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UCSD-CER-11-01

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8 August 2011

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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 uniform 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. It is also known that the divertor region will have a spatial heat flux distribution. The effect of this non uniform heat flux on the T-tube is currently not known. Further analysis can be used to determine if the T-tube is a feasible design concept for future use.

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.

Detailed analysis was performed with a uniform heat flux by testing several configurations of modified components to determine which will give the best performance. This measurement was based on heat transfer, temperature, stress, and the power input required to operate the component. This optimized design was then used to perform analysis with a spatially varying heat flux. The detailed computational fluid dynamics (CFD) and finite element analysis (FEA) results for a spatial varying heat flux are summarized in this paper.

DESIGN MODIFICATIONS

During the course of performing design and analysis of the T-tube divertor concept several further design modifications were considered. The first of these modifications, which is necessary to operate in current conditions, is to have the inner cartridge made from ODS steel rather than tungsten. This is necessary because the inlet temperature is only 600 °C, which is below the DBTT (Ductile Brittle Transition Temperature) of tungsten. This modification will require the outer wall to be modified as shown in figure 1. This outer wall will remain tungsten. The inner cartridge will no longer attach to the wall but rather extend down to the steel manifold. This modification will not affect the performance since the geometry in the heat transfer region will remain unchanged.

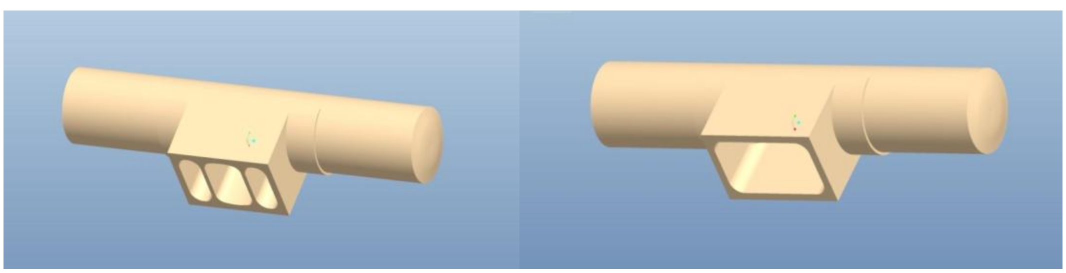

Figure 1 T-tube outer wall modifications

The next two potential design modifications would be to try and create a more uniform temperature distribution along the outer surface. To do this a more even flow velocity through the slots is needed. The first idea was to insert a flow divertor into the cartridge. This would tend to push more fluid toward the ends of the tube. Since currently the highest velocity is in the middle this is an attempt to rectify this. The next idea was to use a non-linear taper in the cartridge. Currently a linear taper is used for simplicity but does not create an optimized flow. When looking at the velocity vectors of the flow through the cartridge, we can see a curved profile where the fluid begins to lose momentum, shown in figure 2.

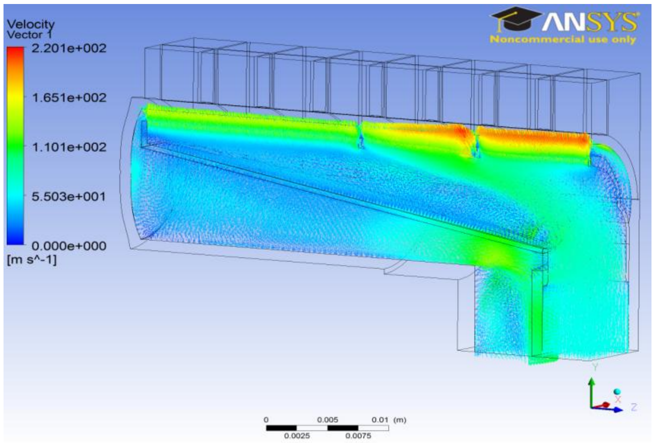

Figure 2 Flow velocity vectors through cartridge

One down side to this design is that the cartridge would be much more difficult to fabricate. These designs were never tested because efforts shifted from looking at a uniform heat flux to looking at a spatially distributed heat flux. When looking at this spatially varying heat flux the majority of the heat will need to be removed from the middle of the structure, so these modifications will not improve performance in this case.

