RING TENSILE TEST OF REFERENCE ZIRCALOY CLADDING TUBE AS A PROOF OF PRINCIPLE FOR HOTCELL SETUP

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ABSTRACT

A proof of principle test on standard Zircaloy cladding tubes was successfully performed at a NRG's reference laboratory. It was performed before placing the ring tensile setup in the hotcell for active materials testing. The most suitable geometry of the half-mandrels was obtained using FEA. FEA was also applied to study the evolution of the maximum applied load along the gauge side of the cladding due to a change in the friction coefficient (μ). The tests were performed at RT and 300 °C for measurement with and without lubricant (MoS), and at 400 °C for measurement without lubricant. The proof of principle tests have successfully measured the UTS and Total Elongation of the cladding tube. Further step will be to test the stability of the lubricant at high temperature up to 450 °C. The ring tensile setup is expected to be installed at the hotcell laboratory by summer 2019.

Introduction

Zircaloy is mainly used as an accident tolerant fuel cladding. It is found to have a strong crystallographic texture and is an anisotropic material in nature. Therefore, for the analysis of failure of high burn-up fuels under RIA conditions by using computer codes, data pertaining to the mechanical properties of the fuel cladding in the hoop direction is required.

One of the most common test setups to investigate the hoop stress in Zircaloy tube is a ring tensile test. The ring tensile test has some advantage over the tube burst test, e. g. uniaxial hoop loading can be applied in the ring tensile test. It has also another advantage, especially in the test of irradiated fuel cladding, such as the requirement of small specimen volume in the test apparatus.

NRG built and installed a ring tensile setup at a standard reference laboratory. The experiments were performed as a proof of principle of the ring tensile setup before moving it to the hotcell laboratory. This paper shows the effort taken by NRG's R&I team to develop and expand our capability in performing cladding research. We expect that we will be able to perform ring tensile test for active cladding tubes and contribute to the development of new cladding materials such as ATF cladding tubes.

Experimental Setup

The ring tensile setup consists of two-half mandrels (Figure 1).The test setup was developed to fit the existing Instron 8562 model at NRG's reference laboratory. The hotcell laboratory has a typical Instron machine, thus, it is expected that there will be no modification needed when placing the half-mandrels at the Instron machine at the hotcell laboratory.

Finite element analysis at room temperature was performed to investigate the optimized radius of the half-mandrel and the most uniform distribution of the maximum applied load at the gauge areas. There are two gauge areas that are placed at the bottom and the top of the half-mandrels. The radii of the half-mandrel used as FEA inputs were 4.125, 4.075, 4.00 and 3.00 mm while the friction coefficients (μ) were 0, 0.03, 0.125, 0.30. The Zircaloy-4 tube used has an inner diameter of 8.25 mm and an outer diameter of 9.5 mm.

The ring specimens were fabricated by taking into account of one of the critical geometry parameters of the ring tensile specimen, i. e. the ratio between the gauge length (l) and width (w). The l/w ratio of 1.5 will result in a uniform strain distribution. However, it has a drawback such that it will have a rapid development of plastic instability [1] [2]. The l/w in the range of 3 - 4 may result in a more uniform strain distribution and delayed necking. Based on this understanding, we decided to fabricate the ring specimen with l/w = 4 [3] [4] [5]. Additionally, the ring specimen has a ratio of the radius and width (r/w) of 1.

The ring tensile measurements were performed at RT and 300 °C for measurements with and without lubricant, and 400 °C for measurement only without lubricant. The lubricant used was MoS lubricant. The strain rate was 5 x 10^{-4} /s and the crosshead velocity was 4.76 x 10^{-4} mm/s.

Figure 1. The ring tensile setup (Instron 8562) at the standard NRG´s reference laboratory.

Results and Discussion

1. Finite Element Analysis

FEA (ANSYS) was performed before fabricating the half-mandrel to study the most optimized geometry of the half-mandrel based on the presence of friction coefficient (μ). Friction coefficient (μ) is one of the main critical parameters in performing the ring tensile test. The presence of a friction coefficient results in an increase of energy needed by the half-mandrels to break the ring specimen. Ideally, the friction coefficient of zero is preferable. However, it will never happen in practice. Thus, it is important to have a good control and reproducibility of friction coefficient during the ring tensile test and try to reduce it by adding a lubricant.

