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DESIGN CONSIDERATIONS OF SCO_2 TURBINES DEVELOPED WITHIN THE CARBOSOLA PROJECT

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ABSTRACT

Due to the great potential in terms of compactness and efficiency, supercritical CO_2 (sCO_2) power systems are being investigated in the field of scientific and industrial energy technology. The German research project CARBOSOLA has been initiated in order to drive the sCO_2 technology development in Europe by providing a test loop for research programs and basic component tests followed by an initial system and component design for a future demo plant. In a first project phase, the economic potential of the sCO_2 technology is evaluated and optimized for different use cases based on initial component designs and corresponding cost assumptions. For the CO_2 turbine, different design aspects have been investigated and assessed leading to first design concept.

INTRODUCTION

The CARBOSOLA project presented in this paper is intended to represent the entry into sCO_2 technology development in Germany. As shown in figure 1, the project structure is divided in two phases. In the first phase an analysis of the expected advantages is carried out. Therefore, the sCO_2 technology will be compared with conventional technologies in the fields of waste and exhaust heat recovery (WHR) and solar-thermal power plant technology (CSP) and subjected to a technical-economic evaluation and optimization. Waste heat recovery is expected to be the most promising market for sCO2 applications in Europe. CSP has been the target application of the DOE-funded SunShot program driving the sCO2-technology development in the United States. The envisaged technology comparison is intended to show what increase in efficiency can be expected when using sCO_2 compared to water/steam and what the electricity production costs are. However, the core of the project is the second phase including the component and system design of a technology demonstrator for the use of secondary heat as well as the theoretical and experimental methods required for further technology development up to commercial maturity. Therefore, the development and commissioning of amodular sCO_2 test loop for generic experimental studies and component test at the Helmholtz Zentrum Dresden Rossendorf are completing the program.

The consortium of the BMWi funded project CARABOSOLA consists of TU Dresden and the Helmholtz Zentrum Dresden-Rossendorf covering the necessary scientific-technical investigations for technology and product development. Furthermore, DLR provides the expertise for the evaluation of solar thermal power plants and Siemens Energy has a high level of competence in the field of thermal energy conversion systems and covers the necessary know-how of all components involved such as turbines, compressors and heat exchangers.

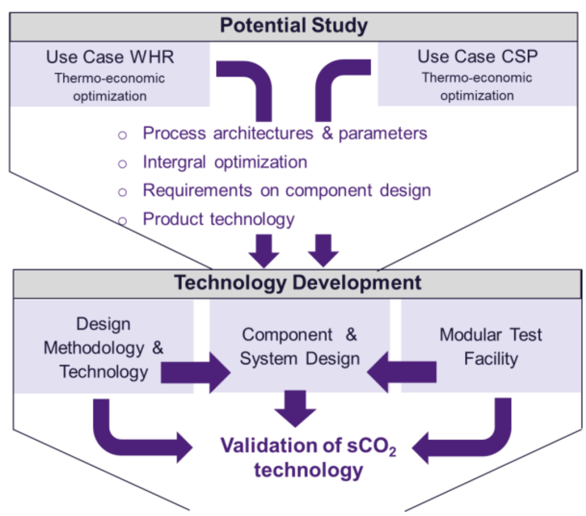

Figure 1: Structure of the CARBOSOLA project

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The focus in this publication are the turbine design considerations derived within the first two work packages, i.e. the potential studies for a waste heat recovery application and a concentrated solar power plant. Design details considering design topology, technology and optimization approaches are discussed.

Beside small lab scale applications, only few sCO2 turbomachinery designs have been realized so far. The only commercial system is the Echogen EPS-100 for waste heat recovery applications (up to 530°C turbine inlet temperature), which utilizes radial inflow turbines to generate approximately 8 MW [8]. However, a radial turbine topology is not likely to be optimal for large scale power generation plants. Within the Sunshot program a 10MW test loop is currently erected using an axial turbine concept [9]. Other conceptual designs studies for large scale sCO_2 turbomachinery have scaled up this concept for 50MW and 450 MW [10]. These concepts are addressing high temperature CSP applications (> 700°C turbine inlet temperature) which constrains the mechanical design to a certain extend.