PUMPING POWER CURVES

One factor that is very important is how the pumping power increases as the surface heat flux increases. To try and visualize this a set of curves were created showing how these values change. In the T-tube we wanted to see how changing the inlet and outlet temperatures will effect the required pumping power. It was seen, in figure 3, that if the delta T is kept constant there is only a very small change is pumping power, but if this is changed then there is a drastic change in pumping power.

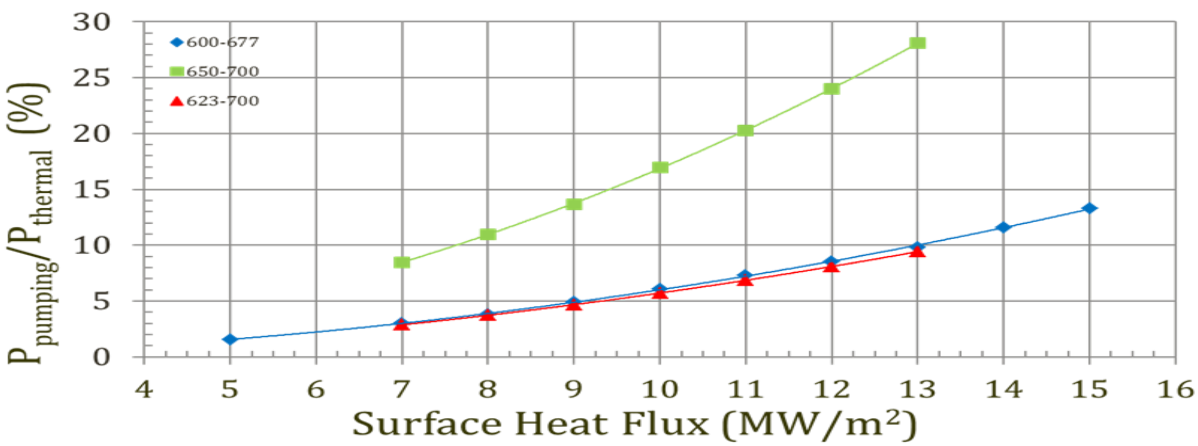

Figure 3 T-Tube Pumping Power Curves

If we look at the corresponding structure maximum temperature we see the same trend emerge. In the case where the delta T had been decreased to 50 °C we see that the temperature is much lower. In this case we do see that the temperature is not only dependent on delta T but also the actual inlet and outlet temperatures, which should be expected.

