Design considerations on a small scale supercritical CO_2 power system for industrial high temperature waste heat to power recovery applications

Giuseppe Bianchi[a*], Savvas A. Tassou[a], Yunting Ge[a], Hussam Jouhara[a], Konstantinos Tsamos[a], Arthur Leroux[b], Maxence de Miol[b]

[a]Centre for Sustainable Energy Use in Food Chains,
Brunel University London, UB83PH Uxbridge, United Kingdom

[b]Enogia SAS, 13015 Marseille, France

[*]e-mail: giuseppe.bianchi@brunel.ac.uk

Abstract

Existing industrial processes are energy intensive environments with a multitude of waste heat streams at different temperature levels whose recovery would undoubtedly contribute to the enhancement of the sustainability of the industrial sites and their products. In particular, the geographical distribution and size of the available waste heat potential is very widespread with most sources being small to medium size, up to 1 MW and fewer of larger size above 1 MW.

Among the waste heat to power conversion approaches, the usage of bottoming thermodynamic cycles based on carbon dioxide in supercritical phase (sCO_2) provides significant advantages compared to more conventional technologies such as the Organic Rankine Cycle (ORC) systems that are nowadays commercially available even at small scale (~10 kWe). However, unlike the large sCO_2 systems that are already on the market (~ MWe), medium and small size ones are still under development.

In the current work, the challenges involved in the design of a small scale sCO_2 system are considered through thermodynamic analysis (1st and 2nd law) as well as by a preliminary turbomachinery design based on the similarity approach. With reference to a simple regenerative cycle architecture, the study provides details on the design parameters and performance trends as well as on the operational and manufacturing constraints that the compressor imposes to the theoretical values.

Introduction

Existing industrial processes are currently responsible for almost the 30% of worldwide primary energy consumptions (2827 Mtoe/yr) [1]. Furthermore, as shown in Figure 1, most of the energy sources that drive the industrial sector are fossil fuel based. Hence, in order to reduce the carbon footprint related to industry and their products, energy efficiency, saving and recovery are the interventions that must be carried out such that international environmental policies will be fulfilled and a more sustainable economy will be pursued.

In particular, waste heat recovery is the process of capturing heat from waste streams of existing industrial process and using this heat directly, upgrading it to a more useful temperature, and/or converting it to electrical power or cooling. Waste heat recovery systems can offer significant energy savings and substantial greenhouse gas emission reductions but for these to materialise technological improvements and innovations should take place aimed at improving the energy efficiency of heat recovery equipment and reducing installation costs. The application potential of waste heat recovery in nowadays industry is remarkable; for instance, in the countries belonging to the European Union (Figure 2), the waste heat recovery potential has been estimated in 370 TWh/yr [2]. However, the technical and economic feasibility of these systems highly depends on the temperature level at which the waste heat is rejected from the industrial process.

Conventional waste heat to power conversion systems are based on Organic Rankine Cycles (ORC) whose working fluids are not suitable for heat sources at high temperature (>450°C). Furthermore, existing ORC systems have a relatively low efficiency at the lower power range increasing the need for alternative technologies that can operate both at higher temperatures and efficiency [3]. Other drawbacks of ORC systems overcome by bottoming thermodynamic cycles using CO_2 in supercritical

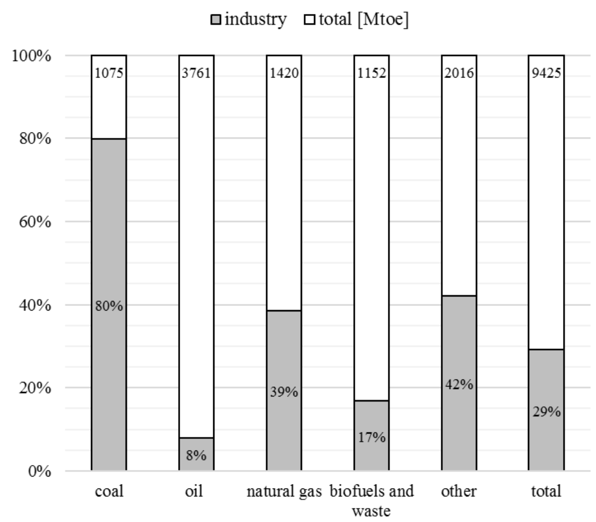

Figure 1 – Share of industry in total final consumption by fuel [1]

