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ABSTRACT

This paper presents performance measurements of a carbon dioxide two-stage reciprocating compressor system using a hot-gas bypass load stand. The experimental data collected was used to characterize the compressors' performance and evaluate oil management between the two compressors. A closed loop compressor load stand has been redesigned to conduct the compressor performance tests. Two hermetic reciprocating compressors, including two oil separators, two intercooling fans, and safety control valves, were connected to the load stand for testing. The first series of compressor tests were conducted for varying suction pressures and discharge pressures. In a second series of tests, the running time was varied and the suction pressure kept constant. Based on the acquired test measurements, compressor performance metrics, such as the volumetric and overall isentropic efficiencies, have been determined. Furthermore, compressor initial clearance factors were calculated, based on available compressor maps, which could be useful for future carbon dioxide system modeling. Also, a simulation model to predict two-stage compressor performance is presented. The entire process can be simulated to provide compressor performance data for different working conditions. Finally, the predicted performance was validated using the acquired test data.

1. INTRODUCTION

Many numerical and experimental studies of carbon dioxide trans-critical and conventional sub-critical refrigerating systems have recently received increased attention as a possible replacement for vapor compression cycle technologies that use fluorocarbon-based refrigerants. Aprea and Maiorino (2008) evaluated energy performance using an internal heat exchanger. The experimental plant employed a semi-hermetic compressor, plate-finned tube type heat exchangers, an electronically controlled back pressure valve, and an expansion valve. Rigola et al. (2005) designed a small cooling capacity unit with an evaporation temperature of 0 °C. A detailed numerical simulation model for hermetic reciprocating compressor performance, validated using conventional refrigerants, was extended to numerically obtain the CO_2 compressor prototypes behavior.

Hubacher et al. (2002) presented several performance measurements of a prototype carbon dioxide compressor using a hot-gas bypass compressor load stand. The compressor was a semi-hermetic, two-piston, single-stage, reciprocating compressor with an estimated cooling capacity of 3 tons. However, Hubacher mainly focused on how to find the volumetric efficiency. Christen et al. (2006) described performance measurements of a prototype carbon dioxide compressor using a compressor load stand based on a hot-gas bypass design were five types of compressors were used for testing. However, there were no simulation results presented in Christen's study to validate the experiment results.

With that said, the purpose of this paper is to fill the gap in open literature with experimental validation for the sub-critical cycle using hermetic compressors. To accomplish this, more detailed experimental results of a closed loop system will be analyzed. Additionally, a suitable intermediate pressure will be found using the approach of pressure recalculation once test conditions are defined. Furthermore, the simulation model presented is expected to predict a two-stage compressor systems' performance.

2. EXPERIMENTAL RESULTS

2.1 Description of system and experiments

The schematic of the closed loop test stand setup is shown in Figure 1. CO_2 was used as the working fluid during this phase of testing. The closed loop was divided into three fluid circuits: the oil return line, the bypass line, and a reduced temperature hot gas line. In the experiments presented in this paper, the reduced temperature hot gas line was used for testing only. However, due to the nature of the system design, the oil return and hot-gas bypass lines are introduced as follows.

In the oil return line, an oil separator having one inlet and two outlets, was used to separate oil that was discharged with the compressed CO_2 by the compressor system. The purified compressed CO_2 was then allowed to continue into the remaining fluid circuits. The separated oil resulting from the oil separation process, was routed to the oil return line for re-entrance to the compressor system. To ensure oil starvation did not occur, an oil measuring gauge was added to monitor oil levels in the test system to indicate if additional oil was needed.

In the bypass line, two metering valves of different sizes were used to control the CO_2 flow rate, which will affect the compressor test conditions accordingly. Coarse adjustment was conducted by a metering valve (NV106) installed in the larger size tube. Flow rate fine tuning was accomplished using a smaller needle valve (NV105).

The heat exchangers installed in the system were used to reject energy received during the compression process. Two heat exchangers were installed in parallel to enable variable cooling capacity. Both heat exchangers used tap water to carry the excess energy away from the system. A membrane tank filled with the system working fluid was also used in the reduced temperature hot gas line. The pressure inside the membrane tank was held at a constant intermediate pressure by enforcing an appropriate back pressure on the membrane. This enforced intermediate pressure provided a reliable anchor point to the cycle. If any system working gas happened to be lost through leakage of the pressure membrane, the gas inside the tank was able to continue holding the test cycle at stable operating conditions. Pressure variation in the membrane tank was not large enough to significantly change the intermediate system pressure. Thus, results from the controlled membrane will not be discussed in this paper.

