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DESIGN OF A SINGLE-SHAFT COMPRESSOR, GENERATOR, TURBINE FOR SMALL-SCALE SUPERCRITICAL CO_2 SYSTEMS FOR WASTE HEAT TO POWER CONVERSION APPLICATIONS

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ABSTRACT

Waste heat to power conversion is a promising approach to reduce the carbon intensity in industry and manufactured goods. In this framework, bottoming thermodynamic cycles using supercritical carbon dioxide as working fluid (sCO_2) might be a suitable and efficient technology to consider especially for heat sources characterized by streams at high temperatures (>300°C). The compactness of sCO_2 turbomachinery is one of the advantages of sCO_2 systems over the conventional technologies; on the other hand, the reduced dimensions limits the bottom end of the power size achievable with such systems. The scarce amount of scientific and industrial literature for electrical power sizes between 50 and 100 kW further demonstrates this. The current research work summarizes the design procedure as well as the technical and technological challenges involved in the design of a single-shaft compressor, generator, turbine unit (CGT) for a sCO_2 system with a 50kWe nominal power output. First an overview of the High Temperature Heat To power Conversion facility (HT2C) under construction at Brunel University London will be presented. Then, highlights of the CGT design are presented in terms of structural and packaging aspects as well as with regards to the ancillary lubrication, drainage and cooling loops.

INTRODUCTION

In the last decades, the need of lowering the environmental impact as well as of optimizing industrial processes has driven research to innovative concepts and equipment with regards to three major areas: energy efficiency, energy saving and energy recovery. Out of these strategies, the energy recovery theoretically implies no changes in the existing energy system but rather aims at converting part of the energy that these systems currently reject to the environment in different forms.

With regards to thermal energy, latest reports state that most the waste processes involve heat sources with temperatures below 300°C [1]. However, the well-known Carnot efficiency states that the potential of a heat source depends on its exergy content and ultimately to its temperature. According to [1], high temperature industrial waste heat potential has been estimated as 3367 TWh worldwide. Using the same methodology, the authors analyzed the European scenario concluding that high temperature industrial waste heat potential amounts to 275 TWh, from nonmetallic minerals (glass, cement), nonferrous materials (aluminum) as well as iron and steel (the industrial sectors where most of the high-grade waste heat processes can be found).

Unlike direct use heat recovery systems, which may require severe modifications to the industrial process in order to use the thermal power recovered within the industrial site or export it over the fence, grid-connected heat to power conversion systems do not necessarily need a local heat user or a storage system. Also, recovery in electrical form can provide greater CO_2 emission savings than recovery in thermal form as for instance in UK, recovering 1 kWh of thermal energy saves 0.204 kg of CO_2 while if the 1 kWh was electrical the savings would be equal to 0.348 kgCO_2 [2].

Existing heat to power conversion systems have mostly addressed medium grade waste heat sources (100-300°C) through steam or Organic Rankine Cycle (ORC) systems. However, for high-grade heat sources, these approaches become less suitable due to the high energy losses involved with two-phase heat recovery or because of the degradation of the organic fluid properties currently employed in these systems. On the other hand, a Joule-Brayton cycle working with CO_2 in supercritical phase would allow to achieve better efficiencies than a steam Rankine Cycle at lower temperatures [3, 4] and at

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lower capital and operating costs. Operating with very dense fluids would indeed lead to compact equipment that would also require less maintenance [5]. Furthermore, the sCO_2 technology would allow a better thermal matching between the temperature glides of working fluid and heat source and thus would achieve a higher 2nd law (exergy) efficiency [6]. For these reasons, simplified or more complex configurations of sCO_2 power cycles are being considered for the next nuclear and fossil fuel power generation (500 to 1000 MWe) [7-9], modular nuclear power generation (300 MWe) [10–13], solar thermal power generation (10 to 100 MWe) [14–21], shipboard propulsion, geothermal, oxy-combustion (1 to 100 MWe) [22–31], and industrial scale waste heat recovery (1 to 10 MWe) [32,33]. The technical feasibility of such systems has been so far assessed mostly at theoretical level and it has been focused on cycle analyses of large power scale sCO_2 systems for power generation applications.

