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Reliability analysis of Solar-Gas Hybrid Receivers for central tower plants

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Abstract

A novel Solar-Gas Hybrid Receiver (SGHR) that combines the function of a solar receiver and a gas boiler in a single device is presented. This concept requires less equipment and maintenance compared to the Solar Gas Hybrid (SGH) concept, in which the boiler and the solar receiver (SR) are independent devices. The economic benefit is attributed to the increased sharing of infrastructures. Additionally, it has less thermal stress, cycles and shocks, which reduce the failure risk. However, the additional benefit in the reliability of these receivers has not been analyzed so far. In this work, a mathematical model of SGHR is presented. It determines the stress in steady state which is used to estimate the allowable transient stress in order to achieve the required 30 years life design. The results show that the SGHR is exposed to lower thermal stress due to much better temperature distribution. Moreover the higher absorber heat flux of SGHR is translated in a higher mechanical stress with could jeopardizes the durability. However the reduced number of cycles and the lower thermal stress of a SGHR allows higher transient stresses than the conventional tube type solar receiver, which lead to more reliable and efficient designs.

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Keywords: hybrid, receiver, solar, Gas, reliability, Molten Salts.

1. Introduction

The large amount of solar energy that reaches the earth is high enough to provide all of the world's energy demands. The use of Central Receiver System (CRS) to transform this energy into electricity is a good initiative to

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reduce the fossil fuels dependence and at the same time reduce the CO2 emission. However, only a small amount of the world's energy comes directly from CRS, mainly due to the high initial investment and the technological risk. On one hand, the foremost cost is associated to the large heliostat field required to collect the solar energy (1). On the other hand, the technological risk is mainly associated to the reliability of the receiver and to the variability of the solar radiation throughout the day, which means the requirement of a storage system to avoid the intermittence and non-dispatchability of power production.

In this work, hybrid receivers Solar- biomass gases are presented as a potential solution to overcoming the cost, reliability and intermittency issues. Both energy sources suffer from high costs. However, their hybridization is interesting due to its complementary nature. On one hand, solar thermal energy has high costs due to the intermittent nature but the fuel itself has zero costs. On the other hand, the biomass gasses supply is not intermittent but the fuel costs are high. Therefore, the hybridization of solar and Biogas fuels allows to increase the dispatchability of only solar power plants and decrease the costs due to the sharing of infrastructures. Moreover, this concept can be extended to cheaper fossil fuels in order to achieve a more economic energy supply. Therefore from now on will be called Solar-Gas hybridization.

Many Solar-Gas hybridization systems have been proposed by several authors and reviewed in (2). An exhaustive review of the possible integration systems has been done in a previous project (3) and it has been selected the one with the higher potential to overcome the cost, the reliability and intermittence issues. The selected integration system is shown in Fig.1. The function of a tube type solar receiver (TTR) and a gas boiler are combined in a single device, henceforth called Solar-Gas Hybrid Solar Receiver (SGHR).The combustion of the gas is carried out at the SGHR simultaneously or consecutively with the solar radiation. Its heat is used to increase the thermal power of the Heat Transfer Fluid (HTF) and to avoid the solar radiation dependence. Then, the heat exchanger (HE) transfers the thermal power of the HTF to the steam of the Power Block (PB). Thus the integration of the gas boiler and the solar receiver into a single device decrease the amount of equipment required to achieve dispatchable energy supply without the need of energy storage. Moreover, the operation of the combustion system is independent on the radiant energy availability, which prevents SGHR from thermal shocks due to solar radiation intermittence. Furthermore, the homogenous temperature reached at the receiver avoids fast degradation of the SGHR, increasing the reliability of the plant.

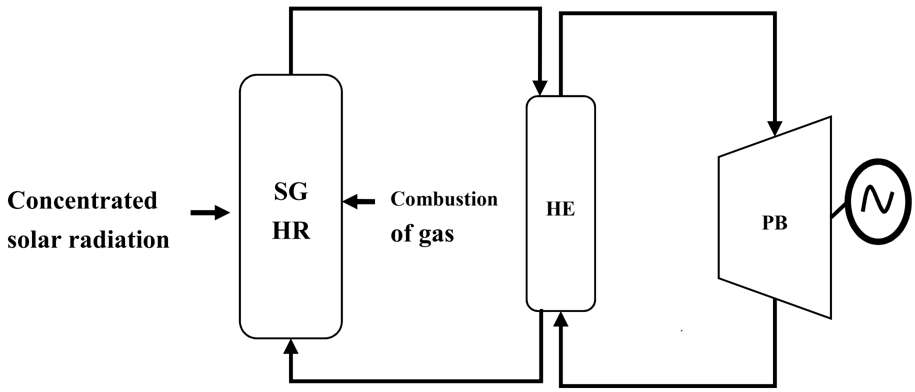

Fig. 1.Integration system of an HRC.

