Lessons from testing the sCO2-HeRo turbo-compressor-system

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Abstract

Motivated by the growing interest in sCO$_2$ Joule-Brayton cycles as a future power plant technology for both energy conversion and safety back-up systems, this paper presents the experimental proof of concept for a small turbo-alternator-compressor unit (TAC). After a brief description of the system as a whole, the design of the TAC is given in detail. This is followed by an outline of the experimental study consisting of the compressor performance map (pressure ratio vs. mass flow rate) complimented with findings on the cycle operation. Comparison between the experimental performance map and the CFD calculations indicates the validity of the CFD code for sCO$_2$ compressor calculations and hence this code is suitable for design and performance map calculations. The paper discusses two different approaches for determining the compressor efficiency and proposes inlet density measurements as a more accurate method compared to temperature measurements. One of the findings of the test campaign is the stable operation of the cycle, which is often believed to be an issue caused by the high dependency of compressor inlet density on temperature. Therefore, one of the contributions of this paper is to explain briefly that compressor inlet state variations should be considered as a function of enthalpy fluctuations rather than temperature fluctuations.

Introduction

The sCO$_2$ Joule-Brayton cycle is considered as a future power conversion cycle since current studies (Ahn et al. 2015) promise a higher thermal efficiency under certain operating conditions. In some applications and under specific conditions they also promise smaller component sizes compared to steam cycles (Ahn et al. 2015). Increasing the flexibility of energy conversion is also discussed (Alfani et al. 2019).

However, the promising benefits are mostly shown by calculations, complimented by first demonstrators. Thus, in order to proof the concept and bring the technology to the market the technical readiness level (TRL) has to be increased gradually.

Bringing the technology to the market requires:

I. Enhancing design tools for components such as heat exchangers and turbomachinery, including its bearings and seals in order to cope with sCO$_2$.
II. Including these models in cycle calculation tools and thereby exploring heat sources for which the sCO$_2$ cycle is particularly suited.
III. Selecting and developing materials especially for sCO$_2$.
IV. Verification of the functionality on the component level as well as building small to medium scale demonstrators.

The European sCO$_2$ alliance (www.sco2.eu) is working on this technology for both energy conversion cycles as well as emergency cooling back-up systems with projects such as sCO2-

HeRo, sCO2-Flex, I-ThERM and sCO2-4-NPP. This publication presents results from the compressor and system testing of the sCO2-HeRo and its follow up project sCO2-4-NPP. This system is designed for nuclear power plants (NPP) as a back-up cooling system (Benra et al., 2016a, 2016b, Starflinger et al. 2017, 2018) that according to Fig. 1 brings decay heat from the reactor core to the ultimate heat sink (Venker et al., 2013, 2014). By utilizing compact heat exchangers, this system can be retrofitted into existing NPP. In the system the turbine extracts at least enough energy to propel the whole system while the sink heat exchanger releases the decay heat to the environment. Therefore, the fundamental idea of the system is that residual decay heat itself propels the cooling system rather than an external power supply.
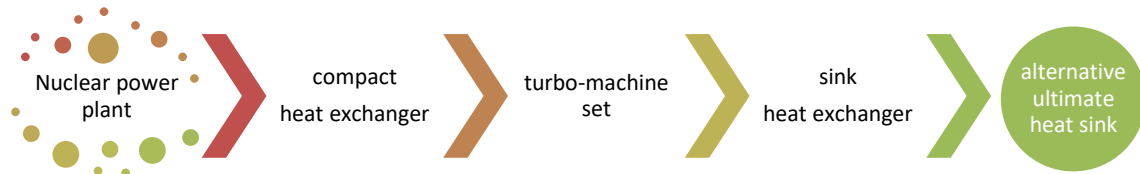

Fig. 1 – Working principle of the sCO2-HeRo system

The heat input is supplied by the reactor core and is transferred with a compact heat exchanger from the water steam side to the cycle. In the experimental loop, the temperature of sCO_2 does not exceed 200 °C and the maximum pressure is around 117 bar. The fluid is expanded in a one-stage radial turbine to the lower nominal pressure of 78.3 bar and heat is released to the environment through a conventional air heat exchanger. The sCO_2 leaves the heat exchanger at about 33 °C before it is compressed in a single-stage radial compressor with a pressure ratio of 1.5. Compressor, alternator and turbine (TAC) are mounted on one shaft and are assembled in a closed housing as shown in Fig. 2. The generator is partially cooled by a CO_2 stream flowing from the compressor impeller outlet to the turbine impeller inlet.

