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# Design of a high-temperature heat to power conversion facility for testing supercritical $CO_2$ equipment and packaged power units

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**Abstract**

This paper addresses the need of bridging between fundamental energy research and industrial exploitation of technologies by presenting a state of the art experimental facility to investigate pilot heat exchangers and plants dealing with high temperature waste heat recovery and conversion. The facility comprises a 830 kW process air heater with an exhaust mass flow rate of 1.0 kg/s at 70 $mbar_g$ and maximum temperature of 780 °C. The heater has a 2.0 m long test section for the installation and characterization of waste heat recovery heat exchangers. The heat sink is a 500 kW water dry cooler with full control of flow rate and temperature of the cooling stream. The high-temperature heat to power conversion facility hosts a 50 kWe power conversion unit based on the simple recuperated Joule-Brayton cycle with supercritical $CO_2$ ($sCO_2$) as working fluid. The packaged, plug and play $sCO_2$ system utilizes a single-shaft Compressor-Generator-Turbine unit. The paper discusses the main design features of the test facility as well as operation and safety considerations.


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*Keywords:* supercritical $CO_2$; waste heat recovery; printed circuit heat exchanger; micro-tube heat exchanger; turbomachinery

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## 1. Introduction

High-grade thermal energy recovery or conversion from waste heat sources is a challenging but attractive opportunity for industry given the high-exergy content, low pay back periods, and consequently, favourable financial returns. With reference to the segmentation proposed by Forman [1], exhausts and effluents at temperatures beyond 300°C are typical in the iron and steel, nonmetallic minerals (glass, cement), nonferrous materials (aluminum) industrial sectors, and account for potential energy recovery of up to 275 TWh in the European Union [2,3] and 3367 TWh worldwide [1].

Despite the environmental benefits, the main barrier to the installation of high-grade waste heat recovery technologies is the economic feasibility of the investment. In fact, the harsh operating conditions call for very high-specifications and bespoke engineering solutions that translate to significant capital expenditures (CAPEX) and reduced returns on investment.

Among the technologies currently being investigated, supercritical $CO_2$ ($sCO_2$) heat to power generation may be a disruptor for high temperature applications. In fact, the high operating pressures in $sCO_2$ devices lead to compact equipment with the potential to reduce capital costs. Moreover, from a thermodynamic perspective, Joule-Brayton cycles benefit from higher energy and exergy efficiency than conventional bottoming thermodynamic cycles such as the Organic or steam Rankine Cycles [4,5]. These facts can translate to greater savings during the operational lifetime of $sCO_2$ plants, leading to higher return on investment.

Present research on $sCO_2$ spans from next nuclear and fossil fuel power generation (500 to 1,000 MWe) [6–9], modular nuclear power generation (300 MWe) [10–12], solar thermal power generation (10 to 100 MWe) [13–17], shipboard propulsion, geothermal, oxy-combustion (1 to 100 MWe) [18–23], and industrial scale waste heat recovery (1 to 10 MWe) [24–26]. 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.

Experimental $sCO_2$ facilities have been increasing in terms of number and potential. As reported in [27], most of the integral test loops are based on simple recuperated cycles [28]. Target pressure ratios range from 1.4 to 2.9 while calculated efficiencies range from 14.7% to 31.5%.

The activities presented in this paper are focused on the development of high-Technology Readiness Level (TRL) small-scale $sCO_2$ systems in the power range around 50 kWe. Reasons for focusing on this segment relate to the widespread nature of the waste heat as well as to the preliminary feasibility studies reported in [29,30]. The novel aspects presented in this work are not only the design challenges and lessons learned in the development of a packaged, plug&play $sCO_2$ heat to power conversion systems but also relate to the development of a state of the art facility for high-temperature equipment testing at pilot scale.