The research presented in this paper, on the other hand, addresses one of the most challenging areas of the sCO_2 heat to power conversion field by presenting the design of one of the first experimental sCO_2 facilities in Europe. The focus is on the design of a small-scale sCO_2 system in the power range around 50 kWe.

OVERVIEW OF THE HT2C FACILITY

The I-ThERM project, funded by the European commission within the h2020 research and innovation program, aims at demonstrating, among the other technologies, the feasibility of a plug&play sCO_2 system for high grade heat to power conversion applications. Accordingly, an industrial scale High Temperature Heat To power Conversion facility (HT2C) has been designed at Brunel University London taking into account not only research challenges but also industrialization aspects.

Unlike existing experimental sCO_2 facilities, whereas the heat input to the working fluid is provided through electrical resistance heaters, a novel feature of the current research is the investigation and development of direct recovery heat exchangers using flue gas as heat source. In order to do that, HT2C is equipped with a process air heater whose main features are listed in Table 1 and shown in Figure 1. Noteworthy ones are the high flexibility of operation as well as power size, which allows to test bottoming systems with power outputs in the order to tens of kilowatts.

Inlet conditions to the high temperature test section, where the sCO_2 system will be installed, are controlled with a proprietary system which relies on primary fan speed and flue gas flow rate as control signals while temperature downstream the process air heater and air flow rate are used for control feedback. A similar control architecture is repeated for the low temperature test section, whose provision is beyond the sCO_2 project.

As heat sink for the H2TC facility, a dry cooler system will be employed. Its features are listed in Table 2 and shown in Figure 2.

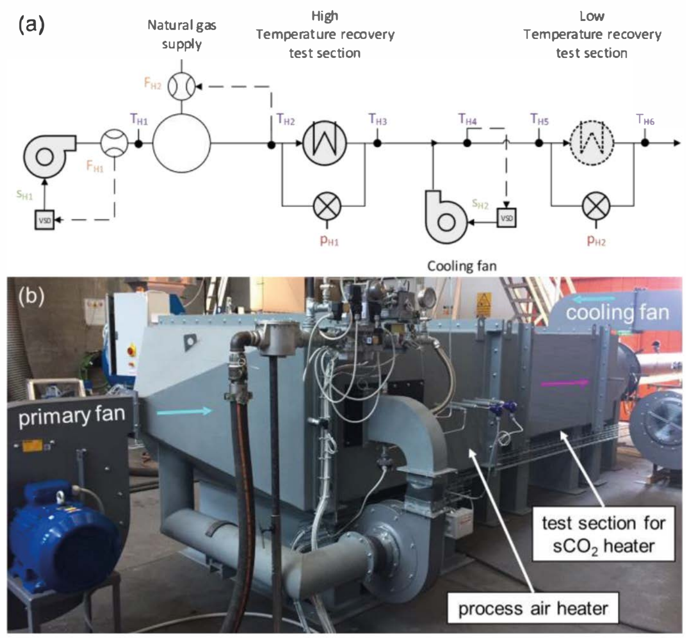

Figure 1: P&ID (a) and picture (b) of the heat source

Table 1: Heat source specifics.

