Proceedings of Montreal 2018
Global Power and Propulsion Forum
7th – 9th May, 2018
www.pps.global

GPPS-NA-2018-0157

Exploration of Combustor Design for Direct Fired Oxy-fuel Application in a sCO2 Power Cycle

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Abstract

Significant interest has developed over the past several years in direct fired oxy-fuel combustion as a heat source for supercritical carbon dioxide (sCO_2) power cycles. This is a promising method for providing the needed thermal energy input while integrating carbon capture directly into the sCO_2 power cycle. This innovative method of energy addition into the sCO_2 has the potential to provide highly efficient power generation while maintaining extremely environmentally friendly emissions. sCO_2 power cycles rely on a very high degree of recuperation when compared to a traditional Rankine or open Brayton cycle. This large amount of recuperation results in a high combustor inlet temperature. This high inlet temperature makes design of the combustor challenging for a variety of reasons. Additionally, since the amount of oxygen is precisely controlled, proportions of CO_2, O_2, and fuel in the primary burning zone can all be controlled independently. This adds considerable flexibility to the design process, which is not typically found in a combustion system using air as the oxidizer. Another major challenge and difference between this combustion system and more typical gas turbine combustion systems in the vast variation in density of the inflowing CO_2 that occurs between the startup state and the design point condition.

The current work focuses on the design of a 1MW thermal sized combustor. This work lays out some of the basic design sizing and cases which should be studied as part of the design effort. The exploration of some of the major geometry features and dimensions are discussed. This design will need to be capable of startup, part load, and full load operation. The maximum exit temperature of this design will approach ~1200°C. Past cycle analysis has shown this temperature to be the maximum temperature the current state of the art recuperators will permit a closed sCO_2 cycle to operate. Simplified combustor geometry is described in detail so that researchers interested in simulating oxy-fuel combustion for sCO_2 environments will have a starting point on which to base their work. The results of this work will be useful for others considering some of the design challenges of a direct fired oxy-fuel combustor for sCO_2 application.

INTRODUCTION

Supercritical carbon dioxide (sCO_2) power cycles offer a potential pathway to increase the efficiency and decrease the costs of power generation. A great deal of interest in sCO_2 power cycles has developed over the past decade. These power cycles offer higher efficiencies than achievable in steam Rankine power cycles. At the same time many of the components, especially the turbomachinery and heat exchangers, are considerably smaller in size when compared to those used in a steam Rankine cycle with comparable power output.

The method of heat addition into a direct fired sCO_2 power cycle also has a number of advantages. Direct fired sCO_2 cycles utilize a combustor which burns fuel and pure oxidizer with the CO_2 acting as a diluent and temperature moderator. The first major advantage is that the energy from the combustion process is released directly into the CO_2 stream, as opposed to a transfer of energy via a heat exchanger. This removes the losses associated with heat exchangers of less than 100% effectiveness. In addition, because the reactions take

place within a closed loop stream of CO_2, the CO_2 produced by the reactions can easily be captured and sequestered. Typically the low side pressure and temperature of a sCO2 power cycle is comparable to the pressure of CO_2 pipelines which makes carbon capture and sequestration very simple and not a major parasitic power loss, as might be in a conventional combined cycle power plant. This efficient carbon separation is offset by the costly need to use an air separation unit to supply high pressure oxygen to the system.

The relative ease with which 100% carbon capture can be implemented on a direct fired oxy-fuel combustor power version of these cycles makes this an extremely attractive technology for future base load power generation. Many of the cycles being considered for use in power generation require a high degree of recuperation. This large amount of recuperation relative to the small amount of thermal input leads to a high temperature stream of CO_2 entering the combustor. The exact combustor inlet temperature depends on the type of cycle which is chosen and the designed turbine inlet temperature. It is believed that combustor inlet temperatures of around 700°C will be the typical design point for these cycles. This elevated combustor inlet temperature poses a significant design challenge for the combustion system, because of material limitations, and auto ignition risks.

In addition to the challenges posed by high inlet temperatures, the high pressure and high CO2 concentration lead to other difficulties. Many chemical kinetic mechanisms available in the literature are not appropriate for such high levels of CO_2 at pressure in the 150 to 300 bar range [1]. Recent work conducted by several investigators is beginning to shed light on the ignition delay time [2] [3]. These investigations are expected to continue to provide useful information at higher pressures and conditions at which oxy-fuel combustors for super critical CO_2 are expected to operate. Further investigations are needed into other aspects of the combustion process, specifically CO oxidation and flame speeds in supercritical CO_2 conditions. Recently there has been some publication of continuous flow combustion testing. Limited results have been presented for continuous flow oxy-fuel combustion testing showing some details of oxy-combustor design for sCO_2 combustors [4] [5] [6]. These works are expected to result in a demonstration of oxy-fuel combustion in a semi-closed cycle within the next couple of years.

