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Analysis of the integral heat transfer characteristics of the MEGAPIE target

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Introduction

The development of the target of an ADS reactor requires detailed knowledge on the cooling of thin-walled thermally highly-loaded surfaces. The related R+D work concentrates among others on developing improved CFD modeling and on detailed experimental investigations of model problems up to full scale target experiments in the KALLA laboratory [1]. In parallel, there is the MEGAPIE project in which a modeled target is developed in an international co-operation [2]. The target shall be irradiated in the SINQ radiation source [3].

Detailed numerical analyses of the window area of the MEGAPIE-target need sufficiently fine grids. With current computers and codes the required detailed analyses can only be performed for some limited areas around the window [4]. Thus, accurate boundary conditions are required for the inlet and outlet of the relatively small computational domain. Such boundary conditions can be provided by approximate 3d simulations of the complete target module which are also suited to investigate more integral heat transfer problems within the module. A number of certain physical models like several turbulence models, heat exchanger and thermal structure models make the transient 3d thermal and fluid-dynamical code FLUTAN [5] attractive to such investigations. In detailed local analyses it was demonstrated that a symmetric flow to cool the window is not acceptable due to the existence of a stagnation point [4]. In addition first complete module analyses showed the existence of strong thermal coupling between the cold and hot fluid paths which may increase the anyway very critical window temperature [6].

The objective of these numerical simulations for a complete target module is to investigate the flow and cooling behavior in a target module with a bypass jet flow across the window. We started with 3d calculations for a simplified representation of an earlier available module geometry, and switched later over to a simplified representation of a newer design [7].

Principles of model geometry

The FLUTAN calculations are basing on a simplified modeling of the complete MEGAPIE target module (Fig. 1). The proton beam is entering from below through the hemispherical steel window into the fluid domain. The spatial distribution of the power deposition by the beam of 541 kW in the liquid lead bismuth and in the steel structures at the lower 30 centimeters of the module is prescribed according to [8]. The window is cooled by liquid lead-bismuth flowing downwards in the annular space and upwards inside the inner pipe. Both channels are separated by the guide tube. The 3d thermal conduction in the guide tube is calculated for materials with thermal conductivities ranging from 1 to 16 W/(m*K). The window is in addition cooled by a bypass jet flow generated by a nozzle near the window. The main and the bypass flows are driven by separate electromagnetic pumps. The heat is removed from the liquid lead-bismuth by one row of double walled heat exchanger pipes.

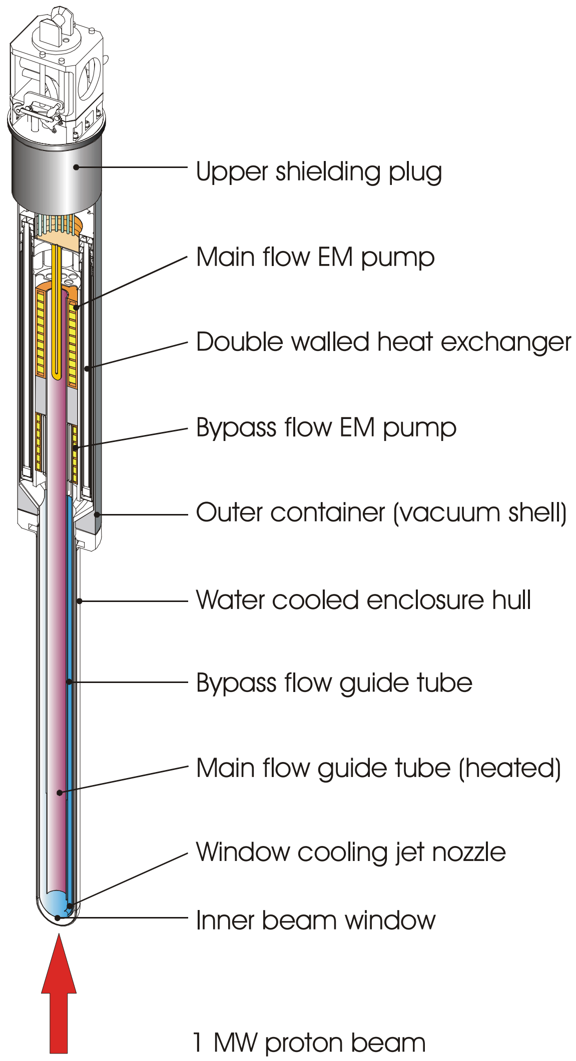

Fig. 1: The MEGAPIE target conceptual design 2001 [7].

