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Evolution law of the friction coefficient and fatigue test of the hold-down spring model for nuclear reactor vessel internals☆

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A R T I C L E I N F O

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A B S T R A C T

A reactor pressure vessel may be opened as much as 200 times during its design lifetime of 60 years. A preloading operation is required when a Hold-down Springs (HDS) are opened and closed; hence, the size of a load is essential for the safe operation of a reactor. The friction coefficient of the HDS and the container flange significantly affect the preload. In this study, a mathematical model of the axial load P and the friction coefficient f are established based on the mathematical derivation of a physical model of the HDS. A set of test devices for simulating the preloading process of the HDS is designed according to similarity criteria. Results of the cyclic loading test indicate that the amount of deformation decreases, whereas the friction coefficient increases, as the times of cyclic loading increases under the same loading condition. The friction coefficient increases from approximately 0.3 at the initial loading to approximately 0.6 at the 200th cyclic loading. The results show that enveloped areas are present in the loading and unloading curves of the HDS because of the instability and slip of the ring structure and the hysteresis of the load. Moreover, an increase in the opening times of the reactor pressure vessel will lead to the same displacement; thus, preload will also increase. The evolution law of the fatigue cycle, the friction coefficient, and the mathematical model of the load provide an effective method for accurately calculating preload. An effective preload ensures the safe and reliable operation of a reactor.

1. Introduction

Reactor internals comprise compacted parts, basket parts, an irradiation supervision tube, and other accessory parts. Compacted parts include a compacted supporting structure, a guide tube, an in-pile temperature measuring device, a compacted hold-down spring (HDS), a supporting plate, and an adjustment pad (Choi et al., 2013; Xue et al., 2016). The HDS is a large-diameter ring-shaped pressure spring with a Z cross section; this ring (Kim and Suh, 2008; Zhai and Guohong, 2013; Yamanoa et al., 2011) is installed between the basket flange and the supporting flange. The basket flange is placed over the nozzle belt flange of the pressure vessel, whereas the outer edge of the upper surface of the supporting flange is fastened by the cap of the pressure vessel, as shown in Fig. 1. When the reactor is running, the reactor internals are subjected to a vertical load because of the flow field (Kaoa et al., 2011). During this period, the compaction force of the HDS prevents the reactor internals from moving in the vertical direction. The compaction force of the HDS acts on the supporting and basket flanges. The magnitude of the compaction force should be appropriate because a weak compaction force will fail to compact the reactor internals, whereas a strong compaction force may damage the flanges (Ehrnstén et al., 2013; Jian et al., 2014).

Studies on the actual operation of active nuclear power plants have shown that the frequent opening of the reactor pressure vessel results in serious wear of the compacted HDS. Each opening will increase wear and tear, along with friction force. The thickness of the HDS decreases as wear increases, which weakens the compaction force and affects the safe operation of the reactor. Therefore, the HDS is designed to minimize wear (Kim, 2010; Sigrist et al., 2006; Lei and Zonghua, 2015) and to ensure that a sufficient preload force is applied. However, the friction force between the HDS and the contact surfaces of the upper and lower flanges directly affects the axial stiffness of the ring. The friction coefficient changes with an increase in the number of times of opening, the axial preload force increases under the same initial displacement condition, and the internal stress value also increases. The friction coefficient changes with an increase in the expected frequency and surface wear, thereby changing the preload (Xia et al., 2016; Zhou et al., 2016). In this paper, a set of HDS friction coefficient testing device is designed, and evolution law of the friction coefficient and the change of the corresponding HDS preload are obtained. The analysis of the change in

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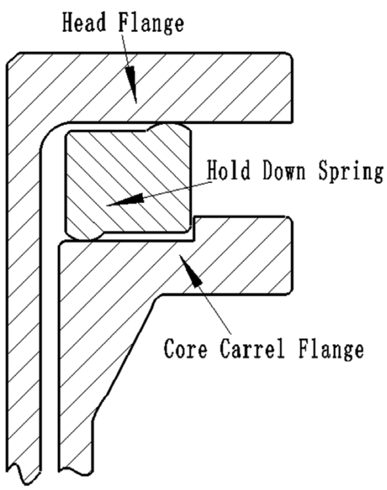

Fig. 1. Installation diagram of the HDS.

the friction coefficient provides an accurate basis for the design, calculation, analysis, and structural selection of the HDS. Moreover, it ensures that the stress intensity of the HDS under the maximum displacement load is controlled within the allowable limits set by the American Society of Civil Engineers (ASME, 1989; RCC-M Edition, 2007).

