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DESIGN OF AN AXIAL SCO2 TURBINE FOR A DEMO PLANT IN AN INDUSTRIAL ENVIRONMENT

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ABSTRACT

Energy conversion processes based on supercritical carbon dioxide (sCO2) are being investigated in the field of scientific and industrial power engineering due to the great potential in terms of compactness and efficiency.

The working fluid CO2 poses a number of challenges for turbine design and aerodynamic optimization. Due to the high density of the working fluid CO2, which is associated with high losses at high velocities, special attention must be paid to aerodynamic optimization. In addition, fluid density can affect rotordynamic stability. The significantly higher demands on seals, both within the turbine and for sealing against the environment, require new sealing concepts that meet the requirements for performance and operational safety.

In this paper the development of an axial turbine for a demo plant application in an industrial environment is discussed in terms of topology, technology and optimization approaches and the resulting design concept is presented. Aero-and rotodynamic details of the specific challenges coming from the working fluid CO2 are discussed as well as material aspects and the system integration in a power cycle concept. Furthermore, the scalability to larger power output and process temperatures is described.

INTRODUCTION

The potentials of the sCO2 technology with respect to compactness and performance are well known in science and the energy industry. Development is taking place in parallel for several application areas with the involvement of industry and science. Disruptive significance is seen in CSP applications, where the use of sCO2 is expected to significantly increase competitiveness [1] as well as for fossil fired and waste heat recovery applications [2].

Due to the high maturity of existing gas and steam power plants with high efficiencies and the market situation for energy prevailing in the past, the demand and market for alternative technologies was low for a long time, especially in Europe. This has changed, especially against the background of the energy transition and several research projects have been started within the innovation program "Horizon 2020" [3-6].

In order to establish sCO2 technology in energy technology in the long term, sophisticated technical solutions with high reliability of the corresponding systems are necessary. It is therefore necessary to adapt existing design and analysis methods to the special thermodynamic properties of sCO2 and to supplement and expand the existing wealth of experience from the development and operation of conventional steam-based systems [9]. Topics include, in the field of materials engineering and mechanical engineering, the development and analysis of suitable heat exchangers, turbomachinery concepts and process modeling, with the special challenges derived in particular from compression and heat transfer near the critical point.

With the CARBOSOLA project, funded by the German BMWK, a sCO2 technology development has started in 2019. In the first two working packages an analysis of the expected advantages has been carried out for two use cases in the fields of waste and exhaust heat recovery (bottoming cycles of combined cycle gas turbine plants) and solar-thermal power plant technology (CSP). For that, a technical-economic evaluation and optimization has been performed for both use cases [7-8] where particularly for the waste heat recovery case a benefit could be derived.

Beside a further work package in which the development and the commissioning of a modular sCO2 test-rig for component development and generic experimental studies has been accomplished [9], the basic design considerations of a potential demonstration plant have been addressed. The gained knowledge, especially concerning the turbine design, defines the basis for the detailed design and realization of a small scaled sCO2 turbine in a demo plant for an industrial application within the Horizon 2020 framework. For the implementation of this EU funded research project, called CO2OLHEAT, a broad consortium of academics and industrial experts covering the necessary equipment such as compressor, turbine, heat exchanger, control systems and system integration has been founded.

The focus in this publication is the design of the 2 MW axial sCO2 turbine for the demonstration plant. In the next section the thermodynamic boundary conditions for the design are briefly described followed by the explanations of the different design aspects in terms of topology, technology, optimization approaches and finally the outlook on upscaling for larger outputs.

THERMODYNAMIC CYLCE DESIGN

A demonstration plant in an industrial environment should be small enough to limit the overall costs and the commercial risks of such an advanced research project. On the other hand, it should be scalable for higher power output and deviating temperatures, so that the knowledge gained can be transferred to commercial systems. Therefore, the thermodynamic boundary conditions were selected for the further procedure based on a use case providing waste heat from an industrial process in the magnitude of 10 MW thermal. The thermodynamic layout of the sCO2 cycle as shown in figure 1 has two expansion sections in series, one driving the turbo-compressor and the other as power generation turbine.

