Coupled Modeling of a Directly Heated Tubular Solar Receiver for Supercritical Carbon Dioxide Brayton Cycle: Structural and Creep-Fatigue Evaluation

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ABSTRACT

A supercritical carbon dioxide (sCO₂) Brayton cycle is an emerging high energy-density cycle undergoing extensive research due to the appealing thermo-physical properties of sCO₂ and single phase operation. Development of a solar receiver capable of delivering sCO₂ at 20 MPa and 700°C is required for implementation of the high efficiency (~50 %) solar powered sCO₂ Brayton cycle. In this work, extensive candidate materials are reviewed along with tube size optimization using the *ASME Boiler and Pressure Vessel Code*. Temperature and pressure distribution obtained from the thermal-fluid modeling (presented in a complementary publication) are used to evaluate the thermal and mechanical stresses along with detailed creep-fatigue analysis of the tubes. The resulting body stresses were used to approximate the lifetime performance of the receiver tubes. A cyclic loading analysis is performed by coupling the Strain-Life approach and the Larson-Miller creep model. The structural integrity of the receiver was examined and it was found that the stresses can be withstood by specific tubes, determined by a parametric geometric analysis. The creep-fatigue analysis displayed the damage accumulation due to cycling and the permanent deformation on the tubes showed that the tubes can operate for the full lifetime of the receiver.

Keywords: Concentrating solar, receiver, solar thermal, structural analysis

1. NOMENCLATURE

t	minimum required thickness (m)
P	working pressure (MPa)
D_i	internal diameter (m)
E	joint efficiency factor, Young's modulus (MPa)
S	maximum allowable stress at working temperature (MPa)
α	thermal expansion coefficient (1/°C)
ν	Poisson's ratio
σ_{eff}	Effective Stress (MPa)
σ_{rr}	Radial Stress (MPa)
$\sigma_{\theta\theta}$	Tangential (Hoop) Stress (MPa)
σ_{zz}	Axial Stress (MPa)
t	long time (hrs)
T	temperature (°C)
$\beta_0, \beta_1, \beta_2, \beta_3$	Larson-Miller coefficients
σ	stress (MPa)

2. INTRODUCTION

With the need to develop cleaner and more efficient power generation technologies, researchers around the globe are trying to design components to be used in power generation installations operating at higher temperatures. The power tower technology, in which the immense quantity of the sun's high grade energy is focused on a central receiver, has received a lot of attention in the past decade [1]. Cavity and external receivers are solar receivers using tubes to absorb the highly concentrated solar energy and transmit the energy to the heat transfer fluid. In this technology, heat transfer fluids such as water, molten salt or air pass

through tubes subjected to concentrated irradiation, and get heated to high temperatures by convection heat transfer [2-5].

Another important exploration is the development of new power cycles, which exploit the favorable thermo-physical properties of thermic fluids. The current power generation cycles which use supercritical/superheated steam give a maximum thermal efficiency ~30-40% [6]. An emerging candidate to improve this efficiency has been the closed-loop supercritical carbon dioxide (sCO₂) Brayton cycles, which has been evaluated to be much more efficient, and at lower temperatures than the conventional steam Rankine cycle [7-11]. Power cycle efficiency ~50% at temperatures which can be obtained by concentrated solar power has made the CSP based sCO₂ cycle a viable option in emerging CSP and power cycle technologies [7-13]. Carbon dioxide (CO₂) with its superb thermo-physical properties, moderate critical pressure, non-toxicity, chemical stability, abundance and low cost has been considered as a heat transfer fluid for nuclear and CSP applications [9, 14, 15]. The structural design of the tubular receivers for sCO₂ for ensuring safety and structural integrity is an extremely challenging task. An analytical methodology to approximate the stress distributions throughout the tube was presented by Neises et al. [1].