Net max power supplied	830 kW
Maximum operating temperature	780 °C
Pressure drop allowed for primary heat exchanger	70 mbar
Fuel	Natural Gas (G20)
Gas input peak design	83.5 Nm³/h

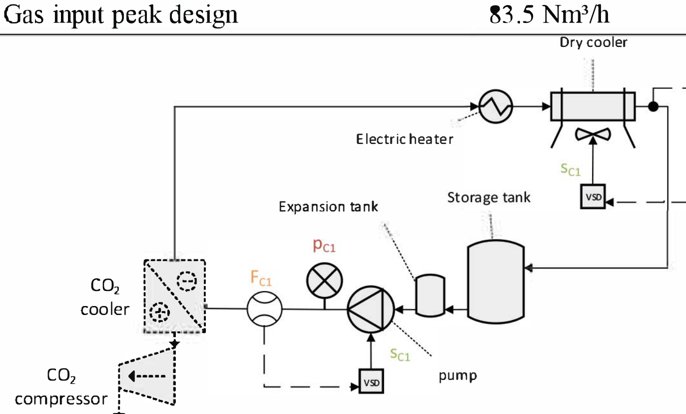

Figure 2: P&ID of the heat sink.

Table 2: Heat sink specifics.

Total cooling duty	500 kW
Water on temperature	60 °C
Water off temperature	20 °C
Mono-Ethylene glycol	25%
Total airflow at standard conditions	35.7 kg/s
Total fin and tube surface area	850 m²
Maximum fluid temperature	100 °C

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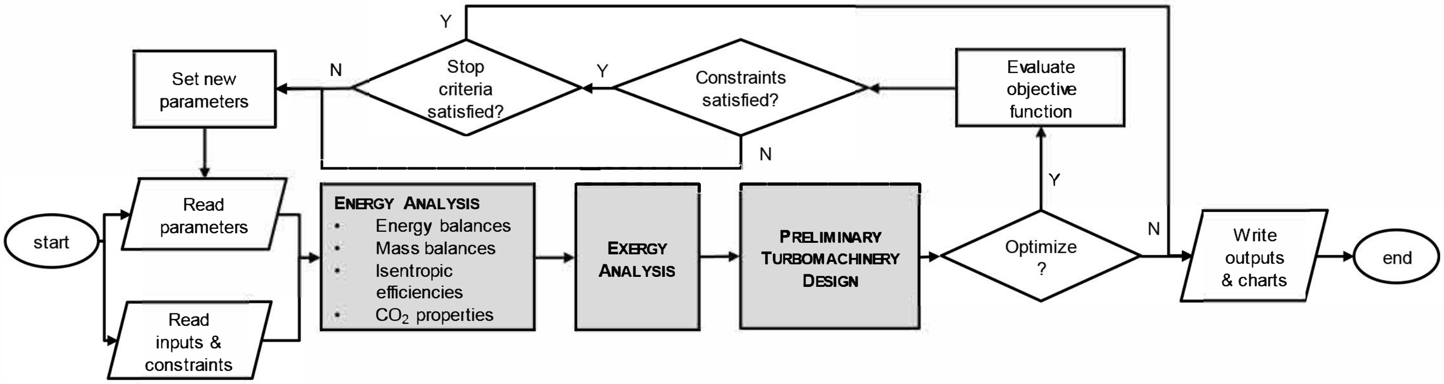

Figure 3: Thermodynamic design procedure

The dry cooler employs variable speed drives for the pump and the fans. A noteworthy feature of the chosen layout is the presence of an electric heater to warm-up the auxiliary fluid in the cooling loop to be used during the startup phase of the tests.

DESIGN OF THE SCO_2 SYSTEM

Within the scope of the I-ThERM project and the testing capabilities of the HT2C facility, the design of the bottoming sCO_2 system was bounded by several constraints that influenced its design. With reference to a theoretical study in which the authors compared several sCO_2 cycle layouts considering technical and economic figures, for the power range of interest it resulted that the most suitable system configuration for the current application was the simple regenerated layout [34]. This architecture requires the lowest number of components (3 heat exchangers, 1 compressor and 1 turbine) and therefore allows to minimize the investment cost of the heat to power conversion unit. In order to further lower the number of components, a single shaft architecture was considered for the Compressor-Generator-Turbine unit (CGT). As a consequence, the strict link between compressor and turbine performances leads to greater control, sealing and lubrication challenges.

