Design Optimization of
Helium Cooled PbLi Blanket for ARIES-ST

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by

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LIST OF SYMBOLS

\( \Delta T_{film} = \) film drop

\( \Delta T_{cond} = \) conduction drop

\( \Delta T_{bulk} = \) bulk temperature rise

\( \dot{q} = \) heat flux

\( h = \) convective heat transfer coefficient

\( k_{He} = \) Thermal conductivity of helium

\( D_h = \) hydraulic diameter

\( \text{Re} = \) Reynold’s number

\( A = \) cross sectional area

\( \mathcal{P} = \) wetted perimeter

\( a = \) width of square channel (equivalent to hydraulic diameter)

\( \rho_{He} = \) helium density

\( P = \) pressure

\( T = \) temperature

\( V = \) velocity

\( \mu_{He} = \) viscosity of helium

\( t = \) first wall thickness

\( k_{steel} = \) thermal conductivity of steel

\( c_1 = \) constant

\( c_2 = \) constant

\( P.P. = \) pumping power

\( T.P. = \) thermal power
\[ f = \text{Moody friction factor} \]

\[ \varepsilon /d = \text{roughness factor} \]

\[ c_p = \text{specific heat of helium} \]

\[ L = \text{length of duct} \]
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ABSTRACT OF THE THESIS

Design Optimization of
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Master of Science in
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High power density is essential in a fusion reactor to provide cost-effective
electric power generation. However, the blanket (the power removal component)
limits the amount of heat flux possible in the reactor due to its own thermohydraulic
and thermomechanical limitations.

High temperature, high-strength, low-activation, ferritic steel alloys with oxide
dispersion strengthening (ODS) are considered as a relatively near-term option for the
structural material of the blanket. In combination with helium, the system offers significant safety and performance advantages for power removal. A helium cooled ferritic steel structure of a dual coolant power core for ARIES-ST is investigated in this work. Thermohydraulic and thermomechanical behavior are analyzed to determine power density limit of the power removal system. Optimization of coolant channel dimensions in combination with artificial surface roughening to enhance heat transfer across the 3 mm first wall maximizes the thermohydraulic limit of the design. Use of ODS steel provides better mechanical properties than MANET suggested for EU-DEMO, which has a similar blanket design. The thermomechanical limit of the proposed blanket is determined by thermal-stress analysis using the commercial code ANSYS, version 5.3. The power density limit of the proposed ARIES-ST blanket is found to be nearly double that of the EU-DEMO design from 1995.
1. Introduction

High power density with high thermal conversion efficiency is essential in a fusion reactor in order to provide acceptable economics. The blanket is the main component that converts energy to a usable form, provides shielding from the environment, and breeds tritium used to fuel the plasma in the reactor. Tritium breeding reactions are as follows:

\[ ^{6}\text{Li} + _{0}^{1}\text{n} \rightarrow ^{4}\text{He} + ^{3}\text{T} \]

\[ ^{7}\text{Li} + _{0}^{1}\text{n} \rightarrow ^{4}\text{He} + ^{3}\text{T} + _{0}^{1}\text{n} \]

There are two main classes of blankets: liquid breeders and solid breeders. Liquid breeders use lithium or a lead-lithium alloy and solid breeders use ceramics containing lithium. It can be shown that the desired level of power density and thermal conversion efficiency cannot be achieved by using solid breeder blankets [1]. Using a liquid breeder material provides the ability to handle high power densities. A dual coolant blanket using the eutectic lead-lithium alloy Pb-17Li as the tritium breeder and breeding zone coolant with a helium-cooled first wall has been studied in the European Union (EU) for several years [2]. This dual coolant blanket design has many advantages and is the type chosen for the ARIES-ST study [1]. The ARIES-ST effort is a conceptual design study of a low aspect ratio Spherical Tokamak (ST), that is part of an ongoing effort to study the feasibility of using magnetic fusion devices in future electric power generation facilities. Figure 1 is a cross-section of the toroidal-shaped ARIES-ST reactor. Power comes from a plasma that is produced in the center of the D-shaped cross-section. The power removal system, or blanket, must be
Figure 1: Reactor Cross-Section
tailored to work efficiently and safely in the high heat flux conditions of the magnetic fusion reactor. A blanket segment is shown in Figure 1 by the red, curved area. Figure 2 shows the blanket cross-section cut by the horizontal axis in Figure 1. Liquid lithium breeder is flowing into and out of the page through twelve square ducts. The eight rectangular ducts along the back supply helium to the first wall.