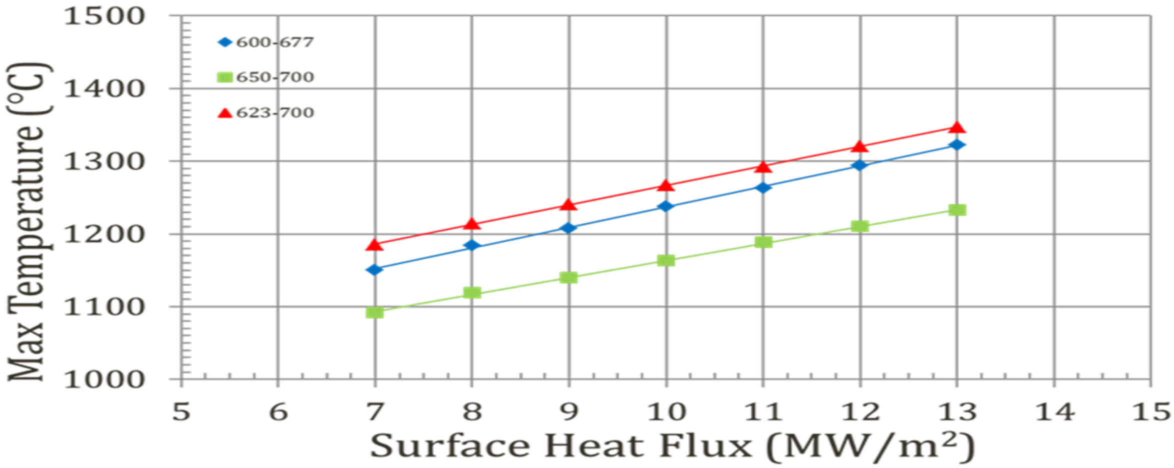

Figure 4 Maximum T-tube surface temperature

In these three cases the slot size was kept constant throughout and the temperature was allowed to increase above the 1300 °C limit. Another case of interest would be to decrease the slot size as the temperature reaches 1300 °C. This is so that the pumping power required to operate this design can be seen at all points. In this case we wanted to keep the tungsten between 800 -1300 °C at all times. In figure 5 we can see that as the heat flux passes 9 MW/m^2 the pumping power begins to increase dramatically. This is because the surface temperature has reached 1300 °C so the slot width is being decreased.

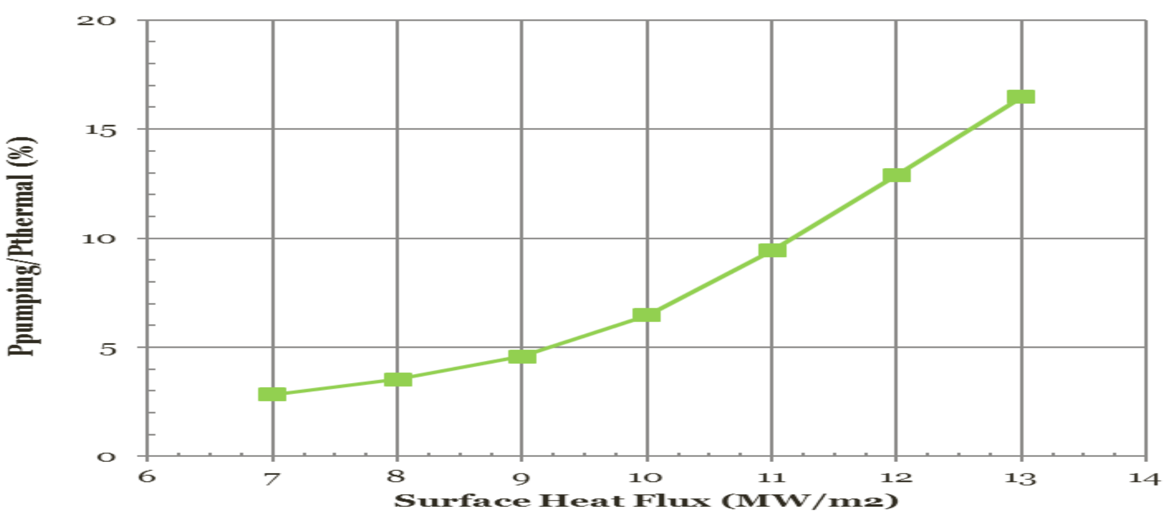

Figure 5 T-tube pumping power with varying slot width

The maximum surface temperature can be seen in figure 6. We see that at 9 MW/m^2 the temperature just passes 1300 °C so the next case was run with a slot size reduced by 50 microns and this continues until 13 MW/m^2.

