Proceedings of Global Power and Propulsion Society
ISSN-Nr: 2504-4400
Beijing Conference 2019
16th – 18th September, 2019
www.gpps.global

GPPS-BJ-2019-0040

BEARING AND SEAL SELECTION FOR A HIGH-TEMPERATURE SUPERCRITICAL CO_2 TURBINE

Jishun Li
Henan University of Science and Technology
li_jishun@163.com
Luoyang, Henan, China

Hal Gurgenci
The University of Queensland
h.gurgenci@uq.edu.au
Brisbane, Queensland, Australia

Zhiqiang Guan
The University of Queensland
guan@uq.edu.au
Brisbane, Queensland, Australia

Jun Li
Henan University of Science and Technology
lijun_ph@163.com
Luoyang, Henan, China

Md. Shazib Uddin
The University of Queensland
m.uddin@uq.net.au
Brisbane, Queensland, Australia

Lun Li
Henan University of Science and Technology
lilunxn@163.com
Luoyang, Henan, China

ABSTRACT

Solar power towers using supercritical CO_2 (sCO_2) Brayton cycles were proposed as the next-generation Concentrating Solar Thermal (CST) technology. One of the challenges facing rapid deployment of CST using sCO_2 cycles remains to be the power block and, in particular, the lack of reliable sCO_2 turbine technology. The main challenge for sCO_2 turbine designers is in the area of component technologies such as materials, bearings and seals. The combination of high temperatures (above 600ºC), high rotational speeds, and high fluid densities create a perfect storm for sCO_2 turbine designers. This paper investigates appropriate bearings and seals for a sCO_2 radial-in-flow turbine proposed for Concentrating Solar Thermal applications.

INTRODUCTION

This paper describes on-going work on mechanical design of a prototype supercritical CO_2(sCO_2) radial turbine being developed for future Concentrating Solar Thermal (CST) power generation in sizes 300 kW to 30 MW. While the focus of our team is on CST, the benefits of sCO_2 cycles are not limited to solar thermal. They are seen as the next-generation technology in a diverse range of applications including nuclear power generation[1], fossil fuel power generation[2], concentrated solar power[3], shipboard power, waste heat recovery, and geothermal power generation[4]. This is the main reason sCO_2 technology is vigorously pursued in USA, where the Department of Energy (DoE) awarded up to $80m in 2016 to design, build and operate by 2024 a 10-MWe sCO2 plant test facility (the STEP facility) in San Antonio, Texas. The STEP project brings together three DoE Offices (Nuclear Energy, Fossil Energy, and Energy Efficiency and Renewable Energy). As suggested by this partnership, the US project is not limited to CST and aims to develop utility-scale sCO2 power cycle technology across a range of markets including coal, nuclear, gas geothermal and solar[5].

As a result of this emerging interest in supercritical CO2 cycles from different power generation sectors, the relevant research effort has increased substantially over the last 10 years. Most of the published research has been directed to thermodynamic optimisation and feasibility analysis for different applications. Crespi et al[6] provides a good summary of research in this area and analyses performance of different sCO_2 cycle configurations as standalone power generators or in topping or bottoming cycles. Turchi et al[7] examined different power block configurations specifically for CST plants using sCO_2 cycles. Brun, Friedman and Dennis [8] collated a collection of articles as a reference textbook on fundamentals and applications of sCO_2 power cycles. Gurgenci[9], Balaji and Gurgenci[10] demonstrated the potential advantages of CST using sCO_2 cycles for off-grid and fringe-of-grid power generation in Australian outback and other places in the world where there may be no secure access to a strong power grid. In fact, the application of load-following standalone renewable power generation has been the main motivation for the work reported in this paper.

In contrast to fundamental thermodynamics and economic studies, the published work on engineering of supercritical CO_2 cycle component technologies is scant. This is probably because there is no working power generator yet using a sCO_2 cycle and a sCO_2 turbine. The few sCO_2 power block implementation projects have been at laboratory scale and were undertaken as experimental campaigns rather than for power generation. Nevertheless, they helped the community develop a better understanding of the underlying issues. The two notable pioneering studies are the 125-kW sCO_2 turbine tested in Sandia National Laboratory[11] and the 100-kW Integrated System Test facility (IST) of the Knolls Atomic Power Laboratory (KAPL)[12]. The work undertaken on these two relatively small test loops identified significant challenges in sCO_2 power block technology that have been motivating the global research effort in this area since then.