Table 1 shows the evolution of the location of the maximum principle stress as a result of the different mandrel radius and friction coefficient (μ). Note that the FEA was performed at RT. The decrease of a mandrel radius resulted in a shift of maximum stress toward the side of ring specimen. While the increase of friction coefficient resulted in a shift of maximum stress toward the edge of ring specimen. See Figure 2 for the explanation of the centre, midway, edge and side locations of the gauge areas. Be aware that by decreasing the half-mandrel radius, it also means that the gap between two half-mandrel increases. It further can be correlated to the more likeliness of the presence of bending stress.

The green marker (Figure 2) showed the most efficient half-mandrel geometry with the maximum stress located in the centre of the gauge areas of the ring specimen (μ = 0.03). The maximum stress moved toward the edge of the gauge area with the increase in the friction coefficient. A high friction coefficient of 0.3 resulted in a presence of two regions at the edge of gauge areas having the maximum stress. Based on the FEA, the two half-mandrels were finally fabricated with a radii of 4.125mm and no-gap between the two-half mandrels.

Table 1. Summary of the geometry of the half-mandrel and friction coefficient (μ) as input for the FEA at RT as well as where the maximum principal stress is located at the gauge areas.

Friction coefficient (μ) / Mandrel radius	0.03	0.125	0.3
4.125	Centre*	Midway*	Edge**
4.075	Edge*	Edge*	--
4.000	Side*	Side*	--
3.000	Side*	Side*	--

*See Figure 2 for the details of each location.

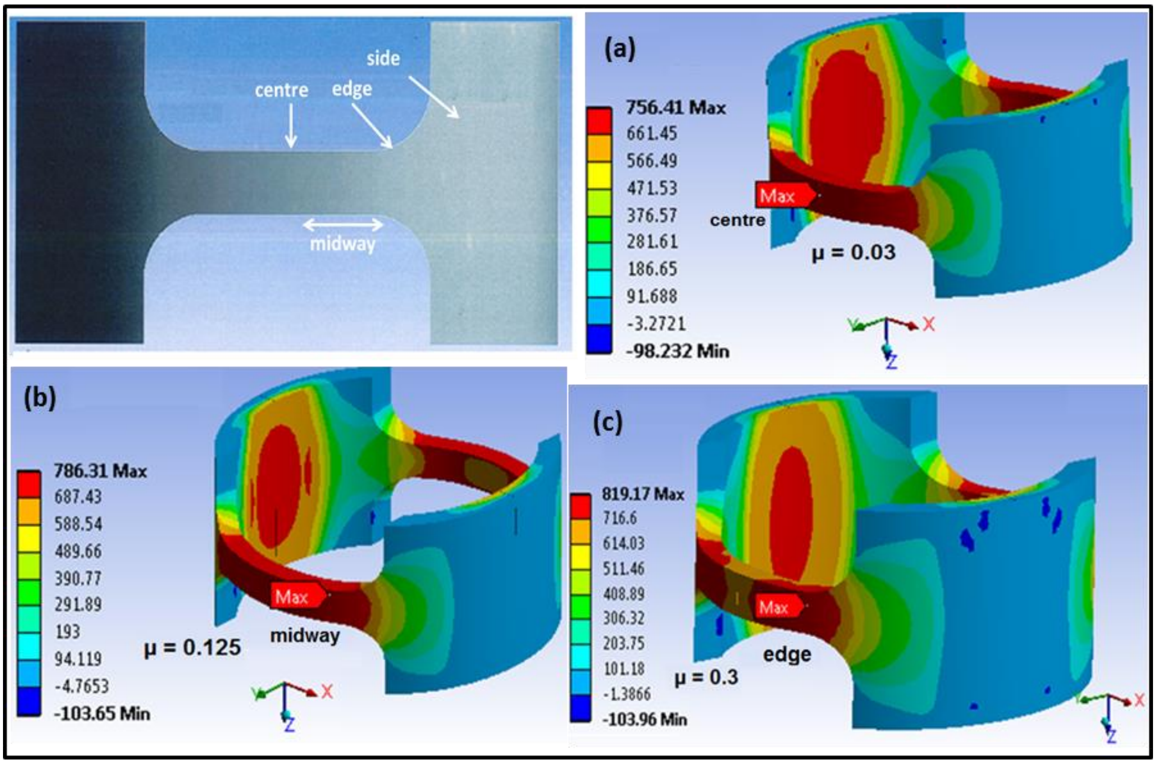


Figure 2. Finite element analysis at RT showing the influence of the friction coefficient (μ = 0.03, 0.125 and 0.3) on maximum stress along the gauge areas. The radii of the half mandrel used was 4.125mm.

