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A vibration model for the primary circuit of VVER-440 type reactors based on finite elements

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ABSTRACT: A theoretical model describing the mechanical vibrations of nuclear reactors can substantially support the early detection of mechanical component faults. The presented finite-element-model comprises the whole primary circuit of the VVER-440 including the steam generators and considers also the fluid-structure-interaction in the downcomer between reactor pressure vessel (RPV) and core barrel (CB). All eigenfrequencies up to 30 Hz and the corresponding mode shapes were calculated. It is shown that the fluid-structure-interaction strongly influences those modes exhibiting a relative displacement between RPV and CB. Moreover, by means of the model the shift of eigenfrequencies due to the degradation or to the failure of internal clamping and spring elements was investigated. By comparing the frequency spectra of the normal and of the faulty structure, it could be shown that a recognition of such degradations and failures even inside the RPV is possible by pure excore vibration measurements.

1 INTRODUCTION

The reason why a theoretical vibration model is urgently needed for VVER-reactors lies in the occurrence of flow induced vibrations of RPV internals. Control element and core barrel (CB) vibrations are known to have occurred in the former East German power plant Greifswald (Altstadt and Weiß, 1993) and also in other East European installations. In some cases these vibrations were accompanied by serious and safety relevant damages. During CB motion at the former Greifswald power station relative amplitudes of up to 5 mm between CB and RPV were found (Liewers *et al.*, 1987). These high amplitudes were a consequence of the plastic deformation of the so called spring pipe segments which are to transmit the clamping force of the reactor vessel top upon the upper flange of the CB. The unsufficient fixing of the CB resulted among others in damaged guide lugs. Due to the fact that CB vibration was accompanied by a quasistatic axial shift of the CB inside the vessel a considerable part of the guide lugs was eroded. Up to 18 mm of material were worn off. Especially the quasistatic axial shift is a safety relevant incident.

To detect the faults in an early stage a theoretical vibration model of the whole primary circuit is needed, permitting

- the description of the normal vibrations of the components, especially to assign the measured vibration frequencies in neutron noise, pressure fluctuations or mechanical displacements to vibration modes of the whole coupled mechanical system

- the determination of physical limits for frequency shifts and amplitude changes as alarm thresholds for on-line vibration monitoring
- the assessment of mechanical loads connected with the failure of a certain component.

2 THE FINITE ELEMENT VIBRATION MODEL

The model comprises the whole primary circuit, including steam generators, loops, coolant pumps, main isolating valves and certainly the reactor pressure vessel and its internals. It was developed using the finite-element-code ANSYS® on a Hewlett-Packard workstation platform. The model has a modular structure, so that various operational and assembling states can easily be considered.

The experimental experience exhibited the frequency range 0-30 Hz to be mainly interesting for the mechanical integrity of the whole system. Thus it is sufficient to assemble the model from 1D pipe elements.

Figure 1 shows a topology scheme of the finite-element-model for the RPV and all the internals: CB, CB support skirt, core basket, upper core structure. The model of the RPV head considers the upper callotte, the control-element standpipe frame and the standpipes themselfes. Each of the 71 nodes connecting two elastic pipe elements has 6 degrees of freedom. Different reactor components are mutually connected by stiffness matrices (12 by 12) which e.g. represent the CB guide lugs (nodes 20 and 39), or the ring foundation (nodes 43 and 71), or the spring pipe segments between the upper flange of the CB and the RPV head (nodes 31 and 50). The parameters of those stiffness matrices were estimated in separate 3D calculations. Extreme stiff connections are modelled by constraints.

A special fluid-structure-element (FSE) was developed to consider the reaction forces of the fluid in the downcomer upon the moving CB and the RPV (Grunwald and Altstadt 1994). It was derived from an approximated analytical 2D-solution of the coupled system of 3D fluid equations and the structural equations of motions. The approximation is specially taylored for narrow annular gaps and uses a linear approach for the radial