A team of Southwest Research Institute, Thar Energy, Georgia Tech, University of Central Florida, and G.E. Global Research Center is currently working to design and build a $1MW_{th}$ (thermal) scale direct fired oxy-fuel combustor for sCO2 applications. Presently, the initial primary objective addressed in this work is the study of the behavior of a sCO_2 swirl stabilized fuel injector in a combustor. Specifically, analyzing, characterizing and understanding the performance, flow behavior and combustion process under the effect of various design variables, including: the combustion chamber geometry, mainstream flow swirl angle, fuel injection scheme, and cooling strategy. To-date various configurations of each of these design variables have been studied in a myriad of computational simulations. The purpose of the discussion in this paper is to show a few early results from RANS simulations of these combustor geometries. A detailed description of the computational setup and modeling details sufficient for other groups to perform their own simulations is presented. This approach will hopefully establish a more-common baseline and promote collaboration and accelerated progress in achieving viable solutions to the direct fired oxy-fuel sCO_2 combustor.

GEOMETRY

The initial geometry for the oxy-fuel combustor is very similar to a traditional style combustor. The envelope of the combustor is cylindrical in form with three main sections comprising the domain considered in this work, namely, the swirler, the combustion chamber and the bypass flow. The swirler is the most upstream portion of the combustor and contains a portion of the CO_2 coming into the combustor. Prior to the swirler, oxygen is introduced and will be mixed with the CO_2. Once oxidizer has been introduced into the CO2, the swirler will then impart rotational swirl into the mixed flow by swirling veins, channels, or some other means. Then the flow is directed into the main combustion chamber.

Fuel is introduced into the system through strategically placed injectors, either during the swirl process or shortly after leaving the swirler. The authors have considered several examples of fuel injection schemes. The one presented in this geometry, termed the partial-premix scheme, injects fuel into the oxidizer/CO_2 within the swirling vane passage. This injection in the vane passage allows for limited fuel and oxidizer mixing before entering the main combustion chamber.

The geometry used to generate the results of this work is depicted in Figure 1 and Figure 2. These figures only show the fluid domain for the computational simulations. The outer diameter of the combustion chamber is 3.0-inches and is 10.0-inches from chamber head to exit. Two 0.05-inch wide dilution cooling slots were used, the most upstream slot located 5.0-inches from the head of the combustor, with a spacing of 1.0-inch between them. The remainder of the bypass CO_2 was introduced as effusion cooling sources at the 5.0-inch long outer chamber wall upstream of the dilution

slots and the chamber head. This is explained in more detail in the following section.

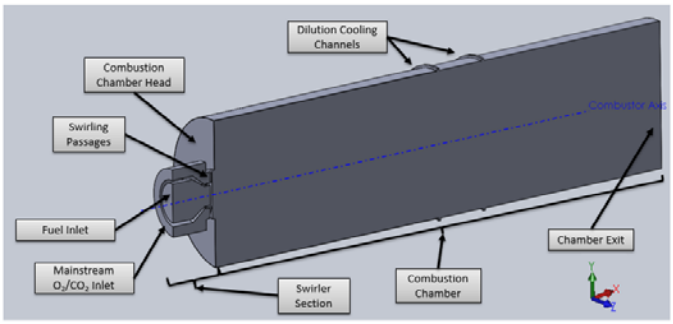

Figure 1. Cross-section of the Simulated Fluid Domain within the Combustor Geometry.

The swirler contains two inlet flow streams. The first is for the mainstream CO_2 and O_2 and has an outer diameter of 1.38-inches and inner diameter of .95-inches. The second inlet is to serve as the plenum for the fuel injectors. As depicted in Figure 2, 16 swirling channels were used to impart swirl onto the main CO_2/O_2 inlet stream. Swirl angle is defined here as measured from the combustor centerline axis; thus, a higher swirl angle results in a greater radial component of the incoming flow streams. Swirl angles of 40° and 30° were used in the results of this effort. In addition, the passages were tilted with a 10° radial angle, toward the combustor centerline axis, *i.e.,* as measured from the top (short-edge) plane of each channel. The swirl channels begin with an exterior diameter of 0.947-inches, as shown in Figure 2. Swirl channels had a cross section of 0.0506-inches by 0.207–inches and extend 0.135–inches in the x-direction. The projected area looking along the axis of the combustor relative to the combustor centerline of the 40° case is shown in Figure 2. It is important to note that the cross sectional area remains constant as the swirler angle is changed. Fuel was injected with a partial-premix scheme. Eight fuel ports of diameter 0.0049-inches were located in half of the 16 swirling passages, alternating every other passage.

The remainder of the pure CO_2 is diverted into a bypass stream, to be introduced into the combustion chamber as cooling and dilution flow. Typically this bypass stream flows in the annulus between the combustion chamber and the outer combustor annulus and flows parallel to the main CO_2 stream. There are multiple methods by which the flow is introduced into the main chamber. Typically, flow enters via slots or holes in the combustion chamber liner. Depending on the size, orientation, and flow rate of the entrance zone, these orifices are used as film and/or effusion cooling for protection on the chamber walls, and as stream dilution and quenching of the hot reaction gases to achieve a suitable final temperature for equipment downstream.