## 2. High Temperature Heat To power Conversion facility (HT2C)

The High Temperature Heat To power Conversion facility (HT2C) at Brunel University London provides the capabilities for testing equipment and systems at pilot scale and addresses the technological gap which commonly exists during the scaling up of the technologies from fundamental research to mass production. An overview of the facility and its communication layout is shown in Figure 1 with reference to a configuration for $sCO_2$ heat to power conversion testing. Details on the $sCO_2$ unit are reported in Section 3.

The heat source of the HT2C is an 830kW gas fired process air heater whose main features are listed in Table 1 while the Process and Instrumentation Diagram (P&ID) is detailed in Figure 2(a). The high flexibility of operation is the key feature of the heat source and the whole facility. Inlet conditions to the high temperature test section are controlled with a proprietary system that relies on primary fan speed and flow rate of flue gas as control signals while temperature downstream the process air heater and airflow rate are used for control feedback. A similar control architecture is repeated for the low temperature test section. The high temperature test section has a maximum length of 2000 mm while the cross-section is 500x500 mm with 200mm alkaline earth silicate wool insulation.

The heat sink of the H2TC facility is a 500 kW dry cooler system. Its features are listed in Table 2 while the P&ID is shown in Figure 2(b). 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 $sCO_2$ tests.

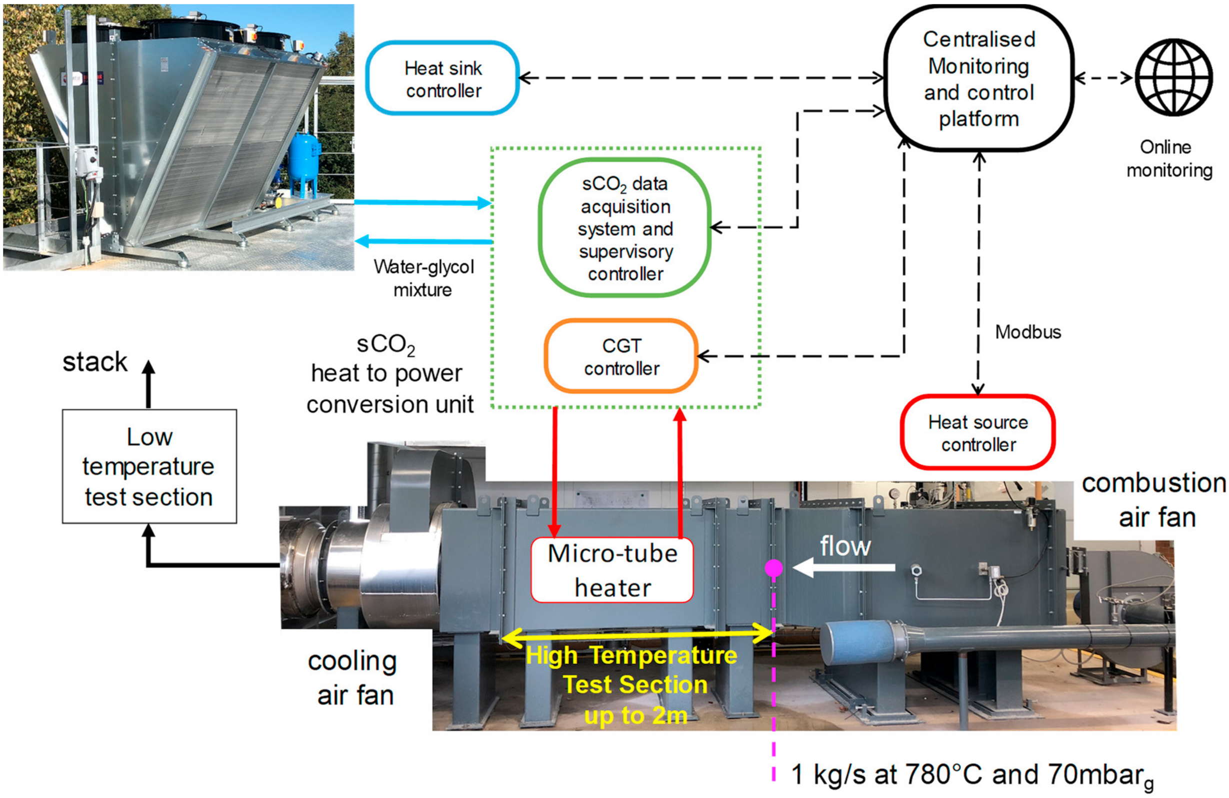


Fig. 1. Overview of the High Temperature Heat To power Conversion facility at Brunel University London with communication layout