This work is concerned with the helium-cooled first wall (FW) of the blanket where most of the surface heat flux is removed. The FW is the perimeter (back side excluded) of the section in Figure 2. Figure 3 shows the FW cross-section. The goal is to determine an optimum configuration to remove the maximum amount of power for efficient use in a thermodynamic cycle while retaining the structural integrity of the design. High temperature, high-strength, low-activation, ferritic steel alloys are considered as a relatively near-term option for the primary pressure vessel and structural material. Combined with helium, the system offers significant safety and performance advantages for power removal. Two methods are used to enhance the efficiency of the blanket. First, surface roughening is used on the plasma facing wall of the coolant channels in order to double the convective heat transfer coefficient. Second, keeping the FW thickness to a minimum will reduce the amount of temperature drop across the FW due to conduction. These two methods help retain a temperature “budget” so that enough is left for a coolant bulk temperature rise that will make the Brayton thermal conversion cycle efficient for electric power generation.

Thermohydraulic and thermomechanical behavior are analyzed to ascertain the optimum sizing of the first wall cooling channels. The power density limit of a workable design is then determined.
Figure 2: Blanket Cross-Section

Figure 3: First Wall Cross-Section
II. Background

The dual-coolant design from the EU-DEMO program was analyzed extensively and all thermohydraulic and thermomechanical limits were met previously with a peak heat flux of 0.5 MW/m$^2$. Based on present goals, the first wall in ARIES-ST will see an average of 0.65 MW/m$^2$ if half the total transport power is deposited. A peak of 0.8 MW/m$^2$ is likely for this steady-state situation [3]. Therefore, the most important design goal is to meet the peak heat flux requirement of 0.8 MW/m$^2$. The second, and equally important, goal is for the design to yield a bulk temperature rise of 133 $^\circ$C in the first wall helium coolant circuit for compatibility with the helium Brayton power conversion cycle [4]. These two goals must be met while staying within the restrictions of the structural material temperature and stress limits. These numbers (heat flux and bulk temperature rise) may become higher as design goals are still evolving, so there is a desire to know the maximum capability of the blanket.

The ARIES-ST blanket is to be constructed of a high-temperature, high-strength, low-activation, ferritic steel alloy (EU designation MANET). The inlet temperature of the helium coolant must be above the Ductile-Brittle Transition Temperature (DBTT) of 300 $^\circ$C for MANET projected for radiation damage lifetime up to 200 displacements per atom (dpa) [5]. The outlet temperature of the coolant must be low enough to keep the peak structure temperature below 600 $^\circ$C. At this temperature limit, the yield strength of the steel is between 200-400 MPa depending on the dpa [6]. (EU-DEMO used 550 and 250 degrees C for the peak and minimum, respectively. Higher temperature Oxide Dispersion Strengthened Steels are under development [7-10]. We allowed a higher DBTT and peak temperature for these materials.)
The strategy for meeting the two goals of high heat flux (0.8 MW/m$^2$) and thermal conversion compatibility (133 $^\circ$C bulk temperature increase) consists of two parts: 1) Maximize the heat transfer from the first wall to the helium coolant, and 2) Minimize the pumping power to push the coolant. The first is achieved by enhancing convection and reducing the conduction drop across the FW by roughening of the coolant channel surfaces to increase the convective heat transfer coefficient and by minimizing the thickness of the FW, respectively. The second is accomplished by allowing surface roughening only on the FW facing portion of the square duct and by optimizing the area and length of the cooling channels. In this manner, the friction factor is increased only by one roughened side of the four-sided channel, thereby keeping pumping power losses to a minimum.