In the Asian continent, the 10-kW loop built at the Tokyo Institute of Technology was a proof-of-concept demonstration of a sCO_2 Brayton cycle and identified high windage losses, and bearing and seal design as the critical challenges for sCO_2 turbines[13]. The SCIEL (Supercritical CO2 Integral Experiment Loop) in KAERI (Korea Atomic Energy Research Institute) was designed and built by a research group from three Korean research institutions KAERI, KAIST and POSTECH[14]. A variety of cycle configurations and turbine inlet temperatures were tested while the compressor inlet temperature was fixed at the near-critical temperature of 33.2ºC and much attention was spent on compressor efficiency. A shrouded radial rotor design was used in the turbine and the rotor shaft was sealed by labyrinth seals and the leakage was extracted. Gas foil bearings and magnetic bearings were considered and gas foil bearings were selected because they were more convenient to utilise at this small size. Contact bearings such, e.g. ball bearings, were not considered due to the very high rotational speed[15]. In China, a number of supercritical CO_2 turbine design studies were published[16-19] in the last couple of years but no actual turbine has yet been constructed.

THE ASTRI PROGRAM

The University of Queensland (UQ) is a partner in the Australian Solar Thermal Research Institute (ASTRI), which is an eight-year, $87 million international research collaboration to develop concentrating solar thermal (CST) power technologies. The UQ is responsible in ASTRI for development of a power block based on a closed supercritical CO_2 cycle.

The aim of the ASTRI project is to bring the sCO_2 turbine technology to a technology readiness level suitable for commercial demonstration at a 1-25 MWe plant size in 2022. Towards this aim, the UQ team is now designing a proof-of-concept 300-kWe sCO_2 turbine and will test it at the CSIRO Newcastle Solar Tower installation. The turbine will have a radial in-flow configuration. The design configuration will be suitable for scaling up to higher plant sizes up to 25-MWe. In a recent study, CST plants using a sCO_2 power block were found to offer the cheapest electricity in this size range for off-grid and fringe-of-grid power generation in Australia[10], the size of which market is estimated to be equal to 6% of the total Australian generation capacity.

Solar power towers using supercritical CO_2 (sCO_2) Brayton cycles were in fact proposed as the next-generation Concentrating Solar Thermal (CST) technology six years ago in a US Department of Energy (DoE) study[20].

One of the challenges facing rapid deployment of CST using sCO_2 cycles remains to be the power block and, in particular, the lack of reliable sCO_2 turbine technology. The main challenge for sCO_2 turbine designers is in the area of component technologies such as materials, bearings and seals. This paper investigates appropriate bearings and seals for a sCO_2 radial-in-flow turbine proposed for Concentrating Solar Thermal applications.

BEARING AND SEAL CHALLENGES

In the 1-25 MWe size range, a radial-in-flow turbine configuration is most appropriate. There are well-established design tools to facilitate the aerodynamic design of the nozzles and the rotor blades. Compared to gas and steam turbines, much higher fluid densities offer a challenge but a number of studies have demonstrated that these challenges can be overcome. The design of the bearings and the seals however is still a formidable challenge.

The combination of high temperatures (above 600ºC), high rotational speeds, and high fluid densities create a perfect storm for sCO_2 turbine designers.

High Temperatures

Referring to Figure 3 of Crespi[6], the thermodynamic cycle efficiency of a sCO_2 cycle does not achieve the efficiency of the advanced ultracritical steam power cycle at tutbine inlet temperatures below 700ºC. It is worth noting that ultracritical steam power generators have to use multiple reheat stages to achieve their high efficiencies at increased cost and complexity and are viable only at utility-scale (e.g. 1000 MW). A supercritical CO_2 power block can achieve similar efficiencies with much simpler power block lay-outs and, using a radial turbine, at sizes as small as 1-MW. In off-grid and fringe-of-the-grid applications where USC steam power is not an option, the sCO_2 can become competitive at lower temperatures as identified in Balaji and Gurgenci[10]. Nevertheless, the long-term potential of sCO_2-based power generation can only be achieved at turbine inlet temperatures exceeding 700ºC. Although our 300-kW prototype is designed for a turbine inlet temperature of 560ºC, the next turbine to be built in the ASTRI program will take CO_2 in at 700ºC.

