INITIAL TEST RESULTS OF A MEGAWATT-CLASS SUPERCRITICAL CO_2 HEAT ENGINE

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ABSTRACT

The first megawatt-class, commercial-scale supercritical CO_2 heat engine, the EPS100, is undergoing validation testing. The individual subcomponents have been modeled and tested, and their performance relative to pre-test predictions has been evaluated. A model of the full system has also been created, and once the measured flow rates of the turbomachinery are used to adjust the relevant flow coefficients, the model compares well to the measured state points of the system.

INTRODUCTION

Supercritical carbon dioxide (sCO_2) thermodynamic power cycles have been studied in significant detail for numerous applications, including nuclear power conversion (Dostal, et al., 2004), concentrated solar power (Seidel, 2010; Turchi, et al., 2013), waste and exhaust heat recovery (Persichilli, et al., 2012; Walnum, et al., 2013), and oxyfuel combustion cycles for primary power (Allam, et al., 2014). Many of these studies have primarily focused on theoretical cycle development, although significant advances have been made in laboratory-scale experimental systems (Wright, et al., 2010) (Kimball, 2011). Note that the references given are exemplary and are not intended to be a comprehensive review of previous work.

Echogen Power Systems, LLC has been developing commercial-scale sCO_2 cycles and systems specifically for moderate temperature thermal power conversion, including industrial waste heat recovery (WHR) and exhaust heat recovery (EHR) applications. These applications are characterized by heat source temperatures in the 300 to 600°C range, and heat that is in the form of sensible enthalpy (that is, $Q = w_{source}(h_{source} - h_{residual})$, where w_{source} is the mass flow rate of the thermal medium, h_{source} is the enthalpy of the heat source at the inlet of the main heat exchanger, and $h_{residual}$ is the unrecovered enthalpy from the source). The unrecovered enthalpy is that which cannot be recovered from the source, due to cycle limitations, technical limitations (e.g., a minimum allowable stack temperature to avoid condensation in the exhaust), or economic factors. The residual enthalpy is permanently lost to the energy conversion process, generally in the form of thermal energy in the exhaust.

This situation stands in contrast to nuclear or concentrated solar power (CSP) applications, in which the residual enthalpy is not lost, but is recycled back to the heat source. Therefore, heat recovery cycles are designed to maximize power output by simultaneously achieving high thermodynamic efficiency (W_{out}/Q), and minimizing the unrecovered enthalpy to the greatest allowable extent. As a result, for the same heat source temperature, the direct conversion efficiency of heat recovery cycles is lower than that of the typically considered cycles (recompression, partial cooling, etc.). However, heat recovery cycles will deliver a significantly higher output power from a sensible enthalpy heat source because of their increased utilization of available enthalpy. As a result, the cycle architecture of heat recovery cycles differs from that of the more commonly studied “recompression” or “partial cooling” cycles.

As part of their commercial development process, Echogen designed the EPS100, a 7 to 8 MW class heat recovery engine. The target application characteristics were gas-phase combustion products with an exhaust temperature in the 500 to 550°C range, and a flow rate of approximately 65 to 70 kg/sec. The minimum allowable exhaust temperature is taken to be 85°C, which is characteristic of natural gas-fired heat sources. The EPS100 is currently undergoing factory validation testing at the Dresser-Rand facility in Olean, NY. The present work includes a description of the EPS100, some of its key operating characteristics, operational experience, and test data.

EPS100 BACKGROUND

Cycle configuration

The production EPS100 cycle configuration is a proprietary architecture that utilizes multiple stages of recuperation and extraction from the primary heat source. For initial testing purposes, the EPS100 is currently configured in a modified simple recuperated cycle, as shown in the process flow diagram in Figure 1. The CO_2 flow is split downstream of the main steam-to-CO_2 heat exchanger into two main streams. Approximately two thirds of the flow is directed toward the power turbine, while the remainder of the flow is directed toward the drive turbine that provides the shaft power for the main CO_2 pump. The system is shown in its test configuration in Figure 2.