Pressure relief valves were installed in each section to protect the test stand from uncontrolled pressure increases. A mass flow meter was installed to enable measurement the systems' mass flow rate. A power meter was also installed to measure the electric power input. All measured data was obtained and displayed using an appropriate DAQ system.

The compressor box of interest, consisting of two hermetic reciprocating compressors used in series to achieve the target pressure in the discharge line, was connected to the test stand. An electronic control box is installed inside the compressor box. The installed electronic control box contains an ON/OFF switch and status indicator lights.

Two adjustable pressure switches are located in the compressor box: one is installed immediately after the first stage compressor and a second is installed in the final discharge line of the compressor box. The first pressure switch is used to control activation of the second-stage compressor. Once the pressure measured by the pressure switch immediately after the first stage compressor reaches a predefined set pressure, the second compressor begins contributing to the system pressure rise. That said, during steady state testing, the initial pressure will be higher than the pressure set value for two-stage compressor activation; leading to both compressors starting and operating simultaneously. The second pressure switch, located in the final compressor box discharge line and used for safety purposes, is charged with the task of shutting the test system down after reaching a predetermined discharge pressure. There are two additional pressure switches, which are used to prevent over pressurization of the first and second compressor stages. These two additional switches are normally closed and connected the electronic box interlock circuit. Whenever pressure exceeds 4800 kPa in discharge pressure or 1100 kPa in intermediate pressure, the interlock circuit opens and the compressors shut down. Two Oil separators and filters have been used after each compressor to provide access for system oil management.

2.2 Test stand instrumentation

All measuring instrumentation is indicated in the schematic shown in Figure 1 . Temperatures were measured with T-type thermocouples with ±0.25 ºC accuracy. The inlet pressure of the open loop test stand was measured with an absolute pressure sensor that has an advertised accuracy of ±0.25%. Other pressures were measured with a pressure sensor gauge that has an accuracy of ±0.13% of full scale. A Coriolis-effect mass flow meter with an advertised accuracy of ±0.1% was also installed in the open loop test stand to measure mass flow rates of the gas. Electric power consumption of the compressors and the cooling fans were measured with power meters with an accuracy of ±0.04%. A data acquisition system was used to convert the incoming voltages from the measuring instrumentation to digital signals for transfer to a personal computer. The computer uses a proper data reduction program for further data analysis.

2.3 Test matrix

A test matrix outlining the operating conditions for the closed loop system is shown in Table 1. The maximum attainable high-side pressure of the closed loop test stand without compressor failure was obtained using CO_2 as the working fluid. Differences in temperature, mass flow rate, and oil slippage as a function of run time were evaluated during these tests.

Table 1: Test matrix of steady state testing with closed loop test stand

Compression Ratio	Low Side Pressure (kPa)	High Side Pressure (kPa)	Runtime (min)	Recording Data
20	105	2100	120	Temperature
30	105	3150	120,240	Pressure, Power
40	105	4200	240,360	Mass Flow Rate
43	105	4550	240,360	Oil Slippage