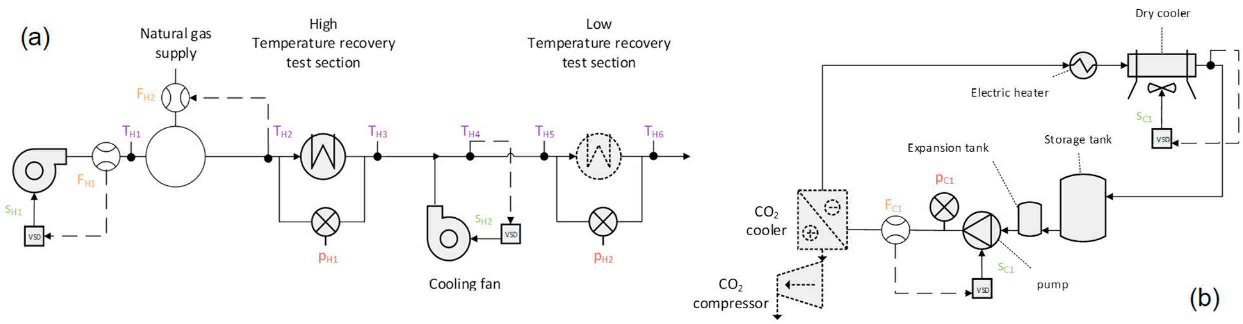


Fig. 2. Process and Instrumentation Diagrams: (a) heat source; (b) heat sink

Table 1. Heat source specifics.

| | |
|---|---|
| Net max power supplied | 830 kW |
| HT test section dimensions (LxWxH) | 2000x500x500 mm |
| Maximum operating temperature | 780 °C |
| Pressure drop allowed for primary HX | 70 mbar |
| Fuel | Natural Gas (G20) |
| Gas input peak design | 83.5 Nm³/h |

Table 2. Heat sink specifics.

| | |
|---|---|
| Total cooling duty | 500 kW |
| Water on temperature | 60 °C |
| Mono-Ethylene glycol | 25% |
| Total airflow at standard conditions | 35.7 kg/s |
| Total fin and tube surface area | 850 $m^2$ |
| Maximum fluid temperature | 100 °C |

The monitoring and control of the whole facility is centralized. A supervisory controller based on the IEC61499 standard allows the other PLC controllers of the subsystems to communicate via the Modbus protocol. High-level information can also be accessible online.

## 3. Packaged $sCO_2$ heat to power unit

In the current study, the HT2C facility has been set up to host the demonstration of a 50 kWe heat to power unit based on a supercritical $CO_2$ Joule-Brayton cycle. The design conditions of the loop are summarized in Table 3 and result from a constrained optimization study available in [31]. In particular, the $sCO_2$ system is based on a simple recuperated cycle with a pressure ratio of 1.7 and maximum operating temperature of 400°C. The recuperator is a commercial printed circuit heat exchanger and the $CO_2$ cooler is a plate heat exchanger. The heater is a novel micro-tube heat exchanger that will be installed in the duct shown in Figure 1.

Table 3. Design conditions of the $sCO_2$ power unit.

| | min | max |
|---|---|---|
| Pressure [bar] | 75 | 127.5 |
| Temperature [°C] | 35 | 400 |
| Mass flow rate [kg/s] | 2.25 | |

The CAD design of the $sCO_2$ system is reported in Figure 3. Except for the $CO_2$ heater, all the components have been enclosed in a 20ft standard shipping container. Pipework is in SS316 while flanged connections are of ring joint type (RTJ). This arrangement fulfills the needs of a packaged solution which is required for a high-TRL compliancy of the developed unit. In fact, the compact solution is also a plug&play one; the connections required are inlet and outlet pipes for heat source and sink, a compressed air supply and electrical connections to the grid. An additional requirement that the $sCO_2$ unit will fulfil is the compliancy to the Pressure Equipment Directive (PED) (2014/68/EU).

