Switched Bed Regenerators for SCO2 Cycles

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Introduction

The supercritical CO_2 (sCO_2) cycle has been shown in recent years to provide significant advantages compared to traditional steam Rankine cycles. One of the most important is the ability to reach significantly higher efficiencies. A plot comparing the efficiency as a function of the turbine inlet temperature for several different cycles is shown in Figure 1 (adapted from [1]).

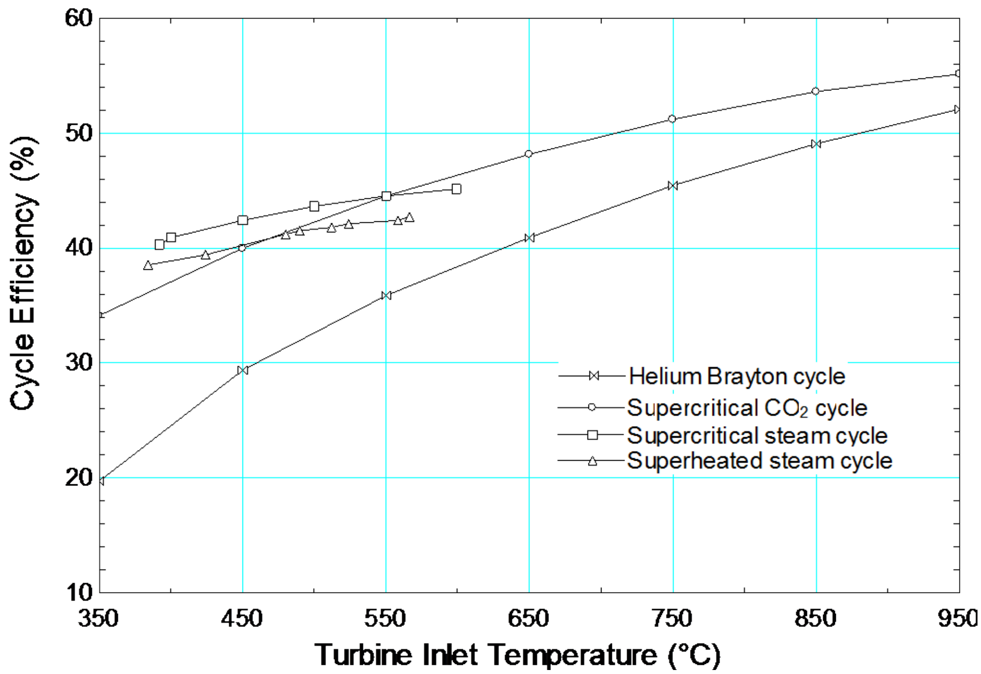

Figure 1 Cycle efficiency as a function of the Turbine Inlet Temperature (TIT) for various working fluids, adapted from [1]

Efficiency gains of the power cycle decrease the cost of generating electricity and can help to make technologies such as solar thermal or nuclear power more cost competitive with fossil energy. Additionally, applying the more efficient sCO_2 cycle to fossil fuel fired power plants would reduce the amount of fuel burned by the plant and thus decrease both fuel costs and emissions. Another advantage of the sCO_2 cycle is its compact size. Designs of turbo-machinery have shown a size reduction of 10x compared to Rankine cycle components of the same power level [2]. This leads directly to a reduction in the plant footprint and the reduced volume of material required for the turbomachinery could lead to a decrease in the capital cost of the equipment [3,4].

Research and development has been done on a variety of the components required for a sCO_2 power cycle including the turbo-machinery and heat exchangers [5–9]. However, Echogen is the only company that currently offers a commercially available sCO_2 cycle. Echogen has created a self contained sCO_2 heat engine used for recovering waste heat that operates at 532°C, producing 8MW of power. This system is a good stepping stone to higher temperatures and power levels but is not representative of a utility scale system. With the goal of proving the benefits of a utility scale sCO_2 power cycle, the Department of Energy has funded a project that seeks to create a 10MWe test facility at the Southwest Research Institute campus [10]. Additionally, Net Power is creating a 50MW direct combustion sCO_2 cycle that can substantially increase efficiency while resulting in no direct emissions [11].

In order to achieve high efficiencies, the sCO_2 cycle requires recuperators with effectiveness greater than 90 percent. The high pressures and temperatures of the cycle limit the materials which can be used to construct the recuperators, and the use of higher strength materials may lead to more expensive components compared to Rankine cycle components. The high amount of recuperation needed and the stringent material requirements result in the recuperative heat exchangers alone accounting for up to 16 percent of the total power block cost [3]. There are two main options currently being considered for the recuperators, the Printed Circuit Heat Exchanger (PCHE), and micro-tube heat exchangers. The PCHE is created by etching a flow pattern onto many thin sheets and using a diffusion bonding furnace to bond them together. This can be expensive because of the intricate etching and bonding process. The micro-tube heat exchangers are similar to conventional shell and tube heat exchangers except with tubes that are less than 0.1 inch in diameter. The small size of the tubes increases the heat transfer coefficient while decreasing the thickness of wall needed. Both of these options are proven commercially available heat exchangers but are costly to build.

As an alternative to these potentially costly recuperator options, a periodic flow regenerator is proposed. A periodic flow regenerator alternatively stores and releases energy from a packed storage bed. In its simplest form a regenerator can be a tube packed with stainless steel spheres. From a manufacturing standpoint this makes construction very simple. There are several industries that currently use regenerators, however there is limited operating experience at the temperature and pressures required for a sCO_2 cycle. In the cryogenic industry, regenerators are used in Stirling cycle, Gifford-McMahon (GM), and various pulse tube cycles. The implementation of regenerators in a sCO_2 cycle is different than in an oscillating Stirling-type cycle where a single regenerator is used in a single fluid volume in order to pre-heat/cool the fluid. In a sCO_2 cycle the flow of the fluid must be periodically switched between two regenerators using valves. While this configuration is more similar to a GM cycle, the GM cycle operates at much lower temperatures and absolute pressures than a sCO_2 cycle [12]. Rotary regenerators are used in both the HVAC industry and in air Brayton cycles to preheat air. However, the rotary regenerator is much different than the switched bed regenerator in that hot material is constantly being introduced into the fluid stream. The rotary regenerator does not work well at high pressure ratios because of the need for a sliding seal. The seals are acceptable for HVAC (0.1 MPa) or air Brayton cycles (0.6 MPa) but the sCO_2 Brayton cycle operates with a pressure differential of approximately 17 MPa which far exceeds the pressure conditions sliding seals can be used [13]. The testing outlined in this paper fills a gap in experimental results for very high temperature and pressure regenerator operation. Previous work has been done to develop a regenerator model and a sCO_2 cycle model featuring regenerators instead of recuperators. This sCO2 cycle model was used to conduct an economic analysis comparing PCHEs and regenerators, the result showed a 21% decrease in the Levelized Cost of Electricity (LCoE) by switching to a regenerator [14]. This paper describes additional work done to demonstrate the regenerator concept for application to sCO_2 power cycles by creating a test facility, performing experiments, and validating the previously developed model.

Regenerator Operation

A regenerator consists of 2 packed beds that alternatively store and return heat to the working fluid. While one of the beds is storing energy, the other is returning it to the fluid. The flow through the beds is controlled with a system of 8 valves. By alternating the flow through the beds a quasi-steady state condition is reached. There are four steps in one complete cycle of regenerator operation as illustrated in Figure 2. This illustration only shows one bed of the two beds, the other bed would be operated 180° out of phase and require 4 additional valves.

