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Effect of Compressor Inlet Temperature on Cycle Performance for a Supercritical Carbon Dioxide Brayton Cycle

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ABSTRACT

Supercritical carbon dioxide (sCO_2) power cycles take advantage of the high density of CO_2 near the critical point to reduce compressor power and increase cycle efficiency. However, properties vary drastically near the critical point leading some cycle designers to select a more stable operating point away from the critical point. The Naval Nuclear Laboratory built and tested the 100 kWe Integrated System Test (IST) to demonstrate the ability to operate and control an sCO_2 Brayton power cycle over a wide range of operating conditions. Since the purpose of the IST was focused on controllability, the design compressor inlet conditions were selected to be 8.2°F (4.6°C) and 270 psi (18.4 bar) above the critical point to reduce the effect of small variations in compressor inlet temperature and pressure on density. This paper presents results of operating the IST over a range of compressor inlet temperatures with either fixed system mass or fixed compressor inlet pressure and the resulting effect on cycle performance.

INTRODUCTION

Supercritical carbon dioxide (sCO_2) power cycles offer the potential for higher system efficiency than other energy conversion technologies, especially when operating at temperatures above 450°C (842°F) [1]. Additionally, sCO_2 power cycles could enable more compact arrangements than other technologies due to the small turbomachinery size enabled by higher fluid density at the turbine. These benefits have led to sCO_2 power cycles being actively developed for a wide range of applications including nuclear, waste heat recovery, concentrated solar power, and fossil power sources.

The high fluid density of sCO_2 near the critical point allows the sCO_2 power cycles to use significantly less compressor power than ideal gas Brayton cycles. However, designing for operation near the critical point can result in issues during operation as compressor inlet conditions change due to the non-linear behavior and dramatic variation in fluid properties near the critical point. For this reason, it is important to assess the behavior of an operational system relative to changes in compressor inlet conditions.

sCO_2 power cycle development efforts at Naval Nuclear Laboratory focused on Brayton cycle thermodynamics, system control modeling, power cycle demonstration, and heat exchanger development. Naval Nuclear Laboratory designed and operated the Integrated System Test (IST) to demonstrate operational, control, and performance characteristics of an sCO_2 Brayton power cycle over a wide range of conditions including rapid transients between steady-state power levels [2-15]. The IST was a simple recuperated closed loop sCO_2 Brayton system with a single recuperator, a variable-speed turbine-compressor, and a constant-speed turbine-generator (Fig. 1). The IST was designed to generate nominally 100 kWe at a relatively modest turbine inlet temperature of 570°F (299°C) and shaft speeds of up to 75,000 rpm. The resulting design cycle efficiency of 12.5% is low relative to proposed larger scale sCO_2 plants due to the modest turbine inlet temperature and low turbine and compressor efficiencies inherent with the small size of the turbomachinery. The performance of the turbomachinery bearings and seals and higher than anticipated windage losses [13] limited IST operation to a maximum output of 40 kWe with a maximum turbomachinery shaft speed of 60,000 rpm. The lower power limit resulted in reducing the normal operating turbine inlet temperature to 440°F (227°C) to increase the control range of compressor speeds achievable within the power limits. While this lower output power increased the range of compressor speeds and recirculation valve positions that could be tested, it further reduced overall cycle efficiency.

