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Simulation of sCO_2 Recompression Brayton Cycles with Regenerators

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ABSTRACT

The implementation of regenerators in sCO_2 recompression Brayton power cycles may provide cost and performance advantages over large footprint, high cost metallic recuperators. A regenerator is a periodic heat exchanger in which the transfer of heat from the hot to cold fluid is temporally decoupled via a thermal energy storage media, such as a packed bed of spheres. This study analyzes the performance of regenerators employed for sCO_2 Brayton recompression cycles for concentrated solar power applications, targeting 50% or higher cycle thermal efficiency with a turbine inlet temperature of 720°C. We present the results of transient simulations of a 10 MWe sCO_2 Brayton recompression cycle power system using regenerators carried out in gPROMS. The regenerator model consists of one-dimensional transient conservation equations that capture the dynamic effects inherent to regenerator operation and process switching. System simulation incorporates this model with valves, turbomachinery, buffer volumes, and other heat exchangers, allowing for the prediction of temperature, pressure, and flow rate excursions caused by regenerators and propagated throughout the rest of the system. Of particular concern are inlet condition dynamics on turbomachinery and heat exchanger hardware. Simulations indicate that regenerator dynamics may inflict a primary heat exchanger (PHX) inlet temperature fluctuation of $\pm$ 88°C every 25 seconds. Adding an additional packed bed downstream of the regenerator can dampen this fluctuation to about 6°C. Regenerator pressurization and depressurization cycling also causes fluctuations in turbomachinery flow rates by as much as 2-9% every 25 seconds, which in turn causes net shaft power fluctuations of about $\pm$ 450 kW.

INTRODUCTION

The U.S. Department of Energy Sunshot program has recently set a target to reduce the levelized cost of energy from concentrated solar power (CSP) to under 5 cents per kWh by 2030 [1]. Reducing power cycle cost and increasing power cycle efficiency are key to this effort. The recompression Brayton sCO_2 cycle (RCBC) is an attractive option, and the U.S. DOE is currently funding research that contributes to achieving a RCBC with >50% thermal efficiency with a turbine inlet temperature near 720°C [2]. However to reach high cycle efficiencies, the RCBC requires large, highly effective, and expensive recuperators [3].

Regenerators may be a low cost alternative to recuperators. A regenerator is a thermal energy storage device that can be operated to act as a heat exchanger. Hot and cold fluids occupy the same physical space but at different times. That physical space may be a packed bed of stainless steel spheres. This arrangement has high surface area and heat capacity and is simple to build using common materials. The hot and cold fluids pass through the bed during the "charge" and "discharge" process, respectively, between which the bed must be pressurized and depressurized to account for the different process pressures (~ 8 MPa and 25 MPa). This results in four sequential processes, as shown in Figure 1. Valves are employed to control the allocation of fluid, and a minimum of four beds must be employed to ensure continuous flow.

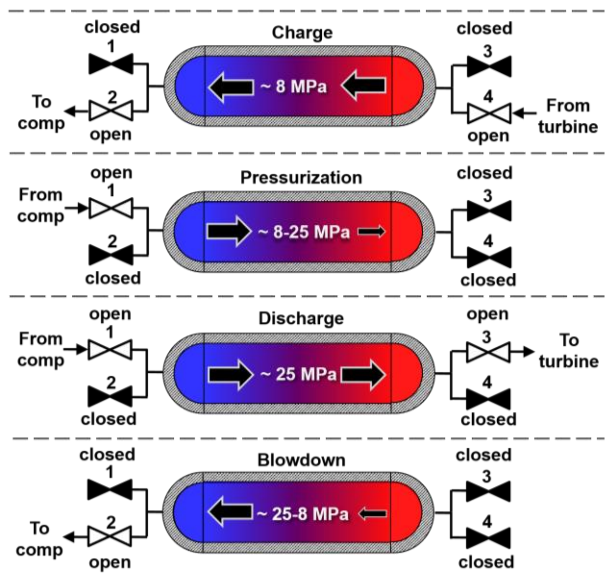

Figure 1. Regenerator operating processes.

Regenerators are inherently transient, and switching time is considerable at large scales. These characteristics must be considered when determining the best operating strategy for regenerators. This work presents the results of a transient one-dimensional regenerator model based on fundamental conservation equations of mass, momentum and energy.

