Seismic Analysis of a Nonlinear Structure with Gaps: Comparison Between Experimental and Computational Results

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Abstract

This paper presents seismic shaking table tests of a scaled vessel supported by stops with gaps and the corresponding computations. It is shown that simple nonlinear beam models give satisfactory results for engineering purposes.

1. INTRODUCTION

Stops with gaps are currently used to support the pressure vessels of nuclear power units. This design is simple and thus low in cost, efficient for high energy loadings such as earthquakes, reliable without significant maintenance, and it allows free thermal expansions if the gaps are large enough.

However, such a design involves two sorts of problems:

- Adjustment of the gaps: The gap sizes must be strictly controlled, to avoid risks of jamming (gaps too small) or inefficiency (gaps too large). This implies a set of operations such as on-site measurement, shim fabrication, periodic checks, etc.
- Structural analysis: The nonlinear behavior of stops complicates the analysis work, especially for the seismic analyses.

Regarding this last point, a way has now been found to perform nonlinear time history (TH) calculations, even for engineering purposes. In this case, the finite element models are rather simple, although they are nonlinear: the components are modeled by beam elements and the stops by nonlinear springs.

In order to study the validity and the limits of such nonlinear beam models, Framatome in cooperation with the CEA and EDF, has initiated an R&D program based on simple dynamic experiments on a mass component supported by stops. The experiments comprise two parts:

- Elementary tests of rebound of the component on its stops. These tests and also the numerical simulations have been reported in a previous SMIRT paper [1];
- seismic shaking table tests, which are reported here.

Ref.[1] L.BORSOI, P.BULAND, P.LABBE: "Impact test of components"; SMIRT 9, session F, Lausanne 1987.

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2. EXPERIMENTAL TESTS

2.1 Vessel Mockup (See Figure 1)

The mockup is a very rough approximation of a French PWR unit steam generator (SG), 1/6 scaled. It is mainly formed by a common steel cylinder, 3 meters long, 0.65 m diameter, and 0.014 m thick, which models the SG external shell. This unpressurized tank is closed at one end by a square plate that represents the solid tube sheet and at the other one by a simple disk. The SG internal mass is represented by additional lead masses, screwed on to the mockup shell. The total mass of the mockup is 1.75 tons.

2.2 Installation on the Shaking Table (See Figure 2)

The mockup is horizontally placed and supported by four vertical cables. Four stops fixed on the shaking table restrain the horizontal motion of the vessel perpendicularly to its axis. These stops work in compression, two by two. The first one limits positive displacements and the second one negative displacements. Two stops enclose the rigid square plate (they are called the "bottom" stops because of the SG vertical position in reality) and the other two enclose the vessel about at its middle (they are called the middle stops).

The contact stiffness when the vessel impacts a bottom stop is essentially provided by the stiffness of the stop itself (made softer by the bending of a plate). On the contrary, the contact stiffness at a middle stop is mainly due to the shell deformation of the vessel.

The shaking excitation is horizontal and monodirectional in the stops direction. The total embarked mass is 3 tons. It must be noted that the mockup between its stops (no contact) is not at all connected to the table.

2.3 Scaling Process

The scaling process is the conventional one used in dynamic analysis which keeps the velocity unchanged. The different scaling factors are indicated in Table 1.

2.4 Shaking Table Tests

The tests were carried out at the CEA facility in Saclay on the VESUVE shaking table. Parametric experiments were made on:

- the nature of the excitation :
  . free-field seismic acceleration corresponding to the NRC spectrum,
  . floor seismic acceleration (the previous one filtered at 42 hertz),
  . stationary "white noise" excitations of long duration;
- the amplitude of the excitation;
- the gap sizes: J = 0.0 millimeters, J = 0.50 mm , J = 1.00 mm, and J = 3.00 mm. (these values of J indicate the cumulated gap of two opposite stops);
- the configuration of the mockup. Some experiments, not reported here, take into account the presence of a branched line that represents the hot leg.

The impact forces on each stop, the accelerations and the displacements at several locations are recorded (see Figure 2).

Ref.[2] L.BORSOI, J.P.THOMAS : "Influence of gaps on seismic behaviour of the Primary System of a PWR"; SMIRT 7, paper F4/7, Chicago, 1983.

