Coupled RF-Thermo-Structural Analysis of CLIC Traveling Wave Accelerating Structures

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Keywords:

Summary

Temperature changes in the CLIC accelerating structure lead to unwanted deformation. One of the aims of the CLIC module team is to optimize the necessary cooling system. The extended thermal program of the module was complemented by these thermal studies on real RF structures tested in X-band high-power test stands at CERN, Geneva (CH). The impact of thermal expansion on the RF characteristics of CLIC traveling wave accelerating structures is studied. The simulation setup in COMSOL is described in much detail. The necessary geometry simplifications are pointed out, mesh configurations are discussed by means of convergence studies, as well as important post-processing quantities are introduced which are common for characterizing traveling wave structures. The present report treats two different prototypes of the TD26_R05 design, a damped traveling wave accelerating structure using 26 cells and two matching cells, with a bending radius of 0.5 mm being applied. One of these prototypes is currently being conditioned in the Xbox2 test stand.
## Acronyms

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<td>TD26_R05</td>
<td>Design of a damped CLIC accelerating structure using 26 cells.</td>
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<td>Prototype of the TD26_R05 traveling wave structure using compact couplers.</td>
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1 Introduction

The CLIC accelerating structures provide very tight manufacturing and positioning tolerances, both of which are sensitive to temperature changes. A careful design of the cooling system is thus mandatory to ensure stable operation and the desired performance of the machine, taking into account that these components provide the largest power density of the accelerator complex.

The impact of thermal expansion on the RF characteristics of CLIC traveling wave accelerating structures is studies. The simulation setup in COMSOL is described in much detail. The necessary geometry simplifications are pointed out, mesh configurations are discussed by means of convergence studies, as well as important post-processing quantities are introduced which are common for characterizing traveling wave structures. The present report treats two different prototypes of the TD26_R05 design, a damped traveling wave accelerating structure using 26 cells and two matching cells, with a bending radius of 0.5 mm being applied. One of these prototypes is currently being conditioned in the Xbox2 test stand.

2 Geometry

Two different prototypes are subjected to thermo-structural-rf simulations both derived from the TD26_R05 design, a damped CLIC traveling wave accelerating structure using 26 cells and two matching cells. The waveguides of each cell provides a bending radius of 0.5 mm towards the propagation direction. The first prototype denoted as TD26_R05_CC considers a single accelerating structure equipped with compact couplers [5]. The waveguides are shortened with respect to the original RF design which yield a phase advance per cell slightly lower than 120 deg. The second prototype denoted as TD26_R05_SS considers two combined accelerating structures equipped with compact couplers and manifolds comprising SiC absorbers. The generally referred super structure is currently being conditioned and tested in Xbox2.

2.1 Simplification

Figs. 1 and 2 shows the original prototype assemblies together with the corresponding simplified model. All modifications of the original models have been carried out using SPACECLAIM [1] and can be summarized as follows:

- Components were suppressed which are expected to have a marginal impact on the studies, for instance, flanges, connectors, nuts, bellows, and washers.
- Tuning holes were filled.
- Cooling pipes and associated details were removed. Note, they are represented by separate line elements in the simulations.
- The alignment and orientation of various waveguides required corrections.
- Slider surface were removed. Note, disks of the TD26_R05_CC prototype assembly provide a very small chamfer on the inner surface for manufacturing reasons, only. For the studies of interest, such details can be ignored.
Figure 1: (a) Original geometry of the TD26_R05_CC prototype assembly, with the highlighted features being modified, removed or replaced for the studies. (b) Simplified model. Note, the holes reserved for cooling pipes are only kept for reasons of illustration.

Figure 2: (a) Original geometry of the TD26_R05_SS prototype assembly, with the highlighted features being modified, removed, or replaced for the studies. (b) Simplified model. Note, the holes reserved for cooling pipes are only kept for reasons of illustration.

- Interfaces between different parts were much simplified. Hence, any holes, groves, bending or other small details in between the geometry parts were removed. This reduces significantly the complexity of resulting meshes.

- In order to achieve a smooth continuation of the waveguides provided by the cavity disks, a bending radius of 0.5 mm is applied at the waveguide edges of the manifolds towards propagation direction.

- Manifolds and waveguides of the TD26_R05_SS prototype assembly are cut to a com-
mon length while removing the SiC absorbers. Note, the present studies focus on the fundamental mode, only. Mechanisms related to higher-order modes are excluded.

- All cavity disks of an accelerating structure were unified to a single geometry part.

It is important to note that the complementary part, that is the inner cavity volume filled with vacuum, is not created in SPACECLAIM but rather directly in the particular simulation tool, hence, in this case COMSOL. This avoids self-intersection errors which often occur when transferring complex geometries between different software despite the fact that a common geometry file standard, such as *.sat or *.stp is used. For this reason, the models are particularly modified in that the interfaces between different parts are as simple as possible while the number of interfaces is minimized.

### 2.2 Cooling Path

Coupled simulation of three-dimensional flow and heat transfer inside the cooling channels as well as the surrounding structure are computationally expensive. Often a semi-analytic approach is used to approximately determine the heat transfer coefficient between the coolant and solid. This approach is typically applied in ANSYS or CST. In contrast, COMSOL provides a fully consistent approach combining the one-dimensional pipe flow with three-dimensional heat transfer in the surrounding structure. Both approaches, which are further described in Sec. 3.2 allow to represent the cooling circuits by paths, hence, line elements as shown in Fig. 3. The positioning and properties of the cooling channels are important aspects for the structure design. This way, it becomes very convenient to optimize the cooling system and parameters. Figs. 3(a) and 3(b) show possible circuit configurations using one inlet and outlet for each prototype assembly.

![Figure 3: Cooling paths modeled by line elements. (a) TD26_R05_CC and (b) TD26_R05_SS prototype assembly.](image-url)
Though both prototype assemblies are based on the same RF design, the number and location of holes preserved for the cooling channels differ as illustrated in Fig. 3(a) and (b) by corresponding cross-sections at the level of input couplers. Any cooling path can be defined based on the highlighted transverse coordinates together with longitudinal positions which are summarized in Tables 2 and 3 in Appendix A.