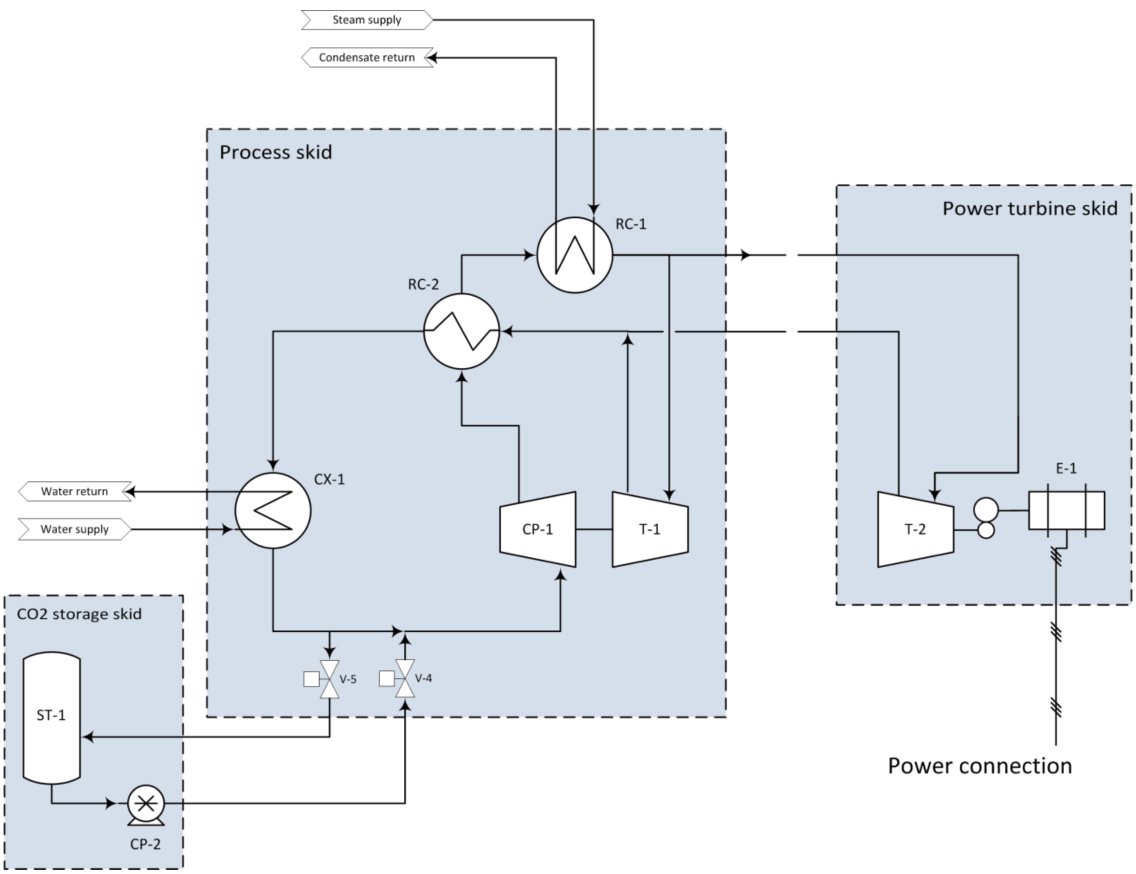

Figure 1: EPS100 test installation process flow diagram (PFD).

Figure 2: EPS100 test installation.

A side note on nomenclature

To call the pressure-increasing device a “pump” is a term of convenience. The EPS100 is designed to operate both in condensing and non-condensing modes, as the supply and temperature of the coolant permits. Under high cooling temperature and/or low cooling flow conditions, the conditions at the pump are in fact supercritical. However, at lower coolant temperature, the “condenser” (again, a term of convenience, as the same device can operate as either a true condenser or as a cooler, depending on the cooling medium temperature and flow) can reduce the temperature of the CO_2 to a low enough temperature to permit the inlet pressure of the pump to reduce to a subcritical temperature. Therefore, the choice to call the respective devices “pumps” or “compressors,” “condensers” or “coolers” is somewhat arbitrary. However, the ability of the control system to operate the system reliably and stably under a wide range of coolant conditions is critical to the success of the heat engine. For consistency, the terms “pump” and “condenser” are used throughout the present work.

Rotating equipment

The EPS100 uses two separate turbines, one (the “drive turbine”) connected directly to the fluid pump, while the other (“power turbine”) is coupled to a four-pole synchronous generator through an epicyclic gearbox for power generation. Of necessity, the power turbine operates at a constant speed (approximately 30,000 RPM), while the turbopump speed can be varied independently over a wide range (< 24,000 to 36,000 RPM) to maintain the required flow rate for the fluid loop in the optimal range for the given heat source and coolant conditions.

The turbopump consists of a hermetically sealed unit, with a single-stage radial turbine directly coupled to a single-stage centrifugal pump. The journal and thrust bearings are a proprietary design that avoids the use of secondary fluids for lubrication. The nominal shaft power rating of the turbopump at full power conditions is 2.7 MW.