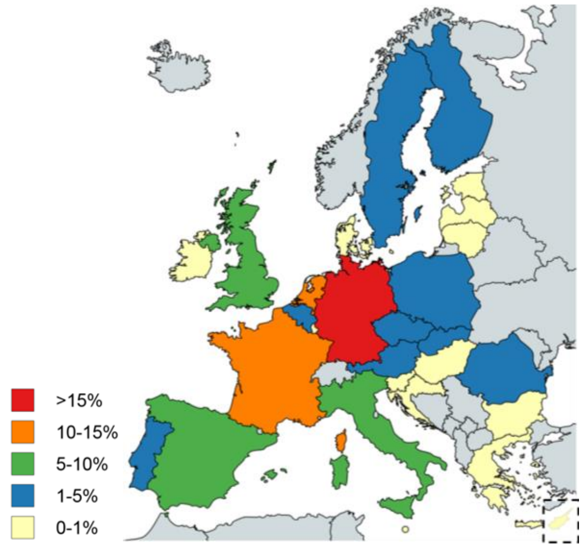

Figure 2 – Waste heat recovery potential in EU28 (370 TWh/yr) [2]

phase are high exergy losses during the phase changes of the organic working fluid as well as issues in environmental indicators (GWP, ODP, ALT), safety and cost of the working fluids.

In the literature, there is a large body of research concerned with cycle analysis as well as thermo-economic optimization of sCO_2 systems, especially for nuclear or concentrated solar power applications. In these studies, energetic and exergetic cycle calculations are performed with reference to multiple plant layouts in order to assess the benefits of multi-stage compression and expansion with and without intercooling or regeneration. In particular, it is concluded that sCO_2 recompressing cycle can achieve the maximum performance compared to other layouts since it reduces the waste heat and increases the recuperated heat by splitting some portion of the flow for a recompressing process to ultimately increase the thermodynamic efficiency [4]. However, if the compressor efficiency is low and maintaining the main compressor inlet condition near the critical point is difficult, the recompression cycle may not be the best choice [5] and a simple recuperated cycle may be preferable.

All the considerations above refer to large scale sCO_2 systems (~MWe), whereas the economies of scale and the greater power production would lead to reasonable payback period despite the complexity of the sCO_2 unit. On the other hand, sCO_2 based small scale applications lack of the technological know-how that is preventing them to be marketable. For these reasons, one of the goals of the H2020 I-ThERM project is the design and development of a 50-100 kWe demonstration unit for high temperature industrial waste heat recovery applications.

In the current work, the thermodynamic design procedure of the small-scale sCO_2 system that will be further developed and tested in later stages of the project is presented. With reference to a simple regenerative Brayton cycle, energy and exergy analyses are coupled with a preliminary turbomachinery design based on the non-dimensional charts that resulted from the similarity theory ultimately summarized by Balje [6]. In particular, the effects of main cycle design parameters are discussed with reference to performance as well as turbomachinery features such as revolution speed and impeller diameter. These quantities, especially the ones related to the compressor, are actually constraints to the performances outlined by the thermodynamics that do not occur so severely in larger scale systems.

Cycle modelling

In small scale applications, the adverse economies of scale necessarily call to minimize the capital expenditure in order to achieve an overall economic feasibility of the recovery solution. Furthermore, as outlined by previous authors, the advantages achievable with a recompressing cycle layout become effective provided that the compressor efficiency experimentally achieved is close to the value considered in the thermodynamic design. Due to the challenges that are going to be discussed in the

