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Original Article

Dynamic characteristics assessment of reactor vessel internals with fluid-structure interaction

Sang Yun Je [a], Yoon-Suk Chang [a,*], Sung-Sik Kang [b]

[a] *Department of Nuclear Engineering, Kyung Hee University, 1732 Deogyeong-daero, Giheung-gu, Yongin-si, Gyeonggi-do 17104, Republic of Korea*
[b] *Korea Institute of Nuclear Safety, 34 Gwahak-ro, Yuseong-gu, Daejeon-si, 34142, Republic of Korea*

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ABSTRACT

Improvement of numerical analysis methods has been required to solve complicated phenomena that occur in nuclear facilities. Particularly, fluid-structure interaction (FSI) behavior should be resolved for accurate design and evaluation of complex reactor vessel internals (RVIs) submerged in coolant. In this study, the FSI effect on dynamic characteristics of RVIs in a typical 1,000 MWe nuclear power plant was investigated. Modal analyses of an integrated assembly were conducted by employing the fluid-structure (F-S) model as well as the traditional added-mass model. Subsequently, structural analyses were carried out using design response spectra combined with modal analysis data. Analysis results from the F-S model led to reductions of both frequency and Tresca stress compared to those values obtained using the added-mass model. Validation of the analysis method with the FSI model was also performed, from which the interface between the upper guide structure plate and the core shroud assembly lug was defined as the critical location of the typical RVIs, while all the relevant stress intensities satisfied the acceptance criteria.
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1. Introduction

Reactor vessel internals (RVIs) perform safety-related functions such as holding up the nuclear fuel assembly, provide coolant passage through the reactor core, and support the control element drive mechanism. In the case of the functional loss of RVIs, the nuclear fuel assembly will be damaged and subsequent failure of the reactor pressure vessel (RPV) due to impact of fallen parts may lead to severe accidents. Hence, during the past several decades, diverse analyses and experiments have been carried on the internals in order to resolve relevant safety concerns. However, from the point of view of design and evaluation, there is still room for improvement of analysis methods related to complicated physical phenomena and insufficient computational accuracy. For instance, realistic dynamic characteristics and behaviors of complex RVIs have not been explicitly taken into account for practical applications.

Many recent studies have focused on the structural integrity assessment of RVIs and other major components. Jhung and Ryu [1] performed response spectra and time history analyses of a simple mechanical component against earthquakes, and compared their results. Several base excitation types were considered and special attention was recommended as a basis for further dynamic analysis. Park et al. [2] examined the modal characteristics of RVIs based on scale-similarity analysis with fluid-structure interaction (FSI). It was observed that the added-mass (A-M) model for submerged structures is considerably dependent on mode shapes and natural frequencies. Sigrist et al. [3] also conducted comparative dynamic analyses with FSI modeling for pressure vessel and internals in a nuclear reactor. They proved that the coupling effect is significant, whereas the effect of added-stiffness on global behavior is negligible. Choi et al. [4] identified dynamic characteristics of a scale-down System-integrated Modular Advanced ReacTor (SMART) by FSI modeling. They showed that the overall natural frequencies in water decrease dramatically compared with those obtained from totally assembled reactor internals in air. In addition, dynamic characteristics were investigated by considering holes and sloshing at free vibration conditions [5–8]. Seismic responses were analyzed for butterfly valves, reactor coolant pump, and reactor internals [9–12], and the FSI effects were evaluated: not only thermal fatigue, but also comprehensive vibration assessment [13,14]. However, most of these numerical studies did not incorporate actual complex structural geometry or realistic analysis conditions, which might contribute to the lack of accuracy.

* Corresponding author.
E-mail address: yschang@khu.ac.kr (Y.-S. Chang).

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In this paper, we examined the effects of the FSI on the dynamic characteristics of the internals in an integrated assembly in a typical 1,000 MWe nuclear power plant. At first, modal analyses were performed using both traditional A-M and alternative FSI models to determine individual mode shapes and frequencies. Subsequently, structural analyses were carried out using design response spectra combined with the modal analyses data. Finally, to ensure structural integrity of the internals after validating the analysis method with the FSI, differences of the two sets of analyses were investigated in detail and resulting stress intensities were compared with the corresponding limit values based on ASME (American Society of Mechanical Engineers) B&PV (Boiler and Pressure Vessel) Code Section III, Subsection NG [15].

2. Analysis conditions and method

2.1. Brief description of RVIs

The RVIs dealt with in this study are classified into two major parts of core support barrel (CSB) assembly and upper guide structure (UGS) assembly. The CSB assembly includes the CSB itself, the lower support structure (LSS), the in-core instrumentation nozzle assembly, and the core shroud assembly (CSA). The CSB is a circular cylinder supported from a ledge on the RPV by a ring flange; the CSB carries the entire weight of the core. There are six equally spaced snubbers at the bottom of the CSB that prevent torsional motion of the internals. The LSS transmits the weight of the core to the CSB by means of a grid beam structure. The CSA surrounds the core and minimizes the amount of bypass flow. By contrast, the UGS assembly is located above the reactor core within the CSB. Its main functions are to align and support the fuel assemblies, maintain the control element assembly shroud spacing, prevent movement of the fuel assemblies in case of severe accident conditions, and protect the control rods from cross-flow effect.

