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Lessons Learned from the Operation of 10MW sCO_2 Compressor Test Facility at the University of Notre Dame

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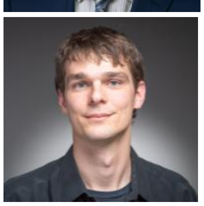
Will Stewart is an experienced engineer who joined Notre Dame Power & Propulsion in 2016 as a senior design engineer. He has been the design lead on multiple compressor rig projects. Will is well-versed in the fields of mechanical design, structural analysis, heat transfer, and manufacturing methods. He earned a Bachelor of Science degree in Mechanical Engineering from Purdue University and a Master of Science degree in Mechanical Engineering from the University of Michigan.



Jeffrey Clark is a Turbomachinery Technician at Notre Dame Power and Propulsion. His work as the build lead focuses on the mechanical assembly, instrumentation, and testing of the axial compressors and turbine component for advanced gas turbines as well as the 10MW sCO_2 Compressor. He earned his Associate of Applied Science from the University of Northwestern Ohio.

Scott Morris is a Professor of Aerospace and Mechanical Engineering at the University of Notre Dame, where he is involved with research involving turbulence, turbomachinery, and aeroacoustics.

ABSTRACT

The construction of a 10MW sCO_2 Compressor Test Facility at the University of Notre Dame was completed in 2023. The closed CO_2 test loop is comprised of a throttle valve, a heat exchanger, a Coriolis flowmeter, and a filter. The loop was built on a mobile cart that facilitates the installation and the removal of the loop to the 10MW variable-speed electric motor drive train in the compressor test cell. A CO_2 inventory system with two 5-ton CO_2 tanks and vaporizers was built outside of the test cell.

A test campaign for a single-stage axial compressor was completed in 2023, and a test campaign for a three-stage axial compressor was completed in 2024. The test campaigns included performance mapping, stall tests, CO_2 real gas effect tests, blade aeromechanical tests, etc. Some challenges encountered during design and testing, such as CO_2 leakage flow management and replenishment, lubrication oil system management, thrust balancing of the rotor, flow rate measurement, and inventory system management are summarized in this paper as lessons learned from the operation of the test facility.

INTRODUCTION

10MW sCO_2 Compressor Test Facility at University of Notre Dame

A closed-loop 10MW sCO_2 compressor test facility was built at the University of Notre Dame. The 10MW open-loop air compressor test facility has been successfully operational at the University of Notre Dame. A CO_2 inventory system and a CO_2 loop on a mobile cart were added to the 10MW compressor facility in a way to allow the 10MW compressor test facility to be used for both air compressor and CO_2 compressor. The figure 1 shows a CAD model view of the test facility. The enthalpy added to the flow by the test compressor was removed by a heat exchanger to control the energy balance in the test loop; in this way the compressor inlet temperature could be maintained at a controlled test condition. The inlet pressure of the compressor is managed by the control of the supply of CO_2 gas from the CO_2 inventory management system and the discharge of CO_2 gas from the loop. A Coriolis flowmeter installed in the loop and a custom-designed bellmouth flow meter in front of the test compressor measured the flow rate to the test compressor. And two pressure relief valves, one at the compressor inlet pipe and the other at the compressor exit pipe respectively, were installed for the safety of the closed-loop from the over pressure. Details of the test loop design were published in Kang et al. [1].

A single-stage axial CO_2 compressor testing and a three-stage axial CO_2 compressor testing have been completed with the test facility and the test results were published by Kang et al.[2] and Kang et al.[3] respectively. Figure 2 shows a cross section of three stage CO_2 compressor test rig. The three-stage CO_2 compressor testing required more than 9MW of electrical power.

