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Numerical Comparison of NASA's Dual Brayton Power Generation System Performance Using CO_2 or N_2 as the Working Fluid

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Abstract

A Dual Brayton Power Conversion System (DBPCS) has been tested at the NASA Glenn Research Center using nitrogen (N_2) as the working fluid. This system uses two closed Brayton cycle systems that share a common heat source and working fluid but are otherwise independent. This system has been modeled using the Numerical Propulsion System Simulation (NPSS) environment. This paper presents the results of a numerical study that investigated system performance changes resulting when the working fluid is changed from gaseous (N_2) to gaseous carbon dioxide (CO_2).

Introduction

As part of the Exploration Technology Development Program's Fission Surface Power project (FSP), the NASA Glenn Research Center procured and tested a Dual Brayton Power Conversion System (DBPCS). Testing of the DBPCS has previously been reported (Ref. 1). The DBPCS has also been modeled extensively using the Numerical Propulsion System Simulation (NPSS) (Refs. 1 to 4) environment.

The DBPCS tested at the NASA Glenn Research Center used nitrogen (N_2) as the working gas; however, the use of carbon dioxide (CO_2) had been suggested due to its higher molecular weight. Unfortunately, the DBPCS test program has been terminated due to a change in the power requirements anticipated for space power generation in the foreseeable future. Nevertheless, the availability of a simulation of the system allowed the qualitative exploration of a change in the working fluid. The Closed Cycle System Simulation (CCSS) software provided a component level representation of the system that allowed a qualitative study of the differences of operation of the system using N_2 and the system using CO_2.

Dual Brayton Power Conversion System Description

An isometric drawing of the DBPCS tested at the Glenn Research Center is shown in Figure 1. All components in the system were commercially available but required modifications to create a closed cycle test rig. The core of the system consisted of two Capstone C30 open-loop microturbine engines where the combustion chamber had been modified to allow the working fluid to be heated by an external heater. This electric heater, shown in Figure 1, was the only common component in the system. Also shown in Figure 1 and provided in Table 1 are the pressure and temperature measurement locations for data

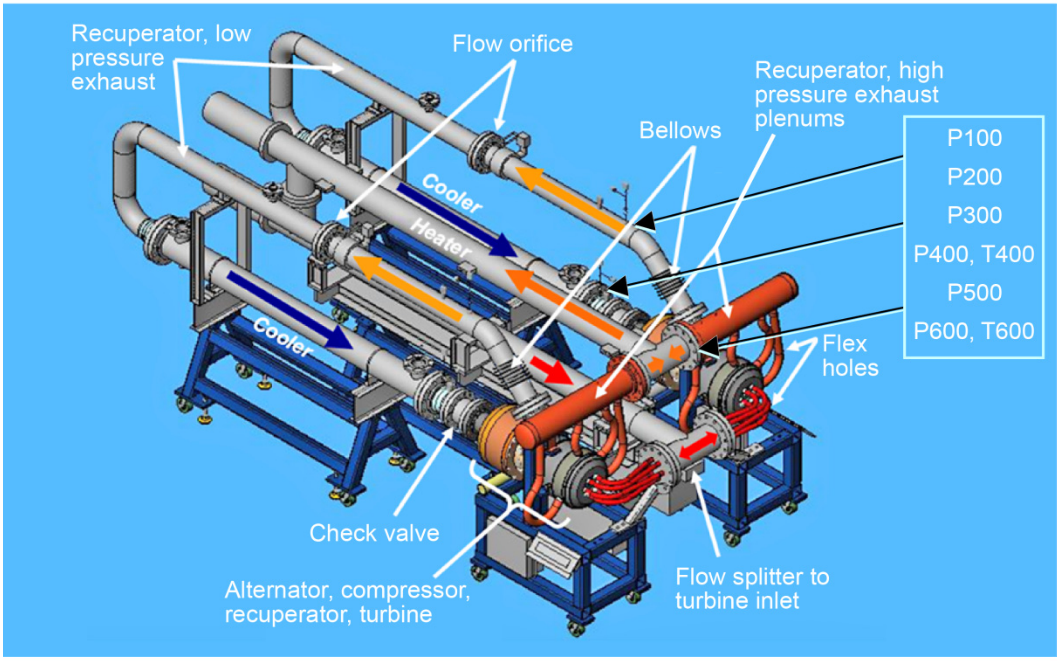

Figure 1.—Dual Brayton power conversion system (Refs. 1 and 5).