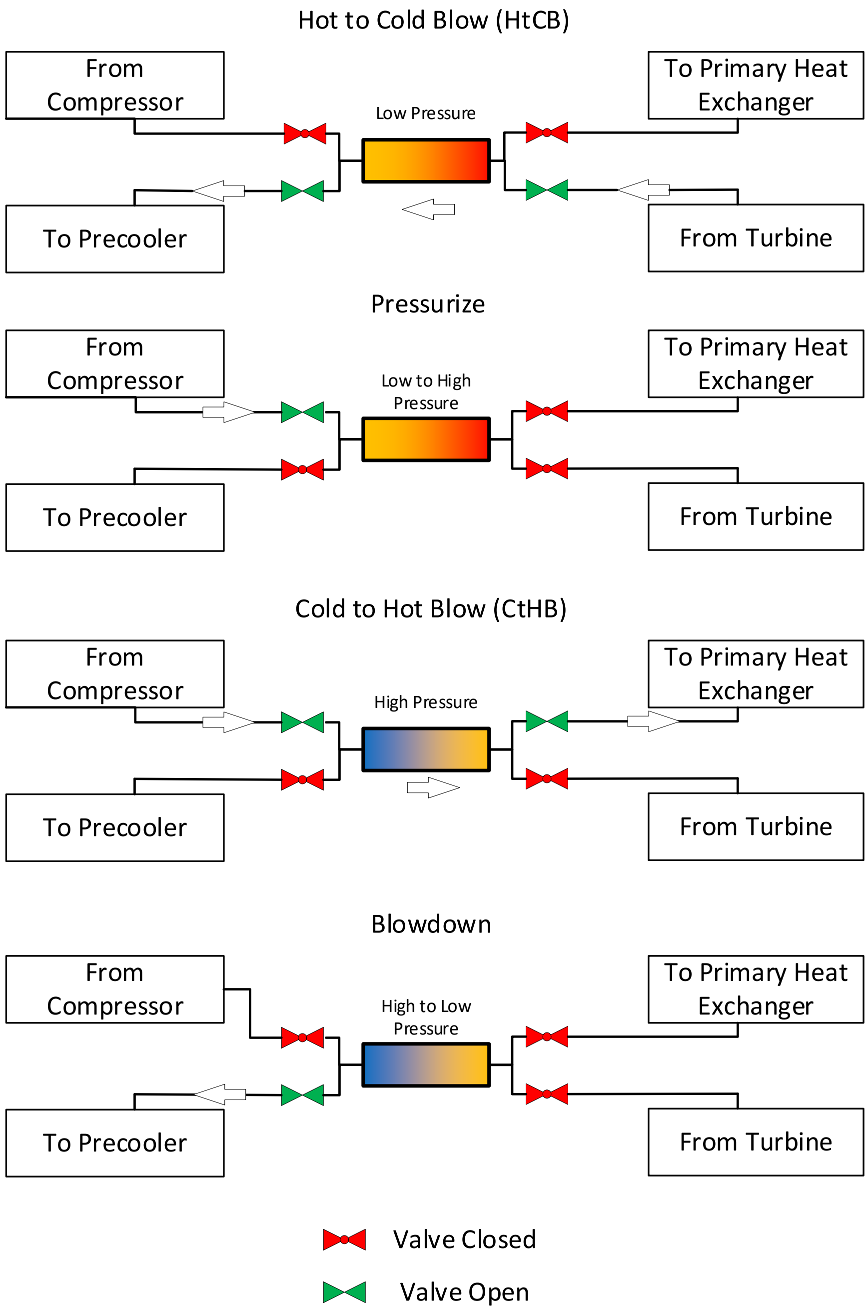

Figure 2 Regenerator operation steps needed for steady state

The Hot to Cold Blow (HtCB) process introduced hot, low pressure fluid coming from the turbine into the bed where energy is stored. The HtCB occurs at low pressure because the hot fluid returning from the turbine which is used to heat the bed is at low pressure, the Cold to Hot Blow (CtHB) occurs at high pressure so the second step in the process involves pressurizing the bed. The CtHB removes the heat stored during the HtCB, preheating the CO_2 before it reaches the primary heat exchanger. Finally, blowdown occur where the excess CO_2 in the bed is removed so the HtCB can occur again at low pressure. This process is repeated for many cycles and the regenerators reach a quasi-steady state.

The pressurization and blowdown steps of the cycle are important to consider as they tend to degrade the performance of the sCO_2 cycle. During the blowdown process, high pressure CO_2 is expanded back to the compressor without ever passing through the turbine where it would otherwise produce power; this mass will be referred to as carryover. Any energy used to compress the carryover fluid is not recovered by the turbine. Depending on the design, carryover can be up to 25% of the total flow rate through the system, causing a significant drain to efficiency. Carryover is highly temperature–dependent; if the temperature in the regenerator is low then the CO_2 has a very high density (and a high pressure driven density difference) resulting in high carryover, however at high temperature the carryover is significantly less.

Regenerator Model

A model is needed to predict the performance of the regenerator. This model should solve quickly and provide reasonable accuracy so that it can be used for optimizing the design of a regenerator in a power block. The model inputs and outputs are listed in Table 1.

Table 1 Regenerator model inputs and outputs

Inputs	Outputs
Bed Length	Number of Transfer Units, NTU
Bed Diameter	Matrix capacity ratio, C_m
Mass Flow	Effectiveness
Inlet Temperatures	Pressure Drop
Inlet Pressures	Carryover

The bed is assumed to be a randomly packed bed of spheres. The heat transfer coefficient and surface area is calculated using [15]. The performance of the bed is highly dependent on the void fraction, or how much of the bed is open to fluid, for a randomly packed bed of uniform spheres this varies between 0.32 and 0.37 [16]. A smaller void fraction cuts down on the amount of carryover, makes better use of the pressure vessel volume, and increases heat transfer. During construction of the regenerators a packing fraction of 0.37 was measured and used for the model.

A model based on dimensionless numbers much like the effectiveness-NTU model for heat exchangers was used [17]. The model takes inputs of Number of Transfer Units (NTU) and matrix capacity ratio (C_m) and uses them to determine the effectiveness of the regenerator. NTU and C_m are defined as follows.

$$NTU = \frac{UA}{\dot{C}_{\min}} \tag{0.1}$$

$$C_m = \frac{m_b c_b}{P_0 \dot{C}_{\min}} \tag{0.2}$$

where UA is the conductance of the regenerator, $\dot{C}_{\min}$ is the minimum capacitance rate of the two streams, m_b is the mass of the energy storage media in the packed beds, c_b is the heat capacity of the bed material, and P_0 is the switching time. The definition for NTU, the dimensionless size of the heat exchanger is defined the same as a conventional recuperator. C_m is the ratio of energy the bed can store compared to the amount of energy passing through the bed, or a non-dimensional measure of how large the regenerator is. The larger the C_m the more energy the bed can store. A lookup table is used to determine the effectiveness as a function of NTU and C_m, contours of constant effectiveness are shown in Figure 3 [17].

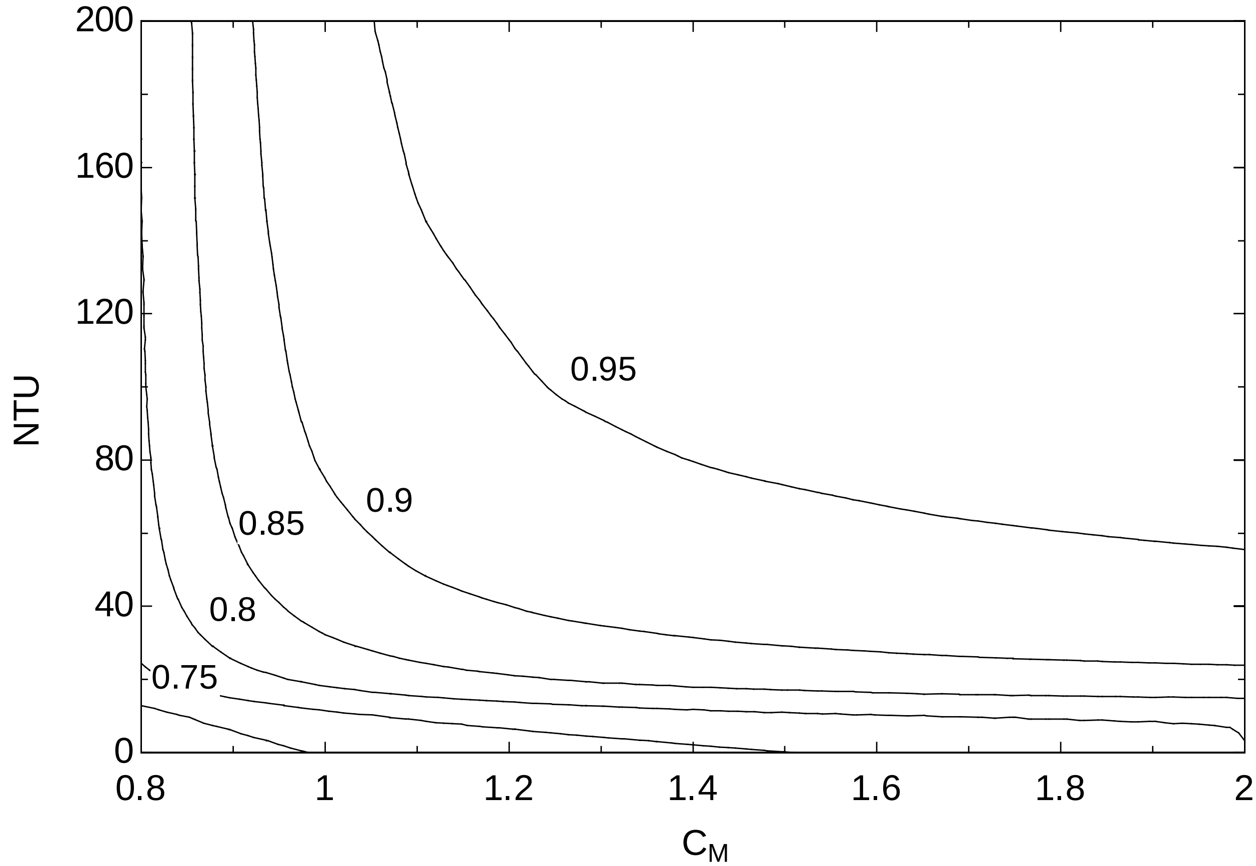

Figure 3 Contours of constant effectiveness as a function of NTU and matrix capacity ratio

The effectiveness can be used to determine the heat transfer rate in the regenerator. Effectiveness is defined in equation (0.3).

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{0.3}$$

Where $\dot{Q}_{max}$ is the maximum possible heat transfer rate calculated by determining what heat transfer would be possible for a perfectly effective heat exchanger operating under the same flow conditions (temperature, pressure, and flow rate).

In addition to calculating effectiveness, the model also calculates pressure drop predicted in the packed bed. The pressure drop is an important design parameter because pressure drop in the bed decreases the power output from the turbine due to the lower pressure differential. The pressure drop in the packed bed is calculated using the Fahiem Schriver correlation [18]. For a more detailed discussion of pressure drop please see [19].

The final performance parameter of importance is the carryover. Carryover is defined as the difference in the amount of CO_2 in the regenerator bed at the end of the CtHB and the start of the HtCB. The easiest way to do this is to integrate the density of the CO_2 along the length of the regenerator. Thus, the accuracy of the model depends on how well the temperature profile in the bed can be determined. The temperature profiles in the bed are assumed to be linear through the length of the bed, with the hot side at the cold stream exit temperature, and the cold side of the bed is at the cold inlet temperature. The equation for carryover is shown in equation (0.4).

$$m_c = \int_0^L \frac{\pi D^2}{4}(1-e_v)\left[\rho(T(x),P_H) - \rho(T(x),P_L)\right]dx \tag{0.4}$$

Where $\rho(T(x),P_H)$ and $\rho(T(x),P_C)$ are the temperatures at location x in the regenerator at high and low pressure respectively. This mass is turned into a leakage mass flow according to equation (0.5).

$$\dot{m}_{co} = \frac{2m_c}{P_0} \tag{0.5}$$

The factor of 2 is added in because for each period the mass in each bed is carried over once. Carryover is one of the most important performance measures since it directly increases the amount of compressor work.

