Recent US activities on advanced He-cooled W-alloy divertor concepts for fusion power plants

M.S. Tillack a,1, A.R. Raffray a, X.R. Wang a, S. Malang b, S. Abdel-Khalik c, M. Yoda c, D. Youchison d

a University of California, San Diego, 9500 Gilman Drive, La Jolla, CA 92093-0417, USA
b Fusion Nuclear Technology Consulting, Flinderweg 3, D-76351 Linkenheim-Hochstetten, Germany
c Georgia Institute of Technology, George W. Woodruff School of Mechanical Engineering, Atlanta, GA 30332-0405, USA
d Sandia National Laboratories, P.O. Box 5800, MS 1129 Albuquerque, NM 87185-1129, USA

A R T I C L E   I N F O

Article history:
Received 11 May 2010
Received in revised form 9 August 2010
Accepted 9 August 2010

Keywords:
Divertor
High heat flux
Power plant
Helium
Tungsten

A B S T R A C T

Several advanced He-cooled W-alloy divertor concepts have been considered recently for power plant applications. They range in scale from a plate configuration with characteristic dimension of the order of 1 m, to the ARIES-CS T-tube configuration with characteristic dimension of the order of 10 cm, to the EU FZK finger concept with characteristic dimension of the order of 1.5 cm. The trend in moving to smaller-scale units is aimed at minimizing the thermal stress under a given heat load; however, this is done at the expense of increasing the number of units, with a corresponding impact on the reliability of the system. The possibility of optimizing the design by combining different configurations in an integrated design, based on the anticipated divertor heat flux profile, also has been proposed. Several heat transfer enhancement schemes have been considered in these designs, including slot jet, multi-hole jet, porous media and pin arrays. This paper summarizes recent US efforts in this area, including optimization and assessment of the different concepts under power plant conditions. Analytical and experimental studies of the concepts and cooling schemes are presented. Key issues are identified and discussed to help guide future R&D, including fabrication, joining, material behavior under the fusion environment and impact of design choice on reliability.

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1. Introduction

A number of different He-cooled W-alloy configurations have been proposed for the inner and outer divertor target plates for MFE power plant applications. They range in scale from a plate configuration with characteristic dimension of the order of 1 m [1,2], to the ARIES-CS T-tube configuration with characteristic dimension of the order of 10 cm [3,4], to the EU FZK finger concept with characteristic dimension of the order of 1.5 cm [5,6]. All these designs utilize tungsten or tungsten alloy as structural material, requiring material development to widen their operating temperature range from a low of ~600–700 °C (governed by W ductility considerations) to enable coupling to an ODS (oxide-dispersion-strengthened) FS (ferritic steel) manifold, to a high of ~1300 °C (governed by the W re-crystallization limit) to provide desirable high-temperature capability. The trend in moving to smaller-scale units is intended to minimize the thermal stress under a given heat load; however, this is done at the expense of increasing the number of units, typically from ~750 to ~110,000 to ~535,000 for the plate, T-tube and finger concepts, respectively, with a corresponding impact on the reliability of the system. The possibility of optimizing the design by combining different configurations in an integrated design based on the anticipated divertor heat flux profile also has been proposed.

Several heat transfer enhancement schemes have been considered that could be utilized in any of the above concepts. They include slot jet, multi-hole jet, porous media and pin arrays. These schemes provide a range of heat transfer capabilities and associated pressure drop penalties. A number of analytical and experimental studies of these different concepts and cooling flow schemes have been performed to help assess their relative performances. This paper considers the recent US effort in this area with the goal of providing a clear summary of the status of the studies, including optimization and assessment of the different concepts under power plant conditions. The different divertor concepts and flow schemes are first presented and analysis of their nominal operating conditions is summarized. Next, analytical and experimental studies of the different concepts and flow schemes are discussed. Key material and joining considerations are then summarized. An assessment of the different concepts under typical power plant con-
2. Divertor concepts

2.1. Configurations

The three main engineering configurations that we considered are presented in Table 1 including typical performance parameters. More detailed analysis is presented in Section 3.

2.1.1. Plate-type concepts

A helium-cooled plate-type divertor design concept has been proposed to provide a larger scale divertor concept that would minimize the number of units and associated joints containing high coolant pressure. The unit was designed to maintain as uniform a temperature distribution as possible based on surface heat flux and volumetric heat generation in order to minimize thermal stresses. An early concept is described in Ref. [1]. A more recent evolution of that concept is described in Ref. [2]; it makes use of a jet flow similar to the design of the finger [5,6] and T-tube [3,4] concepts to cool the heated surface. Its side and back walls are designed to minimize thermal stresses for the given flow conditions and neutron volumetric heating. The plate is made of W-alloy with a neutron volumetric heating. The plate is made of W-alloy with a volume of 150 m$^2$.

Table 1
Comparison of different divertor concepts for a tokamak with an assumed divertor area of 150 m$^2$.

<table>
<thead>
<tr>
<th>Divertor concept</th>
<th>Unit characteristic dimensions</th>
<th>Number of units for a typical tokamak</th>
<th>Allowable incident heat flux (MW/m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Finger</td>
<td>1.5 cm dia</td>
<td>~535,000</td>
<td>&gt;12</td>
</tr>
<tr>
<td>T-tube</td>
<td>10 cm × 1.5 cm</td>
<td>~110,000</td>
<td>~10–12</td>
</tr>
<tr>
<td>Plate</td>
<td>100 cm × 20 cm</td>
<td>~750</td>
<td>8–10</td>
</tr>
</tbody>
</table>

An early concept is described in Ref. [1]. A more recent evolution of that concept is described in Ref. [2]; it makes use of a jet flow similar to the design of the finger [5,6] and T-tube [3,4] concepts to cool the heated surface. Its side and back walls are designed to minimize thermal stresses for the given flow conditions and neutron volumetric heating. The plate is made of W-alloy with a volume of 150 m$^2$. A 5-mm castellated W armor region; it consists of a number of ~1-m long poloidal channels with a 2.2-cm toroidal pitch, as illustrated in Figs. 1 and 2 (a typical toroidal dimension would be ~20 cm). A thicker back region and a stagnant He insulating gap are used to provide the correct steady-state temperature equilibrium and to minimize thermal stresses.

Analysis of this concept is described in Section 3.1.1. Assumed design parameters and the results of thermo-mechanical analysis are summarized in Table 2. For a surface heat flux of 10 MW/m$^2$, a volumetric heat generation of 17.5 MW/m$^3$ (typical of a tokamak with an average wall load of ~3.2 MW/m$^2$), He inlet/outlet temperatures of 600/677 °C and He pressure of 10 MPa, all of the structures remain within their allowable temperature and stress limits while the He pumping power is kept below 10% of the thermal power. These results are encouraging as they indicate the possibility of accommodating up to 10 MW/m$^2$ with this concept. Modeling also has been performed to address concerns over the dynamic stress in case of heat flux transients or during startup and shutdown, because of the difference in thermal time scale between the thin (with end-of-life armor) region at the plasma side and the thicker back region (used to provide larger volumetric heating for better steady state thermal balance). These are further discussed in Section 3.

2.1.2. T-tube concept

The ARIES T-tube concept was developed to provide a mid-size unit capable of accommodating at least 10 MW/m$^2$, as described in detail in Refs. [3,4]. As illustrated in Fig. 3, the T-tube is ~15 mm in diameter and ~100 mm long and is made up of a W-alloy inner cartridge and outer tube on top of which a W castellated armor layer is attached. Both W alloy pieces are connected to a base ODS FS unit through a graded transition to minimize thermal stresses. The helium coolant is routed through the inner cartridge first and then pushed through thin slots (~0.4 mm) to cool the heat-loaded outer tube surface. The design provides some flexibility in accommodat-

![Fig. 1. Cross-section of unit cell and drawing of divertor plate module showing castellated W armor, inlet and outlet flow, jet cooling and stagnant He thermal insulation region. Arrows denote the flow of coolant [2].](image_url)
ing the divertor area since a variable number of such T-tubes can be connected to a common manifold to form the desired divertor target. The required He flow rate is $\sim 6 \text{ kg/s per m}^2$ of divertor plate surface with a corresponding pressure drop of $\sim 0.11 \text{ MPa}$ through the jet region. For He inlet/outlet temperatures of 600/677 $^\circ\text{C}$ and a surface heat flux of 10 MW/m$^2$, the maximum W alloy temperature is within the 1300 $^\circ\text{C}$ limit and the total stress intensity (primary and secondary stresses) is below 370 MPa for the entire geometry [3,4]. The performance of the T-tube concept can be increased by scaling; it was found that the stress and temperature limits can be satisfied by reducing the dimensions in the same proportion as the heat flux is increased. However, since the thicknesses are already small (e.g. the outer tube thickness is 1 mm), there is limited margin to increase the T-tube performance much beyond 10 MW/m$^2$. Analysis results are shown in Section 3.

2.1.3. Finger concept

The finger concept has been developed in part to minimize thermal stresses and allow for higher heat flux accommodation through the use of small modular units [5,6]. Different flow configurations have been investigated, including the HEMJ (He-cooled Modular divertor with Jet cooling) and HEMS (He-cooled Modular divertor with Slot array) designs, as illustrated in Fig. 4 [6]. They both include small hexagonal armor tiles of tungsten (5 mm thick) brazed to a W alloy thimble ($\Omega 15 \times 1 \text{ mm}$), forming a cooling finger. Each finger is cooled with He at 10 MPa and 600/700 $^\circ\text{C}$ inlet/outlet temperature supplied via a manifold system, with a pressure drop of 0.12 MPa through the jet region. Different solutions for fixing the cooling fingers to the manifold plate made of oxide dispersion-strengthened (ODS) steel have been evaluated, for example with a Cu casting transition piece to compensate for the large mismatch in the ther-

![Fig. 2. Longitudinal section through the target plate channel (length scaled by 1/4) showing the tapered flow channel.](image)

![Fig. 3. Schematic diagram and CAD drawing of the ARIES T-tube divertor concept [3,4].](image)
mal expansion coefficients of W and steel. The concept is designed to accommodate an incident heat flux of at least 10 MW/m² and perhaps higher.

In the EU finger divertor concept, the joining of the W alloy in the thimbles to the ODS steel of the manifold is very difficult because the thermal expansion coefficient of W is about 2.5 times larger than that of ferritic steel. New modifications and improvements to the modular finger design have been made by the authors to allow brazing the fingers made of W alloy directly into the W plate to avoid connections of dissimilar materials with different thermal expansion coefficient. This is illustrated in Fig. 5. The outer diameter of the thimble is enlarged from 18 to 20 mm while keeping the same size of the W armor and the cartridge. A 1-mm thick cylindrical ring made of W alloy is added inside of the W-alloy thimble covering the mantle of the thimble to provide an additional barrier against high-pressure helium leakage. The W armor, W-alloy thimble and W-alloy ring are brazed together into one armor unit, and then brazed into the W front plate as shown in Fig. 6. In this way, the modified finger design avoids the transition joints between W-alloy and steel at the modules, and can be considered as providing double containment of the high-pressure helium in the most critical regions. However, the slightly larger dimensions of the double-wall finger may impose temperature and stress penalties and affect performance; confirmation with thermo-fluid and thermo-mechanical analyses is needed.

