Numerical Simulation Analysis for the Divertor Plate of DEMO Reactor

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(Received: 22 October 2009 / Accepted: 26 January 2010)

Abstract: The hydraulic simulation and structure coupled analysis were used to explore critical heat flux (CHF) margin of the W (tungsten) armour and the accommodation of the thermal stress between the cooling tube and the W armour. The △t (wall thickness of the F82H tube) variation or as a constant were considered to analysis the heat removal capability of the cooling tube. It was found a nonlinear distribution of the peak T (temperature) for the W armour and the top cooling tube with the tube bore rising. And an allowable heat flux with 5 MW/m^2 or under the value would be acceptable based on present engineering consideration (△t=1mm, d=10mm and the coolant velocity=10m/s). The structure coupled analysis (d=10mm) indicated the primary stress of the cooling tube was safety, less than 50MPa, but thermal stress would be closed to 3Sm (F82H,550℃) due to the thermal expansion between W armour and F82H tube. Based on the coupled results, a thinner tube would be better than a thicker one by considering thermal conducting and thermal stress. Finally, for the issues on CHF and thermal stress, the possible optimizations were discussed involved the material choice, the structure of the cooling channel and the water conditions.

Keywords: divertor plate, numerical simulation, hydraulic analysis, coupled analysis, DEMO.

1. Introduction

Japan DEMO-SlimCS, a core dimensions are similar to that of ITER and the power generation capability would reach the gigawatt level [1]. The allowable heat flux of the divertor is limited to about 10MW/m^2 [2]. The critical heat flux (CHF) issue of the divertor armour is a key requirement influencing the choice of its cooling channel geometry and coolant conditions [3].

The present concept of the divertor plate is based on tungsten (W) mono-block (armour) and ferritic steel (F82H) cooling pipes [4-5]. A F82H substrate would be in the bottom, and two F82H cooling tubes would be installed both in the W armour and the substrate. The water condition is 15MPa, 290℃. Particularly, the distance from top surface of armour to the outer surface of F82H tube is fixed (h=5mm), as shown in Fig.1. The key concern focused on the CHF issues and the structure intensity of the cooling tube inside the armour. CFD (computer fluid dynamical) method was used for hydraulic analysis based on Fluent code, and the finite element method was used to detail the structure analysis by using ANSYS code.

2. Hydraulic simulation

2.1 CHF based on tube bore

2.1.1 △ t consideration

The △t is related to the structure intensity of the cooling tube, so the two cases were taken into account, △t_1=variation; △t_2=constant. The △t_1 varied along with tube pore rising proportionally. The △t_2 was fixed with 1mm.

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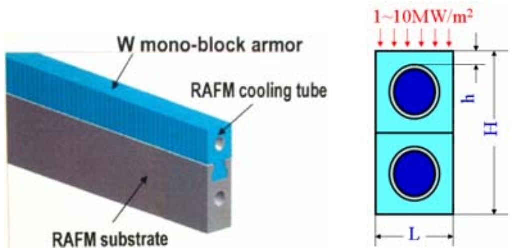

Fig.1 Concept of divertor plate

For △t_1 case, the relationship between the water pressure and the yield strength are clarified as followings, which could result in a series of △t reference values. Here,

D ---outer diameter of cooling tube, mm
d ---pipe bore, mm
K---coefficient of the matching condition
P_f--- water pressure on inner surface of cooling tube, MPa
P --- safety value of water pressure, MPa

For F82H material, the maximum values of S_Y (yield stress), S_u (ultimate tensile stress) at 550℃ are as followings: S_Y=354MPa, S_u=378MPa.

In terms of this, the coefficient K of formulas and the relevant values are as followings:

a) Mises condition $P_f = \frac{2}{\sqrt{3}} S_Y \ln K$ *(1)*

b) Tresca condition $P_f = S_Y \ln K$ *(2)*

c) ASME code Sec.VIII div2(for high pressure, P>0. 4S)

$$P = 1.5P_f,\ P = S\ln K,\ S = \min\left\{\frac{S_Y}{1.5}, \frac{S_u}{3}\right\} \quad (3)$$

(Here, 0.4S=50.4MPa)

d) Diameter relationship $D = K \cdot d$ *(4)*

Therefore, for the above equations, (1), (2) and (3), the K=1.073, 1.043 and 1.126 respectively. Considering the water pressure is only 15MPa, and for easy comparison, the K was chose as 1.2 finally, namely, △t=0.1d. So, when d=8, 9, 10, 11, 12, 13mm, The D=9.6, 10.8, 12, 13.2, 14.4, 15.6 mm respectively. For second case, $\triangle t_2$=constant, namely, △t=1mm was fixed for each changed d value, and the D=10, 11, 12, 13, 14, 15 mm correspondingly.