50 MW SCO_2 TURBINE FOR WHR APPLICATION

For the waste heat recovery use case, the exhaust heat of two aeroderivative gas turbines was defined as heat source for the optimization of a sCO_2 cycle. Those gas turbines are often applied for mechanical drive of compressors in the oil and gas industry. Based on preliminary results of a techno-economic cycle optimization [1], whereas the cycle consists of one recuperator and a split primary heater, an initial turbine design has been carried out. The thermodynamic boundaries of the considered design case are listed in table 1.

Table 1: Thermal boundary conditions for a sCO_2 turbine in use case one (WHR)

Thermal boundary condition	Value
Inlet pressure [bar]	240
Inlet temperature [°C]	350
Mass flow [kg/s]	427
Inlet volume flow [m³/s]	2,1
Outlet pressure [bar]	64
Nominal shaft power [MW]	51

A first approach of the turbine geometry was derived by scaling a high-pressure barrel type steam turbine, whereby the scaling factor fs is derived equation 1.

$$f_s = \sqrt{\frac{\dot{V}_{Steam}}{\dot{V}_{CO2}}} \qquad (1)$$

With this methodology, i.e. dividing all lengths and diameters and multiplying the rotational speed of the original turbine with fs, the axial and circumferential flow velocities inside the turbine are approximately equal to the initial, high efficient steam turbine leading to an appropriate geometry for further optimizations. In figure 2 a general sketch of the obtained turbine design is shown. The turbine topology is a single-flow, double-shell design with inner and outer casing. The outer casing is of the barrel type with a circumferential split which allows rotational-symmetrical design without local material buildup even for high steam temperatures and pressures thus avoiding unsymmetrical deformation and thermal loading. Since the inlet temperature is quite low this design is assumed to be conservative and provides potential for further cost optimization in a next phase.

The drum blading comprises only 7 reaction stages whereby the exact degree of reaction is optimized individually for each individual stage (3DVTM: 3D blade with variable stage reaction). Figure 3 shows a sketch of the bladepath. Due to the scaling approach reasonable blade heights can be achieved leading to isentropic stage efficiencies up to 92,5 %.

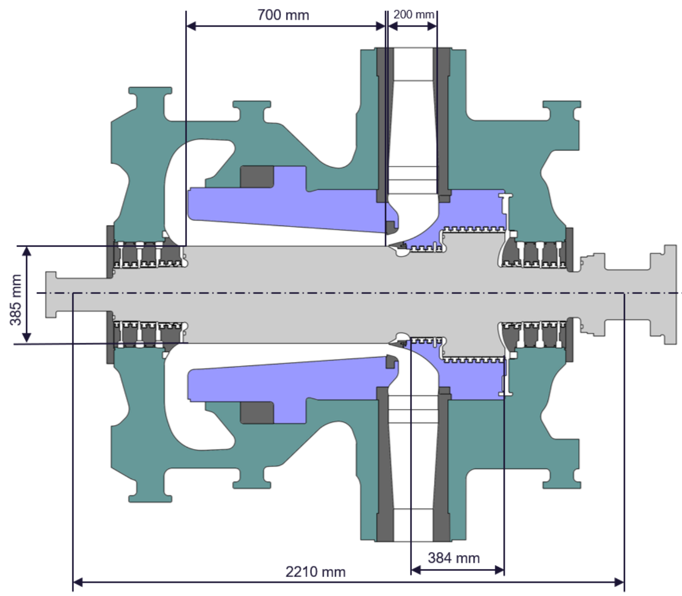

Figure 2: Sketch of the scaled high-pressure turbine

The use of 3DVTM blading represents the most advanced method of optimizing the high-pressure and intermediate-pressure blading. Siemens no longer works on the premise which has been applied for over a hundred years stipulating that the degree of reaction in a stage is an input variable for design calculations.

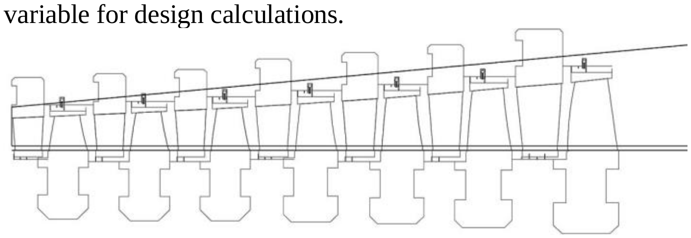

Figure 3: Sketch of the CO_2 turbine bladepath consisting of 7 stages 3DVTM blades

The use of numerical optimization methods has enabled Siemens to individually vary stage reaction and stage loading for every stage in order to obtain maximum efficiency. This allows efficiency to be increased by up to one percent as compared to blading with 50 % reaction. The 3DVTM blading is equipped with state-of-the-art 3D blade profiles such as 3DS blades (3DSTM: 3D blade design with reduced Secondary losses) for highest efficiencies.