The maximum load obtained from the FEA with $r_{mandrel}$ = 4.125mm was further plotted versus displacement as seen in Figure 3. FEA with friction coefficient of 0 was added. The maximum applied load of the friction coefficient of 0 is similar to that of the measurement with friction coefficient of 0.03.

In order to have better understanding of the influence of the friction coefficient on the maximum load, the percentage of the increase in maximum load is plotted versus the friction coefficient (Figure 4). The blue line represents the data from FEA and the red area is the range of the friction coefficient of MoS lubricant (μ = 0.06 – 0.125). The curve would be then used to validate the efficiency of the MoS lubricant to reduce friction coefficient during the ring tensile measurement at room temperature.

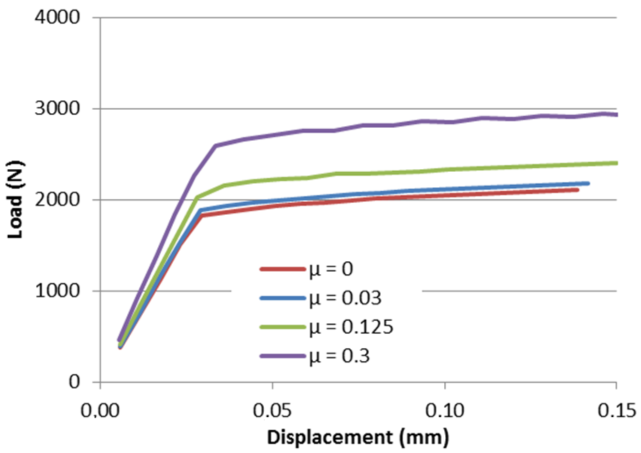


Figure 3. Load versus displacement curves from FEA at RT showing the influence of friction coefficient (μ) on the maximum load ($r_{mandrel}$ = 4.125mm).

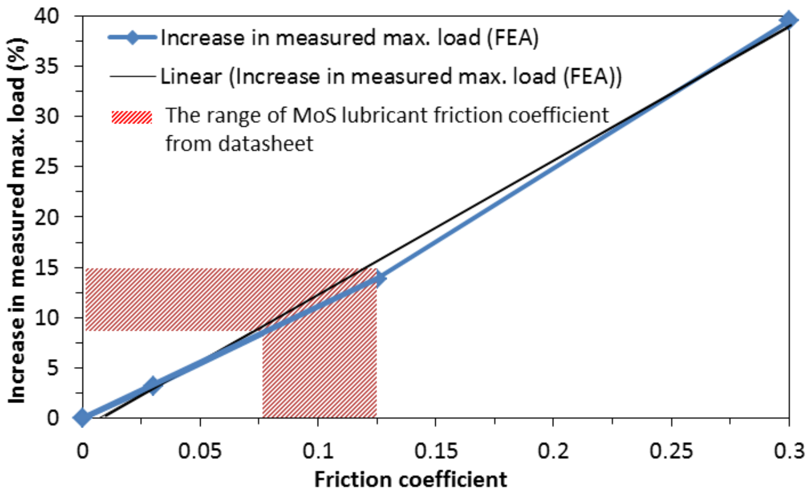
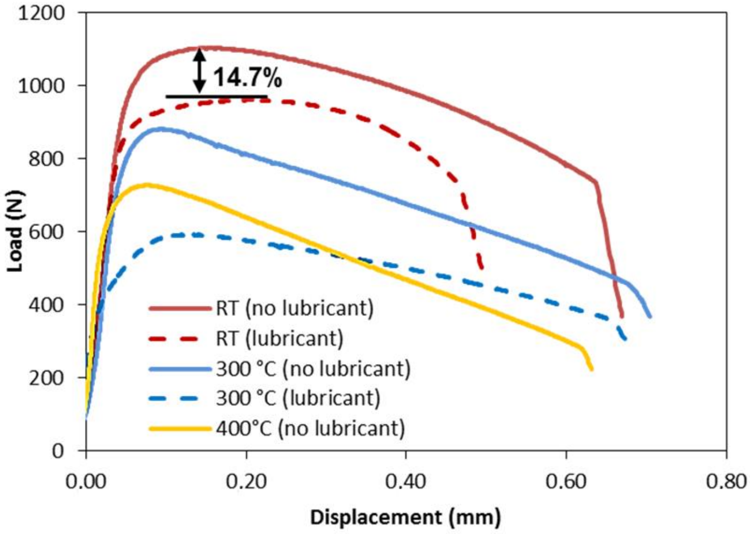