As per Figure 3, which summarizes the design steps undertaken, the thermodynamic design of the sCO2 system has been carried out through a general design platform for simple regenerated layouts whose development has been presented in [35]. The in-house design code is based on energy and mass balances and it embeds turbomachinery design correlations based on the similarity analysis that allow to estimate speed and diameters of the impellers of the radial turbomachines.

The optimization libraries of the Engineering Equation Solver environment [36], in which the design platform has been developed, were used to identify the nominal operating conditions of the sCO_2 systems that are listed in Table 3.

These specifics were eventually used to design the heat exchangers as well as the turbomachinery. In particular, the

Table 3: Design point of the sCO_2 system.

min/max pressure [bar]	min/max Temperature [°C]	CO_2 mass flow rate [kg/s]
75/127.5	35/400	2.25

printed circuit technology was considered for the recuperator but not for the cooler due to recent developments in plate heat exchangers for refrigeration systems which have made this technology able to withstand high pressures at temperatures up to 250°C. This allowed a significant saving in the overall costs of the sCO_2 unit. The recuperator is shown in Figure 4. The CO_2 loop has additional connections to a CO_2 gas cooler to investigate the effects of a direct heat rejection at later stages of this research activity.

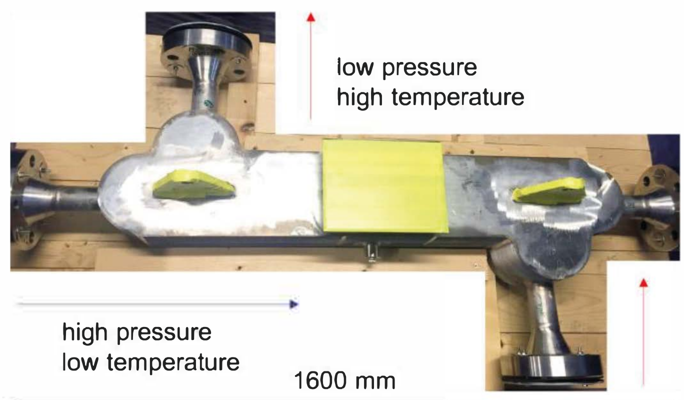

Figure 4: Printed circuit heat exchanger used as recuperator

The CO_2 heater had to fulfill requirements of compactness, low pressure drop on both sides and relatively low cost. These design trade-offs are expected to be achieved using a novel micro-tube heat exchanger technology which can easily be adapt to many high temperature heat to power generation applications.