In figure 4 an exergy loss analysis of the initial turbine design is shown. It can be deduced that in addition to the losses of the blading, the proportions of inflow, thrust balance piston and leakage cannot be neglected. It must be emphasized that the loss portion of the leakage does not include any additional power for potential recompression. A more detailed consideration of this aspect will follow in the further course of this paper.

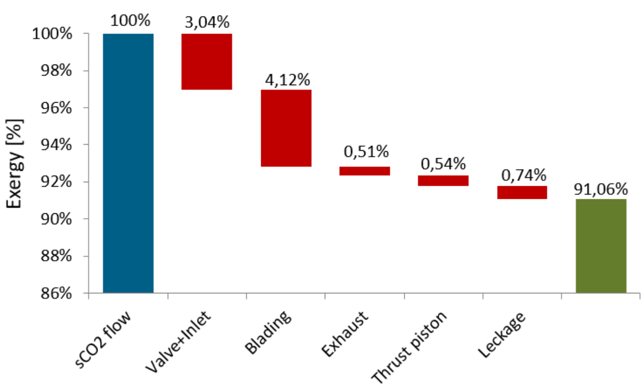

Figure 4: Exergy loss analysis of initial turbine design

Due to the high density of the working medium CO_2 associated with high losses at high velocities, special attention must be paid to the optimization of the inlet and exhaust. geometry Compared to a steam turbine for the same application (and thus same net output) and assuming same flow velocities and the same pressure loss coefficient of the inlet geometry the exergy loss of the CO_2 turbine would be approx. 16 times higher than the exergy loss of the steam turbine as documented in table 3.

Table 3: Exergy loss calculation for a generic pressure loss with ζ=1 and 50 m/s for a CO_2 turbine and a steam turbine

	CO_2 turbine	steam turbine
mass flow [kg(s]	470	29,4
pressure [bar]	250	49
temperature [°C]	350	350
Density [kg/m³]	213,6	18,8
ζ [-]	1	1
velocity [m/s]	50	50
pressure loss [bar]	2,7	0,2
Δs [kJ/kg K]	0,0020	0,0020
Exergy loss [KW]	272	17

Based on the above described scaling approach an initial inlet diameter of approximately 2 x 130 mm is obtained corresponding to a flow velocity of 78 m/s which is a common value for steam turbines. However, assuming the same flow velocity in the inlet valves this leads to an exergy loss of approximately 1,6 MW. In figure 5 the results of a calculus of variations is summarized. It is obvious that a reduction of the inlet flow velocity can reduce the exergy losses significantly. Balancing losses and design requirements, an inlet diameter of 200 mm was corresponding to a flow velocity of 33 m/s was chosen.

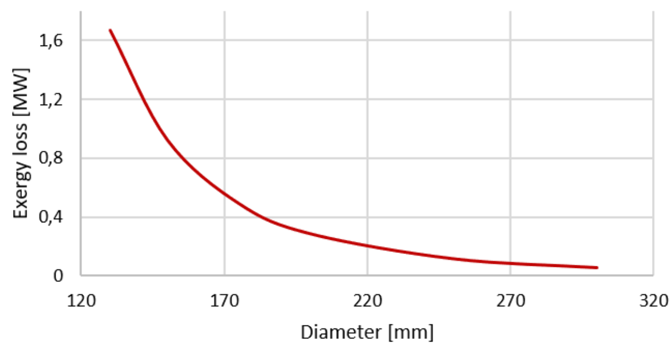

Figure 5: Exergy losses caused by pressure losses of the inlet and valve depending on the diameter

Beside the impact on the pressure losses, increased flow velocities may have an impact on the rotodynamic stability of the turbine. According to Hecker [2] asymmetrical inflow geometries may lead to local velocity differences which cause radial pressure differences. Due to the high fluid density these pressure differences are significant higher compared to steam turbines. The aerodynamic design of inlet and exhaust is therefor of particular importance. The influence of flow-induced lateral forces must be investigated in detail in further steps.