The first version is built with permanent lubricated rolling bearings providing a simple and cost-efficient solution. To mitigate the risk of operating lubricated bearings in a supercritical environment, the pressure in the housing can be reduced and the leakage flow can be compressed back into the cycle.
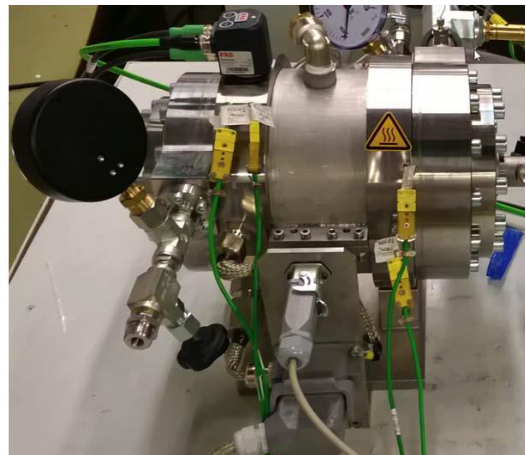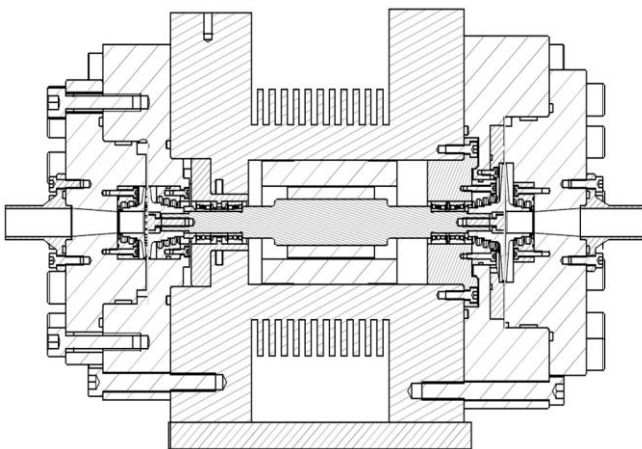

Fig. 2 – Cross sectional view of the TAC (left) and assembled machine (right)

Caused by the small heat input of 200 kW the optimal rotational speed for the turbomachinery aerodynamics is around 200,000 rpm. This is already challenging from a rotor dynamic and electrical point of view but is impossible to operate considering the windage losses in the housing. Therefore, the rotational speed is reduced to 50,000 rpm (Schuster et al., 2016, Hacks et al., 2018a). Thereby, the non-dimensional size parameters of the compressor highlighted with a red star in Fig. 3 are far away from the optimal efficiency values indicated by the contour lines in the same figure. This causes an efficiency penalty. In passing by it is interesting to see that so far most demonstrators (coloured dots) have flow coefficients below the optimum value. Thus, the full potential of sCO$_2$ cycles could not be demonstrated so far and larger systems are required.
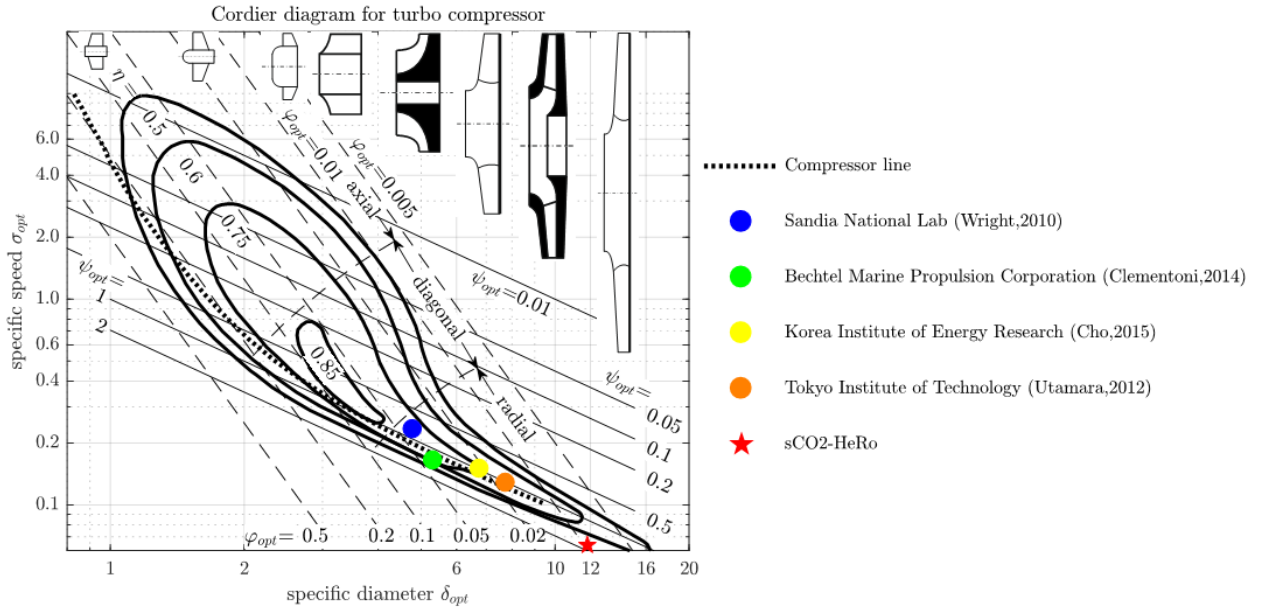