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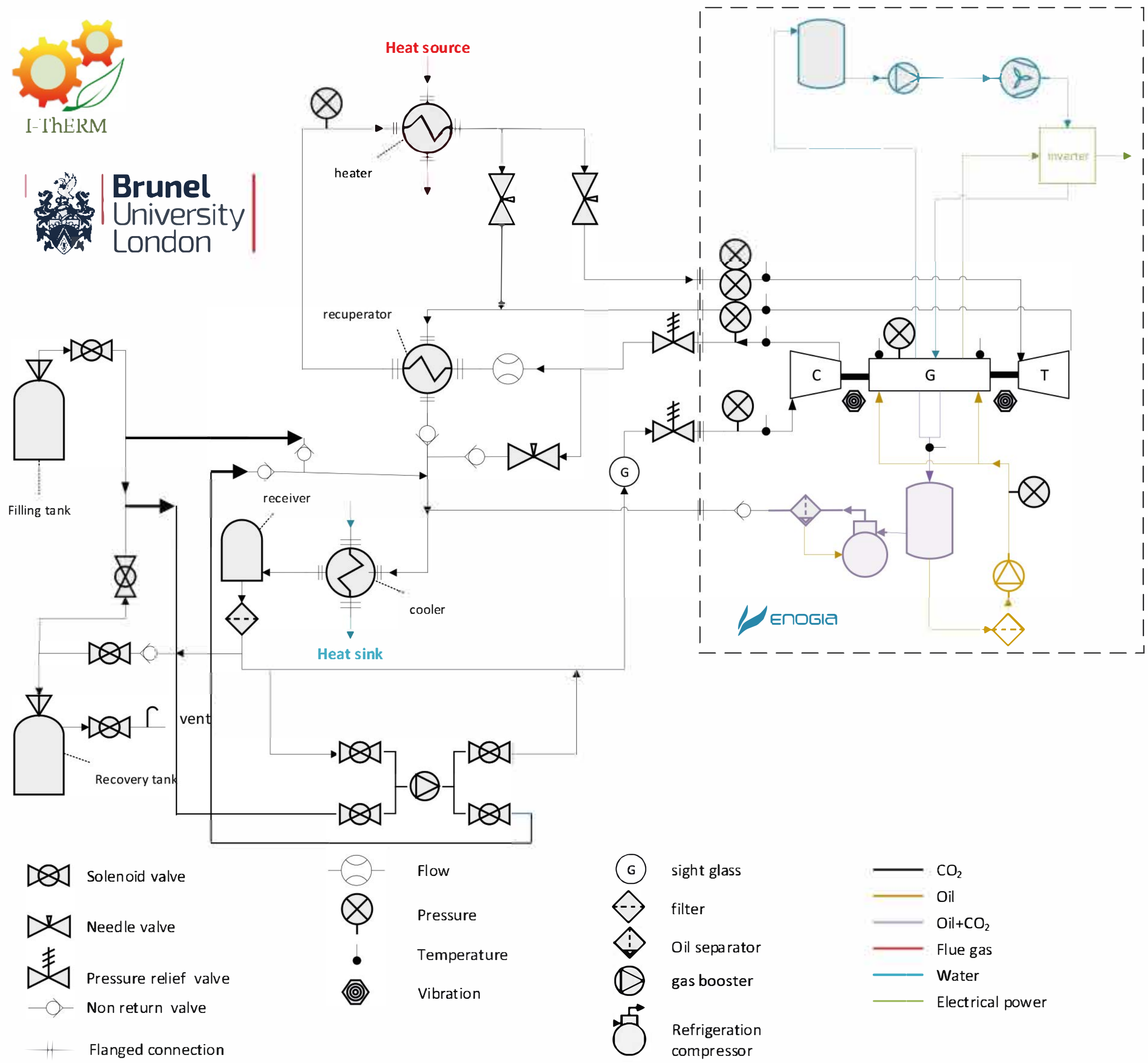

Figure 5: Process & Instrumentation Diagram.

To conceive the process and instrumentation diagram (P&ID) reported in Figure 5, startup, transient operation and shutdown modes had to be considered. In this step, precious pieces of scientific literature were the references [37] and [38], in which the research group of the US Naval Nuclear Laboratory shared their expertise about their testing activities on a 100 kWe two shaft recuperated closed sCO_2 Brayton cycle. In order to support the development of suitable controls for such complex apparatus, a transient model of the experimental rig using the methodology presented in [39] is being developed. In terms of hardware, motorized needle valves are installed downstream the compressor and the heater. According to [37], acting on the recirculation valve downstream the compressor proved to be the most effective strategy to operate the rig during off-design conditions; this device sends part of the compressed working fluid upstream the cooler but not directly to the compressor inlet to prevent any temperature shift from the critical region. As concerns the set of valves downstream the heater, their purpose is to either by-pass or to throttle the flow conditions upstream the turbine.

With regards to the ancillaries of the sCO_2 loop, two vessels and a gas booster were considered to charge the system beyond the critical pressure as well as to potentially vary the charge during operation. The gas booster is also meant to ease the startup procedure by providing a little flow at low heat conditions such that any liquid within the circuit is vaporized before starting the turbomachinery. Reasons for choosing a gas booster against a CO_2 piston pump was mainly cost, around 15 times higher in the second case. Pressure relief and solenoid valves complete the configuration together with static components such as filters, receiver, oil separators and sight glass

Design codes taken into account for all the components and the whole CO2 loop were the ASME BPV Code Section VIII Division 1 or the PD 5500 directives.