At such high temperatures, oil-based lubrication is not an option. Gas foil bearings are feasible but they have to be placed outside the pressure boundary to avoid the high windage losses associated with high-density fluid rotating in these bearings at very high speeds. This was a lesson identified very early in the SNL tests[11].

High Rotational Speeds

The high rotational speed is a concern for sCO2 turbines especially at small to medium sizes. This is unfortunate because this size range is a significant market opportunity. The sCO_2 power block has the potential to make CST a competitive commercial option in this important market, which CST so far has failed to capture mainly because of its dependence on steam turbines. Steam cycles cannot be commercially and efficiently be deployed at sizes small enough to serve the current commercial demand in off-grid and fringe-of-grid markets[9].

Supercritical CO_2 radial-in-flow turbines offer the potential for modular scaleable and efficient CST power generation in sizes from 1-MWe to 30-MWe. However, these small sizes come with a price. The rotational speed of a supercritical CO_2 turbine is inversely proportional to its size and rotor shafts for small turbines need to rotate very fast so as to deliver acceptable isentropic efficiencies. This is seen in Figure 1 that shows the variation of the isentropic efficiency with the rotational speed.

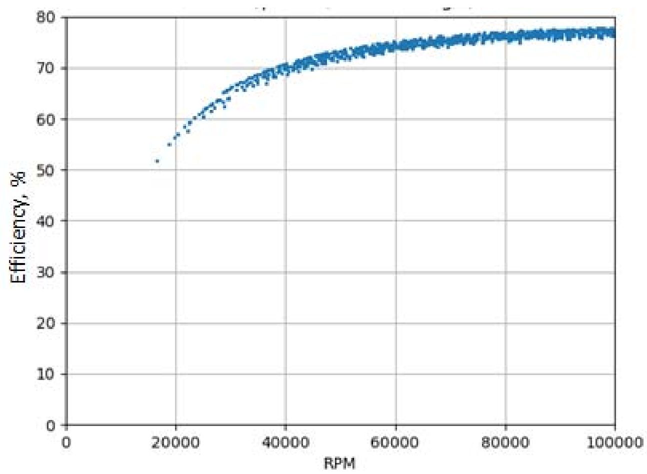

Figure 1 Change of efficiency with rotor sped

Each dot in Figure 1 is a turbine with different geometry parameters (blade angles, etc) and was designed by minimising the inlet Mach number using the meanline design method proposed by Whitfield[21]. As seen in this figure, there is a sharp rise as the speed starts increasing from 20000 but curve gets flatter as the speed gets higher. The speed of 50,000 RPM is the optimum value for the 300-kW turbine in a trade-off between the design of bearings and seals and the rotor efficiency.

Using the 300-kW turbine as a reference, we can plot the speed for larger turbine sizes by using the similarity law:

$$P \propto \rho D^5 \omega^3$$

While using this law, we also note that we would like to keep the linear surface speed of the shaft as small as possible to help with bearing and seal design. The optimum rotor speed against the shaft power can then be plotted as shown in Figure 2. This chart should only be used as only a preliminary guide. The optimum speed at each size needs to be calculated specifically.