Optimizing regenerator

Optimizing the regenerator depends heavily on the objective function chosen. It would be undesirable to optimize for efficiency since that would lead to very high costs. The regenerators need to provide a cost saving compared to PCHE recuperators for the sCO_2 cycle. The optimization function was chosen to be the lowest possible Levelized Cost of Electricity (LCoE) which takes into account both efficiency and first cost. The effectiveness needs to be kept high to recover as much heat as possible, however at the same time the carryover needs to be kept low to reduce compressor work. The parameters that determine effectiveness are NTU and C_m. The NTU can be made very high by using small diameter spheres to increase the surface area and heat transfer coefficient, at the cost of pressure drop. The 10MW design calls for 3mm spheres. The parameter C_m has a drastic effect of cost of the cycle since it is basically a measure of the dimensionless size of bed. C_m can be made large by decreasing the switching time P_0 however at a certain point the switching time becomes too short and the valves do not have time to respond. Also a small switching time increases the total number of cycles the system must survive over its life. A lower limit on the switching time was set at 45s so the valve actuation time (3-5 seconds) is significantly shorter than the cycle time. To have a high effectiveness (greater than 90%) the C_m must be greater than 1.5, however if the C_m continues to increase then carryover also increases, reducing the efficiency of the cycle. The optimization showed that a C_m of around 1.6 resulted in the lowest LCoE. A low pressure drop through the beds leads to higher efficiency, but also increases the cost since the pressure vessel must become shorter with a larger diameter. It was found that a maximum pressure drop of 225 kPa in the bed resulted in the lowest LCoE. For a more detailed description of the optimization of the regenerators please see [14].

Test Facility

The test facility is designed to test a set of regenerators under the same conditions as those in a 10MW power plant operating with a turbine inlet temperature of 720°C. The regenerators are designed to have a heat transfer rate of approximately 10kW depending on the operating conditions. A diagram of the cycle is shown in Figure 4.

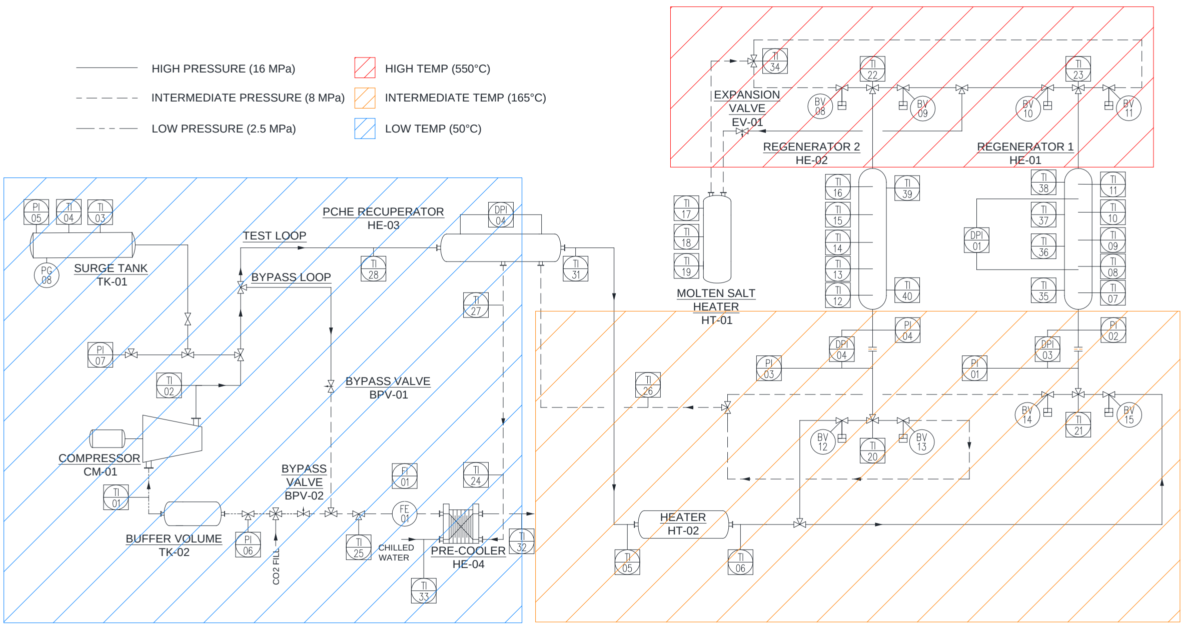

Figure 4 PID of experimental test facility showing different pressures and temperature zones.

Flow Components

The system is powered by a 40HP Hydropac compressor (CM-01) that is capable of providing 1.6 kg/s of flow with a maximum discharge pressure of 16 MPa. The inlet to the compressor is limited to 8 MPa, but the compressor is operated with a lower inlet pressure in order to allow for easier filling from liquid CO_2 bottles. The compressor is a positive displacement 2 piston design where one cycle takes 6s (two strokes). This creates some oscillations in the flow that need to be damped out with buffer volumes. A 12" sch 160 8' long surge tank is attached to the exit of the compressor to damp out some of the flow oscillations. A second smaller tank is attached to the suction side of the compressor to keep the pressure constant entering the regenerator. Together these two surge tanks reduce the flow fluctuations to ±5% of the average value. The large surge tank is also used to set the initial pressure in the system. The surge tank is filled with liquid CO_2 at room temperature and heated until the desired pressure is reached. The alternative is running the compressor while slowly feeding in CO_2 from the bottles which causes wear on the pump and takes a long time.

There are two loops in the system, the test loop and the bypass loop. The pump is capable of providing much more flow than can be used in the test section, so the majority of the flow is passed through the bypass loop. The pressures in the system are controlled by three expansion valves. There are two valves in the bypass loop (BPV-01 and BPV-02), and one in the test loop (EV-01). The two expansion valves in the bypass loop create three separate pressure regions in the test facility; the compressor inlet (low), compressor discharge (high), and the intermediate pressure. The test section operates with one stream at the high pressure and the other at the intermediate pressure. The reason for having three zones is twofold, first the low compressor inlet pressure allows CO_2 to be added using the pressure differential between the bottle and the system, negating the need for a second pump. CO_2 must be continually added to make up for the losses from the seals in the pump and other leakages in the system. Having a low pressure inlet to the compressor also increases the temperature at the exit of the compressor. A high compressor outlet temperature is needed to keep the surge tank (TK-01) at an elevated temperature to keep the pressure in the system high. If the temperature at the exit of the compressor is too low, the flow into and out of the surge tank as a result of oscillations from the pistons causes the temperature to decrease which increases the density of the CO_2 in the tank thus reducing the pressure. The expansion valve in the test loop controls how much flow is going through the test loop. The flow through the regenerator beds are controlled by 8 air actuated needle valves (BV-08 – BV-15) shown in Figure 5.

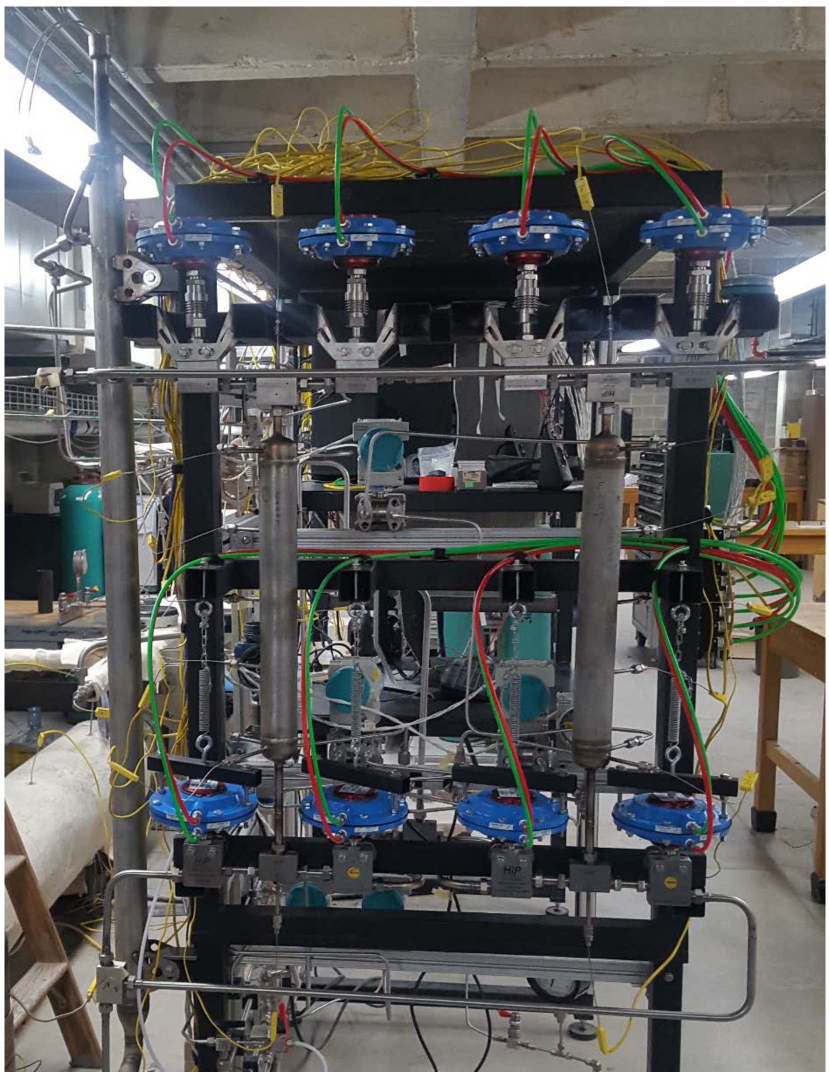

Figure 5 Valves and regenerators mounted on test rig. High temperatures valves are at the top and low temperature valves on the bottom. The pre heater can also be seen to the left of the regenerators.