Because of the sealed nature of the turbopump, the system requires an additional means to initiate circulation of CO_2 in the process loop. A small motor-driven multistage centrifugal pump is used for this purpose.

The power turbine is also a single-stage radial design. However, because the gearbox and generator are external to the main CO_2 process loop, a rotating shaft seal is required to maintain isolation of the process loop. To maintain as low of a leakage rate as possible, a commercial dry gas seal (DGS) is located adjacent to the power turbine impeller. This isolation also permits the use of commercial tilting-pad style journal and thrust bearings for the power turbine shaft.

The gearbox is a commercial compound epicyclic design with a gear ratio of approximately 16.7. This reduces the 30,000 RPM shaft speed to 1800 RPM, which is then used to drive a conventional synchronous generator at 13.8 kV.

Heat exchangers

The recuperators and condenser are all of the printed circuit heat exchanger (PCHE) design, in which multiple chemically etched plates are diffusion bonded into a single core, to which manifolds and nozzles are welded (Le Pierres, et al., 2011). The production exhaust heat exchanger is of finned tube design. However, for the current test configuration, one of the production recuperators has been re-tasked as a steam-to-CO_2 heat exchanger, and is presently used as the primary heat source for operating the system.

Instrumentation and controls

The operation and control of the EPS100 is performed by a proprietary control system and software. The control system hardware is fabricated from industrial programmable logic controller (PLC) modules and centralized processor, and the instrumentation is predominantly industrial-grade pressure transducers, resistance temperature detectors (RTDs), and other measurement elements as required. Uncertainty analysis of calculated parameters uses the manufacturer's rated uncertainty for combined sensor and transmitter – typically 0.1°C for RTDs, 0.5° for thermocouples, and 0.14% of span for pressure transducers. The CO_2 flow rate is measured in several locations using orifice plates with the flow rate calculated using ASME PTC 19.5 procedures. Shaft displacements and speeds are measured using industrial eddy current probes.

The primary control mechanisms for the system are valves located at key locations in the process loop. The most important of these are the pump bypass valve and turbine throttle valves. In addition, the pump inlet pressure is actively managed through the use of inventory control, which utilizes a separate CO_2 storage tank for supply and withdrawal of fluid from the main process loop as necessary to maintain pump inlet pressure at the desired value.

TEST CONFIGURATION

The EPS100 is currently undergoing testing at the Dresser-Rand compressor test facility in Olean, NY, where the plant's high-pressure superheated steam supply is used as the heat source. As a result, the maximum temperature of the CO_2 is limited to a value slightly higher than the steam saturation temperature, approximately 260 to 275°C. Because both turbines are supplied from a common main heat exchanger, the turbine inlet enthalpies are essentially the same. Although this configuration permits operation of the turbopump near its production operating condition, the design point inlet temperature of the power turbine is in the 400 to 485°C range, thus limiting the full power output of the turbine, and requiring its operation well outside its nominal design range.

The generator electrical output is dissipated in a resistive load bank, rather than being connected to an electrical grid. As a result, generator speed control must be provided by the EPS100 system, rather than operating at the speed as stabilized by the connected electrical grid, essentially requiring operation in "island mode."

Initial testing was conducted without the power turbine. In this configuration, the flow that would normally be directed toward the power turbine was expanded through the pump bypass valve directly to the condenser inlet. This permitted testing of the turbopump in relative isolation, including operation up to its maximum speed, power and pressure rise simultaneously. In subsequent testing, both the turbopump and power turbine were tested simultaneously. Other than the obvious distinction of the turbomachinery configuration, the two configurations also differed in the flow distribution through the low temperature recuperator (RHX2), allowing validation of PCHE performance and model capability over a very wide range of operating conditions.

The EPS100 uses standard "food-grade" CO_2 as the working fluid, with a minimum CO_2 content of 99.5%. A typical specification sheet and analysis of the as-delivered CO_2 is shown in Table 1. For analytical purposes, all properties of CO_2 are assumed to be those of the pure fluid, using REFPROP 9.1 (Lemmon, et al., 2013) as the thermodynamic property evaluation code.

