Structural Review for the Key Components of a Capsule Assembly Machine

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1. Introduction

To meet the demands for a high burn-up test at HANARO, capsule assembling and re-instrumentation technologies are required in the HANARO reactor. A mockup of the capsule assembly machine was designed and fabricated in 2003 [1], and pre-operation tests for the developed assembly machine were performed last year [2]. As a part of a structural integrity evaluation for the components related to the capsule assembly machine the structural analysis of the capsule external tube and handling tools is performed using an ANSYS program [3] and a theoretical formula. In order to analyze the stress of the capsule external tube by a clamping torque, the horizontal force applied at a jaw is obtained first, and used as a force applied at the external tube. A shear stress and a buckling load for two kinds of tools with a large slenderness ratio are also obtained. From this analysis, it is found that the stresses of the capsule external tube and the tool are satisfied for the structural integrity.

2. Methods and Results

2.1 Horizontal Force Applied at the External Tube

When a capsule mainbody combines with a protection tube, it should be clamped by using a clamping device to prevent a rotation or shaking of the capsule mainbody during the assembly. Figure 1 shows the schematic view of the force applied at a clamping screw and a jaw. Here the torque (T) is the moment applied at the screw by a special tool, and the minimum torque without the rotation of capsule mainbody is determined as 350kgf·cm form an experiment [4]. To evaluate the structural integrity of the capsule external tube by the clamping force (F), the stress and displacement of the external tube is calculated. The horizontal force due to the clamping torque can be calculated by using a relation of the force as shown in Figure 1 and designed data such as an average radius(1.6cm) of contact surface, a slope angle(23^{o}) of a screw and a friction coefficient(0.3). As a results of the calculation, the concentrated force by the clamping torque of 350kgf·cm is about 585.7kgf. It is transformed into the pressure acting on the surface of the capsule external tube and used as an input force.

2.2 Stress and Displacement of the External Tube

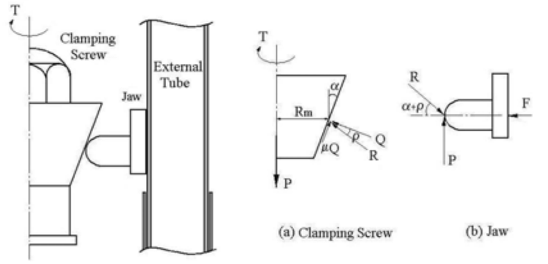

Figure 1. Schematic view of the force applied at a clamping screw and a jaw

The capsule external tube made of STS 316L materials is a cylindrical shell with 58mm in diameter and 2.25mm in thickness and 870mm in length. In the finite element analysis a quarter model of the capsule by considering a geometrical shape is generated. The analysis results show that the stress and displacement of the external tube are linearly increased by increasing the torque. Figure 2 shows the stress distribution of the external tube for a 350kgf·cm torque. The maximum stress is 108MPa and occurs at the contact area between the external tube and the jaw. In this case the maximum displacement is 0.108mm. From analysis results it is found that the stress of the external tube is lower than the yield strength of the materials, and the capsule can be assembled without a plastic deformation of the external tube.

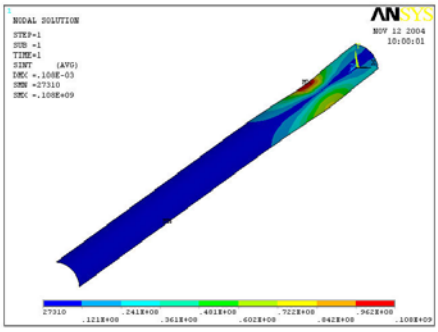

Figure 2. Stress intensity for the external tube of the capsule by the horizontal force

2.3 Shear Stress and Buckling of the Tool

Two kinds of tools according to the shape of the bolts are designed and used to fasten the capsule components. One (A-type) is for the clamping screw (M20×P2.5), and another (B-type) is for the locking bolt which joins the capsule mainbody to the protection tube. The tools are made of STS 304 materials, and the specifications are presented in Table 1.

The shear stress of the tools by the clamping torque can be calculated using the following equation [5];

$$\tau_{\max} = \frac{Tr}{J} \tag{1}$$

Where, T and J are a torque and a polar moment of the inertia, respectively. And r is a distance where the maximum stress occurs. To calculate the shear stress of the B-type tool, the clamping torque used for the locking bolt is 190kgf·cm. Using equation (1), the shear stress of the A and B-type tools is calculated as 84 and 117 kgf/cm^2, respectively. The B-type tool has a larger stress than the A-type one due to the smaller diameter, although the clamping torque is small.

Table 1. Specification of the tool

Tool	Do/Di (mm)	Thickness (mm)	Length (m)	Slenderness ratio (λ)
A-type	34/28	3.0	5	454
B-type	21.7/14.3	3.7	5	770

The tools are a slender column having a large slenderness ratio over λ =450. Thus the buckling characteristics are one of the important items to evaluate the stability of the tools. To obtain the critical buckling load of the tools, an Euler's buckling formula as shown in equation (2) is used and the results are listed in Table 2 with those of the ANSYS analysis.

$$P_{cr} = n\frac{\pi^2 EI}{L^2} \tag{2}$$

Where, n is a coefficient with the edge conditions; fixed-free (n=0.25), hinged-hinged (n=1) and fixed-hinged (n=2.04). For each boundary condition, the analyzed and calculated results show a good agreement each other. In the case of the hinged condition similar to an actual situation the critical buckling load of the A and B-type tools is 2699 and 672.9N, respectively. The buckling load of the B-type tool is small due to the relatively large slenderness ratio. However, we think this tool has enough buckling strength because the force applied to the tool during the assembly is very small. Figure 3 shows the critical buckling mode of the B-type tool for the three edge conditions.

Table 2. Critical buckling load of the tool

Edge Condition	A-type tool		B-type tool	
	ANSYS	Eq. (2)	ANSYS	Eq. (2)
Fixed-Free	674.7	674.8	168.2	168.2
Hinged-Hinged	2699	2699	672.9	672.9
Fixed-Hinged	5518.8	5506	1376.4	1373.0

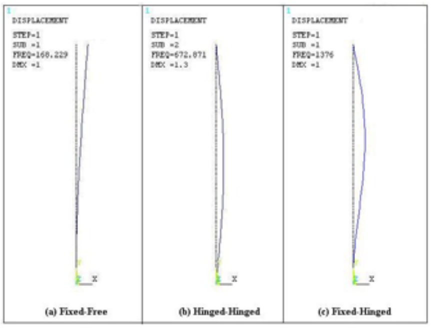

Figure 3. Critical buckling mode shapes of the B-type tool with different edge conditions

3. Conclusion

The structural analysis for key components of a capsule assembly machine is performed. For the clamping torque of 350kgf·cm, the concentrated force applied at the capsule external tube is 585.7kgf, and the analyzed displacement and stress of the external tube are 0.108mm and 108MPa, respectively. It is found that the stresses of the tube and the tool are lower than the yield strength of the materials thus satisfying the structural integrity of the components. The critical buckling loads of the A and B-type tools for the simply supported edge conditions are 2699 and 672.9N thus showing enough buckling strength, respectively.

ACKNOWLEDGEMENTS

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