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Computational Analysis of Seals for sCO_2 Turbomachinery and Experimental Planning

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AUTHOR BACKGROUND

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ABSTRACT

Supercritical carbon dioxide power cycles are expected to have high efficiencies that will hinge, in part, on the availability of low-leakage turbomachinery seals. End seals are expected to have the highest leakage rates and are therefore of greatest concern for impacting overall system efficiency. Seals for supercritical carbon dioxide turbomachinery are difficult to design because of the high pressures involved and the fluid phase characteristics of CO_2 itself, especially when operating close to the critical point. Few experimental results are available and traditional prediction methods are limited due to the assumption that fluid properties stay constant in the seal. Existing methods cannot generally resolve the rapid changes in fluid

properties that occur when operating near the critical point. To overcome this limitation in the literature, a joint experimental and modeling effort is being carried out at the University of Virginia in the ROMAC laboratory with the goal of testing supercritical carbon dioxide seal performance. A novel seal test rig is being designed in which traditional annular seals will be tested at representative operating conditions to gain a deeper understanding of the leakage and power losses that these seals would incur and to provide the opportunity to validate and improve prediction techniques. Here we will review current computational models for predicting seal leakage including those developed at ROMAC. Next, the characteristics of the seal test rig will be presented as configured to test traditional sealing technologies with supercritical carbon dioxide. Predictions using ROMAC codes will then be shared for the expected seal leakage during experiments. The outputs of these planning efforts will inform our experimental design which will also be presented.

INTRODUCTION

Supercritical carbon dioxide (sCO_2) power cycles have the potential to achieve thermal efficiencies over 50% with a compact footprint [1]. Before this technology can be deployed at the utility-scale there are several areas that need further development. This paper and its corresponding poster focus on the need for efficient seals as an enabler of this technology. As established in [2], seals are a particular concern for sCO_2 power cycles. The majority of sCO_2 power cycles envisioned to date have been designed such that fluid conditions at the inlet of the compressor are typically near the critical point, see Table 1 for more details. The fluid properties of CO_2 vary rapidly near the critical point creating difficulties in predicting the seal leakage using existing methods [3].

Table 2. Experimental Low Side Temperature Conditions

Location	Component	Inlet Pressure		Inlet Temperature		Ref.
		[Mpa]	[psi]	[C]	[F]	[-]
Sandia National Labs	Main Compressor	7.7	1115	32.2	90	[5]
	Recompressor	7.8	1130	59.4	139	[5]
IST	Compressor	9.3	1346	35.6	96	[5]
SwRI Sunshot	Pump	8.3	1204	10.0	50	[5]
KAIST	Compressor	7.7	1117	35.0	95	[6]

Bidkar et al. provide a review of seal technology for sCO_2 applications concluding that small machines (shaft less than 6 inches) will be able to use commercially available labyrinth seals [2]. However, labyrinth seal leakage for larger machines is expected to be significantly detrimental to the cycle efficiency. Therefore, they are evaluating the use of hydrodynamic face seals. The labyrinth seal evaluation was performed using a proprietary code, and it is unclear whether or not it is an analytical or computational code. Thimsen et al. also recommend research and development for utility scale ends seals [3]. Yuan et al. constructed a test rig capable of testing a stationary labyrinth seal with sCO_2 [15]. In addition to experiments, they also predicted the leakage using an analytical method and CFD. Their CFD modeling approach matched well with the experiments, which were performed at pressure ratios ranging from 1.1 to 3 with an inlet pressure of 1500 psi. The analytical method they used over predicted the leakage by ~30%.

The primary goal of this research is to improve prediction methods for sCO_2 seals so that seals can be designed with leakage that will maintain the high cycle efficiencies offered by sCO_2 power cycles. The Rotating Machinery and Controls (ROMAC) laboratory at the University of Virginia is currently working towards this goal by performing computational predictions along with building a test rig [4] for experimental validation. Leakage predictions are being performed with ROMAC's RotorLab+ software package and with the commercial computational fluid dynamics (CFD) package, ANSYS CFX. This paper will provide an overview of current efforts plus further background in the three areas of RotorLab+, CFD, and test rig development.

METHODOLOGY

This section provides an overview of the seal dimensions and operating conditions being used to run the computational methods and build the test rig. First the dimensions of the currently installed smooth annular seal are shown in Table 2. There are also plans to perform experiments with hole pattern and labyrinth seals.

Table 2. Seal Dimensions (for single seal)

	Dimension	
	[mm]	[in]
Length	50.8	2.00
Shaft Radius	25.4	1.00
Clearance	0.0254	0.001

To develop meaningful test conditions, the authors consulted with industry and reviewed existing test set-ups. From [5] and [6], the operating conditions in Table 1 were found to be on the low-side temperature portion of the test loop. Future work will be to contact these test sites to see if they will share their seal dimensions and leakage data for additional comparison points. Based on the review of test site conditions, an initial test matrix was developed using a pressure ratio of 3, as shown in Table 3.

