

THERMO-FLUID STUDY OF THE UPC RACE-TRACK MICROTRON COOLING SYSTEM*

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Abstract

The cooling system of the race-track microtron (RTM), which is under construction at the Universitat Politècnica de Catalunya (UPC), has been simulated by means of a computational fluid dynamics (CFD) software. The hydraulic and thermal performance of the system for various operation conditions has been studied. Firstly, the hydraulic model has been validated by comparison with experimental measurements at different flow rates. Then, the cooling fluid temperatures and the pressure losses of the system have been determined and the capacity of the current design to remove the generated heat at nominal power has been confirmed. Finally, the maximum and average wall temperatures and heat transfer coefficients inside the accelerating structure have been calculated. These results have allowed us to localize sections of the cooling system with a low convection due to detached flows where, therefore, a risk of zones of high temperatures exists. An optimization of the cooling circuit with the aim to reduce such high temperature zones has been proposed.

12 MEV RACE-TRACK MICROTRON AND ITS COOLING SYSTEM

The UPC in collaboration with the Skobeltsyn Institute of Nuclear Physics of the Moscow State University and CIEMAT (Madrid) is building a compact electron accelerator of race-track microtron (RTM) type with the maximal beam energy 12 MeV. Its main envisaged application is Intraoperative Radiation Therapy. Modification of this low power consumption machine can be used also for cargo inspection and industrial radiography. The design of the accelerator is described in [1], the course of its systems development was reported in a number of papers, see for example [2-4]. Currently the RTM is at the final phase of its construction.

The UPC 12 MeV RTM is a pulsed accelerator with the output beam energies 6, 8, 10 and 12 MeV, energy gain per turn equal to 2 MeV approximately and low average current, of the order of 10-100 nA. The electron beam is produced in a module called accelerator head, its 3D view is given in Fig. 1 where main elements, like 180° bending magnets, usually called end magnets, and the accelerating

structure (linac) are shown. All magnets are permanent magnets and therefore do not consume power. These elements are fixed on a rigid platform placed inside a vacuum chamber.

The cooling of the RTM is provided by a chiller that comprises a positive displacement pump with a discharge pressure limited to around $6.8 \cdot 10^5$ Pa (6.8 bar) that can be regulated to reach a maximum flow rate of $2.83 \cdot 10^{-4}$ m³/s if necessary. The main function of the RTM cooling system is to provide a safe and stable operation of all accelerator systems, in particular guarantee a temperature stability of $\pm 1^\circ\text{C}$ in the end magnets and linac. In addition, at the stationary regime of RTM operation the temperature differences between different parts of the end magnets and linac must not exceed the same limit of $\pm 1^\circ\text{C}$.

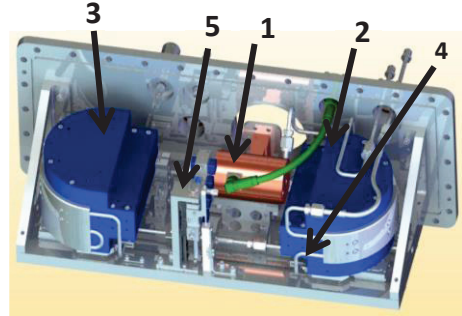


Figure 1: 3D view of the accelerator head of the 12 MeV RTM. The numbered elements are: 1- accelerating structure (linac), 2 and 3 – end magnets M1 and M2, respectively, 4 – cooling tube, 5 – extraction dipoles.

In the present paper we study the performance of the cooling system of the RTM accelerator head. Chilled distilled water is sent through the inlet collector and inlet feedthrough to the vacuum box. Firstly, the cooling circuit, which is a tube of 4 mm internal diameter, passes along the rear walls of the two end magnets, M1 and M2, as one can see in Fig. 1. The tube is partially covered with jackets in order to achieve a good thermal contact with the magnet bodies. After the M2 magnet the cooling tube is connected to the inlet of the linac which has four 4 mm diameter longitudinal cooling channels connected in series. Finally, the outlet of the linac is connected to the output feedthrough and then to the output collector. The connecting tubes are made of stainless steel and also have the 4 mm internal diameter. The total length of the cooling channels and connecting tubes inside the vacuum chamber is about 2 m.

* Work supported by the 2014 SGR846 of AGAUR (Generalitat de Catalunya)

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According to the RTM Technical Design Report [5] the maximum total power which must be removed by the cooling system from all RTM systems is about 4 kW, in particular approximately 1 kW is dissipated in the linac and 0.01 kW in the end magnets. The nominal water flow rate was fixed at $\dot{m} = 0.05$ kg/s. The accelerating structure is normal conducting, thus the dissipated heat power is due to the currents which are induced in its copper walls by the magnetic component of the RF electromagnetic wave. The magnets are permanent and the dissipated power is due to beam losses only.