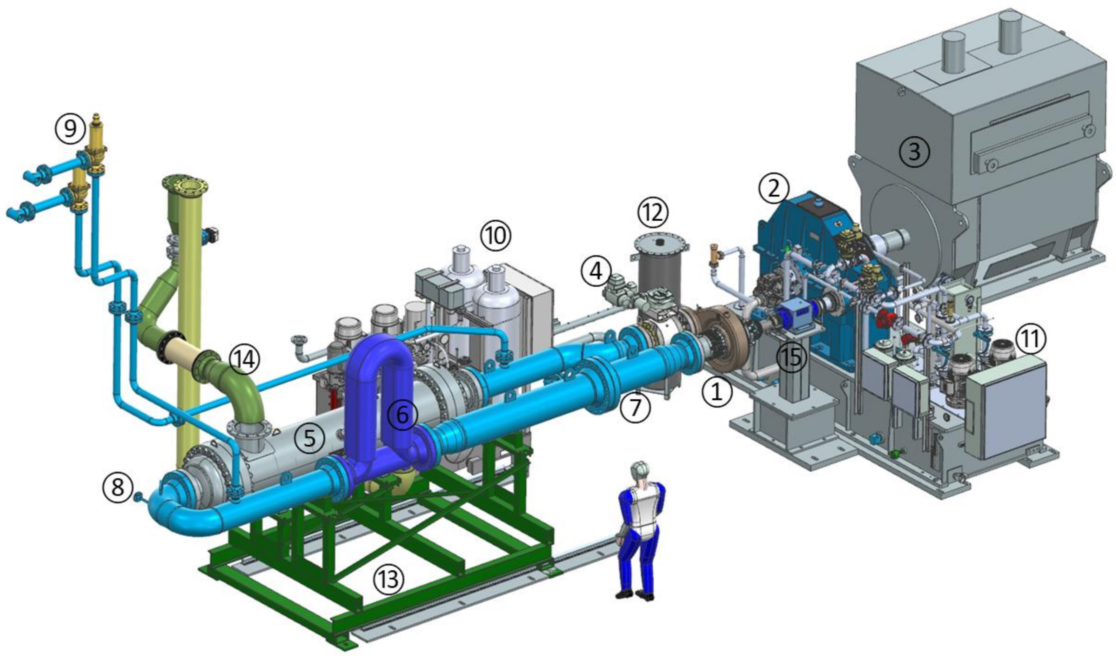


Figure 1 3D CAD model of the closed-loop sCO_2 compressor test facility with a 10MW variable-speed drive motor and a gear box at the University of Notre Dame. 1: Test compressor, 2: speed increasing gear box, 3: 10MW motor, 4: throttle valve, 5: heat exchanger, 6: Coriolis flowmeter, 7: Inlet filter module, 8: CO_2 charging pipe to the closed-loop from the CO_2 inventory system, 9: Pressure relief valves, 10: test rig lubrication system, 11: Lubrication system for gear box and motor, 12: Oil and CO_2 separation tank, 13: portable cart for CO_2 test loop, 14: Water/Glycol pipe for heat exchanger cooling, 15: Torquemeter and its stand

RESULTS AND DISCUSSION

Thrust Balancing

The thrust loading of the three-stage compressor rot from the blades and disks was higher than the thrust loading capability of any commercially available thrust bearing. To reduce the thrust loading to the thrust bearing within its thrust capability., two design approaches have been taken. The first was reducing the pressure difference across the disk by adopting pressure balancing holes. The second was a thrust piston to compensate the thrust loading by introducing thrust force to the opposite direction to the thrust loading generated by the blades and disks. Some of the CO_2 gas from the compressor exhaust were extracted to the left side of the thrust piston (Figure 2) to generate thrust loading to the opposite direction.

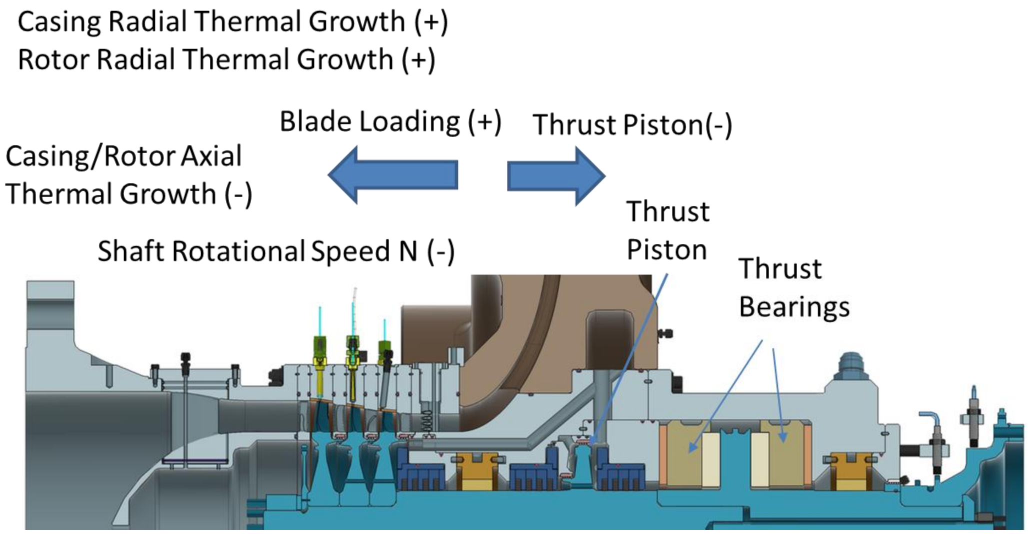


Figure 2 Cross section of the 3 stage CO_2 axial compressor test rig with factors affecting the tip clearance shown. "+" means increasing tip clearance and "-" means decreasing tip clearance.