Fig. 3 – Non-dimensional size parameters of the sCO2-HeRo and similar TAC, Cordier diagram reproduced from Balje (1981)

The small through-flow coefficient results in flat velocity triangles with small flow angles and leads to large separation areas if thin blades are applied. To overcome this, thick blades are used which improve the guidance of the flow as shown in Fig. 4. The flow still separates close to the trailing edge but this can be accepted considering the non-optimal compressor size parameters. At the current state of research, there is no reason to trace the need for the thick blading back to the sCO$_2$ properties but strong reason to believe it is caused by the small flow coefficient. However, the knowledge on deaccelerating flows close to the critical point is still limited but with progressing research, it will become clear if a special blade geometry is beneficial for this kind of flow. The whole geometry has been disclosed (Hacks et al., 2018a) so that the test results can be used for computer code validation.
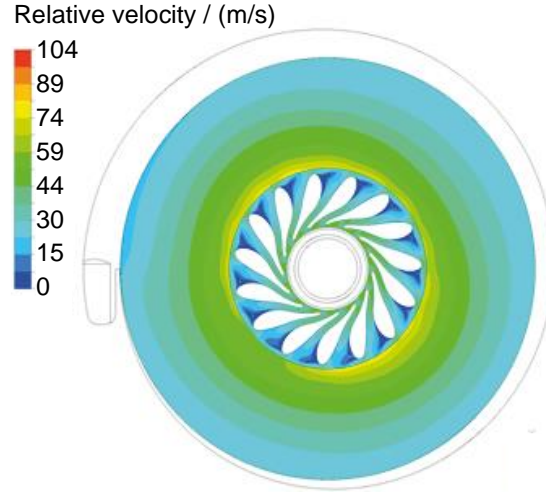
Fig. 4 – Cross-sectional view of the sCO2-HeRo compressor showing the relative velocity

Compressor testing

The compressor testing is conducted at Research Centre Rez, using the SUSEN sCO_2 experimental loop (Hajek et al., 2014, Vojacek et al., 2018). This test facility is particular suitable to study key aspects of sCO_2 cycle components (turbomachines, heat transfer, erosion, corrosion etc.) with a wide range of operation parameters (Vojacek, 2018) such as temperatures up to 550 °C and pressures up to 300 bar. The main cycle components during the compressor test are the turbomachine from Fig. 2, electrical heaters and water coolers. The pressure in the system is controlled either by heating/cooling or by inventory control. During the compressor test, the pressure in the central housing of the turbomachine can be lowered in order to reduce friction and power requirement. A piston pump returns the leakage from the TAC to the cycle.

The performance map of the compressor is generated from pressure, temperature and mass flow rate measurements before and after the compressor. Hacks et al. (2018b) describe details about the measurement equipment and the measurement uncertainties. Since the fluid properties such as density vary greatly with pressure and temperature close to the critical point, a variation of the inlet conditions is important during the testing. Thereby, high inlet density 660 kg/m³ (p_0 = 79.3 bar, T_0 = 31.5 °C), design inlet density 566 kg/m³ (p_0 = 78.3 bar, T_0 = 33 °C), and low inlet density 328 kg/m³ (p_0 = 77.3 bar, T_0 = 34.5 °C) cases are investigated. The resulting pressure ratio as a function of the mass flow rate is depicted in Fig. 5 for two rotational speeds. For the design inlet density the measurement uncertainty is shown which is relatively low for the mass flow rate but much larger for the pressure ratio. This is caused by the good measurement accuracy of the direct mass flow measurement via a Coriolis mass flow meter on one hand and the calculation of a small pressure ratio from two pressure measurements on the other hand.