Table 1. Specification limits and typical analysis results for CO_2 supply

Food Grade Carbon Dioxide			
Requirement	Specification Limit	Result	
Purity	99.5% Minimum	99.99	%
Total Sulfur Content	0.5ppm Maximum	0.02	ppm
Carbonyl Sulfide	0.5ppm Maximum	n/d	ppm
Hydrogen Sulfide	0.5ppm Maximum	n/d	ppm
Sulfur Dioxide	5ppm Maximum	n/d	ppm
Total Hydrocarbon Content as C1	50ppm Maximum	0.01	ppm
Carbon Monoxide	10ppm Maximum	n/d	ppm
Nitric Oxides ($NO+NO_2$)	5ppm Maximum	n/d	ppm
Oxygen	50ppm Maximum	n/d	ppm
Acetaldehyde	0.5ppm Maximum	0.15	ppm
Non-Volatile Residue	10ppm Maximum	n/d	ppm
Dewpoint	-68°F Maximum	-149.9	°F

DATA AND ANALYSIS REVIEW

At the time of the present study, testing of the EPS100 is still in process. Most of the test objectives for steady state and transient performance have been accomplished, including operation of the turbopump at its maximum shaft power, speed and discharge pressure simultaneously, and operation of the power turbine at full speed, and electrical output power levels of up to 2.35MWe to date. System endurance testing is currently underway. The data and analysis review consists of comparison of the subcomponent performance to the pre-test predictive performance models. In addition, the overall system performance is compared to the overall cycle model predictions.

Turbomachinery

Full heat and work balances are used to evaluate turbomachinery performance. As an example, the first-law control volume for the power turbine is shown in Figure 3. Flow enters the turbine from two main sources – the main inlet flow, and a buffer seal gas flow that maintains isolation of the main process gas from the dry gas seal. For purposes of calculating the isentropic efficiency of the turbine, the main inlet flow is used as the normalizing factor. The overall heat and work balance can then be written:

$$W_{shaft} + Q_{loss} = w_{inlet}(h_{inlet} - h_{outlet}) + w_{seal}(h_{seal} - h_{outlet})$$

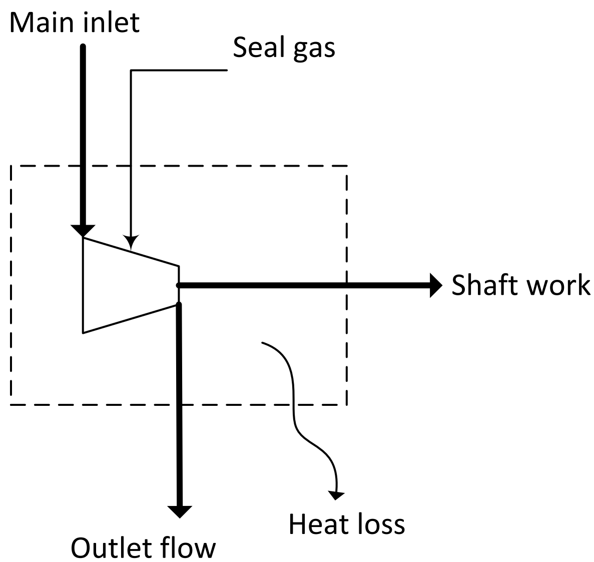

Figure 3: First law control volume for power turbine.

Heat losses for this case are to the environment and to an oil cooling jacket used for thermal isolation for the bearings and seals. The oil cooling loss is measured by flow rate and temperature rise of the oil, while environmental losses are estimated by an empirical natural convection model. Both losses are of the order of 30 kW, representing a small fraction of the aerodynamic power. The isentropic efficiency can then be calculated by dividing W_{shaft} by the isentropic work represented by the main turbine inlet flow:

$$W_s = w_{inlet}\left(h_{inlet} - h_{outlet,s}\right)$$

The primary correlating variable for radial turbine efficiency is U_t/C_o, where U_t is the tip velocity of the turbine blades, and C_o is the "spouting velocity," calculated from the isentropic enthalpy drop through the turbine:

$$\frac{U_t}{C_0} = \frac{N \cdot \pi D_t}{\sqrt{2\left(h_{inlet} - h_{outlet,s}\right)}}$$

The measured efficiency of both turbines is plotted relative to a reference curve derived (and extrapolated) from NASA TP-1730 (McLallin & Haas, 1980) and shown in Figure 4. As can be seen, the agreement is excellent. The power turbine points are representative of significantly off-design conditions due to the limited turbine inlet temperature achievable in the current test installation. At full power conditions, the power turbine will be expected to reach similar efficiency levels to, if not somewhat higher than the turbopump drive turbine, which is operating near its full power operating power.