In this paper, structural design with creep-fatigue analyses of a tubular sCO₂ receiver subjected to high pressure and thermal stresses is evaluated for a lifecycle requirement of 10,000 cycles. The resulting tube stresses are used to evaluate the lifetime performance of the receiver tubes. The Strain-Life approach and the polynomial form of the Larson-Miller creep model are coupled in an in-house developed MATLAB code to study the cyclic loading and estimate the damage to the tubes in order to maintain safety and integrity of the receiver.

3. METHODOLOGY

The ASME Boiler & Pressure Vessel Code (BPVC) provides the rules for the design, fabrication, and maintenance of fired and unfired pressure vessels [16-17]. It also provides a

wide range of methods for high temperature and high pressure applications, the design criteria focuses mainly on traditional (e.g. coal-fired) boilers and super-heaters, which are related, but not similar to CSP receivers [16].

3.1 Material Selection

The ASME BPVC Section II Part D provides the maximum allowable stress levels at a constant temperature. Nevertheless, these values correspond to 80% of the minimum creep rupture stress at 100,000 hours. A safety factor of 1.25 is applied to all pressurized vessels. For a working temperature range of 700-800 °C, Inconel 625 was selected for this analysis, even though it does not show the highest allowable stresses for the desired temperature range, but the availability of the material makes it very suitable for the application . Figure 1 shows a comparison of Inconel 625 to other high-strength nickel alloys that could be used for solar power applications.

Figure 1: Maximum allowable stresses as a function of temperature. These values correspond to the 80% of the minimum creep rupture stress at 100,000 hours [17].

Dostal et al. describe the possible s-CO₂ conditions which can potentially yield a cycle efficiency ~50% [12]. An operating pressure of 20 MPa and an outlet temperature of 700 °C are required from the receiver to achieve the prescribed cycle efficiency.

Since the thermal distribution on the receiver is not uniform, the temperature distribution on the tubes was determined from a contiguous work which describes the results of the computational fluid dynamics analysis. For this work, the “worst” and “best” cases were selected; the details will be presented in the sections which follow.

3.2 Tube Selection

Tube size and wall thickness were selected to maximize heat transfer while minimizing pumping losses. The internal heat transfer coefficient scales as the inverse of the diameter, as the Reynolds number is higher for smaller diameters and convection heat transfer increases with velocity, for a given flow, making small diameters attractive for convective heat transfer. Equation 1 is *ASME BPVC Section 8* design equation for pressurized tubes and pipes, and it was used to select the optimal tube thickness and outside diameter.

$$t = \frac{P \cdot D_i}{2(S \cdot E - 0.6 \cdot P)} \quad (1)$$

Where t is the minimum thickness required, excluding manufacturing tolerance and allowances for corrosion, P is the working pressure, D_i is the internal diameter, S is the maximum allowable stress at working temperature, and E is the joint efficiency factor. For seamless tubes $E=1$. Figure 2 shows the required wall thickness for a given outside diameter at isothermal tube temperature 800 °C with 20 MPa of internal pressure.

For analytical calculations, ~800 °C will be the maximum temperature allowed, since the maximum allowable stress is on the order of 30 MPa. The diameter chosen was the smallest in order to have smallest wall thickness and reduce the thermal stresses across the tubes. For a 12.7 mm (0.5”) outer diameter tube, 1.915 mm wall thickness was estimated and

the nearest standard available tube wall thickness is 2.1082 mm (0.083''). These dimensions will be used for the rest of the analysis.

Using the same approach, the outside diameter for the headers was estimated to be 168.3 mm (6.625'') with a wall thickness of 13.408 mm (0.528'') for an operating temperature of 700°C. The pipe selected for the manifold was 6'' SCH. 160.

Figure 2: Minimum required wall thickness depending on the outside diameter. The estimated values are calculated with equation 1 and the standard values are commercially available tube sizes.