TABLE 1.—DBPCS MEASUREMENT STATIONS (REF. 5)

Measurement	Location, Definition
P100, T100	Heater outlet
P200, T200	Turbine outlet
P300, T300	Recuperator outlet (Cooler inlet side)
P400, T400	Cooler outlet (Compressor intake side)
P500	Recuperator inlet (Compressor outlet side)
P600, T600	Heater inlet (Recuperator outlet side)

acquired during testing. Here the “Pxxx” indicate static pressures and “Txxx” designate static temperature measurements. Instrumentation was symmetrical for both “sides” of the DBPCS. All high temperature sections of the test rig were covered with thermal insulation blankets to reduce heat transfer to the ambient environment.

Unfortunately, physical limitations prevented the placement of thermocouples at the compressor and the turbine exits resulting in a lack of temperature information. The lack of these thermocouples prevented the direct measurement of either the compressor or turbine performance.

Closed Cycle System Simulation

Closed Cycle System Simulation was developed within the NPSS software architecture. NPSS is an aviation industry cycle simulation developed through a NASA/industry consortium that included the major airframe and gas turbine engine manufacturers (Ref. 6). This one-dimensional performance model is characterized by flexibility, adaptability, and ease of use. Johnson and Hervol (Ref. 3) present a detailed explanation of the component models. Although it was developed principally for performance modeling of gas turbine applications, the environment has been used to develop fuel cell, altitude wind tunnel, and closed cycle models. The source code for this model originates from the NASA Closed Cycle Engine Program (CCEP) (Ref. 7), an in-house legacy code.

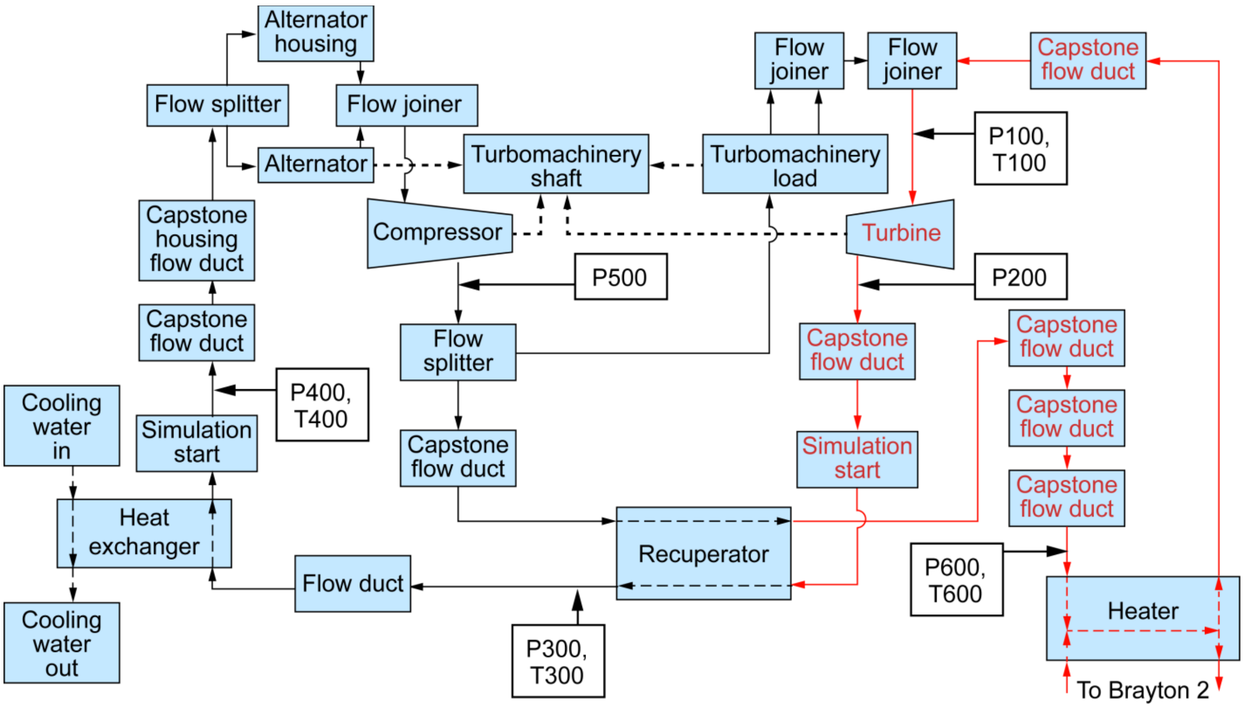

Figure 2.—CCSS model schematic.

