OPTIMIZATION OF THE ARIES T-TUBE DIVERTOR CONCEPT

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The current T-tube divertor design consists of modular helium cooled units. The individual units consist of a steel inner cartridge enclosed in a tungsten alloy outer structure. On top of the outer tube is a layer of pure tungsten armor. Past design and analysis of the T-tube divertor concept has shown that it can accommodate a heat flux up to 10MW/m^2. With recent concerns that steady state or transient heat fluxes may be higher than this in the divertor region, the T-tube concept was modified so that it may accommodate higher heat fluxes.

1. INTRODUCTION

The T-tube is a He-cooled W divertor concept originally proposed by T. Ihli for the ARIES-CS power plant design study.[1] The T-tube was chosen for the design study because it would be able to accommodate a heat flux of 10 MW/m^2 in the divertor region with a relatively small number of poloidally arranged units (110,000) for a power plant.[2] These units will be cooled by helium at a pressure of 10 MPa with an inlet temperature of 600 °C. Originally analysis of the T-tube was done to determine if it would be able to accommodate a heat flux of 10 MW/m^2. Results indicated that this is attainable, although there is a very small safety margin on the temperature limit. Recent research activities are focusing on optimizing designs and determining how far each concept can be pushed. The T-tube configuration has gone through modifications such as tapering the inner steel cartridge to attain a uniform velocity and temperature distribution along the surface. By doing this the thermal stresses will be reduced. The size of the slot jets has also been optimized based on thermo-fluid and thermo-mechanical calculations to find an optimal balance between surface temperature and pumping power. The results from these modifications indicate that the T-tube divertor can handle a surface heat flux up to ~13 MW/m^2 without exceeding the temperature, pumping power, and stress limits.

This paper presents the process of analyzing and optimizing the performance of the T-tube divertor concept. This is done by testing several configurations of modified components to determine which will give the best performance. This measurement will be based on heat transfer, temperature, stress, and the power input required to operate the component. The detailed computational fluid dynamics (CFD), finite element analysis (FEA), thermo-fluid, and thermo mechanical analysis and results are also summarized in this paper.

II. GENERAL DESIGN REQUIREMENTS

The goal in making modifications to the T-tube design is to maximize its ability to withstand a high heat flux (q'' > ~13 MW/m^2) while keeping it within all other design constraints. In He-cooled W divertors there are three main design requirements that must be met. First is the temperature of the W-alloy outer tube structure shown in Fig. 1 (under the W armor). This limit is determined by the recrystallization temperature of the material. In the case of this W-alloy, the limit is ~1300 °C. Next is the combined thermal and primary stresses. This is determined by the ASME code using the 3Sm limit (~450 MPa at 1100 °C). The third limit is that the required power to pump the helium through the structure must be less than 10% of the removed thermal power from the structure. This limit is self-imposed to ensure that there is a net energy gain from the component.

III. T-TUBE DESIGN

III.A. Original Design Description

The total allowable heat flux limit for the T-tube is determined by the temperature of the outer W-alloy wall, the pumping power of the helium coolant, and the combined thermal and primary stresses. The coolant enters the system at 600 °C, which was believed to be the ductile-brittle transition temperature (DBTT) for W.[2] The

W armor is castellated into smaller tiles to prevent stress transferring from the armor to the main structure.

The original T-tube, shown in Fig. 1, was designed to withstand a 10MW/m^2 heat flux. The main structure is composed of two concentric W-alloy tubes. The outer tube is the main structural component while the inner cartridge contains the slot-jets for cooling. On top of the main structural tube is a layer of 5 mm thick pure W armor which has 5 mm x 5 mm castellation for stress relief purposes. The helium coolant enters through the center opening and flows axially towards the ends. There are six slot type impinging jets that the helium flows through as it exits the inner cartridge. There is a 1.2 mm gap between the slot-jets and the surface of the outer tube. After impacting the outer wall, the fluid flows radially around the outside of the cartridge where it then exits through the two outer openings into the steel manifold.