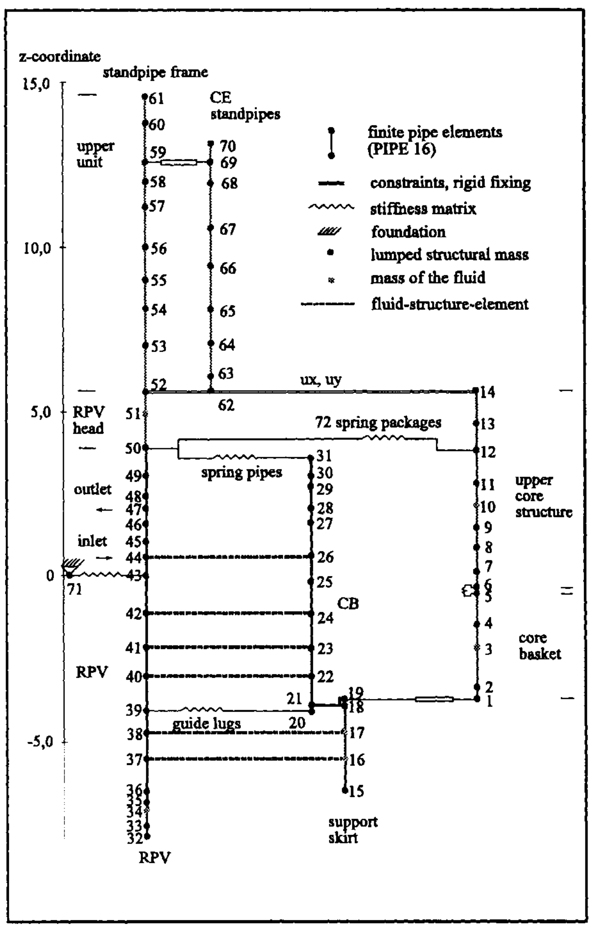

Figure 1. Topology of the finite-element-model for the RPV and ist internals

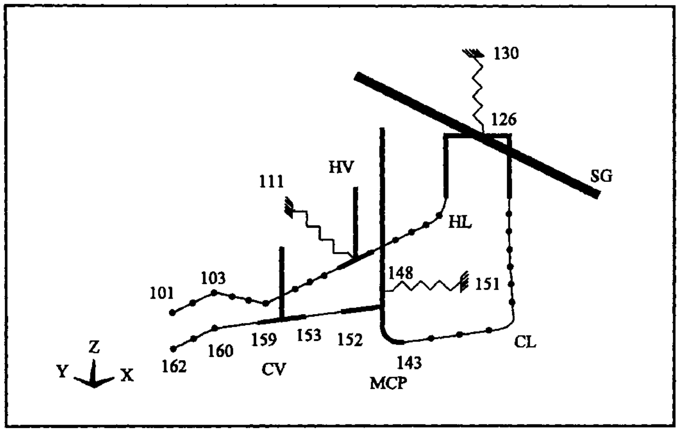

Figure 2. Topology of the finite-element-model for the coolant loop.

component of the fluid velocity. The potential solution for the velocity field is applied to calculate the pressure distribution in dependence on the structural motion. The resulting forces and momenta which act on the structure can in turn be obtained from the pressure field. A FSE has 2 nodes with 8 degrees of freedom and is located between the RPV and the CB nodes (Fig. 1).

In analogy to the RPV with internals, the loop model consists of pipe elements, pipe tee and pipe elbow elements, structural mass elements (which are formulated as 6 by 6 inertia matrices), stiffness matrices and constraints (see Fig. 2 for the topology). The loop model includes the steam generator (SG), the hot leg (HL), the cold leg (CL), the main coolant pump (MCP), the hot isolating valve (HV) and the cold isolating valve (CV). Special attention was put on the SG support and on the connection of the SG with the secondary circuit, both modelled by a separately calculated stiffness matrix between the SG center node and the ground. Neglecting these couplings, one would drastically underestimate especially the first eigenfrequency of the loop. Also the bearings for the MCP and the HV are represented by stiffness matrices. The parameters of the loop model could be adjusted using results from modal analysis experiments performed at coolant loops in the Greifswald NPP and in the Dukovany NPP, Czech, as well.

The vibration behaviour of the RPV and the six coolant loops cannot strictly be separated. Neither the RPV is a fix clamping for the loops nor is the loop inertia neglectable for RPV motions. The total mass of the RPV with all internals is about 600 t and that of one loop is about 290 t. So in general it is to be expected that the mode shapes will be coupled. The connection of RPV and coolant loops is realized by rigid area constraint equations between RPV (node 47, Fig. 1) and the first (hot) nodes of all six loops and between RPV (node 44) and the last (cold) nodes of all six loops respectivly.

3 EIGENFREQUENCIES AND MODE SHAPES

The coupling of all six coolant loops with the RPV results in a great number of eigenfrequencies of the complex system due to the coupling of different degrees of freedom. The whole model then consists of about 450 finite elements with about 2300 active degrees of freedom. In the frequency range up to 30 Hz more than 100 eigenfrequencies are obtained. Many of them are close together but exhibit different mode shapes. Most of the mode shapes are characterized by various loop displacements in many different phase relations with the RPV being almost in rest position. Some mode shapes only exhibit large displacements of the loops and of the RPV as well. For example Fig. 3 shows the lowest mode shape at 3.5 Hz which has significant displacements of the RPV and of the internals.