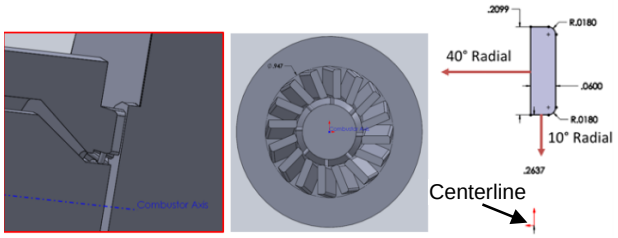

Figure 2. Details of the swirler section of the combustor: Left: Cross-section of the swirler. Center: View of the swirler from combustor exit. Right: Projected area of swirler passage.

The combustor geometry to be manufactured contains significant differences to the computational domain presented here. However, this simplified combustor geometry, can be modified by changing the swirl angle, diameter, fuel injector location, and other parameters to provide a useful test case to explore variations on improved combustor geometries. The authors hope that the geometry provided will be a starting point for those interested in modeling oxy-combustion in a sCO_2 environment.

BOUNDARY CONDITIONS

Boundary conditions used for the modeling of an oxy-fuel combustor design vary substantially with the combustor design. The choice by the combustor designer of oxygen to diluent ratio in the flame zone is a significant difference compared to a more conventional gas turbine combustor. A conventional gas turbine combustor uses the fixed oxygen-nitrogen ratio of the air, and adjusts the fuel flow rate to achieve the desired air fuel ratio. In this case, the designer has control over the fuel, oxidizer, and diluent streams, with the requirement that the equivalence ratio is as close to one as possible while still meeting other performance metrics. This prevents excess oxygen from circulating within the system, which will have a detrimental effect on cycle performance. Design, part load, cold start, and other cases all need to be considered as part of the design process. Presented herein, are the design point boundary conditions, startup conditions, and another condition particular to closed loop operation, termed the “fast ramp start” condition.

The design point boundary conditions for the combustor geometry described in the preceding section are as follows. The downstream fluid boundary was set as a pressure outlet equal to 196 bar, a 2% total pressure drop from the operating pressure of 200 bar. Mass flow inlet boundary conditions were set for the primary CO_2/O_2 mixed inlet and CH_4 fuel plenum. A diagram of the inlet mass flow split is provided in Figure 3. CO_2 entered the swirler inlet at 0.325 kg/s. This is equal to 20% of the total combustor flow (CO_2 plus fuel

and oxidizer). The remainder of the inlet CO_2 is split between the cooling flow to the head and walls of the combustion chamber and the dilution flow. The split used in this model is 0.4875 kg/s or 30% of the total flow for cooling, with 0.7127 kg/s (43.9%) utilized as dilution flow to cool the temperature of the combustor outlet flow. Incoming CO_2 streams were prescribed as 700°C while CH_4 is 200°C. The mixture temperature of the CO_2/O_2 stream was 575°C. These temperatures account for mixing and heat transfer from the high temperature CO_2 surrounding the injector.

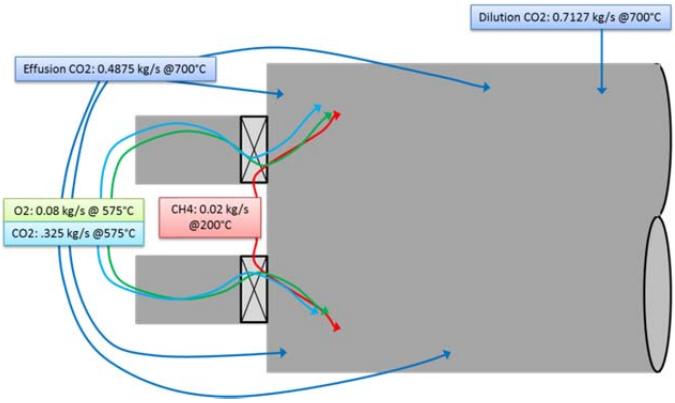

Figure 3. Schematic of combustor with design mass flow splits.

In this case, neither the cooling flow nor the dilution flow contains any oxidizer. This lack of oxygen in either of these streams is a significant difference from a traditional gas turbine design, and makes the distinction between a primary and secondary burning zones largely irrelevant for this design. Other geometries and stream arrangements could be considered where portions of the cooling or other flows contain oxygen. Since a large amount of excess oxygen is undesirable, care should be taken to prevent unreacted oxygen from leaving the combustion chamber. Any unreacted oxygen imposes a negative penalty on the system by increasing the demands on the air separation unit and cycle compressors. Still, complete oxidation of carbon monoxide may call for a mixture which is slightly oxygen rich. Mixtures presented in this paper have 1% excess O_2 by mole fraction to account for this factor.

Apart from the design case, two other extreme types of cases should be considered for modeling of an oxy-fuel combustor for direct fired sCO_2 applications. Examples of these two cases for the mass flow rates needed for a $1MW_{th}$ scale combustor are shown below in Table 1. The operating conditions represented by these cases would occur in actual plant operation during the startup of a sCO_2 plant. During startup, both the mass flow and pressure ratio of the cycle will be at some fraction of the design operating point. For these cases two-thirds of the design pressure ratio and mass flow were selected as a representative startup point, based on internal cycle modeling. The actual startup pressure and mass flow for a given combustor will be dependent on the cycle type, off-design turbomachinery performance, and the heat rejection scheme and ambient conditions. Each combustor designer will need to select conditions appropriate for the cycle for which their combustor is being designed; however, the conditions presented here represent a plausible startup condition for the current application.