It is important to highlight that the tubes, which compose the Heat Collector Element (HCE), of a conventional TTR work not only at high thermal and mechanical stress but also endure daily temperature incursion from high operation temperature to ambient temperature. These severe conditions activate degradation mechanisms such as: low cycle fatigue, oxidation at high temperatures and creep (4).

Although the economic benefits of a similar concept using natural gas instead of biomass gasses were proved in (1), the additional benefit of increased reliability due to reduced thermal cycling and thermal stress has not been analyzed yet and it is what in this paper is presented. The constructive details of the novel SGHR are described in detail and a comparison of its reliability by means of, steady state thermal stress, mechanical stress and allowable transients stress is carried out with respect to the well-known TTR, under similar conditions. For this purpose, a thermo hydraulic model of the HCE for both devices is presented. This mathematical model obtains the thermal and mechanical stress in steady state operation firstly. Then, these results are used to estimate the allowable transient stress in order to achieve the required 30 year life design.

Nomenclature

Q	Power (W)
W	Work (W)
h	Enthalpy (J/Kg)
V	Velocity (m/s)
g	Gravity (m/s^2)
A	Area (m^2)
Comb	Combustion
t	tube
conv	Convection
rad	Radiation
Amb	Ambient losses
Sky	Radiation to the Sky
Sun	Sun radiation
m	Mass flows (Kg/s)
Cp	Heat capacity (KJ/KgºC)
ϕ	diameter (m)
r	radius (m)
in	Inner
out	Outer
T	Temperature (ºC)
ν	Poisson ratio
E	Modulus of elasticity (N/mm^2)
α	Thermal expansion coefficient (10^{-6}mm/mm/°C)
σ	Stress (N/mm^2)
P	pressure (N/mm^2)

2. Concept description

The presented SGHR is composed by a cavity, a billboard TTR, molten salts as HTF, two combustion chambers (in front of and behind the TTR), a fume exhaust duct, and a mobile enclosure (as seen in Fig. 2). This disposition allows heat to transfer to the tubes, the HCE, from both sides, reducing the thermal stress produced by circumferential thermal gradient. On one hand, the front face of the tubes is heated by solar radiation and combustion consecutively in this way: when the direct normal irradiance (DNI) is high enough, the receiver's front face is heated by the solar field, and when DNI is not enough to maintain the operation, the mobile enclosure closes the cavity and the tubes' front face is heated by the combustion chamber. On the other hand, the back face of the tubes is heated by gas combustion continuously. Such combustion is carried out in radiant burners that allow the spatial distribution of the radiant flux to vary to the required level. Finally, the combustion gases are addressed to a heat exchanger in order to recover its heat.

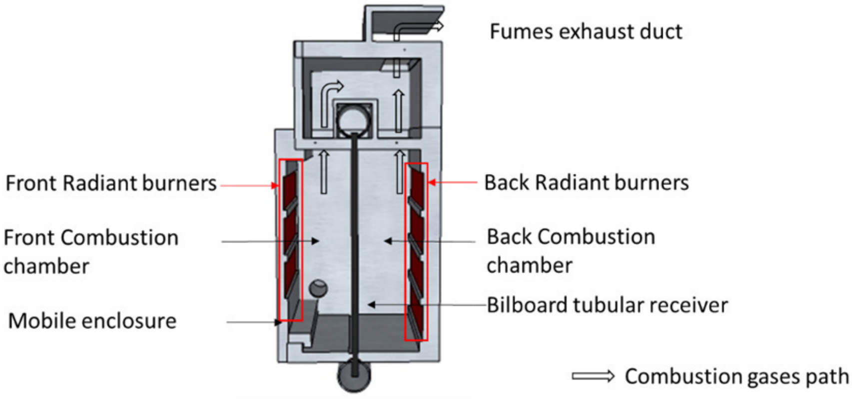

Fig. 2: SGHR components representation.

