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EXPERIMENTAL STUDY OF TUBE/SUPPORT IMPACT FORCES IN MULTI-SPAN PWR STEAM GENERATOR TUBES.

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ABSTRACT

The vibro-impact response of a straight part of a steam generator tube is investigated experimentally and using numerical simulation with the aim to relate tube overall dynamics with excitation and tube-support clearance.

Configuration studied here corresponds to the tube being excited in only one direction at its first resonance presenting an antinode of vibration at the impacted support. Tests show namely that midspan displacement of tube is almost proportional to excitation level and clearance. Impact forces averaged over a cycle of vibration are almost proportional to excitation and poorly dependent on clearance. Results of numerical simulation are in fairly good agreement with test results.

1. INTRODUCTION

Currently there is increased interest in predicting fretting wear of heat exchanger tubes caused by flow induced vibration. As recently discussed thoroughfully by Connors [1] pure fretting taking place when tube is kept in permanent contact with its supports may be evaluated using the Archard's equation [2,3] :

$$V(t) = k\,F_N\,L(t) \tag{1}$$

where
$V(t)$ is the volume of wear at time t,
k is the wear coefficient,
F_N is the normal contact force between tube and support,
$L(t)$ is the total sliding distance.

However very often small clearances exist between tube and support, and intermittent tube-support contacts may occur even for the small vibration induced at normal operating conditions by flow turbulence. This produces an impact-fretting wear process more severe than pure fretting. Connors [1] pointed out the practical interest in extending the Archard's equation to impact-fretting as this would provide at least "a common reference point for comparing wear data from different types of tests". For such a purpose it is clearly necessary to replace the contact parameters in Eq. (1) by some dynamical parameters describing conveniently the highly non linear tube-support interaction. At this respect quite encouraging results have been produced by Ko [4, 5, 6] who has been successfull in correlating impact forces and wear.

In this paper relation between impact forces and the non linear tube motion is investigated on a preliminary basis. It is based on an experimental study of the vibro-impact response of a straight portion of a PWR steam generator tube with tube-baffle interaction at only one support. Results are then compared to numerical simulations based on the method already proposed by Rogers and Pick [7]. Tests to be discussed here are restricted to harmonic excitation of the tube in only one direction at the frequency of its first flexural mode which has an antinode of vibration at the interacting support. Several support holes with radial clearances from 0.25 to 1 mm have been tested. Excitation has been varied from 0.1 to about 10N rms. Midspan peak displacement of the tube was ranging from 0.1 to about 1 mm. A few tests were made in water, the other in air.

Though no great accuracy may be claimed in our measurements data obtained are sufficient for pointing out some simple correlations between tube dynamics (midspan displacement and impact forces) and either excitation level or tube-support clearance. Furthermore numerical and experimental results are in satisfactory agreement with each other.

2. TEST RIG AND INSTRUMENTATION

Test Rig

The test rig is schematically shown in Fig. 1. Detailed geometry of it is reported in Table 1.

Tube			
Diameter	D = 19.05 mm	Young modulus	E = 2 10^{11} Pa
Thickness	e = 1.09 mm	Specific mass	ρ = 8,200 $Kg.m^{-3}$
Length	L = 3,400 mm	Mass per unit length	
Span	l = 1,070 mm		
Material	Inconel 600	(in air)	ms= 0.492 kg/m
Supports			
I shaped beam		Rubber sheet	
Length	: 5 m	Thickness :	7 cm (inserted between I beam and concrete block)
Mass	: ≅ 10^3 kg		
First resonance:	180 Hz		
Concrete block			3 cm (stretched out on the floor)
Mass	: 2 10^3 kg		

Table 1. Geometry

A straight portion of a PWR steam generator tube is inserted through four equally spaced steel supports with either cylindrical or quatrefoiled holes and clamped on a massive I shaped beam. The beam is laying horizontally on two isolating supports consisting in a massive concrete block inserted between a pair of rubber sheets. An independent tank may be used for immersing the tube in water, see Fig.2. Dynamic interaction between the tube and the I beam is very small due to the mass ratio of the two structures.

Instrumentation

Instrumentation is shown in Figure 2. In the test series described here the tube is excited vertically at its midspan by an electromagnetic shaker driven harmonically. As the I beam has no resonance in the frequency range of interest the shaker has been clamped on the beam. Driving force is measured by an impedance head. As expected in the vibro-impacting regime the driving force signal is noticeably perturbed due to tube-shaker interaction, see Fig. 3. However perturbation is essentially restricted to frequencies much higher than driving frequency (about 20 Hz). It has been checked that high frequency rms value of the signal is less than 10 % of the rms value at the driving frequency. Consequently it is believed that tube-shaker interaction is not a too severe problem in the present experiment.

