Experimental Investigation into the Elastic Buckling of a Nuclear Fuel Cladding Tube

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1. Introduction

Outer diameter of a dual cooled fuel rod needs to be increased considerably from that of a conventional one due to the requirement of a surface area increase as well as the formation of an internal flow passage. As was found previously [1,2], the thickness to outer diameter ratio (t/D_o) of the conventional PWR fuel rod is mostly 0.058~0.062. It seemed to be determined from experiences since there was little information about the reason for the t/D_o values. Elastic buckling concern would be one of the reasons. Safety factor is more than 3 if a conventional design formula of the elastic buckling was applied [2]. It is almost impossible to apply those values to a dual cooled fuel rod due to a considerably decreased rod-to-rod gap size and the amount of UO_2 required for a burnup. So, there is a strong need to investigate the fundamental background of the used formula and its conservatism. In this paper, a classical theory of the stability was revisited and an experiment was conducted. Deduction for the used formula of an elastic buckling is explained. Experimental results are discussed by focusing on the safety margin of the used formula. Finally, the safety factor is considered for the presently determined thickness and diameter of a dual cooled fuel rod.

2. Theoretical Background

As has been introduced in the last year [2], the formula for an elastic buckling is as shown below.

$$p_{cr} = \frac{E}{4(1-\nu^2)} \cdot \left(\frac{t_{\min}}{r_{m,\max}} \right)^3 \tag{1}$$

where, p_{cr} is the maximum external pressure when no elastic buckling occurs, E and ν are the elastic modulus and the Poisson ratio of the cladding tube material, respectively. t_{min} and $r_{m,max}$ are the minimum tube thickness and the maximum radius of the neutral surface of the tube.

The safety factor (S) is to be evaluated as

$$S = \frac{p_{cr}}{p_D} \tag{2}$$

where, p_D is the design pressure, i.e. the difference between the external and internal pressure of the fuel cladding tubes.

Eq. (1) was derived from a classical stability theory. Griffin [3,4] and Morgan's [5] works can be consulted. Both considered two more definitions of the elastic modulus such as the secant modulus (E_s) and the tangent modulus (E_t) in addition to the Young's modulus (E). Those definitions can be found in ASTM STD E 111-04 and E 6-08.

Griffin [3,4] used the minimum potential energy theorem together with the variational principle. He assumed the deformation shape of a tube cross section due to the pressure difference was symmetrical. The potential energy was evaluated from the strain energy stored during the deformation and the work done for it. In conclusion, the critical pressure for the deformation (elastic buckling) was

$$p_{cr}^G = \frac{\eta E_s}{4(1-\mu^2)} \cdot \left(\frac{t}{r} \right)^3 \tag{3}$$

where, p_{cr}^G temporarily denotes Griffin's formula, t and r are the thickness and mean radius of a cladding tube, respectively. η and μ are defined as follows.

$$\eta = 1 - \frac{\left(1 - E_t/E_s\right)}{1 + \left(1-4\mu^2\right) E_t / \left(3E_s\right)}, \tag{4}$$

$$\mu = \frac{1}{2} - \left(\frac{1}{2} - \nu \right) \frac{E_s}{E}. \tag{5}$$

Morgan [5] used a similar approach to Griffin's method but he evaluated a critical hoop stress ($\sigma_{\theta\theta,cr}$) when a buckling occurred. It was

$$\sigma_{\theta\theta,cr} = \frac{\xi E}{4\left(1-\nu^2\right)} \cdot \left(\frac{t}{r} \right)^2 \tag{6}$$

where,

$$\xi = \frac{E_s}{E} \cdot \frac{\left(1-\nu^2\right)}{\left(1-\nu'^2\right)} \cdot \left(\frac{1}{4} + \frac{3}{4} \frac{E_t}{E_s} \right), \quad \nu' = 0.5 - 0.2 \frac{E_s}{E}. \tag{7}$$

Since $\sigma_{\theta\theta} = pr/t$, Morgan's formula of the critical pressure for an elastic buckling (p_{cr}^M) becomes

$$p_{cr}^M = \frac{\xi E}{4(1-\nu^2)} \cdot \left(\frac{t}{r} \right)^3. \tag{8}$$

It is readily found that both p_{cr}^{G} and p_{cr}^{M} are deduced to be p_{cr} for Eq. (1) if $E_s = E_t = E$ (i.e., the material is perfectly linear), $\nu = 0.3$ (for Morgan's formula *ad hoc*), and t and r are replaced with t_{min} and $r_{m,max}$, respectively for a conservatism. The details of the derivation can be found elsewhere [6].