MODELING APPROACH

The model is developed in two software platforms – initially in MATLAB for experimental validation [4], and then in gPROMS [5] for system integration and simulation. The four primary conservation equations are summarized in Table 1:

Table 1. Regenerator model conservation equations

Mass	$\dfrac{\partial(\rho_\mathrm{f}\phi)}{\partial t} = -\nabla \cdot \mathbf{J}_\mathrm{f}$	(1)
Momentum	$-\dfrac{dP}{dx} = f_\mathrm{p}\dfrac{\rho v_\mathrm{f}^2}{D_\mathrm{p}}$	(2)
Fluid Energy	$\dfrac{\partial(\rho_\mathrm{f}\phi u_\mathrm{f})}{\partial t} = -\dfrac{\partial(J_\mathrm{f}h_\mathrm{f})}{\partial x} - \overline{h}_\mathrm{fs}\alpha\left(T_\mathrm{f} - T_\mathrm{s}\right)$	(3)
Solid Energy	$\dfrac{\partial(\rho_\mathrm{s}\psi u_\mathrm{s})}{\partial t} = \psi\dfrac{\partial}{\partial x}\left(k_\mathrm{s}\dfrac{\partial T_\mathrm{s}}{\partial x}\right) + \overline{h}_x\alpha\left(T_\mathrm{f} - T_\mathrm{s}\right)$	(4)

The regenerator is discretized axially, and local fluid properties are evaluated with the Span-Wagner Equation of State [7]. The model takes inputs of bed length, bed diameter, particle diameter, hot and cold inlet flow rates, inlet temperatures, and outlet pressures, and the time for each process. It is initialized with a linear temperature profile and run iteratively until periodic steady state is achieved.

RESULTS AND DISCUSSION

The model was compared to experimental data collected for a 10 kWth regenerator built and tested at the University of Wisconsin - Madison. The flow rate, temperature, pressure, and switching times tested ranged from 14.1-41.9 g/s, 40-504°C, 6.5 – 15 MPa, and 30 – 170 seconds [8]. Figures 2 and 3 show model predicted versus experimentally measured effectiveness and pressure drop. The model generally agrees with the experiments within 10%.

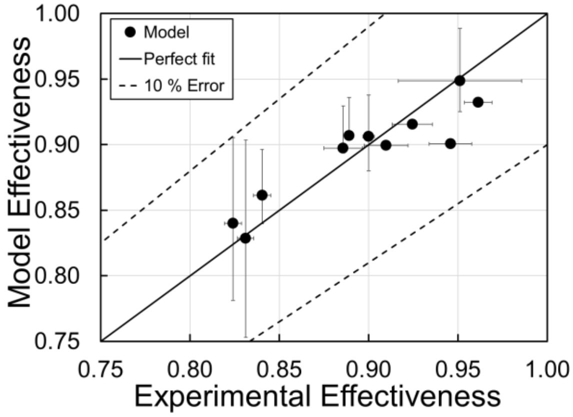

Figure 2. Model predicted vs. experimentally measured effectiveness.

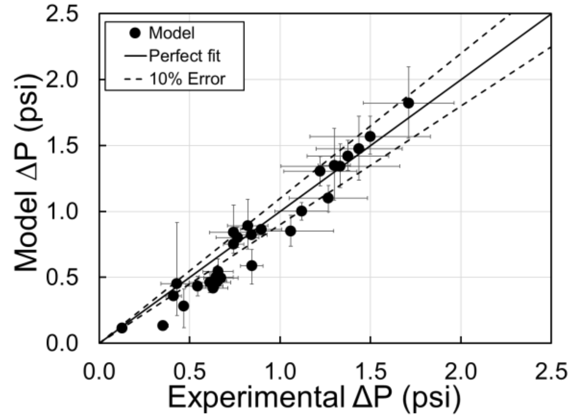

Figure 3. Model predicted vs. experimentally measured pressure drop.