Although any of HTF usually used in TTR could be used in a SGHR, molten salts have been selected in this study due to the complete data base available in literature.

2.1. Mathematical model

A mathematical model has been developed in order to evaluate the SGHR reliability in comparison with the conventional TTR. For this purpose, the thermal and mechanical stress in steady state and the allowable transient stress for a given number of cycles have been determined. Firstly, the temperature gradient and pressure distribution are obtained. Secondly, the thermal and mechanical stresses in steady state operation are determined. And finally, the allowable transient stress for the predicted number of cycles is calculated using a life estimation method.

The temperature gradient is obtained by a steady state modeling approach based on an energy balance on the HCE. It includes the combustion heat and/or concentrated solar radiation, the optical and the thermal losses from the HCE and the thermal gains in the HTF. The thermal model has been solved using the Finite Volume method (FVM) to discretize the domain and the energy conservation is applied at each control volume (CV). As the temperature and pressure of the HTF varies only in longitudinal direction, the HTF has been divided into N_zCVs of equal length with temperature and pressure continuity at the boundaries (see Fig. 3). On the other hand, the metal temperature and heat flux varies along circumferential and longitudinal direction. Consequently the tube has been divided into N_zCVs in the axial direction and N_θCVs in the azimuthal direction (see Fig. 3). The energy balance equation is determined by Eq. (1) at each CV:

$$\dot{Q} - \dot{W} = \int_{cs} (h + \frac{v^2}{2} + gz)(\rho \vec{v} \overrightarrow{dA}) + \frac{\partial}{\partial t} \int_{cv} \left(u + \frac{v^2}{2} + gz \right) (\rho dv) \tag{1}$$

For a CV of the tube, the energy balance equation is given by Eq. 2. The metal properties dependence on the temperature was updated while iteratively solving the problem.

$$\sum_{cv}\dot{q}^{ij} = \dot{q}^{ij}{}_{comb-t,conv} + \dot{q}^{ij}_{comb-t,rad} + \dot{q}^{ij}_{sun-t,rad} - \dot{q}^{ij}_{t-Amb,conv.} - \dot{q}^{ij}_{t-Sky,rad} - \dot{q}^{ij}_{t-HTF,conv.} = 0 \tag{2}$$

For a HTF CV‘" of length L, the energy balance can be expressed as given in Eq 3.The transferred heat ($\dot{q}^{i}_{t-htf,conv}$) to the HTF by the tube CVs has been summed up along the circumference of each ‘i’ ring using the area and temperature of the according ‘j’ element. The fluid temperature, the temperature dependent properties and the heat transfer coefficient of the next HTF CV was calculated using the transferred heat of the previous HTF CV. Fully developed flow and heat transfer are considered. Unlike in the former analysis of the solar receiver tubes, anisotropy of turbulent energy transport has been taken into account by employing theoretical results from the Eddy diffusivity in the different directions which are in satisfactory agreement with experimental data (5).

$$\dot{q}^{i}_{t-HTF,conv} = \dot{m}\left[h^{i}_{HTF} - h^{i+1}_{HTF}\right] = \dot{m}Cp\left[T^{i}_{HTF} - T^{i+1}_{HTF}\right] \tag{3}$$