The valves on the high temperature side of the regenerator have an extended bonnet to keep the packing cool, they are rated to a maximum temperature of 538°C. A slight leak between past the seat of the valve was noticed, but the scale of this leak was not enough to affect the performance of the regenerators. While testing at high temperature and pressure, a leak developed between the body of the valve and the bonnet. Upon inspection it was observed that there was leakage past the 316 seal ring used to seal between the body and bonnet of the valve, it was replaced and there was no further leakage observed. All the fittings on the high temperature side of the test facility are cone and threaded fittings. Since the regenerator is an inherently transient device, the temperature at the exit of the regenerator can swing up to 200°C in 10-15s. This rapid temperature change is shown in Figure 6.

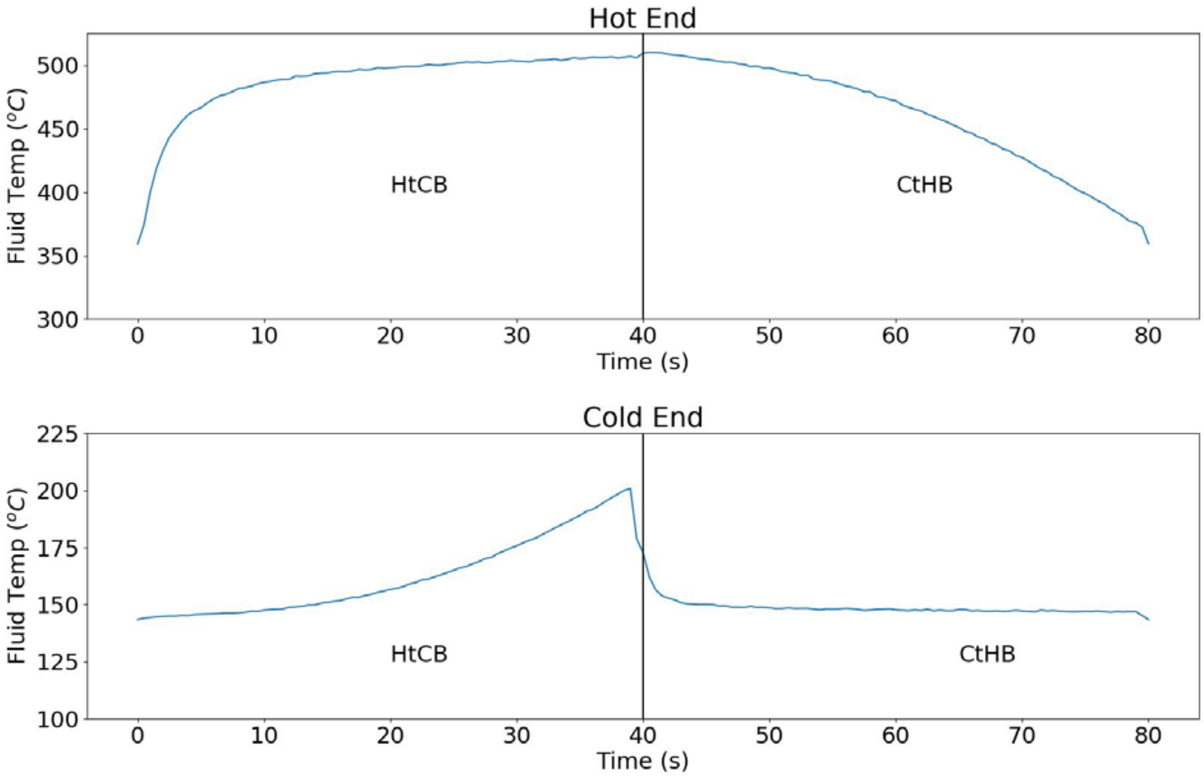

Figure 6 Temperature at both ends of regenerator bed throughout one cycle.

During the CtHB when the temperature falls fastest, the fittings on the hot side of the regenerator were observed to leak substantially. However, after the bed switched over and the HtCB began, the leak stopped and the fittings resealed. It was determined that the large difference in thermal mass between the fitting body and the tubing was causing the tubing to contract faster when cooling resulting in a leak. The solution was add a ¼" liner tube to the 9/16" tubing used in the fittings. The liner greatly increased the resistance to heat transfer from the fluid to the 9/16" tubing resulting in almost no temperature change throughout a cycle, and no leakage.

Heaters and Heat Exchangers

There are two heaters (HT-01, HT02) and two heat exchangers (HE-03, HE-04) used to reach to desired operating conditions. Having two heaters allows the regenerators to be run with an elevated cold inlet temperature, which imitates the high temperature heat exchanger in a Recompression Brayton Cycle (RCBC). To cut down on the amount of heat needed on the cold side, a PCHE (HE-03) is used to recover heat from the regenerators before a tube in tube heater is used to final amount of heat needed (HT-02). This heater can provide up to 6kW of heat at maximum power, which results in a maximum temperature of around 200°C. After the regenerators, a second heater (primary heater) is used to provide heat to make up for the ineffectiveness of the regenerators and any other thermal loss. This primary heater is used to add the heat that would be added in the primary heat exchanger in an actual cycle. The primary heater (HT-01) needs to be able to hold a relatively constant exit temperature even with a variable inlet temperature as seen in Figure 6. A constant exit temperature was maintained by using a large tank filled with molten salt to heat CO_2. The salt heater has 300lbs of molten nitrate salt which is effectively isothermal resulting in a constant exit temperature. 10kW of resistance heaters are located on the outside of the vessel providing heater power. The CO_2 flows through the salt in a ½" tubing that is coiled inside of the salt bath. There is good heat transfer between the outer shell of the heater were the thermocouples are located and the CO_2, with a temperature difference between the heater and the CO_2 of only 30°C. The temperature as a function of time at the inlet and exit to the heater is shown in Figure 7.

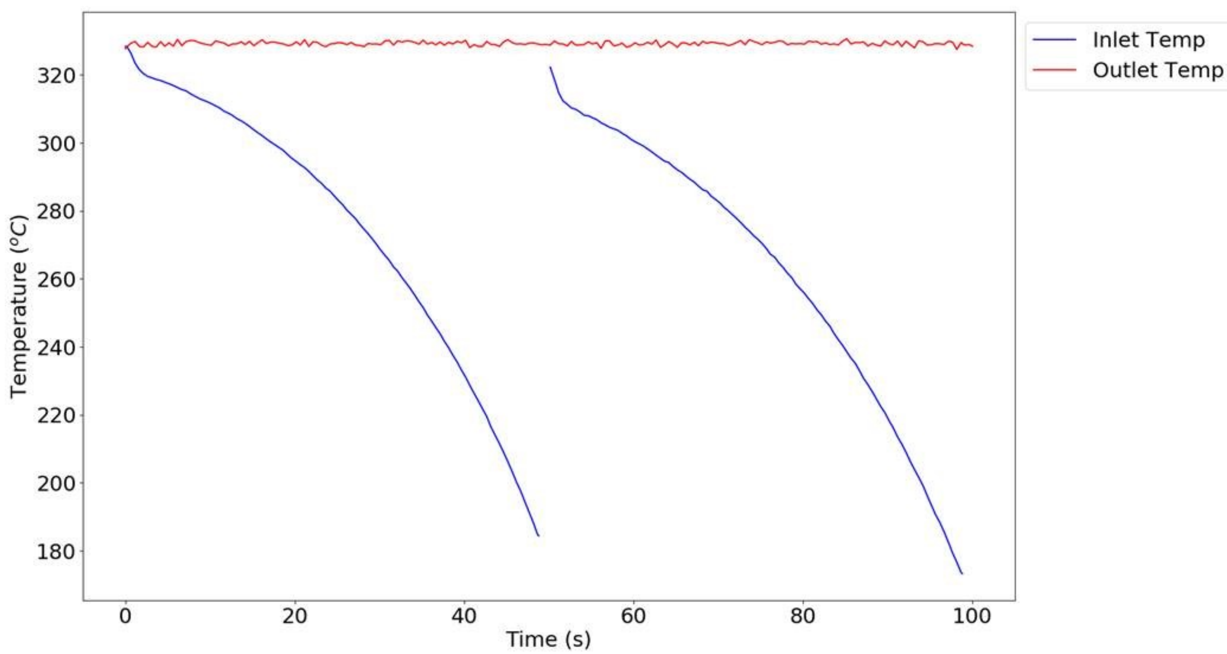

Figure 7 Temperature at inlet and exit of salt heater over an 100s cycle

Figure 7 shows the time shift of the heat load on the heater. At the start of the cycle (0s-10s) only a small heat input is needed to achieve a constant exit temperature. However, near the end of the blow a large amount of heat is needed to reach the constant exit temperature (35s-45s). Figure 7 shows the ability of the salt to maintain a constant exit temperature with this drastically changing inlet temperature.

The heater and all high temperature components are covered in a layer of Kaowool and a layer of pyrogel which effectively eliminated heat loss in the regenerators and greatly reduces temperature loss on the salt heater. The heat loss in the regenerator was measured to be less than 100W which less than 0.2% of the heat transfer in the regenerator.