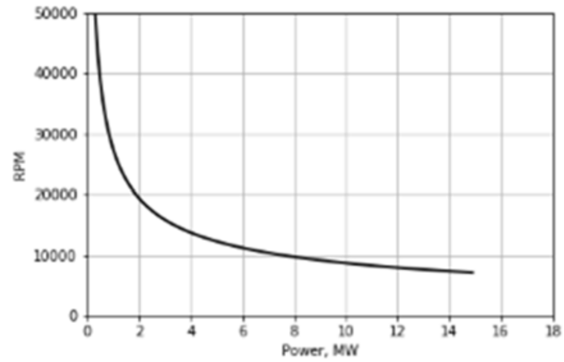

Figure 2 SCO_2 turbine speed versus shaft power

A more subtle effect of higher speed on the bearings is the radial loads on the bearings. To serve as a dispatchable generator with rotating inertia, a sCO_2 turbine must use a synchronous generator. Since synchronous generators operate much lower speeds, the speed of the sCO_2 turbine needs to be reduced to the synchronous speed. This can be done by using a high-speed gearbox. The rotor shaft support bearings would then have to support the pinion power transmission force. We can directly couple to a high-speed DC generator. This would keep the radial bearings loads small but lose the advantage of the rotational inertia of the rotor shaft and its stabilising effect on the AC grid.

High density

One might argue that the above challenges are not unique to sCO_2 turbines. For example, gas turbines are subject to even higher temperatures and turbocharger rotational speeds are comparable to sCO_2 turbines. Lack of suitable bearing technology has not been a problem for them. Bearing selection is a problem for sCO_2 turbines because there is the additional challenge of higher fluid density combined with high-temperature and high-speed conditions. For example, the sCO_2 density at 20MPa and 700ºC is 104 kg/m^3, two orders of magnitude higher than the density of the gas turbine combustion products.

A high-density fluid churning in the bearing passages at very high speeds causes significant problems. Early research identified stability issues and high windage losses if gas foil bearings or similar used in sCO_2 turbines with sCO_2 itself serving as the bearing lubrication fluid, as described in [8] but also in [11, 22].

BEARING AND SEAL CHOICES

Let us now describe how we are proposing to address these challenges in a radial-in-flow sCO_2 turbine, which is the type of turbine most appropriate for modular scaleable CST power plants.

ASTRI Turbine Configuration

There are no commercial options for bearings that will be capable of operating in the hot high-pressure high-temperature

sCO_2. Earlier, the University of Queensland explored the possibility of designing bearings and seals that could be placed within the pressure boundary [23-26] of a sCO_2 turbine. While this would make the mechanical design task easier, the windage losses would be too high. Unfortunately, we are not able to design bearings that can operate with the high-density fluid at low friction.

We decided to place the bearings and the main gas seal outside the pressure boundary. Our selected turbine rotor layout is shown in Figure 3. Standard bearings and standard seals are employed but they are isolated from the high-temperature high-pressure cycle fluid.

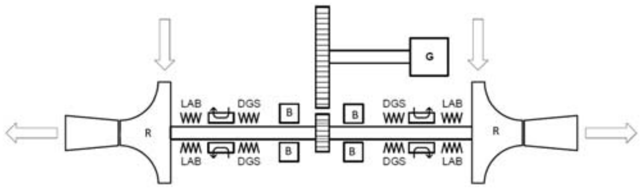

Figure 3 Supercritical CO_2 radial-in-flow turbine lay-out

A disadvantage of this design is the extra unsupported shaft length due to the need for including a space between the dry gas seal and the labyrinth seal. The choice for the bearings and the interaction between the bearing and the shaft dynamics need to be examined carefully in this design.

The University of Queensland (UQ) and Henan University of Science and Technology(HAUST) have a joint project investigating the dynamics of a sCO_2 turbine rotor shaft under different bearing arrangements.

While the turbine design is yet to be finalised, based on preliminary rotor and seal design, it is possible to present a parametric relation between the rotor shaft diameter and the rotor length as given in Figure 4.

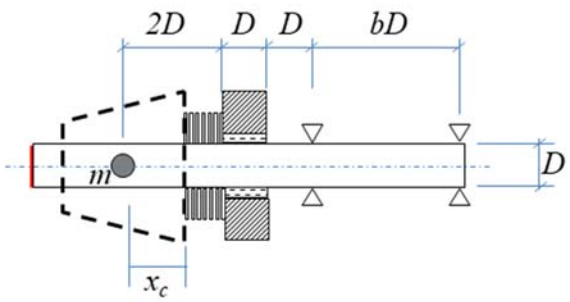

Figure 4 Supercritical CO2 rotor shaft geometry

A closed-form for the gyroscopic whirl of the overhung rotor of mass m can be found using the procedure in rotor dynamics textbooks, e.g. Lee[27]. The lay-out and the shaft geometry is represented by an axially symmetric spring constant that represents the restoration tendency against shaft bending. The value of this constant is estimated by using the relations for a simply supported overhung beam subject to an end moment (e.g. App.14-1 in [28]). Assuming perfectly rigid supports, the critical frequency for the rotor gyroscopic whirl is related to the bearing spacing bD as shown in Figure 5.