The modeling of the first calculations follows the design of February 2000, Fig. 2. The bypass in form of a jet flow across the window surface is modeled by a source with a certain mass flow rate at the position of the nozzle outlet (position of the velocity maximum near the center of the window). The pipe feeding that nozzle is not recorded in the simulation. The annular heat exchanger at the upper end of the target module is modeled by a heat sink distributed uniformly over the heat exchanger cross section, by the correct flow area, and by the two grid plates carrying the (not recorded) vertical heat exchanger pipes. The fluid temperature at the heat exchanger exit is prescribed to be 223 ^{O}C. The standard k-ε model is used with a constant turbulent Prandtl number of σ_t=0.9 to calculate the turbulent heat transfer. Simulations for a concept with bypass require 3d representations. We consider a forced convection case for the first target concept and a natural convection case for the new reference geometry.

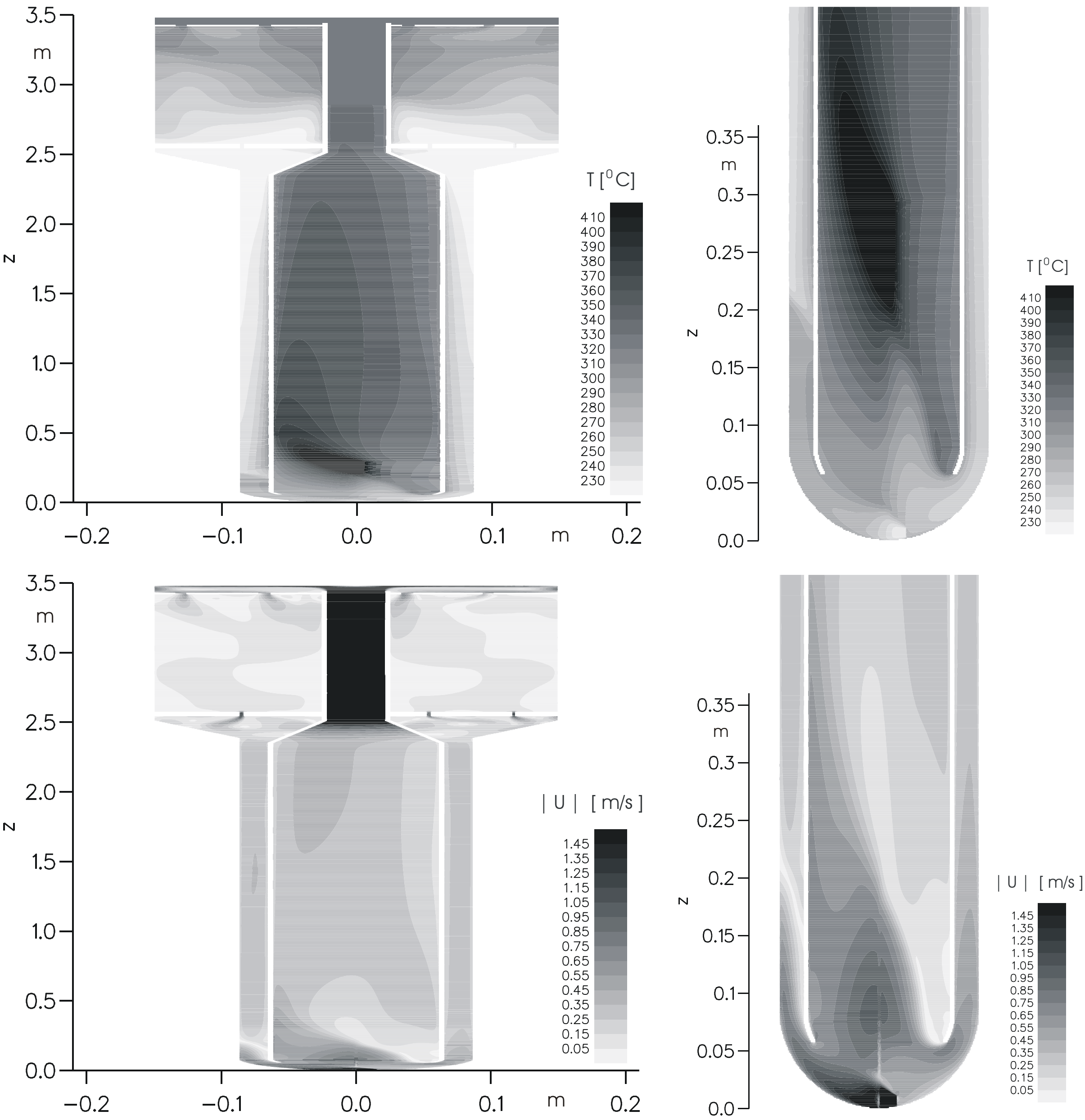

Fig. 2: Temperature and modulus of velocity in whole module and in bottom part. Old geometry, forced convection, bypass flow rate 3.5 kg/s, thermal conductivity 16 W/(m*K) inside guide tube.

Original geometry, forced convection, with bypass

The forced convection results with a bypass of 3.5 kg/s, using a thermal conductivity of 16 W/(m*K) inside the guide tube, are shown in Fig. 2. The pump for the downcomer flow was specified to achieve a flow rate of 35 kg/s. The maximum value of the temperature is with 483° C compared to the cases without bypass [6] far below the boiling temperature. Nevertheless, this is still a quite large and localized value, because the temperatures of the structures should be kept as homogeneously distributed as possible and should not exceed about 400° C at the window for

safety reasons. A thermal power of 152 kW (29% of the power deposition) is conducted through the guide tube from the hot inside flow to the cold annular downcomer flow.
The velocity field shows an asymmetry between the left bottom part of the downcomer and the right part. This is due to the fact that part of the bypass flow injects hot fluid through the gap between window and guide tube on the left into the downcomer against the downwards coming cold fluid. The zero-line in the velocity field in the annulus separates upwards and downwards coming fluid. Thus, the fluid has to redistribute azimuthally in the annulus and consequently forms there a complex 3d vortex. This lower vortex induces a vortex with opposite rotation direction in the annulus above, which is the reason for asymmetries in the u and T-field in the downcomer up to the heat exchanger outlet. The entity of this secondary flow will depend on the flow rate of the bypass, on the nozzle design and orientation, and on the gap widths between guide tube and hemispherical window. This asymmetry makes any local calculation for the window area problematic.

More recent geometry, natural convection, with forced bypass flow
In the next design the upper wider target part with the heat exchanger and the disposition of the pumps was modified in order to reduce the irreversible pressure losses due to the inner contour of the old guide tube, Fig. 3. Calculations were done for natural convection of the main flow with a forced bypass flow rate of 1.18 kg/s and with two different thermal conductivities for the guide tube.