Regenerator Construction

The regenerators for the test system were constructed with the goal of verifying the model so it could be used accurately at the 10MW level. The shell of the regenerator is a 1.5” sch 80 316 stainless pipe with 3 taps welded on for the temperature sensors in the bed. At each end there is an end cap and screen which holds the packed bed into place. One end cap was welded on and the spheres were filled in through the top while inserting the thermocouples into the center of the bed. Once the bed was full, the other end cap was welded into place. The packing fraction is measured by comparing the actual mass of spheres added to the measured volume of the bed, in this case it was about 37% void.

(a) (b)

Figure 8 (a) Packing material of test regenerator (1/8" 304 spheres) (b) constructed regenerator showing bed thermocouples and end pressure and temperature taps

Instrumentation

There are three performance measurements that are needed to confirm the accuracy of the model; effectiveness, pressure drop, and carryover. To measure the effectiveness, the total heat transfer must be calculated. This is done by measuring the temperature, pressure, and mass flow through the regenerators. The temperature measurements were made by K-type thermocouples at various locations along the regenerator bed (TI-07,11,12,16,20-23). The pressure measurements were made with pressure sensor (PI-01). Mass flow was measured by the Coriolis flow meter (FE-01). Pressure drop is measured using a differential pressure sensor (DPI-01). Carryover is not measured directly due to the extremely fast actuation time associated with the valves (less than one second). Instead carryover is calculated by integration of the mass in the bed at two times; however, for the experiment the actual temperature profile in the bed as measured by three thermocouples inserted into the bed (TI-08-10).

Data Analysis

Effectiveness calculations

The effectiveness of the regenerator is calculated using equation (0.6).

$$\varepsilon = \frac{Q_{reg}}{Q_{max,rec}} \tag{0.6}$$

where Q_{reg} is the heat transfer in the regenerator during the CtHB, and $Q_{max,rec}$ is the maximum heat transfer possible. The maximum heat transfer possible is defined as the value that would be obtained if a perfectly effective recuperator were operating at the same conditions as the regenerator; in this way, the effectiveness measured can be compared directly to the value that might be associated with a recuperator (remembering that a recuperator does not suffer from carryover). The CtHB was used to calculate Q_{reg} because this is the part of the cycle where the energy is being recovered from the bed. Q_{reg} is calculated using equation (0.7).

$$Q_{reg} = \int_0^{t_{CtHB}} \dot{m}\left(h_{11} - h_{07}\right) dt \tag{0.7}$$

where *h* is the specific enthalpy of the fluid and the subscripts indicate what temperature was used to calculate the enthalpy. The locations of the thermocouples in the packed bed are shown in Figure 9.

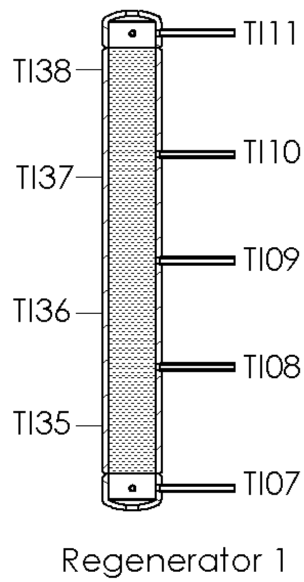

Figure 9 - Thermocouple locations for regenerators

The Coriolis flow meter was used for the mass flow rate in equation (0.7). The maximum heat transfer (denominator of equation (0.6)) was calculated using a sub-heat exchanger approach as described in (Nellis & Klein, 2009). This approach calculates the maximum heat transfer rate from prescribed temperature, pressure, and mass flow rates. The heat exchanger is divided into N segments that are each separately tasked with dealing with 1/Nth of the total duty and each section is analyzed using the effectiveness-*NTU* technique. A diagram of the sub-heat exchanger model is shown in Figure 10.

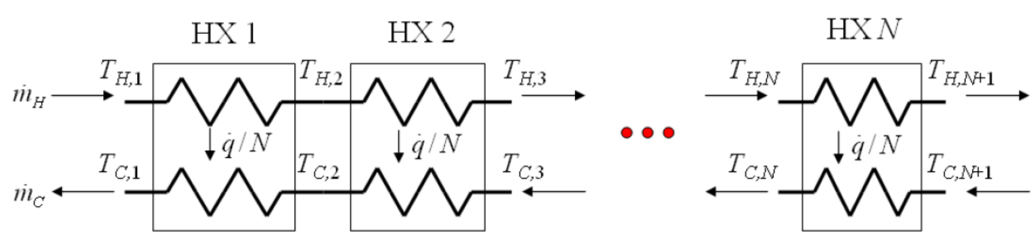

Figure 10 - Diagram of sub-heat exchanger model of a counter-flow heat exchanger

The inlet temperatures and pressures used in the counterflow heat exchanger model are calculated by finding their cyclic average value across a steady state portion of the experimental data. A 12 cycle average for TI11 and TI07 is shown in Figure 11.

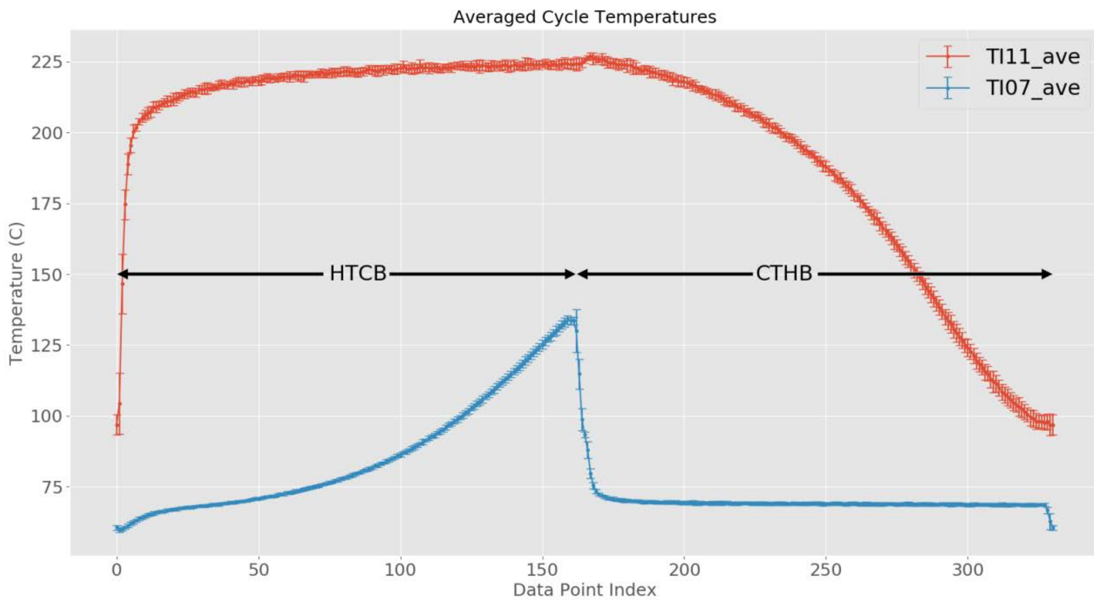

Figure 11 - Cyclic average of TI11 and TI07

The average value of TI11 during the HtCB is used as the hot inlet temperature to the recuperator and the average value of TI07 during the CtHB is used as the cold side inlet temperature. The same procedure is done for the cold and hot side pressure. To calculate an averaged mass flow rate that is comparable to the flow rate in an equivalent recuperator, equation (0.8) was used.

$$\bar{\dot{m}} = \frac{1}{t_{CtHB} + t_{valves}} \int_0^{t_{CtHB}} \dot{m}\, dt \qquad (0.8)$$

where t_{CtHB} is the CtHB switching time and t_{valves} is the time required for pressurization or blowdown as described in "Regenerator Operation". Note that t_{valves} is included in the denominator of equation (0.8) even though the integration time is only t_{CtHB} in order to account for the valve switching time, during which the mass flow rate is zero. In this way, the averaged mass flow rate is more similar to what an equivalent recuperator would experience (i.e., because a recuperator does not need to "stop" and switch it can process a smaller mass flow rate continuously in order to process the same total mass).

Carryover

One characteristic that is inherent and unique to regenerator operation is carryover. Carryover is fluid that remains trapped in the regenerator matrix material after the valves are switched. This affects the overall efficiency of the Brayton cycle because some fluid that was pressurized by the compressor is returned to the compressor without expanding through the turbine.

Experimental carryover was calculated by using the three temperature measurements in the regenerator bed, the two temperature measurements in the cap, and the pressure at the inlet of the bed to calculate a density profile of the fluid in the regenerator. The total void volume of the regenerator was divided into five equal parts and multiplied by the density at each respective location to get a fluid mass. The masses were then summed to get the total fluid mass. To calculate carryover, the mass was calculated at two different times and then subtracted. Since there are two regenerators in the system, this difference is multiplied by two:

$$m_{co} = 2\left(m_{end\ of\ CtHB} - m_{start\ of\ HtCB}\right) \tag{0.9}$$

Results

In total 26 test runs were conducted, each at its own conditions. For each run at least 5 cycles were recorded to eliminate uncertainty due to cycle to cycle variations. The three performance measurements (effectiveness, pressure drop, and carryover) were measured for each run.

Testing Conditions

The experimental testing conditions and effectiveness results are listed in Table 2.