In considering the startup case, it is helpful to remember that CO_2 is chosen as a working fluid because of its very large density difference, which allows for efficient cycle design point operation. This same large density difference can potentially cause problems for a combustor during start up. As can be seen in Table 1 the density of the sCO_2 entering the combustor during startup varies dramatically from the design point case. The fluid entering the combustor during cold start is roughly six times as dense as at design point operation. This will significantly alter combustor residence times. Simultaneously designing a combustor to handle both of these conditions requires careful consideration. The cold start case mass flow rates are specified in Table 1 with $1MW_{th}$ input.

In addition to the cold start case, another case, termed the “fast start” case is also worth considering. This case arises from the need to bring the cycle to thermal equilibrium, and thus design point, as rapidly as possible in order to minimize the inefficient startup operation. As shown in Table 1, this case uses significantly more fuel than the design point in order to rapidly increase the cycle temperature. This case shown here assumes some amount of time has been spent at start up condition, sufficient to elevate the combustor inlet temperature of the CO_2 to 150°C prior to the “fast start.” In the example case, the fuel and oxidizer flow rate is 68% higher than at design point. This percentage was selected by adding enough fuel and oxygen to the system to increase the outlet temperature to that of the design point combustor outlet condition, with a combustor inlet temperature near startup. As with startup the level of combustor thermal duty required will depend on cycle design considerations. Combustor thermal duty during this start up process depends on the desired cycle temperature ramp rate, which may be dependent on component thermal stresses. In addition, practical considerations need to be accounted for, such as: the air separation unit capacity and, the ability of the fuel system to supply oxygen and fuel in excess of design point demands, and the extra cost of these larger systems or potential storage systems. While termed a “fast ramp start,” in reality the combustor may be required to operate in this condition for an extended

period of time, perhaps as long as an hour, depending on the cycle hardware thermal transient.

Table 1. Mass Flows for Several Cases of Interest for Combustor Modeling

	Design Point	Cold Start	Fast Ramp Start
CO_2 Mass Flow (kg/s)	1.53	1.02	1.02
Pressure (bar)	200.00	133.33	133.33
CO_2 Inlet Temp (°C)	700	50	150
CO_2 Density (kg/m^3)	104.2	649.4	203.5
O_2 Mass Flow (kg/s)	0.0806	0.0806	0.1360
CH_4 Mass Flow (kg/s)	0.0200	0.0200	0.0338

MODELING

Once a model was created for the fluid domain, a computational mesh was created using mainly tetrahedron elements, with 5-6 hexahedron elements used for boundary layer inflation along wall boundaries. A quarter section of the domain, with periodic boundary conditions was used to reduce computational load. For the results presented in this document, a mesh of 1.125 million elements was used. Figure 4 shows the computational mesh.

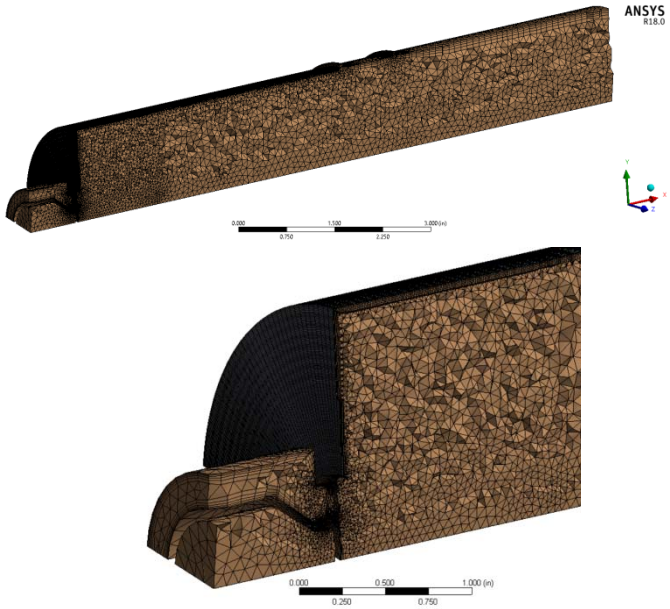

Figure 4. Computational mesh for the Fluent simulations.

The simulation was performed in ANSYS Fluent 18.0. The simulation was completed with a pseudo steady-state RANS model using the realizable k-epsilon turbulence model. Standard wall functions and compressibility effects with the default set of turbulent Prandtl and Schmidt numbers were prescribed.

The downstream fluid boundary was set at a pressure of 196 bar. A mass flow inlet boundary condition was set for the mainstream CO_2/O_2 flow inlet. A perfectly mixed fluid composition was prescribed, assuming O_2 was introduced into the system far enough upstream to become well-mixed with the CO_2. A mass flow boundary was also used for the CH_4 fuel plenum. Dilution CO_2 from the bypass flow was injected into the domain with mass flow boundaries at the two dilution slots. These slots are used to simplify the computational domain, actual dilution ports will most likely be circular holes. The walls of the combustor were set to adiabatic, to simplify the physics considered, which will affect the thermal solution.