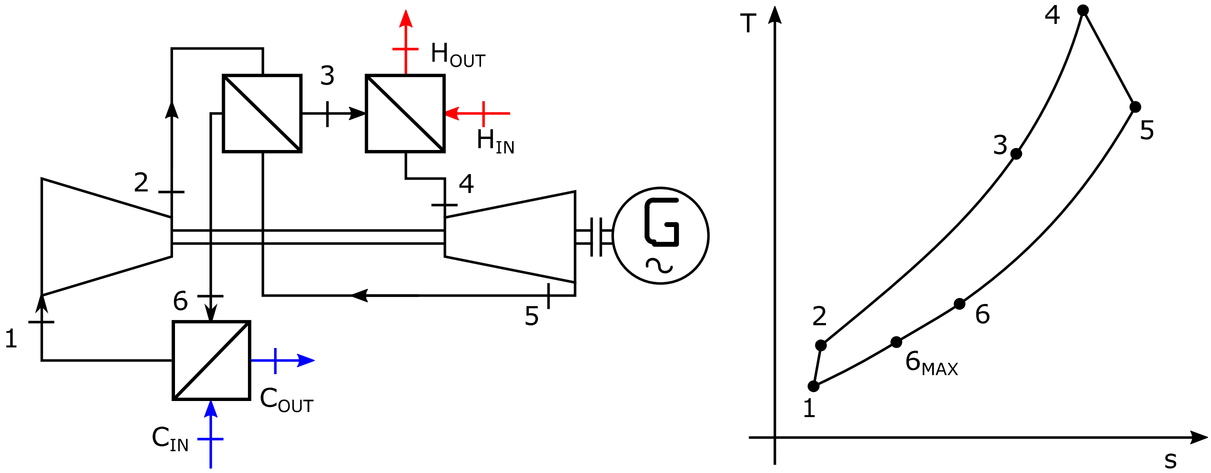

Figure 3 – Simple regenerative Brayton sCO_2 loop and its corresponding entropy diagram

following sections of the paper, this latter assumption might be hardly verified in the testing phase of the sCO_2 system. For these reasons, the cycle layout that was considered is a simple regenerative one. A similar architecture, is also used by the Institute of Applied Energy [7]. However, their configuration employs two regenerative heat exchangers. Despite this configuration allows a better matching of the temperature profiles along the heat exchangers, a similar choice would have increased the investment cost of the sCO_2 system with the risk to ruin the overall feasibility of the waste heat recovery solution.

With reference to Figure 3, that shows the plant scheme and the entropy diagram of the sCO_2 system assuming no pressure and heat losses at the interconnecting pipes, the working fluid is compressed (1-2), undergoes to a heat gain (2-4) to be eventually expanded (4-5) and cooled back to the initial conditions (5-1). The regenerative heat transfer process occurs from the turbine outlet on the hot side (5-6) and from the compressor outlet with respect to the cold side of the heat exchanger (2-3). The heat recovery process takes place in the heater (3-4) while the heat rejection to occurs in the cooler (6-1). In ideal terms, compression and expansion are adiabatic-isentropic transformations while the heat transfer process are isobaric ones. In particular, an ideal regeneration would lead to an outlet temperature on the hot side equal to the inlet temperature of the cold side ($T_{6_{MAX}} = T_2$).

The thermodynamic design of the sCO_2 unit was performed through a steady state model set up in Engineering Equation Solver (EES). The model takes into account heat and pressure losses along the pipes that connect the components of the supercritical system. Heat exchanger effectiveness as well as turbine and compressor efficiencies (isentropic, mechanical and electrical) are additional constant input parameters of the model. The presence of the regenerative heat exchanger could also be investigated by varying the recuperation rate, defined in Eqn. 1 as the ratio between real and ideal enthalpy drops on the hot side of the device.

$$R = \frac{h_5 - h_6}{h_5 - h_{6MAX}} \tag{1}$$

Thermo-physical properties of carbon dioxide are calculated using the fluid library of EES while constant values were considered for specific heats at constant pressure of hot (exhaust gas) and cold (water) sources.

Using the input data, isentropic relationships and energy balances across the heat exchangers, the model calculates pressure and temperature values in all the key points in the cycle, before and after each component. Exergy calculations are performed according to reference [8] while 1st and 2nd laws cycle efficiencies are calculated according to Eqns. (2) and (3), being $\dot{W}_{el,net}$ the electrical net power output, $\dot{Q}_{hot}$ the thermal powered recovered from the hot source and I the irreversibility associated to each component and to all the interconnecting pipes.

$$\eta_I = \frac{\dot{W}_{el,net}}{\dot{Q}_{hot}} \qquad (2) \qquad\qquad \eta_{II} = \frac{\dot{W}_{el,net}}{\dot{W}_{el,net} + \sum_k I_k} \qquad (3)$$

Turbomachinery design

The cycle analysis provides some main design specifics for the design of compressor and turbine, such as pressure ratio and operating temperatures. However, before proceeding to the mean line design, the knowledge of revolution speed and impeller diameter are fundamental parameters to be taken into account in radial machines. Although these values could be specified based on experience or industrial know-how, a common practice is to make use of the similarity relationships that may be present between existing turbomachines and the ones that are going to be designed. This approach has been largely used in the sCO_2 field [9-12] and herein briefly exposed.
The Balje's charts summarize total to static efficiencies of turbines and compressors with respect to non-dimensional parameters, namely specific speed (N_s) and specific diameter (D_s) [6]. Their expressions, proposed in Eqns. 4 and 5 respectively, allow to estimate reasonable values for revolution speed (N) and wheel diameter (D) since adiabatic head (H_{ad}) and flow rate (Q) are known from the cycle analysis.