Figure 4 shows model predicted versus experimentally measured temperature profiles within the regenerator for one set of data. The model matches the measured temperature profiles quite well. This plot also shows that the outlet temperatures are not constant. In particular, the cold outlet temperature ($x = 0$ from 45-90s) drops considerably. This fluctuation would propagate directly to the inlet of the primary heat exchanger (PHX), where it may cause thermal fatigue. This fluctuation could be dampened by incorporating buffer volumes at the periphery of the regenerator, and/or implementing another packed bed in between the regenerator and the PHX. Furthermore, the cyclic pressure swings of the regenerators lead to a pulsating flow through the compressors termed “carryover”. The necessary size of the buffer volumes and extra packed bed, along with the severity of the compressor flow pulsations, can only be determined through transient system simulation.

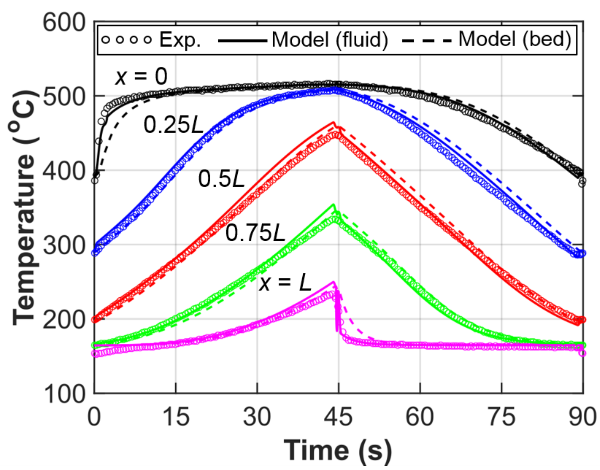

Figure 4. Model predicted vs. experimentally measured regenerator temperature profiles.

Figure 5 shows a schematic of the high-temperature regenerator-valve subsystem within a Brayton recompression cycle as modeled in gPROMS. All turbomachinery and heat exchanger models employ steady state equations. Compressor and turbine off-design is captured using the performance maps previously published [9-11]. The low-temperature recuperator is modeled using a sub-divided counterflow heat exchanger model. The primary heat exchanger is modeled with a counterflow NTU-effectiveness correlation, assuming that the heat transfer fluid is a NaCl-KCl-$ZnCl_2$ molten salt [12]. The pre-cooler is modeled as a multipass, cross flow, finned tube heat exchanger following the approach given in [13]. The buffer volumes are modeled transiently as well-mixed tanks, and the packed bed thermal transient reducer (TTR) is modeled with the same equations as the regenerator.

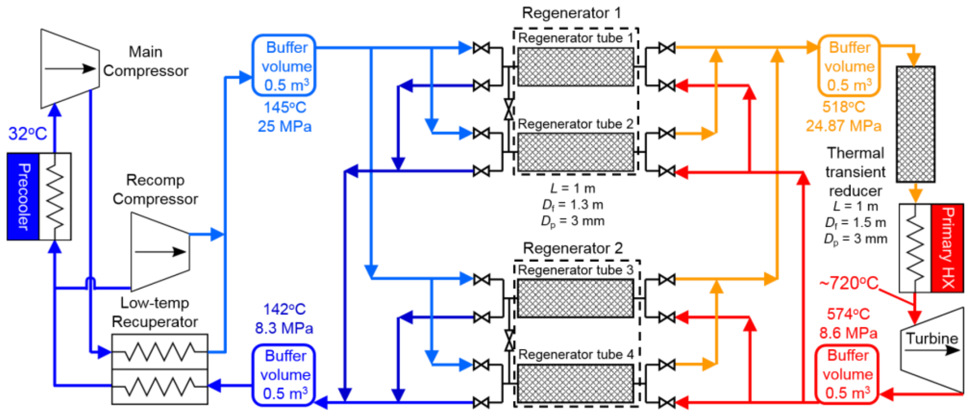

Figure 5. Schematic of the high-temperature regenerator in a Brayton recompression cycle.

In this study, regenerators replace only the high temperature recuperator, as preliminary studies indicate this maximizes efficiency. Bed linking valves are incorporated to allow regenerator beds to equalize in pressure before switching, thereby reducing overall carryover mass. The system model is given initial conditions from the MATLAB model and integrated until periodic steady state is reached.