A schematic depicting one half (side) of the simulated system is shown in Figure 2 (Ref. 1). With the exception of the heater, the simulation duplicates this schematic twice, once for each side of the DBPCS. Relative locations of the DBPCS measurement stations are also shown.

Development of the model required definition of the physical dimensions of the components (pipes, bellows, etc.), turbomachinery performance maps, and recuperators' and coolers' performance, working fluid properties, and the components' layout (bends, etc.). Inputs required to run the models include charge pressure (constant, 90 kpa), heater exit temperature (840 to 950 K), water coolant temperature (294 K), coolant mass flow, and shaft speed (50 to 90 krpm) (Ref. 1). An important software modification for the present study was the addition of CO_2 gas property tables. The inclusion of these tables allows an easy transition between N_2 and CO_2 modeling.

Turbine and compressor maps are required for this software. They can be provided in a number of ways but for this project, the maps were generated analytically using Centrifugal Compressor Off Design Code (CCODP) (Ref. 8) and Radial Turbine Off Design Code (RTOD) (Ref. 9). These codes, written at the Lewis Research Center (now Glenn Research Center) use simple one-dimensional calculations and correlations of experimental data to provide off-design performance of centrifugal compressors and radial turbine. Inputs include operating conditions, working fluid and machine geometry.

Modifications to CCSS

Modifications made to CCSS to allow the software to run CO_2 were relatively minor and consisted of the following four general areas:

1. Insertion of the turbomachinery maps for CO_2,
2. Modifications to the element that describes ducts in the software,
3. Modifications to the model and run files that allow an easy change between working fluids,
4. Removal of the N_2 specific correction factors that improved the correlation between the simulation and the experiment. Unfortunately, no experimental data was available to develop CO_2 correction factors.

Insertion of the turbomachinery maps was straightforward. However, the maps provided did not have the same component inlet reference conditions. The N_2 compressor pressure ratio map was provided as inlet pressure over outlet pressure while the CO_2 map was given as outlet pressure over inlet pressure. The software was modified to accommodate these types of differences between turbomachinery performance maps.

The duct element originally used in CCSS was upgraded with modifications to the calculations of duct turning losses and Reynolds number. In addition, losses for sudden expansions and contractions were added. The duct between the turbine exit and the cooler inlet was split into two ducts to allow more accurate modeling of the frictional losses in the bellows. Finally, the model representation was improved by adding the ability to model the splitting of ducts into a set of parallel ducts (such as between the heater and the turbine). These modifications did not significantly change the results for the ducts in this simulation. Total pressure difference $\left(\left(Pt_{\mathrm{original}} - Pt_{\mathrm{revised}}\right) / Pt_{\mathrm{revised}}\right)$ that resulted from the more detailed loss representations for any particular duct in the system was less than ±5 percent.

Changes to the model and run files were straightforward and simple. A switch was added to specify the working gas that directed the model to read the correct turbomachinery maps, gas tables, and molecular weight.

Finally, N_2 correction factors previously added to the DBPCS model were removed. As has been previously noted (Ref. 1), these correction factors had been added to the simulation to make it more closely predict the actual DBPCS performance. This allowed a more realistic comparison between N_2 and CO_2 predictions. It is unlikely that the same correction factors would apply to both working fluids. Furthermore, no experimental data exist for DBPCS operation with CO_2, thus no CO_2 specific correction factors could be created.

The DBPCS was one of the most complex systems ever modeled within the NPSS environment. Several new modules needed to be developed. Nevertheless, the final representation proved to be robust and representative. The NPSS environment proved to be very adaptable.

Comparison Between Experimental Data and Unscaled Predicted Results

Figures 3 and 4 compare unscaled predicted station temperatures and measured station temperatures for the 80000 rpm, 950 K operating condition (temperature set at the heater exit). Here, the most notable observations are that there exists in the heater a pressure drop that is not accounted for in the model and, more importantly, the measured pressure drop—and energy taken out of the flow by the turbine—is significantly lower in the measured data than in the predicted results. The temperature plot shows the temperatures out of the recuperator are significantly higher than the model predicts, while the pressure drop is lower. Thus the model under predicts recuperator thermal performance and appears to over predict turbine performance. Unfortunately, the lack of experimental compressor and turbine exit temperature data make it impossible to identify overall system performance prediction errors with certainty.