Due to their simplified representation by line elements, the holes preserved for the cooling channels are not required neither for the semi-analytic nor fully consistent approach. Consequently, these features are not present in considered geometries which has the advantage that a modification of the pipe diameter does not imply to modify the geometry. It is worth noting that a certain mesh refinement around the cooling channels is required for accurate heat flow calculations. For this reason, the original channels are used as described in Sec. 4.1.

2.3 Preparation

Independent of the simplification steps described in Sec. 2.1, the geometry requires further preparation after importing in the particular simulation software. For instance, the volume surrounded by the structure and filled with vacuum is part of this preparation step in order to avoid self intersecting geometries and, thus, meshing errors. Fig. 5 sketches the workflow applied in COMSOL. Note, this process strongly depends on the features provided by the particular software.

![Workflow](image)

Figure 5: Geometry preparation in COMSOL.
After importing the simplified prototype assemblies from Sec. 2.1, the different parts, such as disks, cooling blocks, and couplers first become unified. A subsequent healing process removes slider surfaces and short edges from the structure before its complementary geometry is created. The latter one constitutes the vacuum part used for the RF simulation.

Additional geometry parts are added such as a line and cylinder along the particle beam. Likewise cylinders are placed at the preserved locations of the cooling channels while the circuit itself is defined by a polygon whose coordinates are stored in a separate file (*CoolingCircuit.txt). Other than the cooling path and vacuum part, all additional geometry parts serve for mesh refinements in order to provide a higher accuracy of the electromagnetic field quantities along the particle beam propagation as well as the heat transfer coefficient between the coolant and surrounding solid. Several

Boolean operation and partitions are applied to resolve overlaps, define interfaces, associate domains of common material to objects which are eventually split into virtual domains for the meshing process (Sec. 4.1). Generally, there are two strategies to finalize a CAD model. Either interfaces between different geometry parts are shared such that continuity of the field and fluxes is preserved or the coinciding boundaries are paired but still belong to the individual domain. The choice has consequences in the geometry description, physics settings, and mesh as discussed in Sec. 4.2.

Predefined selections of multiple domains boundaries and edges facilitate the settings for physics, mesh, and post processing. They further ensure a consistent configuration even after modifying or replacing the model provided it is similar to the original one. A complete list of selections by means of the TD26_R05_CC prototype assembly is given in Tables 4–7 in Appendix A.

Finally, virtual operations are used to compose surfaces, ignore edges or dedicate specific geometry part for meshing, only, such as the cooling channels. They are generally used to improve the mesh quality by avoiding narrow vertices, short edges, thin surfaces or other small features [4]. In particular, surface compositions of the iris boundaries significantly improve the mesh quality.

3 Physics

3.1 Electromagnetic Waves

The CLIC traveling wave structures are feed via two symmetrically arranged waveguide couplers as illustrated in Fig. 6(a). While propagating through the structure, a fraction of energy carried by the emerged wave is dissipating into the cavity wall due to the finite conductivity. The remaining fraction of energy is extracted by two waveguide couplers using the same arrangement as for the input side. Note, the unloaded case shall be considered in the present studies, that is, no energy transfer to the particle bunch is involved.

The wave propagation can be described by the wave equation for the electric field intensity in frequency domain according to

$$\nabla \times \nabla \times \mathbf{E} - k_0^2 \mathbf{E} = 0, \quad \mathbf{r} \in \Omega_{\text{vac}},$$

(1)

The spatial domain $\Omega_{\text{vac}}$ refers to the inner volume of the cavity filled with vacuum (Appendix A, Table 7). Accordingly, the wave propagates at the speed of light which is described
by the wave number \( k_0 = \omega / c_0 \) where \( \omega = 2\pi f \) is the angular frequency. The latter one is given by the frequency of the exciting TE_{10} waveguide mode at both input ports which satisfy the following eigenvalue problem on the particular port boundary \( \partial \Omega_p \)

\[
\nabla \times \nabla \times H_n + (\beta^2 - k_0^2) H_n = 0, \quad r \in \partial \Omega_p, \tag{2}
\]

with \( H_n \) being the component of the magnetic field intensity perpendicular to the boundary and in parallel to the normal unit vector \( n \). The generalized index \( n \) specifies the periodicity of the solution in both transverse direction spanning the port boundary. Note, the frequency is chosen such that only the TE_{10} mode is able to propagate through the waveguide according to the propagation constant \( \beta \) perpendicular to the boundary \( \partial \Omega_p \). The eigenmode solution of (2) at both input and output ports serve as boundary conditions for the problem (1). Though the output ports do not provide any inward propagating wave, the solutions are required to describe an infinite continuation of the corresponding waveguide. Fig. 7(a) shows the port notation by means of the TD26_R05_CC prototype assembly. Port 1 and 2 serve as input ports providing the TE_{10} mode, both at the same face while port 3 and 4 serve as the output for the traveling wave. It is important to note that problem 2 does provides a unique solution for the specific mode except for a phase which can be a multiple of \( \pi \). It is, thus possible that an artificial phase offset of \( \pi \) may be required between the input ports in order to excite the traveling wave in the structure. This is typically verified by an initial RF simulation of only the port modes.

In order to account for energy dissipation in the cavity wall an impedance condition is applied at the common boundary between the vacuum filled domain \( \Omega_{\text{vac}} \) and surrounding structure \( \partial \Omega_{\text{solid}} \), which is given by

\[
\mathbf{n} \times \mathbf{E}(\mathbf{r}) = Z_s \mathbf{n} \times \mathbf{H}(\mathbf{r}) \quad \mathbf{r} \in \partial \Omega_{\text{solid}} \cap \partial \Omega_{\text{vac}}. \tag{3}
\]

The surface impedance of normal conductors calculates as [13, p. 429]

\[
Z_s = (1 + j) \sqrt{\frac{\mu_0 \omega}{2\sigma}} = \frac{1 + j}{\sigma \delta}, \tag{4}
\]

with the permittivity constant \( \mu_0 \), angular frequency \( \omega \), electric conductivity \( \sigma \) given by the material, and penetration depth \( \delta \).
Symmetries of the model allow to significantly reduce the problem size. Typically the structures are designed by considering only a quarter as illustrated in Fig. 7(b). The tangential components of the magnetic field intensity $\mathbf{H}$ provided by traveling wave must vanish at these symmetry planes $\partial\Omega_{\text{sym}}$ according to

$$\mathbf{H}(\mathbf{r}) \times \mathbf{n} = 0, \quad \mathbf{r} \in \partial\Omega_{\text{sym}},$$

with the unit vector $\mathbf{n}$ being normal to the particular boundary. It is worthwhile to note that the symmetry planes get lost once the deformed structure is considered since the distribution of temperature and, thus, thermal expansion are generally not symmetric. For this reason, only the complete model shall be considered in the following.