Table 3. Test conditions

Case	Seal Type	Inlet Temp		Inlet Pressure		Outlet Pressure	
[#]	[-]	[C]	[F]	[MPa]	[PSIA]	[MPa]	[PSIA]
1	Smooth	32.22	90	7.58	1100	2.53	366.7
2	Smooth	32.22	90	8.27	1200	2.76	400
3	Smooth	32.22	90	8.96	1300	2.99	433.3
4	Smooth	37.78	100	7.58	1100	2.53	366.7
5	Smooth	37.78	100	8.27	1200	2.76	400
6	Smooth	37.78	100	8.96	1300	2.99	433.3
7	Smooth	43.33	110	7.58	1100	2.53	366.7
8	Smooth	43.33	110	8.27	1200	2.76	400
9	Smooth	43.33	110	8.96	1300	2.99	433.3

ROTORLAB+ SIMULATIONS

The first part of the analysis was completed in software developed by ROMAC at the University of Virginia for smooth and hole-pattern seals, DamperSeal. DamperSeal is a seal analysis tool released in 2016 as part of a software package called RotorLab+ 4.0. This code uses Hirs bulk flow theory [7] on liquids, gases, or a combination of both to perform analyses of smooth and hole-pattern seals. More details on the bulk-flow model as well as the seal geometry used in DamperSeal can also be obtained from Migliorini et al [14]. RotorLab+ serves as the graphic user interface where the user can prescribe inputs, run the analysis tool, and obtain results. The primary parameters that were analyzed for this study include leakage values and power loss for a smooth seal at specific operating conditions.

In order to limit the number of simulations, the shaft speed was held constant at 10,000 rpm. Fluid properties were generated with NIST REFPROP [8] for three different temperatures and pressure differentials above the critical point of CO_2. The operating conditions that were varied for each case include inlet pressure, outlet pressure, and inlet temperature. Likewise, gas viscosity and compressibility factors at each temperature and pressure were obtained and specified accordingly. Outlet conditions were predicted by assuming isenthalpic expansion across the seal. The molecular weight of CO_2 was set to be 44.01 kg/mol and the pre-swirl ratio and the inlet pressure loss coefficient were both set at 0.25. The analysis was set to run for 5000 iterations to ensure convergence. A residual of 1E-05 was achieved.

The prediction of outlet conditions showed very different gas compressibility factors at the inlet and outlet. This is not unusual for sCO_2, however DamperSeal only takes one compressibility factor as an input. Therefore, it was decided to perform a sensitivity study to see how the code behaved by using the compressibility factor K based on inlet pressure, outlet pressure, and the average of the two values. The compressibility factors for each case are shown in Table 4. The compressibility factor varied by as much as 71% from the inlet value.

Table 4. Comparison of Gas Compressibility Factors

Case	Inlet Temp		Inlet Pressure		Viscosity	Gas Compressibility, K			% Diff from Inlet K	
[#]	[C]	[F]	[MPa]	[PSIA]	[µPa-sec]	Inlet K	Average K	Outlet K	Average K	Outlet K
1	32.22	90	7.58	1100	31.982	0.291	0.394	0.497	35%	71%
2	32.22	90	8.27	1200	52.356	0.214	0.287	0.361	34%	69%
3	32.22	90	8.96	1300	57.399	0.218	0.273	0.328	25%	50%
4	37.78	100	7.58	1100	21.228	0.508	0.596	0.683	17%	35%
5	37.78	100	8.27	1200	26.819	0.383	0.476	0.570	24%	49%
6	37.78	100	8.96	1300	42.66	0.264	0.342	0.420	30%	59%
7	43.33	110	7.58	1100	20.224	0.574	0.655	0.735	14%	28%
8	43.33	110	8.27	1200	22.367	0.504	0.587	0.670	17%	33%
9	43.33	110	8.96	1300	26.742	0.414	0.499	0.584	20%	41%

The results obtained for all three-temperature values at three different pressures are presented in the tables below. Several cases did not converge, and are marked as DNC. Table 5 shows a comparison of leakage rates, and Table 6 a comparison of power loss. Leakage predictions varied up to 8% depending on which compressibility factor was used. The power loss had much larger variations, up to 35% different from the values obtained using the inlet compressibility factor.