Horizontal and vertical transverse displacements of the tube are measured by a set of ind--tive Kaman sensors. A pair of sensors is located near excitation at tube midspan, the other next the impacted support. On the test rig meaningfull displacements as small as 10µ have been obtained.

Impact forces are measured using a 3 axial dynamometer Kistler inserted between the beam and the impacted support. When installed on the rig forces from about 0.1 to a few 100N with time scales as short as a few 0.1 ms have been successfully measured. Nevertheless impact signals are noticeably perturbed by the inertial response of the support. As described in the next section this has been efficiently corrected for the vertical component by ensemble averaging the signal over about 150 cycles of vibration.

All the signals are tape recorded and numerized for processing on a computer. A sampling frequency of 3,000 Hz is sufficient for describing the major features of the impact impulses which actually also display still higher frequency content.

3. EXPERIMENTAL RESULTS

Natural Modes of Vibration of the Tube

The tube is firmly maintained by a screw at supports numbered 1, 2, 4 in Fig.1 and free to impact at support 3 where it is approximately centered through the hole. Computed and experimental characteristics of the first 12 flexural modes of the tube in air are reported in Table 2.

Computation					Experiment
N°	f (Hz)	Modal mass	Modal stiffness	Modal displ. at S 3	f (Hz)
1	22.8	0.431	$8.9\ 10^3$	1	23.8
2	62.9	0.496	$7.8\ 10^4$	0.2	63.
3	84.8	0.238	$6.8\ 10^4$	0.	(non excited)
4	127.	0.475	$3\ 10^5$	-0.9	127.
5	209.	0.519	$9\ 10^5$	-0.37	208.
6	245.	0.271	$6.5\ 10^5$	-0.036	255.
7	322.	0.470	$1.9\ 10^6$	-0.84	315.
8	448.	0.535	$4.2\ 10^6$	-0.52	445.
9	450.	0.277	$2.7\ 10^6$	0.02	(non analysed)
10	613.	0.474	$7\ 10^6$	0.77	"
11	785.	0.570	$1.4\ 10^7$	-0.66	"
12	854.	0.284	$7.2\ 10^6$	-0.017	"

Table 2. Modal characteristics of the tube (in air, support 3 : free)

Modes considered here have a generalized stiffness less or of the order of the equivalent impact stiffness. The latter is assumed to correspond to the local ovalisation of the tube cross section ($K_i \cong 6\ 10^6\ Nm^{-1}$). Modal computation has been performed using the finite element beam program TEDEL [8]. Good agreement between computed and experimental frequencies has been reached by assuming no displacement and a stiffness in rotation of 6,000 Nm at each screwed support.

The tube is excited at its first resonance (23 Hz in air and 18 Hz in water), corresponding modal shape is shown in Fig. 4.

Main Features of the Time History Signals

Fig. 5 shows some plots of vertical displacement next an impacted support with radial clearance of 0.25 mm. At low excitation level less than 1N rms the signal is almost sinusoïdal at the driving frequency (about 23 Hz in air and 18 Hz in water). When excitation is increased the signal becomes progressively more disturbed, displaying namely several bounces at each half cycle. Due to support limitation peak amplitude remains essentially constant (about 0.28 mm). Tube motion remains essentially vertical provided that excitation and tube-support clearance are sufficiently small. At midspan displacement is noticeably less distorted than at the support and its amplitude is steadily increasing with excitation, see Fig. 6.

Figure 7 shows a typical vibro-impact response plot including tube vertical displacement next the support (dashed line) and the vertical component of the force at the impacted support (full line). This plot is referring to an excitation level of 10 N rms and a peak midspan amplitude of about 0.7 mm, support hole is quatrefoiled with radial clearance of 0.25 mm. The force signal includes two distinct components, namely :

A high frequency noise limited in amplitude to about 15 N is clearly associated with the inertial response of the Kistler dynamometer loaded by the support, wheight of which is about 20 kg.