Table 2 - Experimental Testing Conditions

	EFFECTIVENESS	HOT TEMP (C)	COLD TEMP (C)	HIGH PRESSURE (MPA)	LOW PRESSURE (MPA)
1	86.5%	253.2	70.5	7.8	3.4
2	88.0%	217.1	69.4	7.4	2.7
3	88.2%	222.6	68.8	7.7	4.5
4	95.8%	110.7	67.0	9.0	5.0
5	87.3%	326.7	40.3	6.4	5.0
6	95.3%	221.9	65.5	7.1	4.9
7	91.9%	401.0	74.6	6.4	5.0
8	93.0%	503.2	165.2	7.0	5.2
9	93.0%	307.9	51.2	8.5	5.9
10	89.6%	383.1	81.8	14.5	8.7
11	91.3%	480.6	149.3	14.9	9.1
12	93.0%	487.5	148.4	14.9	9.1
13	94.3%	489.2	147.3	15.0	9.2

The inlet temperature to the high temperature regenerator (i.e., the turbine outlet temperature) in a typical RCBC is about 560°C, however due to the components used for construction in the test facility, the maximum temperature possible in the salt heater was 550°C. Since the CO_2 temperature going to the salt heater is not constant, the CO_2 return temperature starts low and then reaches its highest point at the end of the cycle. For a salt heater temperature of 550°C the average CO_2 temperature leaving the heater is 480°C-490°C. The pressure in this case is about 9 MPa on the low pressure side, and 15 MPa on the high side. The low pressure side is close to the 8 MPa typical of a RCBC. The high side pressure is lower than a typical RCBC which usually has high pressures of 25 MPa due to the rating of the compressor. However, even though the pressure and temperature are slightly lower than in a RCBC the properties of CO_2 are well known in this region and the model would still be considered valid. Future testing plans include going to 25MPa with upgrades to the compressor.

In addition to tests at high temperature and pressure, a second set of tests was done where temperature, pressure, and mass flow were held constant and the switching time was changed. The result is a set of data where only the matrix capacity ratio changes between runs. The test was conducted at two mass flow rates, which results in a difference in NTU between the two data sets. The conditions are shown in Table 3.

Table 3 Test conditions for switching time increment

TEST	EFFECTIVENESS	HOT TEMP (C)	COLD TEMP (C)	HIGH PRESSURE (MPA)	LOW PRESSURE (MPA)	$\dot{m}$ (KG/S)	NTU
14-19	85.2%-99.5%	244.4	71.4	9.5	8.7	0.028	18
20-26	82.3%-99.8%	236.8	70.9	9.9	9.1	0.016	22

The regenerators held up well under testing, there was no noticeable damage to the bed after the 26 tests that correspond to over a hundred hours of operation. The biggest problems were leakage in the fittings as a result of the temperature change of the CO_2 flow. Sleeving the tubes to isolate them from the CO_2 flow and therefore prevent the large temperature changes solved the problem with leakage. Leakage was also noticed at the threaded connection between the body of the valve and the bonnet. At larger scales with welded connections the temperature driven leakage of the fittings and valves is not expected to be a problem.

Effectiveness

The experimental effectiveness results are plotted against the NTU-Cm-Effectiveness model in Figure 12.

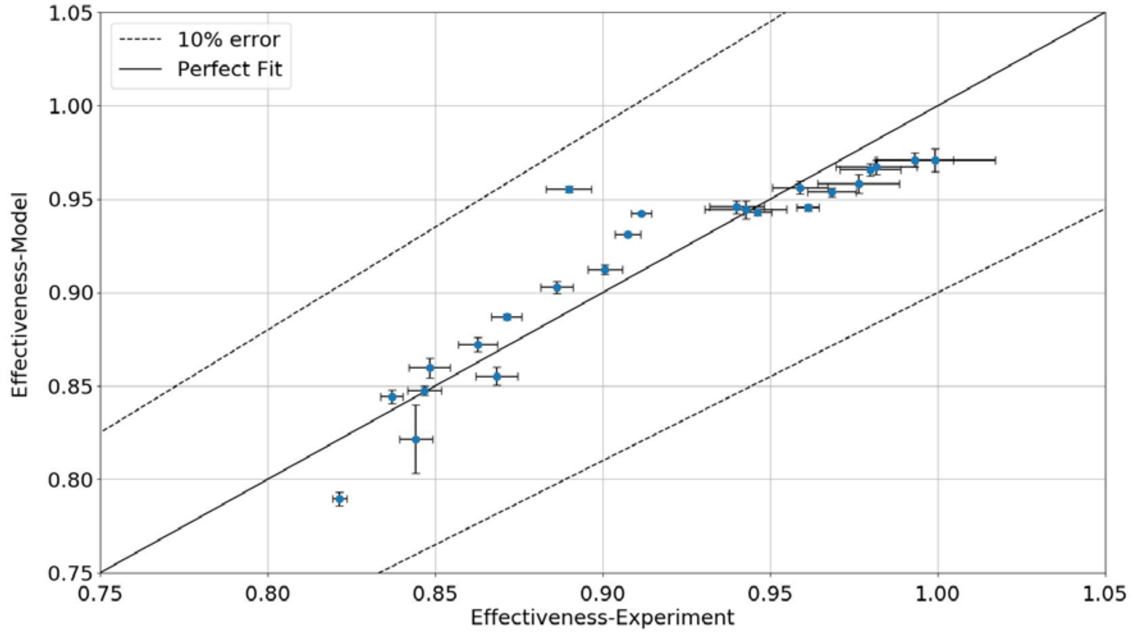

Figure 12 - Model vs experimental effectiveness

Figure 12 shows that all of the data points measured fall within 10 percent of the model. In addition, there seems to be a general trend where at lower effectiveness the model is over predicting effectiveness, while at higher effectiveness the model is under predicting performance. For a 10MW cycle the regenerator effectiveness will be greater than 95% meaning that the model will almost always under predict performance, which leads to a conservative estimate of regenerator performance. It is possible that the wall of the regenerator bed is contributing to the discrepancy in the regenerator performance. The error bars are significantly larger at higher effectiveness due to the effect of switching time. To get high effectiveness, the switching time must be short which means more carryover spikes of fluid through the regenerator which increases the uncertainty of the mass flow measurement. Since the high effectiveness runs are at shorter switching times, non-steady behavior in the heaters as a result of the control scheme also cause a higher cycle-to-cycle variation that leads to uncertainty, whereas the longer cycles averaged out the fluctuations. Since there is good agreement with the model, and error resulting in a conservative estimate of performance it was determined that the effectiveness-NTU-C_m model could be used for design purposes without any corrections.

Pressure Drop

The experimental pressure drop through the regenerator is plotted against the predicted pressure drop in Figure 13 and Figure 14 for the HtCB and CtHB respectively.

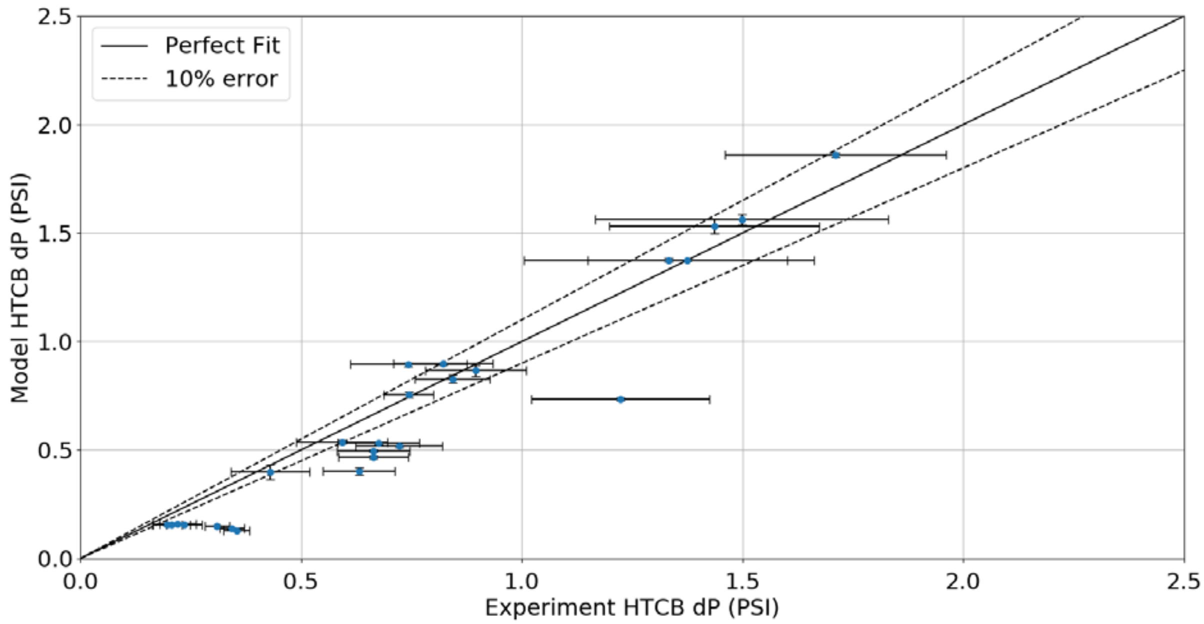

Figure 13 - Model vs Experimental pressure drop through the regenerator for the HtCB