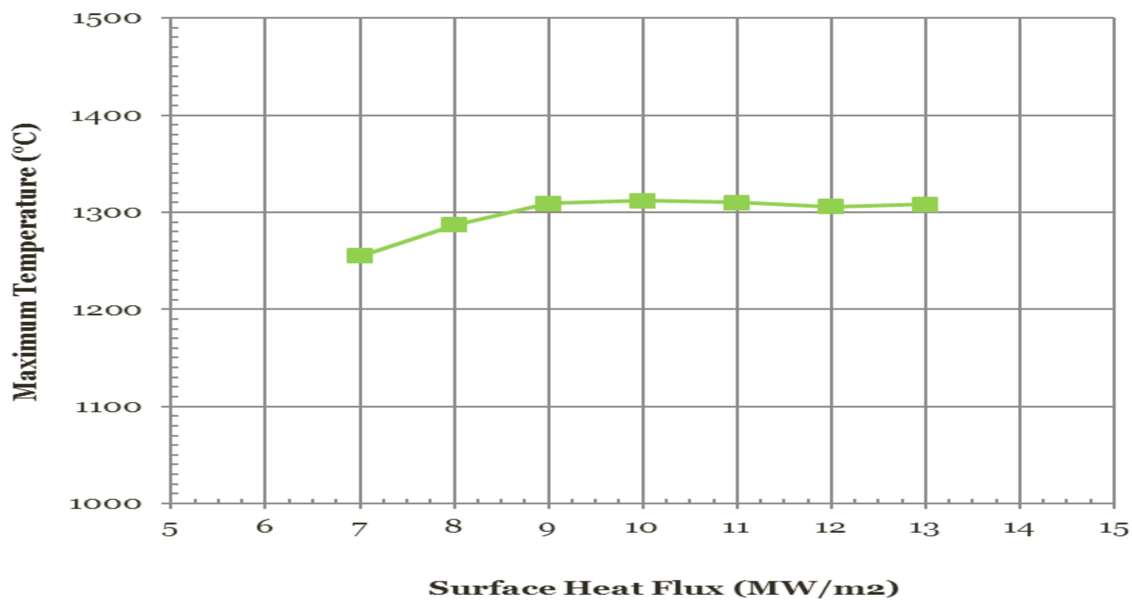

Figure 6 T-tube maximum surface temperature with varying slot width

Table 1 shows how the slot size changes with increasing surface heat flux. These slot size changes corespond to the large slope changes in the graph of pumping power vs surface heat flux. For this particular design we see that the heat flux limit is reached at just over 11 MW/m^2 because this is where the pumping power surpasses the 10% limit. From here on the pumping power is increading very rapidly and by even at just 13 MW/m^2 the pumping power is already at ~17%.

Table 1 Slot Size Variation

Surface Heat Flux (MW/m^2)	Slot Size (Microns)
7	450
8	450
9	450
10	400
11	350
12	300
13	250

SPATIAL HEAT FLUX VARIATIONS

It is known that the heat flux in the divertor region will not be a constant uniform value. Because of this it is important to analyze a case with a spatialy varying heat flux to see the differences. The most current heat flux distribution is shown in figure 7. Here we see that the predicted maximum heat flux value is ~11 MW/m^2 and this value is only seen in a very small region. As you move in either direction this value decreases drasticly. For a simplified analysis still only one quarter of the model will be used. For this 11 MW/m^2 was used in the center and it was decreased at each tile. The profile used for the analysis is shown in figure 8. In this figure each drop is moving to the next tile on the tungsten armor. And overall this coresponds to an average heat flux of ~8 MW/m^2