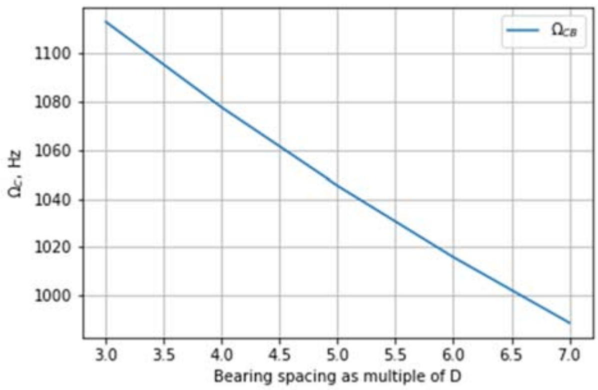

Figure 5 Rotor whirl critical frequency as a function of the bearing spacing

As expected, the critical frequency is the highest for minimum bearing spacing. However, there is a minimum bearing span required to cover the widths of the bearings themselves and the width of the pinion. Therefore, even ignoring the compliance of the bearings, the calculated critical frequencies are lower than and close to the operating frequency of 50000 RPM (833 Hz). The bearing compliance of course will reduce this further and this means crossing over at least one critical frequency during start-ups.

This emphasises the importance of using the appropriate bearings. It also means that the rotor dynamics need to be carefully examined when the bearing choices are finalised.

Bearing Choices

Only non-contact bearings can be considered for the ASTRI turbine. Rolling element bearings are not an option because their life expectation would be too low at our high rotor speeds. There are three non-contact bearing options:

- Oil-lubricated tilted-pad bearings (TP)
- Oil-lubricated hydrostatic bearings (HS)
- Magnetic bearings (MB)

Both radial and thrust bearings are needed. Although the back-to-back rotor configuration of Figure 3 minimises the axial force because the two axial loads from the two rotors cancel each other, a thrust bearings is still needed.

In selecting appropriate bearing technology, the bearing stiffness and the damping values are important parameters when trying to control the rotor dynamics. Assuming y- and z- are the two orthogonal radial directions, the bearing forces are related to radial displacements and rates of displacements through stiffness and damping matrices as in the following equation:

$$\begin{Bmatrix} F_y \\ F_z \end{Bmatrix} = -\begin{bmatrix} k_{yy} & k_{yz} \\ k_{zy} & k_{zz} \end{bmatrix} \begin{Bmatrix} y \\ z \end{Bmatrix} - \begin{bmatrix} c_{yy} & c_{yz} \\ c_{zy} & c_{zz} \end{bmatrix} \begin{Bmatrix} \dot{y} \\ \dot{z} \end{Bmatrix}$$

Table 1 lists typical stiffness and the damping values for the turbine choices to be considered. Ball bearings (BB) are included in Table 1 for comparison only (they are not considered as a viable option for high-speed turbines). The coefficient values in Table 1 are for a 300kW/50000RPM turbine and a bearing span shaft diameter of 60mm. Bearings were sized for a 18-tooth pinion placed at mid-span taking the torque to a high-speed gear box that takes the speed down to enable synchronous generation. The purpose of this table is to aid a parametric analysis as part of the preliminary design process. Therefore, the stiffness and damping coefficients are listed only in order of magnitude precision. Magnetic bearings can be actively controlled to a certain extent. Therefore, no value is provided for magnetic bearings.