Figure 4. Friction coefficient influence on the maximum applied force and the percentage (%) of the maximum load. Note that this is taken from FEA at RT.

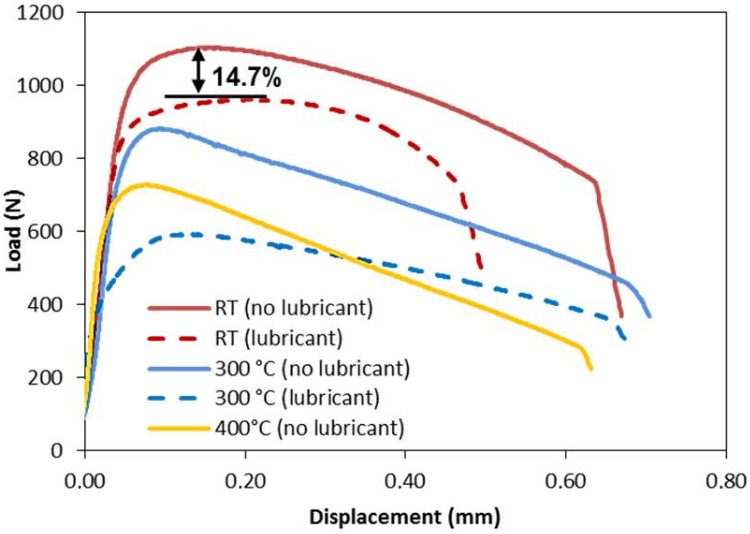
2. Ring Tensile Measurement

Several steps were taken into account to obtain the correct values of the load and displacement. The steps include the correction of the zero-displacement and machine



compliance.

Figure 5 Figure 5 shows the load versus displacement curves after the necessary



corrections. From

Figure 5, we could see that the increase of the maximum load at RT due to the absence of lubricant reached up to 14.7%. By putting this value into Figure 4, we would get the friction coefficient value of ~ 0.11. This value is in the range of the friction coefficient of the MoS lubricant from the datasheet.

The MoS lubricant also reduced the maximum applied load needed to break the specimen at 300 °C. It can be seen from Figure 5 blue line and blue dash line. The maximum applied load decreased up to 40 % when the lubricant was added. However, as the FEA was only performed at RT, we are not able to validate this value yet. The only thing that we are able to confirm is that MoS lubricant can withstand a temperature up to 300 °C.

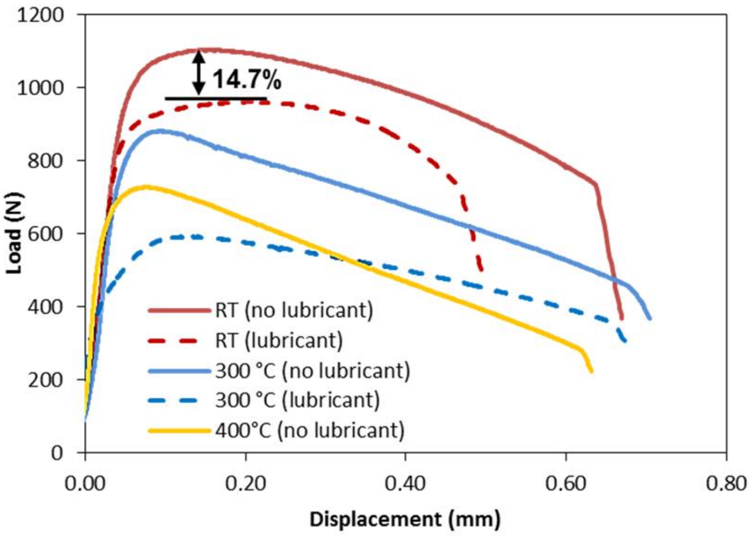


Figure 5. Load versus displacement curves of several ring tensile measurements of the Zircaloy-4 after the correction of the zero-displacement and machine compliance.