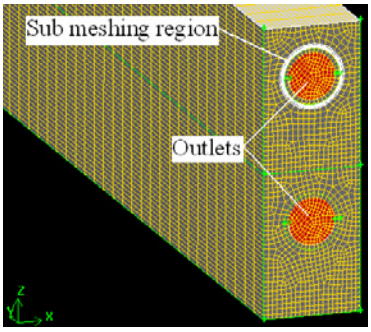

Fig.2 Model meshing (d=10mm, △t=1mm)

2.1.2 CFD model introduction

For the CFD models, according to the design concept of divertor plate, a 1000mm long model with a cross-section L×H=21×62 mm was used. Between the outer surface of the top cooling tube and the top surface of the armour, sub-meshing is considered on the interface closed to W side. Because the high heat flux from the top amour side to the top cooling tube would result in a big temperature gradient, sub-meshing could detail the temperature changes. The interfaces between the solid zones and the coolants were set for fluid-solid coupled analysis. The element hex/wedge were chose for irregular and regular zones respectively, meshing with cooper method was for saving the mesh number. For the two kind of △t cases, totally 12 CFD models were obtained. CFD model meshing example is as shown in Fig.2 (case d=10, △t=1: 257347 elements and 273000 nodes).

The inlet parameters are as followings: initial temperature, 290℃, inlet velocity was 10m/s. The average neutron heat flux was 10MW/m^2 on the top surface of amour. The water properties like the density and thermal conductivity were as a function of temperature and pressure.

In the calculation, the 3ddp (three-dimensional double precision) calculator was chose for the slenderness ratio model. The standard k-ε model was used for the turbulence calculation with a high Reynolds number (Re≥8.07×10^5) model.

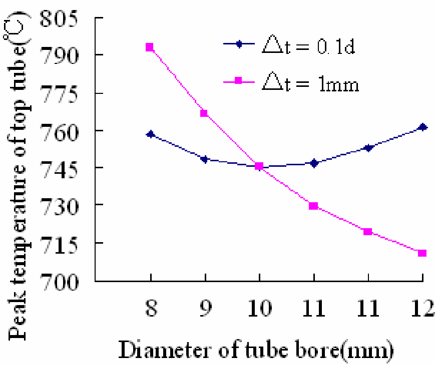

Fig.3 Comparison with two kind of △t for peak T of top tube

2.1.3 Analysis results

For the calculation results, when the d varied along with △t proportionally, the peak T of W ranged 1184~1205℃, the peak T of top tube was ranged 746~761℃. But the temperature trends were not as the linear distribution, it acted as a quadratic function form, the bottom point was the case d=10, △t=1. From Fig.3, when the △t was under the value 1mm, the peak T was as a drop trend distribution (negative exponential) and on the contrary, the trend would be changed to positive exponential distribution. It could be deduced if the △t< 1mm, the peak T distribution would be depended on the d value, and if the △t>1mm, the wall thickness would be the dominated factor, namely, when it is increased, the heat removal capability of the cooling tube would be hindered although the tube bore d was increased. However, for the △t=constant, only the d increasing, both the amour and the top tube would act as a negative exponential distribution on peak T: the range 1128~1233℃for W, and the range 711~793℃ for the top tube. According to the temperature limit of W amour (1200℃) and the F82H (550℃), it was almost no big problem about the W armour (~1200℃ at 10MW/m^2), but the top cooling tube could not accommodate the high CHF, its peak T>700℃ at a CHF=10MW/m^2.

2.2 CHF based on heat flux

2.2.1 Model introduction

Based on the above analysis, a d=10mm, △t =1mm would be preferred choice for CHF margin calculation comparing to the others due to the engineering consideration. Therefore, this case would be considered by applying the heat flux 1~10MW/m^2 to explore the possible CHF value on the top surface of armour. The CFD model setting method was the same as the 2.1.2 involved the relevant boundary conditions.

2.2.2 Analysis results

It was found there was a linear distribution on the peak T for both W and F82H (top tube) along with the heat flux increasing. When the q=5MW/m^2, peak T of W was 743℃, and the Peak T of top tube was 517℃, the corresponding case for q=10 MW/m2, 1169℃ and 746℃ respectively. And it showed the peak T of W amour meets 1200℃ requirement although the q was up to 10MW/m^2.