The stiffness values in Table 1 are normalised against the stiffness of the flexible overhung shaft supported by rigid bearings for a bearing span of 4*D*, which is computed as:

$$k=7.1 \times 10^6 \text{ N/m}$$

The damping coefficients in Table 1 are normalised against the critical damping coefficient, which is calculated as

$$C_c=2m\omega_{bwc}=2 \times 0.817 \times 6770 = 11060 \text{ kg/s}$$

Table 1 – Stiffness and damping coefficients for different bearing options

Bearing	k_{yy}	k_{zz}	k_{yz}	k_{zy}	c_{yy}	c_{zz}	c_{yz}	c_{zy}
BB	35	35	0	0	0	0	0	0
TP	10	6	14	1	3×10^{-4}	6×10^{-4}	6×10^{-4}	6×10^{-4}
HS	200	200	50	-50	15	8	-5	7

The observations from Table 1 should apply to a range of turbines beyond the size considered in this study. It is observed that the tilted-pad bearing stiffness values are within an order of magnitude of the shaft stiffness. Therefore, for such bearings, the rotor dynamics must include the coupling between the shaft and the bearing dynamics. The cross-coupled stiffness terms (k_{yz} and k_{zy}) need special attention because they may be a source of instability. The damping terms are relatively small and will not have a significant influence on rotor dynamics.

Seal Choices

Two types of seals are used in the ASTRI turbine as shown in Figure 3. Starting from the rotor, the first seal seals the high-temperature gas. This is a labyrinth seal. Labyrinth seals are not perfect seals and they always leak. The labyrinth seals on the ASTRI turbine are designed with relatively high clearances to accommodate thermal expansion as well as vibrations of this overhung shaft. The leaks from the labyrinth seal are controlled by the pressure of the cooling sCO_2 flow that circulates through the cooling zone. Labyrinth seals are well-understood and pose no special challenges in this application.

The second seal sets the pressure boundary by isolating the cool high-pressure cooling circulation from the bearings. At a speed of 50000 RPM, a dry gas seal is the only viable option.

Both seals will influence rotor dynamics at varying degrees. The dry gas seal is adjacent to the bearings and its influence is expected to be limited. The labyrinth seal is separated from the supports by the length of the cooling zone and its stiffness may be a significant influence and must be included in rotor dynamics computations once its design is finalised.

CONCLUSIONS

Design and selection of bearings and seals for sCO_2 turbines offer specific challenges to the turbine designer as described in this paper. These challenges, while formidable, are not unsolvable and there are viable off-the-shelf options provided the turbine lay-out is designed to isolate these off-the-shelf options from the high-temperature high-density gas.

Stiffness and damping of the bearings and seals need to be carefully considered and analysed in full rotor system dynamic modelling. The stiffness and damping characteristics for the supercritical turbine bearings and seal choices were listed to help informed selection.

ACKNOWLEDGMENTS

The authors acknowledge funding from the Australian Renewable Energy Agency (ARENA) through the Australian Solar Thermal Research Institute (ASTRI). The authors also acknowledge funding from Key Technology of High Speed Supercritical CO_2 Turbine Henan Centre for Outstanding Overseas Scientists; and the National Natural Science Foundation of China (No.51375148) for financial support. The authors would also like to acknowledge the contribution of Mr Hugh Russell (The University of Queensland) as the creator of the turbine design shown in Figure 3.

1. Dostal, V., *A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors*. 2004, MIT.
2. Moullec, Y.L., *Conceptual study of a high efficiency coal-fired power plant with CO2 capture using a supercritical CO2 Brayton cycle.* Energy, 2013. **49**: p. 32-46.
3. Turchi, C.S., *Supercritical CO2 for Application in Concentrating Solar Power Systems*, in *Supercritical CO2 Power Cycle Symposium*. 2009: Troy, NY.
4. Atrens, A.D., H. Gurgenci, and V. Rudolph, *CO2 Thermosiphon for Competitive Geothermal Power Generation.* Energy & Fuels, 2009. **23**(1): p. 553-557.
5. *US Department of Energy Supercritical CO2 Team*. 2019 [cited 2019; Available from: https://www.energy.gov/supercritical-co2-tech-team.
6. Crespi, F., et al., *Supercritical carbon dioxide cycles for power generation: A review.* Applied Energy, 2017. **195**: p. 152-183.
7. Turchi, C.S., et al., *Thermodynamic Study of Advanced Supercritical Carbon Dioxide Power Cycles for Concentrating Solar Power Systems.*