2.1.4. Integrated divertor concepts

The heat flow to the divertor is governed by a number of parameters including the fusion power and assumed core and edge radiation fractions. The resulting heat flux footprint on the divertor usually follows a Gaussian-like distribution, whereby only a small fraction of the divertor would see very high heat fluxes [7]. This brings up the possibility of optimizing the heat flux accommodation and reliability measure (based on number of pressure joints or units) by utilizing the smaller-scale designs for the high heat flux region and the larger scale design (plate) for the lower heat flux region. This could be achieved by separately designing the high heat flux regions with the smaller units and the lower heat flux region with the plate designs and routing the coolant so as to cool these regions in parallel. A more interesting idea is to integrate the smaller scale unit within the plate design and routing the coolant through the integrated unit. This provides the flexibility of including the high heat flux region at any poloidal location along the design, and even of having the whole plate made up of an array of small-scale units for maximum performance.

An example of such a concept is based on the plate design but with an integrated number of finger units in the high heat flux zones, as illustrated in Fig. 7 [7]. A key feature of this concept is the
use of the plate design from the low heat flux zone (<6–7 MW/m²) as manifold for the finger-elements in the high heat flux zone, avoiding in this way the need for transition-pieces between W alloy to ODS steel with large differences in the coefficient of thermal expansion and without a common temperature window.

For an assumed high heat flux zone of 25 cm length, the number of finger units within the ~750 integrated plate components is ~87,820 (the thimble diameter here is 18 mm, slightly larger than the 15-mm diameter from the original design of Fig. 4); this compares with the 535,000 finger units required for full target plate coverage. The performance of the finger units in this concept is equivalent to the separate finger concept case for similar power plant divertor operating conditions.

This design concept can be extended to have the finger units covering the whole plate. This provides the possibility of designing for better He containment, as well as to minimize the transition between W alloy and the ODS ferritic steel tube and manifold. Such a concept is illustrated in Fig. 8.

Details of the performance analysis of this integrated concept are presented in Section 3.

2.2. Heat transfer enhancement techniques

For a long time, the use of helium cooling for divertor targets was believed to be limited to a maximum heat flux of about 5 MW/m². The main reason for this was the small temperature window for tungsten alloy structures between the re-crystallization temperature (~1300 °C) and the embrittlement temperature under neutron irradiation (~800 °C). The heat transfer coefficient achievable with “normal” channel flow is limited by the required pumping power of values below 20,000 W/m² K. For a heat flux of 5 MW/m² and a heat transfer coefficient of 20,000 W/m² K, the difference between the wall temperature and the bulk coolant temperature would amount to 250 K, which is about the maximum allowable value in order to remain within the temperature window described above. In order to achieve higher heat transfer coefficients, different ways to enhance heat transfer with helium cooling have been explored, including micro-channels, artificial surface roughening, cooling fins, an array of pins with cross-flow, impinging jets, and the use of porous media in the cooling channels.

2.2.1. Flow through micro-channels or a pin-fin array

Different concepts of helium-cooled divertor plates had been evaluated in a study by Hermsmeyer and Kleefeldt [8]. Based on that evaluation, an advanced concept of a plate divertor employing heat transfer enhancement by either flow in micro-channels or through a pin-array had been proposed by Hermsmeyer and Malang at the ISFNT-6 in San Diego [1]. In this concept, the divertor surface heat flux limit was raised to ~10 MW/m² by flowing the helium through an extremely small gap of 0.1 mm width over a length of ~10 mm. With a helium velocity of 200 m/s a heat transfer coefficient of 29,000 W/m² K had been calculated, requiring a pumping power <10% of the thermal power removed.

Since the feasibility of such a small gap was questionable (considering the potential for plugging by impurities), a second method for heat transfer improvement had been proposed. In that method an array of small pins is attached to the cooling surface and the helium flow is oriented perpendicular to these pins. The reference solution was a 60-degree arrangement of the pins with a diameter of 1 mm, a pitch to diameter ratio of 1.2, and a length of the pins of 2 mm. With a maximum velocity of 120 m/s between the pins a heat transfer coefficient relative to the original plate surface of 61,000 W/m² K has been calculated. The main reason for this high value is the increase of the cooling surface area by about a factor of 5 compared to the original plate surface. The calculated pressure drop is about 2.3 of that in a micro-channel with 0.1-mm width.

The original finger design concept also employed heat transfer enhancement using a pin array [9]. This concept, named HEMP (Helium cooled Modular divertor with Pin array), and a modification of it based on flow through parallel slots (HEMS) became the reference concept for the divertor development at the Forschungszentrum Karlsruhe (FZK) [10].

2.2.2. Jet flow through a slot

Impinging flow from a slot jet has been proposed for the plate-type and T-tube designs as well as the HEMS variant of the finger design. For example, the T-tube concept, proposed by Ilki, employed jet flow through a slot for the ARIES-CS divertor [3]. The tubes are cooled at the inner surface by a helium jet generated by a slot with 0.6 mm width. The average heat transfer coefficient and required pumping power are similar to the values calculated for the finger concept using jets through multiple holes.
2.2.3. Jet flow through multiple holes
Jet flow through holes is an alternative to jet flow through a slot, in which the impinging jet geometry is more flexible and the dimension of the orifice is easier to maintain. In the HEMJ finger concept [5] a number of cylindrical He jets hit the cooling surface, providing in this way at least the same heat transfer enhancement as the previous concepts but with lower pressure drop and simpler fabrication.

Examination of the different jet cooling techniques (jet through slot, jet through multiple holes) in the areas of thermal-hydraulics, thermal stresses, and fabrication indicates that a third version could be interesting where the slot is replaced by a linear row of circular holes in order to avoid the problems of machining and maintaining dimensional control with a slot <0.5 mm in width. The analyses of plate concepts using this technique showed that surface heat fluxes up to 10 MW/m² are allowable without exceeding temperature and stress limits, but the thermal stresses are somewhat higher than with the finger- or the T-tube concepts.

2.2.4. Use of porous media
Two design concepts using porous media have enjoyed significant scrutiny in the US fusion energy sciences program: the porous foam-in-tube (PFIT) design and the short flow path foam-in-tube (SOFIT) design. The PFIT concept employs axial flow in a tungsten tube (see Fig. 9). Thermal conduction into the foam ligaments allows the foam to function as a fin as well as a turbulence promoter by disrupting the thermal boundary layer along the tube wall. In addition, flow bypass along the walls is prevented. Fabrication and experimental results are presented in Section 4.2.2.

The SOFIT concept directs axial flow from an inlet tube toward the high heat flux surface through a slot. The flow path is similar to that employed in the plate concept (Section 2.2.1), but the microchannels or the pin-arrays in that concept are replaced by porous media. Fig. 10 shows the configuration of the modules wherein the helium flows circumferentially through the foam through a 2-mm-wide slot separating the inlet and outlet plenums. The flow path through the porous medium in this case can be short, so the pressure drop is inherently low.

3. Performance assessment

3.1. Typical operating parameters

Different divertor concepts and techniques for heat transfer enhancement were presented in Section 2. The original analyses of these concepts were based on somewhat different sets of parameters, such as the cooling parameters, volumetric heat generation rates, material limits, and armor thicknesses. In order to provide a
3.2. Thermo-fluid and thermo-mechanical analysis

Detailed 3-D CFD analyses were performed using CFX [11] in order to optimize geometries and thermal performance of divertor concepts including the plate-type, T-tube, finger, and combination finger and plate designs. The heat transfer associated with jet impingement cooling was predicted based on a standard turbulent flow model, $k-e$ with wall enhancement. Results obtained by the CFD code show good consistency with experimental validations performed at Georgia Institute of Technology [12]. Extensive 3-D finite element (FE) thermo-mechanical simulations were performed using ANSYS Workbench [13] to further optimize the divertor configuration with respect to the material temperature, stresses, deformation and pumping power limits.

3.2.1. Plate design analysis

As illustrated in Figs. 1 and 2, the inlet and outlet manifolds are inserted into the slot channel (20 mm wide) extending all the way along the longitudinal (poloidal) direction, and they are tapered in order to balance the velocity ($\sim$ a few cm/s) in the longitudinal direction. The helium coolant is routed through the inlet manifold with low velocity and then flows through thin slot-jets ($\sim$0.5 mm) with high jet velocity to cool the front plate, which is subjected to a heat flux of 10 MW/m². The flow of helium in both inlet and outlet manifolds is uniform along the 1-m poloidal cooling channel, resulting in nearly uniform and constant jet velocity. Therefore, only a half cooling channel and 2-cm poloidal channel length were utilized in the CFD simulation model. The results of the thermal-fluid and thermo-mechanical analyses are based on a surface heat flux of 10 MW/m², a volumetric heat generation of 17.5 MW/m³, helium inlet/outlet temperature of 600/700°C, and helium pressure of 10 MPa. Fig. 11 shows the velocity distributions for the slot-jet cooling scheme. The maximum jet velocity is about 258 m/s, and the corresponding maximum heat transfer coefficient at the internal surface is $4.19 \times 10^4$ W/m²K while the pumping power is less than 10% of the thermal power. Other flow configurations [2], such as micro-channel and multiple-jet cooling were also investigated; the velocity and heat transfer coefficient for the micro-channel and multiple-jet cooling schemes are higher than those of the slot-jet cooling scheme, but with a correspondingly higher pumping power.

The 3-D FE model of the plate for calculating stresses and deformations included a full length of the channel (20 mm wide and ~1 m long in the poloidal direction) with a W armor layer (~5 mm thick) on the front plate. Castellation of the armor was assumed (5 mm toroidal x 5 mm poloidal) to minimize forces transferring from the tiles to the channel structure and to reduce thermal stresses. The ODS insert manifolds were excluded from the thermo-mechanical model because of their lower temperature and stress. The heat transfer coefficient and the temperature at the interface of the He and W structure were obtained from CFX thermo-fluid analyses and used as thermal boundary conditions for computing the temperature field in the FE model. A coupled thermo-mechanical analysis was then performed. The following mechanical boundary conditions were applied: (1) no channel bending and in-plane back side, and (2) symmetry condition in the middle of the channel. This results in conservative results since the entire surface at the back plate was restrained from free bending.

The results of thermal analysis show that the maximum temperature of the W armor is 1853°C and the maximum temperature at the interface between the W armor and the front plate {underneath the W armor} is 1295°C, which is within the 1300°C re-crystallization limit assumed for the W alloy. The maximum combined stress (primary + thermal) is 359 MPa for the 5 mm x 5 mm tile castellation, which is within the assumed $S_m$ limit of 465 MPa at the corresponding temperature. The maximum pressure stress is 110 MPa (within the assumed $S_m$ limit of 155 MPa), maximum thermal stress is 310 MPa, and the maximum total deformation is about 1.3 mm.