Using the previous notation of active and passive ports, the scattering parameter to characterize the RF reflection and transmission are defined as

$$s_{11} = \frac{\iint_{p_1} (\mathbf{E} - \mathbf{E}_{p_1}) \cdot \mathbf{E}_{p_1}^* \, dA + \iint_{p_2} (\mathbf{E} - \mathbf{E}_{p_2}) \cdot \mathbf{E}_{p_2}^* \, dA}{\iint_{p_1} \mathbf{E}_{p_1} \cdot \mathbf{E}_{p_1}^* \, dA + \iint_{p_2} \mathbf{E}_{p_2} \cdot \mathbf{E}_{p_2}^* \, dA}$$

(6)

$$s_{21} = \frac{\iint_{p_3} \mathbf{E} \cdot \mathbf{E}_{p_3}^* \, dA + \iint_{p_4} \mathbf{E} \cdot \mathbf{E}_{p_4}^* \, dA}{\iint_{p_3} \mathbf{E}_{p_3} \cdot \mathbf{E}_{p_3}^* \, dA + \iint_{p_4} \mathbf{E}_{p_4} \cdot \mathbf{E}_{p_4}^* \, dA}$$

(7)

Here, $s_{11}$ is the RF reflection from both input ports combined while $s_{21}$ is the RF transmission from both input ports to both output ports. Furthermore, $\mathbf{E}$ is the total electric field intensity which solves (1) and $\mathbf{E}_{pi}$ is the specific eigenmode solution satisfying (2) on the boundary of port $i$. Note, the concatenated consideration of input ports is necessary in order to satisfy the definition of scattering parameters, that is, one port is excited while the remaining ports.
are matched. As the electromagnetic field inside the structure is simultaneously excited by two ports it is not possible to evaluate for example the reflection for only port 1. However, the concatenated evaluation of the reflection for port 1 and 2 according to (6) is valid. Note, definitions (6) and (7) correspond to the scattering parameter of the symmetric case in Fig. 7(b), with only one port being excited. A second input port is implicitly given by the symmetry condition as it mirrors the first input port.

With integral operators defined on each port cross section, the evaluation of scattering parameters according to (6) can be implemented in COMSOL as follows:

$$S_{11} = \frac{(i\int_{Port1}(\text{emw}.\text{Ex} - \text{emw}.t\text{Emode}x_1)*\text{conj}(\text{emw}.t\text{Emode}x_1) \\
+ (\text{emw}.\text{Ey} - \text{emw}.t\text{Emode}y_1)*\text{conj}(\text{emw}.t\text{Emode}y_1) \\
+ (\text{emw}.\text{Ez} - \text{emw}.t\text{Emode}z_1)*\text{conj}(\text{emw}.t\text{Emode}z_1)) \\
+ i\int_{Port2}(\text{emw}.\text{Ex} - \text{emw}.t\text{Emode}x_2)*\text{conj}(\text{emw}.t\text{Emode}x_2) \\
+ (\text{emw}.\text{Ey} - \text{emw}.t\text{Emode}y_2)*\text{conj}(\text{emw}.t\text{Emode}y_2) \\
+ (\text{emw}.\text{Ez} - \text{emw}.t\text{Emode}z_2)*\text{conj}(\text{emw}.t\text{Emode}z_2))) \\
/ (i\int_{Port1}(\text{emw}.t\text{Emode}x_1*\text{conj}(\text{emw}.t\text{Emode}x_1) \\
+ \text{emw}.t\text{Emode}y_1*\text{conj}(\text{emw}.t\text{Emode}y_1) \\
+ \text{emw}.t\text{Emode}z_1*\text{conj}(\text{emw}.t\text{Emode}z_1)) \\
+ i\int_{Port2}(\text{emw}.t\text{Emode}x_2*\text{conj}(\text{emw}.t\text{Emode}x_2) \\
+ \text{emw}.t\text{Emode}y_2*\text{conj}(\text{emw}.t\text{Emode}y_2) \\
+ \text{emw}.t\text{Emode}z_2*\text{conj}(\text{emw}.t\text{Emode}z_2))))$$

The electromagnetic wave module emw provides direct access to all previously mentioned RF quantities. The surface integral operator defined on the port boundary n is denoted as \text{iiPort}n(). A detailed description of the integral operators in COMSOL is given in [11].

Using the method of periodic voltage standing-wave ratio described by Kroll et. al. [6]
the cell-to-cell phase advance and internal reflection calculate as

\[ \psi_{\text{unit}}(z) = \arccos \frac{\Delta^+(z)}{2} \]  

(8)

\[ R_{\text{unit}}(z) = \frac{2 \sin \psi_{\text{unit}}(z) - j\Delta^-(z)}{2 \sin \psi_{\text{unit}}(z) + j\Delta^-(z)} \]  

(9)

with the quantities \( \Delta^\pm(z) \) being superposed terms of the longitudinal electric field at the beam center, defines as

\[ \Delta^\pm(z) = \frac{E_z(z + L_{\text{unit}}) \pm E_z(z - L_{\text{unit}})}{E_z(z)}. \]  

(10)

Here, \( L_{\text{unit}} \) refers to the cell or unit length, which is about 8.332 mm for the TD26R05 design. By using the spatial \( \text{at3()} \) operator of COMSOL, which provides field components at arbitrary coordinates, the cell-to-cell phase advance along the beam axis can be evaluated as follows:

\[ \text{Phadvu} = \frac{180}{\pi} \arccos \left( \frac{\text{at3}(0,0,z+\text{UnitLength}, \text{emw.Ez}) + \text{at3}(0,0,z-\text{UnitLength}, \text{emw.Ez})}{\text{at3}(0,0,z, \text{emw.Ez}) / 2} \right) \]

Both, the cell-to-cell phase advance and internal reflection based on the periodic voltage standing wave ratio are used to adjust the matching cells in the design phase. It is, thus of particular interest to study the impact of thermal expansion and contraction on these quantities along the center axis. Fig. 9(a) and (b) show respectively the cell-to-cell phase advance and internal reflection for each individual structure of the considered prototype assemblies. Despite of the common RF design, the phase advance of the TD26R05CC prototype assembly is slightly below 120 deg. This is due to the waveguides being shortened and rounded as shown in Fig. 3. Note, the nominal design provides reflections which are generally lower than shown in Fig. 9(b) due to the mesh used. Though it is not fine enough to provide accurate reflections at a level below 1 %, it is suitable to study, in particular, the

![Figure 9](image-url)

Figure 9: (a) The cell-to-cell phase advance and (b) internal reflection for different prototype assemblies. The quantities are derived from the periodic voltage standing wave method [6].
impact of thermo-structural processes on the RF behavior. Detailed information about the mesh and convergence studies are discussed in Sec. 4.