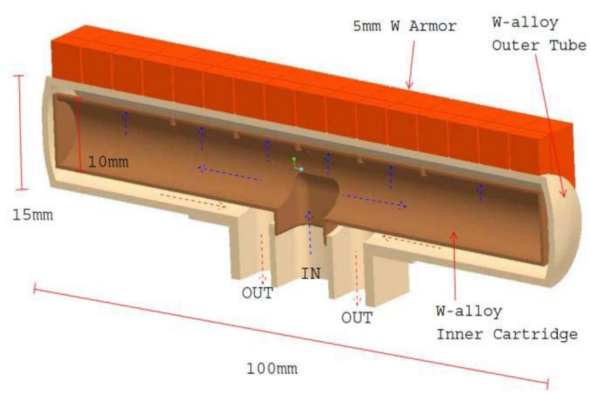

Fig. 1.Original T-tube design with non-tapered cartridge (half cross section).

The main design limits of the T-tube are the temperature of the outer structural tube (1300 °C), the combined primary and secondary stresses (450 MPa at 1100 °C), and the pumping power (10% of thermal power). With the original design subject to a 10 MW/m^2 heat flux the max temperature is 1283 °C, the equivalent stress is 284 MPa, and the pumping power is ~5.7% of the removed thermal power. Both the stress limit and the pumping power limit are far from being surpassed. The main problem is the temperature. At 1283 °C it is very close to the 1300 °C limit. In order to optimize this design, all three parameters should provide roughly the same margin away from their limit. For this initial design it can be said that the maximum allowable heat flux is 10 MW/m^2, but with a very small margin of safety. In many cases this would be more than sufficient, but in cases where the heat flux is either above this or below this, the performance of this design concept are still unknown.

III.B. Design Modifications

III.B.1 Inner Cartridge Modifications

The original design was sufficient for heat fluxes up to 10MW/m^2, but it can be optimized for pushing higher performance. For this concept to handle a larger heat flux than this, modifications must be made.

Recent work has determined that the DBTT of W is ~800 °C. Because of this the inner cartridge cannot be made from W with an inlet temperature of 600 °C. Thus the cartridge will be made from ODS steel which will connect directly to the steel manifold.

The next cartridge modification, shown in Fig. 2, was to make the inner cartridge tapered from an inner diameter of 10 mm in the center to 2.5 mm at the ends. The cartridge will be tapered at the bottom which will keep the distance from the slots to the wall constant. In the original design it was found that at the ends of the cartridge the fluid was beginning to lose most of its momentum and circulate at the bottom while the higher velocity fluid would flow above. By removing this excess area, not only will this circulation be removed but the velocity in this region will be increased. With this, a more uniform velocity will be seen as the fluid enters the slots, as seen in fig 3. This will create a more uniform temperature distribution along the outer wall. Although the average temperature of the wall may remain the same, the maximum overall temperature will decrease, which is the deciding factor in whether a design will survive or not.

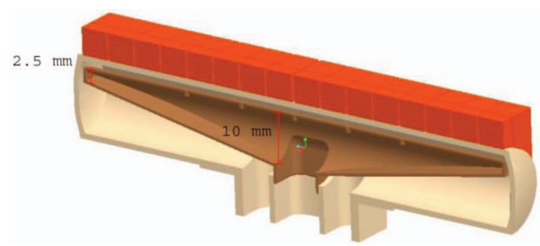

Fig. 2. Modified T-tube design with cartridge tapered from 10 to 2.5 mm. All other dimensions remain the same.

Another result from having a more uniform velocity profile is that in the slots the maximum velocity will be slightly reduced. This is another benefit to the tapered design. As the maximum velocity in the slot decreases, the pressure drop though the structure will decrease slightly. It is shown later that the pressure drop and the pumping power are proportional to one another, so with this decrease in pressure drop, a decrease in required pumping power for the same mass flow rate is seen. In physical terms, this means that more helium can be pumped through the cycle with the same energy input, leading to greater cycle efficiency.

III.B.2 Slot-Jet Modifications

In a further attempt to increase the heat transfer coefficient, the slot-jet was modified. The original design contained a slot with a uniform width of 0.5 mm. By decreasing the size of the slot, the heat transfer coefficient will increase due to the increase in velocity. This will cause the maximum temperature of the wall decrease. The down side to this modification is that it leads to higher pressure drop through the structure. As discussed in the above section, the pressure drop is directly related to the pumping power, and as the slot size is decreased the pumping power required will increase and possibly exceed the limit. Thus an optimal slot width must be determined. This will be done by using the data obtained from the thermo-fluid simulations to plot temperature vs. heat flux and pressure drop vs. heat flux. As the slot width is varied a point will can found where both the pumping power limit and temperature limit occur simultaneously.