The results obtained from numerical analysis are encouraging as they indicate the possibility of accommodating up to 10 MW/m² with the plate-type divertor concept while taking advantage of larger divertor units and fewer joints. If the cooling conditions were changed to be the same as those listed in Table 3, such as helium inlet/outlet temperature of 600/700°C, the accommodating capability of the plate-type divertor concept might be lowered down to a range of 7–9 MW/m².

Concerns exist as to the dynamic stress in case of heat flux transients or during power plant startup and shutdown because of the difference in thermal scale between the thin (with end-of-life armor) region at the plasma side and the thicker back region (used to provide larger volumetric heating for better steady state thermal

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum allowable W structure temperature</td>
<td>1300°C</td>
</tr>
<tr>
<td>Minimum allowable W structure temperature</td>
<td>800°C</td>
</tr>
<tr>
<td>Inlet He temperature</td>
<td>600°C</td>
</tr>
<tr>
<td>Outlet He temperature</td>
<td>700°C</td>
</tr>
<tr>
<td>He pressure</td>
<td>10 MPa</td>
</tr>
<tr>
<td>Maximum surface heat flux</td>
<td>10 MW/m²</td>
</tr>
<tr>
<td>Neutron power generation</td>
<td>17.5 MW/m³</td>
</tr>
<tr>
<td>Allowable pumping power/thermal power</td>
<td>&lt;10%</td>
</tr>
<tr>
<td>Sacrificial W armor thickness</td>
<td>5 mm</td>
</tr>
</tbody>
</table>
To address these concerns, transient thermal and stress response of the divertor during the power plant startup and shutdown operations was investigated based on the ARIES-I[14] startup and operating scenario. Steady-state conditions were assumed to be attained at ∼2500 s. The neutron wall loading and surface heat flux were ramped up at ∼2100 s from a very low value (∼0) to full power values (with the heat flux \(q_s = 10\) MW/m² and the neutron volumetric heating \(q_v = 17.5\) MW/m³) over the 400-s fusion power ramp-up phase.

Detailed 3-D transient CFD (using CFX) and thermo-mechanical analyses (using ANSYS Workbench) were performed to explore the allowable heat flux at the divertor plate. The detailed results [15] indicate that the divertor plate can accommodate such gradual ramp-up and ramp-down scenarios within the given stress and temperature constraints. However, faster transients such as in the case of a sudden shift in the heat flux footprint (over 1–10 s) are of concern and should be addressed to better understand the limits of such a concept.

### 3.2.2. T-tube analysis

The helium-cooled T-tube divertor concept originally proposed for the ARIES-CS compact stellerator power plant [3,4] was extended to a general tokamak power plant by increasing the thickness of W armor from 0.3 mm to 5 mm, because higher erosion for tokamak power plant would be expected. To verify the application of the T-tube divertor concept in a tokamak power plant, thermal-fluid and thermo-mechanical analyses were performed. The helium inlet/outlet temperature was assumed to be 600/677 °C, the surface heat flux on the target surface was 10 MW/m², and the neutron volumetric heat generation was 17.5 MW/m³. Considering symmetry, only one quarter of the tube was analyzed in the CFX and FE model.

The standard \(k-\varepsilon\) model with wall enhancement for turbulent flow was assumed.

Results for the helium velocity and helium temperature distributions in the T-tube based on 3-D CFX analysis are shown in Figs. 12 and 13. The maximum helium velocity though the jet is ∼230 m/s and the maximum local coolant temperature is about 1152 °C. The corresponding pressure drop, also obtained from CFX is ∼0.12 MPa and resulting pumping power is ∼5.7% of the thermal power (smaller than design limit of 10%).

ANSYS Workbench was utilized to calculate the stresses and deformations within the T-tube configuration for an internal helium pressure of 10 MPa and a heat load of 10 MW/m². The interface temperature between the wall and coolant is directly coupled to the ANSYS solid model for stress evaluation. The armor layer was assumed to be bonded to the W alloy tube, and castellated into 5 mm × 5 mm in order to minimize the stress transferring from the W armor to the tube structure. As illustrated in Figs. 14 and 15, the maximum armor temperature is ∼1782 °C and the interface...
temperature between the W armor and W alloy tube structure is \( \sim 1240 \, ^\circ C \), which is within the 1300 \( \, ^\circ C \) limit (re-crystallization limit). The total stress intensity (primary and secondary stresses) is \( \sim 342 \) MPa for the entire geometry, which is less than the \( 3S_m \) limit of an anticipated W alloy at the corresponding temperature. However, if the He outlet temperature increased from 677 \( \, ^\circ C \) to 700 \( \, ^\circ C \) while keeping all other parameters the same, the T-tube still can accommodate a surface heat flux of 10 MW/\( \text{m}^2 \) and maintain the material temperature and stresses with design limits. The T-tube concept can be further optimized to improve its thermal performance, for example by tapering the cartridge while keeping the same size of the outer tube in order to obtain uniform jet velocity, heat transfer coefficient and temperature. In this way, the thermal stresses can be reduced.

3.2.3. Finger design analysis

As illustrated in Figs. 5 and 6, some new modifications and improvements of the finger concept have been made by the authors, based on the EU finger modular divertor concept [16]. The purpose of these modifications is to effectively provide a double containment of the high-pressure helium at the most critical location, and to avoid the large number of joints between materials with largely different thermal expansion coefficients. However, enlarging the outer diameter of the W thimble from 18 mm to 20 mm and adding a 1-mm thick cylindrical ring covering the mantel of the thimble would possibly impose both temperature and stress penalties and affect divertor performance. Extensive thermo-fluid and thermo-mechanical analyses based on the new modifications have been performed to verify its performance. The parameters used in the simulations are those listed in Table 3.

The maximum helium jet velocity obtained from CFX modeling is \( \sim 250 \, \text{m/s} \), and the resulting maximum heat transfer coefficient is \( \sim 5.84 \times 10^4 \, \text{W/m}^2 \, \text{K} \). The distribution of the heat transfer coefficient over the whole surface area is shown in Fig. 16. The pumping power is \( \sim 7.5\% \) of the thermal power (less than 10\% design limit). The corresponding fluid and solid material temperature are shown in Figs. 17 and 18. The maximum temperature of the W armor is 1848 \( \, ^\circ C \) and the maximum temperature of the W thimble (the structural component under the W armor) is 1222 \( \, ^\circ C \), which is within the 1300 \( \, ^\circ C \) re-crystallization limit assumed for W alloy.

The combined stress distribution is shown in Fig. 19. To minimize the forces transferring from the W armor to the thimble, the W armor is assumed to be castellated with small triangles 4 mm deep having 0.25 mm gaps. The maximum combined stress (primary + secondary stress) of the W armor is 429 MPa, exceeding the \( 3S_m \) limit at the corresponding temperature. However, the peak stress only occurs at the local straight angles of the W armor and it can be avoided by making smooth transition surfaces. The maximum combined stress of the thimble is 360 MPa, within the assumed \( 3S_m \) value of 530 MPa at the corresponding temperature. The maximum pressure stress (not shown) is 90 MPa, also within the assumed \( S_m \) limit. The results indicate that the double wall provides an effective double containment between the plasma chamber and the high-pressure helium (10 MPa) without impacting the temperature and stresses significantly, because it does not change the ratio of heated and cooled surface areas.

Fig. 15. Combined stress distribution (primary + secondary) of the 1/4 T-tube.

Fig. 16. Distribution of the heat transfer coefficient.

Fig. 17. Temperature distribution of the helium coolant.
3.2.4. Integrated concept analysis

To assess and confirm the performance of an integrated unit, thermo-fluid and thermo-mechanical analyses concentrated on the back cooling channel (manifolds), which is underneath the hot components [7]. The hot components, i.e., the modified fingers that face the high heat flux, have been investigated individually and the results are described in Section 3.2.3. Examples of the thermo-fluid and thermo-mechanical analysis for the back cooling channel were based on the combination of the finger and plate, as illustrated in Fig. 7. In the CFD model, only one finger unit was considered because each finger is identical in geometry and thermo-fluid behavior. Fig. 5 shows the single finger unit and its coolant manifolds, and Fig. 20 shows the CAD model utilized in the CFD modeling.

Results from CFX calculations show the velocity in the manifolds is \(\sim 47\) m/s and the corresponding pressure drop is \(\sim 0.01\) MPa which is very small compared to that of the finger (\(\sim 0.13\) MPa). The temperature at the interface of the fluid and solid wall surface are almost uniformly distributed in the poloidal direction and the temperature difference between the front and back plate is \(\sim 40^\circ\) C; therefore, the full length of the manifolds (1-m long) was utilized in the structural analysis. The maximum temperature of the cooling channel is 742 \(^\circ\) C. The maximum combined primary and secondary stress is 470 MPa, which is localized at the ends of the manifold and is well below the design limit (35m = 754 MPa for W alloy at a temperature of 740 \(^\circ\) C). The maximum primary and thermal stresses are 195 MPa and 285 MPa individually, and are also within the allowable stress limits.

4. Studies of different engineering units and cooling configurations

4.1. Thermal-hydraulic studies

Detailed experimental studies, supported by numerical simulations, have been performed at the Georgia Institute of Technology as part of the ARIES studies to evaluate the thermal performance of leading helium-cooled divertor designs, including:
All of these concepts use impinging jets of high-pressure helium (He) to cool the back of the plasma-facing surface (the divertor pressure boundary), thereby transferring the heat incident on the divertor to the power cycle. Either a single planar (2D) jet or an array of round jets is used. Numerical simulations using commercial computational fluid dynamics (CFD) codes have shown these three designs to be capable of maintaining adequate thermal control for surface heat fluxes up to 10 MW/m². Such heat fluxes can be accommodated because of the extremely high heat transfer coefficients (HTC) provided by the impinging jets; HTC values in excess of 40 kW/(m² K) were predicted by the CFD codes near the jets’ stagnation points. Such values were judged to be “outside the experience base” for gas-cooled engineering systems. Hence, this research effort has been aimed at experimental validation of the numerical models.

The general approach used in this research consists of the following basic steps:

(a) A test section nearly duplicating the geometry, dimensions, and thermal conductivity of the actual divertor module is designed, constructed, and instrumented;
(b) The test section is placed in an air test loop, where heated experiments are performed. The test conditions (air inlet conditions and flow rate) are varied over a wide range so that the governing non-dimensional parameter (jet Reynolds number) would span its expected nominal operating value in the actual helium-cooled divertor design;
(c) The experimentally measured temperature distributions and pressure drop are used to validate the numerical CFD model predictions.

The experimental data obtained in this investigation, together with the validated numerical models represent the main contribution of this research, inasmuch as the validated numerical models can be used with confidence to predict the thermal performance of the same divertor designs with helium cooling over a wide range of operating conditions, including the effect of non-uniform incident heat flux. The validated models can also be used to optimize the geometry and/or coolant operating conditions of the actual divertor by quantifying the effect of such modifications on the thermal performance. The following sections provide details of the work performed on each of the three leading divertor design concepts, namely, the HEMJ, the T-tube, and the plate designs.

4.1.1. The helium-cooled multi-jet (HEMJ) divertor
Consistent with the three basic steps outlined above, studies have been performed to examine the thermal performance of the helium-cooled multi-jet (HEMJ) divertor design developed by FZK (Fig. 4 [left]) [5,22].