Table 5. Leakage Comparison

Case	Inlet Temp		Inlet Pressure		Leakage [kg/s]			% Diff from Inlet K	
[#]	[C]	[F]	[MPa]	[PSIA]	Inlet K	Average K	Outlet K	Average K	Outlet K
1	32.22	90	7.58	1100	0.147	0.140	0.135	-5%	-8%
2	32.22	90	8.27	1200	0.158	0.151	0.145	-5%	-8%
3	32.22	90	8.96	1300	0.170	0.164	0.159	-4%	-6%
4	37.78	100	7.58	1100	0.143	0.139	0.136	-3%	-5%
5	37.78	100	8.27	1200	0.160	DNC	DNC	DNC	DNC
6	37.78	100	8.96	1300	0.173	0.166	0.160	-4%	-7%
7	43.33	110	7.58	1100	0.141	DNC	0.135	DNC	-4%
8	43.33	110	8.27	1200	0.157	0.153	0.149	-3%	-5%
9	43.33	110	8.96	1300	0.173	0.167	0.163	-3%	-6%

Table 6. Power Loss Comparison

Case	Inlet Temp		Inlet Pressure		Shear Power on Rotor [kW]			% Diff from Inlet K	
[#]	[C]	[F]	[MPa]	[PSIA]	Inlet K	Average K	Outlet K	Average K	Outlet K
1	32.22	90	7.58	1100	23.0	18.0	14.9	-22%	-35%
2	32.22	90	8.27	1200	34.8	27.4	22.8	-21%	-34%
3	32.22	90	8.96	1300	37.1	31.0	26.7	-16%	-28%
4	37.78	100	7.58	1100	13.4	11.7	10.5	-13%	-22%
5	37.78	100	8.27	1200	18.8	DNC	N/A	DNC	DNC
6	37.78	100	8.96	1300	29.7	24.1	20.4	-19%	-31%
7	43.33	110	7.58	1100	11.8	DNC	9.7	DNC	-18%
8	43.33	110	8.27	1200	14.4	12.7	11.4	-12%	-21%
9	43.33	110	8.96	1300	18.6	16.0	14.1	-14%	-24%

The results for leakage variations with inlet pressure are plotted in Figure 1 below for each temperature value analyzed in this study. In general, it can be observed that leakage increases with increasing pressure differential, which is expected. The leakage decreases with increasing temperature at low-pressure differentials but such relationship is not consistent as pressure gradient is increased. Further analysis on the impact of temperature on leakage under high-pressure gradient conditions will be part of a future study.

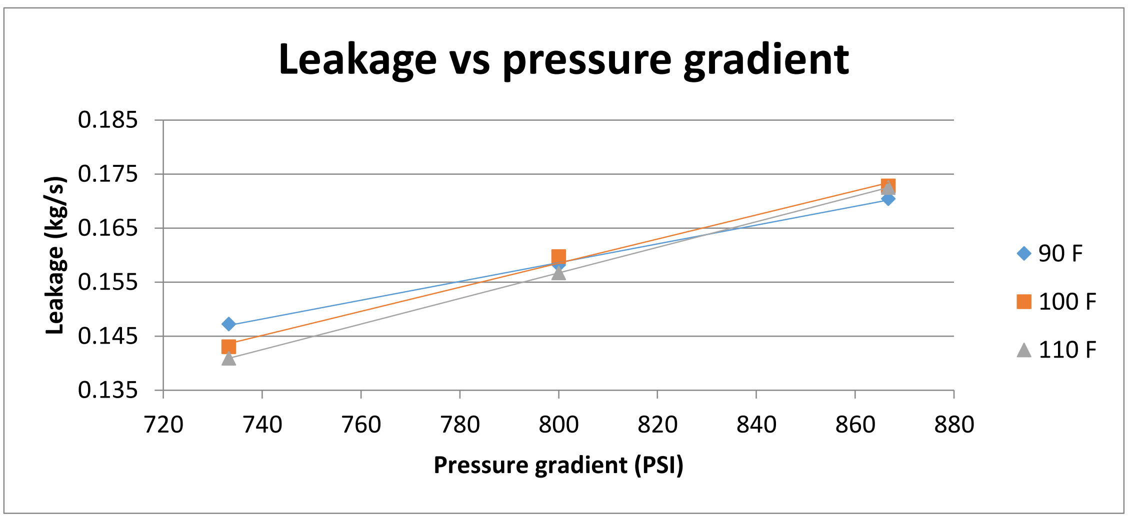

Figure 1. Leakage vs. pressure gradient at three different temperature values

CFD SIMULATIONS

Performing CFD simulations close to the critical point presents numerical challenges with regards to the sharply changing fluid properties of carbon dioxide in this operating range. It is thus expected that the equation of state used will have a large impact on the simulation results. Zhao et al. [9] performed a review of equations of state for simulating sCO_2 Brayton cycles and found that the Span-Wagner method was the most accurate. Several CFD studies focused on the simulation of centrifugal compressors for sCO_2 cycles [10-13] and each resulted in the creation of a user-defined table of fluid properties using NIST-REFPROP's Span-Wagner implementation. The resolution of these tables was shown to be important; Ameli et al. [10] performed a sensitivity study and noted that sharply changing properties would be difficult for the solver and Baltidjiev et al. [11] noted using a resolution of 0.1 K and 0.1 bar. These studies used various CFD codes, thus additional calculations were performed to provide fluid properties in the format required for each code. Saxena et al. [13] was the only one of these papers to mention compressor seals, noting that it was planned to be a labyrinth seal but did not include the seal in the simulation.