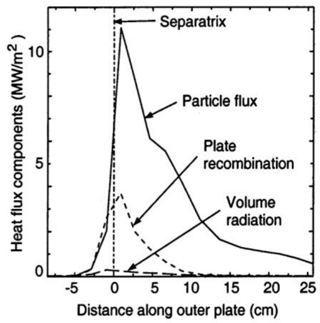

Figure 7 Spatial heat flux distribution in divertor region

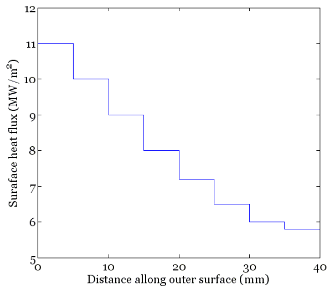

Figure 8 Simplified heat flux distribution in divertor

In this analysis we first wanted to see the effect of changing friom a uniform profile to this now varying heat flux. In order to change the average heat flux, the value on each tile was either increased or decreased by 1 MW/m^2. Since this is the only current estimate, this effectivly shifts the entire profile up or down by 1 MW/m^2 at at time. Figure 9 shows the comparison the pumping power curves and figure 10 shows the coresponding maximum temperatures of the two. In the uniform case we have the tapered design with a 450 micron slot and 1 mm armor. We see that here it is reaching the pumping power limit at just over 13 MW/m^2. With the non uniform heat flux case we start with the same design but we see that at 8 MW.m^2 it has already reached the 1300 °C temperature limit. Because of this the slot size must be decreased. This is why we see the jumps in pumping power required. Each jump coresponds to a decrease in slot size. Table 2 shows the change in slot size as heat flux is increasing.

Table 2 slot size variations for non-uniform heat flux

Average Surface Heat Flux (MW/m^2)	Slot Size (Microns)
7	450
8	450
9	400
10	400
11	350
12	300
13	250