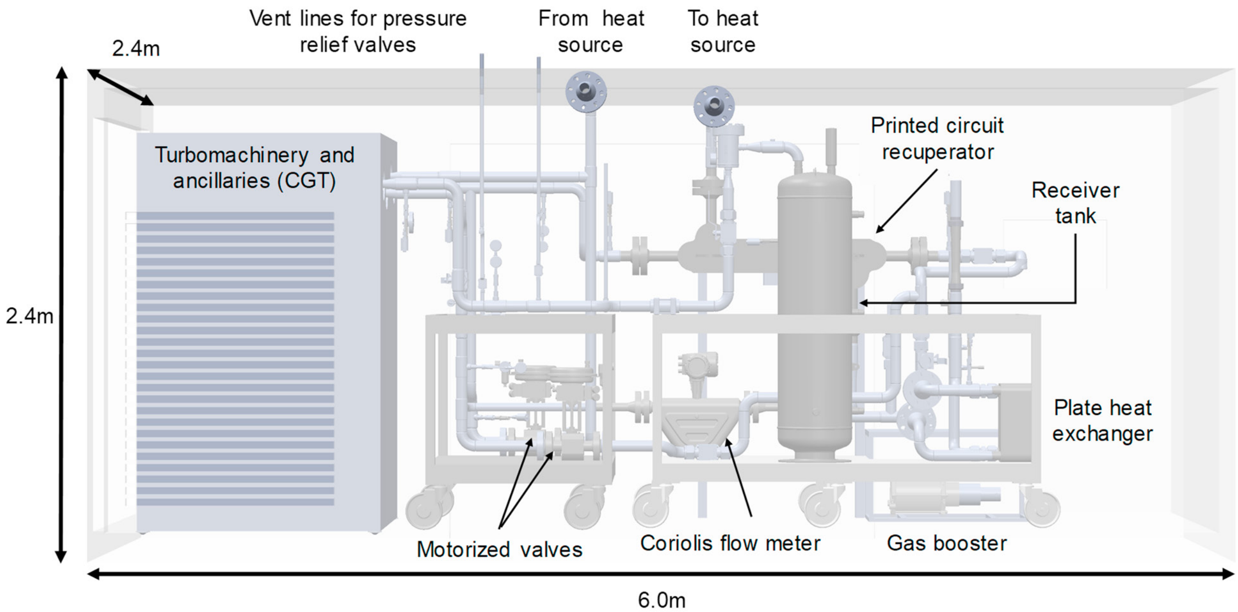


Fig. 3. High-temperature $sCO_2$ heat to power conversion system packaged in a standard 20ft shipping container

The instrumentation selected to characterize the performance of components and the whole $sCO_2$ system is shown in the P&ID of Figure 4. In particular, to assess the performance of heat exchangers, a high-accuracy piezo resistive pressure transducer upstream of the device is coupled with a differential pressure transducer across the heat exchanger. As for the the temperature transducers, 3-wire resistance temperature detectors (RTDs) type PT100 are considered throughout the $CO_2$ loop except at the measurement locations across the compressor where 4-wire RTDs are considered. Flue gas temperatures are measured with K-type thermocouples. A Coriolis gas flow meter is installed downstream of the compressor by-pass. Assuming the design conditions to be measured parameters, based on the transducers accuracies summarized in Table 4, Table 5 reports the expected uncertainty propagation of the key performance parameters. From this analysis, it can be concluded that the quantity that mostly affects the measurement uncertainty is the pressure at the inlet of the compressor.