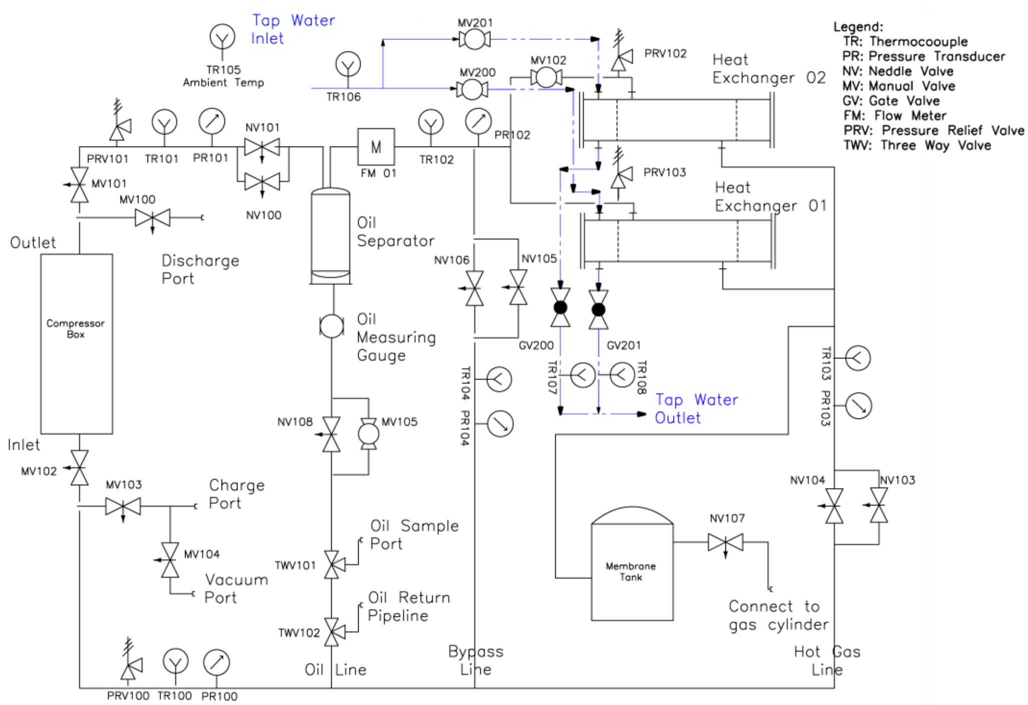

Figure 1: Schematic of the closed loop system

2.4 Experimental results

During the presented tests, the following compressor measurements were recorded: compressor mass flow rate, suction and discharge temperatures, suction and discharge pressures, intermediate pressure and temperature, and compressor power consumption. Using these measurements, the two-stage compressor system performance was evaluated. While multiple tests were conducted under various pressure levels and time durations, this paper only presents the results of the 2100 kPa testing over the 120 min testing period. The other test results lead to similar conclusions.

The steady state test with 2100 kPa high side pressure was conducted and the total run time for complete test execution was 140 minutes. The total testing time included 2 minutes for releasing pressure, 10 minutes for charging process, 120 minutes for steady state testing, and 8 minutes for shut-down testing. During the pressure release stage, only the cooling fans were set to operate. Both compressors were shut down simultaneously during the shutdown stage. Figure 3 shows the variation of the CO_2 pressures as a function of time in the closed loop test stand. It can be noticed that the initial pressure in system was approximately 1050 kPa, releasing to 630 kPa or lower to satisfy the prerequisite condition for compressor start-up.

After both compressors began to operate, it took several minutes to reach steady state. Discharge valve manipulation (MV101) was used to set the target high side pressure. Once steady state with high side pressure 2100 kPa was acquired, the intermediate pressure was set to approximately 840 kPa.

Figure 2: Closed loop system test stand

After two-hours of testing, the compressor box was shut down manually and all pressures in system returned to an equilibrium state. The new equilibrium state was higher than the initial equilibrium system pressure due to the refrigerant charge. That said, it was necessary to release the increased system pressure first before moving on to the next test to enable compressor start-up.

Figure 4 shows the variation of measured CO_2 temperature as a function of time during closed loop, steady state testing. After the compressor system was started, it was found that the 1st stage discharge temperature increased significantly to a value of 125 °C by the end of the test. The compressor thermal protector after 1st stage compressor is engineered to shut down if the temperature exceeds 130 °C. The 2*nd* stage discharge temperature, increased significantly, but to a much lower overall value than the 1st stage discharge temperature. As is shown in Figure 4, differences between these two discharge temperatures became larger during testing. It can be noticed that the temperature variations after the heat exchanger in both stages were relatively small due to the intercooler. Note that the intercooler was at a temperature close to ambient temperature during testing. Also, a small jump of the 2*nd* stage discharge temperature can be noticed in the right corner of Figure 4. This small temperature jump was due to the cooling fans being shut off with heat remaining in the compressor system after shut-down.