A series of very sharp impulses reaching about 90 N in maximum amplitude occurring periodically in coincidence with the tube-support upwards and downwards contacts. Detailed features of tube motion and impact impulses are changing rather randomly from one vibration cycle to the other. Furthermore lack of tube centering through the support may also induce noticeable differencies between upwards and downwards impacts, especially at low vibration level. Horizontal transverse force component also presents rather sharp impulses certainly associated with tube support contacts as seen in Fig. 8. However inertial response is also important and no clear correlation with the overall tube motion has been evidenced. This could be expected as horizontal force is essentially depending on impact oblicity, the latter being poorly correlated with the main vertical tube motion.

Parametric Analysis of the Overall Tube Dynamics

In order to study general features of the vibro-impact response of the tube in relation with excitation and tube-support clearance it has been found more convenient to define at first an equivalent cycle of vibration by ensemble averaging the signals over about 150 consecutive cycles. This data processing is very efficient in reducing the spurious inertial response of the dynamometer as it may be verified in Fig. 9 which shows the vertical component of the equivalent impact force corresponding to the time history displayed in Fig. 7.

Starting from such an equivalent cycle which conveniently characterizes the steady dynamic state, tube dynamics may be described using the following quantities : F is the rms value of the excitation signal as measured by the impedance head, in Newton, δ is the peak value of the tube displacement at midspan in millimeter, EIF is the equivalent impact force per cycle of vibration in Newton, estimated by taking the integral of the absolute value of the equivalent impact force (vertical component) over the equivalent cycle, as shown in Figure 8.

Clearly no great accuracy may be claimed concerning such parameters, in particular the last one. By repeating several times the same series of tests, after dismounting and remounting operations of the tube dispersion less than 40 % concerning EIF has been observed.

As crude as our measurements may be they are still providing valuable information on the tube overall non linear dynamics, as discussed below.

Amplitude of Vibrations

Midspan tube displacement is plotted versus excitation F for four distinct values of the tube-support radial clearance in Fig.10. At sufficiently low excitation depending on clearance no impact is occuring and a resonant response limited by damping is observed. Then increasing F impacts are taking place and tube amplitude δ is increasing with excitation, however at a much smaller rate than it would do in the linear regime. At first this rate is steadily decreasing then it becomes almost constant beyond a certain excitation also depending on clearance. In this fully developed impacting regime tube response is no more resonant near the first mode and at a fixed excitation its amplitude is almost proportional to tube-support clearance value. However in our experiment for the largest clearance values of 0.75 and 1 mm and above F = 5 N the tube tended to have a rather ill defined whirling motion precluding thus the data processing in use here. Such experimental difficulties are likely to be associated with lack of tube centering.

Equivalent Impact Force per Cycle

Fig. 11 shows the EIF plotted versus excitation for a radial clearance of 0.25 mm. The vertical component is observed to increase almost linearly with excitation, reaching about 2 N per cycle of vibration when excitation is 10 N rms. The horizontal transverse component is negligibly small. As already pointed out above this is largely a consequence of the ensemble averaging process performed. Result obtained would merely indicate that angles of oblique impacts are highly scattered leading to a nearly equal probability for having either positive or negative horizontal impulses. No satisfactory data reduction has been yet sorted out for interpreting this horizontal signal.

Fig. 12 shows the vertical EIF plotted versus excitation for the different clearances tested. It seems that provided that motion is fully controled by impacting and that it is still not changed into a complex whirling motion. EIF is increasing almost linearly with excitation and is not very much dependent upon clearance.

Fluid Medium

A few test series have been performed the tube being immersed in water. These tests have been conducted using a cylindrical support hole with radial clearance of 0.25 mm. No significant difference could be noticed in the fully developed impacting regime in comparison with similar tests in air, concerning both tube displacement and equivalent impact force. Similar conclusion has been already produced by Shin et al. [9].

4. NUMERICAL SIMULATION

Method

Non linear dynamics of the impacting tube has been computed using the numerical scheme described by Rogers and Pick [7] briefly outlined as follows : Motion is projected on the modal basis associated with the tube free to vibrate at the impacted support. Impacting is modeled very simply by an equivalent linear spring acting in the direction of motion as soon as impact occurs, i.e. when tube-support radial distance is becoming negative. Impacted support being very stiff spring stiffness is deduced from a static computation of the local tube ovalisation when subjected to a local normal load, this is a classical problem see for instance [10]. Integration of the differential system is performed step by step in time according to a DeVogelaere explicit algorithm [11]. The modal basis may be safely restricted to modes with generalized stiffness less or of the order of the contact spring stiffness, time step must be less than period of the highest frequency mode used.