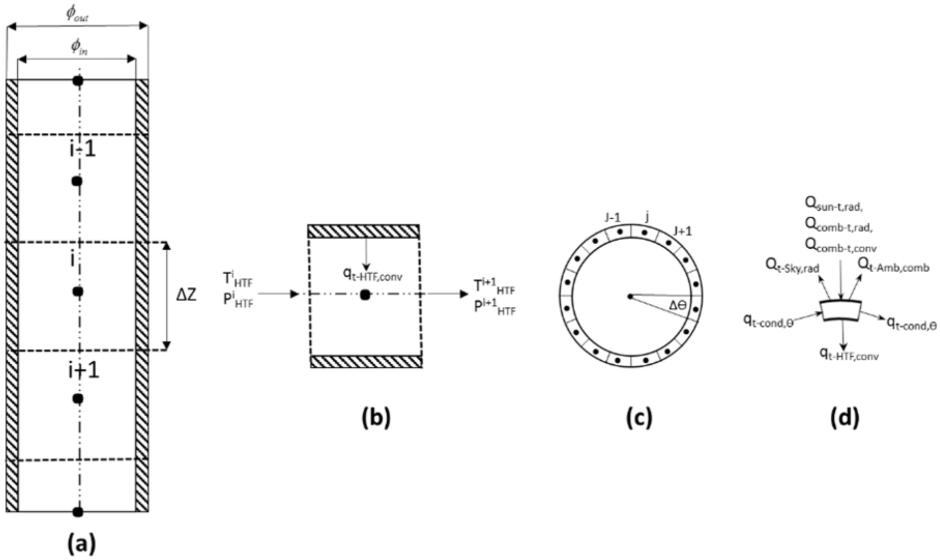

Fig. 3.Discretizitation and energy balance of the HCE: a) Longitudinal discretization, b) energy balance at HTF CV, c) azimuthal discretization, d) energy balance at the tube CV.

The circumferential heat flux along the tube wall can be considered negligible and therefore the outer and inner surface temperature of each metal CV can be calculated as:

$$T^{ij}_{in} = T^{i}_{HTF} + \dot{q}^{ij}_{t-HTF,conv.}\frac{\phi_{out}}{\phi_{in}}\frac{1}{h} \quad (4) \qquad T^{ij}_{out}(\theta) = T^{ij}_{in} + \dot{q}^{ij}_{t-HTF,conv.}\frac{\phi_{out}}{2K}\ln\left(\frac{\phi_{out}}{\phi_{in}}\right) \quad (5)$$

The stress due to the HCE temperature difference in axial direction can be relieved by carefully designed strain relief structures. However, the stress due to the cross-wall temperature difference is difficult to avoid since it is the driving force for the heat transfer. In consequence the thermal stress is analyzed as plane-radial stress problem. Thermal stress in the HCE appears due to the temperature difference across the tube thickness and the circumferential tube temperature difference. In absent of stress relaxation, the temperature gradient in the tube wall makes the outside to expand more than the inside which places the outside under compression and the inside under tension. This balanced stress gradient is biaxial in the z (axial) and ϴ (tangent) directions. The maximum thermal gradient stress values occur at the surfaces and are expressed as:

$$\sigma^{ij}_{th,\max,z} = \sigma^{ij}_{th,\max,\theta} = E\alpha\left[\frac{T^{ij}_{out} - T^{ij}_{in}}{2(1-\nu)} + \left(\frac{T^{ij}_{out} - T^{ij}_{in}}{2} - T_{avg}\right)\right] \quad (6)$$

The pressure variation along the segment "i" is calculated as the sum of friction losses, vertical height difference and pressure change due to acceleration of the fluid as shown in Eq 8.

$$\Delta P^i = \underbrace{(\frac{f_{fric}\Delta z}{\phi_{in}})G^2/(2\rho^i)}_{\Delta P_{fr}} + \underbrace{\rho^i \; gsen(\alpha)\Delta z}_{\Delta P_{gz}} + \underbrace{G^2(1/\rho^i\rho^{i-1})}_{\Delta P_{ax}} \quad (7)$$

Where f_{fric} is the Darcy friction factor, and can be estimated for a turbulent pipe flow with the Colebrook equation. Tube type receivers are composed of several tubes arranged in parallel, with the total head loss across the system being constant. However, due to the heterogeneity of the solar flux distribution, each tube receives different amounts of heat being the outlet temperature and pressure of each tube different for the same inlet conditions. Therefore, the mass flow rate at the inlet of the tube has been adapted for each tube with an iterative algorithm in order to achieve constant head loss. Mechanical stress induced in the tube is directly proportional to the operating pressure of the HTF and it is calculated from:

$$\sigma_{P,z} = \frac{P^i_{in}\phi_{in} - P^i_{out}\phi_{out}}{\phi_{out} - \phi_{in}} \quad (8)$$

$$\sigma_{P,\theta} = \sigma_{P,z} + \frac{\phi_{in}{}^2\phi_{out}{}^2\left(P^i_{in} - P^i_{out}\right)}{r^2(\phi_{in}{}^2 - \phi_{out}{}^2)} \quad (9)$$

$$\sigma_{P,r} = \sigma_{P,z} + \frac{\phi_{in}{}^2\phi_{out}{}^2\left(P^i_{in} - P^i_{out}\right)}{r^2(\phi_{in}{}^2 - \phi_{out}{}^2)} \quad (10)$$