Figure 5 shows variation of the pressure ratio, i.e. the ratio of discharge pressure over suction pressure for each stage as a function of time for the closed loop system. It can be noticed that the pressure ratio of the 1*st* stage compressor was reached immediately after both compressors started working. This pressure ratio dropped during the test and was relatively stable at approximately 6. However, the pressure ratio of the 2*nd* stage compressor continued a gradual increase until the system was turned off. The 2*nd* stage pressure ratio was smaller than the 1*st* stage pressure ratio, which resulted in a higher discharge temperature in 1*st* stage compressor, as shown in Figure 4. The total pressure ratio was stable at approximately 18 in steady state. As discussed before, the compressors hold residual pressures at the end of the test. These residual pressures are why the 1st stage pressure ratio is larger than 1.

Measured data in the test stand side is also analyzed to allow a better understanding of the different measurements in the closed system loop. Figure 6 shows the variation of CO_2 pressures as a function of time in the test stand. Two pressures: high side pressure and low side pressure existing in this plot. During the presented two-hour test, both pressures appeared to be very stable. That said, expected pressure drops existed in Coriolis-effect mass flow meter and pipelines.

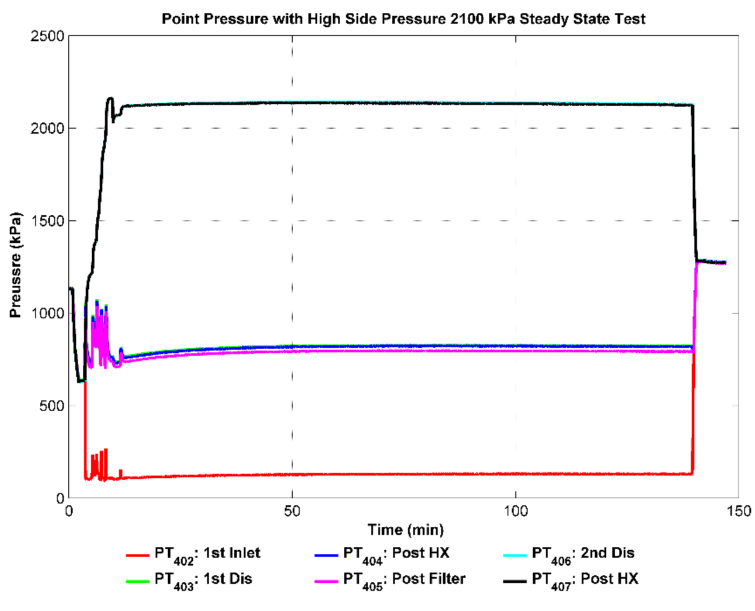

Figure 3: Point pressure with high side pressure 2100 kPa steady state test

Figure 7 shows variations of the measured temperatures as a function of time in the test stand for multiple measurement locations. All temperatures, except for TR_{100} and TR_{103}, appear to have stabilized at approximately 23 °C. Note that 23 °C was close to ambient temperature during testing. The two heat exchangers, used to reject heat from the hot discharge gas using 6 gal/min tap water, caused the post-membrane (TR_{103}) temperature measurement to decrease.

2.5 Oil management

An oil management analysis was conducted during each test. The differences of the oil weight in the discharge filters during each test were recorded to develop a general understanding of how oil migrated within the system during testing, as well as study general factors that affect oil slippage. This analysis helped to indicate how much oil was discharged with the compressed CO_2 and if additional oil was needed. To facilitate the slippage study, two filters were installed after each compressor stage. Oil weight data of the various tests performed are shown in Table 2.