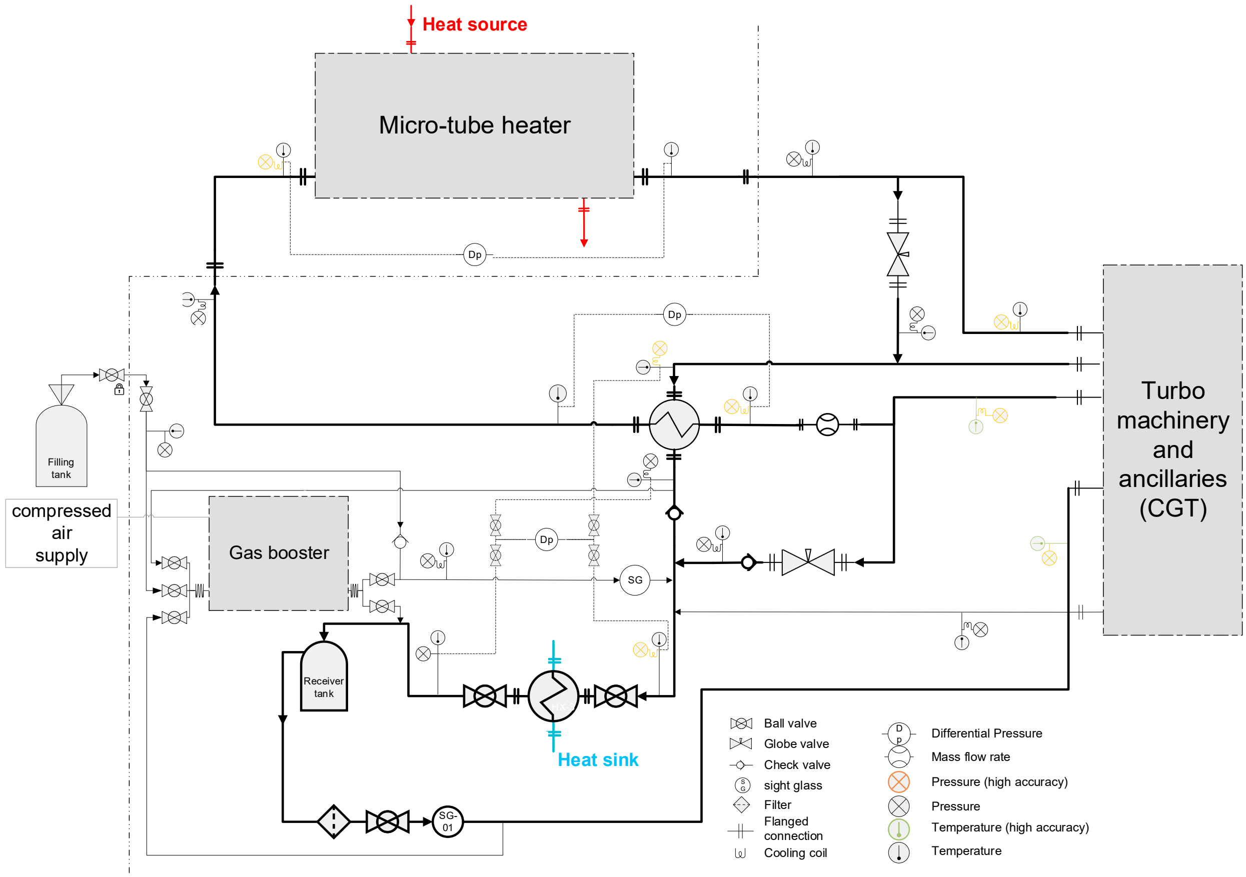


Fig. 4. Simplified P&ID of the $sCO_2$ heat to power conversion system shown in Figure 3

Table 4. Summary of transducers accuracies.

| Accuracy | high | standard |
|---|---|---|
| High pressure transducers | 0.17 bar | 0.52 bar |
| Low pressure transducers | 0.10 bar | 0.34 bar |
| Temperature transducers (RTD) | 0.03K | 0.06K |
| Differential pressure transducer | 1.9mbar | |
| Mass flow rate (Coriolis) | 0.35% of measured value | |