3. TEST ANALYSIS

3.1 Finite Element Model (See Figure 3)

This model is similar to current models developed for production analyses. The vessel is modeled by conventional beam elements - the rigidity matrix includes bending, torsion, and shear effects and the mass matrix is consistent. Concentrated translational and rotational masses are added for modeling lead masses and vessel ends. Each stop is modeled by a single nonlinear impact element constituted by a linear spring K in line with a gap J associated in parallel with a viscous dashpot C. The beam line comprises eight nodes. Due to the modeling and the direction of the excitation, only translational degrees of freedom (DOF) UY and rotational DOF RZ are active.

3.2 Computations

Framatome structural codes are used for the computations. The time history (TH) responses are calculated by a nonlinear modal superposition method. The first 10 modes of the vessel without stops stiffnesses are superimposed and the basic time step is 0.33 millisecond. Parametric studies have shown that these parameters are fully sufficient to ensure correct convergence. The duration of a scaled seismic excitation is about 4 seconds.

3.3 Stiffness K_1 and K_2 of the Stops

The major difficulty of modeling consists in determining the contact stiffnesses K_1 (bottom stops) and K_2 (middle stops). Additional spurious compliances due to the mounting on the shaking table, such as the load cell, the stops supports, etc., make the rigidity lower than designed.

It was hoped that a preliminary sine wave sweep test of the mockup with zero gaps - the vessel was just in contact on the stops - would allow identifying K_1 and K_2. In fact, the structure is still so nonlinear in this configuration that no resonance could be observed. Clear resonances were obtained only by prestressing the stops on the vessel. But this creates additional rotational constraints RZ of the mockup, which made inefficient simple analyses for determining K_1 and K_2.

Therefore the rigidities K_1 and K_2 were derived from TH experimental impact forces F(t). A convincing method consisted in plotting for each impact the momentum variation $\int F(t).dt$ versus the impact maximum F_{MAX}. The slope of the straight line deduced from a linear regression, in association with TH computations, permitted estimating the rigidities K_1 and K_2 (see Figure 4).

3.4 Dashpots C_1 and C_2

The dashpots values C, which give the damping force $F_D = C.\dot{X}$, are deduced from an analogy to a single DOF: $C = 2.\xi.K/\omega$, with K the contact stiffness, ω the pulsation related to K and the impacting mass, and ξ the reduced damping.

Verifications of C based on energy considerations are made afterwards from the TH results.

4. COMPARISONS

4.1 Simplification of the Impact Force Shapes

As pointed out for rebounds tests [1], the real impact force shapes depend on stops locations. The bottom stops provide half sine wave shapes whereas the middle stops give more complicated shapes, taking into account shell modes of the vessel. The beam model can give only half sine wave shapes, since no vessel shell modes are represented.

4.2 Chaotic Motion

The seismic motion of such a nonlinear structure is chaotic: a small perturbation can drastically modify the sequence of successive impacts. Consequently, the approximations of the model make impossible a complete TH superposition of experimental and computational responses. Among these inaccuracies are the exact gap sizes - gaps are very small and thus very difficult to strictly control - the unknown initial distribution of the cumulated gap between the two opposite stops, the behavior of impact elements that can render only normal localized forces F_y by ignoring all local rotation and friction that can occur, and the approximate knowledge of parameters K_1 and K_2 as well as C_1 and C_2.

Nevertheless, when the driven response of the structure becomes more preponderant than the free response, some TH convergence can be established. But actual TH results are not mandatory for engineering purposes.

4.3 Impact Forces

As an example, Table 2 gathers different results related to Test 13: NRC seismic excitation with gaps J = 0.50 mm. Computations a,b, and c take into account adjusted values for K_1 and K_2 (cf.§ 3.3) and are parametric analyses concerning damping. Computations d and e correspond to preliminary K_1 and K_2 values. The best results are obtained from computation b, with 6% damping. But generally the different magnitudes are well rendered even with rough approximations of K_1 and K_2.

4.4 In-structure Response Spectra

Figures 5 and 6 show in-structure response spectra calculated with 2 % damping and related to the previous test 13, computation b. The convergence is very accurate for the spectrum at the "top" (Fig.5 - node 80), for which shell modes are not present. The overall shape of this spectrum has been discussed in [2]. Evidently the convergence is less correct for the spectrum at the middle (Fig.6 - node 40) which involves shell modes (especially a peak at 100 Hz- see [1]). Nevertheless, the order of magnitude is respected.

5. CONCLUSION

From an engineering point of view, simple nonlinear beam models are satisfactory for simulating the seismic behavior of vessels restrained by stops: the impact forces are well rendered and the order of magnitude of seismic excitations that are transferred to the substructures is correct.