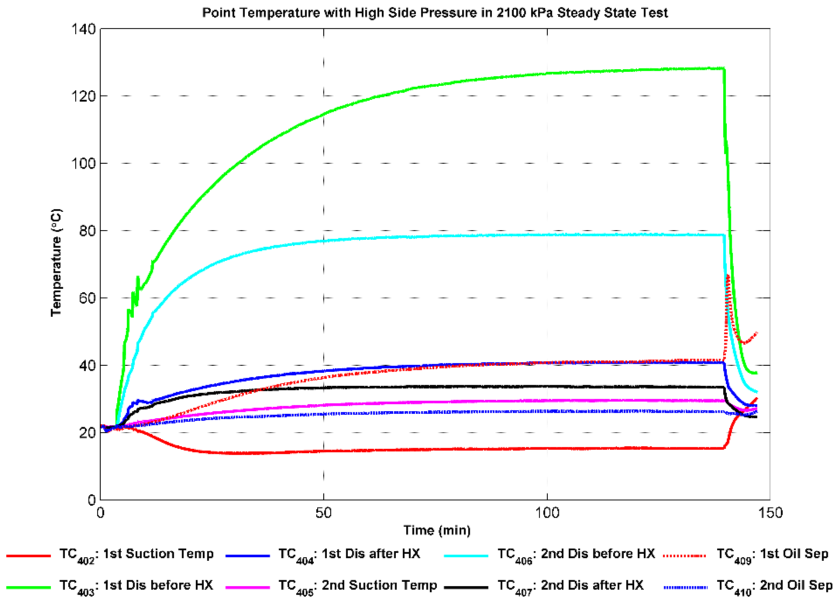

Figure 4: Point temperature with high side pressure 2100 kPa steady state test

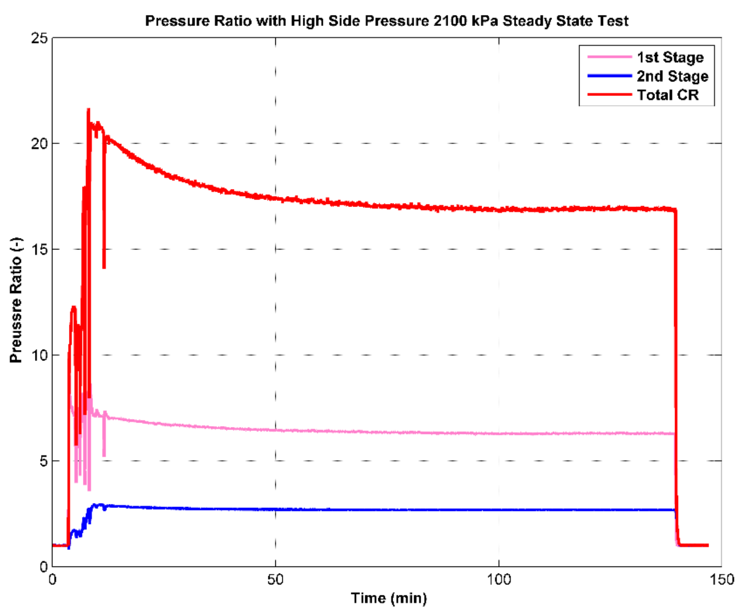

Figure 5: Pressure ratio with high side pressure 2100 kPa Steady State Test

A majority of the oil was discharged by the 1st stage compressor. Also, as expected, more oil was collected by the filters during the tests that required higher target pressures. That said, the importance of analyzing the measured data to estimate which factors affect oil slippage the most becomes apparent. It should be noted that most of the oil was collected by the first filter, which is not shown in Table 2.

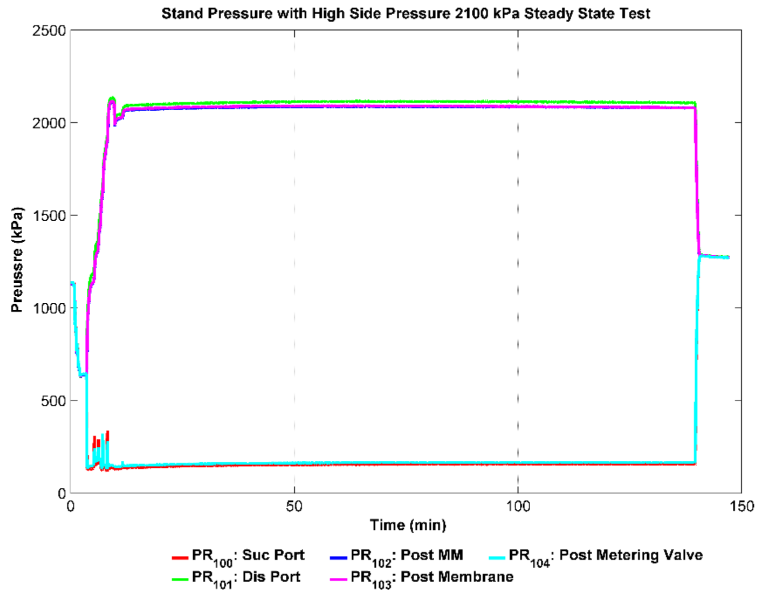

Figure 6: Stand pressure with high side pressure 2100 kPa steady state test

Based on the testing performed, it can be concluded that more oil is collected during testing in a two-stage compressor system with higher target high-side pressures. However, if the same target pressures are compared, more oil will be discharged in the lower pressure range by the 1st stage compressor than discharged in the 2*nd* stage compressor.