Table 5. Estimated measurement uncertainty at design conditions.

| Uncertainty | Power | Efficiency |
|---|---|---|
| Compressor | 2.66% | 3.32% |
| Turbine | 0.43% | 0.47% |
| Heater | 0.36% | |
| Recuperator (cold/hot side) | 0.36%/0.35% | |
| Cooler | 0.61% | |

The envisaged startup sequence has been discussed in [31] but is also summarized here for the benefit of the readers. A gas booster instead of a more expensive $CO_2$ pump has been considered not only for ease of charging the $CO_2$ circuit but also to allow a 5-10% of nominal flow rate during startup. Low temperature heat is provided to ensure that $CO_2$ is not in liquid phase before starting the compressor.

Two electric actuated globe valves will be installed in the compressor and turbine by-pass branches as suggested by the research group of the US Naval Nuclear Laboratory [32,33]. In a previous release of the design [31], a third valve upstream of the turbine was also considered to be able to set the inlet pressure during the tests. However, besides the additional budget implications, since globe valves lead to a noticeable pressure drop even when fully opened, the valve upstream of the turbine was eventually discarded to avoid adverse impacts on overall system performance.

## 4. Compressor, Generator and Turbine (CGT) unit manufacturing and assembly

The Compressor, Generator and Turbine unit, herein referred as the CGT, is the core of the $sCO_2$ heat to power conversion system. Its design involved diverse expertise and it is reported in more detail in [31,34]. The results of the aerodynamic design are summarized in Table 6. Figure 5, shows the unshrouded compressor and turbine impellers, which were machined from material block and milled to desired shapes derived from the aerodynamic design.

Table 6. Summary of the aerodynamic design.

| | | Compressor | Turbine |
|---|---|---|---|
| Rotor | Diameter | 55 mm | 72 mm |
| | No. of blades | 7 | 14 |
| Nozzle | No. of blades | 11 | 17 |
| Isentropic efficiency (total-static) | | 76% | 70% |

(a) (b)