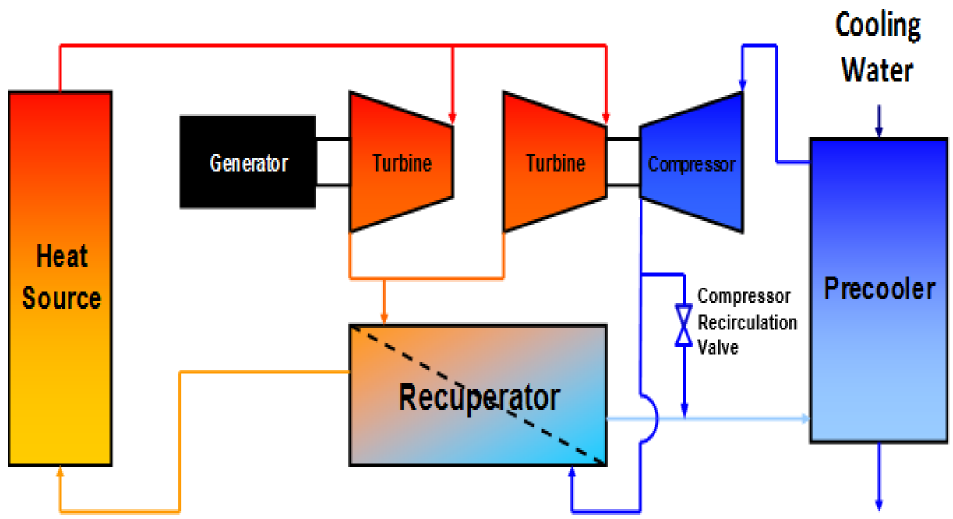

Fig. 1 IST simple recuperated Brayton Cycle

Normal IST power operations were performed with a thermal-hydraulic lead control system. In this control mode, the power output from the system was directly controlled by changing the turbine-compressor speed through feedback control of the compressor recirculation valve. The turbine-compressor speed setpoint was determined from a lookup table based on the commanded system power level; there was not a direct feedback mechanism to account for differences between commanded and actual system output power. The turbine-generator speed was maintained at a constant 60,000 rpm setpoint with the applied load being regulated to maintain the speed.

TEST CONFIGURATION

Three parametric tests were performed with the IST to isolate and analyze the effects of compressor inlet temperature on cycle, compressor, and recuperator performance:

- Constant compressor speed (constant mass) – The compressor speed setpoint was fixed and the compressor recirculation valve position was allowed to change to maintain the shaft speed to the setpoint as the compressor inlet temperature was varied. This condition is relevant to systems with a single shaft in which the compressor must always run at fixed speed due to generator frequency.

- Constant compressor recirculation valve position (constant mass) – The compressor recirculation valve position was fixed and the compressor speed was allowed to change as compressor inlet temperature was varied. This was done to show how compressor speed must vary to compensate for temperature changes in constant resistance systems that do not have the need or features to maintain a fixed speed.

- Constant compressor inlet pressure – Again, the compressor speed was held constant, but in this test the system mass was adjusted at each compressor inlet temperature to maintain a constant compressor inlet pressure to compensate for the effect that varying the compressor inlet temperature has on system pressure and therefore compressor inlet density as could be done in a system with an inventory control system.

All three tests were performed over the maximum range of compressor inlet temperatures that could be tested while maintaining system output power between 5 and 40 kWe to insure stable operation and maintaining the compressor recirculation valve above the minimum controllable operational position of 6% open. Each test was started at a baseline compressor inlet condition of 96°F (35.5°C) and 1330 psia (90.5 bar) and a turbine-compressor speed setpoint of 48,650 rpm. These conditions equated to a compressor recirculation valve position of approximately 30% open and a system output of 30 kWe. Compressor inlet temperature was varied in 2°F (1.1°C) steps at a slow rate and maintained at each setpoint for a minimum of 3 minutes to ensure a stable steady-state condition was achieved.

RESULTS AND DISCUSSION

The range of achievable compressor inlet temperatures was different for each of the three tests. For the constant compressor speed test, the achievable range was from 90°F (32.2°C) to 102°F (38.9°C) with the lower temperature limited to maintain margin to the critical point and the upper limit based on the compressor recirculation valve reaching its minimum position and no longer maintaining turbine-compressor speed at higher temperatures. The constant compressor recirculation valve test had the smallest range of 94°F (34.4°C) to 100°F (37.8°C) with both limits based upon keeping the system power output within the defined limits. The constant compressor inlet pressure test had the lowest temperature tested at 88°F (31.1°C) with the temperature being allowed to approach closer to the critical temperature at the lower end given the significant margin to the critical pressure, but on the upper end was limited to a maximum temperature of 100°F (37.8°C) by the ability of the compressor recirculation valve to maintain the compressor speed setpoint.