Another loss contributor according to figure 4 is the thrust equilibrium piston which compensates the axial forces of the moving blades due to the reaction blade technology. The entire pressure difference of the turbine is applied over the piston. The resulting leakage flow of the contactless seals, e.g. labyrinth seals, results in an exergy loss due to the throttling of the flow. According to Hecker et al. [3] no isenthalpic throttling over the sealing can be assumed and fluid friction effects need to be considered. The gas friction in labyrinth seals can be expressed by the wall shear stress by the following equations according to [3].

$$\boldsymbol{\tau} = \boldsymbol{cf} \times 0{,}5\, \boldsymbol{\rho w^2} \quad (2)$$

$$\boldsymbol{cf}\,(\boldsymbol{Re}) = \boldsymbol{A} \times [\![\boldsymbol{Re}]\!]^{(-0{,}2)} \quad (3)$$

This effect, which is well known from steam turbines, has been evaluated in a comparative calculation for a rotor geometry according to table 4 for both a steam leakage flow and a CO_2 leakage flow. The resulting windage losses and the related sealing mass flows are shown in figure 7 whereas the temperature increase over the isenthalpic temperature is provided in figure 6. The temperature rise of the CO_2 leakage flow is 13 K higher compared to the steam case. Since the CO_2 leakage flow across the sealing is 6 times higher compared to the steam leakage, the higher windage losses result only in this moderate temperature rise. Therefore, special attention must be paid, when optimizing the piston sealing, especially for higher turbine inlet temperatures. The resulting temperature rise must be considered in the mechanical design of rotor and casing. For the investigated WHR application this aspect is assessed to be uncritical due to the comparable low inlet temperature of the turbine.

Table 4: Parameters of a labyrinth sealing used for a comparison of the fluid friction effect for steam and CO_2

Parameter	
Mean labyrinth diameter [mm]	430
Length of labyrinth [mm]	384
Rotor frequency [1/s]	96

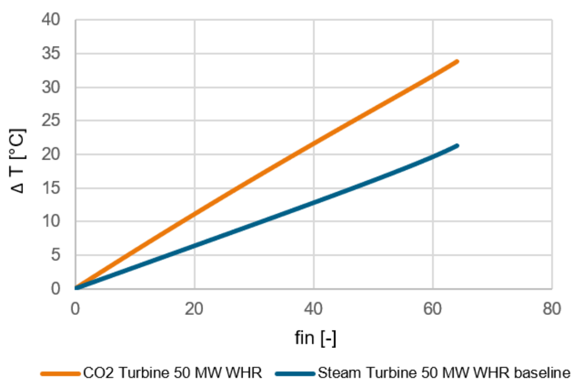

Figure 6: Temperature increase over the isenthalpic temperature for a labyrinth sealing for steam and CO_2

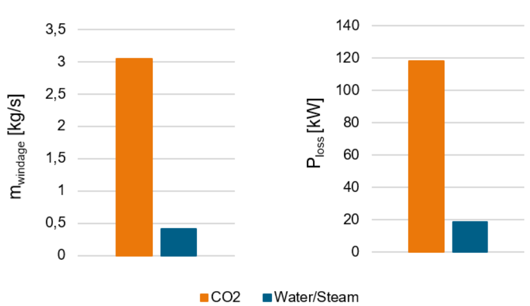

Figure 7: Comparison of leakage flow and fluid friction losses for a CO_2 turbine

For the optimization of the thrust piston geometry for the turbine design according figure 2 the length of piston equipped with labyrinth seals was stepwise increased leading to a reduction of the leakage flow. However, taking the overall exergy loss including the fluid friction into account an optimum piston length of approximately 330 mm, was chosen as shown in figure 8.

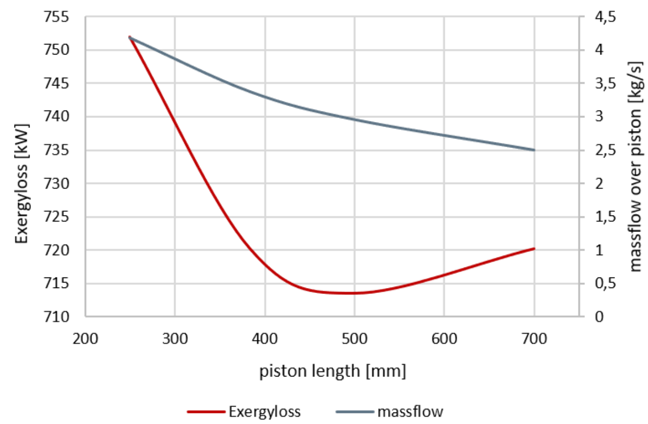

Figure 8: Impact of the piston length on the exergy loss taking fluid friction into account

Another important turbine component for realizing high efficiencies is the sealing technology. Dry gas seals are referenced in most sCO_2 applications due to their low leakage, and friction losses [5]. On the other hand, conventional turbine sealings such as labyrinth and brush sealings are beneficial with respect to costs and reliability. Assuming coated labyrinth sealings as baseline, a first approach of the sealing system was defined according to the sketch in figure 9.