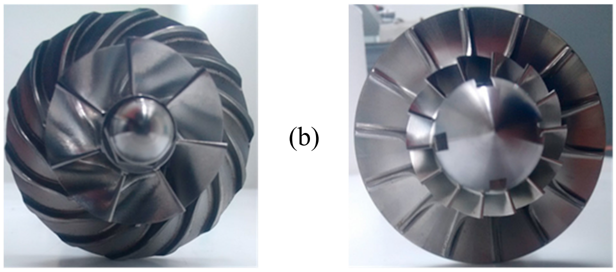

Fig. 5. Impellers: (a) compressor; (b) turbine

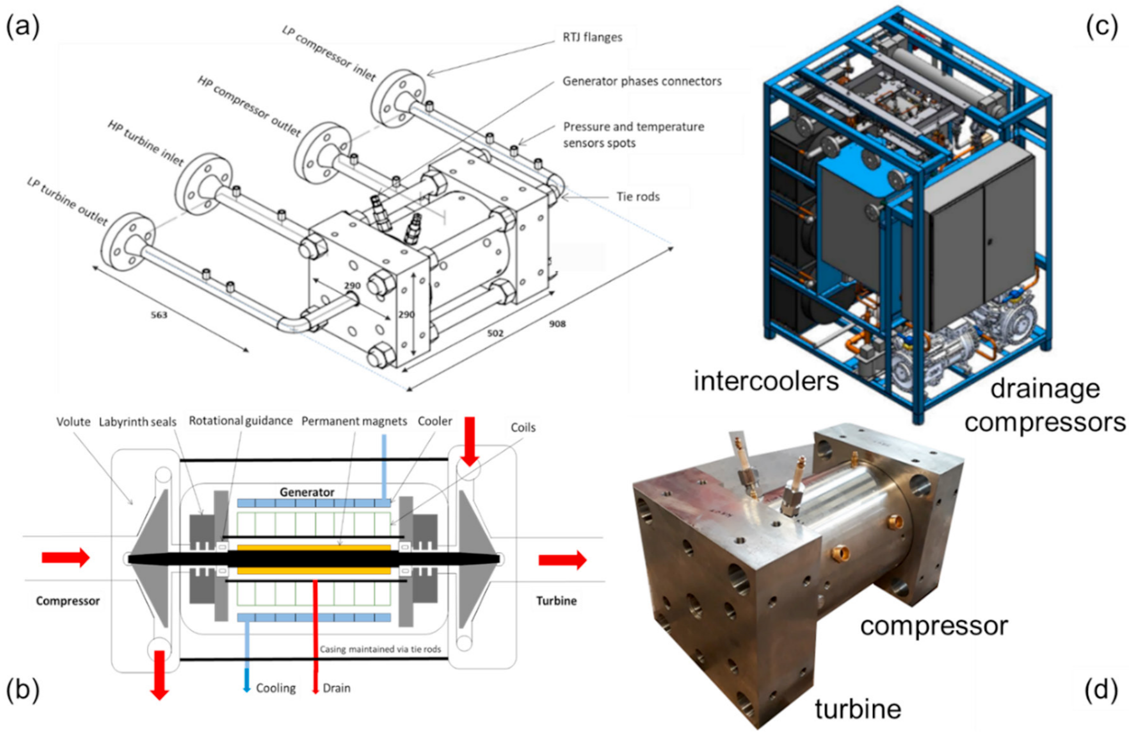


Fig. 6. CGT unit: (a) turbomachinery and generator dimensions and interfaces, (b) operating principle, (c) turbomachinery and ancillaries (d) picture of the manufactured and assembled turbomachinery and generator unit

Figure 6(a), shows the full CGT assembly whose overall dimensions are 290x290x502 mm (WxHxL). An emergency pneumatic valve has also been considered between the compressor outlet and compressor inlet. The purpose of this valve is to shortcut the compressor and, consequently, prevent any high pressure flow going to the turbine in case of faults in the controls or grid failures.

As shown in Figure 6(b), the CGT is rather complex due to the $CO_2$ and ancillary loops required for lubrication, drainage and cooling [31]. Among them, two piston compressors with plate intercoolers have been considered to inject the drainage, after oil separation, back to the main loop. The choice of a dual-stage intercooled compression was to limit the drainage recovery temperature at a value suitable for refrigeration compressors (<120°C). These ancillaries are shown in Figure 6(c) together with the whole CGT unit.

## 5. CONCLUSIONS AND FUTURE WORK

Energy recovery from high temperature sources has a wide application potential ranging from the steel industry to nuclear and concentrated solar power. Nonetheless, the economic feasibility of high temperature heat recovery or heat to power conversions is often hindered by the high capital expenditure required for the design and manufacture of high-specification bespoke equipment. In order to promote the transition towards higher technology readiness levels of fundamental research concepts and technologies, this paper has presented the design and layout of a state of the art high-temperature facility suitable for testing heat recovery and heat to power conversion equipment at pilot scale. The facility comprises a 830kW gas fired process air heater and a 500 kW water dry cooler as heat sources and sink respectively. There are two test sections for heat exchangers testing at low and high temperatures and a centralized control and monitoring system based on Modbus protocol to supervise the whole facility.

In this work, the facility serves to provide heat gain and rejection to a 50kWe supercritical $CO_2$ unit based on a simple recuperated Joule-Brayton cycle. The core of the $sCO_2$ unit is a compressor-generator-turbine unit (CGT) composed of a single shaft turbomachinery. The experimental rig is packaged in a 20ft container and will be CE marked according to the Pressure Equipment Directive (PED) (2014/68/EU). A preliminary uncertainty analysis showed a maximum uncertainty of 2.3% on the compressor isentropic efficiency mostly due to the inlet pressure measurement.

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