Modeling is on-going at ROMAC to develop a CFD simulation of the smooth seal. To represent the annular seal and save on computational resources, we are modeling an annular slice, 2 elements thick in the circumferential direction. Simulations are being performed with ANSYS CFX using the k-epsilon turbulence model. CFD will be performed for each of the test cases outlined in Table 3.

TEST RIG

A seals test rig [4] is under construction at UVA's ROMAC laboratory. The rig is designed with a replaceable test section to enable testing of multiple seal designs. The two initial sets of seals to be tested are smooth and hole pattern seals with a 1 mil clearance. It is also planned to test labyrinth seals as these are typically employed in industry, for sCO_2 applications [13]. Complete dimensions of the seals are shown in Table 1. The test rig is built in a laboratory with high ventilation which provides a safe atmosphere in case carbon dioxide leakage occurs. The test rig is shown in Figure 2 which has a rated working pressure of 1500 psi.

Currently the test rig is undergoing final assembly with plans to commission using air. The long-term plan is to adapt it to use sCO_2 as the working fluid and to perform the same test matrix as analyzed with RotorLab+ and soon to be simulated in CFD. The test rig supply and exit lines will need to be adjusted before it is ready to run with sCO_2 as the working fluid. Tanks of carbon dioxide will be used to supply the experiments. A pump will be necessary to elevate the CO_2 up to the desired test pressures. The tank will be located outside with piping routed to the test rig. Temperature control will be necessary to control the experimental entrance temperature. The initial plan is to use a cylinder heating wrap to pre-heat the cylinder. Currently the rig does not have exit piping, but it was designed to accommodate such, so piping will be added to collect the exit CO_2 and direct it to a vent. Temperature and pressure control into the test rig is expected to be a challenge, thus other heating and cooling designs may need to be considered.

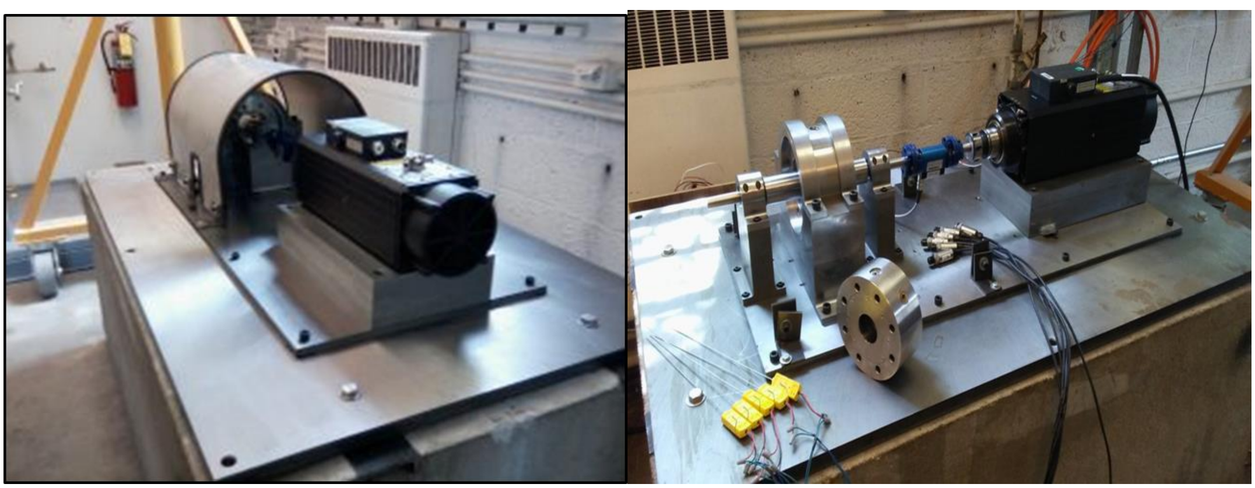

Figure 2. Seals Test Rig, left) Assembled, right) Disassembled

SUMMARY

In summary, ROMAC is working toward improving prediction methods for sCO_2 seals. This will be achieved by comparing simulations from RotorLab+ and CFD against experiments. To date, simulations have been performed in RotorLab+, with CFD model development in progress. The next step of the work is to finish construction and commission the seal rig using air as the working fluid. This will be followed by converting the rig to use sCO_2 as the working fluid and then to perform and compare against the predictions shown in this paper.

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