Pump efficiency is the ratio of isentropic work to actual work, as evaluated at the pump inlet. The internal loss of fluid to the bearing cavity is book-kept separately as a parasitic flow in the overall cycle model. The primary correlating parameter for pump efficiency is the nondimensional flow coefficient, defined as:

$$\varphi = \frac{w_{inlet}}{\rho_{out}(2\pi N)R_t^3}$$

As shown in Figure 5, the measured efficiency of the pump is somewhat higher than the predicted value. Pump inlet flow rate is measured by a summation of the measured turbine and auxiliary flows in order to avoid the flow disturbance created with an obstruction meter at the pump inlet. The overall work balance of the turbopump (where turbine work should equal the pump work plus any heat or work losses) for a typical test case is shown in Figure 6. The parasitic loads are measured by the enthalpy increase of the fluid passing through the bearing cavity, and correspond closely to the difference between turbine work and pump work.

The measurement uncertainty for turbine efficiency ranges from 0.3 to 0.8 points, and for pump efficiency is approximately 0.9 points. The uncertainty in efficiency due to heat losses from the components is less than 0.1 points.

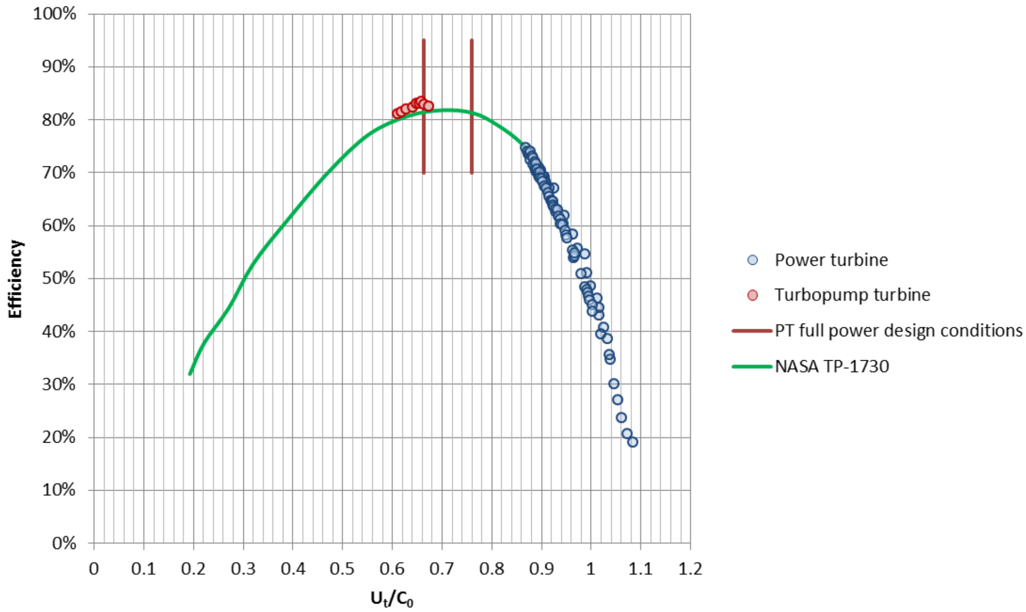

Figure 4: Turbine performance vs NASA TP-1730 curve. Note that TP-1730 curve ends at approximately U_t/C_0=0.9.

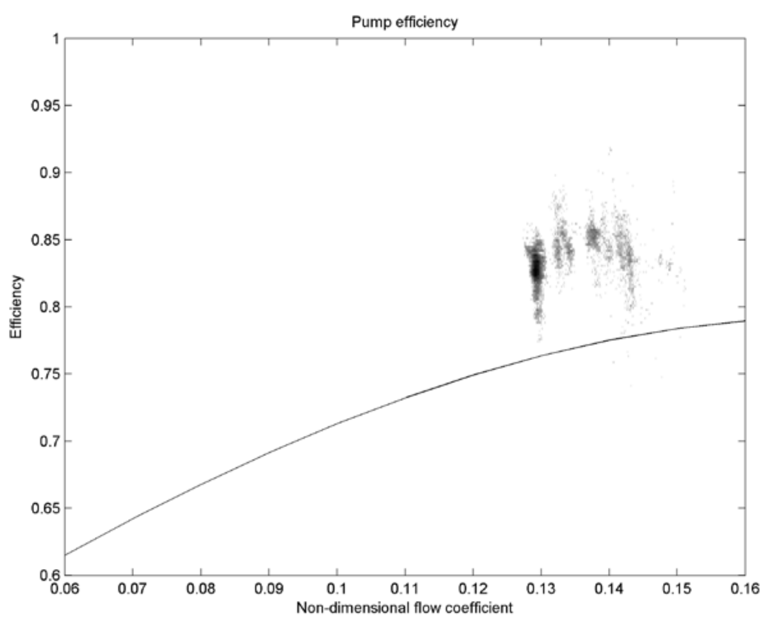

Figure 5: Measured pump efficiency vs flow coefficient. Note that this is a "cloud plot" of the density of approximately 25,000 data points - the gray scale indicates the density of data points that fall within a given x,y coordinate set.