Compressor Inlet Pressure and Density

The effect of compressor inlet temperature on inlet pressure and density will be discussed before looking at cycle and component performance as these parameters are important to assessing some of the behaviors seen in the data. The system mass was fixed in the constant compressor speed and constant compressor recirculation valve tests which resulted in a variable compressor inlet pressure as the compressor inlet temperature was changed. As seen in Fig. 2, the compressor inlet pressure varied linearly with compressor inlet temperature with lower pressure at lower temperature. This is due to the increased compressor inlet density resulting from the lower temperature causing more of the CO_2 in the system to be located within the precooler and thereby lowering the pressure throughout the system. The effect seen in the IST is potentially amplified compared to other systems due to approximately 40% of the system volume being located in the two shell-and-tube heat exchangers that make up the IST precooler; a system with a compact heat exchanger for a precooler is expected to be less impacted by changes in compressor inlet temperature.

The change in compressor inlet pressure with compressor inlet temperature results in a small operating range for compressor inlet density for the tests with a fixed system mass. The range in compressor inlet densities for the constant compressor inlet pressure test is four times that of the range for the tests with a fixed system mass as can be seen in Fig. 3. This difference in density range will come into play during the discussion of cycle performance differences between the two tests with a fixed turbine compressor speed.

Cycle Performance

As expected, the power output from the system increased as the compressor inlet temperature decreased as is shown in Fig. 4. The higher density at the compressor inlet results in a higher system flow rate thereby increasing the power produced by the turbine-generator. The output power of the constant compressor recirculation valve test was affected most by changes in compressor inlet temperature due to the change in compressor speed which resulted in both a change in flow as well as compressor pressure rise. The constant compressor inlet pressure test was slightly more affected by compressor inlet temperature than the constant compressor speed test due to the larger change in system flow resulting from the larger change in compressor inlet density.