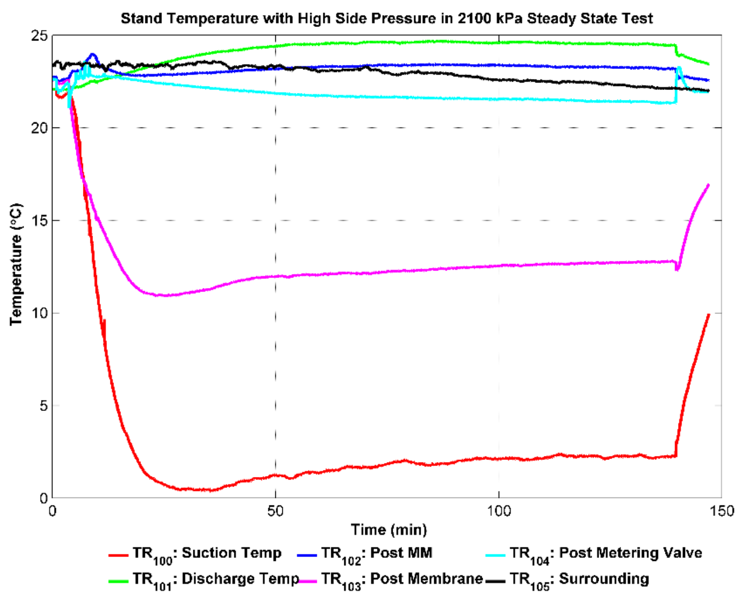

Figure 7: Stand temperature with high side pressure 2100 kPa steady state test

Table 2: Oil slippage analysis

1st Stage (g)	2nd Stage (g)	Total (g)	High Side Pressure (kPa)
9.1	0.4	9.5	2100
11.9	1.6	13.5	3150

3. SIMULATION MODEL

A simulation model to predict the two-stage compressors' performance has been developed. In this model, the initial clearance factor for the compressor was calculated based on available compressor maps that were used as an input to the steady state model. The entire compression process has been simulated to provide the compressor performance data as a function of time. Predicted performance will be validated using test data.

3.1 Clearance volume factor

The clearance volume factor is a significant value, which affects the performance of the compressors. It must be noted that the clearance volume factor is an attribute of the compressor itself and can be found through experimentation. The approach used here is to find an estimated value based on published compressor performance maps for use in the simulation model.

Compressor map data given by the compressor manufacturer provides the relation between mass flow rate, suction, and discharge conditions. Equation (1) is used to calculate the clearance volume factor. The clearance volume factor C is the slope of the function curve between the volume efficiency and the pressure ratio.

$$\eta_v = 1 - C\left[\left(\frac{P_d}{P_s}\right)^{1/m} - 1\right] = \frac{\dot{m}_e}{\dot{m}_{th}} \tag{1}$$

3.2 Intermediate pressure recalculation

In a two-stage compressor system, intermediate pressure observes the principle that the discharging gas volume from the 1*st* compressor will completely transfer to the 2*nd* compressor. However, the corresponding intermediate pressure strongly depends on the relationship between two compressor displacements. It should be noted that when the discharge condition of 1*st* stage compressor changes, the gas volume of 2*nd* compressor and the intermediate pressure will response accordingly. Therefore, the intermediate pressure value must be iteratively adjusted in the simulation to satisfy both compressor displacements. The intermediate pressure between any two stages in a two-stage compressor system can be calculated using Equation (2)：

$$V_{hj}\frac{\eta_{vj}\lambda_{Tj}\lambda_{Pj}\lambda_{lj}}{\lambda_{\varphi j}}\frac{P_j}{P_1}\frac{T_1}{T_j} = V_{hj+1}\frac{\eta_{v,j+1}\lambda_{T,j+1}\lambda_{P,j+1}\lambda_{l,j+1}}{\lambda_{\varphi,j+1}}\frac{P_{j+1}}{P_1}\frac{T_1}{T_{j+1}} \tag{2}$$

where *j* is referred to the 1*st* stage and *j* + 1 is referred to 2*nd* stage in a two-stage system.