2.2. Modeling details

In order to investigate the dynamic characteristics of the RVIs, three-dimensional finite element (FE) models were developed using a general purpose program (ANSYS Workbench version 17.2; ANSYS Inc., Canonsburg, PA, USA). Fig. 1 depicts a typical assembled model consist of the CSA, UGS, CSB, and LSS with encompassing fluid, as well as the RPV and four support columns. Since the amount of cooling water moving to the upper head of the RPV through the UGS was less than 0.1% of the total water quantity, it was not modeled due to effective computational time, as well as negligible flow effect [16]. Each subcomponent with coolant, and the RPV with support columns, were modeled using solid structure elements (element type Solid 185) and fluid elements (element type Fluid 221) in the ANSYS library (ANSYS Inc.), respectively.

Boundary conditions were set by considering the states of the subcomponent assembly according to the design documents, as shown in Fig. 2. Especially, the upper flange of the CSB was assembled with a reactor upper head by alignment keys; these keys were connected to top of the UGS by a hold down ring. Thus, the degrees of freedom (DOFs) of the upper CSB were fixed along the vertical and circumferential directions. UGS fuel alignment plate and CSA guide lug were coupled by insert pins so that the jointing parts were fully fixed. Additionally, since the CSB snubber lugs were tied to the vessel core stabilizing lugs and the cold leg nozzles were welded to the supports, the DOFs of the lower CSB were fixed along the circumferential direction and the bottoms of the support column were fully fixed. Materials of the RPV and RVIs considered in this study are SA508 carbon steel and TP 304 stainless steel, respectively; their properties at operating temperature with damping ratios are summarized in Table 1.

2.3. FSI analysis methods

There are two well-known methods to evaluate the interaction between a structure and an acoustic fluid: one is pressure-based formulation and the other is displacement-based formulation. Although the latter formulation is easier to implement due to the construction of symmetric mass and stiffness matrices, it suffers from spurious resonances. While the former formulation generates nonsymmetric matrices and unnecessary eigenmodes of the fluid, it has the advantage of fewer unknowns [17]. In this research, both methods were examined for comparison.

At first, the following dynamics equation was employed to consider the fluid as an A-M [2,18]:

$$[M_s]\{\ddot{u}\} + [C_s]\{\dot{u}\} + [K_s]\{u\} = \{f_e\} + \left\{f_f\right\} \tag{1}$$

where $[M_s]$, $[C_s]$, and $[K_s]$ denote the mass, damping, and stiffness matrices, respectively, and $\{\ddot{u}\}$, $\{\dot{u}\}$, and $\{\ddot{u}\}$ are the acceleration, velocity, and displacement vectors for the structure. In Eq. (1), the total load applied to the structure is given by sum of the external force vector, $\{f_e\}$, and the hydrodynamic force vector, $\{f_f\}$, at the fluid-structure (F-S) interface. The hydrodynamic force can be obtained from the integral of the fluid pressure vector $\{p\}$ with respect to the infinitesimal interface area dS, as in Eq. (2). Here, $\{Np\}$ is the approximating shape function for the spatial variation of the pressure, $\{n\}$ is the unit vector normal to the fluid-structure interface, and $[M_a]$ is the A-M matrix:

$$\left\{f_f\right\} = \int_S \{N_p\}^T\{n\}\{p\}dS = -[M_a]\{\ddot{u}\} \tag{2}$$

Substituting Eq. (2) into Eq. (1) yields:

$$([M_s] + [M_a])\{\ddot{u}\} + [C_s]\{\dot{u}\} + [K_s]\{u\} = \{f_e\} \tag{3}$$

Thus, the dynamic characteristic equation including the A-M causes a reduction of the frequencies.

Alternatively, the FSI can be evaluated by considering the coupling structural and fluid behaviors. The well-known structural dynamics equation was modified to consider the fluid pressure transferred to the structure:

$$[M_s]\{\ddot{u}\} + [C_s]\{\dot{u}\} + [K_s]\{u\} - [R_{int}]\{p\} = \{f_e\} \tag{4}$$

where $[R_{int}]$ is the stiffness matrix of the fluid interacting boundary. By contrast, fluid momentum and continuity equations were simplified using the acoustic wave equation to take into account the coupling mass matrix at the interface, as in [4,18]:

$$\left[M_f\right]\{\ddot{p}\} + \left[C_f\right]\{\dot{p}\} + \left[K_f\right]\{p\} + \rho[R_{int}]^T\{\ddot{u}\} = \{0\} \tag{5}$$

where $[M_f]$, $[C_f]$, and $[K_f]$ denote the mass, damping, and stiffness matrices of the fluid, respectively, and ρ is the fluid density, so that $\rho[R_{int}]^T$ represents the coupling mass matrix at the F-S interface. Consequently, we simultaneously solved the following discretized matrices, for which the aforementioned fluid elements (element type Fluid 221 in ANSYS library) were adopted:

$$\begin{bmatrix} [M_s] & 0 \\ \rho[R_{int}]^T & \left[M_f\right] \end{bmatrix}\begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{Bmatrix} + \begin{bmatrix} [C_s] & 0 \\ 0 & \left[C_f\right] \end{bmatrix}\begin{Bmatrix} \{\dot{u}\} \\ \{\dot{p}\} \end{Bmatrix} + \begin{bmatrix} [K_s] & -[R_{int}] \\ 0 & [K_f] \end{bmatrix}\begin{Bmatrix} \{u\} \\ \{p\} \end{Bmatrix} = \begin{Bmatrix} f_e \\ 0 \end{Bmatrix}$$

2.4. Acceptance criteria

The ASME code [15] categorizes stress components into general primary membrane stress intensity, P_m, local membrane stress

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