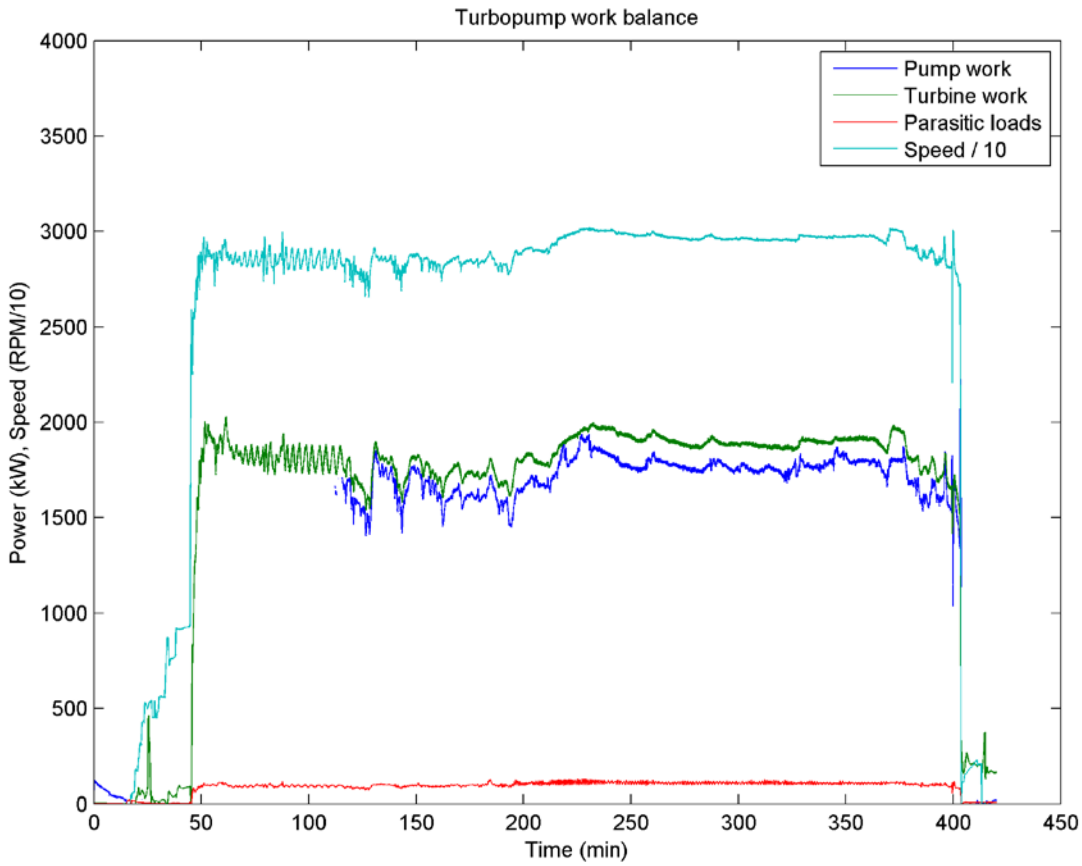

Figure 6: Turbopump work balance.

Heat exchanger performance

Heat exchanger performance could be evaluated on the basis of its comparison to the manufacturer's stated performance at the operating point. In actuality, the operating conditions vary widely, and rarely align directly with the rating point. For scaling purposes, a simplified model of the heat exchanger overall UA and pressure drop has been created both to correlate the performance data, as well as to provide a predictive tool for cycle modeling of off-design conditions.

The numerical form of the model consists of a two sets of parallel, small diameter semi-circular tubes in a strictly counterflowing arrangement, and separated by a thin sheet of 316L stainless steel. Simple Dittus-Boelter convection correlations are used for the fluid heat transfer coefficients in single-phase flow (Incropera & DeWitt, 1996). For two-phase (condensing) conditions a plain-tube correlation is used (Shah, 2009). The overall heat transfer coefficient is then calculated by a simple thermal resistance model. A 20-node discretized model is used to calculate the overall heat transfer performance due to the rapid variation of heat capacity of CO_2 under some operating conditions. Pressure drop is calculated by using a simple Moody friction factor formulation of pressure loss in a tube, as modified for the semi-circular tube geometry.