The remainder of the cooling CO_2 not passing through the swirler or dilution enters the domain as effusion cooling along the 5.0-inches of the combustor liner upstream of the dilution ports and in the combustor head. The effusion cooling was implemented as CO_2 mass and energy sources at cells nearest the outermost walls. This was accomplished in Fluent by creating thin volumes of fluid directly adjacent to the outer wall in the CAD geometry. These thin volumes were then meshed individually as unique zones in the CFD domain. Mass and energy sources were then assigned to the volumes, such that the total integrated mass of the volumes was equal to the balance of the prescribed total flow minus the primary inlet. Mass enters the flow uniformly distributed in each cell and was assigned zero initial velocity. The total energy flux of the source was set by determining the energy necessary to achieve the desired incoming temperatures within the volume under flowing, but non-reacting conditions. This technique of modeling effusion cooling is one method to achieve both mass and energy sources with no-slip wall conditions in Fluent, which is not achievable by simply prescribing the combustor liner walls as mass flow inlets.

Fluid properties were calculated using the ideal gas equation of state, kinetic-theory for mass diffusivity, polynomial regressions for specific heat and other transport properties, and mixing rules for mixture properties. The ideal gas assumption is expected to yield errors of 10% for density and specific heat, as independently estimated and as reported by Manikantachari *et al*. [7] as compared to NIST REFPROP and other cubic equations of state. Chemical kinetics were modeled using a custom reduced reaction mechanism with 12 species and 25 reactions provided by Wenting Sun of Georgia Tech. Calculating fluid properties with these assumptions were considered adequate for these initial studies, and future works will likely include better estimations. The mechanism was created leveraging the USC Mech II kinetic mechanism. USC Mech II was first reduced, then optimized by an in-house code at Georgia Institute of Technology and finally

validated against auto-ignition calculations using detailed USC Mech II. The stiff chemistry solver in Fluent was used for CFD simulations.

RESULTS AND DISCUSSION

Presented in this section are selected sample RANS results from the previously described geometry, at design point boundary conditions. Two other geometries, one with a smaller diameter and one with a higher swirl angle, are also discussed. It should be remembered that these results are generated from the sample geometry, which has several potential issues which would need to be corrected before attempting to use the geometry for an actual combustion experiment.

Figure 5 shows a contour of the temperature field in addition to a velocity vector plot in the lower half of the figure. The velocity vectors clearly show the strong recirculation zone which forms in the center of the combustor. The temperature contour shows that this recirculation zone contains very high temperature gas. The mixing of this high temperature gas with the fresh unburned fuel and oxidizer stabilizes the flame. In addition to the flame stabilization, several other features

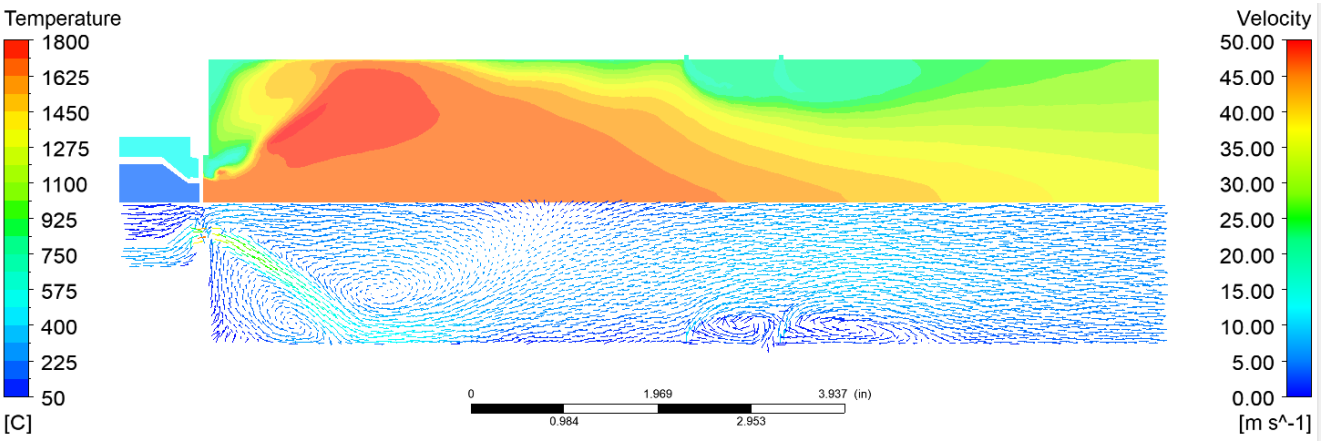

Figure 5. Temperature contour (top) and velocity vector plot (bottom) for 30° case with 3-inch combustor diameter

are worth mention from this figure. The recirculation zone in the corner of the combustion chamber is much colder and does not contribute to the flame stabilization and primarily contains CO_2 from the combustor head cooling. When looking at the temperature contour it is apparent that there are a couple of concerning areas in which very high temperature gas is in close proximity to the combustor walls. In this case, the walls are set to adiabatic temperature, so the magnitude of the hot spots may be lessened when heat transfer effects are included. Temperatures at the level observed in these results would likely pose a problem in an actual hardware test.