The plasma-facing surface is a tungsten (W) armor plate attached to a W alloy (WL10) cap. A cylindrical steel cartridge with 24 (0.6 mm diameter) holes surrounding a single 1.0 mm diameter hole in the center is secured below the cap. Helium at 600 °C and 10 MPa enters the cartridge and is accelerated through the 25 holes to create jets that impinge on the capped inner surface of the W alloy. The He flows through a 0.9-mm annular gap between the cartridge and the cap and exits the divertor at nearly 700 °C.

The test module used to simulate the HEMJ divertor module consists of three main parts (Fig. 21): (1) the test element, which includes a jet cartridge and a concentric thimble, duplicating the HEMJ design geometry and dimensions; (2) the tee, which provides the fittings for the coolant (air) and the instrumentation to monitor the flow; and (3) the heater, which consists of a “bottle-shaped” copper (Cu) block (concentrator) housing one electric heater. The concentrator provides a uniform incident heat flux onto the outer surface of the thimble, thereby simulating the incident heat flux on the divertor.

The thimble was manufactured from C36000 free-machining brass, which has nearly the same thermal conductivity as the W alloy cap and W armor tile of the HEMJ design. The Cu concentrator is designed to produce a uniform axial heat flux at the thimble surface. The block, manufactured from free-machining C14500 Cu, consists of a 50 mm diameter section gradually contracting to a 17 mm diameter “neck” region, which is brazed to the top of the thimble. A FAST-HEAT® cartridge heater with a maximum output of 750 W is placed in the center of the block. Power to the heater can be controlled to produce incident surface heat fluxes up to 2.0 MW/m².

The entire test section is embedded in a 12.5 cm diameter cylinder of rock wool insulation that extends 5 cm beyond the top of the heater block.

Four E-type thermocouple (TC) probes are inserted into the brass thimble at varying depths spaced by 90° to measure the temperature distribution over the cooled surface (Fig. 22). In all cases, the bead center is located 0.5 mm from the cooled surface. The jet cartridge can be rotated relative to the thimble. The angular position of the cartridge, θ, is measured from the location of TC #1 (Fig. 23). The angular distribution of temperature at the four instrumented radial positions is therefore obtained by repeating experiments with different angular, or θ, positions.

The air flow loop used to conduct the experiments is shown in Fig. 24. Air from a compressed-air line enters the loop at a pressure of ~100 psig (724 kPa). The inlet pressure is measured with an analog test gauge with accuracy of ±0.5 psi (3.4 kPa). The inlet temperature is measured using an E-type thermocouple. The exit
Fig. 22. Sketch of the TC locations within the thimble [17,18].

Fig. 23. Photo of the angular indexer (left) and angular position $\theta$ (right) [17,18].

Fig. 24. Schematic of the air test loop [17,18].
pressure and temperature are measured with a pressure transducer and E-type TC, respectively. A needle valve downstream of the cross-fitting is used to control the mass flow rate through the test section. The mass flow rate through the test section is measured by a positive displacement gas flow meter, from which the air is vented directly to the atmosphere.

Experiments were conducted at various heat flux values with a wide range of flow rates and air inlet conditions. The Reynolds number $Re$ based on the central jet diameter within the cartridge varied between $1.4 \times 10^4$ and $5.9 \times 10^4$; these values span the expected value for the HEMJ divertor at the nominal operating conditions ($2.1 \times 10^4$).

The numerical model used to simulate the test section included the electric heater, Cu concentrator, brass thimble, jet cartridge, tee, insulation, inlet tube, and outlet connector (Fig. 25). Only half of the test module was modeled because of its two-fold symmetry. The finite element model included nearly $1.5 \times 10^6$ grid cells with nearly $7 \times 10^5$ nodes. The grid spatial resolution was adjusted depending on the expected parameter gradients within the various regions; a much finer grid was used in the jet impingement region (Fig. 26).

Typical results showing the experimental and numerical azimuthal temperature distributions for TC #1 are shown in Fig. 27. Only the temperature distribution for $\theta = 0$–$60^\circ$ is shown because of the six-fold symmetry of the hole pattern. The experimental and predicted surface temperature distributions are qualitatively similar with only small differences in the absolute temperature values (note the highly expanded vertical scale). Overall, TC #1 has the lowest temperature values of the four TCs. At $\theta = 0^\circ$ and $60^\circ$, TC #1 is in close proximity to the stagnation point of one of the jets; hence, the low temperature readings. At $\theta = 15^\circ$ and $45^\circ$, TC #1 is nearly halfway between two adjacent jet stagnation points, and so the temperatures are elevated at these angular positions.

While the azimuthal temperature distribution at a given radial position appears to be relatively uniform, the jet impact pattern produces large gradients in the local heat transfer coefficients, which, in turn, impacts the local heat flux. Fig. 28 shows azimuthal variations of the experimental and numerical HTC values for the case shown in Fig. 27. Clearly, given the size of the test module, it is impossible to experimentally measure the local surface heat flux variations on the cooled surface; hence both sets of HTC values shown in Fig. 28 use the numerically calculated heat flux values.
Fig. 29. Variations of the experimental (symbols) and numerical (lines) Nusselt number values at the four instrumented radial locations for $\theta = 0^\circ$ [17,18]. The color (grayscale value) of the lines and symbols given in the legend denote the incident heat flux.

to determine the corresponding HTC. Nevertheless, the fact that the experimental and numerical temperature values are closely matched suggests that the local HTC values must also be matched. It should be noted that for dynamically similar conditions with identical geometries (i.e., the same Nusselt number and characteristic dimensions), the HTC values for He are nearly a factor of 12 greater than those for air, because of its higher thermal conductivity.

Fig. 29 shows variations of the Nusselt number $\nu$ at $\theta = 0^\circ$ for the four instrumented radial positions over the full range of Reynolds numbers examined in this investigation. The experimental data and numerical predictions are in good agreement. High values of $\nu$ are obtained at TC locations #1 and #4 due to direct jet impingement at those locations. The $\nu$ values at the centerline (TC #4) are the highest because the center jet diameter (1.0 mm) is larger than that of the other jets (0.6 mm), and hence has a higher $Re$. The $\nu$ values depend slightly on the incident heat flux because of changes in the physical properties. The difference between the Prandtl number $Pr$ values of air and He (0.71 vs. 0.66, respectively) should have little effect on $\nu$ since $\nu \propto Pr^{n}$ where $n < 1$ for these turbulent flows.

The experimental and numerical pressure drop values are compared in Fig. 30. The differences are generally less than $-10\%$, indicating good agreement over the entire range of $Re$ covered in this investigation.

4.1.2. T-tube divertor studies

Experimental and numerical studies have been performed to examine the thermal performance of the T-tube divertor [18,19]. A schematic diagram of the T-tube divertor module, which was selected by the ARIES-CS team [3], is shown in Fig. 3. It consists of two concentric tubes with a 1.25 mm annular gap; a flat W armor layer is placed atop the outer tube. Both ends of the tubes are capped. The inner tube (cartridge) has a narrow (0.5 mm wide) slot along its entire length. The coolant (He at 10 MPa and 600 $^\circ$C) enters the inner tube through an inlet port located midway along its length and is accelerated through the slot towards the inner surface of the outer tube. The stagnation point flow generated by the impingement of the quasi-2D rectangular jet on this heated surface cools the divertor with a moderate pressure drop. Downstream of the stagnation location, the He forms a turbulent wall jet along the inside surface of the outer tube, and is then removed through the two exit ports near the center of the module. The coolant exits the module at $\sim$9.9 MPa and 680 $^\circ$C.

Consistent with the three basic steps outlined in Section 4.1 above, a test module duplicating the geometry and dimensions of

Fig. 30. Comparison between experimental values (symbols) and numerical predictions (line) of the pressure drop across the HEMJ test module over a range of Reynolds numbers [17,18].
the T-tube divertor was designed, constructed, and instrumented (Fig. 31). The test section consists of two main parts:

1. A concentric cartridge, which includes the T-block, inner slotted tube, and two end caps. The brass T-block assembly duplicates the entire T-tube divertor module including the tungsten armor and the pressure boundary. Brass was selected since its thermal conductivity nearly matches that of tungsten.
2. A C14500 Cu-alloy block (concentrator), housing three 500 W cartridge heaters. The concentrator is tapered to focus the heat load onto the much smaller area of the T-block, thereby increasing the incident heat flux on the T-block. Only the top surface of the brass cartridge is heated, duplicating the one-sided heating configuration of the actual T-tube divertor.

Fifteen E-type TC probes are embedded in the T-block to measure axial and azimuthal wall temperature profiles. These probes are placed on planes at three different axial locations, corresponding to 1/4, 1/2 and 3/4 of the axial extent of the T-block. At each plane, five probes are distributed around the periphery of the pressure boundary equally spaced by 30° (Fig. 32). The TC beads are placed at \( \theta = 0° \) (the impinging jet stagnation point) and \( \theta = 30° \), 60°, 90° and 120° away from the stagnation point with a 0.5 mm radial offset (edge-to-edge) from the pressure boundary surface. Six E-type TC probes are embedded in the “neck” of the concentrator to measure the incident heat flux. The TCs are located on two planes (“A” and “B”) at 1/3 and 2/3 of the Cu block length; the TC beads are 3.0, 7.0 and 12.0 mm from the surface contacting the T-block. The probes are embedded halfway in the material. The entire test module is then insulated with 5 cm thick panels of mineral wool.

The T-tube test module was placed in the same air test loop of the HEMJ investigation. The probes are embedded halfway in the material. The entire test module is then insulated with 5 cm thick panels of mineral wool. The probes are placed on planes at three different axial locations, corresponding to 1/4, 1/2 and 3/4 of the axial extent of the T-block. At each plane, five probes are distributed around the periphery of the pressure boundary equally spaced by 30° (Fig. 32). The TC beads are placed at \( \theta = 0° \) (the impinging jet stagnation point) and \( \theta = 30° \), 60°, 90° and 120° away from the stagnation point with a 0.5 mm radial offset (edge-to-edge) from the pressure boundary surface. Six E-type TC probes are embedded in the “neck” of the concentrator to measure the incident heat flux. The TCs are located on two planes (“A” and “B”) at 1/3 and 2/3 of the Cu block length; the TC beads are 3.0, 7.0 and 12.0 mm from the surface contacting the T-block. The probes are embedded halfway in the material. The entire test module is then insulated with 5 cm thick panels of mineral wool.

Experiments were conducted at various heat flux values with a wide range of flow rates and air inlet conditions. The Reynolds number case \((Re=1.9 \times 10^4)\) had the heat transfer coefficient value of the He-cooled T-tube divertor at the nominal operating conditions \((1.9 \times 10^4).\) The standard k-ε turbulence model was used in all calculations. These results pertain to all the three axially instrumented locations, where planes #1 and #3 are symmetrically located along the T-block and plane #2 is in the center (Fig. 32). As expected, the experimental data suggest that the flow is symmetric with respect to the axial centerline (compare results for the instrumented planes #1 and #3). Furthermore, azimuthal symmetry is observed since the same azimuthal TC locations are mirrored at axially symmetric instrumented planes #1 and #3 (Fig. 32). The highest temperatures occur at \( \theta = 0° \) and 30°, the two locations that are directly subject to the incident heat flux.