Traveling wave structures are further characterized by the stored energy per unit length, \( U \), dissipated power per unit length, \( P_{\text{diss}} \), and the power flow through the structure cross section, \( P \). Like the previously introduced cell-to-cell phase advance and internal reflection, all these quantities are functions of the longitudinal coordinate \( z \), and are specified by the unit length, hence, the distance from cell to cell as illustrated in Fig. 6(b). Let \( h \) be a rectangular functions defined as

\[
h(z) = \begin{cases} 
  \frac{1}{L_{\text{unit}}}, & 0 \leq z \leq L_{\text{unit}}, \\
  0, & \text{otherwise.}
\end{cases}
\]  

(11)

and \( \delta \) be the delta distribution, the following relations apply for the stored energy, dissipated power, and power flow, respectively

\[
U(z) = \frac{\varepsilon_0}{2} \int \int \int_{\Omega_{\text{vac}}} h(z-z')E(r')E^*(r')dV', \quad r' \in \Omega_{\text{vac}},
\]  

(12)

\[
P_{\text{diss}}(z) = \frac{1}{2} \Re\{Z_s\} \int \int_{\partial\Omega_{\text{vac}}} h(z-z')H(r')H^*(r')dA', \quad r' \in \partial\Omega_{\text{vac}},
\]  

(13)

\[
P(z) = \frac{1}{2} \int \int \int_{\Omega_{\text{vac}}} \delta(z-z')E(r') \times H^*(r')dV', \quad r' \in \Omega_{\text{vac}}.
\]  

(14)

The first two integrals can be directly evaluated in COMSOL using the time averaged energy of the electromagnetic wave \( \text{emw.Wav} \) and the surface loss density \( \text{emw.Qsh} \) according to

\[
\begin{align*}
\text{intWav} &= \iiint_{\Omega_{\text{vac}}} \left(h((\text{dest}(z)-z)/\text{UnitLength}) \ast \text{emw.Wav}\right)/\text{UnitLength} \\
\text{intQsh} &= \iiint_{\Omega_{\text{vac}}} \left(h((\text{dest}(z)-z)/\text{UnitLength}) \ast \text{emw.Qsh}\right)/\text{UnitLength}
\end{align*}
\]

The operator \( \text{dest}() \) allows to exclude a variable from the integration, hence to distinguish between \( z \) and \( z' \) in (12)–(14). Fig. 10(a) and (b) show the stored energy and dissipated power

![Figure 10](a) Stored energy per unit length and (b) dissipated power per unit length for different prototype assemblies. An average power of 1 W at each input port is assumed.
per unit length, respectively, for the TD26_R05 design with marginal differences between the prototype assemblies. It is worth noting that the use of a rectangular function according to (11) provokes numerical noise due to its discontinuous character. This becomes particularly apparent for the dissipated power by several spikes as depicted in Fig. 10(b). In order to suppress such numerical fragments a slightly smoothed rectangular function in contrast to (11) can be used.

The power flowing through the cross section of the structure is easiest evaluated by a projection of the Poynting vector from the entire domain onto the $yz$-plane and further projected on the $z$-axis as sketched in Fig. 11. Note, this approach which equals an integration of the Poynting vector in the transverse $xy$-plane for each $z$ along the structure, is numerically favorable and much more accurate than solving the volume integral in (14) due to the discontinuous character of the delta distribution. Note, the step-wise projection from the three-dimensional Poynting vector to the one-dimensional power flow by using an intermediate projection onto the $yz$-plane is for implementation reasons, only. The projection operators provided by COMSOL reduce the dimension by one when mapping a desired field quantity [10]. Once the projection operators are defined the power flow in longitudinal direction is calculated by a concatenated use according to

$$\text{intPoavn} = \text{projZ} (\text{projYZ} (\text{emw.Poavn}))$$

with $\text{emw.Poavn}$ being the time averaged local power flow in $z$-direction. Fig. 12(a) shows the power flow through the structure cross section for the different prototype assemblies. The resulting profiles are approximately the same for all accelerating structures of the considered prototype assemblies. However, there is a slight difference with respect to the power flow through the first matching cell.

Further important quantity are the intrinsic quality factor $Q$ defined as the ratio of stored energy per unit length and corresponding energy dissipated in the cavity wall over one RF

![Figure 11: Projection of an arbitrary field quantity defined in the inner volume filled with vacuum (a) onto the $yz$-plane (b). This approach is used to evaluate the power flow through the cross section of the structure as it requires the integration of the Poynting vector in the transverse plane for each $z$.](image)
Figure 12: (a) Power flow through the structure cross section. (b) Longitudinal RF voltage per unit length on the center axis. An average power of 1 W at each input port is assumed.

cycle according to

\[ Q(z) = \frac{\omega U(z)}{P_{\text{diss}}(z)}, \]  

the group velocity \( v_g \) which is the speed at which RF power propagates through the traveling wave structure, given by

\[ v_g(z) = \frac{d\omega}{dk} = \frac{P(z)}{U(z)}, \]  

and the filling time \( t_{\text{fill}} \) defined as

\[ t_{\text{fill}} = \int_0^L \frac{dz}{v_g(z)}. \]  

The longitudinal RF voltage per unit length on the beam axis \( \Omega_{\text{beam}} \) is defined by the integral of the longitudinal electric field component taking into account the phase advance for a particle at the speed of light, and is given by

\[ E_{\text{acc}}(z) = \left| \int_{-\infty}^{\infty} h(z-z') \mathbf{E}(r') e^{jkoz'} \mathbf{e}_z dz' \right|, \quad r' \in \Omega_{\text{beam}} \]  