For present application the twelve modes given in Table 2 have been used. Integration has been made over 150 cycles of vibration for obtaining quite satisfactory steady results, using 400 equal time steps per cycle. Numerical time histories of displacement and contact forces display essentially the same features as the experimental ones. For quantitative comparison, they have been reduced according to the same ensemble averaging procedure.

Midspan Displacement

Figure 13 allows for convenient comparison between numerical and experimental peak midspan displacement values. This quantity is plotted versus excitation level at fixed clearance (0.25 mm) in Figure 13a and plotted versus radial clearance at fixed excitation (3.5 N rms) in Figure 13b. Owing to the various sources of error already mentioned it may be concluded that general agreement between experiment and computation is good, concerning in particular the evolution trends.

Impact Forces

In Fig. 14 it may be verified that computed EIF is varying almost linearly with excitation remaining quite independent upon radial clearance in the fully developed impacting regime. Such a behaviour is also strongly suggested by the experiment, see Fig. 12.

Comparison between numerical and experimental EIF is made in Fig. 15 where experimental data for a radial clearance of 0.25 mm are taken as a reference. Here also agreement is found to be quite good.

5. CONCLUSION

The general objective of this work is to characterize the vibro-impact response of heat exchanger tubes for analysing ultimately the impact-fretting wear process. More specifically our aim is to bring out simple dynamical parameters which could be entered in a wear law of the Archard's type.

Obviously this is not an easy problem as tube motion may display a large variety of dynamical behaviours depending on excitation and on rather tiny local aspects of tube-support interaction which is highly non linear in nature.

Consequently we focussed at first our investigation to the idealized situation described above. Main results obtained are found to be quite encouraging.

At moderate vibration level tube midspan displacement remains essentially proportional to excitation amplitude and to tube-support clearance. Actual impact forces may be conveniently averaged to produce an equivalent impact force per cycle of vibration which is also proportional to excitation but poorly dependent on tube clearance. At this respect it is worth emphasizing that the rms averaging process used by Ko [4, 5, 6] leads to different results. This is simply because detailed time history of individual impulses is highly dependent on excitation and clearance and no simple relation holds between first and second order averages for such signals.

Concerning the Archard's wear law it would be tempting to replace the steady normal force of contact by the equivalent impact force defined here. The same way an equivalent sliding distance proportional to tube-support clearance could be suggested. Clearly such an approach has to be verified by experiment.

The experimental and numerical techniques described in this paper will be applied to other dynamical situations, in particular to more realistic tube whirling motion.

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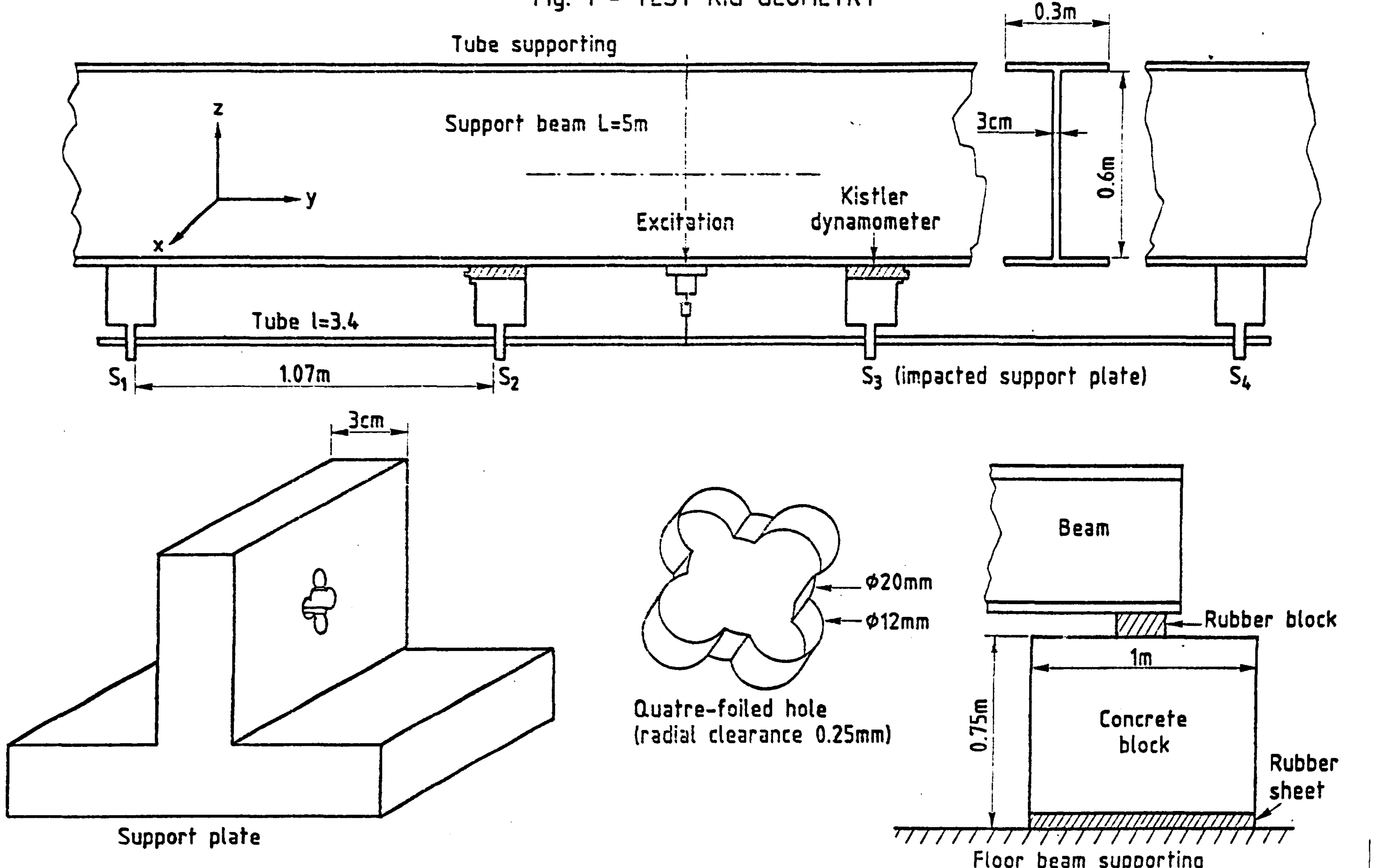
Fig. 1 - TEST RIG GEOMETRY