These differences do not explain the differences between the predicted power output for this condition of 26.54 kW and the measured power output of 18.15 kW. This represents a substantial difference in power outputs. Possible causes for this discrepancy are currently being examined.

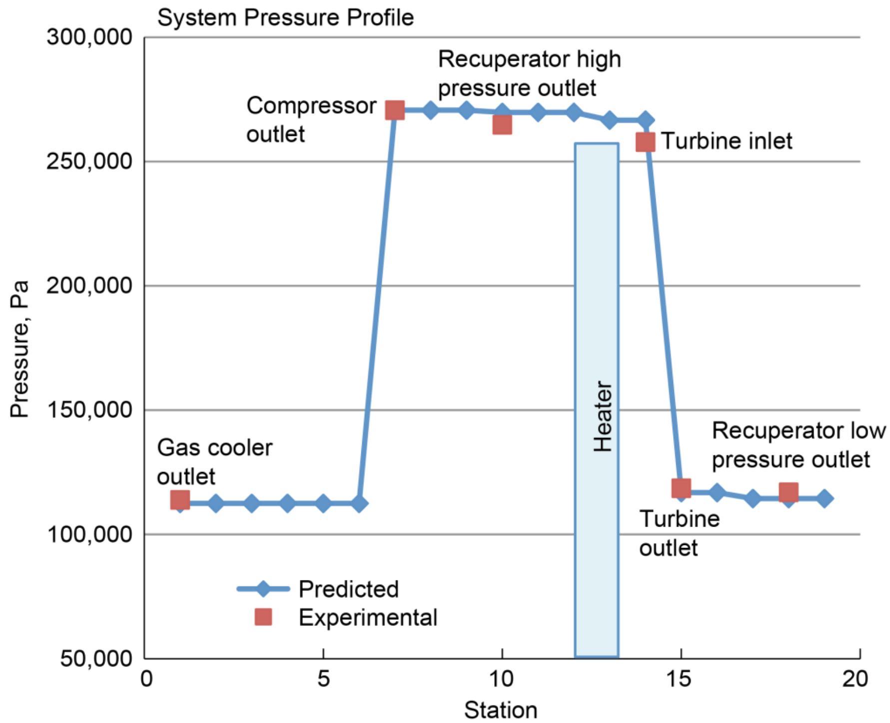

Figure 3.—Pressure versus unscaled simulation station.

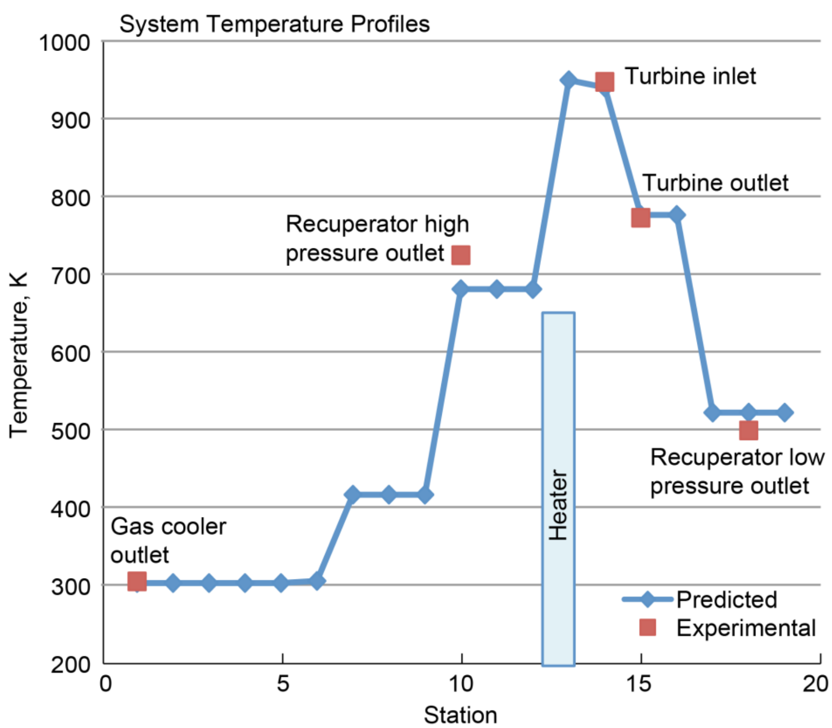

Figure 4.—Temperature versus unscaled simulation station.