Further data analysis is to define the stress and strain. The stress is translated from the applied load as follow,

$$\sigma = \frac{F}{2A_0}$$

in which *F* is the applied load and A_0 is the gauge area. The measured applied load is divided by two as the measured load is recorded from two half-mandrels.

The strain is translated by dividing the actual displacement by the effective gauge length,

$$\varepsilon = \frac{L_{actual}}{L_{eff}}$$

The L_{actual} is the corrected displacement after machine compliance and zero-displacement corrections. The L_{eff}, "effective" gauge length, is determined by dividing the final plastic displacement by the plastic strain, as explained by Daum, et al. (2002) [5].

Figure 6 shows the results of the calculated Ultimate Tensile Strength (UTS) and Total Elongation (TE) of the Zircaloy-4. It is clear that the increase of the temperature influences the UTS as well as the TE. The addition of the lubricant is also found to decrease the energy needed to break the specimen. It also means that the measurements with the lubricant result in a lower stress in comparison to that of without the lubricant.

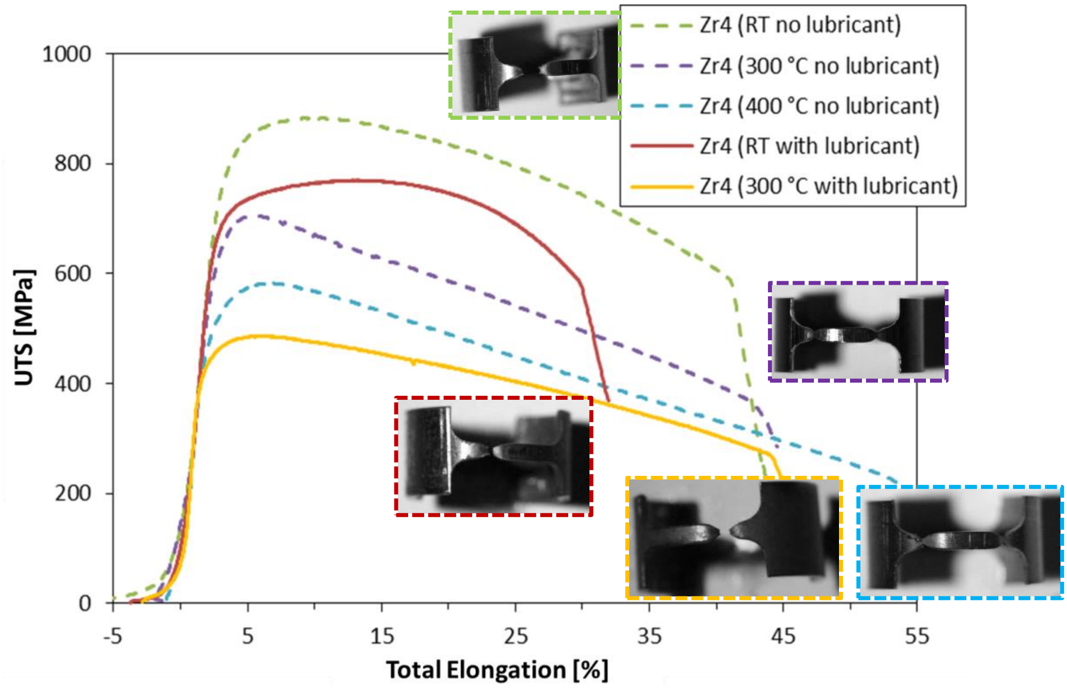


Figure 6. The Ultimate Total Strength (UTS) versus Total Elongation (TE) curves of ring tensile measurement at several temperature conditions with and without MoS lubricant.

The UTS and TE values were shown in Figure 7 and Figure 8, respectively. Some literature data were also added as a comparison showing the conformity of our results to the earlier measurements performed by other groups.