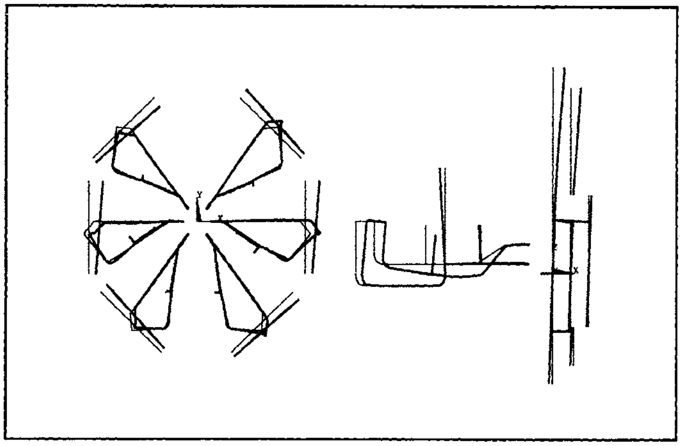

Figure 3. Coupled mode of the primary circuit at 3.5 Hz.

For the investigation of the RPV and its internals it is suitable to reduce the model size by means of substructure technique to exclude such modes which differ from each other only in their phase relations of the loops. Within the reduced model the six coolant loops are represented by a substructure of 12 degrees of freedom. Table 1 gives an overview of the modes of the RPV and its internals in the frequency range up to 30 Hz. The longitutional modes (z-direction) excepted, generally each mode shape occurs twice (x-z-plane and y-z-plane).

Table 1. Eigenfrequencies and mode shapes of the RPV and ist internals

No.	Frequency / Hz		Verbal description
	FSE	no FSE	
1/2	3.5 3.65	3.5 3.65	in-phase pendulum motion of RPV, CB and CB-internals; 1. beam mode of the upper unit in-phase; loops in-phase
3/4	4.6 4.7	4.6 4.7	in-phase pendulum motion of RPV, CB and CB-internals; 1. beam mode of the upper unit in-phase; loops in anti-phase
5/6	8.2 8.3	8.2 8.3	small in-phase pendulum motion of RPV, CB and CB-internals; 1. beam mode of the upper unit in anti-phase; large cold leg displacements
7/8	10.3 10.5	10.5 10.7	in-phase pendulum motion of RPV, CB and CB-internals; 1. beam mode of the upper unit in anti-phase; hot leg displacements
9/10	**13.75** **13.75**	**26.0** **26.0**	anti-phase pendulum motion of the RPV with respect to all RPV-internals, elastic deformation of CB and CB-internals, 2. modes of standpipe frame and standpipes
11/12	16.0 16.0	15.8 15.8	2. beam mode of the CE standpipes
13	18.6	18.6	z-vibration of all internals, small z-amplitude of the RPV, in-phase pendulum motion of RPV, CB and CB-internals, 2. beam mode of the upper unit
14/15	**23.6** **23.6**	not existing	anti-phase beam modes of RPV (lower part) and CB; elastic deformation of CB-internals

To demonstrate how important the consideration of fluid-structure-interaction (FSIA) is, the eigenfrequencies are listed for two cases: with and without FSEs. The first eigenfrequency connected with a significant relative displacement between RPV and CB was calculated to be at 26 Hz without FSIA. Consideration of FSIA shifts that