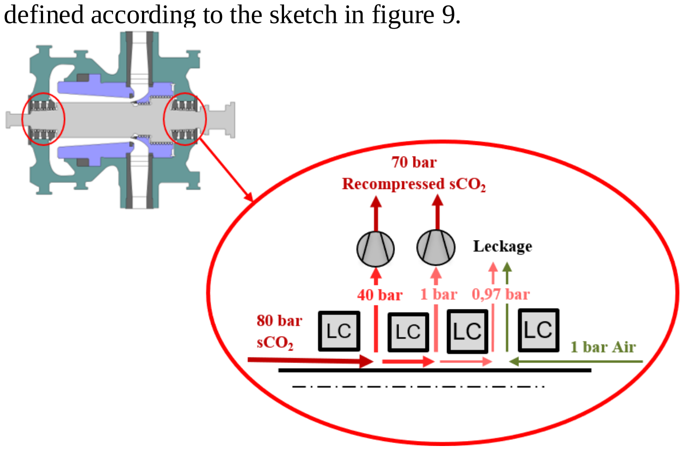

Figure 9: Initial calculation of sealing system based on coated labyrinth sealing technology (LC)

In this layout the CO_2 is throttled from the lower cycle pressure of approximately 80 bar via two sealings to defined pressure levels of 40 bar and 1,03 bar (slight overpressure) where the sealing flow is partly recompressed and piped back into the power cycle. For the chosen geometry and pressures the required aux load amount to approx. 850 KW. At the end of the shaft, a slight vacuum is set between two further sealing elements in a suction hood from which the CO_2/Air mixture is discharged. In this case the CO_2 leakage amounts to 1,3 tons CO_2 per day outside the machine house. Both the high demand for auxiliary energy and the leakage rate show that conventional sealing technology is not sufficient to seal CO_2 turbines. In a next step brush seals have been considered for the first two sealing sections at each shaft end, but the results could only be improved slightly. In contrast to that the use of a dry gas seals results in a significant reduction of the leakage flows as summarized in table 5. Furthermore, no auxiliary compressors are required in this setup. On the other hand, dry gas seals require a clean gas supply [3] as shown in figure 9. It was assumed to extract this gas supply from the high-pressure line behind the recuperator and connect it via a pressure reduction and filter device with the sealing system. Overall, it can be stated that dry gas seals seem to be the best technology for sCO_2 turbines. However, the remaining leakage is not to be neglected from an operational point of view, as regular replenishment is necessary. Further system improvements must therefore be investigated and developed.

Table 5: Thermodynamic comparison of different sealing technologies for 50 MW sCO_2 turbine

Parameter	Labyrinth	Brushes	DGS
Leakage flow [t/d]	1,3	1,3	0,52
Aux. power [kW]	852	436	-
Exergy loss [kW]	1502	811	108

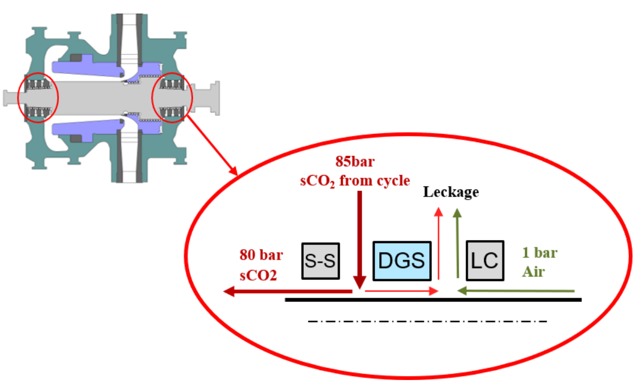

Figure 10: Dry-gas-sealing (DGS) system combined with tip to tip seal (S-S) and a coated labyrinth (L-C)

With the above described improvements, the power weighted isentropic efficiency (including leakages) could be improved from initial 87,5 % to 90,5 % as illustrated in figure 10. Compared to the reference cycle based on water/steam this is approximately 5 % pts better compared to a steam turbine for the same use case. Due to the short bearing span and the by orders of magnitude lower volumetric flow at the exhaust, the estimated cost are about 50 % less than that of a steam turbine.