Figure 6 shows that flow rates fluctuate by $\pm$ 4.4%, 8.6%, and 1.5% for the main compressor, recompression compressor, and turbine, respectively. These fluctuations are caused by regenerator switching, and motivate investigation into the magnitude and rate of flow rate fluctuations that these turbomachines could handle. Figure 7 shows temperature fluctuations at the TTR inlet and outlet and turbine inlet. The TTR reduces fluctuations from $\pm$ 88°C to $\pm$ 5.8°C. The PHX reduces the temperature fluctuation even further to $\pm$ 2.3°C at the turbine inlet.

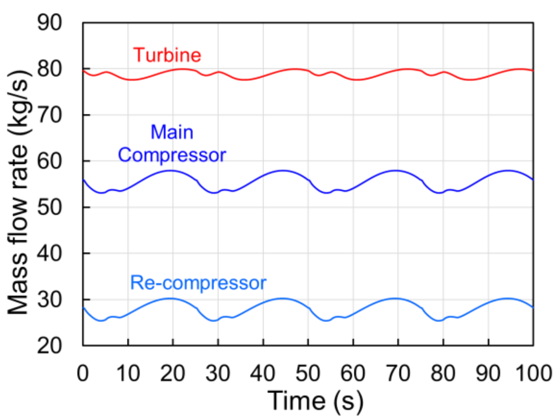

Figure 6. Turbomachinery flow rate throughout one regenerator cycle.

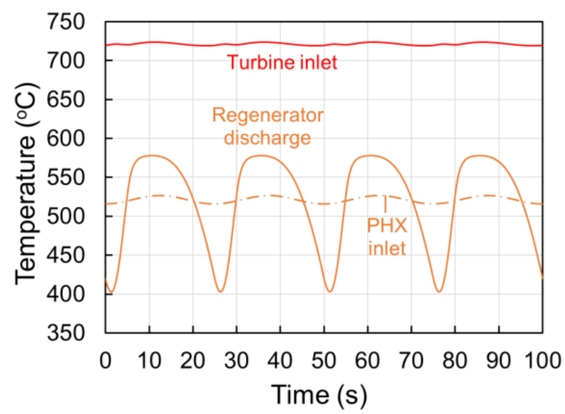

Figure 7. Temperature at the TTR inlet, TTR outlet, and turbine inlet.

Figure 8 shows turbomachinery and net power throughout one regenerator cycle. Fluctuations in recompressor and turbine power lead to a net power fluctuation of ± 4.5%, which motivates investigation of rotor dynamics and control. These fluctuations are impacted by the performance characteristics of the turbomachinery, and performance maps employed here may not be representative of 10 MWe scale equipment. Furthermore, it is expected that at larger power plant capacities (e.g., > 100 MWe), an increase in the number of regenerator beds and staggered operation should substantially reduce the model-predicted power fluctuations illustrated here.

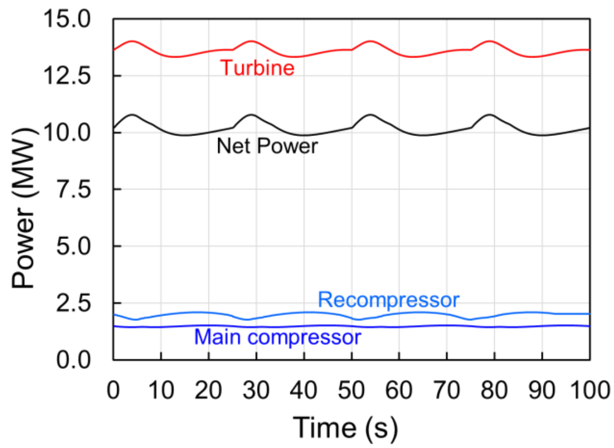

Figure 8. Turbomachinery and net power throughout one regenerator cycle.

CONCLUSIONS AND FUTURE WORK

This study has employed detailed transient simulation to show that regenerators can replace recuperators in sCO_2 Brayton recompression cycles to render high efficiency. Regenerators cause fluctuations in turbomachinery flow rate and power and in the PHX inlet temperature. These fluctuations can be reduced by introducing buffer volumes to the system, and the addition of a packed bed between the regenerator and PHX can dampen thermal transients to an acceptable level. Future work will involve assessment of allowable fluctuations on hardware, particularly flow rate and power on turbomachinery at both 10- and 100-MW scales.

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