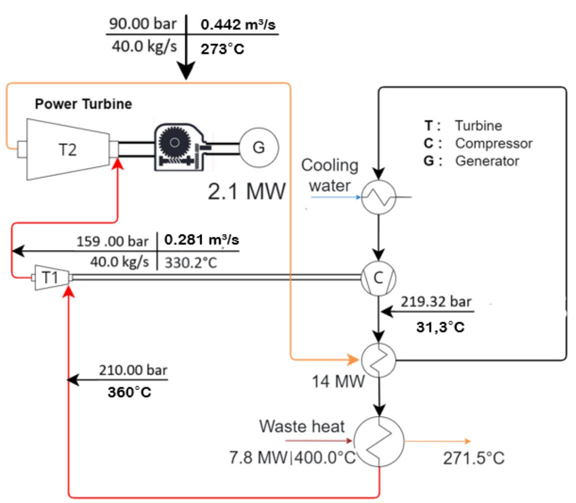

Figure 1: Preliminary thermodynamic cycle layout used for turbine design study. The thermodynamic parameters of the CO2LHEAT project deviate from this to a certain extent due to the project-specific boundary conditions.

This concept requires on the one hand a very detailed alignment of the characteristics and operational aspects between both machines. On the other hand, the thermodynamic conditions at the inlet and the outlet of the power generation turbine still allow a feasible axial turbine design with an output of approximately 2,1 MW, which can be upscaled to larger power outputs.

TURBINE DESIGN STUDY

A design study for the sCO2 power turbine was initiated based on the assumed thermodynamic requirements according to figure 1. To this end three alternative concepts were designed according to figure 2 and evaluated with respect to manufacturing and assembly aspects, rotordynamic and mechanical criteria, costs and scalability. These are two variants in barrel design (1,3) and one concept with a horizontal split joint (2).

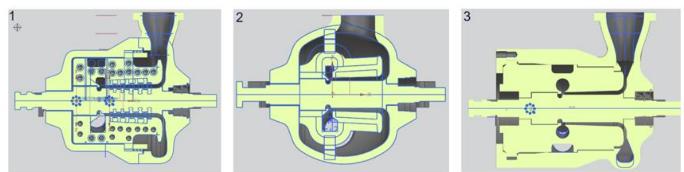

Figure 2: Comparison of design concepts for a sCO2 turbine

In principle, barrel type turbines exhibit lower radial deformations and are therefore particularly suitable for high-pressure applications. However, production and assembly are comparatively complex. In this respect, turbines with a horizontal parting line are more favorable, but they react less favorably to high pressure differences. Consequently, it can become quite challenging to avoid leakage at the half-joint for such a design. Particularly the gas tightness to the ambient is seen to be a decisive factor, which could not be fully guaranteed for design 2. The relatively large space that needs to be provided for the dry gas seals makes it hardly possible for design 2 to place the stud bolts at the half joint in such a way to avoid leakage. Additionally, this also imposes further difficulties with respect to scalability. This described contact separation is shown in figure 3 for design 2.

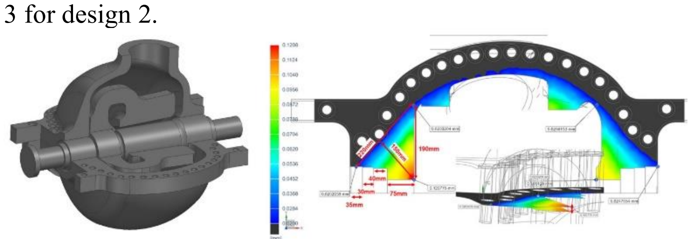

Figure 3: Contact separation at horizontal joint for design 2

The aerodynamic design, which will be discussed in more detail in the following section, is also unfavorable in design 2. Finally, the larger radial deformation leads to larger requirements in radial clearances and therefore give rise to higher losses. The resulting decision matrix according to table 1 documents the choice for design 1 as the favorable concept for the sCO2 turbine