III.B.3 W Armor Modifications

One design parameter that is unknown is the thickness required for the W armor. The armor is required because of erosion that occurs over time because of a strong flux of charged particles that escape from the plasma, but it is not known at what rate this will occur. In past designs 5 mm castellated armor has been used, but some believe that this may be overly conservative. In this analysis a 1 mm castellated armor was analyzed and compared to the previously used 5 mm armor. With this it will be determined how significant this parameter actually is in determining the maximum attainable heat flux.

IV. ANALYSIS

For this analysis all 3-D models were prepared using Pro ENGINEER Wildfire 5.0. All simulations were done using ANSYS 12.0. The thermal and mechanical simulations were done using ANSYS Workbench and the CFD analysis was performed using ANSYS CFX.

IV.A. CFD Thermo-fluid Simulations

For the CFD analysis there are two main parameters in which the performance of the design is based on. First is the maximum temperature on the outer structural tube. The other is the pressure drop, Dp, which relates directly to the pumping power, $P_{pumping}$, by Eq. (3). The CFD simulations were run with the standard k-ε turbulence model and scalable wall function. The k-ε turbulence model can be used with confidence when there are no large stagnation areas and the flow remains sub-sonic. The scalable wall function is a very good approximation for simulations where the near wall velocity is not near zero.[5]The criteria for both of these models are met so they can be used with confidence. Experimental analysis has also been performed to validate the accuracy of these models.[3]

In Eq (1), q'' is the surface heat flux and q''' is the volumetric heat generation. In Eq (2), ṁ is the mass flow rate of the helium and in Eq(3), Δp is the pressure drop.

$$Q_{tot} = q'' \cdot A_{Plasma\ Surface} + q''' \cdot V_{total} \quad (1)$$

$$\dot{m} = \frac{Q_{tot}}{C_p \cdot \Delta T} \quad (2)$$

$$P_{Pumping} = \Delta p \cdot \frac{\dot{m}}{\rho} \quad (3)$$

The parameters used in these simulations are listed in Table I. The volumetric heat generation, inlet temperature, outlet temperature, and He pressure are all assumed to be constant. As the heat flux, q'', is changed, the mass flow rate also changes based on the mass balance in Eq. (1) and Eq. (2). The slot width was varied while keeping the surface heat flux constant to find the optimal value.

TABLE I. Design Parameters

Volumetric Heating (MW/m^3)	17.5
Inlet Temperature (°C)	600
Outlet Temperature (°C)	677
Slot Diameter (mm)	0.25, 0.4 ,0.45, 0.5
He Pressure (MPa)	10
Heat Flux (MW/m^2)	8, 10, 11, 12, 15
Inlet Mass Flow Rate (g/s)	6.7, 8.4, 9.2, 10, 12.4

IV.B. Cartridge Modification Analysis

The cartridge was modified in order to create a more uniform velocity distribution throughout the cartridge. Fig. 3 shows the velocity vectors through the entire structure. The top is the original design and the bottom is the tapered design. In the tapered design the maximum velocity is 210 m/s, which is slightly less than the maximum velocity from the non-tapered deign. But the velocity at the end has increased from 130 m/s to 160 m/s. This more uniform velocity will create a more even heat transfer coefficient along the inside face of the structure wall resulting in a more uniform temperature distribution .

Another advantage of the tapered design is a reduced pumping power requirement. With a limit of 10% of the removed thermal power, the maximum allowable pressure drop was calculated to be 0.21 MPa from Eq. (3). As the maximum velocity decreases, the pressure drop through the slot decreases. This lower velocity allows for an increase in mass flow rate through the system for the same pumping power.