The key point lies in the determination of the contact stiffnesses K, which must be used in the model. But in the present study, a certain imprecision about K was quite acceptable.

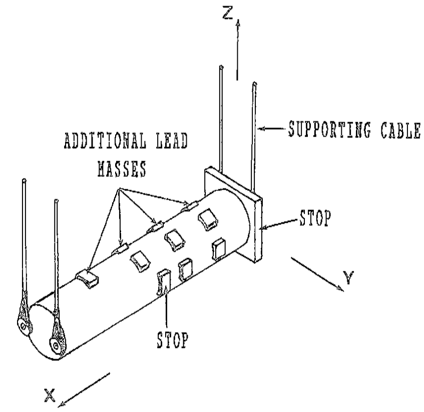

Fig.1: Vessel Mockup

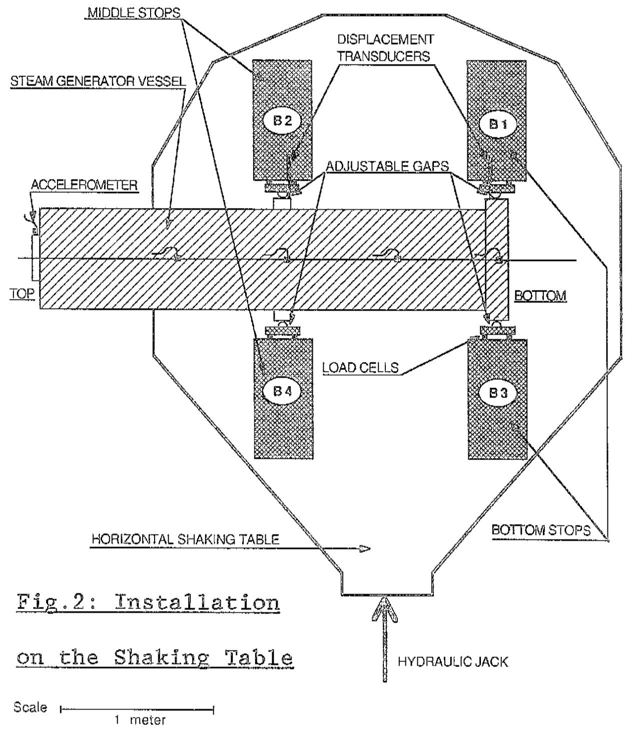

Fig.2: Installation on the Shaking Table

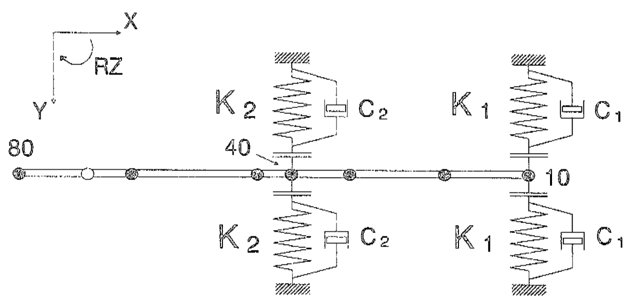

Fig.3: Finite Element Model

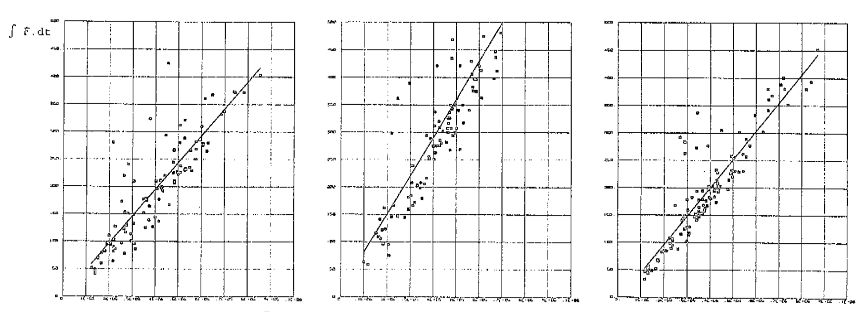

Fig.4: Adjustment of the Contact Stiffness

Quantity	dim.	Scal.fact.
length	L	$\lambda = 1/6$
area	L^2	$\lambda^2 = 1/36$
volume	L^3	$\lambda^3 = 1/216$
density	$M.L^{-3}$	$\lambda^0 = 1.$
mass	M	$\lambda^3 = 1/216$
velocity	$L.T^{-1}$	$\lambda^0 = 1.$
acceleration	$L.T^{-2}$	$\lambda^{-1} = 6.$
load	$M.L.T^{-2}$	$\lambda^2 = 1/36$
stress	$ML^{-1}T^{-2}$	$\lambda^0 = 1.$
time	T	$\lambda^1 = 1/6$
frequency	T^{-1}	$\lambda^{-1} = 6.$
rigidity	$M.T^{-2}$	$\lambda^1 = 1/6$