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Table 2: Decision matrix for the three designs

	Gas Tightness	Pressure Losses	Rotorforce	Procurability	Manufactur-ability	Assembly	Scalability	Sealing
D 1	+	++	+	+	+	o	+	+
D 2	--	-	o	+	++	++	-	--
D 3	+	++	+	+	-	o	+	-

As mentioned above dry gas seals have been selected and an in-House solution was integrated in the outer casing of the barrel turbine. Dry gas seals are referenced in most sCO_2 applications due to their low leakage, and friction losses [10] and in a previous assessment [11] it was as well concluded that dry gas seals seem to be the best technology for sCO2 turbines. On the other hand, the remaining leakages are not to be neglected from an operational point of view, as regular replenishment would be necessary, and a recovery system is inevitable. As sealing gas CO2 is chosen which is extracted from the cycle upstream the turbine and thus must be purified or treated to meet the necessary pressure and temperature requirements. The resulting system technology must safely supply the seal with gas in all operating and fault conditions. An illustration of the basic design for the 2MW CO2 turbine is shown in figure 4, which also shows the compactness of the machine very well.

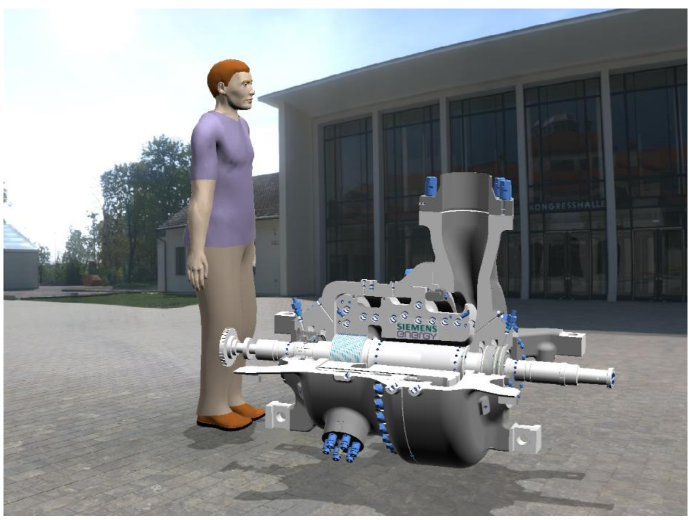

Figure 4: 2 MW sCO2 demo turbine

Furthermore, current dry gas seal concepts are limited in size so that for larger upscaling a further technology development might be necessary.

AERODYNAMICS – TURBINE INFLOW & EXHAUST

Particular attention must be paid to the aerodynamic design of such a sCO2 turbine, since the specific thermodynamic properties of the working medium show major differences compared to steam. Especially the high fluid density leads to increased flow losses. Under the simplifying assumption of an incompressible fluid the pressure loss can be deduced from energy conservation and written as,

$$\Delta p_{tot} = \zeta \frac{\rho}{2} c^2 \qquad (1)$$

with ζ being the dimensionless pressure loss coefficient, the density ρ and the flow velocity c. Although, the loss coefficient is depending on the Reynolds Number, this effect is not very strong, when comparing a regular steam cycle to sCO2. The Reynolds Number is already very high for steam and thus the loss coefficient is not changing much for higher Reynolds Number as present in this application. However, it can be seen that the pressure loss is linearly proportional to the density. Depending on the exact thermodynamic boundary conditions, the fluid density of the presented turbine is up to 4 times higher than the density in a comparable steam driven turbine. Thus, it is necessary to reduce the flow velocity by a factor of roughly two, in order to keep the overall losses on a similar level. Additionally, care should be taken to avoid regions with locally high flow velocities as these are usually the main sources for increased pressure loss. Particular attention is therefore paid to the aerodynamic design of the inflow and exhaust flow of the turbine.