While the incident heat flux produced by the Cu concentrator is nearly one-dimensional, significant azimuthal temperature gradients are observed due to conduction in the brass T-block; similar gradients are to be expected in the actual W alloy divertor module. The numerical predictions of the temperature profiles show reasonable agreement with the measured values (note the expanded scale). The code predictions are slightly higher than the measured values. The differences can be attributed to the contact resistance at the TC tips.

The temperature distributions shown in Fig. 33 indicate that the center axial location of the module (plane #2) is slightly cooler than the edge planes (planes #1 and #3). This is primarily due to the coolant flow distribution along the slot, where the coolant velocity at the midpoint is expected to be higher because of the location of the inlet port (Fig. 3).

The surface temperature profiles, together with the numerically predicted local heat flux values, were used to determine the local surface heat transfer coefficient profiles. The experimental (open symbols) and numerical (filled symbols) values of the azimuthal HTC are shown in Fig. 34. As expected, the highest local heat transfer coefficient is achieved at the impinging jet stagnation point \((\theta = 0°)\); the HTC values then rapidly decrease away from the stagnation point.
point. The azimuthal gradients in surface heat transfer coefficient profiles shown in Fig. 34 are due to the impinging jet cooling effect which is considerably more significant than the surface temperature gradients due to conduction noted in Fig. 33. Again, it should be noted that for dynamically similar conditions with identical geometries (i.e., the same \( \text{Nu} \) and characteristic dimensions), the heat transfer coefficients for He are nearly a factor of twelve higher than those for air because of its higher thermal conductivity.

The local HTC values were used to calculate the corresponding Nusselt numbers. The characteristic dimension is the hydraulic diameter of the slot (twice the slot width). Fig. 35 shows \( \text{Nu} \) as a function of Reynolds number at the impinging jet stagnation point for the three instrumented axial planes. All the experimental cases (symbols) are included. For \( \text{Re} = 1.9 \times 10^4 \) and \( 3.04 \times 10^4 \), two different values of the incident heat flux were tested—(0.55, 0.71) \( \text{MW/m}^2 \) and (0.75, 0.81) \( \text{MW/m}^2 \), respectively. As expected, the heat flux used in the experiment has a negligible impact on \( \text{Nu} \); the change, if any, is attributed only to changes in the physical properties. Again, the difference between the Prandtl number values of air and He (0.71 vs. 0.66, respectively) should have little effect on \( \text{Nu} \), as discussed previously for the HEM studies.

Table 4 compares values of experimentally measured and numerically predicted pressure drop in the test module for all the test cases. The pressure drop estimates are in good agreement with the experimental data, with the exception of experiment #1, where the pressure drop is very small because of the low \( \text{Re} \) and therefore subject to the largest experimental errors. Nevertheless, the numerical predictions are, in all cases, slightly lower than the measured pressure drop values for all the \( \text{Re} \) tested. This can be attributed to “minor losses” associated with changes in the cross-sectional areas of the fittings, which were not included in the numerical model.

4.1.3. Plate-type divertors

Recently, the Georgia Tech group has focused on variants of the He-cooled flat-plate divertor, a design originally developed at the Karlsruhe Research Center (FZK) [1]. The major advantage of this particular design is the large surface area of each module (2000 cm\(^2\)), which reduces the number of modules needed to cool the \( 100 \text{ m}^2 \) area of the divertor by more than two orders of magnitude compared with either the T-tube or HEM designs, with module surface areas of about 13 cm\(^2\) and 2.5 cm\(^2\), respectively. The resultant reduction in the complexity of the manifold systems required to supply and discharge the coolant is a major advantage for a power plant. Numerical simulations have shown that this design can, with appropriate modification, withstand incident heat fluxes up to 10 MW/m\(^2\) without exceeding either the maximum temperature limit of \( \sim 1300 \) °C or the maximum stress limit of \( \sim 400 \) MPa for the W alloy [2].

As shown in Fig. 3, the most recent plate-type divertor module consists of 9 inlet/outlet manifold units separated by two W-alloy side plates up to 3 mm in thickness brazed together to cool an area 200 mm \( \times \) 1000 mm, or 2000 cm\(^2\) [left]. Helium enters the module as indicated by the blue arrow [left], flows through the inlet manifold [right, In], and is accelerated as a quasi-2D rectangular jet through a 0.5 mm wide slot [right, blue arrow], impinges (and stagnates) upon the underside of the surface attached to the W armor, then flows down along the outer surface of the inlet manifold [right, red arrows] into the outlet manifold [right, Out], exiting the module as denoted by the red arrow [left].

Given the difficulties in manufacturing this modified geometry, the experimental studies closely reproduce instead the original geometry of a single inlet/outlet manifold unit as proposed by Malang (Fig. 36 [left]) [1]. A single unit was studied at prototypical dimensions because the results are sufficient, with appropriate scaling, to determine the thermal performance of the entire plate, which is determined by the characteristics of the jet impingement and the pressure drop across the unit.

The experimental module (Fig. 36 [right]) consists of an aluminum (Al) cartridge (gray) inside a brass shell (orange) with a thermal conductivity that closely matches that of the tungsten armor and the W-alloy side plates; the brass shell simulates both the W armor and the pressure boundary of the manifold unit. The inlet and outlet manifolds are modeled as 19 mm \( \times \) 15 mm \( \times \) 76.2 mm rectangular chambers inside the Al cartridge. The test section is attached to an air flow loop (see Fig. 24) where air at a known inlet pressure and temperature enters the inlet manifold, then is driven through either a slot with a width \( H \) or an array of circular holes of diameter \( d \) to produce a planar jet or a series of round jets, respectively. The jet(s) then impinge upon and cool the inner surface of the brass shell, which is heated by three 750 W cartridge heaters embedded in a copper-alloy heat concentrator. Temperature distributions near the cooled inner surface of the brass shell are measured by five E-type TCs embedded 1 mm away from the cooled surface (Fig. 37).
Experiments were performed at incident heat fluxes \( q'' = 0.21–0.85 \text{ MW/m}^2 \) and \( Re = 1.1 \times 10^4–6.8 \times 10^4 \). The range of \( Re \) studied here spans the actual Reynolds number of 3.3 \( \times 10^5 \) for the plate-type divertor (for \( H = 0.5 \text{ mm} \)) at its proposed nominal operating conditions. Temperature readings from six E-type TCs in the “neck” of the concentrator were used to determine \( q'' \). The experimental heat fluxes are significantly lower than the actual heat flux values (up to 10 MW/m²) because of the need to limit the maximum temperature of the heat concentrator to less than 500 °C (roughly half the melting temperature of the Cu alloy). Nevertheless, for dynamically similar systems, the heat flux should have a relatively small effect on the Nusselt number, primarily due to the change in the coolant physical properties from their values at the mean bulk temperature.

As detailed in Gayton [23], these studies concluded that:

- The array of holes had a higher pressure drop (rescaled to a common test section pressure of 414 kPa) than the 2 mm wide slot at a given mass flow rate in and out; and
- Although the presence of the foam significantly increased the effective heat transfer coefficient, by about 40% at \( Re = 3.6 \times 10^4 \), it also increased the pressure drop across the manifold unit by almost 50%.

Based on these results, the most recent Georgia Tech experimental studies have considered a plate-type divertor with a pin-fin array integral to the cooled surface and extending across the entire gap to the outer surface of the inner cartridge [21]. Like the metallic foam, the array of fins increases the area of the cooled surface, but with relatively minor increase in the pressure drop across the manifold unit. These studies considered four different configurations with two different planar jet/slot thicknesses (i.e., 0.5 mm and 2.0 mm), and two different gap media (i.e., a 2 mm “empty” gap, a 2 mm wide bare strip in the center of the array to provide clearance for the impinging planar jet, and a hexagonal array of 808 1 mm diameter \( \times 2 \text{ mm} \) tall cylindrical pin fins on a 1.2 mm pitch). The integral pin-fin gap configuration includes a 2 mm wide bare strip in the center of the array to provide clearance for the impinging planar jet (Fig. 38). The array of pin fins greatly increases the total surface area cooled by the impinging jet; the selected array dimensions correspond to an area ratio \( \alpha = 3.76 \).

This area ratio is defined to be the total area of the sides of the fins \( A_f = 5.08 \times 10^{-3} \text{ m}^2 \) plus the area of the flat surfaces between the fins \( A_g = 9.54 \times 10^{-4} \text{ m}^2 \) (note that the area of the fin tips is not included in this area) divided by the area of the bare surface \( A = 1.59 \times 10^{-3} \text{ m}^2 \). The Al cartridge touches the top of the brass fins, which are directly etched by electron discharge machining (EDM) into the inner surface of the brass shell. The brass shell was again heated by three 750 W cartridge heaters embedded in the Cu-alloy heat concentrator.

The four different flow configurations were evaluated in an open-air flow loop at \( q'' = 0.22–0.75 \text{ MW/m}^2 \) and \( Re = 1.2 \times 10^4 \) to \( 4.9 \times 10^4 \) based on the hydraulic diameter of the slot (twice the slot width) and the average velocity of the air through the slot, \( \bar{V} \).
heat flux values given here are the electrical power input to the cartridge heaters divided by the area of the inner surface of the brass shell, i.e., A. As in the initial phase of the plate-type divertor studies, the range of Reynolds numbers examined spans the actual Re for the plate-type divertor at its proposed nominal operating conditions, but the heat fluxes are again significantly lower than the actual heat flux (10 MW/m²).

Fig. 39 compares the actual heat transfer coefficients \( h_{\text{act}} \) as a function of Re for the \( H = 0.5 \) mm (red triangles) and 2 mm (blue circles) planar jets impinging on the bare (open symbols) with the effective HTC \( h_{\text{eff}} \) for the pin-covered surfaces (filled symbols). At each of the 5 TC locations, the local HTC \( h_l \) was defined as follows:

\[
h_l = \frac{q''}{T_{\text{in}} - T_{\text{act}}}.
\]

Here, \( q'' \) is the surface heat flux (based for both the bare and pin-covered surfaces on \( A \)), \( T_{\text{act}} \) is the cooled surface temperature at that TC location determined by extrapolating the actual TC reading through 1 mm of the brass shell, and \( T_{\text{in}} \) is the temperature of the coolant at the test section inlet. The actual HTC was then the average of the \( h_l \) values at the 5 different TC locations; in all cases, the values for \( h_l \) varied by no more than ±5%. The surface heat flux is assumed to be uniform and equal to the incident heat flux at the brass shell surface as measured by the TC readings (i.e., temperature gradient) in the neck of the heat concentrator.

For the pin-covered surfaces, the effective HTC represents the value of the heat transfer coefficient that a bare surface would need to have to achieve the same base surface temperature as that for the finned surface subject to the same incident heat flux. Assuming that the tips of the fins are adiabatic, the effective HTC is then:

\[
h_{\text{eff}} = \frac{\Delta P + A \eta}{A} h_{\text{act}}
\]

where \( \eta \) is the pin-fin efficiency. The \( h_{\text{eff}} \) values for the finned surface cases are expected to be higher than the corresponding HTC values for the bare surface, since the temperature of the cooled surface is reduced by the presence of the fins (at a given \( q'' \)).