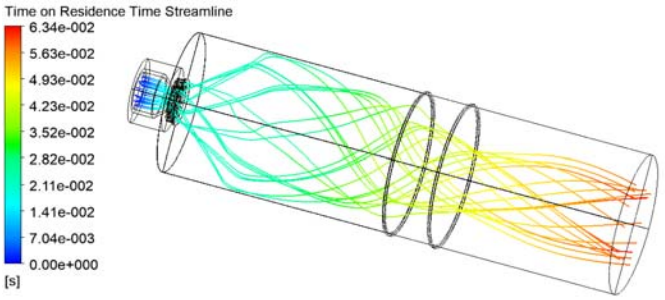

Figure 6. Streamlines originating in the fuel plenum, colored by residence time

Figure 6 shows streamlines which originate in the fuel manifold. A significant portion of the residence time shown in this figure occurs within the fuel manifold. Figure 7 shows the position and temperature of a streamline trace as a function of residence time. This figure shows that the total residence time in the combustor is just under 0.04 seconds. At approximately 0.02 seconds of residence time the dilution flow is injected. As this mixes with the hot combustion gasses, the reactions in the flow will rapidly stop.

Figure 8 shows the temperature maximum and average of the combustion gasses along the length of the combustor. This temperature plot indicates that the temperatures in the zones of maximum heat release are extremely hot. These temperatures are caused by high concentrations of oxygen and fuel. The elevated temperatures are distributed fairly evenly through the reaction zone. It is important to realize that this is a RANS simulation and does not fully capture the unstable nature of the structures formed in the shear layer, which will substantially affect the maximum temperature. Average temperatures exceed 1700°C at approximately 1.75-inches from the injector plane.

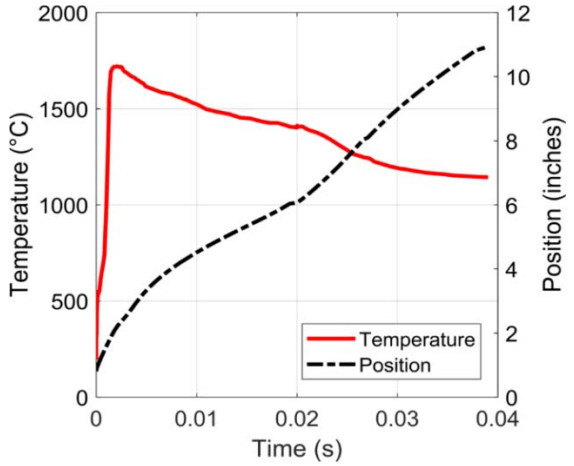

Figure 7. Temperature and axial position as function of time

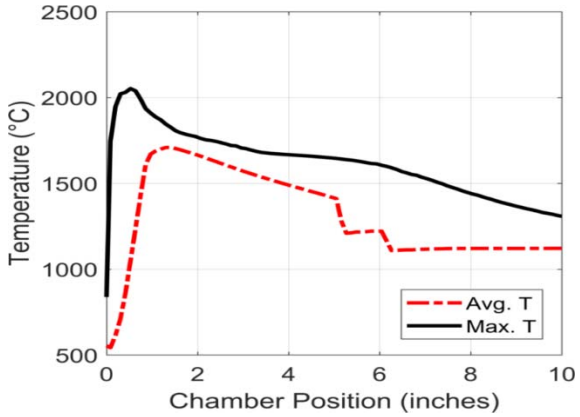

Figure 8. Average temperature and maximum temperature through the combustor

Figure 9 shows a comparison of results from two additional simulations, to explore the influence of the combustion chamber diameter. The top portion of this figure shows the results from a 3-inch diameter combustor, while the bottom figure shows the results from a smaller 2-inch diameter combustor. It is important to note that these combustor geometries have somewhat different boundary conditions. The flows entering the swirler in both of the geometries pictured in Figure 9 compose 38% of the total flow, and the dilution holes have been deleted from the geometry. This tends to result in a cooler recirculation zone as compared to the case shown in . Apart from that difference, the geometry is the same as the previous discussion. This significant change in the O_2/CO_2 ratio illustrates the ability of the designer to explore this parameter, a flexibility which is not available in a typical gas turbine combustor design. Among other possibilities, this allows the designed to control the primary zone temperature, while still maintaining an equivalence ratio of 1.0. Figure 9 shows that the larger diameter combustor tends to have a larger recirculation zone due the reduced axial velocity. In this case the larger recirculation zone pulls in a substantial amount of cooling flow which reduces the recirculation zone temperature. In addition, the corner recirculation zone is larger and contains more products of combustion in the larger 3-inch diameter case.