The yield strength of the structural material restricts the size of the coolant channels and the minimum thickness of the first wall. Pressure in the coolant channels creates membrane and bending primary stresses that limit the size of the coolant channels and limit how thin the first wall can be. The heat flux and thermal differential across the first wall set up thermal secondary stresses that limit the heat flux and how thick the first wall can be. With the primary and secondary stresses requiring opposite conditions on the first wall thickness, a dimension must be chosen to reach middle ground. A first wall that is thinner than the 5mm suggested by [11] is an improvement since large thermal stresses are reduced by lowering the temperature difference across the FW (The ITER FW is 3mm). Channel dimensions can be adjusted to alleviate the primary stresses that occur due to a thinner first wall.
III. Thermohydraulics Analysis

The power density limit from a thermohydraulics point of view is governed by two key parameters. One is the maximum temperature of the steel structure and the other is the amount of pumping power one wishes to sacrifice in order to circulate the helium coolant. Each will be treated with separate derivations and the results compared.

The thermohydraulics analysis is simplified by modeling the FW coolant circuit as a straight, square duct of length, $L$. Pumping power losses in the conduits leading to the first wall are not significant enough to warrant complicating the model. More pumping power will be required to force the coolant through these ducts. However, they may be made larger and smoother to keep pumping losses to a minimum since these areas will not be affected by high heat flux.

The two variables of interest are the hydraulic diameter of the square first wall (FW) duct and the velocity of flow. One would like to find the optimum hydraulic diameter and velocity for maximum power removal.

The following is the temperature budget expressed in terms of the minimum and maximum allowable temperatures in the structure ($T_{\text{steel}}^{\text{max}}$, $T_{\text{He}}^{\text{inlet}}$) and the various heat transfer temperature drops ($\Delta T_{\text{film}}$, $\Delta T_{\text{cond}}$, $\Delta T_{\text{bulk}}$):

$$T_{\text{max}} = T_{\text{inlet}} - T_{\text{outlet}} + \Delta T_{\text{film}} + \Delta T_{\text{cond}}$$

(1)

where,

$$T_{\text{outlet}} = T_{\text{inlet}} + \Delta T_{\text{bulk}}$$

(2)

resulting in:

$$T_{\text{max}} = T_{\text{inlet}} + \Delta T_{\text{bulk}} + \Delta T_{\text{film}} + \Delta T_{\text{cond}}$$

(3)
Whether using the EU DEMO temperature limits: \( T_{\text{steel}}^{\text{steel max}} = 550^0 \text{C} \) and \( T_{\text{He}}^{\text{inlet}} = 250^0 \text{C} \), or the proposed \( T_{\text{steel}}^{\text{steel max}} = 600^0 \text{C} \) and \( T_{\text{He}}^{!\text{inlet}} = 300^0 \text{C} \), the temperature budget works out to be \( 300^0 \text{C} \) in either case. Equation (3) then becomes:

\[
300 = \Delta T_{\text{bulk}} + \Delta T_{\text{film}} + \Delta T_{\text{cond}}
\]

(4)

For the ARIES-ST design, a \( \Delta T_{\text{bulk}} = 133^0 \text{C} \) is required in order to make the thermodynamic cycle compatible. Equation (4) becomes:

\[
167 = \Delta T_{\text{film}} + \Delta T_{\text{cond}}
\]

(5)

The convective film drop is

\[
\Delta T_{\text{film}} = \frac{\dot{q}}{h}
\]

(6)

where the convective heat transfer coefficient is:

\[
h = 0020 \frac{k_{\text{He}}}{D_h} \text{Re}^{0.8} F \quad \text{from [12] times enhancement factor, } F
\]

\[
k_{\text{He}} = 25(10)^{3} T^{0.72} \frac{\text{W}}{\text{mK}}
\]

\[
D_h = \frac{4A}{\mathcal{P}}
\]

\[F = \text{enhancement factor}
\]

\[A = \text{cross sectional area}
\]

\[\mathcal{P} = \text{wetted perimeter}
\]

\[D_h = a \quad \text{for square duct with dimension } a
\]

A square duct is analyzed in order to avoid effects caused by the aspect ratio of rectangular sections. A hydraulic diameter that nearly approximates a circle should be
used (square or rectangle with low aspect ratio) to provide accurate heat transfer results [12].

\[
\text{Re} = \frac{\rho_{He} V D_h}{\mu_{He}} = \frac{\rho_{He} V a}{\mu_{He}}
\]

\[
\rho_{He} = (48.75) \frac{P[\text{atm}]}{T[K]} \frac{\text{kg}}{m^3}
\]

\[
\mu_{He} = 32(10)^{-7} T^{0.72} \frac{\text{kg}}{m\text{s}}
\]

\[V = \text{flow velocity}\]

Enhancement of the convective heat transfer coefficient by a factor of 2 has been demonstrated by [13, 14] through the use of low transverse ribs. The factor, \(F = 2\), is included in constant \(c_1\) in Equation (8) below.