$$N_s = \frac{N\, Q^{1/2}}{H_{ad}^{3/4}} \qquad (4) \qquad\qquad D_s = \frac{D\, H_{ad}^{1/4}}{Q^{1/2}} \qquad (5)$$

Moreover, for a given turbomachinery technology in the Balje's charts, it is possible to draw a line that, for a given specific speed, provides a value of specific diameter that ensures the maximum efficiency. These lines, called Cordier's lines, were graphically retrieved from the Balje's charts for the specific speed ranges strictly related to radial machines and taken into account in the sCO_2 model using Eqns. 6 and 7 for compressor and turbine respectively.

$$D_{S,comp} = 2.719\, N_{S,comp}^{-1.092} \qquad (6) \qquad\qquad D_{S,turb} = 2.056\, N_{S,turb}^{-0.812} \qquad (7)$$

Hence, being the turbine the most affecting component for the actual energy recovery process, for a given value of its specific speed the model calculates the revolution speed of the turbine from Eqn. 4 and its diameter using Eqns. 7 and 5. On the other hand, since the configuration of the compressor-turbine-generator (CGT) will be a single shaft one, knowing the revolution speed from the turbine calculations allows to compute the compressor specific speed, specific diameter and impeller diameter using Eqns. 4, 6 and 5 respectively.

Results and discussion

Due to a limited amount of equations, in the cycle analysis of a sCO_2 system, a significant amount of design choices need to be performed in order to limit the investigation to a small number of design configurations. In particular, constant input data for these calculations were heat exchangers effectiveness, turbine and compressor isentropic efficiencies, mechanical and electrical efficiencies as well as pressure drops in pipes and heat exchangers, and thermal losses in pipes. All these pieces of information can be retrieved either from experience or with reference to literature values, such as the ones provided in reference [5]. Furthermore, input data related to the hot and cold sources coupled

with the sCO_2 system are additional boundary conditions to the cycle analysis since they define the temperature ranges between which the waste heat recovery unit will operate.
Hence, for a given design and cycle layout configuration, a number of design parameters can be investigated and optimised. In the current work, the effects of cycle pressure ratio and turbine inlet temperature (TIT) are reported in Figure 4 while the influence of recuperation rate and inlet temperature of the hot source are summarised in Figure 5. Both the sets of results were calculated neglecting thermal losses in pipes as well as pressure drops in heat exchangers and pipes.
In Figure 4 the analysis was carried out with reference to an exhaust gas mass flow rate of 1 kg/s at 700 °C and assuming a constant temperature difference between exhaust gas and CO_2 outlets. Hence, the energy balance at the heater respects Eqn. 8.

$$\dot{m}_h c_{p_h} (T_{H_{IN}} - T_{H_{OUT}}) \varepsilon_{htr} = \dot{m}_{CO_2} (h_4 - h_3) \quad (8)$$