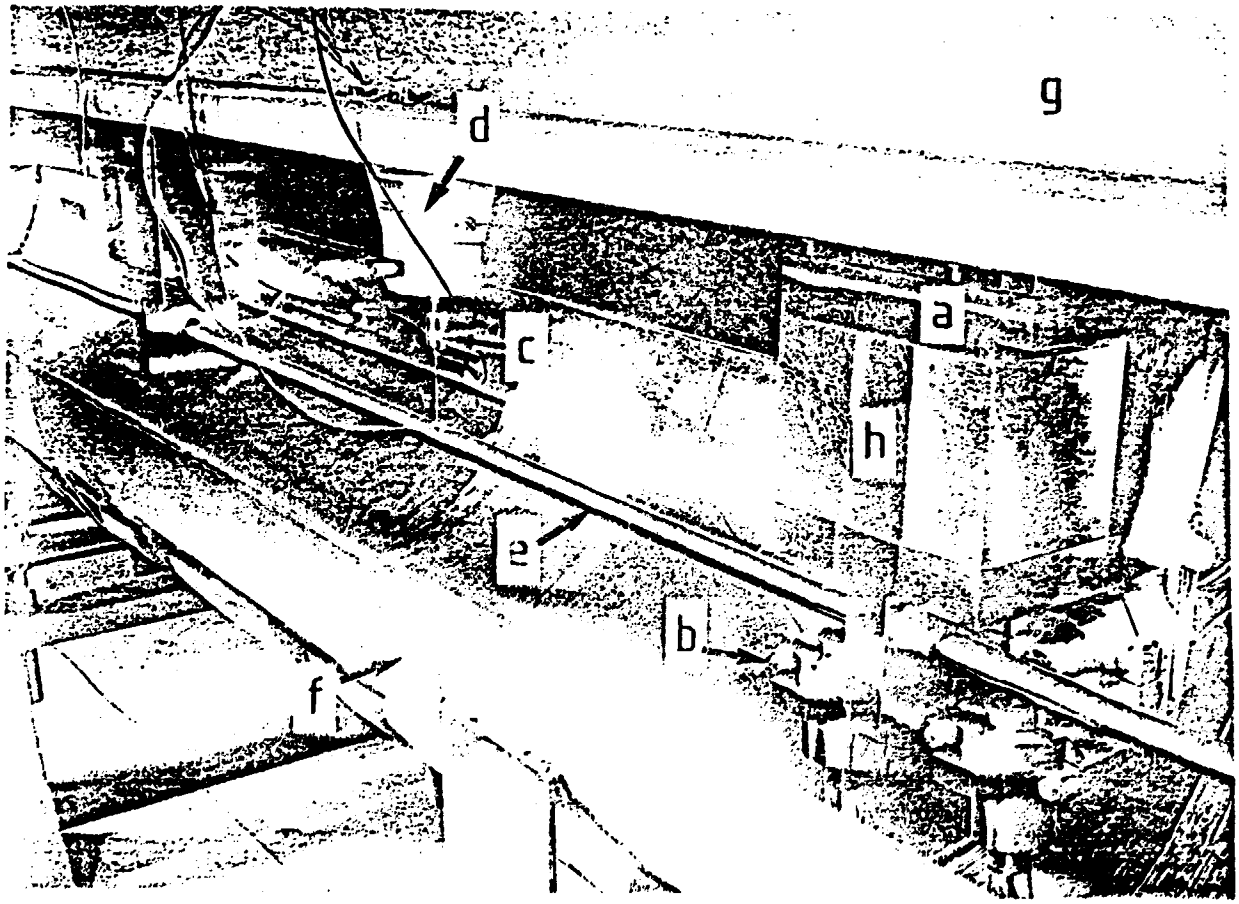


FIGURE 2 - INSTRUMENTATION

a : Three axial dynamometer KISTLER

b : Displacement sensor KAMAN

