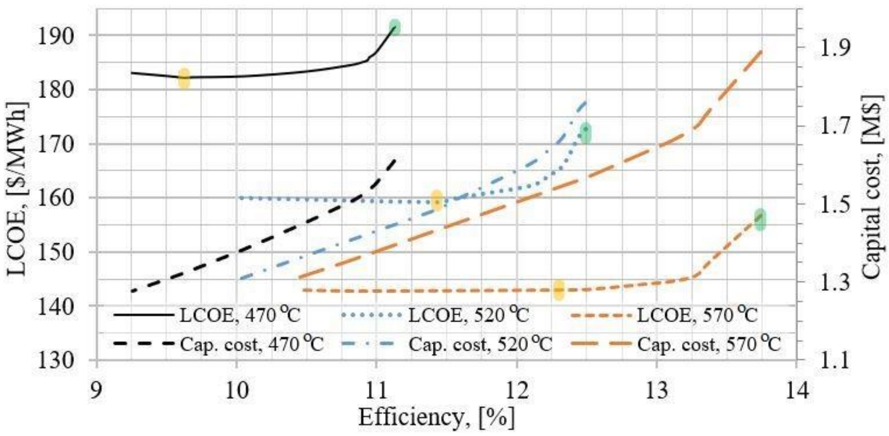

Figure 5: Pareto front for the LCOE-efficiency pair and corresponding capital cost of the system. T_{amb}=20 ºC.

5. CONCLUSION

In this work, the performance of an authentic scheme for waste heat recovery (WHR) based on the combined inverted Brayton - organic Rankine cycle (IBC-ORC) has been thoroughly investigated from the techno-economic point of view in the assumption of operation in design conditions. First, the scheme has been modelled under different reference ambient temperatures, showing a 30 % reduction of electric system efficiency (from 14.6% to 10.3%) with an increase of ambient temperature from 5 to 35 ºC. The ambient temperature affects the system because of the usage of air as the system coolant. Furthermore, the dependence is non-linear due to the effect of water condensation, occurring in the zone of minimal temperature and pressure in the duct of the inverted Brayton cycle in the studied range of these parameters. For a number of sets of operational conditions, this effect causes a non-typical two-maxima profile of the plot for the correlation between IBC produced power and pressure ratio in the IBC turbine. The optimal distribution of the IBC and ORC generated power also changes with the ambient temperature, starting from 47 % share of IBC at 5 ºC, ending up with 32.5 % for 35 ºC, as with lowering of this parameter, IBC improves performance more intensively than ORC. Besides that, the Pareto fronts identifying the trade-off between electric efficiency and levelized cost of energy (LCOE) under three temperatures of the incoming exhaust gas (470, 520, 570 ºC) have been plotted for the proposed system assumed to be used with the capacity factor equal to 0.9. The curves of the fronts have a bend where the LCOE starts to increase rapidly with a small growth in the efficiency. The zone in the adjacency of this bend should be recommended for the majority of the investment cases. For instance, for the 520 ºC exhaust temperature, the recommended configuration has the system electric efficiency around 12.15 % with the LCOE equal to 162.4 $/MWh and 1.6 M$ capital cost, whereas maximal efficiency equals to 12.49 %, and minimal achievable LCOE is 159.5 $/MWh. However, in the case of primer capital investment shortage, points which are not on the Pareto fronts may be chosen as they correspond to system configurations with lower capital cost than points of the fronts. For example, for the same temperature, the LCOE may be equal to 160 $/MWh with 10 % efficiency when the capital cost is equal to 1.3 M$. As future steps, off-design simulations will contribute to the further development of the technology. In particular, the performance of the system during a year based on real climatic data may give a more accurate economic assessment. Besides, the estimation of the effect on the primary engine and the combined heat and power (CHP) mode assessment may demonstrate other prospects of the combined IBC-ORC scheme.

NOMENCLATURE

η	efficiency	%
H	enthalpy	kJ/kg
$\dot{m}$	mass flow	kg/s
P	power	kW
T	temperature	^{o}C
Q	heat flow	kW

Subscripts
amb - ambient
av - available
g - gas
gen - generator
in – inlet
sys – system

Abbreviations
BC – Brayton cycle
CF – capacity factor
CS – combined scheme
HE – heat exchanger
IBC – inverted Brayton cycle
LCOE – levelized cost of energy
MCT – module costing technique
ORC – organic Rankine cycle
PR - pressure ratio
TOL – thermal-oil loop
WHR – waste heat recovery

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