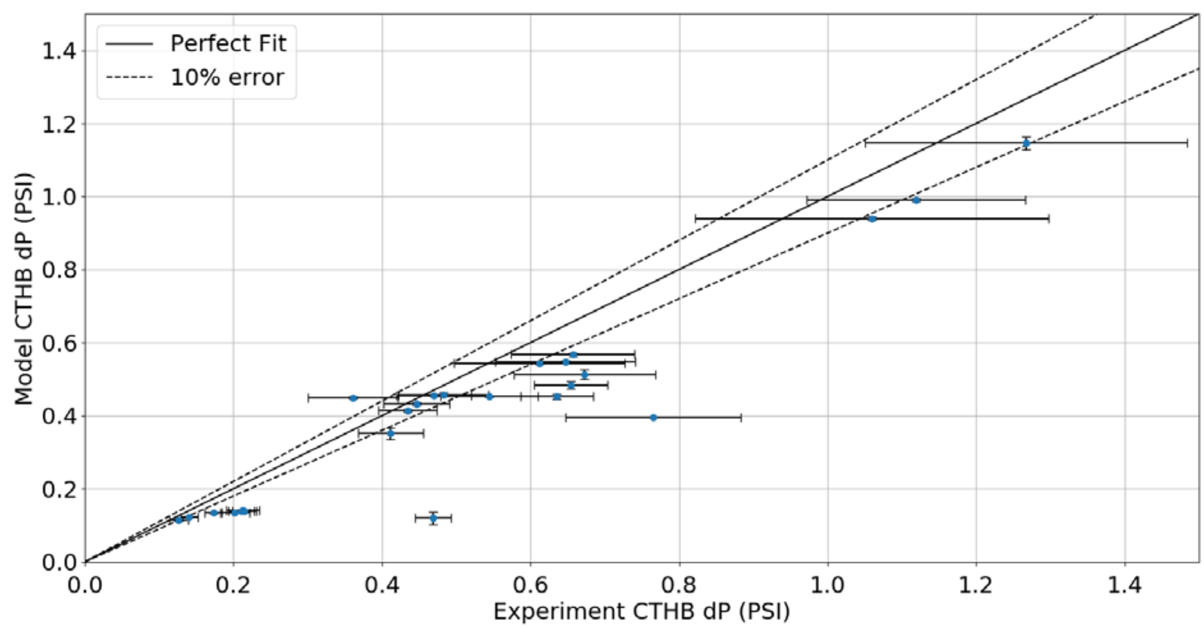

Figure 14 - Model vs experimental pressure drop through the regenerator for the CtHB

There is some observed error in the agreement between the modeled and measured pressure drop across the packed bed. In both the HtCB and CtHB when the experimental pressure drop is less than 1 psi the model under predicts pressure drop. One of the problems here is the test facility, the compressor does not have a constant mass flow rate and that means that every 3 seconds (the time it takes for one stroke of the compressor) the flow is changing by about 10 percent. This variation in mass flow can be seen a variation in pressure drop over the bed and the result are the very large error bars seen in Figure 13 and Figure 14. Another issue is the very small pressure drops that are being measured, they are on the same scale as the corrections needed to correct for the static pressure in the pressure taps. If the beds were designed to have a higher pressure drop the measurements would also be more accurate. Since the properties in the bed are well known and pressure drop through a packed bed has been exhaustively researched the mismatch between the experimental data and model is not a large concern [20–22]. Future tests will have a pressure drop that can be measured more accurately. A simple correction to ensure conservative results would be to increase the modeled pressure drop by 20 percent.

Carryover

Figure 15 shows the carry over results versus a NTU-Cm-Effectiveness model.

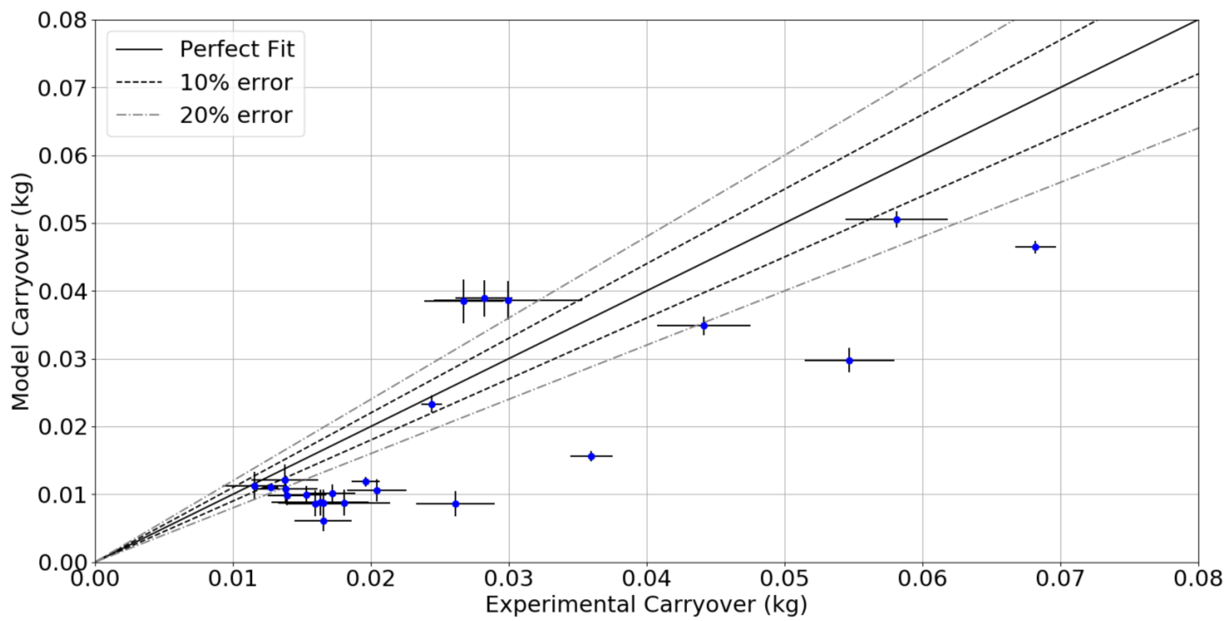

Figure 15 - Experimental vs model carryover results

The carryover experimental results generally do not agree well with the model predictions. The model assumes a linear temperature distribution whereas the experiment calculates the carry over using the actual, measured temperature distribution. The disagreement between model and experiment is therefore directly related to the non-linearity of the temperature distribution associated with the experimental conditions. An example of the model versus experimental temperature distribution is shown in Figure 16.

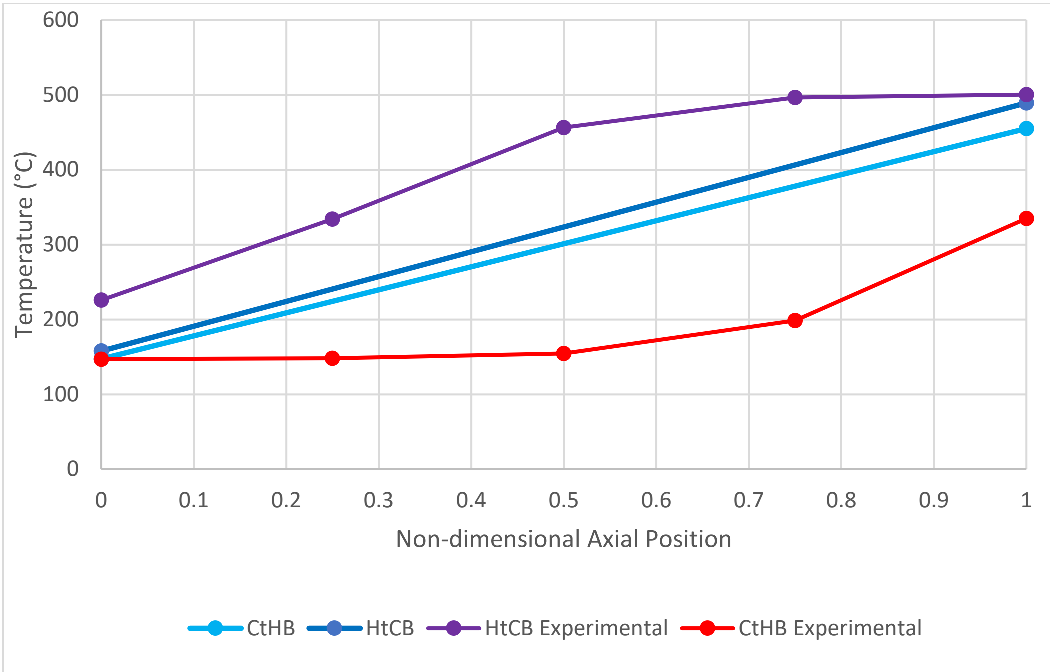

Figure 16 - Comparison of assumed linear temperature distribution and the experimental temperature distribution

The problem is clearly the difference between the actual final temperature and the model final temperature illustrated in Figure 16. Unfortunately, the effectiveness-NTU-C_m model does not predict the temperature profile in the bed so a correction is needed to make the model match the experimental results. Analysis of experimental data showed that there was no simple correlation between the shape of the temperature profile and the operating conditions of the regenerator, so the temperature profile used in the model is still assumed to be linear. A linear temperature profile for the HtCB and CtHB each only have one free temperature, since the inlet temperature for both are set by the cycle operating conditions. Thus, the cold exit temperature of the HtCB can be increased (lowering the amount of minimum mass in the bed), while the hot exit temperature of the CtHB can be decreased (increasing the amount of maximum mass in the bed). The free temperature of both HtCB and CtHB is changed by the same amount, dT. EES was used to calculate what dT was needed to get the same amount of carryover as the experimental data [23]. A corrected temperature profile for the example shown in Figure 16 is shown in Figure 17.