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THE COMPRESSOR-GENERATOR-TURBINE UNIT

The Compressor, Generator and Turbine unit, herein referred as to CGT, is the core of the sCO_2 heat to power conversion system. Its design involved varies expertise and it is summarized in the following paragraphs. The first step of the procedure was to select the most suitable turbomachinery technology based on the design specifics of Table 3.

The selection of the turbomachinery type and architecture was performed with reference to technical and economic criteria. Based on the non-dimensional theory for turbomachinery, proposed by Balje [40] and applied to the sCO_2 field by Fuller [41], a first conclusion that was drawn was to discard positive displacement machines which are mainly suitable for high pressure ratios and small flow rates. As concluded in [41], for low power sCO_2 turbomachinery (<0.3MWe) single stage radial turbine is preferable. In fact, an axial multistage turbine would have lower speed and high efficiency but it would also require additional costs that can be justified only for large scale systems. On the other hand, a radial outflow turbine would have low revolution speeds but also low efficiency, high axial loads and high bending stresses. Instead, a radial inflow turbine should have a high efficiency for low pressure ratio, but also high revolution speed and high axial loads. Additional selection criteria for the turbomachinery technology were efficiency and cost. After a comparison made with the support of Table 4, the radial inflow turbine appeared to be the most suitable architecture for the current application.

Table 4: CGT architecture comparison.

	Axial multistage	Radial outflow	Radial inflow
Efficiency	+ + +	+	+ +
Cost	- - -	-	-
Ranking	3	2	1

A synchronous permanent magnet machine was selected for the electrical generator due to the high revolution speeds of the turbomachines as well as the authors' know-how on the design of such devices for small-scale ORC applications. To minimize the costs of the CGT unit, a single shaft arrangement was considered and resulted in the architecture shown in Figure 6. The Figure also shows the shaft guidance systems and passive seals. The pressure reduction to minimize windage losses is made by the creation of a cavity around the generator, surrounded by passive seals. A drain continuously vacuums this central part and keeps it at sub-critical pressure. The good efficiency of the static passive seals is ensured by the compact architecture which allows an accurate placing, close to the rotating shaft. The thermal insulation coupled with active cooling protects the bearing system located on the turbine side and ensures proper operation of the generator.

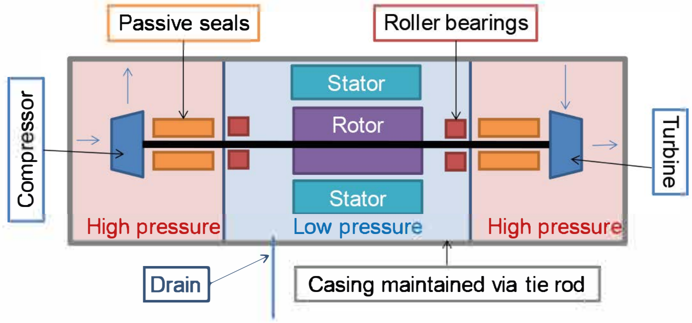

Figure 6: CGT layout.

AERODYNAMIC DESIGN

Several methodologies of varying complexity were employed for the design of the radial compressor and turbine. The design procedure is summarized in Figure 7 while more extensive details are provided in reference [35].

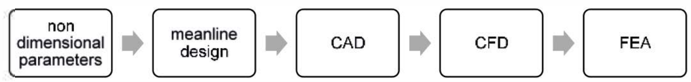

Figure 7: Turbomachinery design procedure.

