Development of a Computer Code for Horizontal Two-Dimensional Seismic Analysis of PWR Core

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SUMMARY

Recently, the use of large artificial seismic waves has led to the possibility of grid corner collisions of PWR fuel assemblies in case of declined excitation with respect to the X, Y axes of a core. In order to evaluate this phenomena, two dimensional energy transmission in a core plane should be considered. For this purpose, the author has developed a new code HORIZON employing a full core model.

Mode-superpositioning method of analysis (hereafter, modal method) was used. To verify adequacy of the use of this method for non-linear problems (colliding vibration), a comparison with the ordinary direct integration method (Houbolt) was performed. The results are as follows:

1. Modal method can follow the colliding vibrational process of fuel assemblies without any conspicuous error, while Houbolt method shows larger numeric damping in both frequency and amplitude aspects.
2. Rayleigh damping (ordinarily combined with the direct integration methods) is far more suppressing vibration than modal damping.

The main concern in such a large scale analysis is how to save calculation time. For this reason, several lower modes were considered to be dominant in whole vibrational behavior and higher modes were omitted, of which adequacy was verified in some test case. The results are as follows:

3. The larger the number of grids becomes, the more we can reduce the number of modes to be considered. Approximately, lower half modes should be considered for an actual assembly model.

The additional study employing a full core model was performed using sinusoidal input. It indicates that there is much less probability of grid corner impacts for inside fuel assemblies than peripheral ones. For the more definite judgement, further study using simulated seismic waves is needed.
1. Introduction

One of the most important items to be ensured in a seismic analysis of a PWR fuel assembly is the integrity of grids. They would collide each other between adjacent assemblies, and must be assured not to be damaged by any postulated seismic waves so much as to prevent the insertion of RCCAs (Rod Cluster Control Assemblies).

Many codes having been developed for this purpose employ a one-row core model, and consider parallel excitation with respect to the row (the same as FUVIAN\textsuperscript{11}).

Adequacy of the one-row model has been explained as follows:

1. As for impacts between a side and a side of grids in X, Y direction (Fig.1), a one-row model is most conservative because all excitation force acts on those direction.

2. As for impacts between a corner and a corner, or a corner and a side of grids in \(\pm 45^\circ\) direction (Fig.1), a one-row model isn't necessarily conservative since it doesn't consider these directions. But fuel assemblies are stuffed in a core (with a gap ~ 1 mm), so a relative displacement between adjacent assemblies would be small. Additionally, a corner occupies much smaller portion of the outer line of grids than a side (Fig.1). Consequently, there will be hardly any chance for these impacts.

The assumption (2) isn't the case for HTGR fuels which have a regular polygon cross section. For this type of fuels, some codes considering two dimensional energy transmission have been already developed and confirmed to be valid by tests.

But recently, the use of large artificial seismic waves has lead to the possibility of grid corner impacts in case of declined excitation with respect to the row. So, a seismic proving test is planned using the test facility for large components of nuclear power plants at Nuclear Power Engineering Test Center. Here, declined excitation will be input to a miniaturized core model constructed from prototype fuel assemblies which have the same specifications as an actual fuel assembly except that Sb-Pb and Pb pellets are loaded instead of UO\textsubscript{2} pellets.

In order to confirm the assumptions (1), (2) analytically, the author has developed a new code HORIZON employing a full core model. HORIZON is a revised code of FUVIAN and considers two dimensional energy transmission in a core plane.

This code is programmed on CRAY–1 which has realized the very fast calculation of this kind of large scale problem.

2. Modeling

2.1 Fuel Assembly and Core

Fig.2 (a) shows a layout of an actual PWR fuel assembly (17x17, 9 Grids). This type has 264 fuel rods, 24 control rod guide thimbles and 1 instrumentation tube in a 17x17 array. Nine grids are allocated to maintain the spacings of rods, thimbles and a tube. Upper and lower Nozzles settle the position of a fuel assembly in a core.