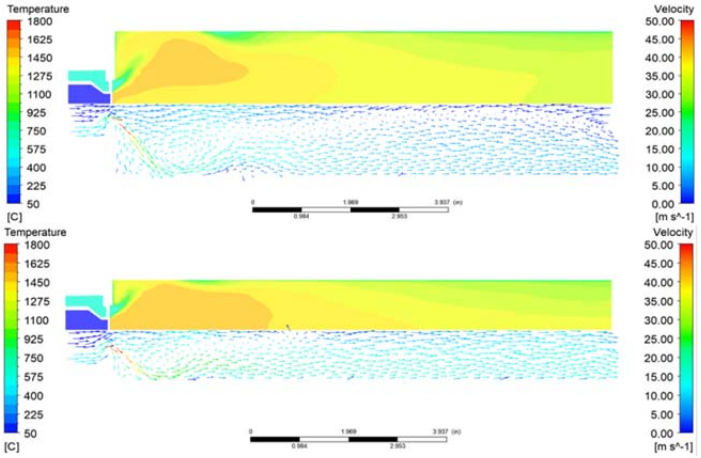

Figure 9. Temperature contours and velocity vector field of various combustor diameters. Top: 3-inch combustor diameter, Bottom: 2-inch combustor diameter

Another variable which can be studied is the swirl angle of the swirler. Figure 10 shows two different angles of swirl, both with a 2-inch diameter combustor and the same swirl mass flow of 38% from the previous figure. The upper image shows the 30° swirl (which is a lower amount of swirl) and does not cause as rapid of a radial spread and has a slightly weaker recirculation than the 40° swirl case seen on the bottom. The stronger recirculation means that the recirculation bubble is smaller and the center is closer to the head of the combustor. Also of note in comparing the two temperature contours, is how close the high temperature combustion product gases are to the chamber wall in the 40° case as compared to the 30° case. With a larger radial momentum component at the exit of the swirler, the higher swirler cases tend to strip away the protective film cooling layer much more significantly. If unaddressed this could lead to potential thermal related failure in the combustor liner.

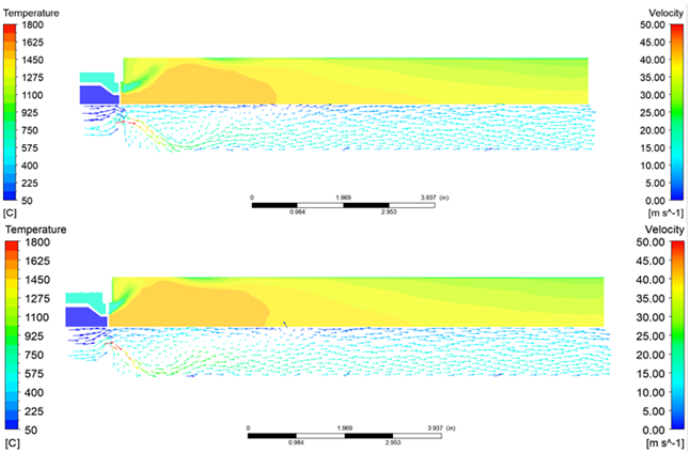

Figure 10. Temperature contours and velocity vector field showing effects of swirl angle: Top: 30° swirl angle, Bottom: 40° swirl angle.

Several different off design conditions were considered as part of this combustor development process. Two of these conditions which are believed to pose a unique challenge have been provided in Table 1. These cases are the startup cases, where the combustor is started while the cycle is operating at partial pressure and mass flow rate. The startup conditions for a sCO_2 power cycle will require the loop to be started with high pressure and low temperature. The temperature and pressure entering the combustor at these conditions result in dense phase sCO_2 entering the combustor. sCO_2 at these conditions exhibits strongly non-ideal state behavior. Simulations presented here were performed with ideal gas assumptions, as a point of interest to examine the results, with the knowledge that real gas equations of state are needed at these conditions. Both of the off-design cases presented here, should be simulated using a real gas equation of state.

Results from the cold start conditions are shown in comparison to the baseline design point case below in Figure 11. Given the much cooler inlet temperatures, it is expected that the temperatures throughout the combustor are considerably cooler than those found in the design point case. Of the interesting observations and potential issues with this combustor condition is the large difference in inlet density, which results in much slower flow through the swirler. This allows for an extended mixing period of the fuel and oxidizer in the swirler passage, and in this case a small zone of hot gas attached can be seen anchored on the trailing edge of the swirler vane. This attachment could potentially cause damage to the combustor, as the simulation indicated heat release occurring in this zone. A similar sort of attachment can be seen in the fast ramp startup case as seen in the bottom of Figure 11. The fast ramp case results in extremely high temperatures, and the simulations seem to indicate that the fuel flow rate chosen is probably too aggressive, and a lower fuel flow rate is needed.

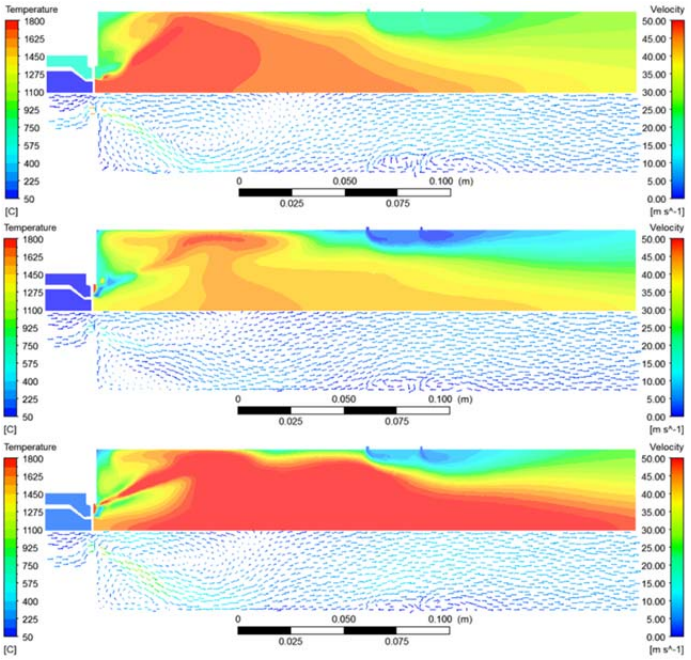

Figure 11. Top: 30° swirl design point case. Middle: Cold start case. Bottom: Fast ramp startup case