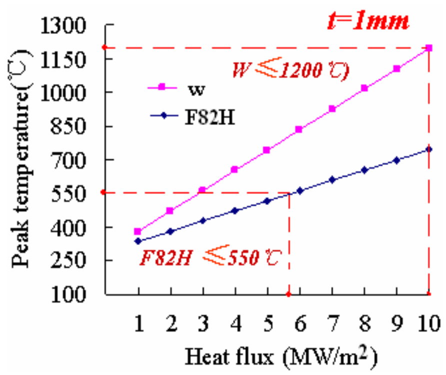

Fig.4 A possible CHF exploration (d=10mm, △t=1mm)

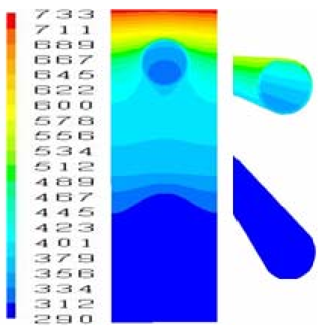

Fig.5 Temperature distribution (5MW/m^2)

Obviously, the problem existed with the top tube, when the q≥ 6MW/m^2, the peak T of F82H would exceed 550℃. Present result indicated a q=5.5 MW/m^2 would be feasible in terms of the temperature consideration, as shown in Fig.4. But based on engineering consideration, q=5MW/㎡ would be acceptable. Fig.5 shows the temperature field distribution.

3. Structure coupled analysis

Structure simulation focused on the stress distribution (primary, secondary) and the material deformation which could clarify the structure intensity resulting in a reasonable design. The top cooling tube was stressed on. Presently, it was considered with the d=10mm, △t=0.5~1.2mm for comparison. A q=5MW/m^2 was applied.

3.1 FEM model

2D model could be used to detail the mechanical analysis. A 2D thermal element plane 55, four nodes with a single degree of freedom, temperature, at each node was chose to perform the thermal analysis firstly, and by the element switch (thermal to structure), structure element plane182 was used to calculate the primary stress and the thermal stress. Meshing was by quad/free method, and the accuracy grade was chose as 1.

3.2 Simulation results

The structure coupled analysis indicated the primary stress of the cooling tube was under 50MPa, ranged 23~33MPa. However, the secondary stress-thermal stress would be increased along with the △t rising, and when the △t=1mm, the thermal stress would be 373MPa, and closed to the 3Sm=378MPa (F82H, 550℃), as shown in Fig.6-Fig.7. The elongation of the cooling tube in Y direction contribute most of the thermal stress rising. Analysis results indicated the primary stress was safety, according to the Sm of F82H (Sm=126MPa, 550℃) and for the secondary stress, a thinner tube would assume less value comparing to a thicker one. In addition, when q＞5MW/m^2, along with the temperature gradient rising, the thermal stress distribution would be more than that of q=5 MW/m^2 due to the bigger thermal expansion.

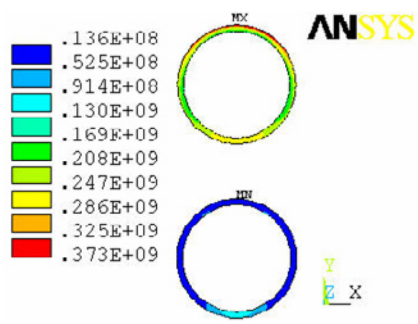

Fig.6 Thermal stress (d=10mm, △t =1㎜)

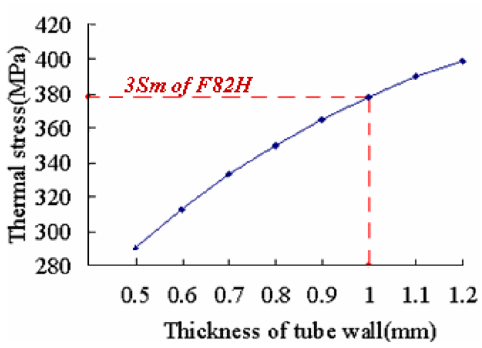

Fig.7 Thermal stress with the △t increasing (q=5MW/m^2)