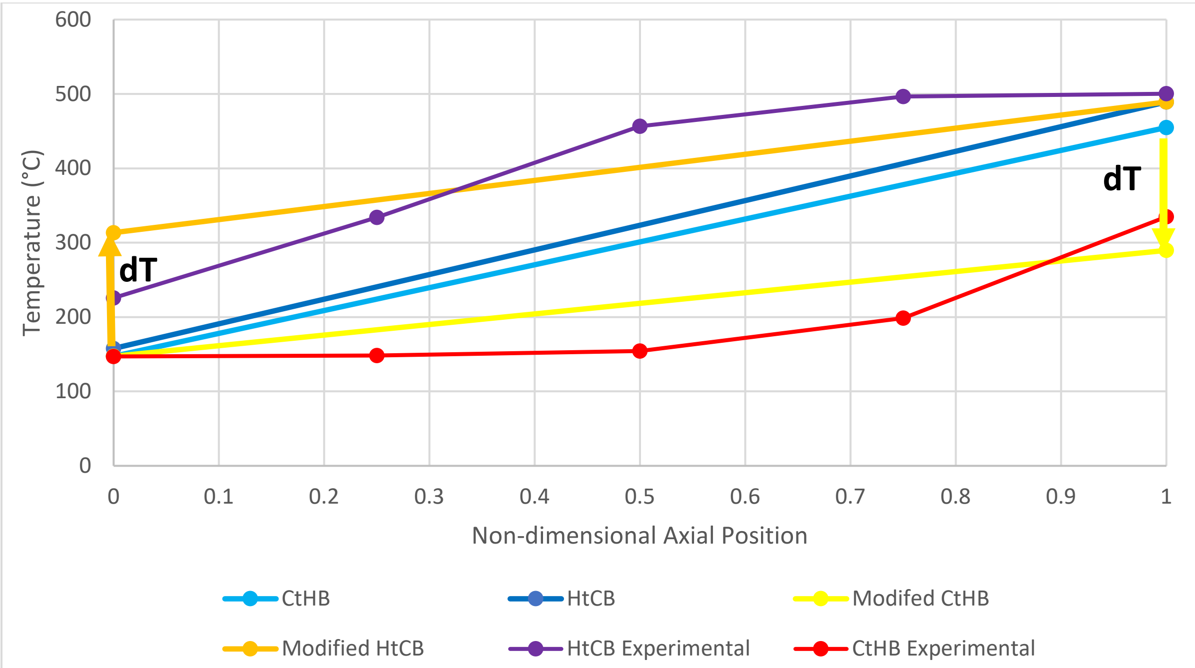

Figure 17 Temperature profile of example run showing temperature profiles at minimum and maximum mass in regenerator as well as corrected temperature profiles

Correcting the temperature profiles make them fit much closer to the experimental temperature profiles. An equation is needed for dT so it can be used to correct carryover in the model. The linear regression tool in EES was used to correlate dT to the inlet temperatures, inlet pressures, and matrix capacity ratio (C_m) [23]. The results are shown in Figure 18.

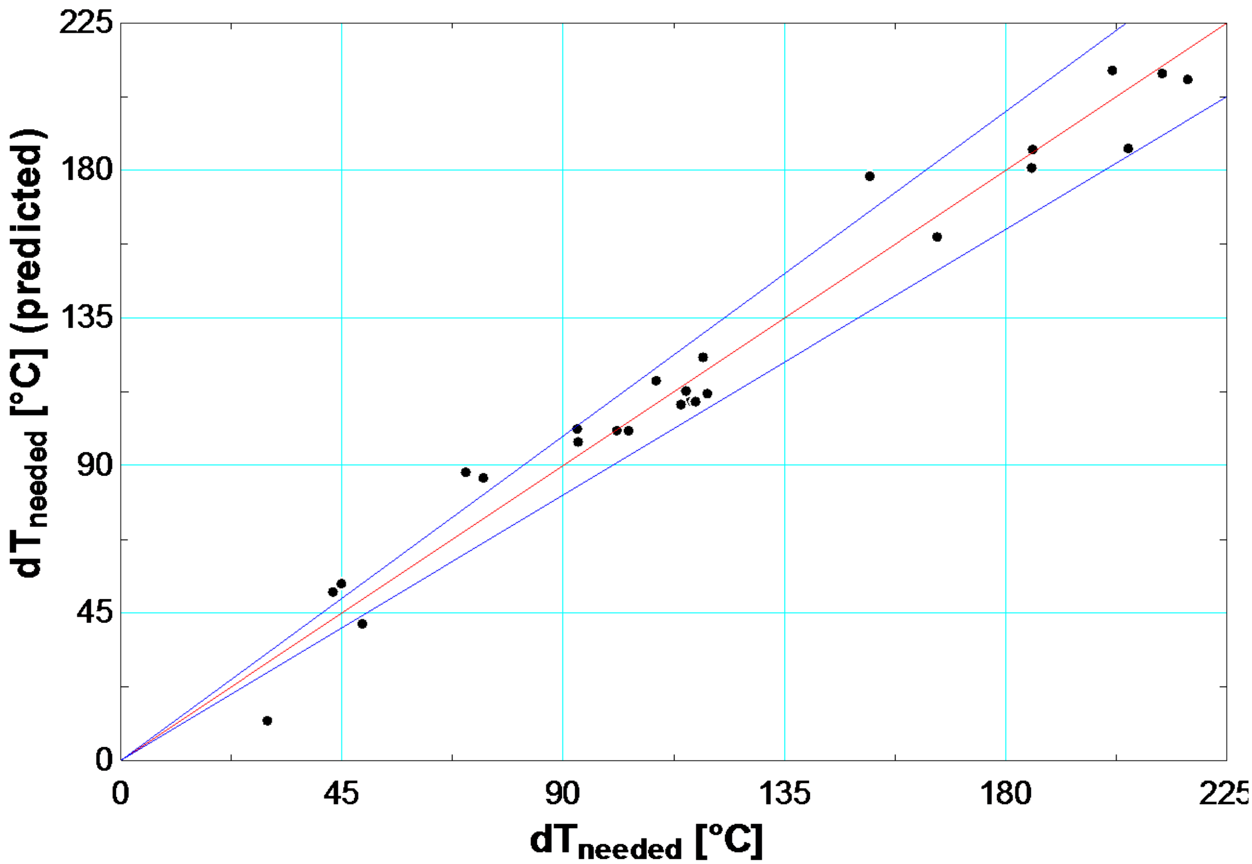

Figure 18 Differential temperature needed to correct carryover

The x axis shows what dT is needed to perfectly match the carryover in the model to the experimental data. The y axis shows what dT the linear regression will return using the inlet parameters (temperature, pressure, and matrix capacity ratio) for each run. Again, lines of 10% error are added. Most of the points are within 10% of the correct value. Using this correction method, the carryover can be recalculated as shown in Figure 19.

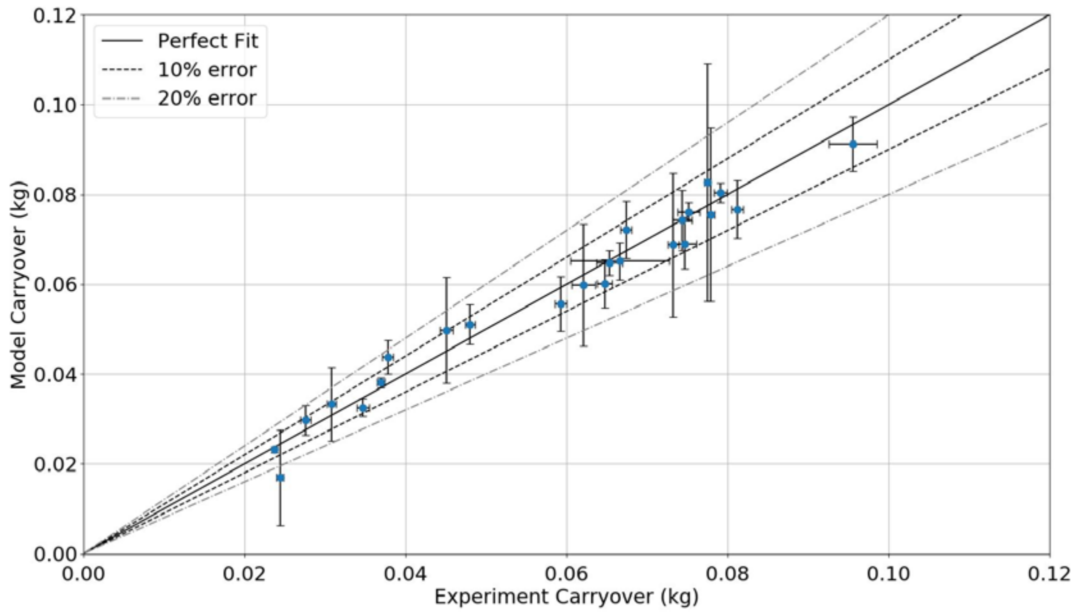

Figure 19 Experimental vs model carryover results with corrected temperature profiles

After correcting the carryover the model and experiments match very well, with almost all points within 10% error and all points being within 20% error. Some tests show very high uncertainty in the carryover since the correction factor dT is a function of inlet temperatures, pressures, and matrix capacity ratio. In particular the cases that have the largest uncertainty also have the largest matrix capacity ratios (greater than 3). An economically designed regenerator will have a matrix capacity ratio of approximately 1.7 so the uncertainty is still reasonable for regenerators with a matrix capacity ratio near the optimal matrix capacity ratio. The corrected model shows good agreement for all of the data points collected.

Conclusions

Regenerators have been shown to offer a 21% reduction in power block LCoE compared to PCHE recuperators at a 10MWe system size [14]. Since sCO_2 regenerators are a novel concept, a $10kW_{th}$ test facility was created to validate the effectiveness, pressure drop, and carryover of the model. In creating the facility and making measurements, great care was taken to ensure the data could be easily reduced into useable performance metrics. The effectiveness model matched well with the experimental data, with error of less than 10 percent for all runs. Pressure drop was harder to measure due to the very small magnitude (1 psi), however most measurements were within 20 percent of the model, and the model performed better at higher pressure drop. The 10MW system has much higher pressure drops (33psi) and the model will likely work well at those conditions. Future tests will be conducted at higher pressure drop for a better match with the 10MW system. The carryover provided to be the most difficult measurement because the pressurization and blowdown processes occur so quickly, making it difficult to measure directly by integrating flow. Instead carryover is calculated based on the temperature and pressure in the bed. Since the model uses a linear assumption for the temperature distribution it does not provide a good match with the actual, highly nonlinear temperature profile in the bed. As a result, the value for carryover is off. This issue is still being investigated. For now, a conservative correction of doubling the carryover is used. With this exception, the result of this work is a validated regenerator model that can be used for optimization at higher power levels.

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