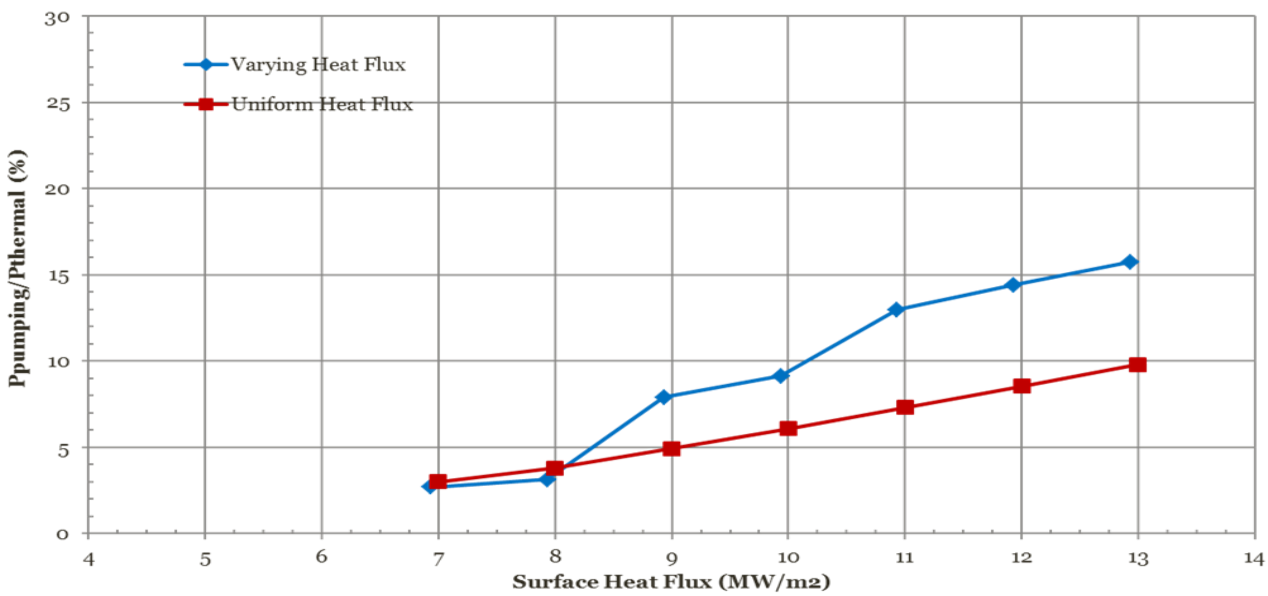

Figure 9 Constant vs. varying surface heat flux pumping power curves

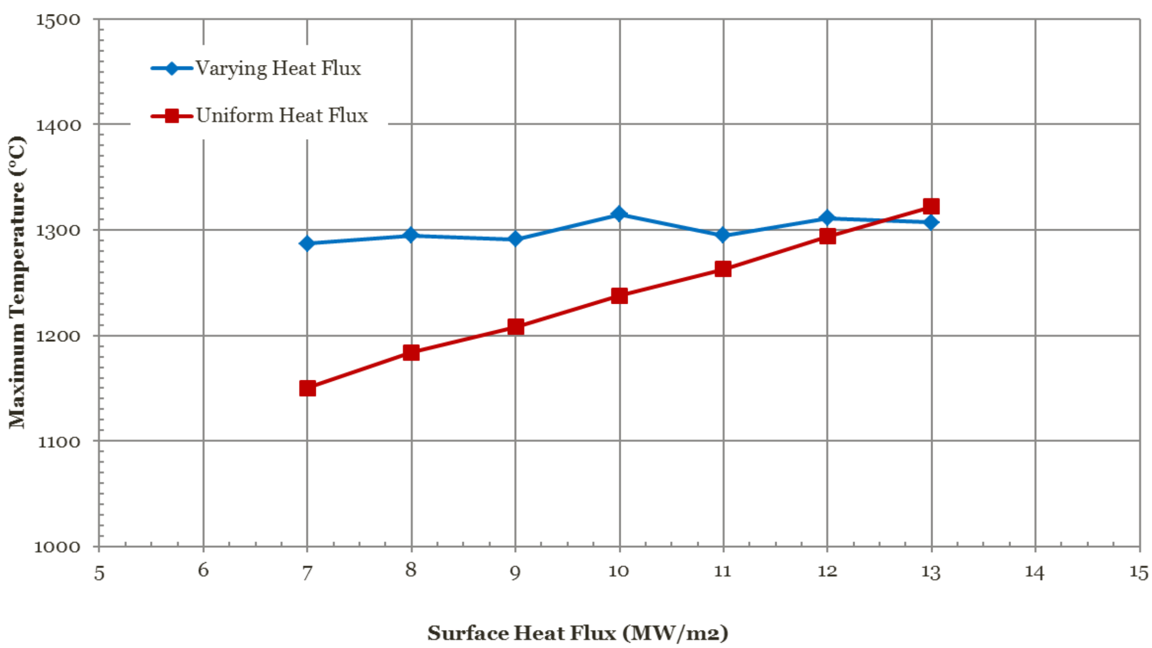

Figure 10 Constant vs. varying surface heat flux temperature curves

Here we see that the tapered design is only optimized for a uniform heat flux. For this non uniform profile, the majority of the heat that needs to be removed is at the center. With this we see that with the non uniform heat flux it can take up to 50% more pumping power to remove the same amount of thermal power from the structure. Since the tapered design is optimized to try and remove the same amount of heat from all parts of the structure, it is not very efficient with a non uniform heat flux. Because of this the original non tapered design may actualy perform better than the tapered design.

ORIGINAL DESIGN VS. TAPERED DESIGN

With the results of how the tapered design performed with a non uniform heat flux it can be seen that this is not the optimal design. Based on previous analysis, it is believed that the original design will perform better than the tapered design with a non-uniform heat flux. To determine this the same pumping power curves were generated for the original non-tapered design and compared to those previously genereated for the tapered design. Again multiple slot sizes had to be used to maintain the surface temperature at ~1300 °C. these values are shown in table 3.