As expected, the \( h_{\text{eff}} \) values for the pin-fin covered surfaces at a given Re are much greater than the HTC values for the bare surface: the pin-fin array, which increases the cooled surface area by 276%, gives an increase in HTC of up to ~160% at the highest Re. The \( H = 2 \) mm planar jet gives slightly higher heat transfer coefficients than the \( H = 0.5 \) mm jet for the pin-covered surface. This may be due to the presence of the 2 mm wide clear area in the center of the pin-covered surface, which allows the narrower plane jet to spread more after impingement before flowing through the array of pins. For a given set of coolant flow conditions (i.e., Re), the ratio between the effective HTC for the finned surface and that for the bare surface can reach the surface area ratio \( \alpha = 3.76 \) (where \( \alpha \) is the ratio of the surface area of the pin-fin covered surface to that of the bare surface) only if the fins have an efficiency \( \eta \) of 100% (i.e., are infinitely conducting) and the actual surface HTC remains unaffected by the presence of the fins. Both of these conditions are never achieved in reality; hence, in all cases, the HTC enhancement ratio is less than \( \alpha \).

Fig. 40 compares the normalized pressure drops \( \Delta P \) as a function of Reynolds number Re. The legend is the same as that for the previous figure.
This maximum heat flux is:

\[ q''_{\text{max}} = \frac{T_S - T_{\text{in}}}{R_f} \]

For these estimates, the plasma-facing side of the pressure boundary, i.e., the front plate of W alloy, is assumed to operate at the maximum allowable temperature for that alloy, so \( T_S = 1300 \, ^\circ\text{C} \), and the coolant, He is assumed to enter the manifold at an inlet temperature \( T_{\text{in}} = 600 \, ^\circ\text{C} \). The total thermal resistance, \( R_f \) is the sum of the convective and conductive thermal resistances:

\[ R_f = \frac{1}{A_f} \left[ \frac{1}{h_{\text{eff,He}}} + \frac{\Delta x}{k_W} \right] \]

where \( h_{\text{eff,He}} \) is the effective HTC expected when using He as the coolant, \( \Delta x = 0.2 \, \text{cm} \) is the assumed thickness of the pressure boundary, and \( k_W \) is the thermal conductivity of the W alloy plate. For the pin-fin array case, the expected \( h_{\text{eff,He}} \) is equal to the value obtained for air rescaled by the ratio of the thermal conductivities \( k \) of the two gases (at their respective inlet temperatures) and corrected for the difference in fin efficiency for helium versus that for air:

\[ h_{\text{eff,He}} = \frac{k_{\text{He}}}{k_{\text{air}}} \left( \frac{A_f}{A_{\text{He}}} + \frac{\Delta x}{k_W} \right) / \left( \frac{A_f}{A_{\text{air}}} + \frac{\Delta x}{k_W} \right) \]

The thermal conductivity \( k_W \) was taken to be that of pure tungsten at 1300 °C, or 101 W/(m K). For the same pin-fin geometry and solid thermal conductivity, the fin efficiency for the He-cooled case \( \eta_{\text{He}} \) will be considerably lower than the corresponding value for air-cooled fins \( \eta_{\text{air}} \) at a given \( Re \). As shown in Fig. 41, the fin efficiency decreases as \( Re \) increases, with \( \eta_{\text{He}} \) and \( \eta_{\text{air}} \) decreasing from about 66% to 44% and 97% to 91%, respectively, as \( Re \) increases from \( 1.2 \times 10^4 \) to \( 4.8 \times 10^4 \). This effect is directly attributed to the higher HTC obtained with He, which increases the temperature gradient along the fin, thereby decreasing its efficiency.

Fig. 42 summarizes these estimates, showing \( q''_{\text{max}} \) as a function of \( Re \) for the \( H = 0.5 \, \text{mm} \) (red triangles) and \( 2 \, \text{mm} \) (blue circles) planar jets of helium impinging on the bare (open symbols) and pin-covered (filled symbols) surfaces. As expected \( q''_{\text{max}} \) for the bare surface at a \( Re \) near the prototypical value of \( 3.3 \times 10^4 \) is about 13 MW/m², well in excess of the nominal design value of 10 MW/m². These estimates suggest, however, that adding a pin-fin array to the jet impingement surface could increase the range of operating heat fluxes for the divertor to values of 17–18 MW/m² at the prototypical \( Re \) with nearly a 40% increase in pressure drop. The results also show that the maximum allowable heat flux values tend to “saturate” at higher Reynolds numbers. In other words, the enhancement in the maximum allowable heat flux due to the addition of pin fins decreases as \( Re \) increases because of the decrease in fin efficiency. At lower heat fluxes (~13 MW/m²), adding the fins would allow one to operate at a lower coolant mass flow rate (i.e. Reynolds number), thereby decreasing the pressure drop significantly.

It should be noted that the maximum allowable heat flux will be limited by the thermal stresses within the structure. Hence, the values shown in Fig. 42 represent lower bounds dictated only by heat transfer considerations.

4.1.4 Summary

The divertor thermal-hydraulic studies described in this section comprise numerical and experimental studies of the three leading helium-cooled divertor designs, namely, the HEMJ, T-tube, and plate-type divertor designs. Each of these studies has three common components: (a) A test section nearly duplicating the geometry, dimensions, and thermal conductivity of the actual divertor module is designed, constructed, and instrumented; (b) The test section is placed in an air test loop, where heated experiments are performed over a wide range of test conditions (air inlet conditions and flow rate) so that the governing non-dimensional parameter (jet Reynolds number) spans its expected nominal operating value in the actual helium-cooled divertor design; and (c) The experimentally measured temperature distributions and pressure drop are used to validate numerical predictions from a CFD model.

The results of this investigation verify the thermal performance goals (10 MW/m²) of these three divertor designs, as predicted by previous CFD studies. The validated numerical models examined in this investigation can be used with confidence to predict the thermal performance of the same divertor designs over a wide range of operating conditions. They can also be used to optimize the divertor designs by quantifying the effects of dimensional changes due to manufacturing tolerances, changes in geometry, and/or local vari-
ations in the coolant operating conditions. The experimental data suggest that for the plate-type divertor design, the thermal performance can be significantly enhanced by the addition of pin fins within the gap, albeit at the cost of a moderate increase in pressure drop.

4.2. High heat flux testing

4.2.1. Background

The US program in high heat flux testing of helium-cooled tungsten heatsinks for divertor applications concentrated on micro-fin and porous media devices. In 1993, Sandia constructed the closed helium flow loop, the HeFL, at Sandia’s Plasma Materials Test Facility as part of a Cooperative Research and Development Agreement with Creare, Inc. This loop operates at a maximum working pressure of 4 MPa and can provide up to 22 g/s of helium for low pressure drop devices. During the 1990s, the US effort, mostly in the Small Business Innovation Research Program (SBIR) sponsored by DOE concentrated on electron beam high heat flux testing of helium-cooled copper devices with the objective of demonstrating that helium cooling could be just as effective as water cooling in heat removal. These devices achieved heat transfer enhancement through a combination of turbulence promotion and extended areas for convective cooling. Several designs were tested, including the normal flow heat exchanger developed by Creare, Inc. [24], the porous metal Faraday shield developed by Thermacore, Inc. [25,26] and the copper micro-channel heatsink developed by General Atomics [27]. The Faraday shield was the first multi-channel device tested for parallel flow instabilities [28]. In the latter half of the decade, General Atomics developed a helium-cooled vanadium module that was tested to a maximum heat flux of 9 MW/m² [29]. We tested the first helium-cooled tungsten device fabricated by Thermacore, Inc. in 1999. The device absorbed a maximum heat flux of 12 MW/m² and produced a maximum surface temperature of 1100 °C using 20 g/s of helium at 4 MPa [30]. We investigated parallel flow instabilities at higher helium temperatures by performing a test with parallel flow in side-by-side modules.

In 2003, Ultratmet, Inc. provided the first axial flow, circular cross-section, all-tungsten device with internal tungsten foam. The connections failed prematurely during the test setup. Although we did gather flow data, it was impossible to obtain high heat flux test data. In 2005, we tested a similar device from Ultratmet, now equipped with niobium connection tubes, to a heat flux of 24 MW/m² with 20 g/s helium flow at 4 MPa [31]. During cool down at a slightly higher heat flux, brittle failure resulted in the rapid blowdown of the helium loop. Recently, in 2008, we tested three square-cross-section, molybdenum devices in circumferential flow. These were low pressure drop heatsinks with molybdenum foams of various densities from 18 to 23% and foam structures from 45 to 100 pores per inch (ppi) [32]. In these tests, molybdenum served as a surrogate for the more brittle tungsten. The thermal conductivity of molybdenum is 8–10% less than tungsten. This means the thermal performance is close to that of tungsten, but without the fear of a brittle failure at high helium pressures.

The remainder of this discussion will focus on the test results obtained for the foam-filled round tungsten tube and the flat surface molybdenum devices in circumferential flow. Currently, preparations are under way at the PMTF to test multi-channel helium-cooled molybdenum panels developed by Ultratmet over much larger areas of roughly 70 × 80 mm at elevated temperatures. An upgrade to the HeFL resulted in higher mass flows of 100 g/s at 4 MPa helium and piping to the 1.2 MW EB-1200 electron beam system now allows for larger area (1 m²), higher total absorbed power (400 kW), and higher temperature operation (1000 °C) than previously possible. In addition, testing of the ARIES T-tube concept fabricated by Plasma Process, Inc. [33] and testing of the 9-finger HEM module developed by FZK [34] will occur this year as part of an IEA exchange for Nuclear Technology of Fusion Reactors (NTFR). We hope to report these results in a series of papers next year.

4.2.2. Foam-in-tube (FIT) design

Ultratmet provided four tubular CVD tungsten, axial flow, helium-cooled devices for high heat flux testing at the PMTF. The tubes were fabricated under a DOE SBIR project. A sketch appears in Fig. 9, and Ref. [31] provides a thorough description of the module and the test results. Each tube was 152 mm long with a 12.7-mm-ID and a 15-mm-OD. Samples had foam lengths of either 38 or 51 mm. The foam was integrally bonded to the tube. Ultratmet vapor deposited the tungsten tube wall uniformly around the circumference of the foam to a thickness of roughly 1.1 mm. This ensured that thermal conduction into the foam ligaments would allow the foam to function as a fin as well as a turbulence promoter by disrupting the thermal boundary layer along the tube wall. Determination of the pressure drop flow characteristics and an evaluation of the steady state heat removal capability were the main objectives of our testing. The CVD tungsten tube ends were joined to pure niobium tubes (19.1 mm OD) by a thick CVD layer. The niobium enabled use of standard compression fittings for attaching the device to the stainless steel tubing on the HeFL.