The size of the turbine, and the corresponding mass flow, requires only one feed pipe to fulfil the demand on the flow velocity mentioned in the former section. From an aerodynamics point of view, an inlet volute is the preferred solution of such a one pipe arrangement, as the volute gives a very homogeneous flow field. Consequently, the losses in such a design are on a very low level as well (e.g. shown by Hecker [12]). The main advantage of design 1 over design 2 is the much easier possibility to realize this feature. In particular, the resulting torque from the flow in the volute can be transferred to the outer casing much easier in design 1, whereas in design 2 usually a dead-end opposite to the feed pipe is introduced to reduce the torque. However, this dead end mostly results in increased flow losses.

The flow field for both designs has been investigated with the help of computational fluid dynamics (CFD) methods to determine the exact flow losses. Total pressure and total temperature are set at the inlet of the model; the design mass flow is prescribed at the outlet. The upstream effect due to the flow blockage of the blading has been modelled via a porous medium. This can be shown to accurately predict the flow in the inlet chamber itself (see Sievert [13] and Hecker [12]). Meshing has been done with a hybrid approach, i.e., an unstructured tetrahedral mesh in the inlet chamber and an extruded mesh for the porous medium. All walls adjacent regions are meshed with prism elements to accurately account for the boundary layer of the flow. The overall mesh size is approximately 3 Mio nodes. Grid element size and boundary layer size were defined according to [15]. CFD calculation has been done as steady state analysis using Ansys 19.2 and CFD solver CFX. All walls have no-slip boundary condition. Convergence has been considered to

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be achieved when average residuals were less than 10^{-4} and mass flow and energy imbalances were below 0.01%. Turbulence was modeled applying k-ω turbulence model with wall functions (Hecker [11]). Accurate resolution of the boundary layer can be shown from the y^+ values (dimensionless wall distance of mesh node next to wall) which have a maximum of 45. This is well within the logarithmic region of the turbulent boundary layer and therefore sufficient for the usage of wall functions. The fluid is modelled with real gas properties from a gas table using the REFPROP [14] library. Spatial discretization is done using a hybrid scheme of almost second order accuracy.
The flow field for both design variants is shown in figure 5