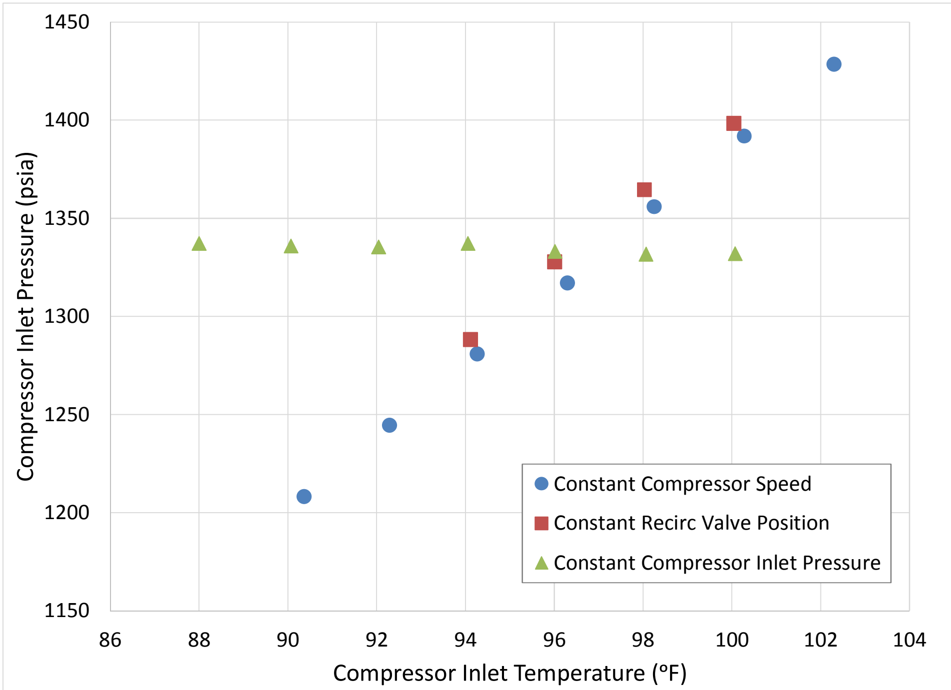

Fig. 2 Compressor Inlet Pressure Sensitivity to Compressor Inlet Temperature

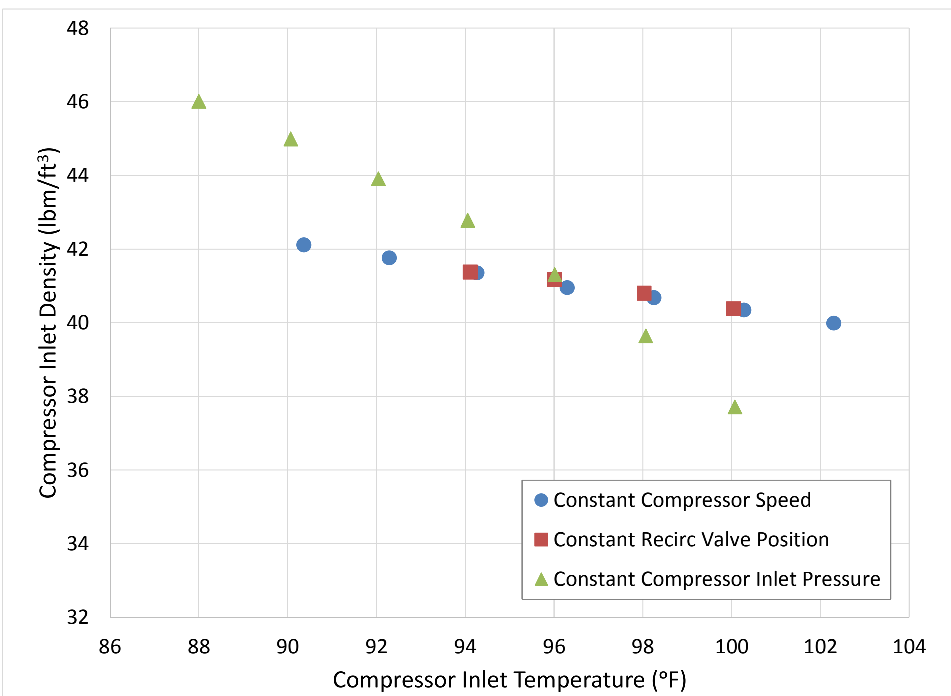

Fig. 3 Compressor Inlet Density Sensitivity to Compressor Inlet Temperature

Cycle efficiency follows a similar trend to the system power output as shown in Fig. 5. While the trend is the same, the slope of the curves is flatter due to the change in system flow resulting in an increase in the heat input to the cycle. Since a portion of the turbine flows is being used to power the compressor, only a portion of this additional heat input is being used to produce output power in the turbine-generator.

