DEVELOPMENT OF HIGH EFFICIENCY HOT GAS TURBO-EXPANDER FOR OPTIMIZED CSP SUPERCRITICAL CO_2 POWER BLOCK OPERATION

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Abstract

GE Global Research in collaboration with Southwest Research Institute is working on development of a supercritical CO_2 (sCO_2) turbo-expander for application to a sCO_2 based power cycle for concentrated solar power (CSP) conversion. The proposed cycle uses sCO_2 as both the heat transfer fluid in the solar receiver and the working fluid in the power block. The lower thermal mass and increased power density of the sCO_2 cycle, as compared to steam-based systems, enables the development of compact, high-efficiency power blocks that are compatible with sensible-heat thermal energy storage, and can respond quickly to transient environmental changes and frequent start-up/shut-down operations. These smaller, integrated power blocks are ideal for modular tower mounted CSP solutions in the 5-10 MW range. With funding under the Sunshot initiative, the authors are developing this high-efficiency sCO_2 turbo-expander for the solar power plant duty cycle profile for the sCO_2 Brayton cycle. The scalable sCO_2 expander design closes a critical technology gap required for an optimized CSP sCO_2 power plant and provides a major stepping stone on the pathway to achieving CSP power at $0.06/kW-hr levelized cost of electricity, increasing energy conversion efficiency to greater than 50%, and reducing total power block cost to below $1,200/kW installed. High power density of the sCO_2 working and its impact on turbine design are presented in detail.

Background

Because of the highly cyclical nature of the CSP plant operation, a sCO_2 hot gas turbo-expander must be able to operate at high efficiency over a wide range of part load conditions, must be able to handle rapid transient heat input swings, and have very fast start-up capabilities so as to optimize the plant's on-line availability. Several sCO_2 cycle options for CSP applications have been proposed over the last several years. One possible configuration of the sCO_2 cycle is shown in Figure 1.