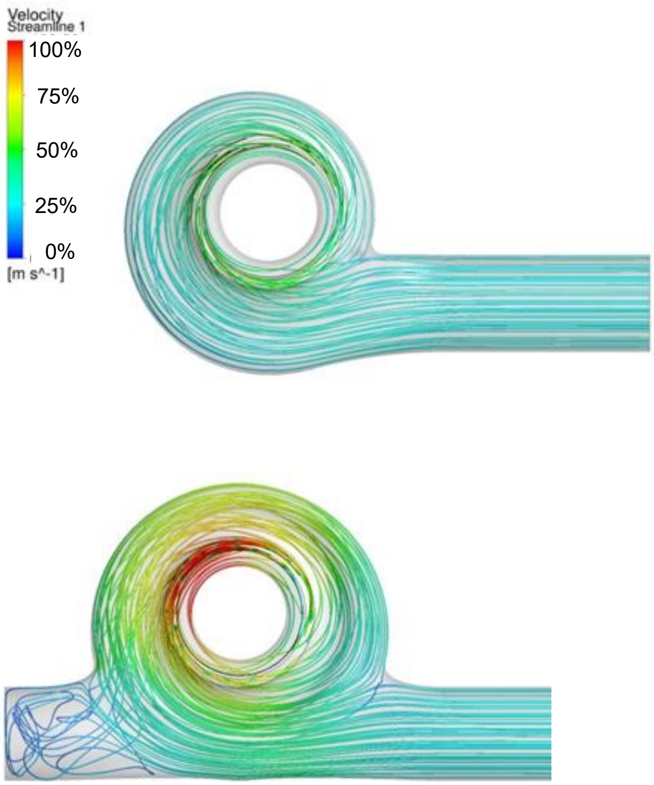

Figure 5: Comparison of inlet flow fields for design 1 and 2

It can be seen from the streamline plot, that the flow field in the volute of design 1 is much more homogeneous around the circumference, with flow incidence angles to the first stator vane only varying between -6° to 3°. Additionally, the maximum velocity in design 2 is much higher. Consequently, the pressure loss coefficient for design 1 (ζ=0.2) is reasonably lower compared to design 2 (ζ=2.1). As a nice-to-have the resulting forces on the rotor are also lower for design 1.

The turbine exhaust flow has also been investigated with care to reduce pressure losses and transient effects caused by flow separation in the diffuser section of the exhaust. A preliminary layout of the diffuser has been done with the help of a very simplistic 2D-CFD approach based on a potential flow solver combined with a boundary layer solver to predict flow separation. This preliminary design is then investigated and further improved with a fully 3D-CFD approach like the one described in the former section. The velocity profiles in axial, radial and circumferential direction, and the total temperature, which are taken from a separate simulation of the last stage alone, are used as boundary conditions at the diffuser inlet. At the outlet of the exhaust the static pressure is prescribed. Again, a hybrid mesh with prism elements in the near wall region has been used, resulting in an overall mesh count of 6 million nodes. Turbulence closure has been done using the k-ω turbulence model (compare Musch [15]). Fluid properties are taken from the REFPROP library again. The approach for discretization and convergence control are the same as applied for the inlet. It can be seen from the flow field in figure 6, that the objective to avoid flow separation in the diffuser could be achieved. Overall flow losses are on an acceptable level, with a pressure loss coefficient of ζ=0.3 according to the definition above.