2. Deformation analysis and experiment design of the HDS model

2.1. Deformation analysis of the HDS model

The HDS has a rectangular cross section, which will produce a small angle θ without any other deformation under the action of a uniformly distributed torque M_t along the neutral l ine; M_t will generate circumferential stress on the circumference (Hua et al., 2016). The analysis of stress on the HDS shows that the HDS can be regarded as a twist caused by a uniformly distributed torque along the neutral line, with reference to the early derivation of the friction coefficient (Linjun et al., 2016). When load is applied to the HDS, torque M_t is generated from the total load P of the axial force as follows:

$$M_t = pa = \frac{Pa}{2\pi}, \tag{1}$$

where a is the distance from the stress point to the supporting point of the HDS.

After the HDS is subjected to the axial force P, circumferential bending stress is generated in the cross section and the maximum value is obtained at four angles. Moreover, an axial displacement δ occurs. For the torque balance of the HDS, the maximum bending stress σ_{Max} and the angle θ are expressed as (Linjun et al., 2016)

$$\sigma_{\theta\,\max} = \frac{Pah}{4\pi J_x}, \quad \theta = \frac{Pa^2}{2\pi EJ_x}. \tag{2}$$

The axial displacement δ can be expressed as the section twist angle θ multiplied by the distance D/2:

$$\delta = \frac{D}{2}\theta = \frac{Da^2P}{4\pi EJ_x}, \tag{3}$$

where h is the effective height of the HDS, Jx is the moment of inertia of the section, and D is the equivalent diameter of the HDS.

In the loading process, sliding would occur at the acting point of force and thus friction force, i.e., fp (f is friction coefficient) exists. A counter moment fpH would be produced, in Fig. 2. The countertorque causes a decrease in the axial displacement δ under the same axial force P; that is, the stiffness of the HDS is improved. In accordance with Eqs. (2) and (3) and Reference (Linjun et al., 2016), the δ displacement

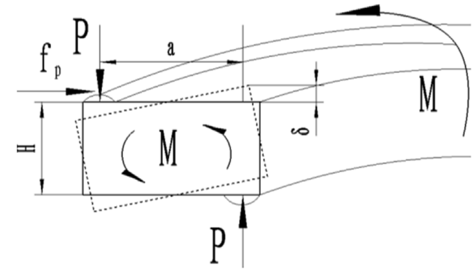

Fig. 2. HDS stress diagram that considers friction.

formula for the HDS and the $\sigma_{\theta\max}$ formula for the maximum circumferential bending stress on the four angles of the upper and lower edges of the HDS under friction force are expressed as

$$\delta = \frac{aD(a - fH)P}{4\pi J_x E}, \tag{4}$$

$$\sigma_{\theta max} = \frac{(a - fH)hP}{4\pi J_x}. \tag{5}$$

The flexibility coefficient λ_f and the stress coefficient k_{af} under friction force can be derived using Eqs. (4) and (5) as follows:

$$\lambda_f = \frac{\delta}{P} = \frac{Da(a - fH)h}{4\pi J_x E}, \tag{6}$$

$$k_{af} = \frac{\sigma_{\theta\,\max}}{P} = \frac{(a - fH)h}{4\pi J_x}. \tag{7}$$

The mathematical models of the load and the maximum stress and displacement are expressed in Eqs. (6) and (7), respectively. The flexibility coefficient λ_f, the stress coefficient k_{af}, and the friction coefficient of the HDS can be obtained through experiments.

2.2. Test methods and device design

In the test, the material selected for the HDS of the reactor internals is SA182-F6NM martensitic stainless steel. Two spring sizes, A1000 and A1400 are tested. The dimensions of these HDS are provided in Figs. 3 and 4.

All model types cannot be tested in the laboratory during the experimental study because of the large size of the HDS. Therefore, a model should be scaled down according to model similarity criteria (Uryukov et al., 2004). When tested, the model ratio is 1:10 and the elastic modulus of the same material is equal.

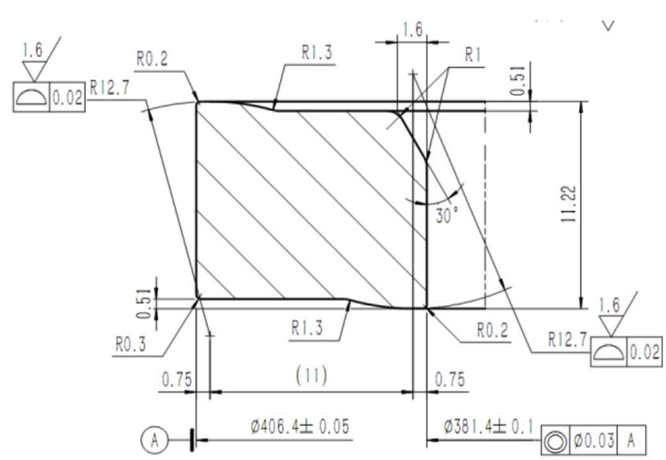

Fig. 3. A1400 size structure.