In the model development process, the number, diameter and length of the tubes are varied to match the design point heat exchanger performance (UA) and pressure drops. After the model has been matched to the design point, its geometry is held fixed while the operating conditions are varied. The model prediction of UA is then compared to the measured values (Figure 7). The agreement between model and data is very good over a broad range of operating conditions.

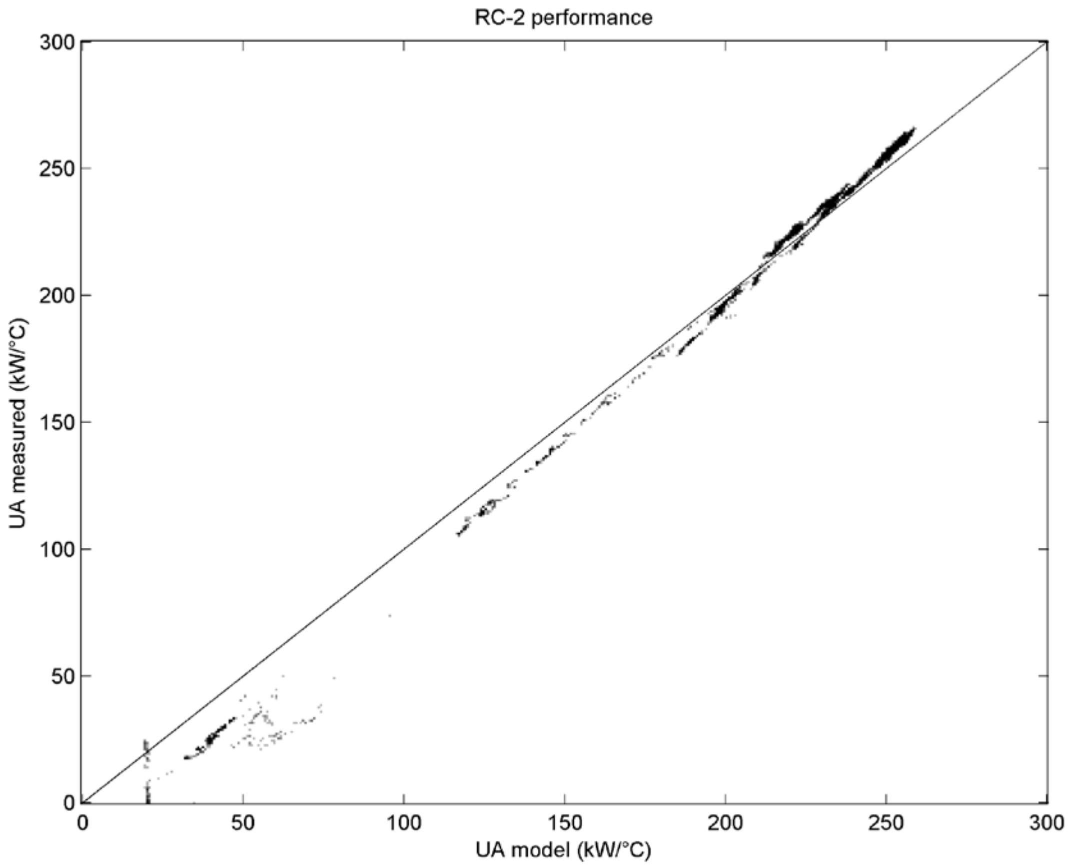

Figure 7: Cloud plot of recuperator (RC-2) performance, actual vs predicted.

Model comparison to data

An overall heat and mass balance model has been created for the simple recuperated test configuration of the EPS100. In addition to the submodels described above, pressure and heat loss models for the interconnecting piping are included using classic pipe flow pressure drop correlations, and assuming natural convection heat transfer from the external surfaces of the pipes.

Several parameters are used as boundary conditions for the model. These same parameters are those used in the control of the system to establish the desired steady-state operating conditions:

- Pump inlet pressure, which is actively controlled during the operation of the EPS100 through inventory management.
- Pump outlet pressure is actively controlled during normal operation through the action of pump bypass and turbine throttle valves.
- Steam-to-CO_2 heat exchanger high-pressure outlet temperature is controlled by modulation of steam flow and pressure.
- Power turbine inlet pressure is set by throttle valve position.
- Power turbine speed is controlled by modulating the resistance of the load banks.