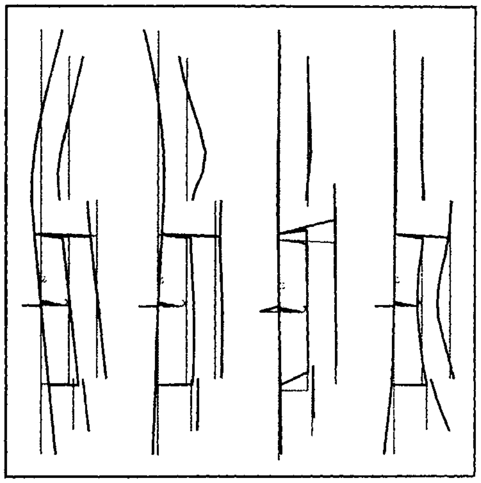

Figure 4. Modes of the RPV and internals a) 10.5 b) 13.75 c) 18.6 d) 23.7 Hz

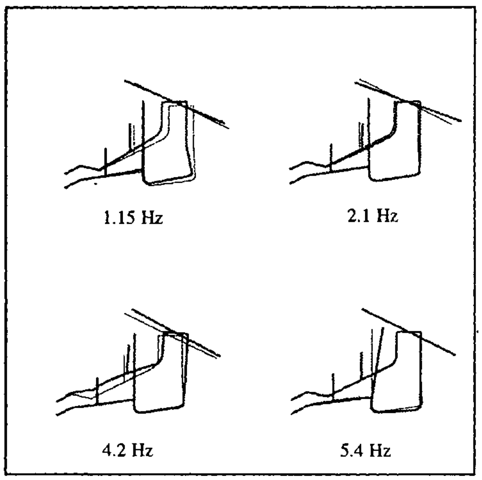

Figure 5. Modes of a coolant loop at 1.15 Hz, 2.1 Hz, 4.2 Hz and 5.4 Hz

eingenfrequency down to 13.75 Hz. The eigenmodes 9, 10, 13, 14 and 15 (Fig. 4b-d) are particularly important for the detection of possible degradations of the internal clamping elements like the guide lugs and the spring pipe segments. Mode 13 at 18 Hz is above all a z-vibration of all the internals while the vessel amplitude itself is rather small. Also the relative displacements connected with the pendulum component of this mode can be neglected. The modes 9/10 and 14/15 are related to anti-phase motions of the RPV with respect to all internal components. Furthermore these modes are characterized by elastic deformations of the internals.

The modes of a single loop were calculated with the loop-FE-model assuming the coupling nodes to the RPV (Fig. 2, nodes 101 and 162) to be in rest position. The finite-element-model of the loops has already been adjusted with modal analysis experiments performed as well in the former Greifswald power station as in the Dukovany plant in Czech. In the frequency range up to 30 Hz about 20 eigenfrequencies were found. Figure 5 e.g. depicts 4 mode shapes for an empty loop (without coolant) calculated with the model. The mode shape connected with the lowest eigenfrequency at 1.15 Hz is a pure tangential displacement of the whole loop with respect to the RPV. The mode shape at 2.1 Hz essentially corresponds to a torsional vibration of the steam generator around the collector nozzles. At 4.2 Hz the mode shape is a radial steam generator vibration with respect to the RPV. The eigenfrequency centred at 5.4 Hz is mainly determined by the beam mode vibration of the whole coolant pump in the CL plane.

4 SIMULATION OF MECHANICAL FAILURES

To reveal which modes are best suited to detect degradations of the safety relevant internal components, sensitivity studies were performed by varying certain parameters of the finite-element-model. To recognize the basic tendencies in frequency and amplitude shifts when RPV clamping elements degrade, it is sufficient to use the reduced model with the approximate consideration of the loops. Figure 6 indicates how the forced vibration spectrum of RPV bottom displacement in y direction would change in case of two broken guide lugs (in total there are 8 guide lugs). As expected, above all the eigenfrequencies

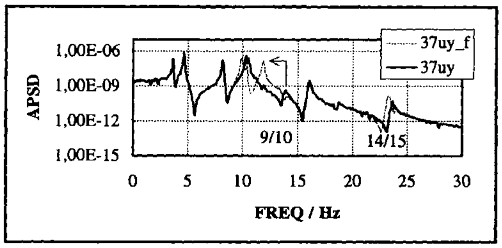

Figure 6. Change of spectrum of RPV bottom displacement (node 37, uy). Fracture of two guide lugs.

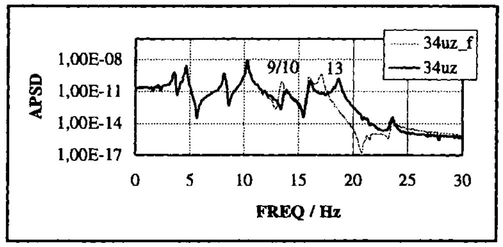

Figure 7. Change of spectrum of RPV bottom displacement (node 34, uz). Failure of two spring pipe segments.

pertaining to anti-phase pendulum motion do respond to a loss of guide lug stiffness.There is almost no effect in the other eigenfrequencies. Figure 7 shows the change of the spectra of the RPV bottom displacement in z direction in case of the failure of two spring pipes (in total there are 6 spring pipes). A significant effect is observed in a down shift of the eigenfrequency of mode 13 at about 18 Hz and a rather small shift of the modes 9/10.

5 CONCLUSIONS

The results obtained with the vibration model for a VVER-440 reactor hint at the opportunity to establish a monitoring procedure that is capable of distinguishing between guide lug and spring pipe failures, because these two types of failures are mapped in different and separated frequency regions. Moreover it seems that it is possible to detect these failures by performing excore vibration measurements, what is important since incore vibration measurements are difficultly to be implemented over longer time periods. By measuring the displacements for x- and y-direction even the localization of the failed spring pipe or guide lug might become achievable.

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