Details on the flow testing and reported pressure drops appear in reference [31]. The flow tests characterized tubes containing both 10 ppi and 20 ppi foam, 20% dense, of 38 and 51 mm lengths, as well as a tungsten tube with no foam.

The 30 kW electron beam test system (EBTS) provided the applied heat flux from a 2-mm-DIA rastered beam spot at 30 keV. The EBTS applied heat load centered on the specimen over a 38-mm-length of the tube over the porous tungsten foam. The heat load has a cosine distribution on the circular tube with a peak along the top centerline of the tube surface normal to the beam. Real-time calorimetry provided a measure of the beam power absorbed in the helium. This was obtained by measuring the mass flow through an orifice flow meter and the temperature difference between the inlet and outlet helium streams. One one-color and two two-color spot pyrometers provided a measure of the surface temperature in the center of the heated area. A scanning infrared camera provided thermal images of the temperature distribution across the tubes. The sample received a maximum absorbed heat flux of 22.4 MW/m² along the top surface centerline normal to the beam. Helium at 4 MPa and an inlet temperature of 40 °C, flowing at 27 g/s produced a pressure drop of 92 kPa and a 51 °C rise in the helium temperature. The average absorbed heat flux over the entire projected surface area was 14 MW/m². Fig. 43 presents the peak surface temperature response for a sequence of increasing heat loads. The surface temperature reached a maximum of 2295 °C measured by 2-color spot pyrometry while the flowing helium removed 7.2 kW near steady state at the end of the 100 s loading. The results indicate that tungsten heatsinks can remove a significant amount of power, but only by operating at very high temperatures with a significant delta-T in the helium. The measured heat removal would be lower if the design temperature restrictions in Table 3 applied to the experiments without correction for mass flow rate.

An infrared thermogram appears in Fig. 44. During the cool down, the tungsten tube experienced a catastrophic brittle failure resulting in rapid blowdown of the HeFL. As shown in the photograph in Fig. 45, the heated portion of the sample containing the foam fragmented into a multitude of small pieces. Only the niobium connection ends remained intact. The sample was cycled through the DDBT as many as 50 times. The thermal stresses exceeded the yield strength by a factor of ∼3 before failure. These
results further illustrate the brittle nature of pure tungsten and its challenges as a structural material in high temperature divertor applications. Development of more ductile tungsten alloys or dispersion-strengthened composites is necessary before tungsten can function reliably in a high temperature divertor heatsink.

4.2.3. Short flow path foam-in-tube (SOFIT) design

Due to the risk associated with high heat flux testing of brittle tungsten, Ultramet and Sandia decided to fabricate the next generation heatsink from molybdenum. Molybdenum has about the same thermal properties of tungsten, but because the DBTT is near room temperature, it does not suffer from brittle fatigue. Yet, it allows for high temperature operation in the range comparable to tungsten. Molybdenum is not acceptable as a reactor material due to high activation. Here it serves as a surrogate for a future, more ductile tungsten alloy. In 2008, Ultramet delivered four single-channel, all molybdenum devices to the PMTF for high heat flux testing. These were 32 mm square by 254 mm long with the interior foam mesh over 127 mm of the axial length with a single, 2-mm-wide × 127 mm long slit centered under the heated area, which spanned less than 100 mm during the tests. The thickness of the top wall (heated) was 2 mm thick and others were 3 mm.

Fig. 44. An infrared thermogram shows the temperature distribution on the outside of the tungsten tube after 100 s exposure to 22.4 MW/m² heat flux.

Fig. 45. (a) Photograph inside the EBTS vacuum chamber shows the 100-ppi foam PFIT module before testing. (b) Post-test photograph of the PFIT after brittle failure and blowdown of the HeFL shows remnants of the plumbing connections.
Fig. 46. Flow characteristics for the SOFIT device with various foam properties reveal extremely low pressure drops that are not a function of axial length.

Fig. 47. Thermogram of the SOFIT under heat loading illustrates the effects of flow instabilities on the non-uniform surface temperature distribution.
Flow conditions in testing were similar to the axial flow device, 22 g/s helium at 4 MPa. The electron beam rastered over a 100 by 32 mm heated area in the center of the foam. Initially, the surface temperature limit was set at 900 °C due to concerns about the braze joint. The highest heat flux achieved with this limit was 3.6 MW/m² on the 100-ppi device.

The first test campaign, using 22 g/s of mass flow and the original HeFL blower, revealed an inherent flow instability in these devices with a 2-mm-wide slot, observed in our IR camera as a hot zone on the heated surface on one side of the flow slot. The thermograms in Fig. 47 illustrate this behavior and compares images after a small shift in the raster pattern caused the hot stripe to shift to the opposite side. The size and uniformity of heat flux distribution, gun power and total helium flow rate did not change during these shots. We attribute the non-uniformity in surface temperature to asymmetric helium flow from the center slot. Expansion of the helium directly under the heated area decreases its density and increases the pressure drop if no bypass flow can occur. We surmise that the hotter zone indicates a region of low mass flow (lower density and higher temperature) while helium on the opposite side with higher density and greater mass flow cools more effectively. Also, the foam does not cover the full axial length of the device, and, particularly near the ends of the heated area, the flow will likely move toward the cooler unheated regions where there is less flow resistance.

Subsequent testing used our new and larger HeFL blower and a higher back-pressure at the sample exit. In these tests, also at 22 g/s and 4 MPa, the flow was stable with no hot stripes on the surface. For the 100 ppi sample, a heat flux of 8.5 MW/m² resulted in a 1510°C peak surface temperature using 28 g/s of helium at 4 MPa and 64°C inlet temperature. Fig. 48 summarizes the surface temperature response as a function of absorbed heat flux. The heat removal would be lower if one imposes the limits for helium and tungsten temperatures suggested in Table 3. However, at the higher pressure of 10 MPa stated in Table 3, the increased mass flow rate likely would produce comparable heat removal at lower temperatures. Data taken during flow instabilities appear in the rectangles. The surface temperature distribution obtained with a higher back-pressure and no flow instabilities appears in the thermogram in Fig. 49.

4.2.4. Discussion

These experiments highlight several important points about helium-cooled devices. First, Fig. 42 shows that these devices remove a greater heat flux when they can operate at very high temperatures and the delta-T (T_{out} - T_{in}) is large. Restrictive conditions, either by a limit on helium inlet and outlet temperatures (for effective electricity generation) or the minimum and maximum structural temperatures in order to avoid embrittlement as well as re-crystallization, limit the performance of all the concepts described.

Second, although a very low pressure drop requires less pumping power, which is an advantage for a power plant, flow instabilities can arise more easily where bypass flow is possible. This can be alleviated by increasing the back-pressure or careful design to eliminate bypass flow completely. Third, the devices with foam integrally bonded to the heated faceplate perform 20–30% better than those that only promote turbulence. The higher the thermal conductivity of the foam ligaments, the more pronounced is the fin effect, so the effect will be greater for copper than for the refractory metals. Fourth, higher flow rates mean increased turbulence and result in better heat transfer and higher pressure drops.

Finally, design integration is exceedingly important, and high heat flux testing is a necessary to validate any design concept. Manifolding and careful fabrication are as critical to performance as...
reliability. Design “compromises” that ease fabrication but permit instabilities or maldistributions in the flow are not acceptable. Substantive issues related to joining and materials properties directly affect reliability. The device must be leak tight at high helium pressure, survive thermal stresses and multiple thermal cycles (perhaps through the DBTT of tungsten armor), and mitigate or accommodate grain growth in the heated zones. Successful resolution of these issues for a clever concept is necessary to realize a deployable, robust helium-cooled plasma facing component (PFC). Minimizing the number of joints and the potential for leaks, especially between dissimilar materials with large differences in thermal expansion must be one goal for successful development of helium-cooled PFCs. For PFCs with large temperature gradients and high thermal stresses near pressure boundaries, we need more ductile materials (particularly for tungsten alloys) with high fracture toughness than are now available.

5. Material considerations

5.1. Properties under irradiation and required development

Dedicated high fluence irradiation tests are mandatory before a suitable structural material can be selected. However, even without irradiation, more data from mechanical tests of the candidate materials are necessary to judge the feasibility of divertor target concepts based on these materials.

There are different requirements on the properties of a suitable tungsten alloy for divertor targets, depending on the different radial locations in the plate:

(A) Plasma facing surface

This front region reaches the highest material temperature and considerable erosion by energetic particles. It is designed with a sacrificial layer of ~5 mm giving an expected lifetime of a few years. High mechanical strength is not required for this layer, but blistering by implanted He atoms can be a crucial issue. There are suggestions to use a kind of porous tungsten in order to have micro-channels for the release of helium without blistering. It may be possible to achieve this by using tiles fabricated by Hot Isostatic Pressing (HIPing) of very small grain size powder, or by Powder Injection Molding. A suitable material for such elements is probably pure tungsten.

(B) Region between sacrificial tiles and the surface cooled by the He flow

This is usually the most critical zone with high primary stresses caused by the coolant pressure and large thermal stresses caused by a large temperature gradient. Either thin-wall tubes (T-tube concept) or cylindrical thimbles (finger concept) are used in this zone. The maximum temperature in these elements must be maintained below the re-crystallization temperature, and the minimum temperature at the cooler side above the embrittlement temperature of the material. Considering the large heat flux in this region, a re-crystallization temperature of at least 1300°C and an embrittlement temperature below 800°C is mandatory, and must be maintained under high fluence neutron irradiation.

Candidate materials are tungsten-rhenium alloys, dispersion strengthened tungsten with lanthanum-oxide, (for example WL10 composed of W-1%La2O3), and Vacuum-Metallized (VM) tungsten, doped with small amounts (in ppm-range) of aluminum and potassium silicate.

Rhenium leads to a lower thermal conductivity and relatively high activation by neutron irradiation. Tungsten with lanthanum oxide has a rather limited temperature window, and VM-tungsten can be fabricated only as thin wires and sheets (up to 2 mm thickness).

Based on the present knowledge of the mechanical behavior of the candidate materials, the most promising material for the thin-walled thimble of the finger-concept is VM-tungsten. The mechanical properties of this material depend strongly on the wall thickness of the sheets fabricated by rolling with the interesting aspect that thinner is better. This can be illustrated with the following properties of unirradiated wires of VM-tungsten:

<table>
<thead>
<tr>
<th>Wire diameter</th>
<th>Tensile strength at 20°C</th>
<th>Transition temperature</th>
<th>Incipient re-crystallization</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6 mm</td>
<td>&gt;2100 N/mm²</td>
<td>&lt; -50°C</td>
<td>~1800°C</td>
</tr>
<tr>
<td>1.0 mm</td>
<td>&gt;1800 N/mm²</td>
<td>&lt; 0°C</td>
<td>~1500°C</td>
</tr>
</tbody>
</table>

A potential drawback of VM-tungsten is its strong de-lamination under bending loads, especially in Charpy impact tests. This tendency is more pronounced here than with other tungsten alloys, resulting in a higher transition temperature. However, for a component like a thimble, loaded mainly with tensile stresses, delamination is of much smaller concern.

The proposed fabrication method for the thimble is to start with a 1-mm thick sheet rolled in alternating directions and to obtain the desired shape of the thimble by deep drawing or press rolling at a sufficiently high temperature.