In HORIZON, a fuel assembly is simulated as idealized single beam having 8 impact elements at each grid position (Fig.2 (b)). This model is the same as FUVIAN–2 except for number of degrees of freedom (FUUVIAN–2 model has two degrees of freedom, X, ROTY while HORIZON model has 2nd, X, Y, ROTX, ROTY). In this model, all rods and thimbles are assumed to move together. This method has been pointed out giving appropriate displacement prediction but excessive impact force. In FUVIAN–3 model, relative displacements between a rod and a rod were considered and impact force was predicted successfully,\textsuperscript{12} but this method increases the total number of degrees of freedom. However, displacement is more dominant a factor in colliding vibrational behavior than impact force (impact force can be corrected as long as displacement solution correspond to the results of experiment), and it is necessary to minimize a number of degrees of freedom because HORIZON considers far more assemblies than FUVIAN.

An impact model at grid position is expressed by a combined element of an elastic spring and a frictional element (Fig.2 (c), of which validity has been confirmed experimentally in FUVIAN–2, 3.\textsuperscript{11}\textsuperscript{12})

Fig.3 (a) shows a one-row core model of FUVIAN. In HORIZON, model is extended two dimensionally in a core plane like Fig.3 (b).

In this model, the following assumptions were put:

1. A barrel is simulated as a closed rigid box filled with water.
Upper and Lower ends of beams are fixed in both lateral and axial directions and are elastically supported in rotation.

Water in a barrel is assumed to move together with a barrel.

These assumptions are the same as FUVIAN. Besides these, there are other three assumptions:

The lateral vibrational characteristic of a fuel assembly is uniform in all directions.

There is no torsional motion in a fuel assembly.

Each grid collides only with grids of the nearest neighbor in X, Y direction and of the second nearest neighbor in ±45° direction.

(5) and (6) can be adequate from the same reason as 1. (2), namely fuel assemblies are stuffed so closely.

Excitation wave can be input from both upper and lower ends of beam independently in any direction in a core plane.

A vibrational equation for one fuel assembly is expressed as Eq. (1):

\[
[M] \begin{bmatrix} \ddot{X} \\ \ddot{Y} \end{bmatrix} + [C] \begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix} + [K] \begin{bmatrix} X \\ Y \end{bmatrix} = -M \begin{bmatrix} \ddot{X}_0 \\ \ddot{Y}_0 \end{bmatrix} + \begin{bmatrix} Fx \\ Fy \end{bmatrix} \tag{1}
\]

where,
- \([X] = \text{Translational and rotational displacements vector of a beam relative to a barrel, X, ROTY.}\)
- \([Y] = \text{Y, ROTX}\)
- \([X_0] = \text{Acceleration vector of a barrel in X direction}\)
- \([Y_0] = \text{Acceleration vector of a barrel in Y direction}\)
- \([M], [C], [K] = \text{Mass, damping, stiffness matrix of a beam.}\)
- \([Fx], [Fy] = \text{Combined impact force in X direction at each grid}\)
- \([Fx], [Fy] = \text{Combined impact force in Y direction at each grid}\)

Interaction between fuel assemblies at grid position is contained in \(\{F\}\), so equations in X and Y directions for each assembly can be treated independently except when impact force is calculated.

### 2.2 Calculation of impact force

As explained in assumptions 2.1, (5) and (6), there are eight impact forces for each grid. Fig.4 shows these forces around grid \((i, j)\) as follows:

Impact with the nearest neighbor (X, Y direction)

- \(FX (i, j), FX (i, j+1), FY (i, j), FY (i+1, j)\)

Impact with the second nearest neighbor (±45° direction)

- \(FC1 (i+1, j), FC1 (i, j+1), FC2 (i, j), FC2 (i+1, j+1)\)

Existence and value of forces are determined from relative displacements, gap, relative velocities, impact stiffness and a coefficient of restitution of adjacent grids. Calculation logic is summarized in Table 1 in case of FX (i, j) and FC1 (i+1, j). Other six forces can be derived in the same way and all forces are combined like Eq. (2) and (3), thus we can get impact forces acting on grid \((i, j)\) in X, Y direction \((FX (i, j), FY (i, j))\).

\[
FX (i, j) = FX(i, j) - FX(i, j+1) - FX(i, j+1) - FC2(i+1, j+1)
\]

\[
FY (i, j) = FY(i+1, j) - FY(i, j) + FC2(i+1, j+1) + FC2(i+1, j+1)
\]

All impact forces are determined simultaneously at a certain time step from Table 1 and are fixed as constants in Eq. (1) during this time step.
3. Mathematical Method

The integration of Eq. (1) is mode-superpositioning time history method which was used in FUVIAN and of which validity has been confirmed experimentally. There are other ordinary direct integration methods for nonlinear transient dynamic analyses such as Newmark and Houbolt.