Finally, the allowable transient stress for a required 30 year life design has been determined with a well-established life prediction model based in UNE-EN 12952-3:2001 norm, which considers the degradation due to creep, thermo-mechanical fatigue, and oxidation at high temperature. The transient stress is obtained subtracting the steady state stress to the fatigue strength for the estimate number of cycles.Fatigue, oxidation and creep resistance at high temperature are available from several sources. In this work the durability limits established by the UNE-EN 12952-3:2001 norm have been used. The mentioned norm is applicable for high temperature pressure vessels up to 800ºC. Moreover, in order to consider the effect of three mechanisms, a damage accumulation method has been used.

In order to solve the mathematical model, the presented equations have been implemented within the Visual Basic environment. The inputs required in the model are: HTF inlet pressure, temperature and mass flow rates, inside and outside diameter tube, tube's material, number of cycles and heat flux spatial distribution. The same HTF inlet temperature has been considered for both, the TTR and the SGHR. However the number of cycles and heat flux spatial distribution are considered different due to the specific operation condition of each compared element.

Regarding the operation conditions of both receivers, it is described as follows. In the case of the TTR: during a sunny day the Direct Normal Irradiance (DNI) is concentrated by the solar field in the HCE front face while the back face is not illuminated. The non-uniform radiation distribution along the outside HCE is shown in Fig. 4 (a). The longitudinal distribution has been obtained with WinDelsol software considering one aiming point strategy and maximum radiation peak of 1200 kW/m^2. A small amount of the concentrated sunlight is reflected by HCE surface and the high absorptivity coating absorbs the remaining. A part of the gained heat is transferred to the HTF by forced convection, across the wall thickness by conduction and the other part is lost by natural convection and thermal radiation to the ambient in the front face while the back face remains adiabatic. A TTR must operate cyclically due to its nature. It is heated to reach the operation temperature and then it is allowed to relax up to the thermal equilibrium at least daily. In addition, the TTR is cooled due to clouds and heated again several times per day. 36.000 cycles have been considered for a design life of up to 30 years.

In the case of the SGHR: during a sunny day the DNI is concentrated by the solar field in the HCE front face. At the same time, the heat produced in the combustion chamber reaches the HCE back face by radiation. The radiation circumferential distribution is shown in Fig. 4 (b). A small amount of the concentrated sunlight is reflected by HCE surface and the remaining is absorbed by the high absorptivity coating. A part of the gained heat is transferred to the HTF by forced convection, across the wall thickness by conduction and the other part is lost by natural convection and thermal radiation to the ambient in the front face. When the DNI is not high enough to maintain the operation the mobile enclosure closes the cavity, and the tubes front face are heated by the front radiant burners. Therefore, the only thermal cycles of the SGHR will be during the maintenance and technical shut-downs of the plant. In this case, 60 cycles have been considered.

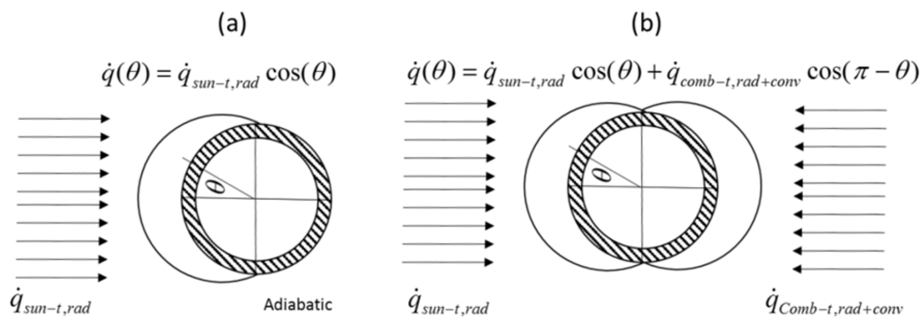

Fig. 4. Circumferential heat gain: a) TTR. b) SGHR

3. Results

The TTR and SGHR reliability has been compared by means of steady state operation stress and the allowable transient stress for a design life of 30 years. The HCE of the TTR and SGHR are composed by 10 meter longs panels arranged in series. The panels are composed by austenitic stainless steel, BS3059 grade CFS1250 tubes arranged in parallel with 25mm outside diameter and 2 mm of wall thickness that is in the usual range of TTR (6). The molten salts temperature at the inlet of the first panel has been fixed in 300 ºC and the mass flow rate and inlet pressure have been adapted in order to achieve 565ºC and 5 bar of pressure at the outlet of the 2nd, 3rd or 5nd panel. Fig. 5 shows the temperature evolution of HTF CV along the 2, 3 or 5 panels. As can be seen, the temperature evolution of TTR and SGHR are superposed for the 3 cases. However, the mass flow rate of the SGHR is almost the double in all the cases due to the fact that heat gain is twice than of TTRs. Therefore for a same panel of tubes the SGHR produces the double of thermal energy.