The conduction drop through the FW is:

\[
\Delta T_{\text{cond}} = \frac{\dot{q} t}{k_{\text{steel}}}
\]

where

\[
k_{\text{steel}} = 26.2 \frac{W}{mK} \text{ for MANET}
\]

\[t = \text{thickness of FW in meters}\]

Substitution of equations (6) and (7) into equation (5) yields the following:

\[
167 = c_1 \dot{q} a_{02} V^{-08} + c_2 \dot{q}
\]

where:

\[
c_1 = \frac{1}{(2)(002)k_{He} \sqrt{\frac{\rho_{He}}{\mu_{He}}}}
\]

\[
c_2 = \frac{t}{k_{\text{steel}}}
\]
Rearranging equation (8) yields:

\[
\dot{q} = \frac{167}{c_1\left(a^{0.02}V^{-0.8}\right) + c_2}
\]  

(9)

Equation (9) is evaluated for \( a = 0.1, 0.025, 0.01, 0.001, 0.0001, 0.00001 \) and 0.000001 and varying \( V \) and plotted in Figures 4-10. Material properties are evaluated at \( 350^\circ C = 623 \) K. Any point on or below the dashed line is a case where the maximum steel temperature will not exceed \( 600^\circ C \) (for a \( 300^\circ C \) inlet temp).

Pumping power is used as the governing parameter for the second derivation. The amount of pumping power must be specified as a percentage of thermal power removed. The following work uses 10\% as the pumping power limit to prevent the pumping power from substantially degrading the plant net efficiency. This requirement could be lifted to allow higher heat flux capability, but would detract from the economic advantage of this high power density.

Therefore,

\[
\frac{P.P.}{T.P.} = 0.10
\]

(10)

where the pumping power (in Watts) is:

\[
P.P. = VA\Delta p = V\alpha^2 \frac{fL\rho_{He} V^2}{2D_h} = \frac{V^3 qfL\rho_{He}}{2}
\]

(11)

and the thermal power (in Watts) is:

\[
T.P. = \dot{q}aL
\]

(12)
where

\( \dot{q} \) is the surface heat flux at the FW in W/m²

\( a \) is the width of the square duct

\( L \) is the length of the duct

Substitution of Eq. (11) and (12) into (10) yields:

\[
\dot{q} = \frac{V^3 \rho \nu_e}{0.2} \quad (13)
\]

The friction factor in the turbulent regime (\( Re > 10^4 \)) may be expressed explicitly by the following from [15]:

\[
f = \frac{1}{-18 \log \left( \frac{69}{Re} + \frac{\varepsilon/d}{37} \right)^{1.11}} \quad (14)
\]

where

\( \varepsilon/d \) = roughness factor

The friction factor in the laminar regime (\( Re < 2 \times 10^3 \)) is simply:

\( f = 64/Re \)

The overall friction factor is comprised of an \( f \) for the smooth walls and an \( f \) for the rough wall as follows:

\[
\begin{align*}
\varepsilon & \quad \text{smooth wall} \\
\beta & \quad \text{rough wall}
\end{align*}
\]
\[ f = \frac{af_a + 2bf_b + af_c}{2a + 2b} \] where \( c \) is the roughened wall and \( a \) and \( b \) are smooth.

For a square duct with side length \( a \) and one roughened wall, this simplifies to

\[ f = 0.25 f_{\text{rough}} + 0.75 f_{\text{smooth}} \]

where \( f_{\text{rough}} \) (commercial steel) and \( f_{\text{smooth}} \) (drawn tubing) are chosen

\( \frac{E}{d} \) \(_{\text{rough}} = 0.0018 \text{ m}^2/\text{m} \) and \( \frac{E}{d} \) \(_{\text{smooth}} = 0.00006 \text{ m}^2/\text{m} \), respectively from [15].