Preliminary speed and dimensions of the impellers were determined from the non-dimensional theory [40] with the relevant equations included in the thermodynamic design shown in Figure 3. Based on these figures and the design specifics resulting from the cycle analysis (pressure ratio, mass flow rate, isentropic efficiencies), a mean-line design of the turbomachines was carried out to calculate blade angles and overall dimensions of the flow passages. Three-dimensional geometries of the impellers and of the diffusers were generated through CAD and CFD studies set up in in Star-CCM+ package. In particular, using 3D steady state RANS CFD simulations, based on the mixing plane approach and taking into account real gas properties for CO_2 [35], more accurate estimations of the isentropic efficiencies were made and compared to the ones assumed in the cycle analysis. Iterations were subsequently carried out to maximize the isentropic efficiency and to avoid inconsistences with the cycle analysis. The results of this procedure are summarized in Table 5.

Table 5: Summary of the aerodynamic design.

		Compressor	Turbine
Rotor	Diameter	55 mm	72 mm
	Number of blades	7	14
Nozzle	Number of blades	11	17
Isentropic efficiency (total-static)		76%	70%

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Figures 8 and 9 show the compressor and turbine impellers which are unshrouded and machined from material block and milled to desired shapes derived from the aerodynamic design.

Figure 8: Compressor impeller

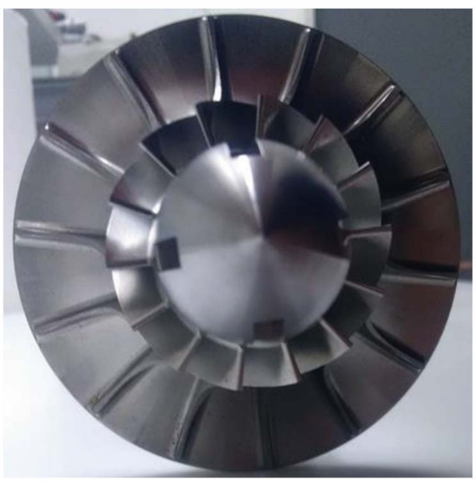

Figure 9: Turbine impeller

STRUCTURAL DESIGN

The rotating speed directly affects the rotational guidance technology. In order to optimize performance and reliability, high precision roller bearings have been employed. The natural frequencies and gyroscopic effect on the rotating parts are simulated by Finite Element Analysis (FEA). The stiffness of the bearings was simulated according to manufacturer's data and calculated load.

This study allowed to retrieve the frequency spectrum and, in turn, to identify the safe operating regime in terms of revolution speed. Furthermore, the analysis allowed to design the shaft. As shown in Figure 10, it has a bigger diameter in the central part due to the permanent magnets of the electrical generator. As an example, Figure 10.b, shows the stress induced by a white noise load.

The structural design of the CGT included also included FEA simulations to verify the structural integrity of static parts like the casing. These analyses were carried out assuming a pressure 1.5 times the maximum value at nominal operating conditions. Figure 11, shows local Von Mises stress in the casing. This picture further enables visualization of the openings for the electrical generator, the drainage as well as the lubrication channels for the bearings. Figure 12, shows the full CGT assembly whose overall dimensions are 290x290x502mm (WxHxL).

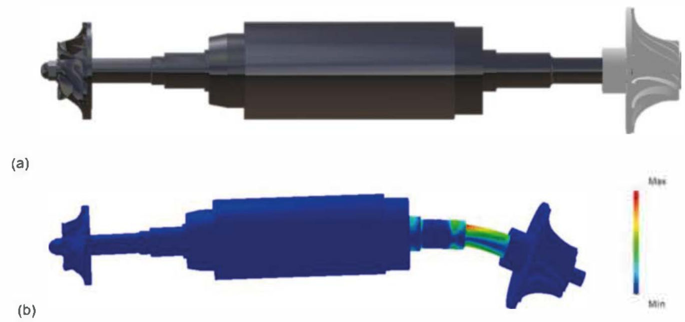

Figure 10: Shaft layout (a) and FEA simulation results to white noise load profile (b).