Furthermore, the tubes used in CSP receiver tubes are subjected to non-uniform temperature gradients radially, tangentially and axially. At steady-state, the heat transfer rate and the thermally-induced stresses across the tube-walls are proportional to the wall-temperature difference [1]. This non-axisymmetric irradiance produces non-uniform wall temperatures which are common to CSP, but not to any other high-temperature applications. There are three sections in the ASME BPVC which are capable of handling design at higher temperatures. The three main drawbacks of using the code exclusively and without any modifications inclined to CSP applications are:

- 1   1. Although *Section I* considers the design of power boilers and super-heaters, it is
2   mainly design for power plants which typically are convectively heated by flue gas at
3   relatively low rates of thermal flux.
- 4   2. The large safety requirements developed for nuclear components in *Section III*,
5   *Division I – Subsection NH* will require further simplifications since the level of
6   conservatism in the creep-fatigue analyses is not necessary for CSP applications [17].
- 7   3. *Section VIII-Division I* does not consider high-cycle fatigue [17] and therefore the
8   Design-by-Rule methodology does not capture the complete analysis required for
9   solar thermal receivers.

10   Nonetheless, *Section VIII-Division II* is a secondary set of guidelines which can be more
11   appropriate for CSP design because they consider a Design-by- Analysis methodology.

12   Aside from analysis required to design for the resulting stresses due to the thermal and
13   pressure loads, the tubular receivers experience fatigue and creep damage accumulation
14   which will produce failure. Since CSP receivers operate diurnally, they experience a
15   significant cyclical damage exposure which accumulates fatigue on the tubes. Additionally,
16   the receivers will operate at temperatures where creep damage becomes relevant in design.

17   In order to account for this accumulated damage, a set of simplified design rules
18   based on the nuclear code were developed for CSP receivers and were documented in an
19   interim design standard for solar energy applications [16]. This approach simplifies the
20   design methodology for a creep-fatigue analysis with a cumulative damage approach [16].

$$\sum_{j=1}^p \left(\frac{n}{N_d}\right)_j + \sum_{k=1}^q \left(\frac{t}{T_d}\right)_k \leq D \quad (2)$$

22   The creep-fatigue damage is given by equation 2 for p number of unique loading
23   cycles, and q number of unique creep loads, where N_d is the number of allowable and n is the
24   number of applied cycles at known loading cycle j , T_d is the allowable creep rupture time and

t is the applied load time at loading condition k . Grossman et al. [19] highlights that this creep-fatigue analysis is based entirely on test data and not on the specific processes leading to creep-fatigue failure. Therefore, D , the total allowable accumulated damage is a material property and varies between alloys. Literature suggests that for Inconel 625, $D \approx 1.0$ [20]. For this work, 10,000 diurnal cycles have been investigated. As mentioned by Neises et al. this is nearly equivalent to 240,000 hours (~30 years) of service life [1]. Due to the high complexity of the cyclic loads in the receiver, approximately 100,000 hours are assumed to be operational time and 140,000 hours are non-operational time. Starting and shutting down times are considered transient periods, which require more detailed analyses and are out of the scope of this present work.

4. FINITE ELEMENT MODELING

The procedure presented by Neises et al. [1] focuses on the development of an analytical model using the pressurized cylindrical vessel equations. The calculated resulting radial, tangential and axial stresses throughout the tube account for the pressure and thermally induced stresses. The results obtained from the analytical models were used to build a finite element analysis (FEA) structural model using ANSYS Structural.

4.1 Static Structural Model

A static structural stress model was developed in order to understand the stress distributions that the tube experiences under the loads described in the next subsection.

4.1.1 Boundary Conditions

The boundary conditions described in Table 1 are initially used in the static structural model and are applied to the tube as shown in Figure 3. The thermal distribution was obtained from ANSYS Fluent using constant temperature inner and outer walls. Both ends of the tube were allowed to freely expand by specifying weak springs boundary condition.

Table 1: Parameters used as for 20 MPa and 700°C outlet pressure and temperature.

Parameter	Value (Units)
O.D./Thickness	12.7/2.1082 (mm)
I.D./O.D. Temperatures	700/720°C
Internal Pressures	20 (MPa)
E (Young's Modulus)	163 x10 ³ (MPa)
α (thermal expansion coefficient)	18.6 x10 ⁻⁶ (1/°C)
ν (Poisson's ratio)	0.322 (-)

Figure 3: Boundary conditions applied for 20 MPa and 20 K radial temperature gradient.