Equation (13) is plotted in Figures 4-10. Any point on or above the solid curve is a case where 10\% or less of the thermal power removed is used for pumping the coolant.

By comparing the results from Equation (9) in the same plots one can see the “window” in which the blanket can theoretically operate if \( \Delta T_{\text{bulk}} = 133 \text{ ^\circ C} \) and stress limits in the structure are ignored. Staying above the pumping power curve and below the steel temperature curve, an eye-shaped zone is created. It is within this zone that the blanket must operate to meet the temperature and pumping power restrictions. The optimum size for coolant channels from a thermohydraulic stance is on the order of a \( 10^{-4} \) meter hydraulic diameter according to the results in Figure 11, the maximum capability of the cases in Figures 4-10 (the location where the solid and dashed lines intersect at maximum heat flux are plotted for each case). The \( 10^{-4} \) m hydraulic diameter is capable of removing over 1.2 MW/m\(^2\).

A 25mm channel size was chosen in the EU-DEMO design since the first wall provides a structural role in addition to power removal. A 25 mm channel was used by [11] for the EU-DEMO proposed dual-coolant design and was studied extensively with
detailed stress analysis. Since the 25 mm channel shows a peak heat flux capability of over 0.95 MW/m$^2$, it exceeds the design requirement of 0.8 MW/m$^2$. For this reason, the design proposed by [11] will be modified slightly and used as the reference case for the thermomechanical analysis that will follow in Chapter IV. A 20% increase in $h$ could be obtained by reducing the channel size. The penalty is reduced bending strength, a particular concern in the event of an internal coolant leak which would pressurize the module. Design measures such as a rupture disk could alleviate the requirement of a 25mm channel size.

If we fix $\Delta T_{\text{bulk}} = 133^\circ$C, then the length, $L$, of the duct must be calculated from the definition of the bulk temperature rise:

$$\Delta T_{\text{bulk}} = \frac{T.P.}{\rho_{He} AV_c p} = \frac{\dot{q} L}{\rho_{He} aV_c p}$$

(14)

where

$$c_p = 5192 \frac{J}{kgK}$$

Therefore,

$$L = \frac{\rho_{He} aV_c p (133)}{\dot{q}}$$

For the reference case with a 25 mm duct, $L = 13.7$ meters.

If one were not concerned about achieving a large bulk temperature rise for use in the thermal conversion cycle, the blanket could theoretically remove a much larger amount of power. For example, if the bulk temp rise were 13 degrees, the blanket could remove 1.16 MW/m$^2$, a 21% increase over 0.96 MW/m$^2$.

The blanket proposed by [11] uses the same diameter duct with a 5mm FW. The limit for the 5mm FW is 0.65 MW/m$^2$. Therefore, a 48% gain in power removal to 0.96
MW/m$^2$ is attained over the EU DEMO model by reducing the FW thickness by 40% (2mm) from 5mm to 3mm. Roughened plasma facing walls were used in both cases, so no further gain resulted due to the artificial roughness.

The power removal limits determined by the thermohydraulics analysis are likely to be higher than what the structure can withstand due to stress limits. Besides the high temperature gradient in the FW, primary stresses resulting from the 8MPa helium pressure within the duct will limit the amount of heat flux the blanket can withstand. The stress analysis in Chapter IV will determine if the thermohydraulic limits presented here can be attained or not. The heat flux value used for stress analysis is 0.95 MW/m$^2$ in order to be comfortably within the thermohydraulic limits.
IV. Thermomechanics Analysis

IV.1. Introduction

In this chapter, stress analysis models are formulated in 2-D and 3-D using ANSYS 5.3 [16]. The results will show whether the design proposed (25 mm channel with 3mm FW) can perform under the thermal load determined to be thermohydraulically feasible (0.95 MW/m$^2$) in the previous chapter.