Table 1: Scaling Factors

		TEST 13	COMPUTATIONS a	b	c	d	e
NUMERICAL DATA	K_1		16.	16.	16.	16.	11.
	K_2		6.8	6.8	6.8	10.	11.
	ξ target %		4.0	6.0	8.0	6.0	6.0
	ξ_1 verif.%		4.0	6.0	8.0	7.0	7.0
	ξ_2 verif.%		5.0	7.0	9.0	6.5	5.5
BOTTOM STOPS	no.of peaks	98	111	106	102	109	106
	F max.	8.2	10.	8.1	7.2	7.9	7.4
	F mean	3.9	4.5	3.9	3.6	3.7	3.6
	$\int$ F.dt max	410.	520.	430.	360.	400.	440.
	$\int$ F.dt mean	185.	230.	195.	175.	180.	210.
MIDDLE STOPS	no.of peaks	60	68	61	61	66	76
	F max.	8.0	9.3	7.2	6.0	8.1	9.4
	F mean	3.9	4.6	3.9	3.4	4.1	4.3
	$\int$ F.dt max	800.	940.	890.	670.	840.	920.
	$\int$ F.dt mean	410.	540.	450.	380.	400.	390.

Rigidity K_1 and K_2 in 10^7 N/m
Force F in 10^4 N
Momentum variation $\int$F.dt in 10^1 N.s

Table 2: Comparison of Impact Values

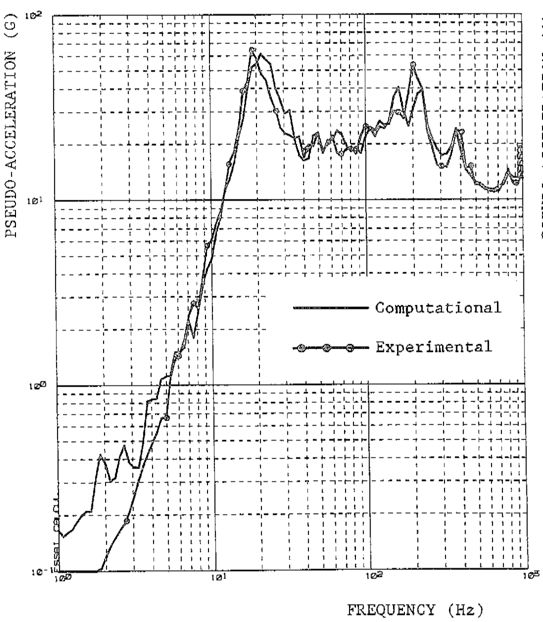

Fig.5: Top Response Spectrum

Node 80 - 2% Damping - Test 13

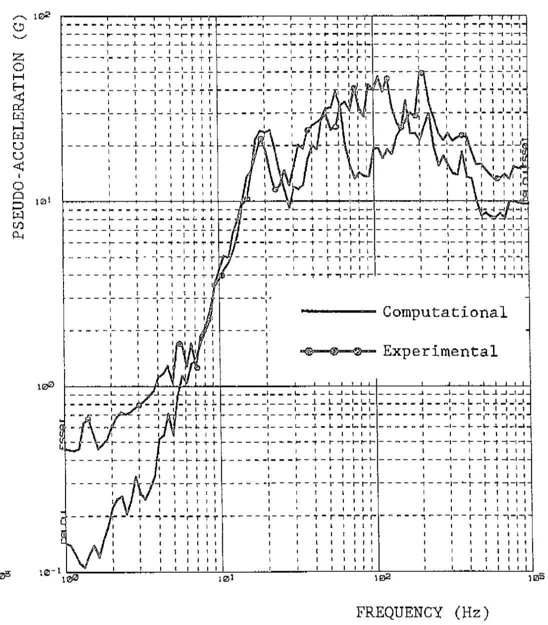

Fig.6: Middle Response Spectrum

Node 40 - 2% Damping - Test 13