4.1.2 Mesh Independence

A mesh independence analysis was performed using the boundary conditions established before. The goal of the mesh independence analysis was to obtain a percent difference below 1% between the static structural model and the analytical stress levels. For a 1 m long tube with an O.D. of 12.7 mm and wall thickness of 2.1082 mm, about 67,000 elements (Figure 4) were required to yield grid independence.

Figure 4: Mesh used for the FEA structural modeling

4.1.3 Model Validation

Figures 5-8 show comparison of the stress gradients across the tube wall due to the applied pressure and thermal loads. As required, the modeled stresses must correspond to the analytically calculated stresses using the method presented by Neises et al. since it is the only method to validate the more complex model presented in the following section. As previously identified by Neises et al., the tensile stresses are more dominant in the inner part of the tube.

Resulting Stresses due to Internal Pressure

Figure 5: The resulting stresses due to the internal pressure of 20MPa only.

Figure 6: The modeled stresses due to the internal pressure of 20MPa.

Resulting Stresses due to Combined Loading

Figure 7: The resulting stresses due to the internal pressure of 20MPa and 20 K induced thermal load.

Figure 8: The modeled stresses due to the internal pressure of 20MPa and 20 °C induced thermal load.

Analytical and modeled stress levels are comparable in magnitude and follow the same type of distribution throughout the tube. As a result, a more representative thermal-structural model can be performed by coupling a non-axisymmetric temperature distribution more representative from a tubular receiver, which is the ultimate goal of this work.

4.1.4 Fatigue and Creep Accumulated Damage

Using the stress values obtained from the models and Figure 9, the number of cycles to failure can be estimated. Although the number of cycles to failure is high, Neises et al. suggest the value of the fatigue accumulated damage to be 0.1, as an added safety factor [1].

Figure 9: Stress Level and Temperature vs Cycles to Failure of Inconel 625 [20].

The Larson-Miller model is a method to extrapolate the short-time creep-rupture data to predict long-time life as a function of temperature and stress level [21]. The method expresses the logarithmic long-time, $\log(t)$, as a function of the reciprocal of the absolute temperature T and logarithmic stress, $\log(\sigma)$ as shown in equation 3.

$$\log(t) = \beta_0 + \beta_1 \frac{1}{T+273.15} + \beta_2 \frac{1}{T+273.15} \log(\sigma) + \beta_3 \frac{1}{T+273.15} \log(\sigma)^2 \quad (3)$$

The Larson-Miller coefficients were obtained from the curve fit generated from the ASME BPVC stress tables. Two methods for Larson-Miller were used to fit the data, the linear model assumes $\beta_3 = 0$, while the polynomial model is a quadratic fit on the data. Figure 10 shows how the quadratic model can provide a better fit to the ASME BPVC stress values [21].

Figure 10: Creep models and ASME BPVC stress levels [17].

Following the same procedure described by Ortega et al. [22], the creep damage accumulation model can be performed using the temperature and stress levels at every node on the grid previously used. This methodology can be applied for the static structural analysis

and incorporating the creep-fatigue damage accumulation models, a non-uniform thermal-structural model was developed to best represent the thermal load on the receiver tubes.

4.2 Static Thermal-Structural Model

A static thermal-structural analysis is a type of finite element analysis (FEA) that couples the thermal solution or temperature distribution and numerically approximates the resulting stress distributions throughout a designed component. These stress levels are used to estimate the creep-fatigue accumulated damage using the same methodology used by Neises et al [1]. The contiguous work containing the results of a coupled optical-thermal-fluid modeling analysis performed using ANSYS Fluent provides the thermal distribution to be used for this analysis of the tubes of receiver [23].