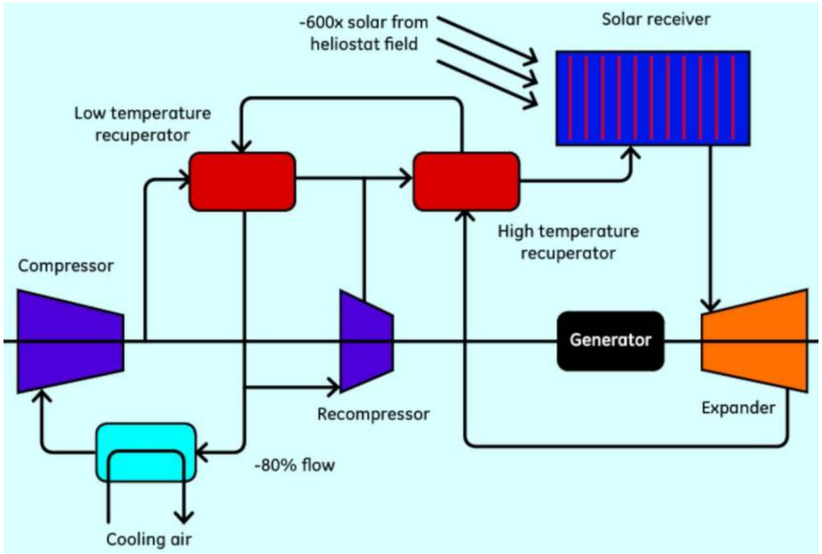

Figure 1: Recompression sCO_2 Cycle

This configuration uses sCO_2 as both the heat transfer fluid in the solar receiver and the working fluid in the power cycle. Brayton cycles using sCO_2 as working fluid have (a) a high degree of heat recuperation; (b) flat isobars near the critical point, and hence, a low average heat rejection temperature; (c) high fluid density near the compressor inlet, resulting in low compressor work; (d) relatively high temperature at the solar receiver; and (e) pressure ratio in this cycle is around three compared to fifty or more in a Rankine cycle. These advantages result in a highly efficient cycle in comparison to the Rankine cycle or steam cycles (supercritical or otherwise). The design is compatible with sensible-heat thermal energy storage, if desired. The lower thermal mass makes startup and load change faster for frequent start-up/ shut-down operations and load adaption than a steam based system. The high power density in sCO_2 enable power generation rates at comparatively much low volume flow rates, resulting in compact, low weight, and low cost power block machinery that can be placed on top of a CSP tower. This power density in sCO_2 is much higher compared to steam or air, enabling compact, lightweight, and low cost receiver and power cycle designs, making it ideally suited for a tower-mounted modular CSP solution in the 5-10 MW range (Ma, 2011).

The earliest uses of sCO_2 as a working fluid in a closed-loop recompression Brayton cycle was proposed by Combs in 1977 (Combs, 1977) for shipboard applications, in which Combs concluded that a substantial reduction in fuel consumption was possible. More recently, sCO_2 cycle testing has been performed at Sandia National Laboratories (Wright, 2010; Kolb, 2011; Conboy, 2012) and at Knolls Atomic Power Laboratory (DOE, 2012; Kimball, 2012). Sandia National Lab is currently operating an sCO2 test-loop to investigate the key technology issues associated with this cycle (Wright, 2010). In the testing to date, the turbo-expander has reached maximum speeds of 45,000 rpm at 315°C, peak flow rates of over 4.1 kg/s (9 lb/s), and pressure ratios of just over 1.65. The data from these tests indicate that the basic design and performance predictions for the recompression cycle are sound; however, the reliability and performance scalability of the turbo-machinery to the demanding requirements for high efficiency CSP applications are currently not addressed (Wright, 2010; Kolb, 2011).

To make this technology commercially viable, it is imperative to advance the design of the sCO_2 cycle from small laboratory scale to the multi-MW range. The size sCO_2 Brayton cycle is being designed to match current modular solar fields and has been identified as being commercially highly competitive (Ma, 2011). The oil and gas industry has developed technologies for compressing and pumping CO_2 at supercritical pressures for other applications, and hence, compression technology required for the sCO_2 Brayton cycle is considered a moderate risk (Moore, 2007). In contrast, industrial scale turbines for operation on sCO_2 do not have a precedent in the industry beyond some small demonstration radial turbomachinery units currently being run in labs.

This Sunshot Initiative program is funded by the Department of Energy (DOE) Office of Energy Efficiency and Renewable Energy (EERE) SunShot office under the CSP power block Funding Opportunity Announcement (FOA). The goal of this program is to meet the following performance and cost goals:

- Net cycle efficiency > 50%
- Dry cooled
- Power Block Cost < $1,200/kWe

Turbo-machinery Design

The power cycle operating conditions are optimized for a CSP plant operation in Dagget, CA as a reference location. For most relevant operation of a CSP power plant, the design point performance should be optimized for DNI (Direct Normal Irradiance) weighted ambient temperature. This would result in highest power block efficiency at most probable ambient conditions. The optimal turbine pressure ratio for given turbine inlet conditions is a strong function of the ambient temperature, therefore, the selection of design point ambient conditions is critical for turbine design activity. This section documents the aero design of a nominal 10 MW (net power) expander for a supercritical CO_2 power cycle designed for use in a concentrated solar plant. The intent is to capture the final design configuration only and not the many iterations that were made on the path to the final design.

Machinery component general layout

The overall power block for CSP installation using recompression CO_2 cycle has the following rotating machinery: expander, main compressor, re-compressor, and generator. These components can be organized in various different layouts and rotational speeds and the system configuration would provide different overall thermal conversion efficiencies for the same primary component designs. The layout options for recompression cycle turbomachinery are listed in Table 1.