Note, this quantity corresponds to the accelerating gradient in case of unloaded standing wave structures. Moreover, the shunt resistance \( R_{\text{sh}} \) and geometric shunt resistance \((R/Q)\) are calculated from the longitudinal voltage gradient respectively as

\[ R_{\text{sh}}(z) = \frac{E_{\text{acc}}^2(z)}{P_{\text{diss}}(z)} \]  

\[ (R/Q)(z) = \frac{E_{\text{acc}}^2(z)}{\omega U(z)} \]
Finally, the accelerating gradient of unloaded traveling wave structures is given by [7]

$$
\tilde{E}_{\text{acc}}(z) = E_{\text{acc}}(0) \sqrt{\frac{v_g(0)}{v_g(z)}} \sqrt{\frac{(R/Q)(z)}{(R/Q)(0)}} \exp \left( \frac{\omega}{2} \int_0^z \frac{dz'}{v_g(z')Q(z')} \right).
$$

(21)

In order to reduce post processing time, it is recommendable to export only the stored energy $U$, dissipated power $P_{\text{diss}}$, and longitudinal voltage $E_{\text{acc}}$ each per unit length as well as the power flow through the structure cross section, $P$. The remaining quantities can be subsequently derived without re-calculating the integrals of (12)–(14) and (18).

### 3.2 Heat Transfer and Pipe Flow

The heat transfer in the structure is characterized by the inward heat flux which corresponds to the RF losses as discussed in Sec. 3.1, the convection through cooling channels, and eventual heat dissipation to air. The minor but not negligible contribution of the latter one is difficult to quantify by simulations as it depends on the complex ventilation system of the tunnel. Figs 13(a) and (b) shows respectively the total heat deposit of the RF field as well as heat dissipation into air for various case scenarios of the TD26_R05_CC prototype assembly.

![Graph](image1.png)

**Figure 13:** (a) Time averaged RF power dissipated into the total cavity wall of the TD26_R05_CC prototype assembly as a function of inserted RF power at both input ports. (b) Corresponding heat dissipation into air assuming different flow rates through the single cooling circuit as illustrated in Fig. 3(a). A fixed heat transfer coefficient of 5 W/(m²K) accounts for the surrounding air convection.
with the cross section averaged fluid velocity $\mathbf{u}$ and pressure $p$ being solved for. Both quantities are defined along the one-dimensional domain of the cooling path $\Omega_{\text{pipe}}$. The second term in (22) accounts for surface shear stresses [2, p. 340]. The friction factor $f_D$, also denoted as Darcy friction factor, strongly depends on the flow regime as expressed by the Reynolds number:

$$\text{Re} = \frac{\rho |\mathbf{u}| d}{\mu}. \quad \text{(24)}$$

The friction factor of a fully developed laminar flow ($\text{Re} \lesssim 2300$) is given by

$$f_D = \frac{64}{\text{Re}}. \quad \text{(25)}$$

Turbulent flows may further depend on the relative surface roughness $\epsilon/d$ of the pipe. Both dependencies are implicitly correlated in the phenomenological Colebrook–White equation which is illustrated in Fig. 14 for a wide range of the Reynolds number and varying roughness. The Moody chart provides a convenient way to estimate the pressure drop needed to sustain an internal flow $u_0$ over the length $L$ which follows from (22) according to

$$\Delta p = -f_D \frac{L \rho u_0^2}{d}. \quad \text{(26)}$$

Various approximations of the Colebrook–White equation have been developed over the past decades in order to express the friction factor in an explicit manner. A single correlation encompassing Reynolds numbers from $3 \times 10^3$ to $5 \times 10^6$ is given by [9]

$$f_D = (0.79 \ln \text{Re} - 1.64)^{-2}. \quad \text{(27)}$$
It applies to smooth pipes whose roughness is completely covered by a viscous layer. Note, this case corresponds to the highlighted curve in Fig. 14. Another approximation developed by Churchill [12] applies to all flow regimes including the laminar flow, and is used for the present simulations to evaluate the friction factor locally.

The energy conversation of the coolant and surrounding structure can be described as

\[ c_p \rho u \nabla T = \nabla \cdot (k_{\text{pipe}} A \nabla T) + f_D \frac{\rho A}{2d} |u|^3 + Q_{\text{wall}}, \quad r \in \Omega_{\text{pipe}}, \]  
\[ 0 = \nabla \cdot (k_{\text{solid}} \nabla T), \quad r \in \Omega_{\text{solid}} \setminus \Omega_{\text{pipe}}, \]  

where \( c_p \) is the heat capacity of the coolant at constant pressure and \( k \) is the thermal conductivity of the particular material. The terms on the right hand side of (28) account respectively for the conductive heat transfer, energy dissipation by internal friction, and heat exchange with the surrounding structure. The latter one is calculated according to

\[ Q_{\text{wall}} = h_{\text{pipe}} \pi d \Delta T, \]  

with \( \Delta T \) being the temperature difference to the surrounding structure. The convection heat transfer coefficient \( h_{\text{pipe}} \) is calculated via the Nusselt number which describes the enhancement of heat transfer through a fluid layer by convection relative to conduction according to

\[ h = \frac{Nu \cdot k}{d} \]  

Fig. 15 shows the Reynolds and Nusselt number obtained from thermal simulations of the TD26_R05_CC prototype assembly as sketched in Fig. 3(a), with varying RF power at the input ports and gradually increased flow velocity of water in the single cooling circuit. Note, the

![Figure 15](image-url)

Figure 15: Thermal study of the TD26_R05_CC prototype assembly as sketched in Fig. 3(a), with varying RF power at the input ports and gradually increased flow velocity of water in the single cooling circuit. (a) The Reynolds and (b) Nusselt number, each averaged over the entire cooling path. The error bars result from the corresponding minimum and maximum.
flow becomes close to be laminar at an average velocity of 11/min. Though, the convection provided by the cooling channels generally varies along the flow direction, average values appear fairly accurate in the present cases.