4.2.1 Boundary Conditions

While the internal pressure was kept constant at 20 MPa, the temperature distribution was obtained from a typical flow configuration with an input power of 500 kW described in the contiguous work [23]. The coupling of the optical-thermal-fluid temperature distribution throughout the tube using ANSYS Fluent with ANSYS Structural provides the non-uniform temperatures on the tube for this analysis. Figure 11 presents one of the tubes with the highest surface temperatures from the case under consideration.

Figure 11: Non-uniform temperature distribution on tube's surface.

4.2.2 Stress and Strain Distributions

Figures 12 and 13 show the stress and strain distributions which will be used to approximate the fatigue-creep accumulated damage. As expected, the highest stresses are located on the inner walls of the tube, which is the area of interest.

Figure 12: Strain distribution on the tube at the center of the tube.

Figure 13: Stress distribution on the tube at the center of the tube.

4.2.3 Fatigue and Creep Accumulated Damage

The fatigue and creep damage accumulated in the tube under a non-axisymmetric heat flux load was estimated using the nodal-analysis quadratic Larson-Miller creep model. The maximum creep damage is taken as 0.9, since the fatigue damage was fixed initially at 0.1 [1]. Table 2 shows the results of the nodal-analysis. At this point, the MATLAB code does not produce a graphical output to visualize the life and damage accumulation of the analyzed structure. Nonetheless, it was found that in the first iteration of the Design-by Analysis method, about 86% of the nodes met the required time to rupture. It should be noted that the most efficient case from the contiguous work was analyzed for structural integrity [23]. The meaning that 86% of the nodes meet the requirement is that the receiver life will be limited in the number of hours that the tubes can be used. As expected, the weakest region on the tubes is the internal part of the tube on the side where the irradiance is applied. Although the design criterion for 100,000 hours of operation was not met fully, the flexibility of the methodology

allows for a change in the power input levels to reduce the wall temperatures or increase the tube's wall thickness.

Table 2: Results of Quadratic Larson-Miller Creep Model

Maximum Creep Damage	Number of Nodes in the analysis	Required time to rupture (hrs.)	Number of Nodes that met the required time to rupture	Percent of Nodes that met requirement (%)
0.9	319248	>111,000	275239	86.2%

5. CONCLUSIONS

The feasibility of a high temperature and high pressure supercritical carbon dioxide receiver has been confirmed. Although the Design-by-Analysis methodology requires an iterative process, this methodology will allow the designer to determine the best operating parameters.

A comprehensive thermal-structural FEA is presented. The method employed allows the designer to focus on the weaker sections of the receiver, and adjust the geometry and physical conditions to the temperature and stress levels which the receiver will be subjected to. The results for a specific case show that only 86% of the nodes surpass the required lifetime, nonetheless, there are plenty of ways to achieve the optimal temperature and stress distribution. It is important to note that based on the *ASME BPVC Section 8*, the membrane stresses do not exceed the maximum operating stresses specified. This suggests that creep-fatigue is the primary failure mode. The following suggested routes will be applied in the future work to assure that 100% of the nodes in the model meet the life requirements.

1. Reducing the heat input to the receiver and limiting the heat flux levels will yield lower surface temperature gradients, thus, lower thermal stresses. This would benefit the life-long structural integrity of the receiver, but can compromise thermal performance of the receiver.

2. Increasing the wall thickness would help reduce the stresses due to internal pressure, but increase the thermal resistance across the wall, negatively impacting the thermal efficiency and increasing the thermal stresses. Also, as the thermal resistance increases with the thickness of tube, the temperature gradients in the tube wall increase, resulting in a higher outer temperature and thermal losses.

3. Investigating the behavior with stronger materials. Although Inconel 625 is a high strength nickel alloy which is excellent for this application, newer alloys will be developed and they could offer more strength at these high temperatures and pressures.

The Design-by-Analysis methodology helps to reliably estimate the life of the new generation of solar thermal receiver and serve as reference for further design and evaluation of future direct and indirect tubular receivers.

Lastly, although this procedure focused on the design of a solar thermal receiver, this methodology is geometry independent, so any type of structures could be analyzed by applying the appropriate boundary conditions and constraining the model appropriately.

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