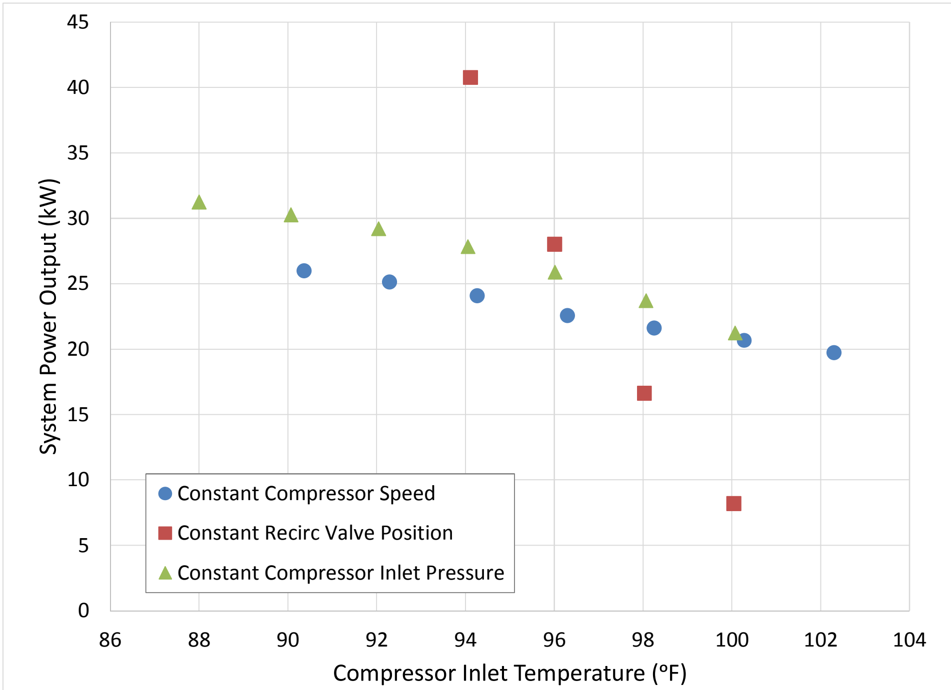

Fig. 4 System Power Output Sensitivity to Compressor Inlet Temperature

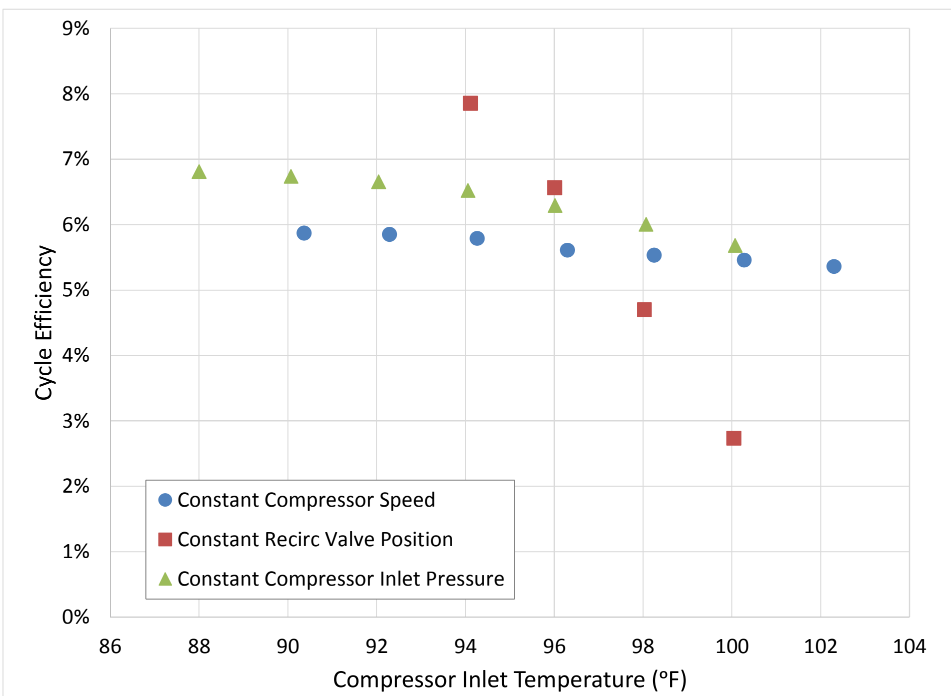

Fig. 5 Cycle Efficiency Sensitivity to Compressor Inlet Temperature

Compressor Performance

The compressor speed and pressure ratio for the different tests are shown in Figs. 6 and 7, respectively. For the constant compressor speed and constant compressor inlet pressure tests, the compressor speed varied slightly around its setpoint of 48,650 rpm due to slow signal response from the control system and the valve controller deadband. For the constant compressor speed test, the fixed compressor speed resulted in a relatively constant pressure rise across the compressor but a varying pressure ratio based on the change in compressor inlet pressure. For the constant compressor inlet pressure test, the pressure ratio follow the same trend as in the constant compressor speed test which meant a higher pressure rise in the compressor with decreasing compressor inlet pressure. In the constant compressor recirculation valve position test, the compressor speed and pressure ratio varied since the loop resistance was held the same as inlet temperature varied. The compressor speed ranged from 43,600 rpm to 53,600 rpm which resulted in a range of compressor pressure ratios between 1.30 and 1.50.

