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Research Paper

# 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 ($sCO_2$) Brayton cycle is an emerging high energy-density cycle undergoing extensive research due to the appealing thermo-physical properties of $sCO_2$ and single phase operation. Development of a solar receiver capable of delivering $sCO_2$ at 20 MPa and 700 °C is required for implementation of the high efficiency (~50%) solar powered $sCO_2$ Brayton cycle. In this work, extensive candidate materials are review 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.



## 1. 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 ($sCO_2$) 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 $sCO_2$ cycle a viable option in emerging CSP and power cycle technologies [7–13]. Carbon dioxide ($CO_2$) 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 $sCO_2$ 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 $sCO_2$ 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

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**Nomenclature**

| | | | |
|---|---|---|---|
| $t$ | minimum required thickness (m) | $\sigma_{eff}$ | effective stress (MPa) |
| $P$ | working pressure (MPa) | $\sigma_{rr}$ | radial stress (MPa) |
| $D_i$ | internal diameter (m) | $\sigma_{\theta\theta}$ | tangential (Hoop) stress (MPa) |
| $E$ | joint efficiency factor, Young's modulus (MPa) | $\sigma_{zz}$ | axial stress (MPa) |
| $S$ | maximum allowable stress at working temperature (MPa) | t | long time (h) |
| | | T | temperature (°C) |
| $\alpha$ | thermal expansion coefficient (1/°C) | $\beta_0, \beta_1, \beta_2, \beta_3$ | Larson-Miller coefficients |
| $\nu$ | Poisson's ratio | $\sigma$ | stress (MPa) |

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.

## 2. 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].

### 2.1. 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 h. 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. Fig. 1 shows a comparison of Inconel 625 to other high-strength nickel alloys that could be used for solar power applications.

Dostal et al. describe the possible s-$CO_2$ 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.

### 2.2. 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. Eq. (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.

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

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 $E = 1$. Fig. 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

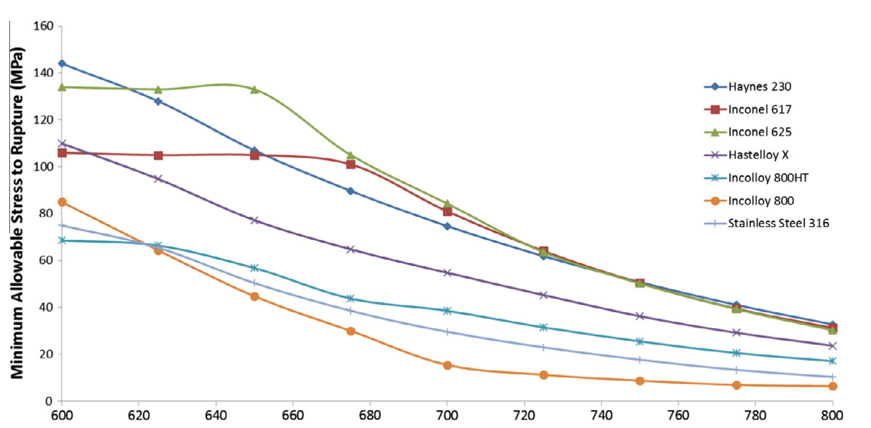


**Fig. 1.** Maximum allowable stresses as a function of temperature. These values correspond to the 80% of the minimum creep rupture stress at 100,000 h [17].

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