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Thermal-hydraulics of helium cooled First Wall channels and scoping investigations on performance improvement by application of ribs and mixing devices

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HIGHLIGHTS

- Existing first wall designs and expected plasma heat loads are reviewed.
- Heat transfer enhancement methods are investigated by CFD.
- The results for heat transfer and friction are given, compared and explained.
- Relations for needed pumping power and gained thermal heat are shown.
- A range for the maximum permissible heat loads from the plasma is estimated.

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ABSTRACT

The first wall (FW) of DEMO is a component with high thermal loads. The cooling of the FW has to comply with the material's upper and lower temperature limits and requirements from stress assessment, like low temperature gradients. Also, the cooling has to be integrated into the balance-of-plant, in a sense to deliver exergy to the power cycle and require a limited pumping power for coolant circulation. This paper deals with the basics of FW cooling and proposes optimization approaches. The effectiveness of several heat transfer enhancement techniques is investigated for the use in helium cooled FW designs for DEMO. Among these are wall-mounted ribs, large scale mixing devices and modified hydraulic diameter. Their performance is assessed by computational fluid dynamics (CFD), and heat transfer coefficients and pressure drop are compared. Based on the results, an extrapolation to high heat fluxes is tried to estimate the higher limits of cooling capabilities.

1. Introduction

High pressure helium gas is considered as the coolant of the first wall (FW) in several blanket concepts (Helium Cooled Pebble Bed (HCPB), Helium Cooled Lithium Lead (HCLL), Dual Coolant Lithium Lead (DCLL)) for DEMO and the ITER Test Blanket Modules (TBMs) [1,2]. The main advantages of helium as coolant are safety aspects (chemically inert, no activation, comparatively low effort to remove tritium), no chemical corrosion, and a flexible temperature range (due to no phase change) allowing to match the permissible operation temperature window of the foreseen structural material. Challenges, compared to water or liquid metals, are seen in the lower product of density and heat capacity and lower thermal conductivity, which leads to larger required pumping power for the same cooling.

Up to date helium cooled TBM concepts were designed for plasma-side incident heat flux densities of 500 kW/m^2 steady state. For the HCPB TBM the spatially resolved heat transfer coefficient (HTC) of more than 6000 W/m^2/K was calculated by 3D Reynolds Averaged Navier Stokes CFD in FW channels 15 mm × 15 mm with

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a surface roughness of R_t = 20 μm and helium velocities of about 80 m/s (100 g/s). Applied to a 3D finite element model for heat conduction, an acceptable maximum structural temperature of 542 °C was achieved [3,4]. For the HCLL TBM similar procedures were applied and yielded broadly comparable results [5]. It should be noted, that the TBM coolant flow layout makes use of a bypass, which diverts a significant percentage of the large mass flow rate needed to cool the first wall, without taking it through the rest of the blanket.

The maximum temperature of the FW structural material occurs systematically on the plasma facing surface at the outlet side of the coolant, since the heat flux density from the plasma side is usually considerably higher than from the breeder zone side (the latter depends on the neutron wall load (NWL) and the internal design of the blanket) and is for example assumed as 60 kW/m^2 in [3].

Recent studies [6] for "EU DEMO1" (with impurity seeding of Ar, Xe, W), where it is assumed that the power from the core and the Scrape-Off Layer (SOL) (together 369 MW) are completely radiated, predict a heat flux density distribution with a peak of 450 kW/m^2 at the divertor dome and at the top of the plasma chamber. Another analysis [7], assuming that the power from the SOL (150 MW) is transferred to the FW by thermal charged particles (with an extreme estimate of the power fall off length of λ_q = 170 mm) predicts a power flux density coming from this contribution with a peak of 640 kW/m^2 in the upper part of the inboard. To arrive at an estimate of the total steady-state FW loads (1) a reasonable splitting of the power conducted or convected across the separatrix into radiated power to the wall and power transferred by charged particles to the wall has to be introduced, and (2) other load types have to be investigated (e.g. via charge exchange neutrals). It is thus assumed in the frame of this study that the peak heat flux density on the FW in steady conditions is above 500 kW/m^2, but below 1 MW/m^2. Additional uncertainties are introduced by the FW shape and the magnetic equilibrium, which both are not converged yet. Additionally, much larger short-term heat loads (expressed for instance as heat impact factors) have to be expected due to edge localized modes (ELMs).

This state of achieved FW helium cooling technology and anticipated level of heat loads along with the associated uncertainties has motivated this study to investigate the realistic limits of the maximum wall heat flux density that can be tolerated for helium cooled FW or limiter designs. To this end, certain heat transfer enhancement techniques applicable to the FW cooling channels are studied by CFD. The channel geometry based heat transfer enhancement techniques enlarge the degrees of freedom the FW designers have for the thermal layout. It can be applied either to increase the maximum permissible heat loads, or to gain additional freedom in the choice of coolant mass flow rate, inlet- and outlet temperatures and channel length.