Table 3 Slot size variation for non-tapered design

Average Surface Heat Flux (MW/m^2)	Slot Size (Microns)
7	500
8	500
9	450
10	450
11	400
12	300
13	300

The results of this analysis are shown in figures 11 and 12. Figure 11 shows the comparison of pumping power curves and figure 12 shows a comparison of maximum surface temperature. In this comparison we see that the original non-tapered design can start at a slot size of 500 microns, where the tapered design needed to start at 450 microns. This is because in the non-tapered design there was a much larger velocity in the center than with the tapered design. This was the goal of the taper and because of this it is not as efficient with this non uniform heat flux. Althoug the non-tapered design has a benefit in pressure drop from using a larger slot size, there are other areas where it is not as efficient. When the original analysis was performed a large area of stagnant coolant was observed near the ends of the tube. This loss of momentum created an increase in the pumping power. This was seen in the original comparison of the tapered vs. non-taped design with a drop in pumping power for the same removed thermal power. This still exists in the non-uniform surface heat flux case. This two factors seem to cancel each other out and we see in figure 11 that both designs have roughly the same pumping power requirement for each value for surface heat flux. There are some slight differences due to the slot size changing at different points on the curve, but overall the two designs performed almost identically.

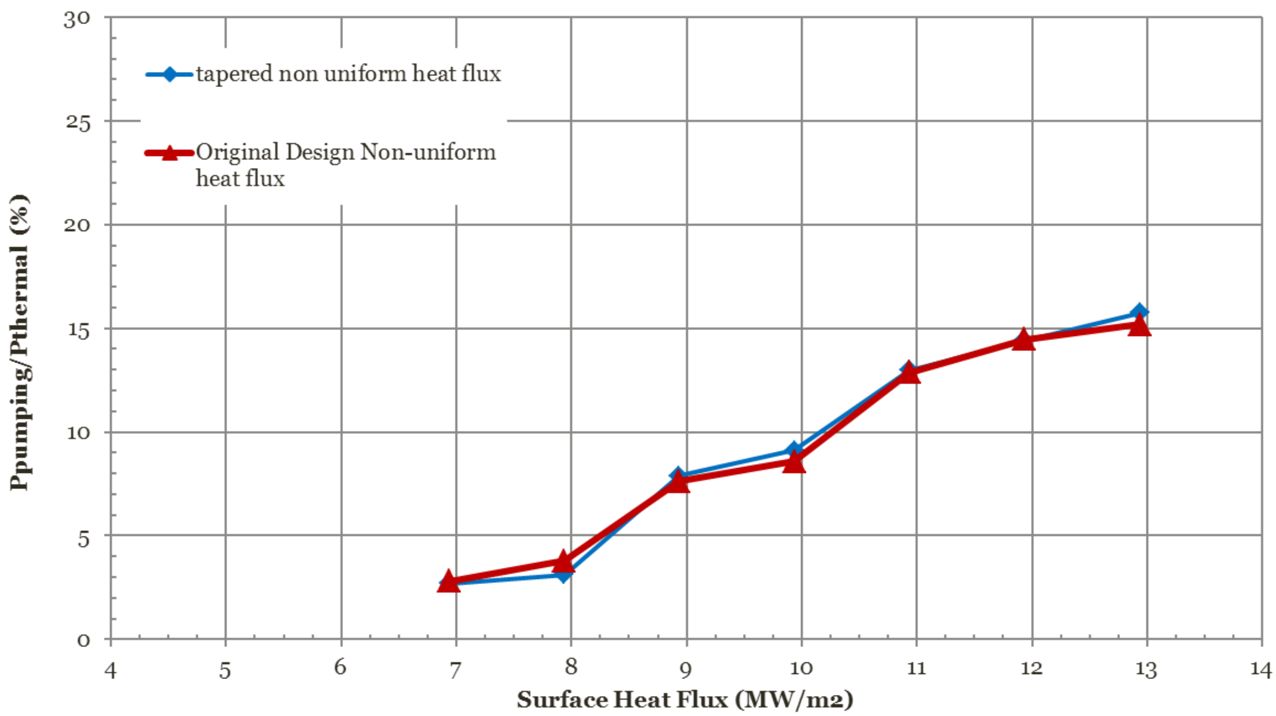

Figure 1 Tapered vs. non tapered with non-uniform heat flux pumping power curves

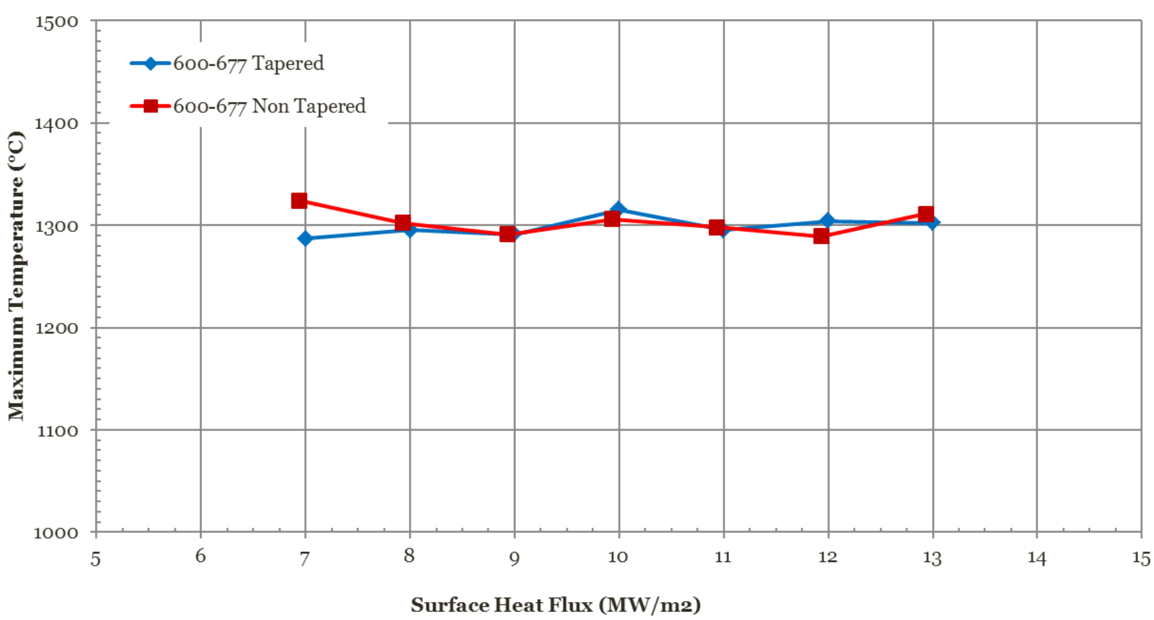

Figure 2 Tapered vs. non tapered with non-uniform heat flux temperature curves

T-TUBE ORIENTATION