Comparison Between Predicted Results Using Nitrogen and Carbon Dioxide Working Fluids

Turbomachinery maps for both N_2 and CO_2 were provided by Jeff Noall of Barber Nichols Inc. (Ref. 10) using NASA legacy codes CCODP (Ref. 8) and RTOD (Ref. 9). These maps were generated using CCODP for the compressor and RTOD for the turbine. The design point for the Capstone compressor is 92000 rpm. However, the compressor performance prediction code was unable to find a solution for the given compressor geometry above 75000 rpm. Therefore, 73000 rpm was arbitrarily selected as 100 percent design speed, allowing a 2 percent overspeed limit for the compressor. Certainly turbomachinery specifically designed with CO_2 as the working fluid would have allowed operation at higher speeds.

Figure 5 shows the predicted compressor pressure ratio and efficiency. The compressor operation is acceptable at lower speeds with the different working fluid. As would be expected, performance using CO_2 begins to drop off above 60 k rpm and is significantly worse with efficiency at 75000 rpm being almost 4 percent worse than at 60 k rpm and 2 percent lower than the equivalent N_2 efficiency. Figure 6 indicates the turbine is operating poorly throughout the operating range and is badly designed for operation with CO_2 in this system.

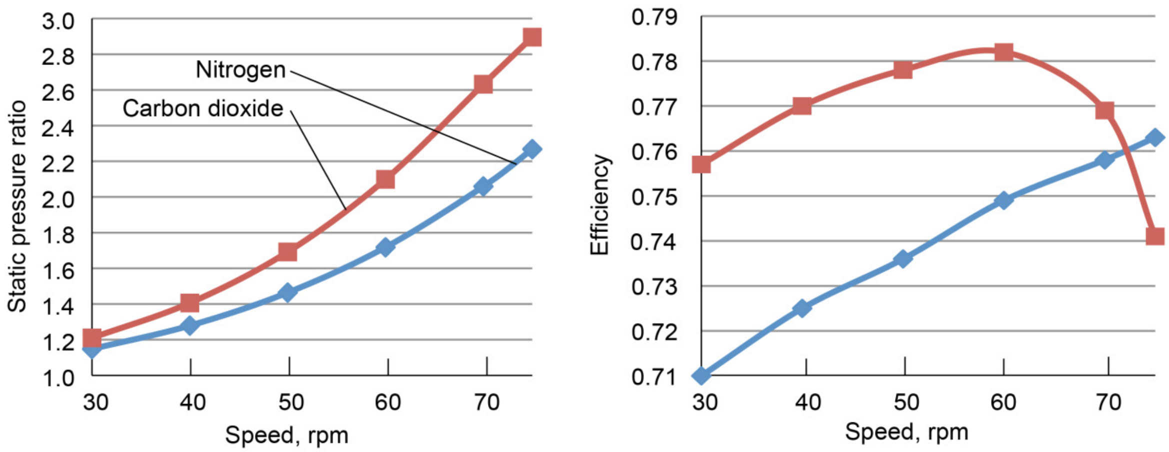

Figure 5.—Compressor performance.