In accordance with the ASME pressure vessel code [17], level A loading allowables are as follows:

1) Allowable Membrane Primary Stress: 1 $S_m$, $t$

2) Allowable Membrane + Bending Primary Stress: 1.5 $S_m$, $t$

3) Allowable Primary + Secondary Stress: 3 $S_m$

with $S_m = \min(\frac{2}{3}\sigma_{0.2}, \frac{1}{3}\sigma_u)$ and $S_m, t = \min(S_m, \frac{2}{3}\sigma_{R,t}, \frac{1}{3}\sigma_{1,t})$ where $\sigma_{0.2}$ is the 0.2% offset yield stress, $\sigma_u$ is the tensile strength, $\sigma_{R,t}$ is the creep resistance, $\sigma_{1,t}$ is the 1% creep strain limit, and $t$ is the time to failure, with $t = 2(10)^4$ hours for steady state operation.

Using the same procedure as [11] in accordance with the ASME code [17], the stresses are computed in two steps. First, only pressure loads are applied to the model to calculate the primary stresses for use in the first two stress criteria, membrane and membrane-plus-bending stresses. Second, thermal loads are applied to the model in addition to the pressure loads and are used to calculate primary-plus-secondary stresses for the third criterion. For each of the two steps in the above procedure, models in 2-D and 3-D will examine the EU-DEMO design presented by [11] that has a 5mm FW and side walls with an 8mm second wall. These examples will benchmark the validity of
the finite element models presented herein by comparing them to the more complex 3-D treatment in [11]. Similar 2-D and 3-D models will then examine the proposed design (3 mm FW and 0.95 MW/m² heat flux).

Material properties for all modeling are as follows: Young’s modulus = 207 GPa, Poisson’s ratio = 0.3, Thermal conductivity = 26.2 W/m-K, Thermal expansion coefficient = 1.2(10)^{-5} 1/K. Pressure load, when applied, is 8 MPa for all examples.

All stress results are von Mises.

Design allowables from [11] for 1.4914 steel (MANET) and suggested allowables for ODS steel are as follows:

<table>
<thead>
<tr>
<th>T[°C]</th>
<th>S_m[MPa]</th>
<th>S_m, 2(10)^4 h [MPa]</th>
<th>S_m[MPa]</th>
<th>S_m, 2(10)^4 h [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>258</td>
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<td>121</td>
<td>41</td>
<td>144</td>
<td>98</td>
</tr>
</tbody>
</table>

The design allowables for ODS are assumed to have an advantage of 50°C over MANET, so the values for MANET are simply shifted by 50 degrees [7-10].

IV.2. Pressure Stress
IV.2.A. EU-DEMO Design

The 2-D model of the FW helium cooling duct from EU-DEMO [11] is formulated in ANSYS using a Plane13, Coupled-Field element. Element behavior is plane strain and the mesh is formed using the Smart Size level 1 (fine) automatic mesher.

Two load cases for pressure stress will be examined. The first is an internal pressure load applied to the interior walls of the coolant channels. The second is the first load case combined with pressure applied to the second wall (the pressure resulting from a helium coolant leak to the larger breeder channels).

Figure 12 shows the stress distribution for the internal pressure load of 8MPa only for two adjacent, 25mm, square channels representing the EU-DEMO design. The left edge has a symmetry boundary condition applied, while all other edges are free. The peak stress is about 90 MPa and occurs in the plasma facing surface. The high stress occurring at the right edge is ignored because it does not represent the actual design which would have another adjacent pressurized channel. Actual stress in the vertical sections is more accurately represented by the vertical member in the center of the model.

A 3-D model of the same load case is shown in Figure 13. This model uses only one and a half channels with the positive-x and positive-z faces subject to a symmetry boundary condition. The reason for using only one and a half versus two channels is to make the model small enough to run on the computer. The channels are 167 mm long and constrained in the y-direction at each end to be similar to the constraints in the model from [11]. A Solid5, Coupled-Field element is used for the mesh with the
automatic Smart Size mesher at level 1 (fine). The peak stress (again at the plasma facing wall while ignoring the unconstrained vertical member) is about 70 MPa. In comparison, the 2-D plane strain approach is slightly more conservative than the 3-D model, but it is still quite accurate for a pressure vessel type problem.

However, when the off-normal pressure case (a high pressure leak to the breeder channels from one of the helium coolant channels) studied by [11] is added to the problem, the 2-D model is no longer sufficient because of beam bending behavior between the $z$ faces as shown in Figure 14. The pressure in this region under normal conditions would not be nearly as high as the 8 MPa that results from a helium leak. The results from this 3-D ANSYS model correspond quite well with those in [11], although there are some differences in the stress distribution. Despite these differences, there is a good confidence level for the ANSYS results even though they are from a much simpler model than the one presented by [11].