In spite of impact forces which were considered as external forces to a beam, vibrational behavior of fuel assemblies can be treated by modal method successfully. Besides that, modal method has some merits as follows:

1. [M] and [K] can be determined directly from the vibrational characteristics of a fuel assembly (as natural frequencies and modal shapes).
2. [C] is assumed to be diagonal and has variables as many as the number of modes to be considered. (In the direct method, Rayleigh damping is generally used and it has only two variables.)
3. Calculation time can be saved by omitting higher modes which can be ignored.

Item (3) is especially favorable in case of large scale analyses. In the following sections, a comparison with direct integration method and the validity check of mode reduction will be described.

3.1 Comparison with direct integration method (Damping = 0.0)

In order to verify adequacy of the use of modal method, a comparison with Houbolt (3rd order) integration method was performed by analyzing the same problem. Here, we have chosen this ordinary direct integration option installed in ANSYS (CRAY Version), a general purpose FEM code. Fig.5 shows the outline of problems where two models, one-grid and three-grids were employed. Constant lateral acceleration of IG was input to both models, and colliding vibration was followed by two methods in the same integration time-step. Fig.6 (a) and (b) show the results of impact force time history occurred in the middle grid of one-grid and three-grids models, respectively. Fairly good accordance is observed in Fig.6 (a), but poor in Fig.6 (b) because in three-grids model, the middle grid behavior is affected by impacts occurring at other two grids step by step. As for Fig.6 (a), the smaller we make time-step of ANSYS, the better accordance with HORIZON becomes. Time-step was small enough to resolve the highest frequency of interest yet Houbolt shows larger numeric damping in both frequency and amplitude aspects.

Table 2 shows the comparison of the first impact force of the middle grid which wasn’t affected by other impacts in a three-grids model. Two methods coincide well enough.

As a result, it can be said that modal method can follow colliding vibrational behavior of assemblies without any conspicuous error, while Houbolt shows larger numeric error in the same time-step condition. That is, modal method can save more solution time and cost.

3.2 Comparison with direct integration method (Damping ≠ 0.0)

In the direct integration method, the viscous damping matrix [C] may be generally given by Eq.(4).

\[
[C] = \alpha [M] + \beta [K]
\]

(4)

This form is known as Rayleigh damping. In modal method, damping is given for each mode as a modal damping ratio \( HS(i) \) \((0.0 < HS(i) < 1.0)\). The relation of two types is given as Eq.(5).

\[
HS(i) = \frac{1}{2} \left( \frac{\alpha}{\omega_i} + \beta \omega_i \right)
\]

(5)

where \( \omega_i \) = angular frequency of \( i \)-th mode.

Therefore, \( \alpha \) and \( \beta \) can be determined from the first two modal damping ratios. But Eq.(5) also indicates Rayleigh damping gives larger effective modal damping ratios in higher modes because of large value of \( \omega_i \). To verify this, another comparison of HORIZON with ANSYS was performed using a three-grids model under existence of damping. HS(1), HS(2) and \( \alpha, \beta \) are set up as follows from Eq.(5):

\[
\begin{align*}
\{HS(1) = 0.20\} & \iff \{\alpha = 5.21\} \\
\{HS(2) = 0.15\} & \iff \{\beta = 3.54 \times 10^{-3}\}
\end{align*}
\]

— 350 —
Other inputs and time-step condition are the same as 3.1. Fig.7 shows the time-history of impact force of the middle grid (correspond to Fig.6 (b)) and indicates Rayleigh damping is far more surprising vibration. That is modal damping is more conservative. In vibration tests of a fuel assembly, natural frequencies, modal shapes and damping ratios are measured up to 5 ~ 7 modes, so modal damping can fit the results more neatly while Rayleigh damping can fit only two modes.

3.3 Reduction of modes

Modal method was observed to be more efficient in saving calculation time, but not enough in case of employing a full core model where we should consider nearly 200 fuel assemblies. So another way to advance cost performance of the code should be discussed.