To contain the CO_2 gas for the thrust piston within the rig and not to let the CO_2 gas leak out to the test cell, the thrust piston was located in the middle of the bearing housing between the two radial bearings. The CO_2 gas from the low pressure side of the thrust piston mixes out with the lubrication oil as it comes out of the bearing housing through the oil and gas outlet.

Lubrication Oil System

The CO_2 gas from the thrust piston must be discharged from the bearing housing as mentioned in the previous section. Also there are CO_2 leakages through the carbons seals that mixes with the lubrication oil in the bearing housing. The CO_2 gas in the bearing housing and the lubrication oil share common discharge ports from the bearing housing.

To make sure the required lubrication oil flow rate for the bearing are supplied to the bearings while the CO_2 leakage flow from the higher pressure source than the oil supply pressure, and to make sure the lubrication oil or CO_2 gas in the bearing housing does not flood over to undesired direction to the compressor section or to the test cell, the pressure at the outlet of the oil/CO_2 mixture must be maintained well below the atmospheric pressure. This was accomplished by a vacuum ejector operated by compressed air as a motif flow. The schematic diagram of the oil and CO_2 gas discharge & separation system is shown in figure 3.

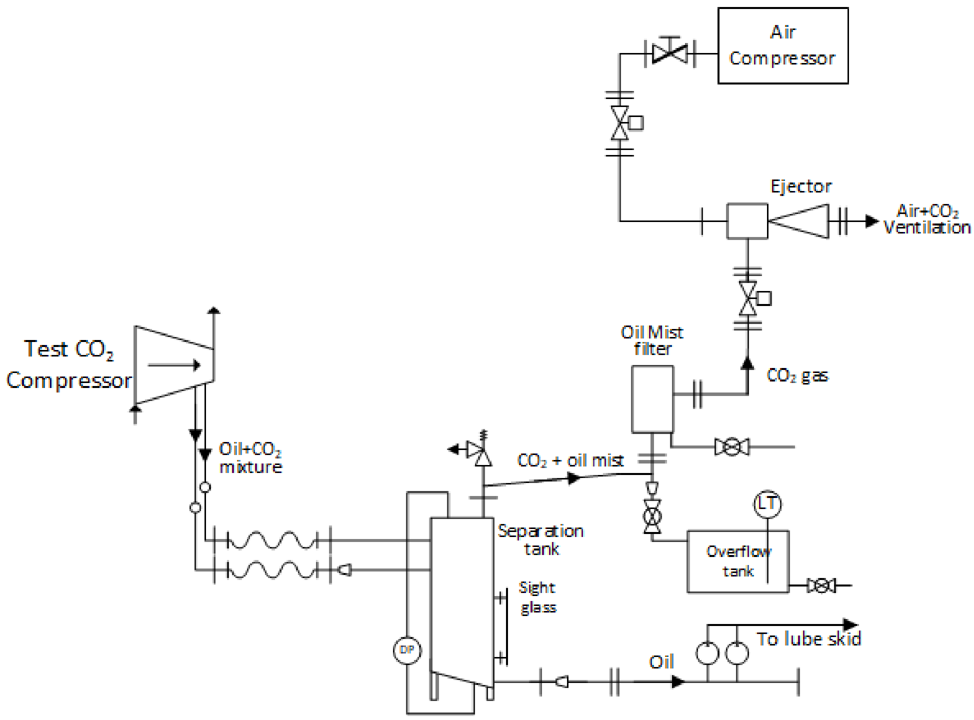


Figure 3 Schematic diagram of the lubrication oil and CO_2 scavenge system for separation of lubrication oil from the CO_2 gas to refeed the oil to the test article

The discharged mixture of oil and CO_2 gas were separated into CO_2 gas and oil by two steps. The first step of separation used gravity as a driver to separate liquid oil from the CO_2 gas as shown in figure 3. The collected oil at the bottom of the separation tank returned to the lubrication oil reservoir to recirculate into the test rig. The second step of separation used separation filter which allowed CO_2 gas pass through it while collecting any oil droplets in the flow. More than 99.9999% of lubrication oil was recovered and recirculated through this two-stage oil and CO_2 gas separation system.

Blade Tip Clearance Management

The tip clearance between the blades and the casing affects performance of the compressor, stall and surge line, and even aeromechanic behavior such as flutter [4]. And it becomes critical when it comes to safe operation of the compressor without rubbing the blades to the casing.

There are multiple parameters that affect the blade tip clearance in this rig. The radial growth of the rotor and casing due to rotation and temperature has been the major cause of the change of the tip clearance. The centrifugal loading to the blades due to rotation tends to decrease the tip clearance. And due to high pressure loading on the blade with the CO_2 gas, the blades deformation in CO_2 compressor affects the tip clearance more than air compressor [5]. Since the radius of the shroud flow path decreases as toward the compressor exit, the relative axial location of the compressor rotor again the shroud casing matters to the tip clearance. Axial displacement of the shaft and the thrust loading on the shaft, thermal growth of the shaft, and thermal growth of the casing contribute to the tip clearance.