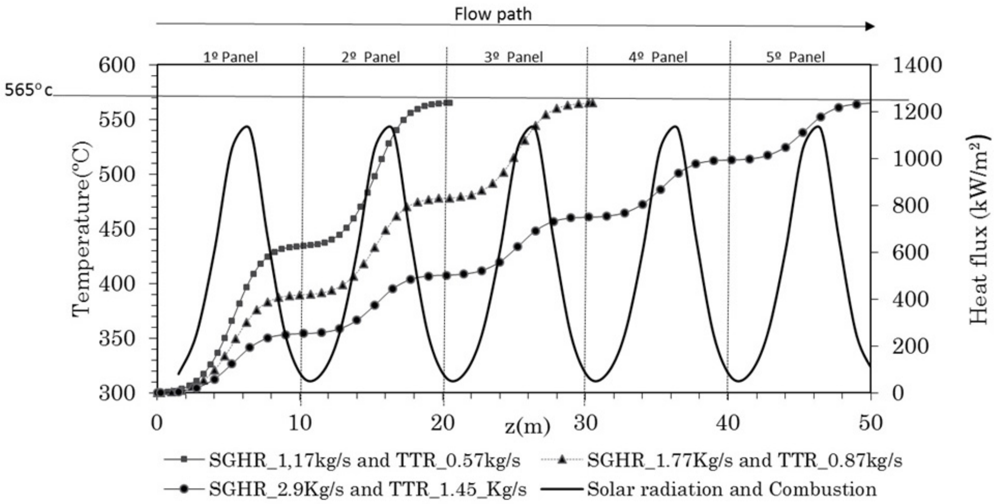

Fig 5. HTF CV temperature distribution along the TTR and SGHR for 2, 3 and 5 panels.

Nevertheless, a higher mass flow rate for the same tube diameter results in higher pressure drops. The inlet pressure must be higher for SGHR than for the TTR as the outlet pressure has been define as a constant value of 5 bars. High inlet pressure can produce significantly high mechanical stress.

Fig. 6 shows the distribution of mechanical stress produced by HTF pressure in the outer crown of the tube. As consequence of the high flow rate required to absorb the heat flux, SGHR withstands higher mechanical stress and therefore its reliability can be jeopardized. Note that in order to reach 5 bar in the outlet pressure, the inlet pressure must be 26.4 for the 5 panels which correspond to 145 N/m^2 unlike in the case of 2 panels TTR that require an inlet pressure of 9.5 bar., which corresponds to 53 N/m^2 in the tube.

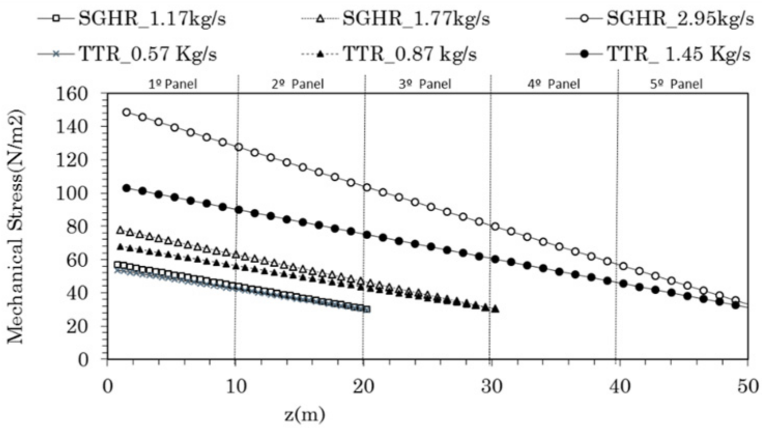

Fig 6. Tube outside crown mechanical stress distribution along the TTR and SGHR for 2, 3 and 5 panels.