Here, influence of mode reduction on impact force was evaluated in case of a three-grid model (Damping = 0.0).

The model has 8 degrees of freedom and the highest mode to be considered was varied from 4-th to 8-th. Fig.8 shows the relation of the first impact force of the middle grid vs. the highest mode number, and indicates influence of mode reduction is small and conservative for the value of impact force.

An actual fuel assembly model has more degrees of freedom, the influence will be smaller and about first half modes would be enough to get good approximation.

4. Test Run for an Actual 4 Loop Core

The additional study employing a full 4 loop core model (containing 193 assemblies) was performed. Inputs of an actual 17x17 9 grids fuel assembly was used and sinusoidal acceleration (amplitude=√2G, period=0.1 sec.) was input in 45° direction with respect to the row.

Fig.9 (a) shows a displacement vector diagram of 5-th grid (relative to a barrel) at 0.1 sec., and Fig.9 (b) shows a X→X' cross section at the same time. They indicate that the relative displacements of adjacent assemblies are small and the probability of grid corner impacts for assemblies located inside area of a core is much smaller than peripheral ones.

5. Conclusion

(1) Mode-superposition method was compared with Houbolt integration method and was confirmed as an efficient and valid way to solve the colliding vibration problem of PWR fuel assemblies.

(2) Modal method gives more accurate answer at the same time-step.

(3) The modal damping is more conservative than Rayleigh damping and has more variables which can be fit the results of vibration tests neatly.

(4) The highest mode to be considered can be reduced to some extent without any conspicuous error, thus calculation time can be saved.

From now on, some analyses using simulated seismic waves will be performed, and general features of two dimensional colliding vibration of a PWR core will be revealed.

6. References


Fig. 1  PWR Fuel Grid (17x17)

(a) Actual PWR Fuel Assembly (17x17, 9 Grids)  (b) Fuel Assembly Analytical Model

Fig. 2  Modeling of Fuel Assembly
Fig. 3  Calculation Model for FUVIAN and HORIZON

Table 1  Calculation Logic of Impact Force [FX(i, j), FCI(i+1, j)]

<table>
<thead>
<tr>
<th>Event/Condition</th>
<th>R.Force Velocity</th>
<th>Impact Force</th>
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</thead>
<tbody>
<tr>
<td>a) v &lt; 100 m/s</td>
<td>ALL CASES</td>
<td>FX(i, j) = 0</td>
</tr>
<tr>
<td>b) v &lt; 150 m/s</td>
<td>ALL CASES</td>
<td>FX(i, j) = 0</td>
</tr>
<tr>
<td>c) v &lt; 200 m/s</td>
<td>0 &lt; v &lt; 150 m/s</td>
<td>FX(i, j) = 0</td>
</tr>
<tr>
<td>d) v &gt; 200 m/s</td>
<td>0 &lt; v &lt; 150 m/s</td>
<td>FX(i, j) = 0</td>
</tr>
</tbody>
</table>

Fig. 4  Interaction around Grid (i, j)
Table 2  Comparison of 1st Max. Impact Force of Middle Grid

<table>
<thead>
<tr>
<th>Model</th>
<th>Max. Impact Force</th>
<th>Impulse Time (s)</th>
<th>Time for Max.</th>
<th>Duration Time</th>
<th>Thrust</th>
<th>Max. Thrust</th>
</tr>
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<tr>
<td>HORIZON</td>
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<tr>
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<td>0.0080</td>
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</tbody>
</table>

Fig. 5  Test Case for Comparison with ANSYS

(a)  One Grid Model  (b)  Three Grids Model

Fig. 6  Comparison of Middle Grid Impact Force Time History with ANSYS (No Damping)

(a)  One Grid Model  (b)  Three Grids Model

Fig. 7  Comparison of Middle Grid Impact Force Time History with ANSYS (Damping HS (1) = 0.2, HS (2) = 0.15)
Fig. 8  Max. Impact Force vs. No. of Modes to be considered. (Three Grids Model)

(a) Displacement Vector Diagram
(b) X-\(X'\) Cross Section

Fig. 9  Test Case for Actual 4 Loop Core Model