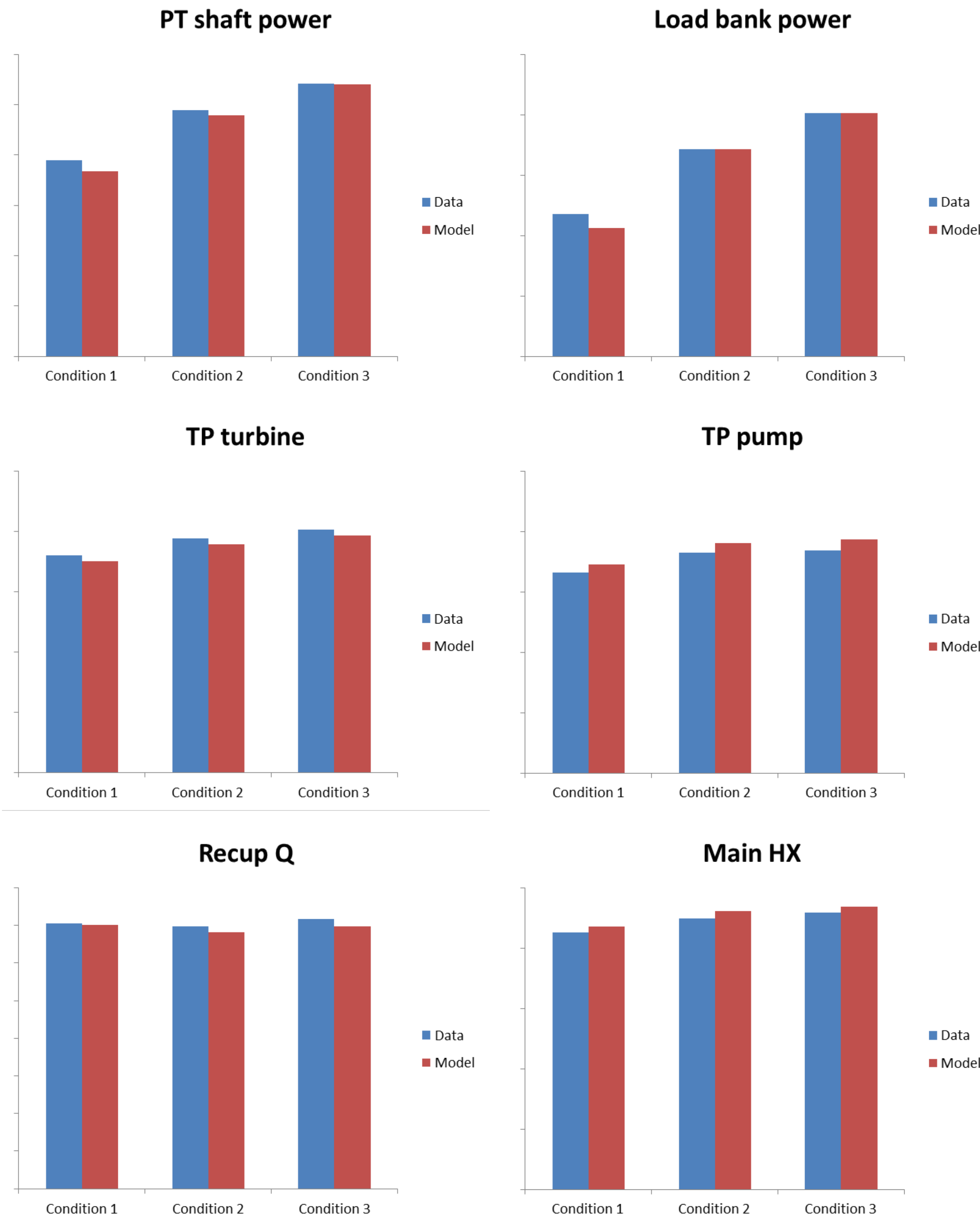

Figure 8: Selected performance parameters over several operating conditions, comparison of model vs data.

Comparisons of several key performance parameters between measured data and model predictions are shown in Figure 8. The pump and turbine flow rates were adjusted from the original performance maps by scalar factors within design uncertainty to match the measured flow rates at Condition 3. These scalar factors were then left in place to predict the performance at the other two conditions. Heat exchanger performance was not adjusted to match data. After these two adjustments were made, good agreement between predicted and measured steady state operating conditions was achieved.

CONCLUSIONS

The initial test results from the EPS100 confirm that a multi-megawatt class sCO_2 heat engine can be designed, constructed, and operated successfully. Off-design performance of key components has been confirmed to meet model predicted values for multiple operating conditions, which provides confidence that the full design-point performance of the system will meet its predicted levels.

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