With reference to Eqn. 8 and within the above mentioned assumptions, for a given cycle pressure ratio (β) a higher turbine inlet temperature (T_4) leads to a lower amount of CO_2 mass flow rate in the supercritical loop. On the other hand, for a given turbine inlet temperature, a higher cycle pressure ratio leads to a lower turbine outlet temperature and, in turn, to a lower potential for regeneration. Therefore, at the sCO_2 heater, more thermal power is exchanged using a higher CO_2 mass flow rate.
For a given cycle configuration, the net power output depends on the specific net work and the amount of working fluid mass flow rate. This fact explains the net electrical power trend that is shown in the top left chart of Figure 4: even though cycle pressure ratio and turbine inlet temperature both enhance energy efficiency of the cycle and its net specific work, because of a lower amount of working fluid that is needed to balance the heat loads at the heater, the net power output decreases at high values of turbine inlet temperature. For instance, with a pressure ratio of 2 and a TIT equal to 400 °C, theoretical electrical power output would be over 70 kW with a 1st law efficiency of 23%. At 500 °C the cycle efficiency would rise to 26% but the net power output would drop to 20 kWe.
Mass flow rate resulting from the energy balance at the heater and cycle pressure ratio affect revolution speed and diameter of the turbomachinery according to Eqns. 4 and 5. In particular, for a given flow rate, the compressor and turbine will rotate faster and be smaller with increasing pressure ratio. On the other hand, for a given enthalpy drop/rise, smaller flow rates will reduce the size of the machine increasing its revolution speed. Speed and size, however, are constrained by technological limitations in the bearings and in the manufacturing process respectively. Furthermore, in small machines useful flow passages tend to have the same dimensions of leakage paths. Therefore, efficiency values largely accepted and achievable for MW-scale machines can be hardly assumed in kW-scale ones. In particular, for a given revolution speed of the CGT, due to the high density of the CO_2, it's the compressor the most limiting machine in terms of size. Figure 4 provides some figures related to compressor wheel size and speed showing that cycle configurations with high efficiency demand small and fast machines whose specifics exceed the operational constraints mentioned above. Reasonable thresholds for cost effective turbomachinery are a revolution speed lower than 100,000 RPM and wheel diameter greater than 40 mm.
Hence, the application of turbomachinery constraints on the thermodynamic design, limit the design configuration to a maximum cycle efficiency of 22% and a net output power range between 30 and 70 kW.
In Figure 5 the influence of recuperation and inlet temperature of the exhaust gas is assessed with reference to a given thermodynamic cycle, i.e. temperature and pressure in the cycle points did not change during the analysis as well as its net power output. On the other hand, a larger recuperation would not only increase the energy efficiency of the cycle but also the exergy one due to a better overall utilization of the waste heat and lower irreversibility in the regenerative heat transfer process. At the same time, a higher inlet gas temperature would increase the amount of CO_2 mass flow rate and, in turn, the recuperator duty. Furthermore, for a given temperature rise on the CO_2 side of the

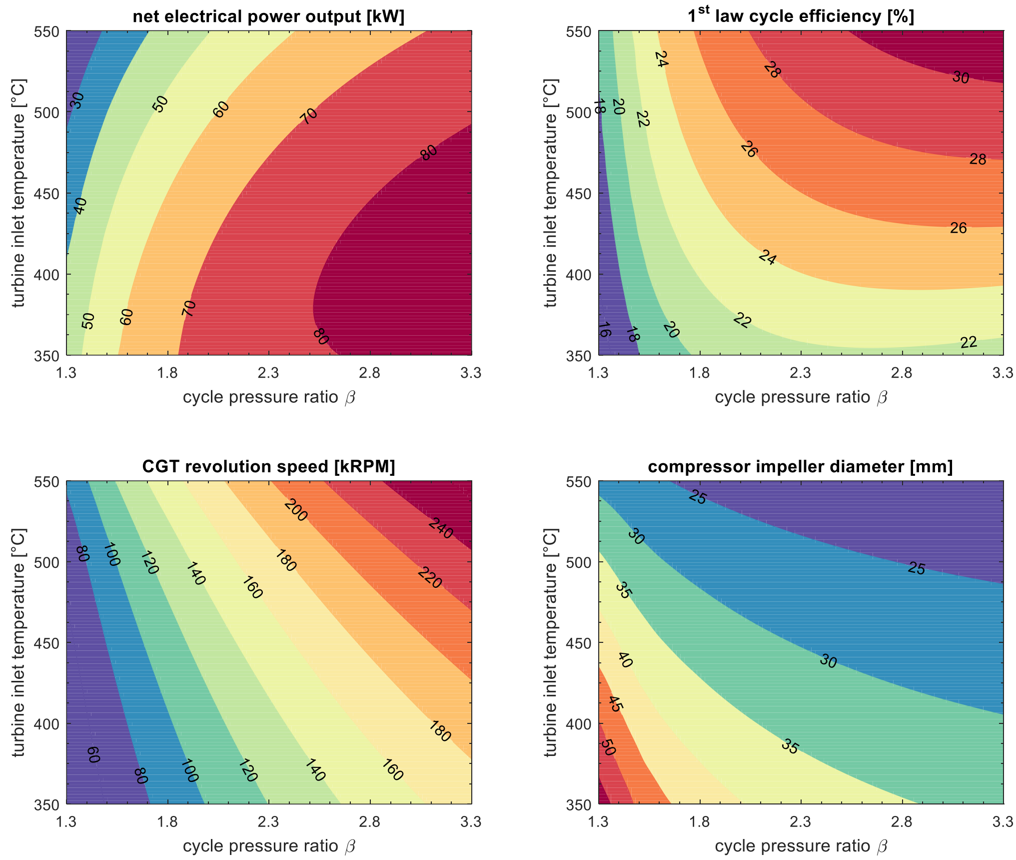

Figure 4 – Influence of cycle pressure ratio and turbine inlet temperature on cycle performance and turbomachinery speed and dimensions