On the other hand, thermal stress is lower for SGHR than for TTR. The thermal stress varies with heat flux distribution and is maximal at the peak radiation position.Fig. 7 shown the maximum thermal stress evolution along the panels. Even if the heat gain is doubled for SGHR, the thermal stress is lower for all the cases. That is due to the fact that the temperature distribution in the SGHR is more homogenous than in the TTR.

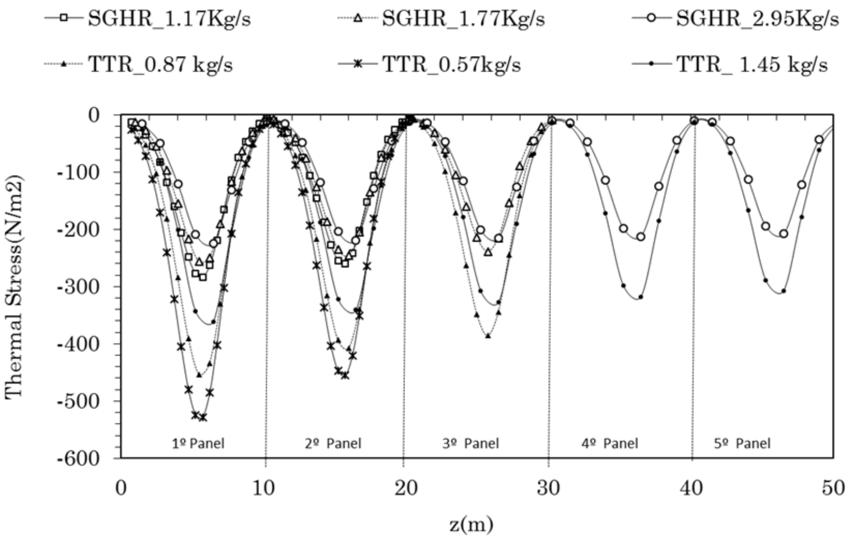

Fig. 7.Maximun thermal stress distribution for the most irradiated CV along the TTR and SGHR for 2, 3 and 5 panels.

Finally, the steady state stress and the number of cycles are used to determine the allowable stress in transients. Fig. 8 shows that the allowable stress is much higher in SGHR than in a TTR, so the SGHR is more reliable than the TTR and the start-up and shut down can be done faster. Moreover, the 2 and 3 panels TTRs allowable stress reach zero value in the radiation peak flux zone. Therefore this TTR design will not withstand the required number of cycles for the given 30 years of operation.

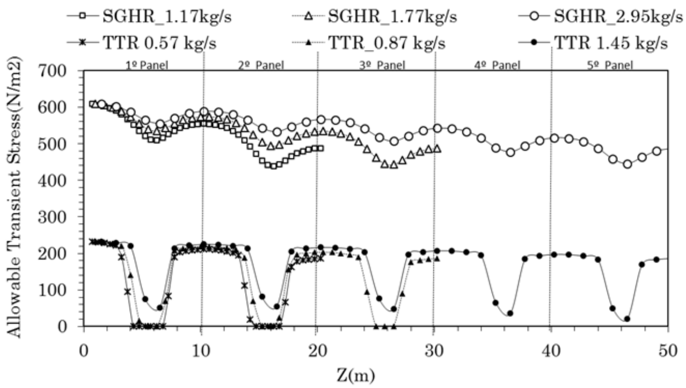

Fig 8 Allowable transient stress for the most irradiated CV along the TTR and SGHR for 2, 3 and 5 panels.

4. Conclusions

The constructive details of a novel hybrid Solar-Gas receiver have been presented. In order to evaluate its reliability it has been compared with respect to the well-known TTR by means of the evaluation of the steady state operation stress and the allowable transient stress for a design life of 30 years. For that purpose a mathematical model has been described and solved by an in house flexible code. The results showed that thermal stress in a SGHR is lower than in a not hybridized TTR. Moreover, the thermal power absorbed in a SGHR is the double for the same surface. On the other hand, the higher required mass flow rate in SGHR increases the pressure drops. Therefore, pressure operation of SGHR is higher and hence the mechanical stress too. The lower thermal stress and number of cycles allows higher transient stress for SGHR. In order to evaluate the SGHR concept that has been presented here a prototype will be constructed and tested in real operation condition in the CTAER installations.

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