For laminar flow in circular pipes an analytic solution of the Nusselt number can be derived which gives

$$\text{Nu} = 3.66.$$  \hspace{2cm} (32)

The Nusselt number for turbulent flow is given by Gnielinski according to [2, p. 441]

$$\text{Nu} = \frac{(f_D/8)(\text{Re} - 1000)\text{Pr}}{1 + 12.7(f_D/8)^{1/2}(\text{Pr}^{2/3} - 1)},$$  \hspace{2cm} (33)

which is valid in the range of $0.5 \lesssim \text{Pr} \lesssim 2000$ and $3 \times 10^3 \lesssim \text{Re} \lesssim 5 \times 10^6$. The Prandtl number $\text{Pr}$ is defined by the heat capacity $c_p$, viscosity $\mu$ and thermal conductivity $k$ as

$$\text{Pr} = \frac{c_p \mu}{k}.$$  \hspace{2cm} (34)

The problem (28)–(29) further requires boundary conditions in order to involve the RF heat source and power dissipation to air. They are given by the following two equations

$$\mathbf{n} \cdot (k_{\text{solid}} \nabla T) = \frac{1}{2} \Re \{Z_s \} |\mathbf{H}|^2, \hspace{1cm} \mathbf{r} \in \partial \Omega_{\text{solid}} \cap \partial \Omega_{\text{vac}},$$  \hspace{2cm} (35)

$$\mathbf{n} \cdot (k_{\text{solid}} \nabla T) = h_{\text{amb}} (T - T_{\text{amb}}), \hspace{1cm} \mathbf{r} \in \partial \Omega_{\text{solid}} \setminus \partial \Omega_{\text{vac}},$$  \hspace{2cm} (36)

where $\mathbf{H}$ is the magnetic field intensity at the common boundary between solid and vacuum domain, $Z_s$ is the frequency and material dependent surface impedance according to Sec. 3.1, $T_{\text{amb}}$ is the ambient temperature, and $h_{\text{amb}}$ represents the convection heat transfer coefficient provided by the environmental air. A detailed tutorial for simulating self consistently the heat transfer in solids combined with one-dimensional flows using COMSOL is provided in [3].

Fig. 17 shows the temperature distribution within the solid and cooling pipes. An averaged RF power of 1 kW in total is applied at the input ports of the TD26.R05_CC prototype assembly while the water flows with an average velocity of 31/min through the single cooling circuit. In order to account for the air convection around the structure, a constant heat transfer coefficient of 5 W/(m²K) is assumed. Furthermore, an ambient temperature of 28 degC is considered while the coolant provides 27 degC at the inlet. Despite of the high thermal conductivity of copper, the heat is very much localized around the irises as illustrated in Fig. 16(b). Given the simulated heat exchange $Q_{\text{wall}}$ and temperature difference $\Delta T$ between fluid and surrounding solid, the corresponding heat flux and heat transfer coefficient can be derived from (30). The implementation in COMSOL looks as:

$$q_{\text{pipe}} = \frac{n_{\text{ipfl}} \cdot Q_{\text{wall}}}{\pi \cdot \text{Pipe Diameter}}$$

$$h_{\text{pipe}} = \frac{q_{\text{pipe}}}{T_{\text{Text}} - T_{\text{2}}}$$

Here, $n_{\text{ipfl}}$ refers to the non-isothermal pipe flow module which provides the variables $T_2$ and $\text{Text}$ being the local temperature of the fluid and surrounding solid, respectively. It further provides access to the heat exchange per length $Q_{\text{wall}}$. The convection heat transfer coefficient of the water only slightly varies over the cooling path as depicted in Fig. 16(d). Consequently, it could be assumed constant for the sake of simplicity. This approach is
Figure 16: Thermal study of the TD26_R05_CC prototype assembly, with the input RF power, water flow rate, and air convective heat transfer coefficient being 1 kW, 3 l/min and 5 W/(m²K), respectively. Temperature profile (a) of the coolant and (b) within the solid. (c) Heat flux and (d) heat transfer coefficient between water and solid.

Generally used in ANSYS where the heat transfer coefficient is estimated from (31) and (33) assuming a constant average flow, thus, constant Reynolds number according to (24) as well as the smooth pipe approximation of (27) to derive the friction factor $f_D$. These assumptions are in particular valid towards larger flow rates and smaller RF input power as concluded from Fig. 15. However, even for an input RF power of 1.5 kW and a flow rate of 1 l/min which is close to the transition from turbulent to laminar flow, the Nusselt number varies by approximately 20%.

A more detailed study about the thermal behaviour of the TD26_R05_CC prototype assembly by means of varying RF input power and water flow rates is shown in Fig. 18. For an
average RF power of 1.5 kW at the input ports, the temperature around the irises may rise by several tens of degree Celsius. Starting from a flow rate of 31/min the temperature rise becomes saturated. It is thus not efficient and with respect to cavitation phenomena even unfavorable to further increase the flow rate. In order to overcome this drastic temperature increase at relatively high average RF power, the cooling system must be located closer to the irises. This approach is used for example by the CLIC-G structures. The convection heat transfer coefficient varies linearly with the flow rate and is approximately constant along the cooling path for all considered cases. The corresponding heat dissipated into the environmental air shown in Fig. 13(b) in the beginning of this section. Its contribution is marginal.
Figure 18: Thermal study of the TD26_R05_CC prototype assembly with varying RF input power and water flow rate. An ambient temperature of 28 degC is assumed while the water temperature at the inlet is 27 degC. (a) Water temperature at the outlet. (b) Maximum temperature provided by the structure. (c) Heat flux between the coolant and surrounding solid. (d) Convection heat transfer coefficient provided by the coolant. The latter two quantities are averaged over the single cooling circuit with the error, with error bars accounting for the corresponding minimum and maximum along the path.

in comparison to the heat evacuated by the coolant which amounts roughly 98% of the RF power dissipated in the structure wall [Fig. 13(a)].
3.3 Structural Deformation

The deformation of an isotropic linear elastic solid with the material specific Youngs modulus $E$, Poisson ratio $\nu$, and density $\rho$ can be described by Navier’s equations according to

$$\frac{E}{2(1+\nu)} \left( \frac{1}{1-2\nu} \nabla (\nabla \cdot \mathbf{u}) + \nabla^2 \mathbf{u} \right) + \mathbf{f} = \rho \frac{\partial^2 \mathbf{u}}{\partial t^2} \quad \mathbf{r} \in \Omega_{\text{solid}}, \quad (37)$$