For the proposed design, off-normal pressure will not be considered as it has already been determined unnecessary. A leak detection device will prevent such leaks from being a problem by relieving the 8MPa pressure before it can do any damage. Therefore, a 2-D treatment is a conservative and sufficient means of modeling the proposed design for internal pressure loads only.

IV.2.B. ARIES-ST Design

The proposed ARIES-ST design (3mm FW with 4mm webs and 3mm second wall) subject to internal pressure load in 2-D is shown in Figure 15. Figure 16 shows the 3-D von-Mises stress for the same load case. Note again that the 2-D model is very
close to the 3-D model when dealing with internal pressure only. The primary membrane condition applies in the web where the maximum stress is 68 MPa and the temperature 500 °C. This is well below the 1 $S_{m,t}$ limit of 183 MPa for ODS steel and is below the 1 $S_{m,t}$ limit of 165 MPa for MANET. The membrane plus bending condition applies at the first wall where the highest stress is 148 MPa at 580 °C. This is well below the 1.5 $S_{m,t}$ limit of 195 MPa for ODS steel, but does not meet the limit of 100 MPa for MANET. Therefore, the proposed design meets the first two criteria for ODS steel, but not MANET.

**IV.3. Combined Thermal and Pressure Stress**

**IV.3.A. EU-DEMO Design**

Figures 17 and 18 show the temperature profile in the EU-DEMO design [11] resulting from a heat flux load on the FW of 0.5 MW/m$^2$ for a 2-D and a 3-D case, respectively. A convective heat transfer coefficient of $h = 6,534$ W/m$^2$K is applied to the plasma facing walls of the interior of the channel while the other walls use $h/2 = 3,267$ W/m$^2$K because they are not artificially roughened. These convective values correspond to a helium velocity of 38 m/s to be consistent with the work in [11]. The bulk temperature of the coolant is 350 °C. Both 2-D and 3-D result in the same temperature profile.

Figures 19 and 20 show the stress distribution for a 2-D and a 3-D case, respectively, with combined thermal and pressure loading. The 2-D model has stresses an order of magnitude higher than the 3-D. Since the 2-D is plane strain behavior, it is basically a model of the 3-D case as if it were constrained at each end of the channel.
Running this situation in 3-D (Figure 21) proves that constraining both ends of the channel results in stresses that are the same as those for the 2-D plane strain model in (Figure 19). Plane stress behavior for the combined loading in 2-D results in stresses that are much lower than those in 3-D as evidenced by Figure 22. Therefore, no 2-D model is useful for the combined loading. To verify that the 3-D model is accurate for the combined thermal and pressure loading, the off-normal pressure is added as before and the results are presented in Figure 23. As with the 3-D pressure only case, the stress field is not identical to that of [11], but the model seems to agree within 10-30% of the minimum and maximum values from [11]. The 2-D models are not valid for the combined load case. For reference, Figures 24 and 25 show 2-D and 3-D stress results, respectively, for the thermal load case only.

IV.3.B. ARIES-ST Design

Figures 26 and 27 show the temperature profile for the proposed ARIES-ST design in 2-D and 3-D, respectively, for a heat flux of 0.95 MW/m$^2$ and convective heat transfer coefficient of $h=16,720$ W/m$^2$-K at the plasma facing wall and $h/2=8,360$ for the other channel walls (these values correspond to a helium velocity of 125 m/s). The bulk temperature is 400 $^\circ$C.

Figure 28 shows the von-Mises stress distribution in 3-D for the combined pressure and thermal loading. The highest stress occurs in the FW and is 430 MPa at a temperature of 575$^\circ$C. This is below the 3 $S_m$ limit of 464 MPa for ODS steel but does not meet the 398 MPa limit for MANET. For reference, Figure 29 shows the stress in 3-D due to the thermal loading only. No off-normal pressure case is considered. The proposed design now meets all of the ASME criteria for ODS steel.
V. Conclusions