Journal of Solar Energy Engineering-Transactions of the Asme, 2013. **135**(4).
8. Brun, K., P. Friedman, and R. Dennis, *Fundamentals and Applications of Supercritical Carbon Dioxide (sCO_2) Based Power Cycles*. 2017: Elsevier. 433.
9. Gurgenci, H., *Supercritical CO2 cycles offer experience curve opportunity to CST in remote area markets.* Energy Procedia, 2013. **49**: p. 1157-1164.
10. Balaji, V. and H. Gurgenci, *Search for optimum renewable mix for Australian off-grid power generation.* Energy, 2019.
11. Wright, S., et al., *Operation and Analysis of a Supercritical CO2 Brayton Cycle*. 2010, Sandia Laboratories. p. 101.
12. Kimball, K.J. and E.M. Clementoni, *Supercritical Carbon Dioxide Brayton Power Cycle Development Overview*, in *ASME Turbo Expo 2012*. 2012: Copenhagen, Denmark.
13. Utamura, M., et al., *Demonstration of supercritical CO2 closed regenerative Brayton cycle in a bench scale experiment*, in *ASME Turbo Expo*. 2012: Copenhagen, Denmark.
14. Ahn, Y., et al., *Design consideration of supercritical CO2 power cycle integral experiment loop.* Energy, 2015. **86**: p. 115-127.
15. Cho, J., et al., *Development of the turbomachinery for the supercritical carbon dioxide power cycle.* International Journal of Energy Research, 2016. **40**(5): p. 587-599.
16. Lv, G.C., et al., *Aerodynamic design optimization of radial-inflow turbine in supercritical CO2 cycles using a one-dimensional model.* Energy Conversion and Management, 2018. **165**: p. 827-839.
17. Zhang, D., Y. Wang, and Y. Xie, *Investigation into Off-Design Performance of a S-CO2 Turbine Based on Concentrated Solar Power.* Energies, 2018. **11**.
18. Zhou, A., et al., *Aerodynamic design and numerical analysis of a radial inflow turbine for the supercritical carbon dioxide Brayton cycle.* Applied Thermal Engineering, 2018. **132**: p. 245-255.
19. Luo, D., et al., *The design and analysis of supercritical carbon dioxide centrifugal turbine.* Applied Thermal Engineering, 2017. **127**: p. 527-535.
20. *Sunshot Concentrating Solar Power Program Review*. 2013. Phoenix, Arizona: US DoE.
21. Whitfield, A., *The Preliminary Design of Radial Inflow Turbines.* Journal of Turbomachinery-Transactions of the Asme, 1990. **112**(1): p. 50-57.
22. Ahn, Y., et al., *Review of Supercritical CO2 Power Cycle technology and Current Status of research and Development.* Nuclear Engineering and Technology, 2015. **47**: p. 647-661.
23. Qin, K., R.J. Gollan, and I.H. Jahn, *Application of a wall function to simulate turbulent flows in foil bearings at high rotational speeds.* Tribology International, 2017. **115**: p. 546-556.
24. Qin, K., et al., *Development of a Computational Tool to Simulate Foil Bearings for Supercritical CO2 Cycles.* Journal of Engineering for Gas Turbines and Power-Transactions of the Asme, 2016. **138**(9).
25. Qin, K., I.H. Jahn, and P.A. Jacobs, *Effect of Operating Conditions on the Elasto-Hydrodynamic Performance of Foil Thrust Bearings for Supercritical Co2 Cycles.* Proceedings of the Asme Turbo Expo: Turbine Technical Conference and Exposition, 2016, Vol 7b, 2016.
26. Zakariya, M.F. and I.H.J. Jahn, *Performance of Supercritical Co2 Dry Gas Seals near the Critical Point.* Proceedings of the Asme Turbo Expo: Turbine Technical Conference and Exposition, 2016, Vol 9, 2016: p. 373-382.
27. Lee, C.-W., *Vibration Analysis of Rotors*. Solid Mechanics and its Applications, ed. G.M.L. Gladwell. 1993: Springer Science.
28. Mott, R.L., *Machine Elements in Mechanical Design*. 5th edition ed. 2016: Pearson.