The accelerating structure consists of four accelerating and three coupling resonance cavities. According to calculations of the detailed distribution of the RF power losses in the cavity walls carried out in Ref. [6] the RF power dissipated in the 1st, 2nd, 3rd and 4th linac segments is equal to 189.5 W, 273.7 W, 273.7 W and 263.1 W, respectively. Each segment comprises the corresponding resonant cavity, in addition the 1st and 4th segments contain annular passages from one cooling channel to another.

As a part of preliminary tests of RTM systems, measurements of the pressure loss in the cooling system of the RTM accelerator head without RF power dissipation in the linac were carried out. For this the inlet and outlet feedthroughs were connected to the chiller and two pressure sensors (model PT5404 from ifm®) and a flowmeter (model PF3W520-F03-1-R from SCM®) between the chiller and the vacuum chamber were mounted. During the measurements the data from the pressure sensors was read and processed with a 10 bits Arduino® Ethernet board. The measured values of the pressure loss for different flow rates are given in Fig. 2.

CFD MODEL AND HYDRAULIC PERFORMANCE

The CFD model of the RTM cooling system that comprises the fluid domain (see Fig. 3) has been developed using the ANSYS CFX® version 14.5 [7]. The annular passages connecting the cooling channels in the 1st and 4th segments are clearly seen in Fig. 3. All no slip pipe walls were considered as adiabatic with the exception of those which are in contact with the end magnets or represent the linac cooling channels. In solving the energy transport equation the viscous dissipation was not taken into account because the friction effect is negligible in comparison with the temperature increase due to heat transfer from the solid surface towards the moving fluid. In the simulations the inlet flow condition was set as a function of the distilled water flow rate at $T_{inlet} = 10^\circ\text{C}$ and the outlet flow condition was set to the average static pressure $p_{out} = 0$ Pa in relative terms. The Shear Stress Transport (SST) turbulence model was used [8]. After a preliminary model sensitivity analysis it was decided to limit the size of the mesh by about 6.2M elements.

Using this model we performed simulations of the total pressure loss Δp in the system in the range from a minimal mass flow rate of $\dot{m} = 0.042$ kg/s until a maximum

one of $\dot{m} = 0.075$ kg/s. For the considered average flow velocities (from $V=3.3$ m/s to $V=6.0$ m/s), the Reynolds number varies from $Re=10137$ to $Re=18247$, so that the water flow is always turbulent.

The obtained results indicate that Δp increases with the flow rate, as it is shown on the plot in Fig. 2, in very good accordance with the experimental results. For the maximal flow rate, the pressure loss is $\Delta p = 6.4 \cdot 10^5$ Pa. Since the location of the pressure sensors was not exactly at the inlet and outlet of the cooling system the measured values have been corrected correspondingly. The error bars correspond to the $\pm 10\%$ interval around the corrected measured values.

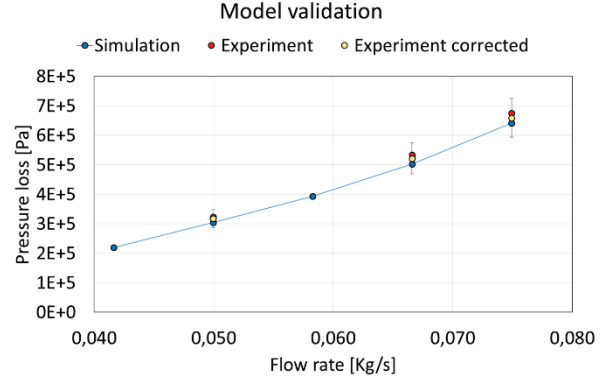


Figure 2: Pressure loss in the cooling system as a function of the flow rate. The continuous curve fits the simulation results indicated with blue dots. Red dots are measured values and yellow dots are corrected measured values.

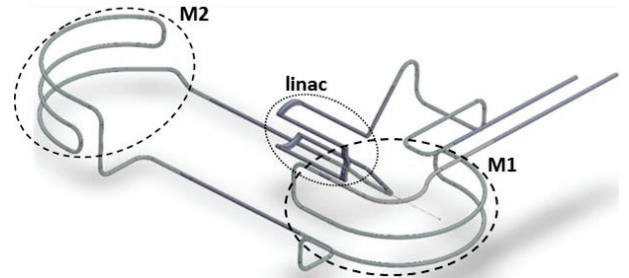


Figure 3: Fluid domain representing the internal diameter of the water piping.