Figure 19: Temperature distribution of the coolant along various paths applied on the TD26_R05_CC assembly, with the input RF power, water flow rate, and air convective heat transfer coefficient being 1 kW, 3 l/min and 5 W/(m²K), respectively.
Figure 20: Thermal study of the TD26.R05.CC prototype assembly with different cooling paths according to Fig. 19. The input RF power and air convective heat transfer coefficient are 1 kW and 5 W/(m²K), respectively. An ambient temperature of 28 degC is assumed while the water temperature at the inlets is 27 degC. (a) Water temperature at the outlet. (b) Maximum temperature provided by the structure.

where \( \mathbf{u} \) corresponds to the unknown displacement vector field. The force per unit volume due to thermal expansion is given by [8]

\[
f = -\frac{E}{3(1-2\nu)}\alpha_T \nabla T,
\]

with \( \alpha_T \) as the temperature coefficient of linear thermal expansion. As in the previous section, only the stationary case shall be considered, that is the right handsight of (37) vanishes. Furthermore a boundary condition must be added in order to allow for a unique solution. The structure is fixed on one end by forcing zero azimuthal and longitudinal displacement along the highlighted circular edges in Fig. 21(a) and (b) for the TD26.R05.CC and TD26.R05_SS prototype assembly, respectively. This particular choice of boundary conditions which are conform to the realistic case, ensure that the beam remains at the center when entering the the structure and deviates from the center while propagating through the structure.

\[
\text{dispx} = \frac{1}{4} \left( \text{at3}(\text{offset,offset},z,u) + \text{at3}(-\text{offset},\text{offset},z,u) + \text{at3}(-\text{offset},-\text{offset},z,u) + \text{at3}(\text{offset},-\text{offset},z,u) \right)
\]

\[
\text{dispy} = \frac{1}{4} \left( \text{at3}(\text{offset,offset},z,v) + \text{at3}(-\text{offset},\text{offset},z,v) + \text{at3}(-\text{offset},-\text{offset},z,v) + \text{at3}(\text{offset},-\text{offset},z,v) \right)
\]

3.4 Moving Mesh and Port Displacement
todo
4 Mesh

4.1 Preparation

4.2 Conformal and Non-Conformal Approach

4.3 Convergence

5 Studies

6 Conclusions

Figure 21: Boundary condition for the structural deformation due to thermal expansion. The structures are fixed on one end by forcing zero azimuthal and longitudinal displacement along the highlighted circular paths. (a) TD26_R05_CC and (b) TD26_R05 prototype assembly.
Figure 22: Radial deformation due to thermal expansion of the TD26_R05_CC assembly with single circuit cooling provided an input RF power, water flow rate, and air convective heat transfer coefficient of 1 kW, 31/min and 5 W/(m²K), respectively. An ambient temperature of 28 degC is assumed while the water temperature is set to 27 degC at the inlet. (a) Perspective view. (b) Cross-sectional view at the structure exit. The center axis displacement along the structure is measured by superposing the transverse displacements over four probe lines close to the irises and highlighted by the black dots.

A Settings

References


Figure 23: Distance between the beam and the center axis of the deformed TD26_R05_CC prototype assembly with single circuit cooling. The input RF power and water flow rate are varied while the air convective heat transfer coefficient, ambient temperature and inlet temperature are fixed to 5 W/(m²K), 28 degC, and 27 degC, respectively.


Figure 24: Cross-sectional view of the radial deformation due to thermal expansion of the TD26_R05_CC assembly at the exit provided various cooling paths, each with an input RF power, water flow rate, and air convective heat transfer coefficient being 1 kW, 3 l/min and 5 W/(m²K), respectively. An ambient temperature of 28 degC is assumed while the water temperature is set to 27 degC at the corresponding inlets.

Table 2: Parameters of the TD26_R05_CC prototype assembly.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cell_Period</td>
<td>8.332 mm</td>
<td>Cell period.</td>
</tr>
<tr>
<td>Pipe_X1</td>
<td>0.0 mm</td>
<td>1st x-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_X2</td>
<td>45.0 mm</td>
<td>2nd x-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Y1</td>
<td>45.0 mm</td>
<td>1st y-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Y2</td>
<td>0.0 mm</td>
<td>2nd y-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Z1</td>
<td>−114.393 mm</td>
<td>1st z-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Z2</td>
<td>−86.0 mm</td>
<td>2nd z-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Z3</td>
<td>86.0 mm</td>
<td>3rd z-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Z4</td>
<td>111.307 mm</td>
<td>4th z-coordinate of cooling path.</td>
</tr>
<tr>
<td>Pipe_Diameter</td>
<td>8 mm</td>
<td>Diameter of cooling channel.</td>
</tr>
<tr>
<td>Pipe_SurfRoughness</td>
<td>0.0025 mm</td>
<td>Surface roughness of cooling channel.</td>
</tr>
<tr>
<td>Pipe_Qvol</td>
<td>6 l/min</td>
<td>Coolant volume flow rate.</td>
</tr>
<tr>
<td>T_ambient</td>
<td>28 degC</td>
<td>Ambient temperature.</td>
</tr>
<tr>
<td>T_inlet</td>
<td>27 degC</td>
<td>Inlet temperature of the coolant.</td>
</tr>
<tr>
<td>HTC_CuAir</td>
<td>5 W/(m²K)</td>
<td>Heat transfer coefficient Cu to air.</td>
</tr>
<tr>
<td>RF_PhaseOffset</td>
<td>0 deg</td>
<td>Phase correction between both input signals.</td>
</tr>
<tr>
<td>RF_PortName</td>
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<td>Port name.</td>
</tr>
<tr>
<td>RF_Power</td>
<td>2100 W</td>
<td>Total RF power entering the input ports.</td>
</tr>
</tbody>
</table>
Figure 26: Configuration of mesh partitions. (a) The original model before applying mesh configurations. (b) Horizontal and vertical transverse planes to split both, the structure and vacuum part in several individual mesh domains in order to accelerate the meshing. (c) Geometry parts for the cooling channels are dedicated for mesh refinements, only. (d) The Model as it is available to the physics interface. Internal boundaries and cooling channels are not present anymore.
Figure 27: 