Table 1: Turbo-machine layout options (IC: Inductively coupled, PM: Permanent magnet, 3600 rpm synchronous generator)

Option	Generator	Compressor	Turbine	RPM
High speed, Optimal	A. IC B. PM	A. Single stage centrifugal B. Multi stage pump	A. Radial B. Axial	Optimized for compressor
High speed, expander only	A. IC B. PM	None	A. Radial B. Axial	Optimized for expander
High speed, Geared	A. IC B. PM C. 3600 rpm	A. Single stage centrifugal B. Multi stage pump	A. Radial B. Axial	Both expander and compressor run at optimal speed
3600 rpm integrated	3600 rpm	Multi stage pump or compressor	Multi stage Axial at 3600 rpm	3600 rpm
3600 rpm – expander only	3600 rpm	None	Multi stage Axial at 3600 rpm	3600 rpm

In the table above all the possible configurations of the turbomachinery layout that can be used in a modular CSP power plant have been included. Using this table, the common components from each layout were selected for preliminary design to feed into rotor-dynamics analysis and initial down-selection.

1. Turbine wheels
    a. Axial turbine designs
    b. Radial turbine designs

2. Compressor wheels
    a. Centrifugal compressors
    b. Dense flow pumps
3. Generator
    a. Synchronous generators (low speed)
    b. Induction generators
    c. Permanent magnet generators
4. Generator cooling analysis
5. Pressure containment / casing design
6. Sealing system design
7. Bearing and rotor dynamics analysis

Using this analysis and component designs, a more detailed study of the 4 feasible designs was performed and the impact of layout was quantified on the overall system performance. The block diagram schematic of 4 feasible designs is shown in Figure 1.2.9. Down-selection activity required that the designs needed to be mature enough to understand the system level impacts of each layout, included detailed aero-design, generator selection, gearbox selection, compressor wheel selection (off-the-shelf), sealing system designs, rotor-dynamics and bearing designs for each design. These details are not presented here.