The machining and assembly of the of the parts contribute to the clearance as well. The runout of rotor shaft or disk, the machining tolerance of the casing, especially the abradable material on the shroud casing above the rotor blades, and eccentricity of the casing and the rotor center contribute to the initial assembly (cold) clearance both in the circumferential average and circumferential distribution of the clearance. Also, the orbit of the shaft rotation is not circular most of the time. It is either elliptical or more complicated shape which makes the minimum tip clearance point in circumferential space changes with speed or with operating condition.

The running blade tip clearance(s) for all the three stages were measured from the in-situ calibrated tip clearance probes (figure 4). Four probes have been installed per stage to measure the running tip clearance.

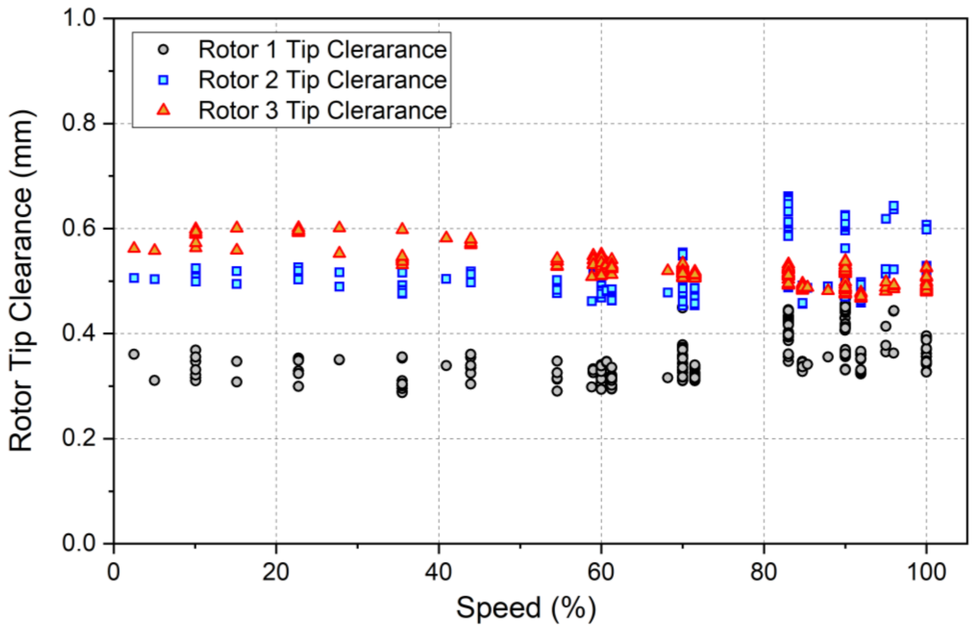


Figure 4 Measured blade tip clearances from 0 to 100% speed

Compressor Inlet Filter

The flowrate through the compressor at the design condition is larger than the total amount CO_2 present in the closed loop which means the CO_2 gas in the loop passes the test compressor more than once in every second. It is not intended but any debris in the closed-loop will circulate inside the loop and can potentially damage any components in the loop. The most critical component of the loop is the compressor blades which can easily be damaged by the foreign object(s) due to its high rotating speed. If that happens, the source of damage circulates within the closed loop more than once per every second. Then the number of debris will increase continuously while damaging the compressor more and more which can lead to a catastrophic failure of the compressor eventually. To avoid this, a cone filter with the mesh size of 240-micron was installed at the inlet of the compressor. The size was determined to generate acceptable pressure loss while protecting the compressor blades from the damage. The filter did collect some particles that would have caused large damage to the compressor during the testing if it were not for the filter.

If the filter actually collects any debris, the pressure drop across the filter increases. The cone type filer can collapse if the pressure drop becomes higher than the permissible limit. To avoid this happening, the pressure drop through the filter has been monitored through a differential pressure transducer during the operation and it was designed to trigger the automatic facility shutdown at the predefined differential pressure.

Blade Vibration Monitoring

Both synchronous and non-synchronous blade vibration can cause damage to the axial compressor blades. The speeds when the synchronous vibration would happen are predictable based on the counts of the airfoils, struts, etc. upstream and downstream of the compressor, but the amplitude of the vibration is hard to predict. And the non-synchronous blade vibration, like flutter, is very hard to predict and the simulation results of the flutter is not reliable. Especially, the aeromechanic behavior of the high density fluid, like CO2, has not been published in the open literature as far as the authors know. For this reason, blade tip timing (BTT) technology has been adopted to monitor and measure the compressor blade vibration during the testing. A synchronous blade vibration of the blade first bending mode with the 6E crossing was observed between 73% to 82% of the shaft speed.

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