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PRELIMINARY DESIGN OF THE FCC-EE VACUUM CHAMBER ABSORBERS*

M. Morrone, C. Garion, R. Kersevan, S. Rorison, P. Chiggiato
CERN, 1211 Geneva 23, Switzerland

Abstract

In the FCC-ee study, it is proposed that electron and positron beams circulate at high current and high energy in a 92-km circumference twin ring. The present operational scenario foresees a first running step at an energy of 45.6 GeV and around 1.4 A current, which would generate copious amounts of synchrotron radiation (SR) power and flux. To guarantee a quick decrease of the photon desorption yields and so a fast vacuum conditioning, it has been proposed to use localized SR absorbers along the vacuum chamber, spaced about 5 m apart. This would also help contain the high-energy Compton-scattered secondaries once the beam energy is increased up to 182.5 GeV, later in the experimental program.

In the preliminary design of FCC-ee vacuum chamber absorbers presented in this work, the SR thermal power is intercepted along around 100 mm of slanted surface. The temperature distribution in the adsorbers is estimated by Finite Element Analysis (FEA) and needs to be assessed to avoid any liquid-gas phase change within the water-cooling circuit. The cooling channels contain a twisted tape that increases the turbulence of water. This results in the desired heat transfer coefficient. The mechanical deformations due to the non-uniform temperature map are presented and analysed as well.

INTRODUCTION

The proposed 100 km-circumference Future Circular Collider (FCC) at CERN features, as a first stage, an electron–positron Higgs and electroweak factory (FCC-ee) operating at center-of-mass energies from 91 GeV (the Z mass) at around 1.4 A to a maximum of 365 GeV (above the tt production threshold). The design choice for the synchrotron radiation (SR) power is 50 MW/beam at all beam energies [1].

It is proposed to intercept this power by SR absorbers placed every 5 m rather than distributing it along the whole vacuum chamber. The main advantage would be a quicker decrease of the photon desorption yield and, therefore, a faster conditioning of the vacuum chambers.

The thermal absorbers should be designed to withstand the temperature and thermal stress induced by the high heat load. A similar design approach can be found for the crotch absorbers of synchrotron light facilities which are water cooled inserts made of copper. The ALBA absorber dissipates a total power of 6.78 kW with a peak power density of 242 W/mm^2 at a beam energy of 3 GeV [2]. The European Synchrotron Radiation Facility (ESRF) absorber removes 8.16 kW of power with a peak density of 177 W/mm^2 at a beam energy of 6 GeV [3]. The absorber of the Cornell High Energy Synchrotron Source Upgrade (CHESS-U) evacuates a total power of around 9 kW with a peak density of 800 W/mm^2 at a beam energy of 6 GeV [4].

DESIGN

The FCC-ee copper vacuum chamber is cooled down using two smooth channels welded on the winglet sides, through which room temperature water flows (see Figure 1). The cooling channel on the absorber side splits into two smaller channels which are integrated into the absorber and not directly exposed to the SR fan.

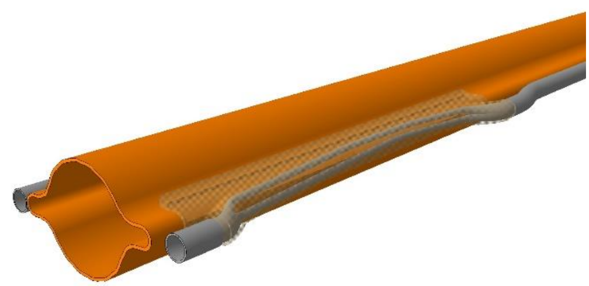

Figure 1: FCC-ee Vacuum Chamber with Transparent View of The Absorber to Visualize the Internal Cooling Channels.

The preliminary design of the SR absorber is displayed in Figure 2. It is a 340 mm long copper insert welded into a custom-made aperture machined into the vacuum chamber.

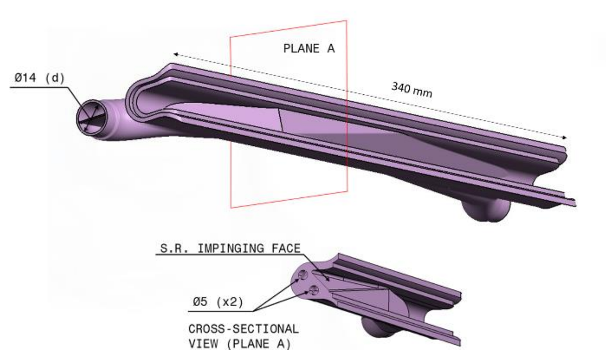

Figure 2: SR Absorber with Cross Sectional View.

The absorbers intercept a high heat load from SR on its 45° slanted surface. Thus, the temperature needs to be maintained below safety levels to ensure the absorber's mechanical integrity and avoid fluid phase transitions. As a

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result, the channels integrated into the absorber have a twisted tape design that significantly improves cooling capacity by enhancing turbulence mechanisms. However, this also leads to an increased pressure drop that requires a design optimization.