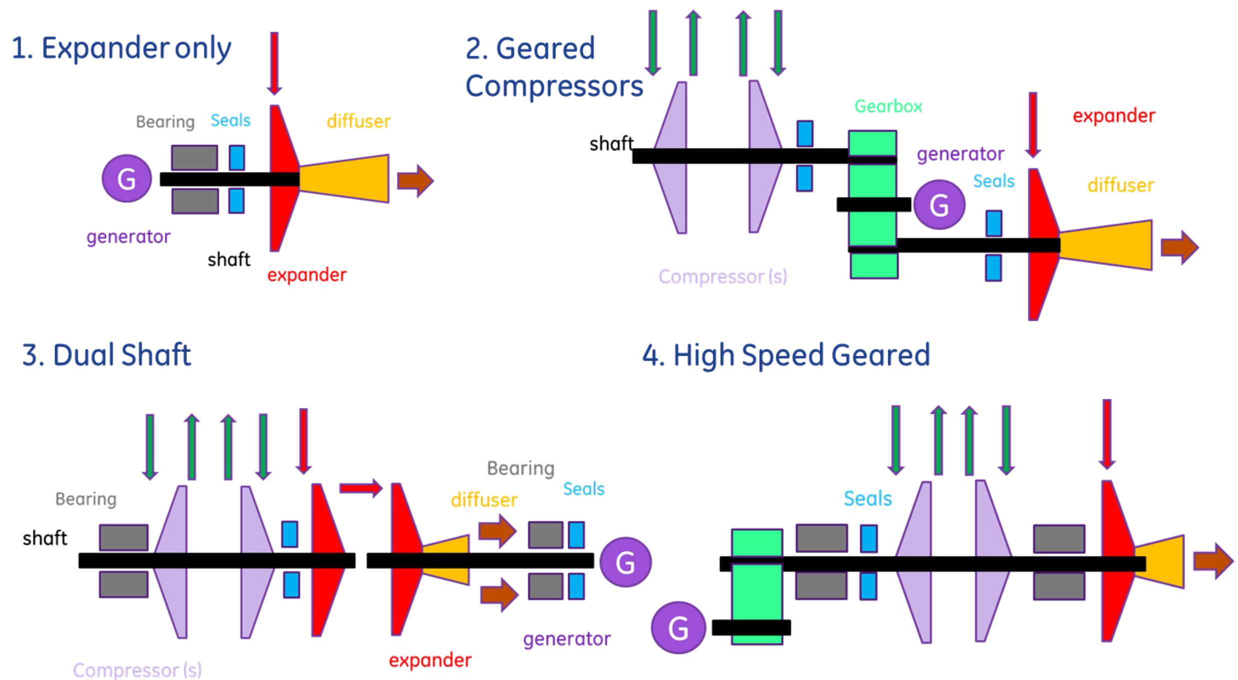

Figure 2: Block schematic of the four feasible layouts for recompression CO_2 Brayton cycle based modular CSP power blocks. (1) Direct drive or geared turbo-generator with undefined motor driven compressor, (2) Geared compressor train with direct drive or pinion geared generator, (3) a dual shaft concept with a single expander stage driving the compressors, while a second shaft with turbo-generator – direct drive or geared, and (4) a single shaft concept with both expander and compressor train running at same speed with a geared generator.

The down-selection of this turbomachinery architecture was based on trade-off analysis using performance and cost comparison divided into the following four (4) categories:
1. Annual energy production (AEP)
2. Cost of the machine and system cost
3. Operation and maintenance (O&M) cost
4. Commercialization criteria

The weightage of each criterion was derived from its weight going into the LCOE calculation. This resulted in clear identification of the best design to be pursed. In addition to the best design, a second option with significantly lower technical risk but also lower scores on the criteria list has been selected a back-up. Following the overall architecture down select, the design team primarily focused only on the turbine design with shaft speed, mass flow rate, leakage requirements, and efficiency targets as boundary conditions.

***Aero Design Tools**:* Initial design concepts were explored using a spreadsheet based tool due to the ease in accommodating the sCO_2 properties via NIST Refprop. Once a candidate design was selected, models were built in the traditional GE design system using TP3, CAFD, and BBP (all proprietary tools for aero design) with equivalent perfect gas properties. To verify the design a TACOMA (proprietary full scale CFD tool) simulation was run using both perfect gas properties and tabular properties for sCO_2 developed for this project using Refprop. TACOMA is the state of the art CFD tool for turbo-machinery flow analysis developed for axial turbomachinery design.

Overall thermal design requirements were established by the thermodynamic cycle for the concentrated solar power plant. In addition to the cycle requirements, mechanical design analysis was performed on intermediate aero designs providing feedback on the levels of static stress due to rotation. This resulted in the need to minimize the blade heights and tip diameters. Additionally, the bending stress on the airfoils due to the gas load from the dense high pressure CO_2 resulted in the need for mechanically robust blade designs. Throughout the conceptual design phase of the turbine several flowpath layouts were considered. Once the design configuration was down-selected to a multi-stage axial design, a study of several designs with 3 and 4 stage counts was conducted for mechanical shaft feasibility. To reduce mechanical stresses and improve efficiency a four stage design at 27000 RPM was chosen shown in Figure 3 and the 3D airfoil designs are shown in Figure 4.