Figure 28: (a) The cell-to-cell phase advance and (b) internal reflection for different prototype assemblies. The quantities are derived from the periodic voltage standing wave method [6].
Figure 29: Distance between the beam and the center axis of the deformed TD26_R05_CC prototype assembly with single circuit cooling. The input RF power and water flow rate are varied while the air convective heat transfer coefficient, ambient temperature and inlet temperature are fixed to 5 W/(m²K), 28 degC, and 27 degC, respectively.
Figure 30: Distance between the beam and the center axis of the deformed TD26_R05_CC prototype assembly with single circuit cooling. The input RF power and water flow rate are varied while the air convective heat transfer coefficient, ambient temperature and inlet temperature are fixed to $5\,\text{W/(m}^2\text{K)}$, 28 degC, and 27 degC, respectively.
Figure 31: Thermal study of the TD26_R05_CC prototype assembly with different cooling paths according to Fig. 19. The input RF power and air convective heat transfer coefficient are 1 kW and 5 W/(m²K), respectively. An ambient temperature of 28 degC is assumed while the water temperature at the inlets is 27 degC. (a) Cell-to-cell phase advance and (b) Internal reflection, both based on the periodic voltage standing wave ratio method.
Figure 32: Thermal study of the TD26_R05_CC prototype assembly with different cooling paths according to Fig. 19. The input RF power and air convective heat transfer coefficient are 1 kW and 5 W/(m²K), respectively. An ambient temperature of 28 degC is assumed while the water temperature at the inlets is 27 degC. (a) Cell-to-cell phase advance and (b) Internal reflection, both based on the periodic voltage standing wave ratio method.
Table 3: Parameters of the TD26_R05_SS prototype assembly (super structure).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cell_Period</td>
<td>8.332 mm</td>
<td>Cell period.</td>
</tr>
<tr>
<td>Pipe_X11</td>
<td>17.5 mm</td>
<td>1st x-coordinate of cooling path, structure 1.</td>
</tr>
<tr>
<td>Pipe_X21</td>
<td>44.0 mm</td>
<td>2nd x-coordinate of cooling path, structure 1.</td>
</tr>
<tr>
<td>Pipe_X12</td>
<td>17.5 mm</td>
<td>1st x-coordinate of cooling path, structure 2.</td>
</tr>
<tr>
<td>Pipe_X22</td>
<td>44.0 mm</td>
<td>2nd x-coordinate of cooling path, structure 2.</td>
</tr>
<tr>
<td>Pipe_Y11</td>
<td>44.0 mm</td>
<td>1st y-coordinate of cooling path, structure 1.</td>
</tr>
<tr>
<td>Pipe_Y21</td>
<td>17.5 mm</td>
<td>2nd y-coordinate of cooling path, structure 1.</td>
</tr>
<tr>
<td>Pipe_Y12</td>
<td>45.0 mm</td>
<td>1st y-coordinate of cooling path, structure 2.</td>
</tr>
<tr>
<td>Pipe_Y22</td>
<td>17.5 mm</td>
<td>2nd y-coordinate of cooling path, structure 2.</td>
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<tr>
<td>Pipe_Z11</td>
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<td>Pipe_Z21</td>
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<td>Pipe_Z41</td>
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<td>Pipe_Z22</td>
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<td>Pipe_Z42</td>
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<td>Diameter of cooling channel.</td>
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<td>Surface roughness of cooling channel.</td>
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<td>T_inlet</td>
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<td>Inlet temperature of the coolant.</td>
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<td>HTC_CuAir</td>
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<td>Heat transfer coefficient Cu to air.</td>
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<td>RF_PhaseOffset</td>
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<td>Port name.</td>
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<tr>
<td>RF_Power</td>
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<td>Total RF power entering the input ports.</td>
</tr>
<tr>
<td>Name</td>
<td>Description</td>
<td>View</td>
</tr>
<tr>
<td>------------------</td>
<td>-----------------------------------------------------------------------------</td>
<td>-----------------------------</td>
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<tr>
<td>Beam Line</td>
<td>Edge along the nominal particle beam.</td>
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<tr>
<td>Beam Cylinder</td>
<td>Boundaries of the cylinder enveloping the beam line.</td>
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<tr>
<td>Cooling Pipe</td>
<td>Edges associated with the cooling circuit.</td>
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</tr>
<tr>
<td>Inner Cooling Pipe</td>
<td>Edges associated with the cooling circuit which are inside the structure.</td>
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Table 5: Selections of the TD26.R05.CC prototype assembly.

<table>
<thead>
<tr>
<th>Name</th>
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</thead>
<tbody>
<tr>
<td>Iris</td>
<td>Boundaries associated with the irises using a cylindrical selection.</td>
<td></td>
</tr>
<tr>
<td>Waveguide</td>
<td>Boundaries associated with waveguides and couplers using a cylindrical selection. The iris boundaries are excluded.</td>
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</tr>
<tr>
<td>Inner Surface</td>
<td>Boundaries between the vacuum part and the structure.</td>
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<tr>
<td>Outer Surface</td>
<td>Remaining boundaries of the structure which are not associated with Inner Surface.</td>
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</table>
Table 6: Selections of the **TD26_R05_CC** prototype assembly.

<table>
<thead>
<tr>
<th>Name</th>
<th>Description</th>
<th>View</th>
</tr>
</thead>
<tbody>
<tr>
<td>Port 1</td>
<td>Boundary of the first input port.</td>
<td></td>
</tr>
<tr>
<td>Port 2</td>
<td>Boundary of the second input port.</td>
<td></td>
</tr>
<tr>
<td>Port 3</td>
<td>Boundary of the first output port.</td>
<td></td>
</tr>
<tr>
<td>Port 4</td>
<td>Boundary of the second output port.</td>
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</tbody>
</table>
Table 7: Selections of the TD26_R05_CC prototype assembly.

<table>
<thead>
<tr>
<th>Name</th>
<th>Description</th>
<th>View</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beam Port 1</td>
<td>Boundary of the first beam pipe port.</td>
<td></td>
</tr>
<tr>
<td>Beam Port 2</td>
<td>Boundary of the second beam pipe port.</td>
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</tr>
<tr>
<td>Vacuum</td>
<td>Domains or boundaries associated with the inner volume filled with vacuum.</td>
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</tr>
<tr>
<td>Cavity</td>
<td>Domains or boundaries associated with the structure.</td>
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