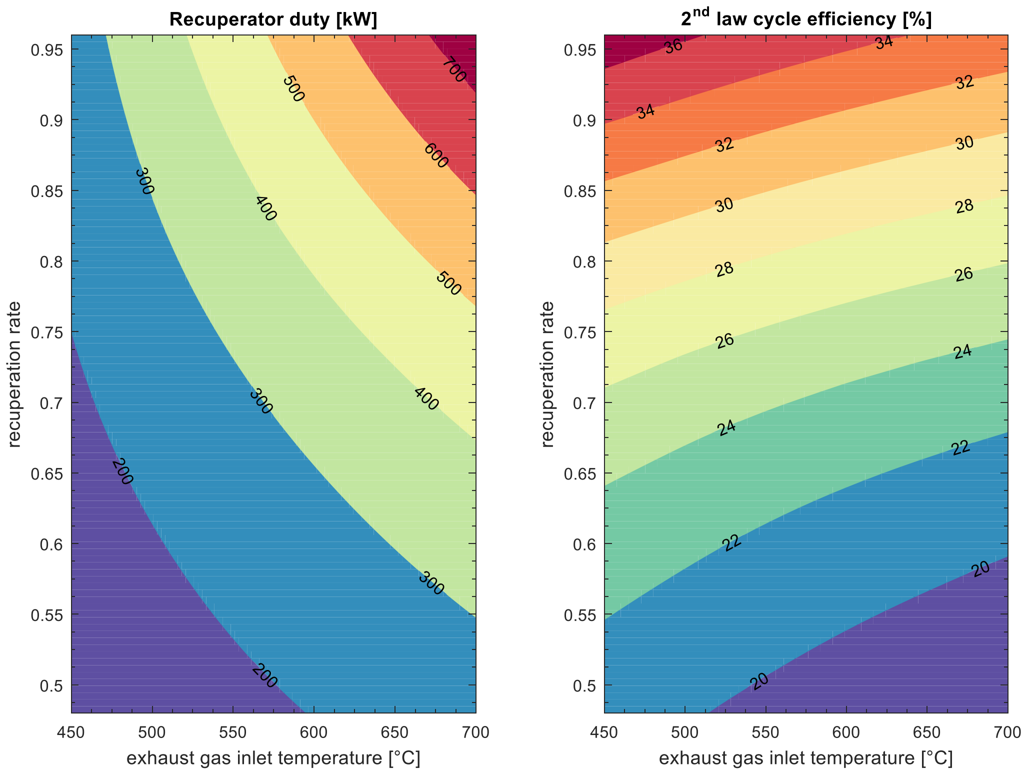

Figure 5 – Influence of recuperation rate and exhaust gas inlet temperature on recuperator and exergy performance of the sCO_2 system

heater, a higher inlet gas temperature would result in larger irreversibility in the heat recovery process due to a worse matching of the temperature profiles.

Conclusions

Bottoming thermodynamic cycles using supercritical carbon dioxide (sCO_2) provide remarkable advantages with respect to conventional waste heat to power approaches. Although large scale sCO_2 systems are already commercially available, in order to tackle the geographical spread of the waste heat recovery potential in industry, medium-small scale systems need to be developed. In this power range technical and economic challenges arise and constraint the design choices proposed by the thermodynamics. This paper assessed these issues presenting a preliminary design procedure for small scale sCO_2 systems based on 1st and 2nd law analysis and the similarity theory for turbomachinery. With reference to a simple regenerated cycle layout, parametric analysis showed that cycle efficiency and net power recovery are highly affected by cycle pressure ratio and turbine inlet temperature. Without any turbomachinery constraint, 1st law cycle efficiency would exceed 30 %. However, this theoretical estimation drops to 20% when impeller manufacturing or bearings limitations are taken into account, being the compressor the most restrictive machine. At the same time, the influence of recuperation on cycle energy and exergy efficiencies is remarkable even though one should be aware of the additional costs that a more aggressive heat exchanger design could provide with respect to the actual benefits gained from technical and economical perspectives.

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