The pressure ratio calculated by CFD is within the measurement uncertainty. Therefore, the computation model is able to predict the sCO_2 flow in compressors in general. Further, the pressure ratio is strongly depending on the inlet density which can readily be seen from the fundamental thermodynamic relation *Tds = dh – dp/ρ*. This states that for the same enthalpy change (roughly the same work input), for the same losses (*Tds*) and an adiabatic process, the pressure change is larger for higher densities.
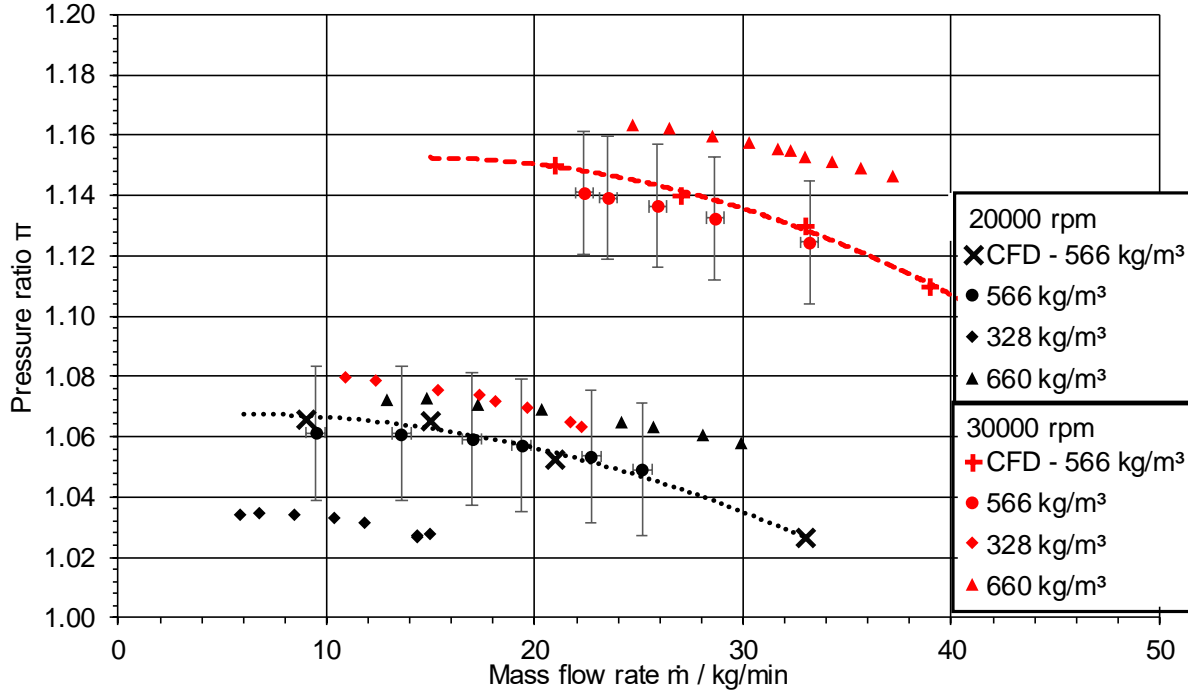
Fig. 5 – Performance map of the sCO2-HeRo compressor for different inlet conditions and rotational speeds. Calculated results compared to CFD calculations.

The steep gradients of some properties near the critical points have a severe consequence for determining the efficiency. By using temperature and pressure measurements at both inlet and outlet the uncertainty in efficiency is 15% for 1 °C uncertainty in temperature. However, the uncertainty can be reduced to less than 1% by using density measurement according to Wahl (2009). These density measurements are foreseen for the sCO2-4-NPP project.

System testing

The system testing aims on investigating the stable operation range of the cycle. This follows directly from the compressor performance test. In this, it has been confirmed that small variations of inlet temperature have a significant impact on the compressor pressure ratio. Thinking in terms of temperature, this means a small change of cooler outlet temperature can have a massive impact on the following components such as compressor, heat source and turbine and in turn might result in unstable cycle operation. This leads to the undesirable hypothesis that stable operation of sCO_2 cycles is hard to achieve. However, the system testing has disproved this hypothesis.