Originally the T-tubes were arranged in the toroidal direction. With this orientation the spatial heat flux variations are large over the length of the structure. One way to remedy this could be to orient the T-tubes in the poloidal direction. This would create an almost constant heat flux on each T-tube. One simple case was analyzed to see the effect of this orientation change. For this case the center rows of armor tiles will have 11 MW/m^2 and the two outer rows will have 10 MW/m^2. With this profile the average heat flux would be 10.333 MW/m^2, which is used to calculate the mass flow rate. It was determined that in this case the maximum temperature on the surface is 1244 C, which is slightly better than the case with a uniform 10 MW/m^2 heat flux. This would need to be verified with further analysis but the initial results are promising.

SUMMARY

With this analysis it can be seen that the T-tube, as currently designed, performs much better with a constant surface heat flux. Although if the spatial heat flux distribution in figure 7 is correct, the T-tube will be able to withstand this with a pumping power of ~5%. If the location of this large heat flux spike is known then the T-tube could be a very attractive design consideration. A set of specialized high performance T-tubes could be used in this region and then when the heat flux levels out further away the tapered tubes could be used. Arranging the T-tubes in the poloidal direction greatly increases the performance with a spatially carrying heat flux. Further design modifications could be performed to optimize the T-tube for this spatially varying heat flux.

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2. A.R. RAFFRAY, S. MALANG, and X. WANG, "Optimizing the overall configuration of a He-cooled W-alloy divertor for a power plant," Fusion Engineering and Design **84** (2009).