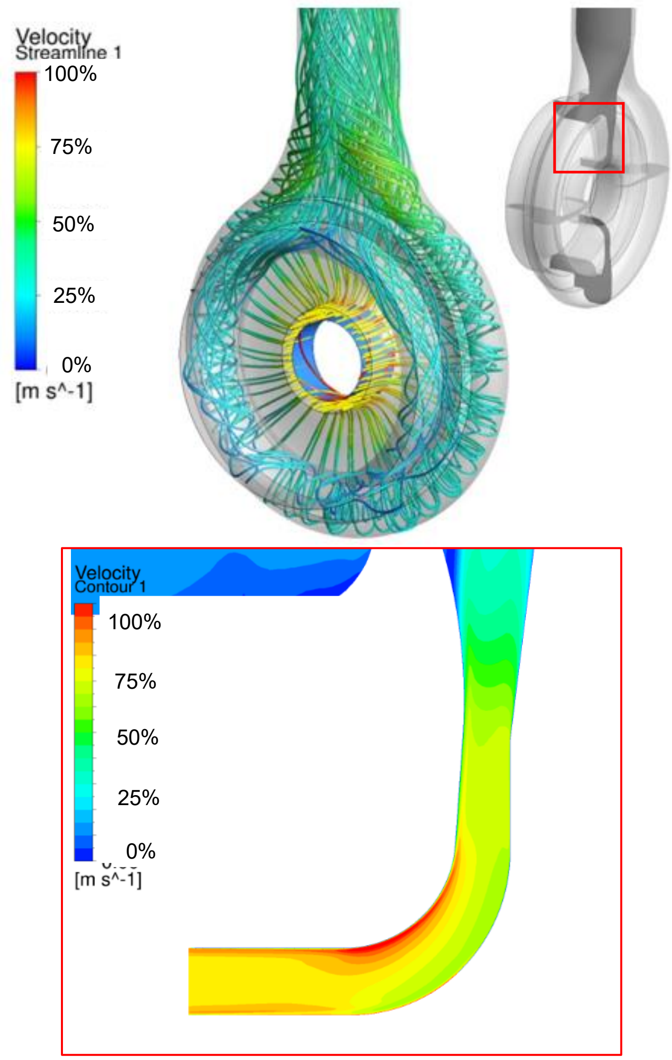

Figure 6: Flow field in turbine exhaust and diffusor

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ROTORDYNAMICS

As for the aerodynamics design, also for the rotordynamic calculations special care must be taken to consider the more challenging fluid properties of sCO2. These fluid properties have a major impact on the fluid forces on the rotor and therefore on possible flow excitation in the seals. The very high fluid density, and the low viscosity, do not allow to use the same semi-empirical methods, applied for steam, to determine the rotordynamic coefficients, which are vital for the assessment of the shaft train. As the standard correlations could not be applied directly, it has been decided to evaluate all seals for the demonstration turbine with the help of 3D-CFD. As an example, the numerical approach is described for the dummy piston seal. Figure 7 shows an overview of the examined sealing section. An eccentricity of 10µm has been assumed for the simulations. Meshing is done applying a sweep method in circumferential direction. Again, an eddy viscosity approach has been applied for turbulence closure, this time using the Baseline (BSL) model by Menter [16], which has been proven by Musch [17] to produce reliable results for CFD simulations of labyrinth seals. From the calculated forces on the shaft, the rotordynamic coefficients for stiffness and damping of the seal can be derived and considered in the shaft train design.

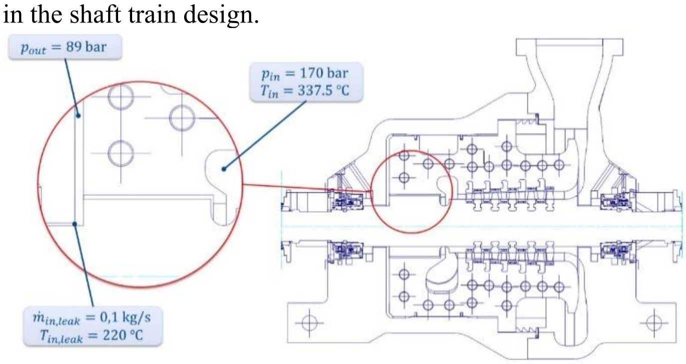

Figure 7: Cross Section of turbine wit close-up of dummy piston

MATERIALS

For the design and the manufacturing of a turbine, operating with supercritical CO2 as working fluid, materials must be selected assuring a safe and reliable operation. To answer the question, whether conventional materials used for (steam) turbines are also suitable for the operation with sCO2, the effects of oxidation and carbonization need to be evaluated. Many scientific studies were carried out in the field of interest in different institutes all over the world in the past about 50 years. Available oxidation data in sCO2 was assessed and published in [2]. In figure 8, published in [2] a limit of suitability of $5x10^{-13}$ g^2/cm^4 s has been defined. However, corrosion rates of $5x10^{-12}$ g^2/cm^4 s are experienced for 9-12% Cr. steels in steam atmosphere. From that it is concluded that these material class can even be operated up to approximately 560 °C.

Positive corrosion resistance is reported based on small additions of silicon [18] and a further improvement can be achieved by oxidation protection layers. For further technology development these measures should be further evaluated by experiments. Initial measurement campaigns are carried out within the CO2OLHEAT project. However, for the above-described waste heat recovery application with inlet temperatures below 400°C, it is assumed that proven materials can be used without any restrictions.