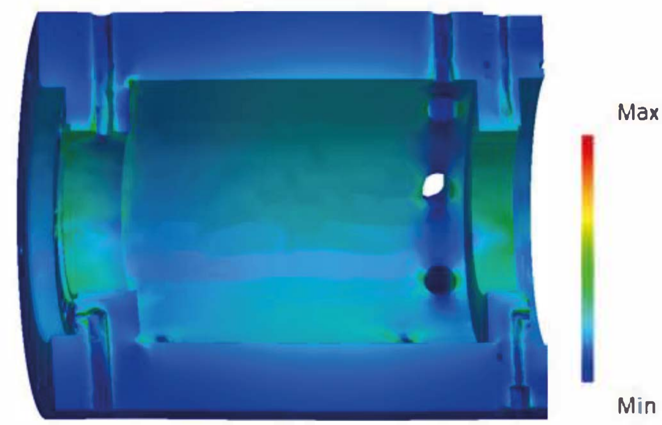

Figure 11: Von Mises stress on the casing.

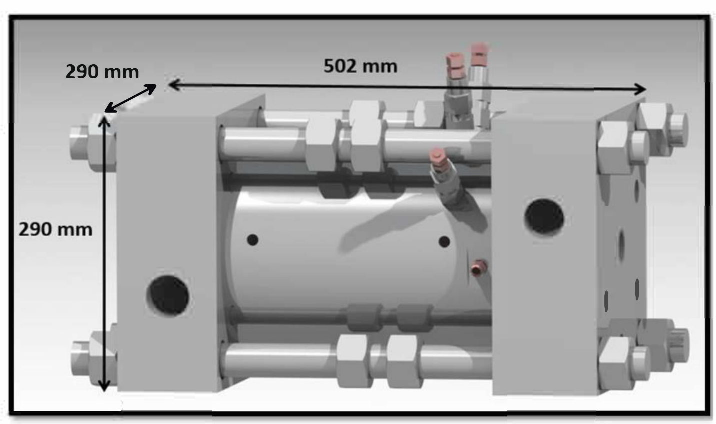

Figure 12: Final design of the CGT

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ANCILLARIES LAYOUT

To ensure a proper operation of the CGT during any operation mode (idle, transient, steady), a number of auxiliary loops were considered. The complete P&ID chart of the CGT is shown in Figure 5. In order to reduce costs and enhance reliability, most of the ancillaries are off-the-shelf components specially designed for CO_2 refrigeration applications.

The cooling loop controls the inverter and generator temperatures using water as a cooling medium and a dry cooler to reject the heat to the ambient. The lubrication loop provides oil to the bearings of the CGT. It consists of an oil tank, and oil pump which supplies oil to the bearings. The oil is recovered inside the generator cavity by the draining loop, and then directed back to the oil tank.

The ancillary loops and the CGT will be housed in a common enclosure. The enclosure will also house the electrical cabinet and instrumentation that monitors pressure, temperature and vibration of the installation. The instrumentation will be connected to the electrical cabinet PLC which monitor and control the operation of the CGT.

CONCLUSIONS & FUTURE WORK

Supercritical CO_2 (sCO_2) power systems are promising candidates for the replacement of conventional steam power plants in any high-grade heat to power conversion application. In addition to the large-scale applications, where the power size of sCO_2 plants will be in the order of megawatts, an attractive power range is also the one between tens and few hundreds of kilowatts. The high compactness of sCO_2 equipment in this range of capacities, however, involves additional design challenges that the current research activity aims to tackle. In particular, this paper has shown the methodology through which a 50 kWe sCO_2 heat to power conversion unit has been designed. The sCO_2 system is based on a simple regenerative Joule-Brayton cycle architecture with pressure ratio of 1.7 and a single shaft Compressor-Generator-Turbine (CGT) unit. This configuration, together with the employment of a plate heat exchanger as CO_2 cooler and of a micro-tube CO_2 heater, allows to lower the investment costs without a substantial reduction in system performance. A future challenge of this research will be an extensive test campaign at steady and transient operating conditions to assess the actual performance of the sCO_2 system as well as its reliability for future industrialization.

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