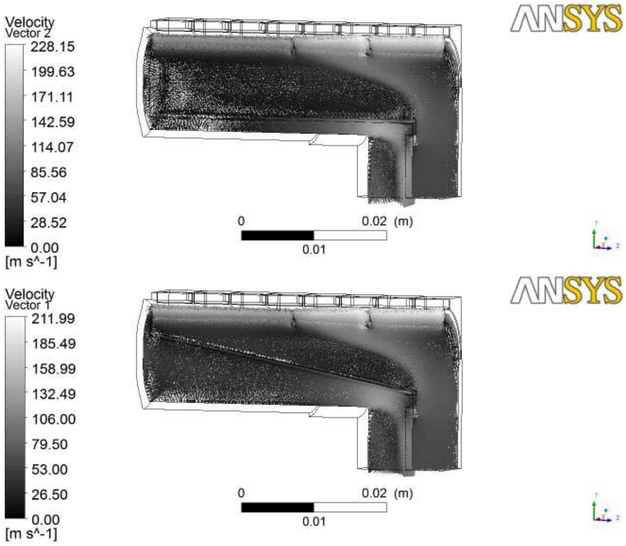

Fig. 3 Velocity vector comparison of the tapered and non-tapered cartridge with all other variables kept constant.

IV.C. Thermo-mechanical Simulations

For performing the thermo-mechanical simulations a simplified model consisting of one fourth of the original geometry was used. This may be done because the model is symmetric about both the XY plane and the YZ plane. This simplification greatly reduces the computing power required to run the simulations. The same mesh was used from the CFX analysis with the fluid geometry removed. To keep consistent from the CFX analysis, the temperature from the He/W-alloy interface was imported into the thermo-mechanical simulation. The boundary conditions placed on the finite element model were as followed: no displacement at the bottom inlet/outlet channel, symmetry about the XY plane, and symmetry about the YZ plane.

TABLE II. Sample Design Results

q''(MW/m^2)	Temp(°C)	Dp(MPa)	Stress (MPa)
8	1234	0.081	281
10	1293	0.124	284
11	1325	0.147	286
12	1356	0.173	289
13	1382	0.205	293
15	1424	0.278	304

To calculate the heat flux at which each limit is reached several different values for surface heat flux were analyzed. Table II shows a sample of the data obtained from the CFD and structural simulations for the original non-tapered design with 5 mm armor and a 0.5 mm slot width. With these sets of data, curves can be made for temperature, pressure drop, and equivalent stress vs. surface heat flux. These curves, shown in fig. 4, fig. 5, and fig. 6, are used to determine the heat flux at which each limit will be surpassed.

Fig. 4 shows where the pumping power limits are reached for each design. This shows that both the tapered cartridge and slot modifications affect the pressure drop while the armor thickness does not. Fig. 5 shows the temperature vs. surface heat flux curves for five design configurations. It is seen that recrystallization temperature is extremely important. A slight change in recrystallization temperature can greatly increase or decrease the maximum attainable heat flux. Fig 6 shows the total stress vs. surface heat flux. The stress is far from the 3Sm limit so further modification need be made to take advantage of this.

TABLE III.

Design Name	*Symbol*
Non-tapered, 5mm Armor, 0.5 mm Slot	●
Non-tapered , 1mm Armor, 0.5 mm Slot	■
Tapered, 5mm Armor, 0.5 mm Slot	♦
Tapered, 1mm Armor, 0.5 mm Slot	X
Tapered, 1mm Armor, 0.45 mm Slot	▲

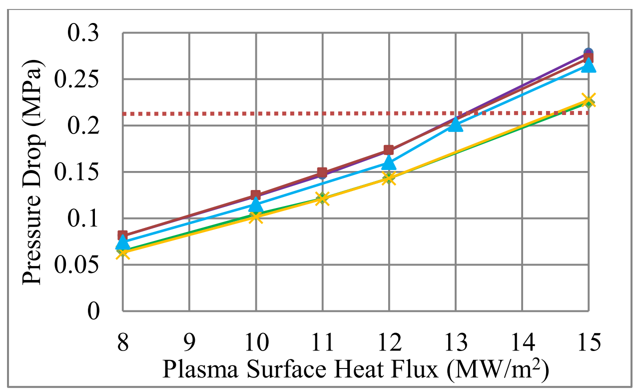

Fig. 4. Pressure drop vs. surface heat flux. Legend provided in Table III.