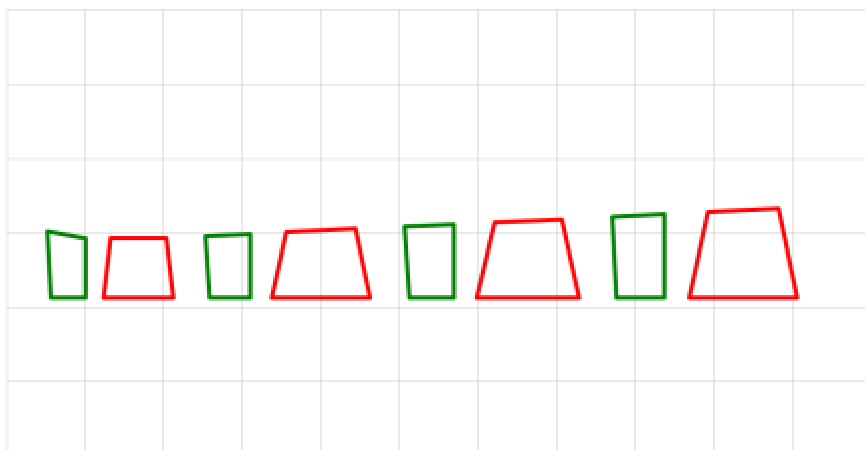

Figure 3: Preliminary flow path layout of the 4-stage axial turbine

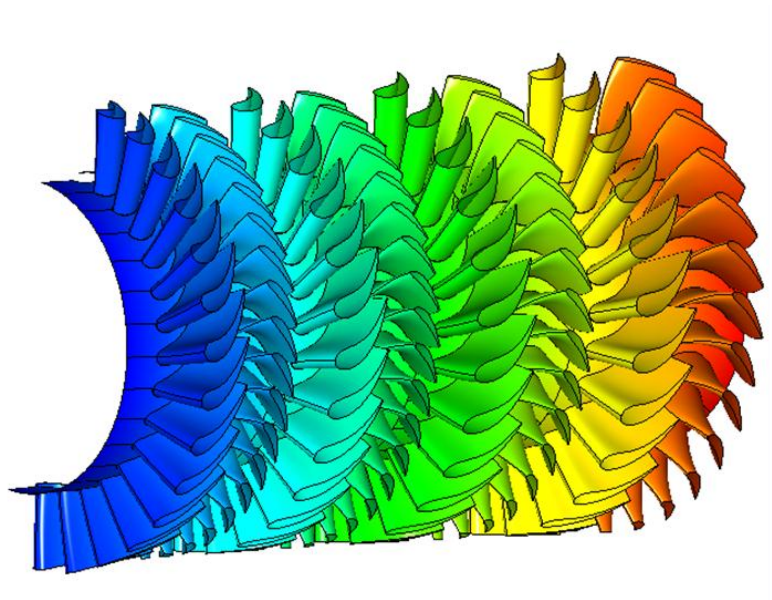

Figure 4: Airfoil designs for 4-stage SunShot turbine

CFD Analysis*:* To confirm the quality of the design, a 3D CFD analysis using TACOMA was performed. This analysis was done with two different fluid models, first with the perfect gas assumptions used in the design tools and second with a tabular CO_2 equation of state generated from the RefProp database specifically for this project. Computational meshes were generated for each blade row containing about a million grid points with an average y+ between 30 and 45. Comparisons of the blade row efficiency and overall turbine group efficiency between the perfect gas CFD computation and the tabular CO_2 CFD computation using the same boundary conditions and meshes show a slightly higher efficiency for the tabular CO_2 case.

Due to the extreme power density associated with a supercritical CO_2 turbine and the high temperatures needed to meet the SunShot cycle efficiency goals, the mechanical design of the turbine is extremely challenging. For the SunShot turbine, some features in the aero design were compromised to help meet the mechanical requirements as discussed in the next section.

Rotor Mechanical Design:

Blade Mechanical Design*:* The airfoil design went through number of iterations to meet geometric design criteria in terms of shape and form factor of the buckets. The bucket concentrated and average stress numbers were then evaluated and compared to selected material properties at temperature. Proprietary methods were used to calculate the representative stress numbers in the airfoils and below the material allowable stress for > 90,000 hour (30 years) rotor life.