(C) Structure of the divertor target plate, serving as coolant manifold

Here a distinction has to be made between the three different divertor target concepts.

1. In the original plate concept there is one material only, serving as sacrificial layer, divertor structure and helium manifold. This makes the material selection especially difficult because in some regions the allowable temperature is limited by re-crystallization, and in other regions by embrittlement. The required wall thickness of the elements of this concept is at least 6 mm. The use of VM-tungsten is not possible, and from all the available materials an alloy like WL10 is probably the only credible choice. With this material, ductility is a crucial issue for this concept, requiring operating temperatures greater than 800°C. Since re-crystallization limits the maximum allowable temperature to values below 1300°C, the maximum allowable heat load is probably 8 MW/m² at most.

The situation is different if the plasma-facing surface is subdivided into small modules minimizing the thermal stresses. There are two concepts employing this technique - the European “finger” concept and the US “integrated design concept.”

2. In the European divertor concept the small fingers are connected to a steel ring, which is finally EB welded to the manifold plate fabricated with ODS-steel. A critical item with this concept is the connection from the fingers (W-thimble) to the ODS steel of the manifold plate. There is a mismatch in the allowable temperatures (>800°C for the W-thimble, <750°C for available ODS-steel), and the different thermal expansion coefficients for the two materials (~4 × 10⁻⁶ K⁻¹ for W, and ~10⁻⁵ K⁻¹ for ferritic steel).

3. In the US integrated divertor concept the manifold plate is fabricated of tungsten, and the W thimbles are brazed directly into the W plate. The operating temperature of this plate is maintained nearly uniform at ~800°C by the exit temperature of the He coolant (700–800°C). Therefore the strength requirements for the plate material are moderate, and re-crystallization is not a relevant issue. For these reasons, “normal” tungsten is probably a feasible material.

A crucial issue of this divertor concept is the transition between the tungsten plate and the ODS steel of the coolant
access pipes at both ends of the plate. Volumetric heat generation at these locations is considerably lower than in the high heat flux region, and the surface heat flux there is nearly zero. A special design of this transition has been suggested, employing a tantalum interface between tungsten and steel. Tantalum matches the allowable temperature range of tungsten (higher) and ODS steel (lower), and has a thermal expansion coefficient between those materials. A conceptual design and a suggested fabrication method for this region are described in the following section.

5.2. Fabrication procedures and issues

5.2.1. Plate

As already mentioned in the previous section, there are different candidate materials and fabrication steps required for the divertor target plates, depending on the divertor concept. For the original plate concept as well as for the US integrated concept, the plate is a panel with a thickness of the order of 6 cm with parallel helium coolant channels, fabricated from a tungsten alloy. There is the possibility to machine the coolant channels with typical dimensions of 2 cm (width) by 5 cm (height) either by EDM (Electrical Discharge Machining) or ECM (Electro-Chemical Machining) into a solid tungsten plate. This would, however, have the decisive disadvantage of rather poor mechanical properties of the plate because the thickness reduction starting with a given block of ~15 cm thickness would not be sufficient to achieve “good” material properties. Better properties can be obtained when the panel is composed of a front plate, back plate and side-walls, each of them <10 mm thick and brazed together with a high temperature brazing material in a furnace.

Fig. 50 illustrates the different pieces involved in the design and in the fabrication and assembly. It includes:

- a front plate with castellation and grooves for brazing the side walls,
- a back plate with grooves for brazing in the side walls of large helium channels, and
- side plates.

An interesting detail to be seen there is the design of the brazing surface with a total area about two times as large as the cross-section of the sidewall (Fig. 50b). By this method, the primary stresses in the brazed area are much lower than in the sidewall itself.

Another important issue is the selection of the brazing material. For the original plate concept, the operating temperature in the brazed area can assume values up to ~1200 °C. For the US integrated divertor concept, the temperature at this location during divertor operation will be <1000 °C, but a slightly higher re-melting temperature will be required when the small finger modules are brazed into the plate. For both applications there are a number of suitable candidate brazing materials with melting points around 1300 °C, allowing the brazing operation to proceed without the risk of re-crystallization of the structural material.

5.2.2. T-tube

The T-tube consists of a W-alloy inner cartridge and outer tube on top of which a W castellated armor layer is attached. The design of such high heat flux divertor components with W-alloy and ODS FS assumes substantial progress in the development of these structural materials. It is envisaged to connect both W alloy pieces to a base ODS FS unit through a graded transition to minimize thermal stresses. The fabrication of complex W components is challenging (especially for such thin-walled W tubes, of the order of 1 mm), and needs to be demonstrated. An effort is ongoing at Plasma Processes Inc. as part of a DoE SBIR grant to fabricate mock-ups of such a component and to then test them in an electron beam high heat flux facility to provide a better assessment of the T-tube performance. The investigated fabrication and joining methods include EL-form and brazing.

5.2.3. Fingers

“Fingers” in the EU divertor concept as well as in the US integrated concept are small plasma facing modules composed of a cup-shaped thimble and a hexagonal tile serving as sacrificial layer for erosion. Typical dimensions of the thimble are a diameter of 15–20 mm and a wall thickness of ~1 mm. In the EU modular concept, this thimble is brazed to a cylindrical ring made of ODS-steel which is EB-welded later to the ODS manifold plate [5,6].

The most critical element of the fingers is the thimble, which gets primary stresses from the helium pressure, and rather high thermal stresses caused by a large thermal gradient resulting from the extremely high heat flux. An important issue is reliability since the thimble is the only barrier between high-pressure helium and the vacuum in the plasma chamber. In order to increase the reliability of the target plates, we suggested to insert a cylindrical tungsten ring into the thimble, providing a double containment at the most critical location where a 1-mm thick wall was the only barrier between the ~10 MPa helium and the high quality vacuum.

With this modification of the finger concept, the anticipated fabrication sequence includes the following steps:

(a) Fabricate the hexagonal tiles from small grain size powder (nano-sized particles?) either by powder HIPing or by injection molding,
(b) Fabricate the thimbles from VM tungsten by deep drawing or press rolling of ~1 mm thick sheets (thin sheet made by rolling in alternating directions),
(c) Put the tiles on top of the thimble and the cylindrical ring inside the thimble,
(d) Introduce a suitable brazing material with a melting point >1300 °C between ring and thimble as well as between thimble and tile, and
(e) Braze the three elements (tile, thimble and ring) together in a furnace.

5.2.4. Integrated concept

The original idea of the integrated concept was to subdivide the divertor target plate into two zones: a high power zone and a zone with lower surface heat flux. For this purpose, the entire plate is fabricated as described in Section 5.2.1, but in the high heat flux zones there are holes machined for the insertion of the modular fingers as described in Section 5.2.3. Fig. 51 shows and exploded view of the various elements of the design concept.

All the small finger modules are brazed into the front plate in a one-step furnace operation. An improved characteristic of this design is that this braze is made between two W alloy parts; in the original finger design, the W-alloy front part is brazed to an ODS FS back part requiring a technique to compensate for differential thermal expansion between the two materials.

For the two heat flux zone concept, the manifolds for the small modules are inserted into the plate unit and aligned with the front plate: the inlet manifold for the low heat flux zone is inserted from the end of the low heat flux zone and aligned with the front plate; and, finally, the outlet manifold for the low heat flux zone is inserted from the end of the low heat flux zone.

5.2.5. Fabrication of the transition zone at both ends of the target plate

In both plate divertor concepts – the original plate concept as well as the US integrated concept – the target plates are fabricated...
An important point is the transition from the W-alloy plate to the ODS steel coolant access pipes. A crucial issue there is to avoid thermocyclic plasticization of the W-alloy/steel joints due to a large mismatch of thermal expansion coefficient of the W ($4-6 \times 10^{-6} \text{K}^{-1}$) and the ODS steel ($10-14 \times 10^{-6} \text{K}^{-1}$). An idea for designing the transition region is to place a Ta-alloy between the W-alloy and the ODS steel because the thermal expansion coefficient of the Ta-alloy is between that of W-alloy and ODS steel. The transition concept is illustrated in Figs. 52 and 53.

The procedure we are considering for making the joint involves the following steps:

(a) explosion welding between the thin ODS ring and the thin Ta-ring,
(b) making seal welds between the thin ODS ring and the ODS manifold,
(c) diffusion welding of the thin ODS ring to the ODS manifold,
(d) brazing the thick Ta ring to the W-alloy plate,
(e) TIG or laser weld between the thin Ta ring and the larger Ta part

6. Summary and conclusions

In this article, we reviewed the current status of divertor design and R&D in the United States using tungsten alloys and helium cooling. Helium is a desirable coolant for use at high operating
Our current understanding of plasma heat loads in a tokamak power plant divertor suggests that steady state values greater than 10 MW/m² are likely, at least in some locations. Our engineering analysis suggests that proper design and use of heat transfer enhancement techniques can accommodate these high values of surface heat flux with acceptable coolant pumping power. By customizing the design for non-uniformities in the heating profile, assuming those profiles are known, the design complexity and pumping power can be minimized. The loading conditions during transient events in tokamaks, such as disruptions, vertical displacement events and edge localized modes, are not presently well known. However, they are undoubtedly more severe than steady-state conditions. Avoidance of such transients cannot be guaranteed, such that more detailed analysis is required to determine the operating limits on high heat flux components over their full range of thermal loads.

Small-scale testing of thermal hydraulics and thermomechanics has been performed in mockup experiments at Georgia Institute of Technology and Sandia National Laboratories. The results of thermal-hydraulic testing confirm the high heat removal capability of He-cooled design concepts and also validate model predictions, providing a high degree of confidence in this aspect of divertor performance. The results of thermo-mechanical testing are less convincing. More effort is required, in particular using advanced tungsten alloys, in order to provide confidence in component survivability under normal and off-normal conditions.

All of the power plant divertor designs discussed in this article utilize tungsten or tungsten alloy as structural material. It is a low-activation material (compared to other refractory alloys such as molybdenum) and possesses a high melting point, high thermal conductivity, and low thermal expansion. Its disadvantages lie in the relatively small operation temperature window, which is dictated by the ductile-brittle transition temperature (DBTT) at the lower boundary and the re-crystallization temperature (RCT) at the upper boundary. The RCT and machinability of tungsten can be improved by adding fine oxide particles. A serious material R&D effort is required: (a) to widen the W alloy operating temperature range from ~600–700 °C (governed by W ductility considerations) to enable coupling to an oxide-dispersion-strengthened (ODS) ferritic steel manifold, to ~1300 °C (governed by the W re-crystallization limit) to provide desirable high-temperature capability; and (b) to develop an ODS ferritic steel offering acceptable strength properties up to about 750 °C (e.g. nano-sized ODS FS).

Acknowledgements

This work was supported in part under U.S. Department of Energy grant numbers DE-FG02-04ER54757 and DE-FG02-01ER54656. Sandia is a multi-program laboratory operated by Sandia Corporation, a Lockheed Martin Company, for the United States Department of Energy’s National Nuclear Security Administration under Contract DE-AC04-94AL85000.

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