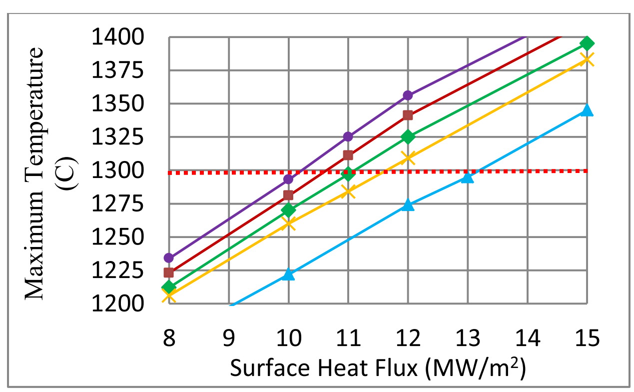

Fig. 5. Maximum temperature of WL10 outer tube vs. surface heat flux. Legend provided in Table III.

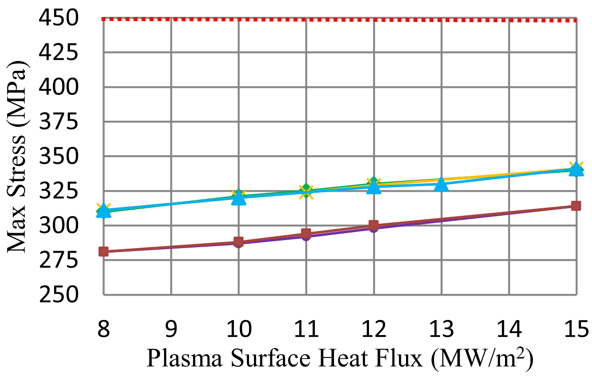

Fig. 6. Equivalent von-misses stress of WL10 outer tube vs. surface heat flux. Legend provided in Table III.

Table IV shows a comparison of the five design configurations and the maximum heat flux that each configuration will accommodate. With the optimized slot, cartridge, and armor in design 5, both the temperature and pumping power limits are reached at the same point so this is the optimal design.

TABLE IV. Design Comparison

1) Non-Tapered, 5mm Armor, 0.5 mm Slot					
2) Non-Tapered, 1mm Armor, 0.5 mm Slot					
3) Tapered, 5mm Armor, 0.5 mm Slot					
4) Tapered, 1mm Armor, 0.5 mm Slot					
5) Tapered, 1mm Armor with 0.45 mm Slot					
Design Number	1	2	3	4	5
Temp Limit (1300°C)	10.25	10.6	11.1	11.7	13.25
Stress Limit (450 MPa)	NA	NA	NA	NA	NA
Pumping Power Limit	13.1	13.1	14.3	14.3	13.25
Max Heat Flux	10.25	10.6	11.1	11.7	13.25

IV.C. 1 mm VS 5 mm Armor

One parameter that is not known is the thickness of the W armor. Throughout the power plant there will be some degree of erosion of the wall from the plasma, but the rate at which this will occur is not known. A comparison analysis was performed using 5 mm armor and 1 mm armor. By reducing the amount of armor for both the tapered and non-tapered designs the maximum heat flux attainable is increased by ~3-5%.

IV.D. Neutron Irradiation Effects

As metals become irradiated generally there thermal conductivity will be decreased. Over the life cycle of a W divertor component this decrease will be ~5-10%. With a loss of 10% in thermal conductivity, the maximum heat flux the T-tube can accommodate will be decreased by ~6%.

V. CONCLUSION

Detailed analysis and optimization of the T-tube concept has shown that it can perform significantly better than previous results had shown. It has been shown that this concept can remain under the temperature, stress, and pumping power limits while being subjected to a ~13 MW/m^2 heat flux as opposed to the previously believed 10 MW/m^2. Although the minimum required thickness for the W armor is not known, 5 mm is believed to be conservative, and it only comes at a loss of ~3-5% in maximum attainable heat flux. The maximum heat flux is also affected by irradiation embrittlement. With a decrease in thermal conductivity of 10% comes a loss in attainable heat flux of ~6%. These are both relatively small losses. The most important factor is the recrystallization temperature. If a W-alloy can be developed with a higher recrystallization temperature the maximum attainable heat flux will be greatly increased. Further analysis and design must be performed to address the issue of possible stress concentrations in the transition zone from tungsten to steel.

ACKNOWLEDGMENTS

The work at UCSD was supported under U.S. Department of Energy Grant number DE-FG02-04ER54757.

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