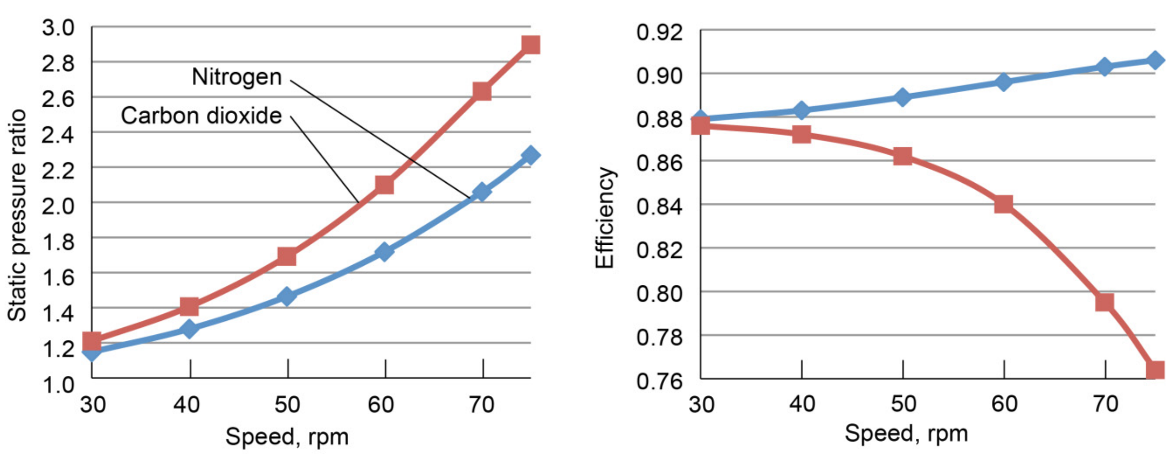

Figure 6.—Turbine performance.

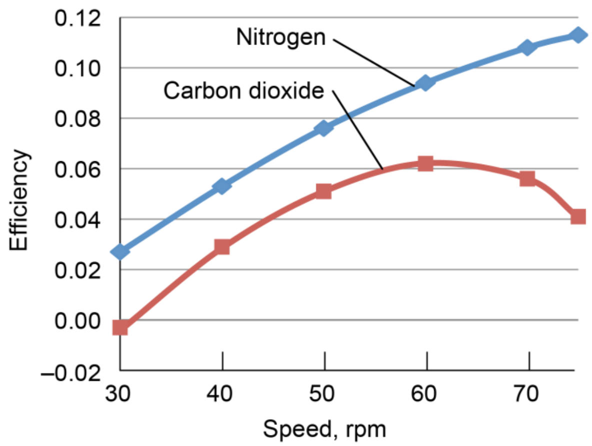

Figure 7.—Cycle efficiency.

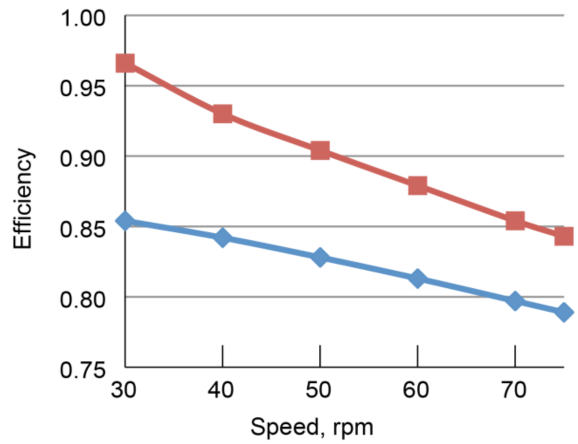

Figure 8.—Recuperator effectiveness.

The turbine configuration is not optimum as the efficiency of the turbine drops almost 14 percent as the speed of operation increases from 30000 to 75000 rpm. Should operation of the DBPCS be desired, the turbine and compressor would need to be redesigned for optimum performance.

Figure 7, the system efficiency shows that the system runs significantly worse with CO_2 as the working fluid with the cycle efficiency, at best, around 2.5 percent lower and at 75 percent rpm the cycle efficiency almost 8 percent lower. This is despite improved recuperator effectiveness (~10 percent) while using CO_2 (Fig. 8).

The lower efficiency for CO_2 operation is a result of the higher molecular weight of CO_2 (MW 44 for CO_2 versus 28 for N_2). The cycle efficiency is calculated using the formula:

$$\eta_{\text{cycle}} = \frac{\text{Power}_{\text{alternator}}}{W * (H_o - H_i)} \tag{1}$$

where η_{cycle} is the cycle efficiency, W represents massflow and (H_o–H_i) calculated the specific enthalpy change across the heater. The significantly larger molecular weight will reduce efficiency without a corresponding increase in power output or change in ΔH. For example, at 50000 rpm CO_2 develops ~35 percent more power but requires ~57 percent more massflow. This coupled with a slightly higher ΔH (~7 percent) results in the significantly lower recuperator effectiveness (ε_{rc}), defined as:

$$\varepsilon_{rc} = \frac{cap_{h*}(Tin_h - Tout_h)}{cap_{c*}(Tin_h - Tin_c)} \tag{2}$$

where *cap* is the heat capacity, *Tin* is the inlet temperature, and *Tout* is the outlet temperature. The *h* and *c* subscripts represent the hot and cold flows, respectively. The hot side and cold side heat capacities will normally be the same.