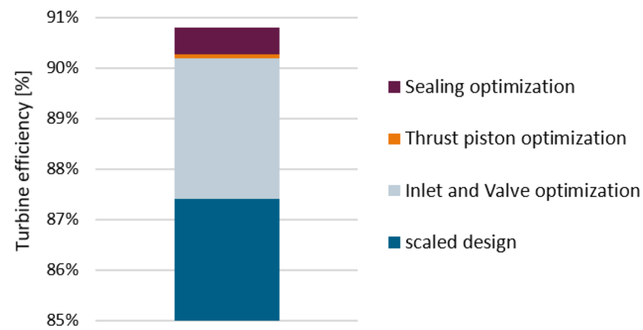

Figure 11: Isentropic turbine efficiency and breakdown of optimization measures

150 MW SCO_2 TURBINE FOR CSP APPLICATION

The second use case which is evaluated in the CARBOSOLA project is a concentrated solar power plant (CSP). With a similar methodology a preliminary techno-economic cycle optimization has been carried out for different upper process temperatures [5]. For a base case at 550 °C turbine inlet temperature and a molten salt based thermal storage system a simple recuperated cycle has been identified as the most economic cycle configuration. The thermodynamic boundaries of the turbine design case are listed in table 6. For comparison purposes the values of the steam turbine in the reference cycle are also listed.

Since the pressure levels and the volumetric flows at the inlet and the outlet fit very well to large high pressure steam applications the turbine topology can be derived from a Siemens H60 turbine, which is a single-flow, double-shell design with inner and outer casing operating at net frequency. With an optimized shaft diameter of approximately 650 mm only 13 reaction stages based on 3DS technology are necessary to provide 148 MW. Under consideration of all pressure losses, internal and external leakages, assuming DGS technology as described above, the power weighted isentropic efficiency amounts to 92 %.

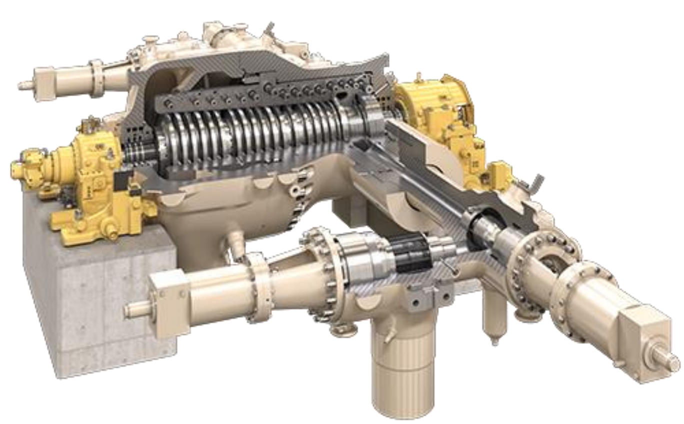

Figure 11: High-pressure turbine from current Siemens Energy portfolio adapted for CSP application with 115 MW net-power.

The compatibility of materials is a major concern for the design and operation of sCO_2 power cycles and its components. The temperature depending corrosion behaviour of materials in CO_2 environment is subject of numerous research studies and Ni-base alloys are discussed as suitable materials. However, these studies are focussing essentially high temperature application of 700 °C or higher [6,7]. An assessment of the turbine material selection at moderate temperatures is currently part of an investigation within the CARBOSOLA project. For the above described base case with 550 °C turbine inlet temperature it is assumed that proven materials such as 1 % chromium steels for shaft and casings can be used.