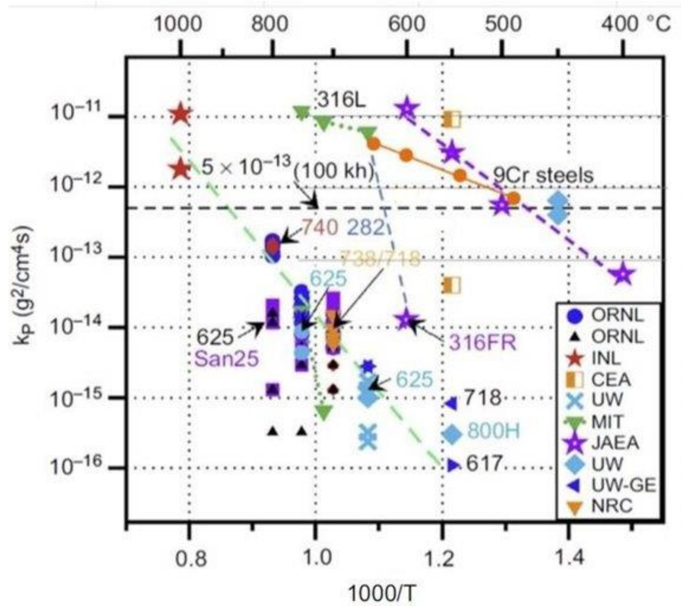

Figure 8: Corrosion rates of different martials in CO2 atmosphere [2]

SYSTEM INTEGRATION

To integrate the sCO2 turbine into the overall system, it is connected to a generator on a base frame via a gearbox and equipped with the necessary auxiliary systems such as the oil module and the sealing and leakage gas system. Figure 9 shows the overall package of the sCO2 turbine.

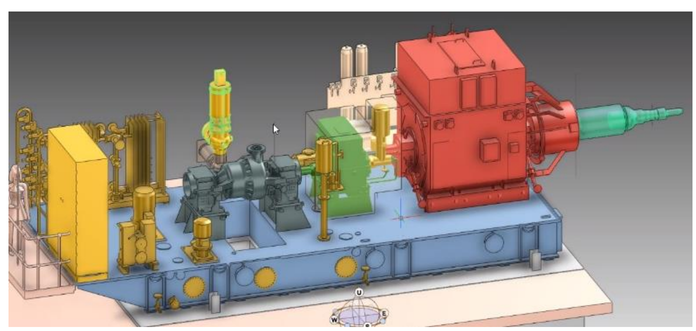

Figure 9: Preliminary set-up of complete sCO2 -turbine train

As already mentioned above, the sealing system is designed to minimize leakages and assures a safe operation in each operational condition of the plant. The chosen dry gas seal (DGS), integrated in the outer casing as shown in figure 7, requires clean sealing gas with defined requirements in terms of

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purity, pressure and temperature. For that, CO2 is extracted from the high-pressure side of the cycle and processed as shown in the preliminary and simplified P&ID in figure 10, whereas redundancies of the components needs to be considered in order to assure a reliable operation of the system. The main portion of leakage gas discharged in a first stage at low overpressure and sent to the recovery compressor of the system. The remaining leakage of $<<10^{-4}$ kg/s is vented into the atmosphere.