In traditional axial turbine applications, blade structural vibrations are damped using frictional forces at the dove tails (location of blade assembly to the rotor. In this application due to high power density in the working fluid (CO_2), an integral shaft design is being pursued. Various options to introduce frictional damping to mitigate the vibrations were evaluated including z-locks machined into the shrouds. These were not feasible due manufacturing limitations and extremely small gap requirements between adjacent shrouds. As a result, the structural dynamics of this design are similar to a blisk with no frictional forces to dampen the structural vibrations. The only damping available for the structure is in the form of material damping. It is therefore critical to evaluate the structure natural frequencies and establish that excitation sources within the turbine have enough margins to avoid resonance. This can be achieved by carefully selecting the stage nozzle counts such that natural structural frequencies are far from resonance with the nozzle passing frequencies. Interference diagrams were used to evaluate the aeromechanic performance of the four continuous integral shrouded blisk stages. An interference diagram calculates the effect of nodal disc vibrations on blade vibrations. It is a plot showing blisk natural frequencies with nodal diameter on the horizontal axis and frequency on the vertical axis. It also plots the first and the second nozzle passing frequencies. Nozzle passing frequency is considered the vibratory force on the turbine blades. It is caused by the working fluid flowing through a nozzle. This force is cyclic given the design of the nozzles. The cyclic force may excite the blisk at its natural frequency and lead to resonance. The interference diagram corresponding to a nozzle count shows the nodal diameter excited for that nozzle count (marked by vertical lines on the diagram), the blisk mode shapes at nodal diameters (zero to half the nozzle count) as well as the first and second nozzle passing frequency. For each of the four stages of the SunShot turbine, a desired percent margin is required for the natural frequencies of the first six modes with respect to the first nozzle passing frequency (NPF) at design speed.

Blisk natural frequency analysis*:* The first few modes for nodal diameters to Number of Bucket were calculated by modal analysis using cyclic symmetry in ANSYS. A single sector model of the blisk was created and imported in ANSYS as shown in Figure 5. A 4- node brick element (ANSYS element SOLID45) was used to mesh the blisk. Constraint equations were used to connect the dissimilar meshes on the blade, the shroud and the disk. Cyclic constraint equations are applied to the specified cyclically symmetric faces on the shroud and the disk. The axial constraints on the two side faces are to prevent any axial movement in the blisk as in reality the shaft is long and any axial movement is negligible.

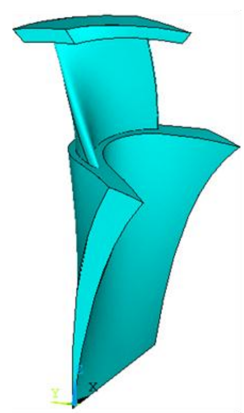

Figure 5: Single sector blisk model

The selection of nozzle count was based on the bucket count, efficiency requirements (all obtained via the aero design process) and the interference diagrams to meet the margin requirement between blisk natural frequency and first NPF. Only the upstream nozzle counts were considered for each stage as a dominant forcing function.

Shaft End Seals: A Dry Gas Seal (DGS) is used to separate the high pressure CO_2 environment inside the turbine region from ambient air and oil lubricated bearings. DGS is a known technology available from various vendors and primarily applied to high pressure compressor to minimize the loss of working fluid to the ambient. In certain situations the fluid leaking out of the DGS can be captured and recompressed using an external compressor. A DGS, suitable for this turbine operating speed and shaft diameter was designed by a vendor. The off-shelf designs of DGS are however limited by the highest operating temperature of the shaft and fluid. The current designs are flooded with high pressure cold flow to avoid damage to the seal components. To achieve this cooling of the shaft from turbine inlet temperatures or higher a thermal management solution is implemented before the DGS to ensure safe and reliable operation of the turbine.

A comparison of this heat transfer Figure of merit (FOM) for CO_2 with air is shown in Figure 6. As CO_2 passes through the critical point, the fluid density increases significantly resulting in very high heat transfer coefficients. This high heat transfer coefficient when compared with the thermal conductivity of the high nickel super alloys being used to manufacture the rotor results in very high Biot numbers between 10 and 40. Due to the high Biot numbers in this application, traditional cooling schemes using counter-flow of cold CO_2 stream in the thermal management region, results in very rapid shaft surface cooling while the core of the shaft is still at a very high temperature. This sets up high radial thermal gradients resulting in high thermal stress in the shaft.