This much larger massflow is likely the result of the difference in molecular weight. Certainly at higher speed where both turbine and compressor move significantly away from optimal operation (Figs. 9(b) and 10(b)), the cycle efficiency for CO_2 operation decreases notably.

Figure 9 provides the compressor maps for the system operating with N_2 (Fig. 9(a)) and CO_2 (Fig. 9(b)). These figures also provide the predicted operating line for the components at speeds between 30 and 75 k rpm. The operating line for N_2 was to the left of the map, beyond the predicted surge line. The predicted operating line for CO_2, however, is well to the right of the surge line. The maps provided reflect the conservative predictions of the CCODP software, which will likely underpredict the true

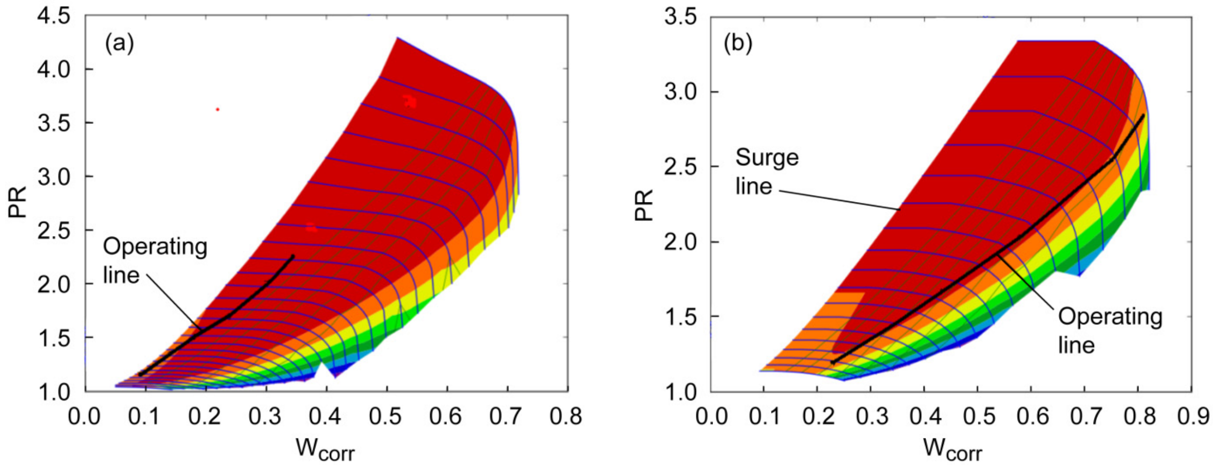

Figure 9.—Compressor operating maps. (a) Nitrogen. (b) Carbon dioxide.

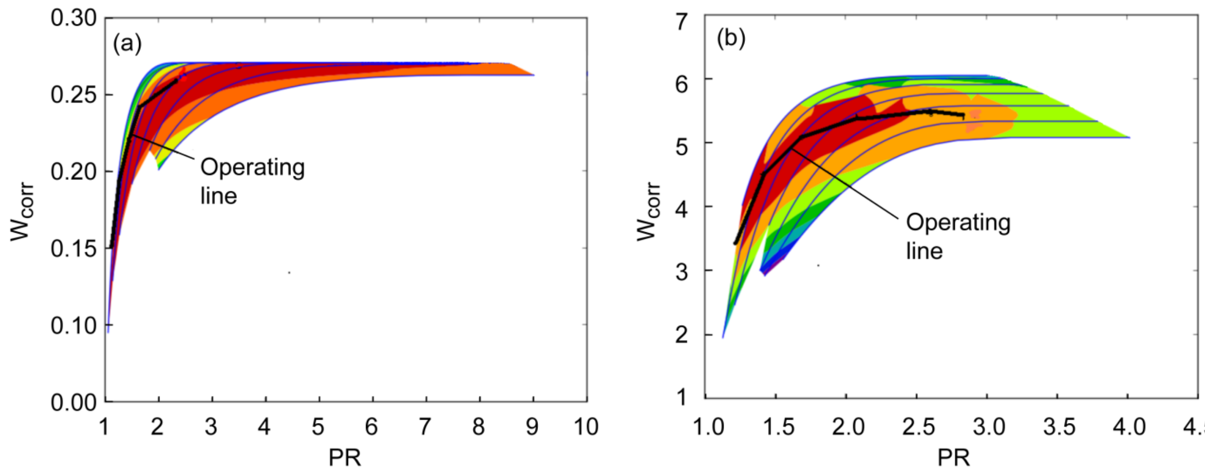

Figure 10.—Turbine maps. (a) Nitrogen. (b) Carbon dioxide.