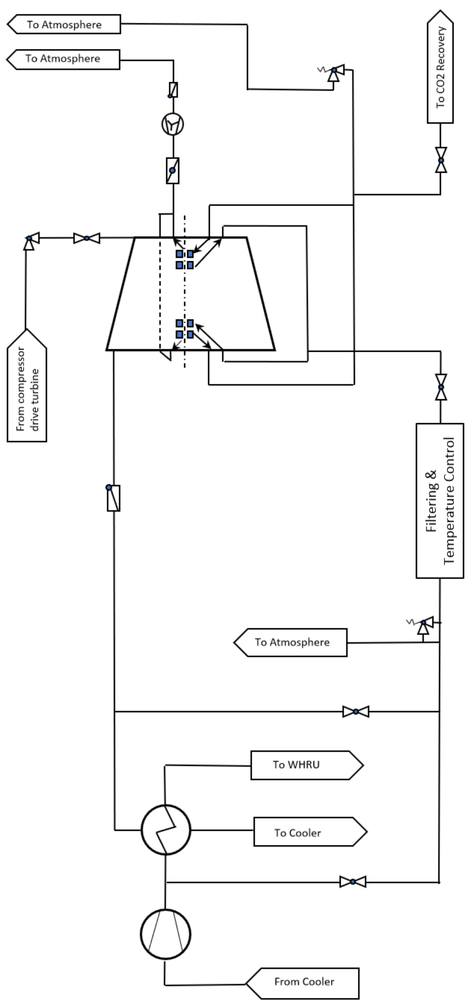

Figure 10: Preliminary and simplified P&ID of sealing system

In addition to the development of the components, the integration and coordination of the respective designs is one of the main challenges in the development of closed Brayton cycles. In this context, off-design load cases, transient processes such as startup and shutdown, and malfunctions are of particular importance. In contrast to the water-steam cycle, there is no integrated storage of the working medium, such as in the hotwell of the condenser or in the feedwater tank. Such an accumulator is therefore essential to enable transient processes such as start-up or shut-down. The further elaboration of the process technology including the corresponding logics and control loops of the entire system will be an essential focus of the joint development of the CO2OLHEAT consortium for the next phase of the project.

UPSCALING

The presented design of a 2 MW sCO2 demo turbine is at the lower end of the sensible application range of axial turbines. Due to the low volume flow and thus low blade-heights, the leakage losses are dominating and leading to a moderate efficiency. However, the goal is to validate a scalable design that can be applied for larger-scale power range. In table 2 the basic results for upscaling the 2MW bladepath design to 10 MW and 50 MW is shown. This leads to a significant improvement of the achievable efficiency. A more detailed description of the blading design methodology can be shown in [11].

Table 2: Basic results of upscaling the developed turbine design

	2 MW	10 MW	50 MW
Di	160 mm	300 mm	500 mm
$h_{1st\ stage}$	16 mm	35mm	60 mm
stages	6	4	11
n	175 Hz	110 Hz	50Hz
η_{Bld}	~ 83 %	~ 91 %	~ 93 %

In figure 11 the sketch of the 50 MW variant is compared with the 2 MW demo turbine in terms of size and the compactness of the 50 MW variant becomes obvious. Upscaling the power output by a factor of 25 increases the rotor length by a factor of ~ 2 only.

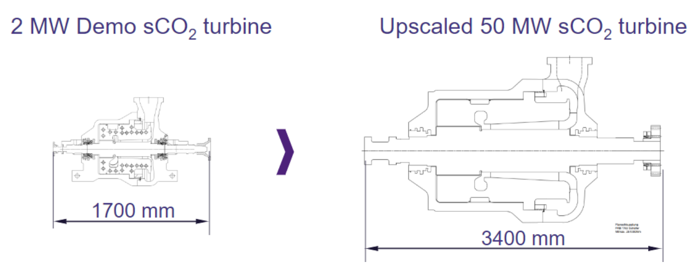

Figure 11: Comparison of scaled 50 MW turbine with 2 MW demo turbine

The basic feasibility of the scaled variants was investigated, whereby technological aspects must be the subject of further investigations and optimizations. For example, the sealing concept must be examined in detail for larger diameters and higher temperatures. In addition, the cycle architecture might be changed for larger applications due to potential restrictions of the

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serial connection of compressor drive turbine and power turbine for higher outputs and temperatures.

NOMENCLATURE

Bld	Blading
c	Velocity
CSP	Concentrated Solar Power
D	Diameter
DGS	Dry-Gas Seal
h	Height
P	Power [MW]
p	Pressure [bar]
P&ID	Piping & Instrumentation drawing
n	rotational speed [1/s]
sCO2	Supercritical Carbon Dioxide
T	Temperature [°C or K]
tot	Total
Δ	Difference
η	isentropic Efficiency (total/total)
ζ	Pressure loss coefficient
ρ	Density

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