CONCLUSIONS

The rapid pace at which sCO_2 power cycles have advanced over the last several years is very promising. One of the most promising technologies associated with sCO_2 power cycles is the direct fired power cycle configuration. This configuration allows for efficient capture of CO_2 produced as a product of combustion. This configuration requires a unique combustor configuration capable of operating in a wide range of conditions. This type of combustor has not yet been experimentally reported on in the open literature. A team of Southwest Research Institute, Thar Energy, Georgia Tech, University of Central Florida, and G.E. Global Research Center is working to develop a $1MW_{th}$ demonstration combustor, which should be operational in the next couple of years.

This paper has provided a discussion of some of the requirements for such a combustor. A geometry example has been laid out in enough detail that an interested party could reconstruct the geometry. The geometries presented here are not the actual combustor geometry which will be built and tested at the $1MW_{th}$ scale, but do allow for interesting observations of the combustion behavior to be examined. This geometry, and variations of it, were used as a case study to examine many of the fundamental sizing and geometry considerations for the design of the $1MW_{th}$ combustor. In addition, three

boundary condition cases which should be considered as part of a sCO_2 oxy-fuel combustor design process were presented and explored in this study. Limited modeling results from RANs simulations were presented, some of the effects of various design choices are discussed at a high level. Some potential flaws with the simple design are discussed. For example, large differences in inlet density between the startup case and design point operation of this type of combustor pose a unique design challenge. In addition, unsafe wall temperatures can result if swirler design, cooling and fuel injection is not carefully considered. Further analysis is required to better understand these issues, and design changes would be needed to address some of these issues. It is hoped that this paper will encourage the further study of methods and novel technics to allow for successful operation at these varied conditions.

Finally, this study provides a much needed description of the geometry and boundary conditions likely to be seen in a direct fired oxy-fuel combustor for sCO_2. These boundaries are specifically relevant to the upcoming $1MW_{th}$ combustor design, but they can be scaled to larger or smaller scales as desired. It is hoped that this description will provide the basis from which other researchers can perform simulations and studies to further collaboration and progress in this exciting field.

ACKNOWLEDGEMENTS

This work was funded by the U.S. Department of Energy under DE - FE0024041.

Works Cited

S. Coogan, X. Gao, A. McClung and W. Sun, "Evaluation of Kinetic Mechanisms for Direct Fired Supercritical Oxy-Combustion of Natural Gas," in *ASME Turbo Expo 2016*, Seoul, South Korea, 2016.

S. Wenting, D. Ranjan, T. Lieuwen and S. Menon, "Investigation of Autoignition and Combustion Stability of High Pressure Supercritical Carbon Dioxide Oxy-combustion," in *UTSR*, Pittsburgh, 2017.

S. Barak, O. Pryor, J. Lopez, E. Ninnemann, S. Vasu and B. Koroglu, "High-Speed Imaging and Measurements of Ignition Delay Time in Oxy-Syngas Mixtures with High CO2 Dulition in a Shock Tube," *Journal of Engineering for Gas Turbines and Power,* vol. 139, no. Dec 2017, 2017.

Y. Iwai, M. Itoh, Y. Morisawa, S. Suzuki, D. Cusano and M. Harris, "Development Approach to the Combustor of Gas Turbine for Oxy-fuel, Supercritical CO2 Cycle," in *ASME Turbo Expo*, Montreal, 2015.

J. Delimont, A. McClung and M. Portnoff, "Direct Fired Oxy-Fuel Combustor for sCO2 Power Cycles: 1MW Scale Design and Preliminary Bench Top Testing," in *ASME Turbo Expo*, Charlotte, 2017.

T. Sasaki, M. Itoh, H. Maeda, J. Tominaga, D. Saito and Y. Niizeki, "Developmnet of Turbine and Combustor for a Semi-closed Recuperated Brayton Cycle of Supercritical Carbon Dioxide," in *ASME-ICOPE*, Charlotte, 2017.

K. Manikantachari, S. Martin, J. Bobren-Diaz and S. Vasu, "Thermal and Transport Properties for the Simulation of Direct-Fired sCO2 Combustor," *Journal of Engineering for Gas Turbines and Power,* vol. 139, 2017.

Z. Liu, X. Gao, M. Karimi, B. Ochs, V. Acharya, W. Sun, J. Delimont and N. Andrews, "DRAFT -Reduced Kinetic Models and Sensitivity to Kinetic Models in the Simulation of sCO2 Oxy-combustion," in *ASME Turbo Expo*, Oslo, 2018.

A. McClung, K. Brun and J. Delimont, "Comparison of Supercritical Carbon Dioxide Cycles for Oxy-Combustion," in *ASME Turbo Expo*, Montreal, 2015.