operating range of a tubomachinery configuration. Since stable operation beyond the surge line is not possible, the CCSS solution algorithms will extrapolate the stable region of operation of the turbomachinery components to find a converged solution. Figure 9(a) shows the extended compressor map for N_2. It is clear that compressor operation on N_2 remains in the higher efficiency regions of the map throughout the range of interest while compressor efficiencies drop are lower during operation with CO_2 at both low and high rpm.

Figure 10 provides the turbine maps for N_2 and CO_2. As with the compressor maps, the turbine operates in high efficiency regions with both gases except with CO_2 at higher speeds.

Figure 11 shows the power output comparison between N_2 as the working gas and CO_2. Table 2 provides specific information at an operating speed of 75000 rpm. Note that the system efficiency provided in Table 2 is simply the electric power provided by both generators divided by the electric power provided to the electric heater.

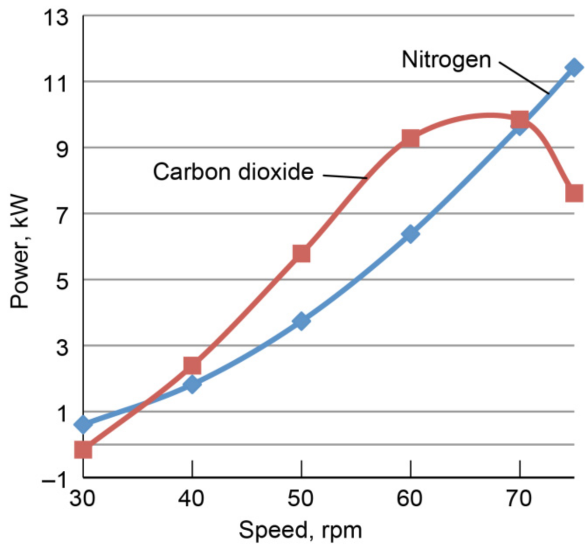

Figure 11.—Power output versus engine speed.

TABLE 2.—OPERATING PARAMETERS AT 75000 rpm

Parameter	CO_2	N_2
Shaft speed, rpm	75000	75000
Turbine power, kW	44.679	31.564
Compressor power, kW	–36.023	–19.342
Power output, kW	7.614	11.425
Compressor pressure ratio	2.896	2.268
Turbine pressure ratio	2.782	2.212
Recuperator effectiveness	0.843	0.789
Turbine efficiency	0.764	0.906
Compressor efficiency	0.741	0.763
System efficiency	0.023	0.047
Cycle efficiency	0.041	0.113

It is interesting to note that even though the turbomachinery performance with CO_2 as the working gas is consistently and significantly worse than that of N_2, overall power output is higher until very high operating speeds. This result suggests that the use of CO_2 as the working fluid in closed Brayton power generation units should be explored experimentally in the future.

Results and Conclusions

The following results and conclusions can be made:

- A closed cycle dual Brayton power system, designed to be operated with N_2, can successfully use CO_2 as the working fluid.
- Turbomachinery designed with N_2 (or air) as the working fluid will not operate as efficiently with CO_2 as the working fluid. Compressor to rotational speed will be limited and the turbine operating points will likely be far from optimum operating conditions.
- Simulations indicate that power production using CO_2 can be higher despite lower operating efficiencies but require turbomachinery designed for the operating fluid to achieve optimum performance.

- This higher power production from the same physical system using CO_2 as a working fluid suggests that systems where space and/or weight are constrained, CO_2 Brayton cycle systems may offer significant advantages.
- The NPSS environment proved readily adaptable to the development of a system that is far from the type of systems that NPSS was originally developed for.

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13. SUPPLEMENTARY NOTES

14. ABSTRACT
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