THERMAL PERFORMANCE

In the thermo-fluid analysis of the cooling system the distribution of the heat power transfer specified in above was used. We performed numerical simulations of the following characteristics:

- Flow temperature increase along the cooling circuit, $\Delta T = T_{outlet} - T_{inlet}$.
- Pipe wall temperature distributions, in particular the average and maximum values, T_{av} and T_{max} , at each linac segment.
- Pipe wall heat transfer coefficients, in particular the average and maximum values, h_{av} and h_{max} .
- Flow velocity distributions.
- Streamlines of the water flow inside the linac channels.

Some of these results are given in Table 1. The maximum wall temperatures T_{max} are always in the 1st segment, in the rest of the segments the temperatures are very close between them and are considerably lower than in the 1st segment. As far as the average wall temperature T_{av} is concerned, for any flow rate its maximum value is in the 3rd segment and the minimum value is in the 1st one. The interval of values of the average temperatures in the linac segments is approximately 1.5° C at the minimum mass flow rate $\dot{m} = 0.042$ kg/s and is about 0.7° C at the maximum flow rate $\dot{m} = 0.075$ kg/s.

Table 1: Flow Temperature Increase ΔT , Maximum Wall Temperatures T_{max} , and Average Heat Transfer Coefficients h_{av} for Different Water Flow Rates

\dot{m} [kg/s]	ΔT_{sim} [°C]	T_{max} [°C]	h_{av} [W/(m ² K)]
0.042	5.78	26.1	5832
0.050	4.82	22.3	6711
0.058	4.15	20.1	7556
0.067	3.64	19.8	8392
0.075	3.25	18.4	9129

The explanation of the fact that the maximum wall temperatures are in the 1st linac segment, even though the heat power generated there is considerably lower than in the rest of the segments, can be deduced from the plots in Fig. 4 where water streamlines through the conduits are plotted being colored as a function of the flow velocity in Fig. 4a and the flow temperature in Fig. 4b. Due to the abrupt change of direction, the flow experiences a boundary layer detachment in the 1st segment entering from the circular conduit into the annular passage and making a 90° turn. Thus, a region with stagnant water and very low velocity in the region close to the wall is created and, consequently, the local convective heat transfer reduces significantly. Such behavior takes place in the two annular connections in the 1st segment and also in the connection path in the 4th segment. As a result, regions with defective flow velocities below 4 m/s appear in these segments.

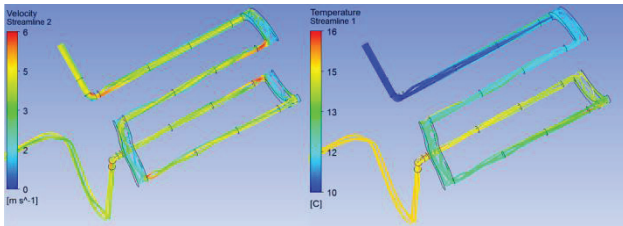


Figure 4: Streamlines of the water flow inside the linac conduits colored as a function of the velocity (a) and the temperature (b).

DISCUSSION OF THE RESULTS AND CONCLUSIONS

We studied the hydraulic and thermal performance of the cooling system of the accelerator head of the UPC 12 MeV RTM. Numerical simulations of the cooling system with the CFD code were performed. For this a 3D model

representing the fluid domain with a mesh with 6.2M elements was developed. The accuracy of the model was checked for a few values of the mass flow rate by comparison of the pressure loss simulation results with the measured ones, and quite a good agreement was found.

The results of the simulations validate the cooling system design. We obtained that the difference of the average temperatures between the linac segments is below 1° C at the maximum flow rate $\dot{m} = 0.075$ kg/s. Thus, the RTM design requirement, that the temperature difference between different segments of the linac does not exceed the limit of $\pm 1^\circ$ C, is met for the maximal flow rate only.

The study of the water streamlines showed that the appearance of regions with high temperatures in the 1st linac segment is due to boundary layer detachments of the flow and low flow velocities there. These effects are caused by 90° abrupt bending of the conduits inside the linac in this segment. The temperature in this region can be reduced by modifying the cooling channel geometry at the transition between the circular pipes and the annular arms, in particular by filleting the two surfaces at their contact edges in order to create a rounded smooth transition that would help the flow to turn without detaching from the surfaces. This improvement of the cooling pipes geometry can be implemented in a future upgrade of the RTM design.

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