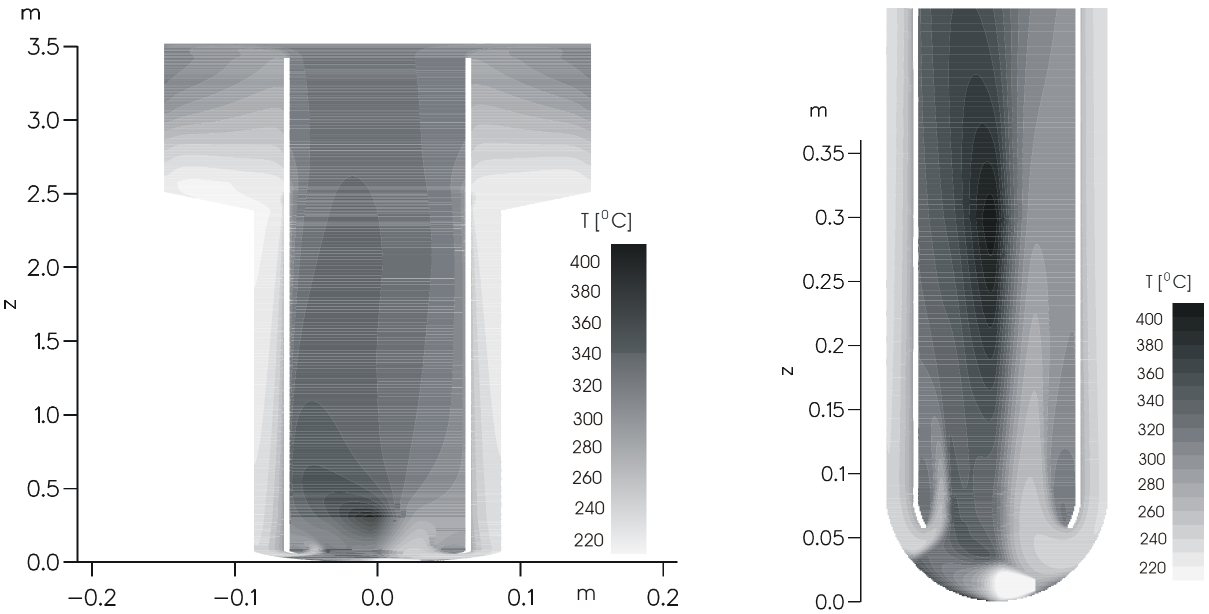

Fig. 3: Temperature field in whole module and in bottom part. New geometry, natural convection, bypass flow rate 1.18 kg/s, thermal conductivity 16 W/(m*K) inside guide tube.

The results show, the heat transferred through the guide tube on the temperature field in this concept is 27.6% at λ=16 W/(m*K) versus 4.3% at λ=1 W/(m*K). Therefore, the window temperature is for the case of λ=16 W/(m*K) about 20% higher, Fig. 3, in comparison with the other case [6]. The maximum fluid temperature remains in both cases also below the boiling temperature. In fact, a mass flow rate of about 40 kg/s is achieved inside the guide tube. This value is even higher than the mass flow rate given in forced convection for the original geometry.

An asymmetry in the flow and temperature field in the downcomer exists also in this design, but it is weaker than in the old geometry mainly because a lower bypass flow rate was chosen in this calculation. Nevertheless, the asymmetries reach again far up in the downcomer towards the heat exchanger area. Thus, also in these cases detailed investigations for the window area need careful treatment of the inlet and outlet conditions at the boundaries of the computational domain.

Conclusions

The 3d FLUTAN calculations with a simplified modeling of the complete MEGAPIE target module for the first design concept confirm that the window can in principle be cooled by using forced convection combined with a forced bypass jet flow across the window. The thermal conduction in the guide tube turned out to be a crucial feature to determine the coolant temperature at the window. The 3d investigations with the next design concept show that the influence of the thermal conduction in the guide tube is only slightly smaller compared to the results for the old geometry concept; this means, reducing the heat transfer through the guide tube could also in this design contribute to the reduction of the window temperature. Following these results one may conclude that after optimization this target window can effectively be cooled by means of a forced bypass jet flow.

The results show also, that the bypass flow induces vortices in the guide tube which cause asymmetries in the downcomer up to the heat exchanger area. As this asymmetry may be changed by including more accurate geometry information in the computational model, especially of the bypass piping system, detailed local numerical investigations for the window area will need special approximations for the inlet and outlet conditions at the boundaries of the computational domain, or to record the complete downcomer up to the lower end of the heat exchanger.

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