c : Impedance head ENDEVCO

d : Electromagnetic shaker

e : Tested tube

f : Water tank

g : Support beam

h : Tube support

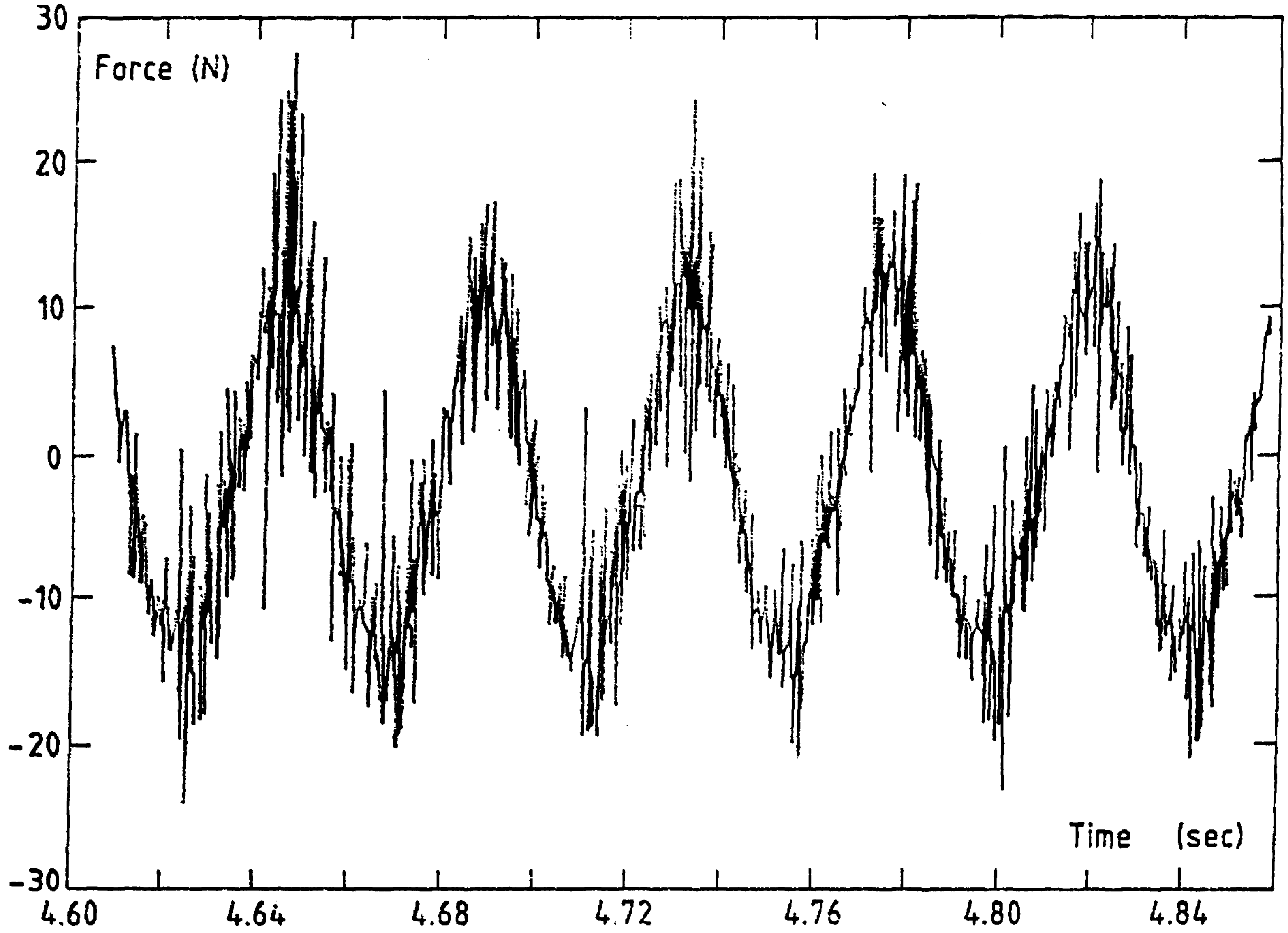


FIGURE 3 - DRIVING FORCE HISTORY IN PRESENCE OF STRONG IMPACTING

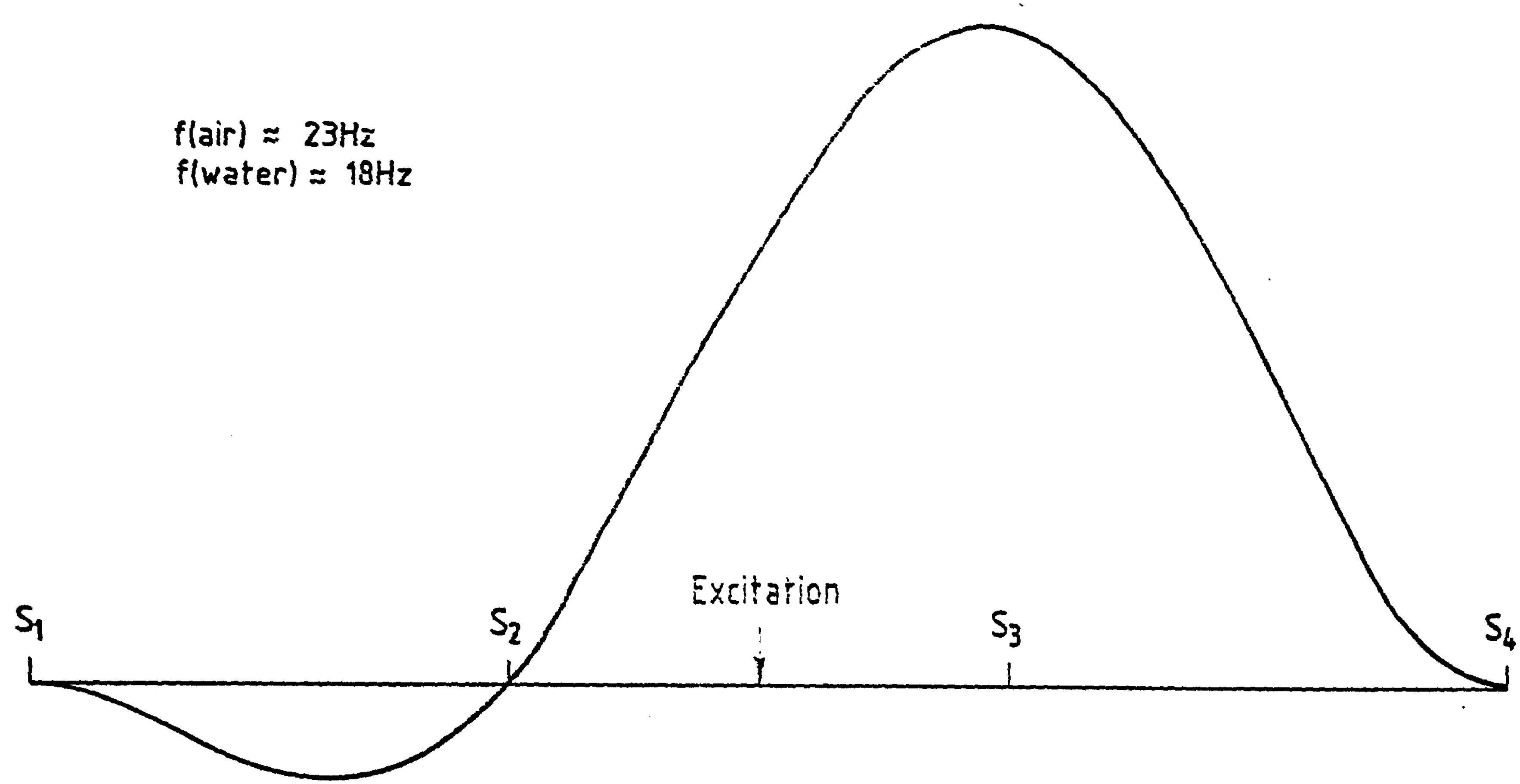