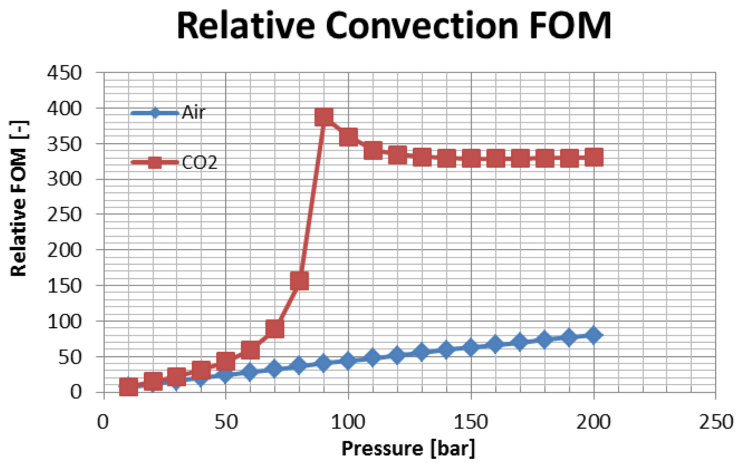

Figure 6: The comparison of relative figure of merit (convection) between air and CO_2 as a function of pressure.

To achieve low stress, below the allowable for shaft material, a proprietary flow arrangement was used to create a predominantly axial thermal gradient in the shaft. A representative thermal gradient in the shaft is shown in Figure 7. Note the axial thermal gradient and low stress numbers on the rotor surface.

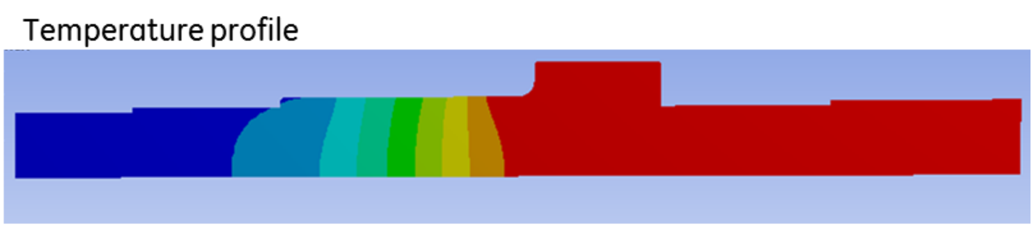

Figure 7: Temperature profile in the shaft and stator piece in the thermal management region.

***Transient start-up and shut-down analysis**:* A transient thermal profile for the turbine inlet temperature was computed based on estimated power block thermal mass and solar irradiation profile at target location (Dagget, CA). The worst case shut down profile was assumed to be a cloud event. With this thermal loading profile, the most limiting turbine rotor components in terms of thermal cycling and LCF life is the turbine stage shroud. In order to allay these concerns, a finite element analysis was performed considering both these loads to evaluate the resultant stresses. The resultant stress profile in the turbine blade row and the shroud are shown in Figure 8. This assessment where the peak concentrated stress is below the allowable stress values for the material at temperature confirms the LCF life of this machine and could be designed >10,000 start and shut down cycles as required.

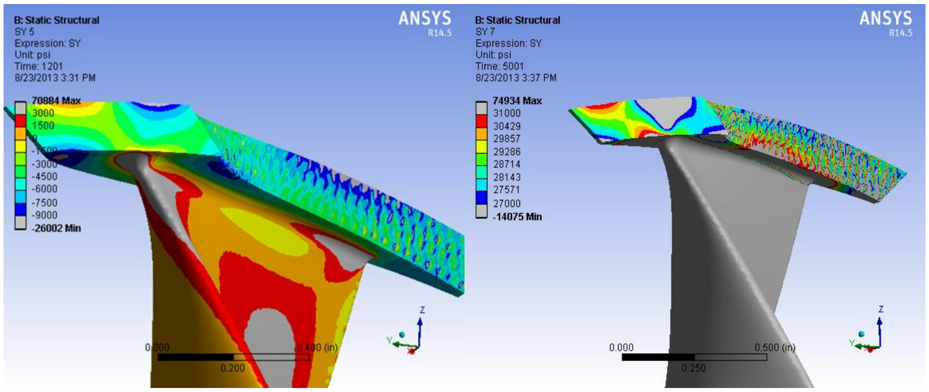

Figure 8: Peak equivalent stress profile in the turbine blade and shroud for (a) After transient to full rated conditions (centrifugal + thermal) and (b) After cool down due to cloud transient (centrifugal + thermal)