The absorber will be manufactured using Laser Powder Bed Fusion (LPBF) additive manufacturing process technology out of a copper grade to be defined. A green laser machine will be used as this enables pure copper to be printed with negligible degradation to mechanical, thermal, and electrical properties [5]. New machines are entering the market with lower energy requirements and smaller beam wavelengths that will provide a low-cost, low-energy solution. This method also has the benefit of being able to construct the internal complex twisted tape design. More traditional manufacturing methods would result in an unstable design with more energy intensive and mechanically destabilizing effects. This process allows no gap with the inner surface of the channel, which has been shown to greatly increase thermal efficiency [6]. In addition, copper materials that are 3D printed are being qualified for ultra-high vacuum (UHV) applications [7]. To the best of the authors knowledge, this would be the world first 3D printed SR absorber.

The maximum pressure difference between the inlet and outlet of the cooling line should be around 2-3 bar, and the inlet water temperature should be around 27°C. The power deposited on the absorber has a Gaussian profile with a peak density of 45 W/mm^2 and an integrated power of 3.5 kW over 109.2 mm of its slanted surface. SR strikes a (<4 mm) strip horizontally across the middle of the absorber surface. This refers to the 91 GeV center-of-mass machine.

The amount of beam induced heat of the FCC-ee vacuum chamber due to bunch length ranges from 100 to 200 W/m, depending on the effective bunch length at collision. The effective bunch length can vary significantly depending on several parameters that are challenging to define with precision at this stage of development. Therefore, in addition to the SR power mostly intercepted by the absorbers, a uniform heating of 200 W/m is considered in this analysis.

Fluid dynamics analysis

Numerical and analytical analyses were utilized to support the design of the SR absorber, specifically for the twisted-tape cooling channels within it show in Fig. 3. The design is based on implicit formulations by Manglik and Bergles, which are in good agreement with experimental observations [8].

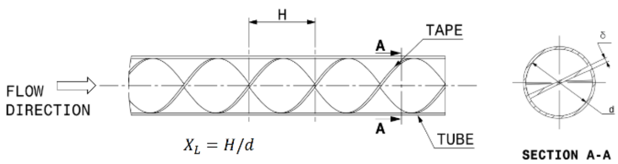

Figure 3: Twisted-tape Design of the Absorber Channels.

The Nusselt number equation used for the design of the cooling channel is given as follows:

$$Nu = \left(1 + \frac{0.769}{x_L}\right)$$
$$0.023\ Re^{0.8}\ Pr^{0.4}\ \left(\frac{\pi}{\pi-4\delta/d}\right)^{0.8}\ \left(\frac{\pi+2-2\delta/d}{\pi-4\delta/d}\right)^{0.2}\ \left(\frac{\mu_b}{\mu_s}\right)^{0.3}$$

Where $x_L$is the twist ratio of the tape, which is defined as the twist period H over the internal diameter of the tube d. Re is the Reynolds number, Pr is the Prandtl number, δ is the thickness of the tape, μ_b and μ_s are the dynamic viscosity of water at the bulk fluid mean temperature and on the surface temperature of the cooling channel, respectively.

The heat transfer coefficient h can be calculated using the following formula:

$$h = \frac{Nu\ \lambda}{d}$$

where λ is the thermal conductivity.

The friction factor of the twisted-tape cooling channels is calculated using the following equation:

$$f = \frac{0.0791}{Re^{0.25}}\ \left(\frac{\pi}{\pi-4\delta/d}\right)^{1.75}\ \left(\frac{\pi+2-2\delta/d}{\pi-4\delta/d}\right)^{1.25} \left(1 + \frac{2.752}{{x_L}^{1.29}}\right)$$

Hence, the pressure drop can be computed as:

$$\Delta P = \frac{4\ f\ L\ \rho\ u^2}{2\ d}$$

Where ρ is the density of water, u is its speed and L is the length of the cooling channel.

It should be noted that the equations above are valid for $Re > 10^4$. Conversely, the heat transfer and pressure drop of the smooth cooling channels were calculated by the Dittus-Boelter and the Blasius correlation, respectively. At this stage of the design, only friction-related pressure losses were considered. The losses associated with bifurcation or bends of the cooling line were not considered as the design is still in its preliminary stages.

The diameter of the smooth cooling channels on the chamber sides was determined from minimizing the pressure drop. A compromise was reached to ensure the water remains below boiling temperature and to limit the pressure drop to 2-3 bar along the cooling line. Table 1 shows the design parameters and the relative fluid dynamics quantities to achieve this compromise, both for the absorber channels and the smooth cooling channels.

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Table 1: Geometric and Fluid Dynamics Parameters of The Absorber and Smooth Cooling Channels

Param.	Absorber channel 0.34 m long	Smooth channel 15 m long
d	$5\ mm$	$14\ mm$
δ	$0.5\ mm$	–
H	$30\ mm$	–
v	$5\ m/s$	$1.3\ m/s$
Re	$2.5\ \cdot 10^4$	$1.8\ \cdot 10^4$
$\dot{m}$	$2\ \cdot 100\ g/s$	$200\ g/s$
h	$3.3\ \cdot 10^4\ W/m^2K$	$5.4\ \cdot 10^3\ W/m^2K$
ΔP	$0.7\ bar$	$0.25\ bar$

Three absorbers can be cooled through a vacuum chamber that is 15 m long, with two main cooling channels running on either side of the chamber. The absorbers contribute to a total thermal power of 10.5 kW (3.5 kW each) while the vacuum chamber to 3 kW (200 W/m x 15 m). In the parallel configuration of the two cooling tubes, the combined mass flow rate is 400 g/s, leading to an average temperature difference of 8°C between inlet and outlet.