3. Experimental

To investigate the conservatism of p_{cr} of Eq. (1), elastic buckling experiments were conducted with using actual cladding tubes for PHWR fuel rods. The reason of using the PHWR claddings rather than the PWR ones was that the pressure capacity of our autoclaves was less than the PWR internal pressure (≈ 15 MPa). The cladding tubes had a minimum thickness of 0.38 mm and a maximum outer diameter of 13.12 mm. The tube specimen length was 20 mm after the end plugs were welded to both ends of the tube specimen. Elastic modulus and the Poisson ratio were 77.4733 GPa and 0.30, respectively, which were obtained from the design data of PHWR fuels and MATPRO data [7] at 350°C. With these, p_{cr} (Eq. (1)) was 4.518 MPa.

The pressure range in the experiment was 2 ~ 10 MPa. After each tube specimen was loaded into an autoclave, the temperature was increased to be 350°C and maintained for one hour. Then, the specimen was cooled down inside the autoclave. The occurrence of elastic buckling was determined from an existence of the collapsed tube surface and/or a diameter change. The pressure inside the tube specimen was set as 0.21 MPa constantly, the critical pressure was the pressure difference between the applied pressure and that when none of the specimens collapsed.

4. Results and Discussions

Experimental result is provided in Table I. And a typical view of the collapsed and non-collapsed tubes is shown in Fig. 1 when the pressure difference was 3.893 MPa. Although one specimen did not collapse in Fig. 1, that pressure difference was regarded to be higher than the critical pressure.

The critical pressure was determined as 3.824 MPa from the experimental result. It is less than 4.518 MPa evaluated from Eq. (1). This means that the conventionally used formula of elastic buckling is non-conservative by around 18.2%. In other words, an actual pressure for an elastic buckling should be less than the evaluated one by more than around 15.4%.

Returning to Eq. (1), the minimum thickness (t_{min}) to outer diameter (D_o) ratio for no elastic buckling should be larger than the following.

$$\frac{t_{\min}}{D_o} \geq \frac{K}{1+K}, \quad \text{where} \quad K = \sqrt[3]{\frac{1.182 \cdot P_D \cdot 4\left(1-\nu^2\right)}{E}} \tag{9}$$

The number, 1.182 in Eq. (9) is used to compensate the overestimation of the critical pressure in Eq. (1).

Presently, the outer diameter and thickness of the outer cladding is determined as 15.9 and 0.87 mm, respectively. When E = 70140.36 MPa, ν = 0.25 for Zircaloy-4 at 350°C are plugged in, the safety factor is evaluated as 2.33. So it is concluded that the present dimension is safe against the elastic buckling problem.

Table I: Result of elastic buckling experiment

Pressure difference (MPa)	Specimen loaded	Number of collapsed tubes
4.065	1	1
3.961	3	2
3.927	1	0
3.893	3	1
3.824	3	0
3.513	1	0
2.272	1	0

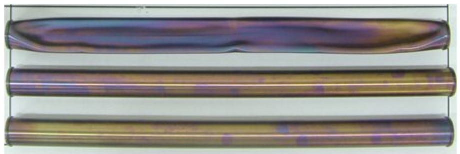

Fig. 1. Typical view of collapsed and non-collapsed tube specimen (at pressure difference of 3.893 MPa).

5. Conclusions

It was found that the conventional formula for an elastic buckling is not conservative enough by around 18.2% after analyzing the experimental result. Nevertheless, it is concluded that present thickness and diameter of a dual cooled fuel rod's outer cladding is safe against an elastic buckling.

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