Given parametric data for one compressor, the recalculation method was used to find the corresponding intermediate pressure for the two-stage compressor system. Based on the calculated intermediate pressure value; power consumption, and gas flow rate can be updated accordingly in each loop. The recalculation process is as follows:

1) Related Coefficient:
In the recalculation method, all related coefficients are assumed to be a constant value for both compressors respectively, as shown in Table 3.

Table 3: Compressor coefficient

Stage	λ_p	λ_T	λ_l	$\lambda_0 = \lambda_p \lambda_T \lambda_l$
1st	0.96	0.96	0.92	0.848
2nd	0.98	0.97	0.92	0.874

2) Displacement guess value:
   Define a displacement guess value to be used for estimating the suction pressure and pressure ratio in each stage. Assume 3 °C increase in the 2*nd* stage due to incomplete intercooling.

$$V_{hj,guess} = V_{hj} \frac{T_1}{T_j} \lambda_{0j} \tag{3}$$

   where $\lambda_{0j} = \lambda_{Pj}\, \lambda_{Tj}\, \lambda_{lj}$ and T_j is the suction temperature in each stage.

3) Equation (4) is used to update pressure and the volumetric coefficient with a guess value. Results are shown in Table 5:

$$P_{sj} = P_{s1} \frac{V_{h1,guess}}{V_{hj,guess}} \frac{\eta_{v1}}{\eta_{vj}} \tag{4}$$

Table 4: Displacement guess value

Stage	$V_{hj}(10^{-5} m^3)$	$T_j(K)$	$\lambda_{0j}(-)$	$V_{hj,guess}(10^{-5} m^3)$
1st	3.436	291	0.848	2.91336
2nd	0.424	294	0.874	0.36703

Table 5: Update pressure and volumetric coefficient

Stage	$P_{sj}(kPa)$	$P_{dj}(kPa)$	$\varepsilon_j(-)$	$\eta_{vj}(-)$
1st	112	886	7.9375	0.6766
2nd	886	2101.5	2.3708	0.8707

4) Convergence Criteria:
   Based on Equation (2), the suction gas volume in each stage was converted to the volume at the suction condition in the 1*st* stage, which is shown in Equation (5):

$$V_j = \frac{P_{sj}}{P_{s1}} V_{hj,guess} \eta_{v1} \tag{5}$$

   In all results of V_j for each stage, if $V_{min}/V_{max} > 0.97$, the recalculation process stops and results converged.

Table 6: Pressure recalculation in each stage

Iteration	Stage	$P_{sj}(kPa)$	$P_{dj}(kPa)$	$\varepsilon_{j,update}(-)$	$V_j(10^{-5} m^3)$	$V_{\min}/V_{\max}(-)$	Converged (N/Y)
1	1st	112	688.7	6.168	2.257	0.8151	N
	2nd	688.7	2101.5	3.051	1.8397		
2	1st	112	845.4	7.597	2.079	0.8699	N
	2nd	845.4	2101.5	2.487	2.391		
…..							
5	1st	112	757.7	6.786	2.177	0.9562	N
	2nd	757.7	2101.5	2.773	2.083		
6	1st	112	768.6	6.884	2.165	0.9793	Y
	2nd	768.6	2101.5	2.734	2.121		

Final results from the last iteration were used for cycle calculations, as shown in Table 7 :

Table 7: Final converged results

Stage	$P_{sj}(kPa)$	$P_{dj}(kPa)$	$\varepsilon_j(-)$
1st	112	768.6	6.884
2nd	768.6	2101.5	2.734

Polytropic coefficients for two compressors operating in series can be obtained using experimental data. These are a function of the pressure ratio and experiment time, as shown in Equation (6) and (7). Equation (8) was used to predict the discharge temperature by using the pressure ratio and polytropic coefficients. P-h and T-s diagram, shown in Figure 8 and 9, can then be derived to show the entire gas cycle. In the cycle diagram, all temperature and pressure in each point, at 100 min running time, can be seen clearly and validated by the measurement data discussed above.