The proposed helium-cooled, 3mm first wall using 25mm square channels with heat transfer enhancement of the plasma facing wall can theoretically remove 0.95 MW/m$^2$ according to both thermohydraulic and thermomechanical analysis. It is noted that a lot of performance, i.e. heat flux limit, is lost due to the desire for a high bulk temperature rise (133 °C). Thermohydraulic analysis has shown that the design can remove over 1.16 MW/m$^2$ if bulk temperature rise were not an issue. In combination with a reduced channel size ($10^{-4}$ m hydraulic diameter), an improvement to 1.22 MW/m$^2$ is feasible. Some of the performance lost due to bulk temperature constraints is already gained over the design by [11]; an almost 50% gain in heat flux is realized over the EU-DEMO design by reducing the first wall thickness by 2mm.

Use of oxide dispersion strengthened (ODS) steel instead of MANET allows the design to reach its full potential. The design is able to withstand the 0.95 MW/m$^2$ heat flux from a thermomechanical standpoint, depending on the fidelity of the thermal-stress analysis results. Comparing the results of the ANSYS models to the work in [11] for EU-DEMO, the stress distribution of the models are quite different but are of similar magnitude for minimum and maximum values. The two models differ by about 30%, but this is good agreement since the ANSYS models use a lot fewer elements than the work in [11].

It has been discovered that 2-D plane strain elements are fairly good at describing the stresses due to internal pressure only. Results are consistently within 25% of 3-D figures and are on the conservative side since the 2-D plane strain is like a 3-D model constrained in one axis. Off-normal pressures induce bending which requires 3-D
analysis. The same can be argued for thermal loads which should only be applied when using 3-D analysis. The plane strain behavior of the 2-D model is overconstraining in thermal analysis and pushes the stresses up an order of magnitude over the actual stress values that are found in 3-D analysis.

Boundary conditions for 2-D analysis are fairly straightforward (constrain one edge), while those in 3-D are not. For 3-D, symmetric conditions on two adjacent outer sides seem to work best, but the quasi-plane strain condition that [11] is able to apply does not seem to be possible when using ANSYS.

Better creep strength and yield strength result from using oxide dispersion strengthened steels that are presently being developed. There are high temperature ferritic steel alloys under development that are expected to have a 50 °C temperature range extension over MANET before yield strength degradation [7-10]. These materials will provide the added confidence needed to push the present design to its thermohydraulic limit while remaining safely within the thermomechanical boundary.
Appendix

Figure 4: Equations 9 and 13 for 0.000001m channel

Figure 5: Equations 9 and 13 for 0.00001m channel
Figure 6: Equations 9 and 13 for 0.0001m channel

Figure 7: Equations 9 and 13 for 0.001m channel
Figure 8: Equations 9 and 13 for 0.01m channel

Figure 9: Equations 9 and 13 for 0.025m channel
Figure 10: Equations 9 and 13 for 0.1m channel

Figure 11: Maximum capability of cases in Fig 4 - Fig 11
Figure 12: Pressure stress in 2-D (EU-DEMO)
Figure 13: Pressure stress in 3-D (EU-DEMO)
Figure 14: Pressure stress with off-normal pressure condition (EU-DEMO)
Figure 15: Pressure stress in 2-D (Proposed design, ARIES-ST)
Figure 16: Pressure stress in 3-D (Proposed design, ARIES-ST)
Figure 17: Temperature profile in 2-D (EU-DEMO)
Figure 18: Temperature profile in 3-D (EU-DEMO)
Figure 19: Combined thermal and pressure stress in 2-D (EU-DEMO)
Figure 20: Combined thermal and pressure stress in 3-D (EU-DEMO)
Figure 21: Combined stresses with channel ends constrained (EU-DEMO)
Figure 22: Combined stresses 2-D plane stress behavior (EU-DEMO)
Figure 23: Combined stresses with off-normal pressure condition (EU-DEMO)
Figure 24: Thermal stress in 2-D (EU-DEMO)
Figure 25: Thermal stress in 3-D (EU-DEMO)
Figure 26: Temperature profile in 2-D (Proposed design, ARIES-ST)
Figure 27: Temperature profile in 3-D (Proposed design, ARIES-ST)
Figure 28: Combined thermal and pressure stresses (Proposed design, ARIES-ST)
Figure 29: Thermal stress (Proposed design, ARIES-ST)
References