3.3 Primary stress comparison

For pressurized water condition, the primary stress is concerned with the cooling tube especially due to its importance to structure safety. A F82H tube with △t=1mm was considered about the primary stress under a inside pressure value=15MPa. Formula (5) is a stress equation for free tube, r_1-inner radius of cooling tube, p_1-water pressure on the inner surface of cooling tube. Based on this, $\sigma_{\Delta t}$ =150MPa when it is no support outside. However, for divertor plate, there is a W armour outside and support the cooling tube, the elongation of the cooling tube would be limited and resulting in a lower stress value. So the simulation results on the primary stress assumed a 23-33MPa should be reasonable and credible. Fig.8 shows the primary stress distribution of d=10mm, △t=1mm and with a W support.

$$\sigma_{\Delta t} = \frac{r_1}{\Delta t} \times p_1 \qquad (5)$$

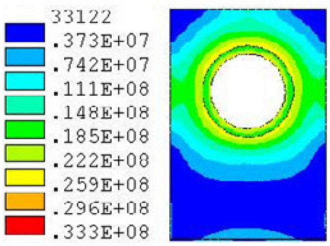

Fig.8 Primary stress with support case (d=10, △t=1mm)

4. Discussion

4.1 CHF issues

According to the CHF laws based on △t cases, it could not be applied a q=10MW/m^2 by increasing the cooling tube bore. A CHF=5MW/m^2 would be acceptable for the W-F82H structure based on the present engineering consideration.

The CHF is related to the thermal conductivity (W/m-K), the cooling tube bore (mm), the △t, the water temperature and the inlet velocity (m/s). For the engineering consideration, v=10m/s is a limit, so the velocity of water could not be increased once more. Therefore, if the CHF need to be increased, the material choice (related to thermal conductivity) would be a good consideration. A further calculation indicated when the thermal conductivity of cooling tube rising, the CHF could be increased at some extent. Because a higher thermal conductivity could result in lower temperatures for a given heat flux in the surface layers closed to plasma side [6]

From the point of view on thermal conducting, a thinner tube could reduce the thermal transfer distance from CHF surface to the coolant, for example, △t=0.5, which could result in a CHF value with 6MW/m^2.

Further more, if the water condition like the inlet temperature could be dropped, it also could improve the CHF for some extent, but this should be based on practical engineering consideration.

Particularly, when the structure of the cooling channel is changed to swirl tape form, the CHF would be up to 10 MW/m^2 or more, but need to consider the thermal fatigue [7-9].

In addition, based on the present size of divertor plate, L×H=21×61mm, when the pitch of cooling tube is increased, for example, pitch=2, the contact area between the high heat flux zone and the cooling tube would be increased, it would improve heat removal capability of the cooling tube, which could result in a higher CHF. However, from the point of view of equivalent diameter (tube bore), this method would be limited.

4.2 Thermal stress issues

According to the structure analysis, the primary stress based on 15MPa water inside the cooling tube was very small compared to the thermal stress. This also was confirmed by the empirical rules in our study. So primary stress would not the key problem for the design. However, the thermal stress caused by the thermal expansion between the amour and the cooling tube is the key concern. The main reason is the difference of the coefficient of thermal expansion (α-10^{-6}/K) of the two materials. So a reduction of the different gap is the best approach. For example, when the W-F82H structure was changed to W-SiC/SiC form, it is found the thermal expansion was smaller than that of W-F82H due to a smaller gap of the △α, and the thermal stress would be dropped, as shown in table 1. In addition, a thinner wall of cooling tube could result in a smaller thermal stress because of less elongation of the cooling tube, this was confirmed the thermal performance and 3D analysis [10]

Table 1 Comparison of different material choice (d=9mm, △t=1mm, q=5MW/m^2)

results / material	Deformation (mm)	Thermal stress (MPa)
W-F82H	0.1499	363
W-SiC/SiC	0.0872	269

5. Summary

1) An allowable q≤5 MW/m^2 from a point of view of engineering consideration would be acceptable for the CHF, a higher CHF could be reached by changing the △t values, and the replacement the material of cooling tube, but it is limited due to the peak temperature trend.

2) Primary stress is small and safety from structure point of view. Thermal stress would be key concern

because of the thermal expansion, because the elongation of the armour and the cooling tube in Y direction lead to a higher thermal stress.

3) The coefficients of thermal expansion (α-10^{-6}/K) of the armour and the cooling tube are the main reason of the elongation which contribute on the thermal stress rising.

4) A thinner cooling tube would be the better choice not only for higher CHF accommodation, but also for less thermal stress.

5) This paper discussed the safety case of stress, for Sm range of F82H, it could be enlarged a more wide case due to most of the material under 550℃. In addition, for the material are plasticity, the alternating stress range and the number of alternating cycles could be considered because the secondary stress limit does not depend on a stress level over one period.

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