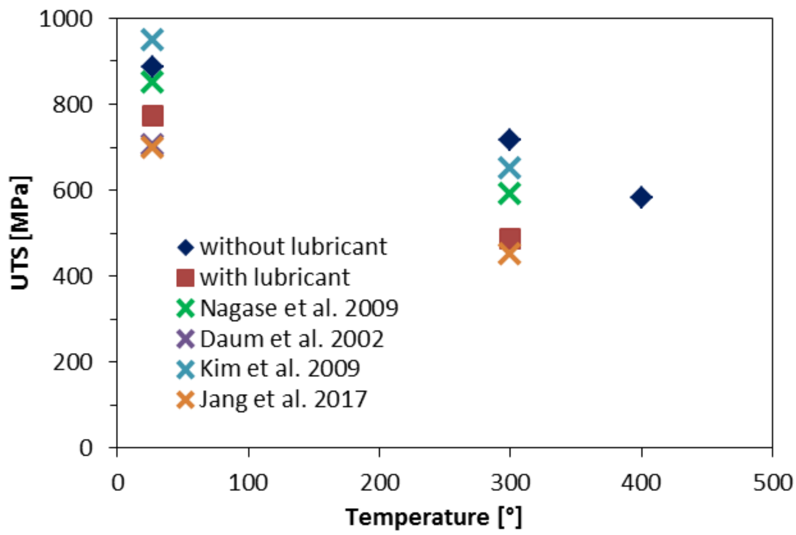


Figure 7. Ultimate Tensile Strength (UTS) as a function of temperature for our experiments in comparison to the values from literatures.

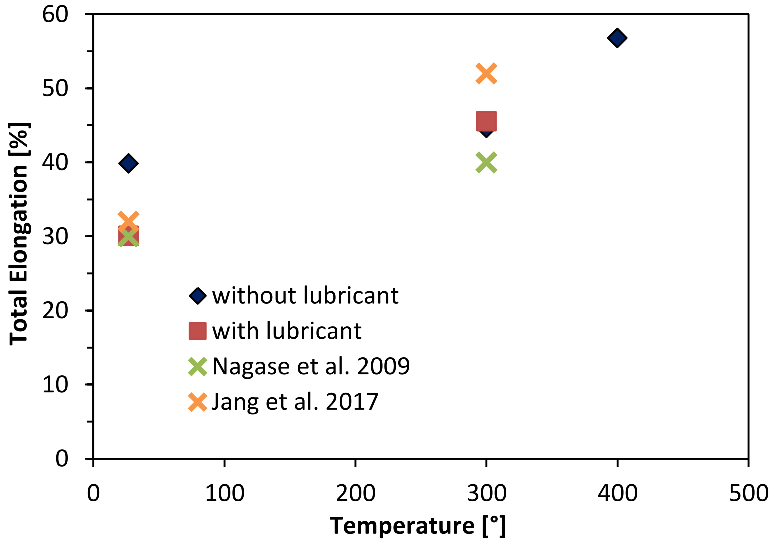


Figure 8. The Total Elongation (TE) as a function of temperature for our experiments in comparison to the values from literatures.

Conclusion and Outlook

A new ring tensile setup has been developed and installed at a NRG´s reference laboratory. The proof of principle tests were successfully performed. FEA analysis was used to investigate the influence of half-mandrel geometry and friction coefficient of the test results.

The half-mandrels used for the measurement has a geometry in a way that there is no gap between each half-mandrel. From FEA, the lack of a gap between the half-mandrel provided the most uniform force along the gauge. The FEA was also used to track the evolution of the maximum applied load due to the friction coefficient. The high friction coefficient resulted in the appearance of the cracks on the both sides of the gauge areas. For the time being, the FEA was only performed at RT.

The ring tensile measurements were performed at RT and 300 °C for measurement with a lubricant (MoS) and without a lubricant, and at 400 °C for measurement only without a lubricant. From the experimental data, the MoS lubricant was found to be able to withstand the temperature up to 300 °C. It was clear as there was a decrease in the maximum applied load at 300 °C due to an addition of MoS lubricant.

Some literature data were also added as comparison showing the conformity of our results to the earlier measurements performed by other groups.

Further research will be carried out to test MoS lubricant at 450 °C. Another lubricant that can be used at a higher temperature up to 450 °C can also be tested. It is foreseen that the ring tensile setup will be installed at the hotcell lab by summer 2019.

Acknowledgement

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