Table 6: Thermal boundary conditions for a sCO_2 turbine and a steam turbine for use case 2 (CSP)

Thermal boundary condition	CO_2 turbine	Steam turbine *)
Inlet pressure [bar]	260	170/34
Inlet temperature [°C]	550	550/550
Mass flow [kg/s]	946	85/79
Inlet volume flow [m³/s]	5,9	1,7/8,7
Outlet pressure [bar]	78	78/0,07
Outlet volume flow [m³/s]	15,6	5,7/797
Enthalpy difference [kJ/kg]	154,7	414/1154
Nominal shaft power [MW]	147	117

*) the second value corresponds to the reheat turbine

By comparing the parameters in table 6, some general statements can be made, which describe how sCO_2 turbines differ from steam turbines and what the special requirements are. Beside the well-known fact that the pressure levels of sCO_2 cycles are higher, especially at the low-pressure side, it is obvious that the expansion line, i.e. the enthalpy difference of the turbine is significant smaller compared to the steam turbine. Consequently, the mass flow in an sCO_2 cycle must be larger compared to a Rankine cycle assuming

the same power output.

In the considered use case, the reference cycle is operating also at a high-pressure level of 170 bar. Comparing the inlet volume flow of both turbines it can be seen that the value of the CO_2 turbine is approximately 3 times larger than the volume flow of the corresponding steam turbine. This leads to significant larger cross-sectional areas of the high-pressure piping and the inlet of the turbine. This difference becomes smaller when the reference steam process operates at a lower pressure as it is the case in the WHR example described in the first section of this paper.

On the other hand, the volume flow at the low-pressure side of the CO_2 turbine is several orders of magnitude smaller compared to the steam turbine. This results in a significantly lower space requirement within the nacelle, which significantly reduces secondary costs such as steel construction, foundations, etc.

Based of the derived turbine designs, a simplified efficiency model was derived in order to support further process optimizations within the two working packages of the CARBOSOLA project [1,2]. A simple mathematical relationship was fitted to data points, where the efficiency is a function of the volume flow and of the isentropic enthalpy difference. The dependency of the volumetric flow, which is well known from axial turbines, is mainly driven by leakage losses of the blades. Due to the comparatively small enthalpy difference across the turbine the impact of internal pressure losses, e.g. in inlet and outlet, and the impact of internal leakages is quite high as shown in the first sections of this paper. The smaller the enthalpy drop of a turbine, the greater the influence of these constant losses on the isentropic efficiency. In particular this consideration has led to a penalty of the turbine efficiencies in reheated cycle layouts of the CSP use case evaluation carried out by Heller et al. [5]. Figure 12 shows the efficiency curve for different inlet volume flows and isentropic enthalpy differences. The consideration of further influencing variables such as blading technology was not carried out for reasons of simplification.

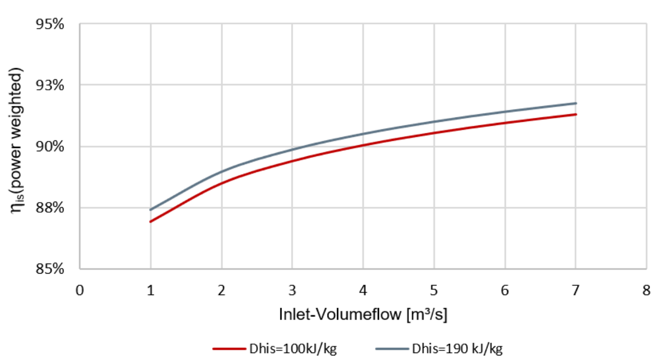

Figure 12: Isentropic efficiency of CO_2 turbine including all internal and external leakages (power weighted).

CONCLUSION AND OUTLOOK

In a first project phase of the Carbosola project, different aspects of the turbine design have been investigated leading to first design concepts for a waste heat recovery and a CSP use case. Due to the high density of the working medium CO_2 and the characteristic low enthalpy drop various aspects such as fluid friction, pressure losses, asymmetrical flow conditions and leakages are of greater importance than for steam or gas turbines and need to be further evaluated. All in all, very compact turbines can be achieved that are highly efficient and have a low footprint, especially due to the low volume flows at the cold end. Further investigations and design optimizations are envisaged in a next step of the project.

NOMENCLATURE

A	Constant
C	Windage coefficient
CSP	Concentrated Solar Power
DGS	Dry-Gas Seal
d	day
f	friction
is	isentropic
L-C	Coated labyrinth seal
P	Power [kW]
Re	Reynolds number
s	Entropy
sCO_2	Supercritical Carbon Dioxide
S-S	Fin-Fin seal
T	Temperature [°C or K]
t	tons
$\dot{V}$	Volume flow [m^3/s]
w	Velocity
Δ	Difference
η	Efficiency
ρ	Density
τ	wall shear

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Bundesministerium für Wirtschaft und Energie

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