$$n_{T,1} = 1.0013 + 2.6944e^{-4} \times t + 7.4677e^{-4} \times PR - 1.2135e^{-7} \times t^2 - 1.9413e^{-5} \times t \times PR \quad (6)$$

$$n_{T,2} = 0.0023416 - 0.016275 \times t + 3.2629 \times PR - 3.9045e^{-5} \times t^2 + 0.019146e^{-5} \times t \times PR \quad (7)$$

where t is duration of time and PR is pressure ratio in each stage.

$$T_{out} = T_{in}\left(\frac{P_{out}}{P_{in}}\right)^{\frac{n_T - 1}{n_T}} \quad (8)$$

where n_T is polytropic coefficient

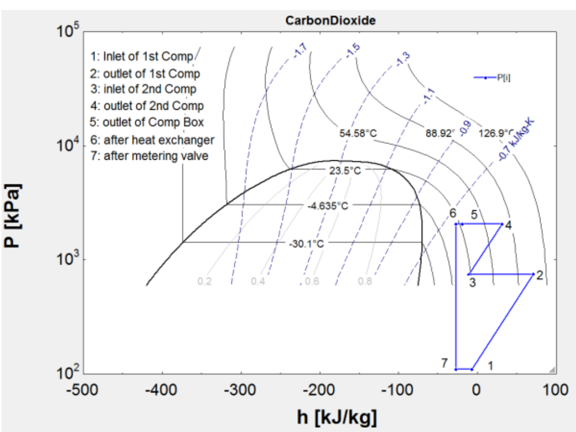

Figure 8: P-h diagram in closed loop system

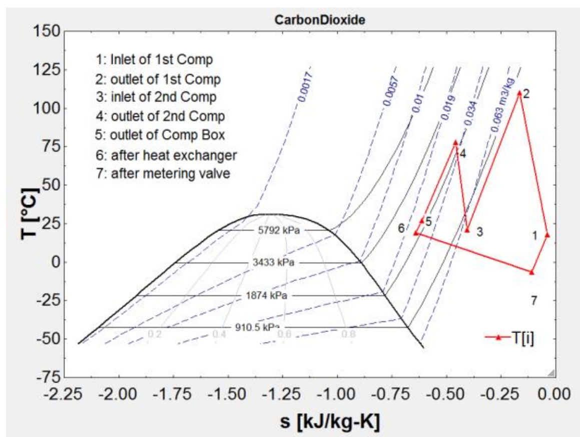

Figure 9: T-s diagram in closed loop system

4. CONCLUSION AND FUTURE WORK

Several experimental measurements with different target pressures and run times have been conducted using CO_2 as the working fluid. The following parameters have been recorded: pressures, temperatures, surface temperatures, and pressure ratios.

At the 2100 kPa steady state test condition, the intermediate pressure is 750 kPa and the maximum discharge temperature is approximately 125 °C.

Oil management studies have been conducted to track oil slippage between the two compressor stages at different target operating pressures and run times. The dominant factor affecting oil slippage in a two-stage compressor system is the pressure ratio. In lower pressure ranges, high pressure ratios will lead to more significant oil slippage.

A simulation model has been created to predict compressor performance and model predictions were validated through the use of the acquired test data. The simulation studies presented in the paper employed a pressure recalculation method and correlation equations which were derived using experimental data. The resulting model is used to analytically predict temperature which potentially leads to some inaccuracy. This inaccuracy is likely a consequence of the temperature calculation being somewhat inaccurate due to a lack of accounting for the 1st to 2nd stage intercooler, as well as heat transfer effects between the compressor shells and gas. Therefore, future work should include usage of a more accurate differential model. Additionally, heat transfer and back pressure calculations should be investigated to form a more complete understanding of the actual two-stage compressor steady-state process.

NOMENCLATURE

η_v	volumetric efficiency	(–)	**Subscripts**	
n	speed of compressor	(r)	d	discharge
m	expansion efficiency	(–)	dis	discharge
V_h	displacement	(m^3)	s	suction
λ_p	pressure coefficient	(–)	e	experimental
λ_l	leakage coefficient	(–)	th	theoretical
λ_T	temperature coefficient	(–)	L	entering tank
λ_ϕ	condensation coefficient	(–)		
C	clearance volume factor	(–)		
ρ	density	(kg/m^3)		
u	specific internal energy	(kJ/kg)		
t_{step}	time in each loop	(s)		
h	convection coefficient	($kJ/m^2 - K$)		
A	heat transfer area	(m^2)		

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