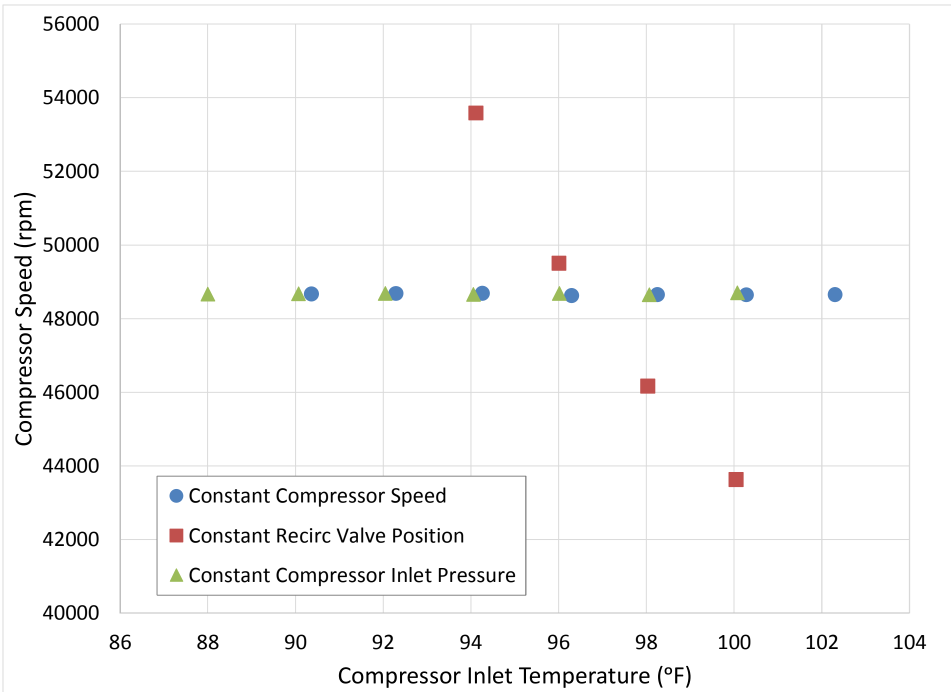

Fig. 6 Compressor Speed Sensitivity to Compressor Inlet Temperature

Previous analysis of the IST compressor performance has indicated that the use of the design compressor maps developed using standard correction techniques developed with ideal gas laws under-predicts the compressor enthalpy rise by approximately 25% [14]. This analysis was performed using test data where the compressor inlet temperature and pressure were maintained very close to the design conditions but over a wide range of compressor speeds. The compressor performance relative to the map predictions was evaluated for all of the compressor inlet temperature points with the results for the constant compressor speed and constant compressor recirculation valve tests shown in Fig. 8. The corrected actual values were consistently 29% ± 2% (N=21) above the predicted enthalpy rise from the compressor map. The corrected actual pressure ratio across the compressor was 6.9% ± 0.6% (N=21) above the predicted pressure ratio using the compressor map. These differences indicate that the behavior seen in the previous analysis is independent of compressor inlet conditions. The extreme variability of properties of sCO_2 in comparison to an ideal gas demonstrates that the standard correct techniques used in the compressor design did not fully account for the behavior exhibited by a supercritical fluid compared to an ideal gas, including a non-constant heat capacity (C_p). Additionally, sCO_2 has a high density and low compressibility, whereas ideal gases exhibit low density and high compressibility. The higher density increased the efficiency of the compressor, which was not properly taken into account for this model and potentially others. In order to make accurate compressor maps for sCO_2 turbomachinery, a better approach is needed. However, the use of an individual chart for each specific compressor inlet condition may not be needed since the standard correction techniques have a consistent offset over a range of compressor inlet conditions and speeds.