Specific design details and metrics:

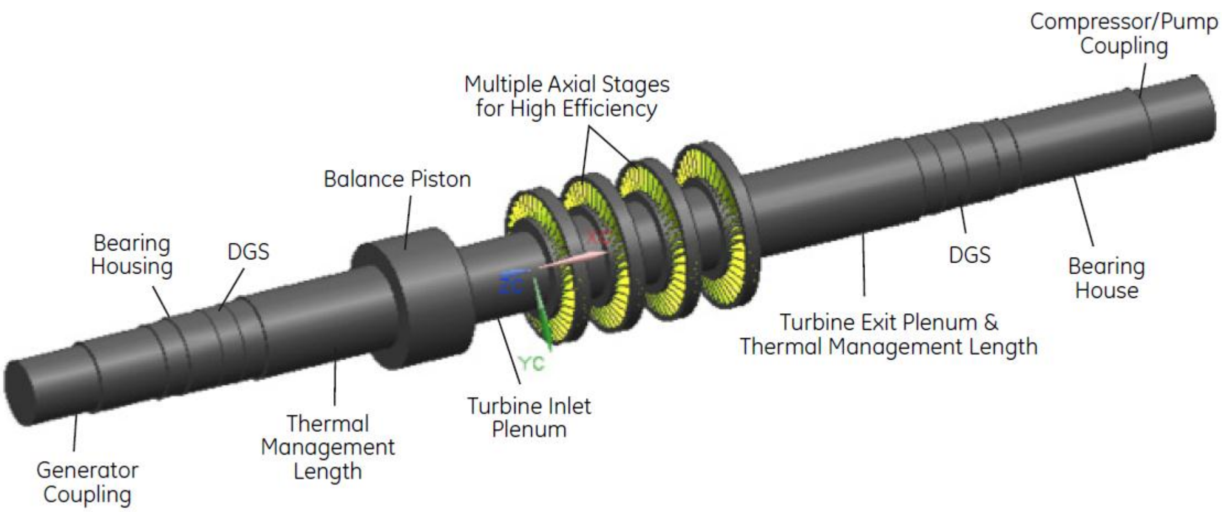

Figure 9: Final SunShot sCO_2 turbine rotor design

- Pressure containment to industry and API standards
- Lateral and torsional rotor-dynamics established per API and industry standards
- Rotor overspeed capability with 20% margin to all conditions anticipated in test loop
  - Thrust bearing and balance piston design expected to meet API 617 standard for anticipated operating conditions in test loop
- Creep and low cycle fatigue life sufficient for expected continuous and transient operation
  - Axial and radial clearances set to meet API 617 at all steady state and transient operating conditions
- Expander isentropic efficiency greater than 85% based on mean line and CFD analysis.

Conclusions:

The design of a high-pressure, high-temperature sCO_2 turbine rotor has been completed and presented here. Specific design challenges due to high power density in the working fluid resulting in highly compact turbomachinery design were presented in detail and include: high torque transmission requirements, small airfoil design and fabrication, challenging aero-design optimization with mechanically safe blade design, and high cycle fatigue life of the rotor. The transient thermal loading of a CSP power plant was also considered and the design is robust to handle multiple fast cloud transient events during the life of the power plant. The overall goals of the development program, to provide high efficiency, compact, low cost turbomachinery package is demonstrated to be achievable. A complete review of the turbine design along with the casing design is currently in progress and will be presented in greater detail at the next symposium.

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