In the test depicted in Fig. 6, initially the compressor inlet conditions are in the so-called (by Kiran et al. (2008)) liquid-like region ($p_0 = 76$ bar, $T_0 = 32.5$ °C). Then, the cooling power is reduced continuously at nearly constant pressure and thereby the inlet state changes from liquid-like to gas-like. The properties measured respectively derived at the compressor inlet are shown in the same figure. Firstly, it can be observed that the inlet pressure is kept nearly constant. Further, the inlet temperature changes by 2 °C and the derived spec. heat capacity c_p increases when the pseudocritical line is crossed. However, no oscillations are observed in the mass flow rate, it only decrease smoothly. In conclusion, changes of cycle operation should not be expressed as changes in temperature but as changes in cooling power. Further, stable operation of sCO_2 cycles is not an issue. A detailed analysis is given by Hacks et al. (2019).
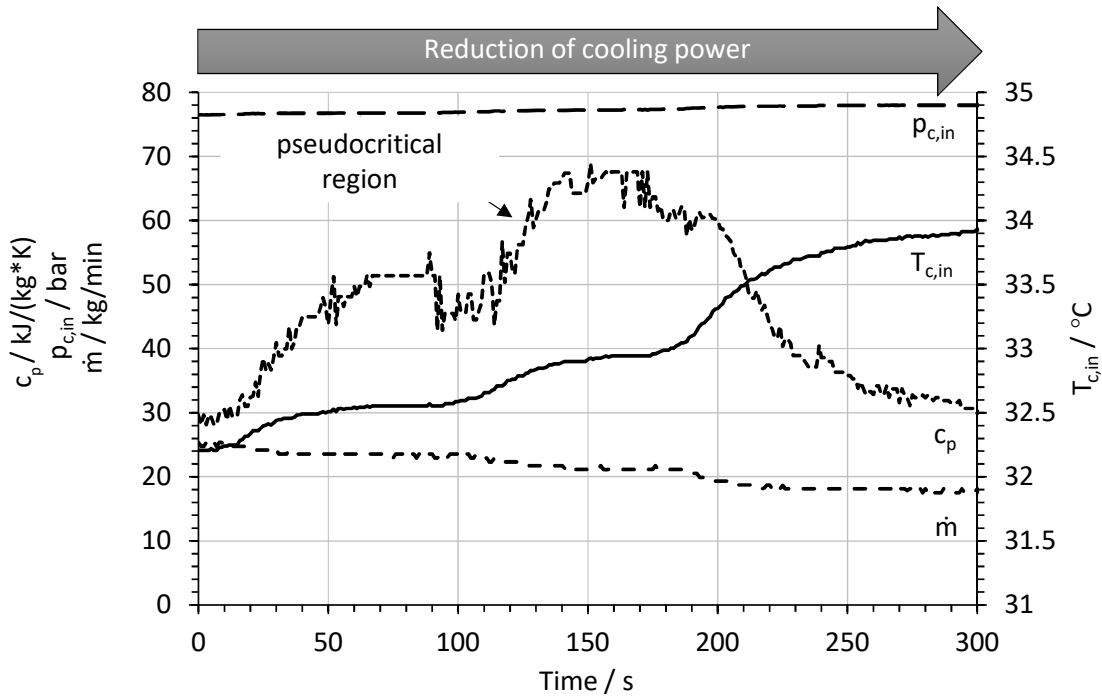
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Fig. 6 – Compressor inlet properties during cooling power variation

Summary and Conclusion

This paper has presented a back-up cooling system based on the sCO$_2$-Joule-Brayton cycle. This system was developed to technology readiness levels TRL 3 (experimental proof of concept) in the sCO2-HeRo project and the aim is to bring it to TRL 5 (technology validated in relevant environment) and parts of it to TRL 7 (system prototype demonstration in operational environment) in the sCO2-4-NPP project. On the 10 kWe scale, the optimum rotational speed for the compressor cannot be reached and therefore a blade design that provides increased guiding for the flow has to be employed. First test results of the compressor are promising and the CFD code is in general able to predict the overall performance. Design speed operation will be achieved within the sCO2-4-NPP project. During the testing, no oscillations in the cycle mass flow rate are observed even if the compressor inlet state is changed from liquid-like to gas-like. Thus, stable operation of the cycle is possible.

Acknowledgment

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