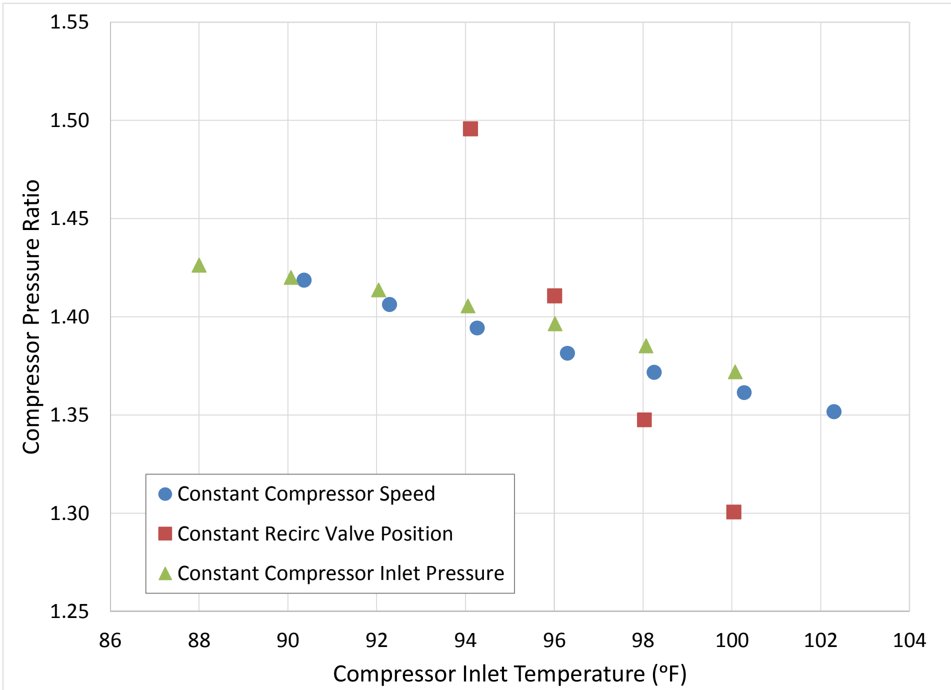

Fig. 7 Compressor Pressure Ratio Sensitivity to Compressor Inlet Temperature

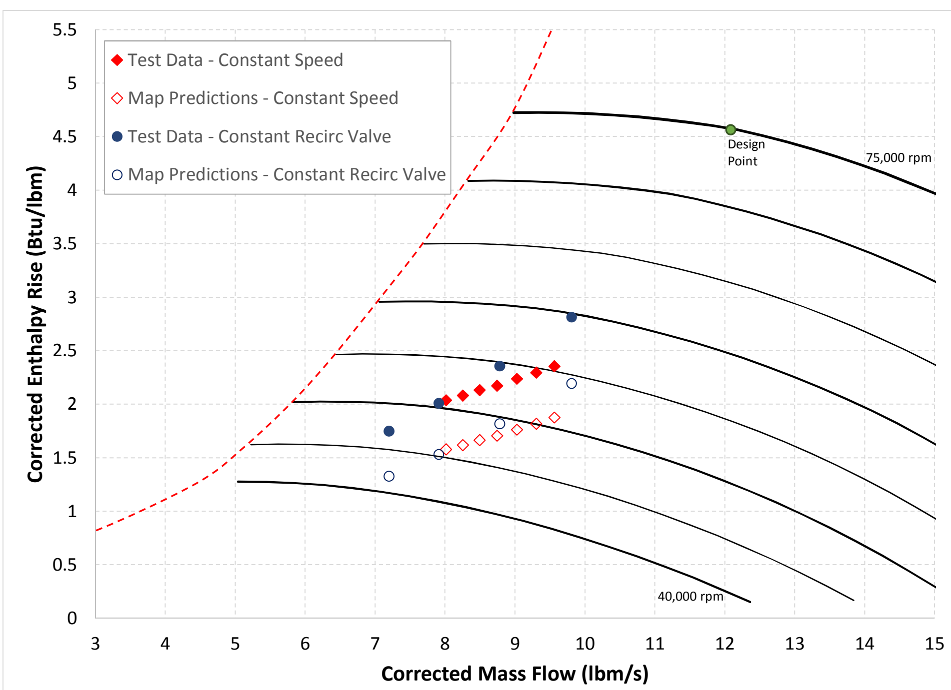

Fig. 8 Comparison of Compressor Performance to Map Predictions

CONCLUSIONS

Naval Nuclear Laboratory has demonstrated a successful ability to operate and control a closed loop sCO_2 Brayton power cycle over a wide range of conditions. By stabilizing several different variable components, the IST evaluated a range of compressor inlet temperatures that reached the performance edges of the system without loss of control. By fixing the recirculation valve position, the IST reached the widest range of output powers in the smallest compressor inlet temperature range. The power fluctuation experienced by the other two tests in response to the change in compressor inlet temperature was much less extreme and failed other equipment bounds rather than the minimum and maximum power limits.

Within the system, the compressor behaved as expected with increased inlet density and pressure ratio increasing the efficiency of the system. When compressor speed varied, the largest response to compressor inlet temperature change was noticed. The compressor map that was used to model the IST compressor was found to under-predict for the enthalpy rise and pressure ratio across the compressor at the corrected speed and flow rate, which is consistent with past analysis performed at near design compressor inlet conditions. It was determined that sCO_2 may not be able to be mapped with the standard correction techniques utilized in the construction of the IST compressor map.

The analysis and conclusions made within this paper may be dependent upon the small turbomachinery size and the specific components used in the IST. The use of large shell-and-tube heat exchangers for the precooler may exaggerate the effect of compressor inlet temperature on compressor pressure ratio and in the end indicate that the differences seen between the constant compressor speed test and constant compressor inlet pressure test would not be as large in a system with a smaller precooler. Additionally, the low turbine inlet temperature and low turbomachinery efficiencies of the IST relative to envisioned commercial operating temperatures and larger scale systems may result in different behavior than would be seen in other potential applications.

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