2. Considerations on the FW cooling

2.1. Requirements of FW cooling

A basic requirement of the FW cooling is to keep the used structural material within its applicable temperature window. The upper limit can be defined where a considerable reduction of mechanical strength (yield S_y or creep rupture S_r) occurs. This limit is often taken to be around 550 °C for the blanket/FW candidate material Eurofer. A lower limit exists, when the embrittlement due to neutron irradiation (in combination with the occurrence of impact type loads) is considered. This limit is often cited to be 300 °C for Eurofer, while even higher temperatures 330–350 °C are recommended [8]. A prudent approach would therefore be to keep the coolant entrance temperature above the lower material limit, and adjust coolant flow rate and heat transfer coefficient (as described in Section 2.2) to keep the material temperature below the high limit.

Stresses in the FW are induced from the internal coolant pressure, constrained thermal dilatation and external loads (attachment, electromagnetic). The first two contributions can be in the same magnitude for the 8 MPa helium cooled flat panel type FW with rectangular cooling channel. Considering these two loads, an optimum wall thickness for the channel cover can be found near the 3 mm used in up-to-date designs. The optimum moves to thicker walls for materials with higher thermal conductivity or for higher coolant pressures and to thinner walls for higher heat loads. Thermal induced mechanical stresses in the order of $\sigma = \Delta T \cdot \gamma \cdot E \cdot (1-\nu)$ with the coefficient of thermal expansion γ, the Young modulus E and the Poisson ratio ν occur due to temperature differences ΔT within the structure [9]. The thermal design of the FW should therefore aim at a homogenous temperature distribution to keep thermal stresses low. Stresses from internal pressure can be affected by keeping the channel width (or diameter) small, by choosing a rather rounded cross section, and obviously by reducing the coolant pressure.

The FW receives a considerable fraction of the total power of the blanket (depending on the ratio of plasma sided heat flux and neutron wall load) which is preferably to be supplied to the balance-of-plant system (in series or in parallel to the blanket) at a high exergy. The FW cooling should therefore aim at an as high as possible ratio of outlet temperature vs. inlet temperature T_2/T_1. The required pumping power to circulate the coolant is proportional to the product of volumetric flow rate and the pressure difference (see Eq. (5)) over the cooling circuit. Both quantities should therefore be minimized by the FW thermal-hydraulic design.

2.2. Basics of FW thermal-hydraulics and consequences

In order to prepare the optimization approaches discussed in the following sections, some simplified relations about the thermal-hydraulics of the FW are introduced. We consider a patch of the FW of the length L_{pf} and width w_{pf}. From the plasma side, the patch receives a uniform heat flux density $\dot{q}_{pf}$, from the breeder zone side $\dot{q}_b$. This patch is cooled by a single channel run through by a mass flow rate $\dot{m}$ of a coolant (helium), with the inlet temperature T_1 and the exit temperature T_2, which are related by Eq. (1)

$$T_2 = T_1 + \frac{(\dot{q}_{pf} + \dot{q}_b) \cdot L_{pf} \cdot w_{pf}}{\dot{m} \cdot \overline{c_{p,fl}}} \tag{1}$$

where $\overline{c_{p,fl}}$ is the mean specific heat capacity of the fluid (5188 J/kg/K for helium). The channel is characterized by its hydraulic diameter d_h and flow cross section A_c. The coolant's thermophysical properties are density ρ_{fl}, heat conductivity λ_{fl} and dynamic viscosity μ_{fl}. In a very simplified approach, where the transversal heat conduction in the structures of the wall (i.e. toward neighboring counter-flow channels, which would additionally cool) is neglected, the maximum temperature occurring on the plasma facing wall (near the channel outlet) can be estimated by Eq. (2)

$$T_{w,p,max} = T_2 + \dot{q}_{pf} \cdot \left(\frac{1}{h_{wf}} + \frac{s_w}{\lambda_w} \right) \tag{2}$$

where $h_{wf} = Nu \cdot \lambda_{fl}/d_h$ is the heat transfer coefficient determined by the cooling channel shape and the coolant flow, and s_w the thickness of the wall between the cooled surface and the plasma facing side, made from a material with the heat conductivity λ_w. The temperature excess over T_2 is thus combined from a temperature difference in the coolant flow's thermal boundary layer ($\dot{q}_{pf}/h_{wf}$) and a temperature difference over the wall ($\dot{q}_{pf} \cdot s_w/\lambda_w$). For the case of the 15 mm × 15 mm smooth channel discussed in Section