The pressure drop over a continuous smooth cooling channel of 15 m is 0.25 bar.

The cooling channel on the absorber side incurs a pressure drop of 0.7 bar per absorber, resulting in a total pressure drop of 2.1 bar for all three absorbers. In addition, the smooth channel incurs a pressure drop of 0.25 bar, resulting in a total pressure drop of 2.35 bar. The two cooling channels are meant to be water pumped independently.

Thermal mechanical analysis

The temperature distribution of the SR absorber is illustrated in Fig. 4.

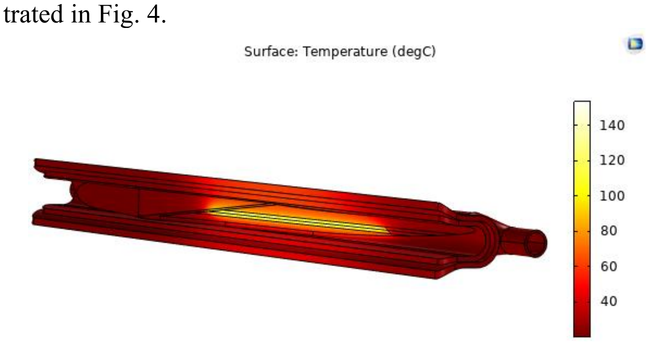

Figure 4: Temperature Distribution of the SR Absorber.

The absorber surface reaches a maximum temperature of approximately 150 °C, which meets the fatigue guidelines proposed by the Advanced Photon Source (APS). These guidelines specify that the maximum temperature rise should not exceed 150 °C [9].

The maximum temperature on the walls of the cooling channel is approximately 65 °C. Therefore, the water does not change phase, which would have compromised the flow regime and the heat extraction. The thermal deformation of one periodic unit of the vacuum chamber (approximately 5 m) containing an absorber is presented in Fig 5. The second cooling channel helps to homogenize the temperature around the absorber area and, therefore, limit differential deformations. The maximum deformation is approximately 0.5 mm at around a 45° angle, which is considered acceptable. The von Mises stress is shown in Fig.6 and the peak value is around 180 MPa.

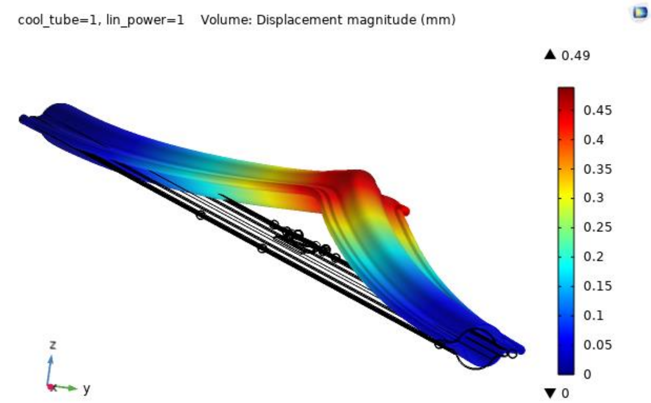

Figure 5: Total Displacement of One Periodic Unit (5 m Long) of the FCC-ee Vacuum Chamber.

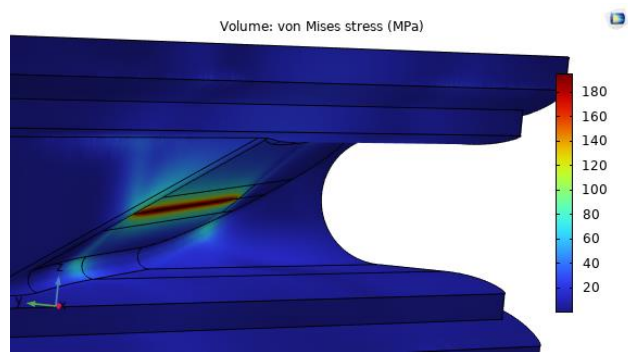

Figure 6: Von Mises Stress of the SR Absorber.

CONCLUSIONS

Overall, the proposed design provides a promising solution for the cooling system of the FCC-ee vacuum chamber absorbers. However, further analysis and optimization are needed to improve the design and ensure its feasibility for the final implementation. This includes considering additional factors such as the effects of bends and bifurcations on pressure drops, the impact of permanent insulation, thermal radiation on the temperature distribution, and the influence of manufacturing constraints on the design. Fluid dynamics tests to validate the analysis will be performed too. Nevertheless, the presented approach offers a foundation for future development of the vacuum chamber cooling system for the FCC-ee accelerator.

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