FIGURE 4 - MODAL SHAPE OF THE FIRST TUBE RESONANCE

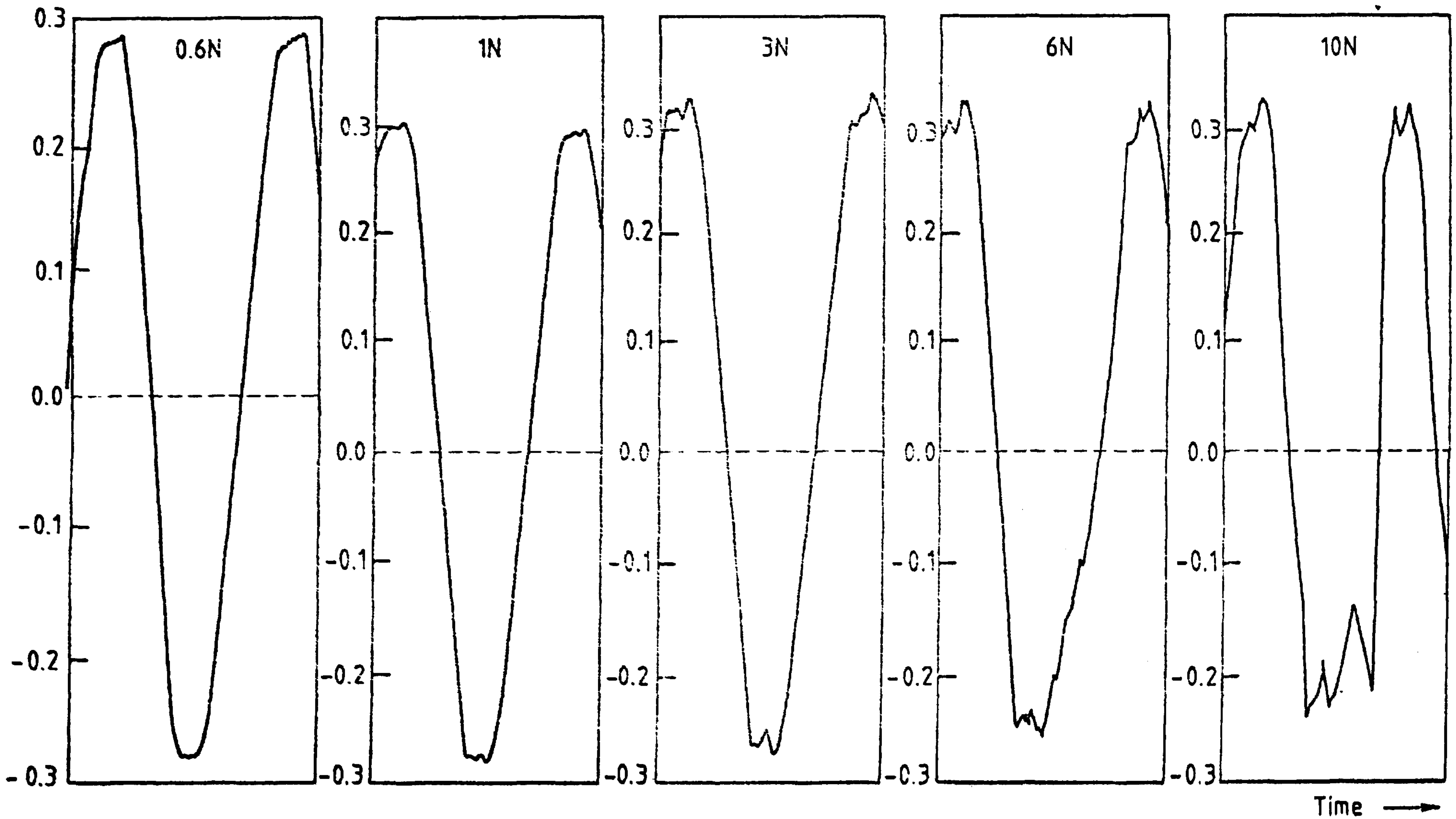


FIGURE 5 - VERTICAL DISPLACEMENT HISTORIES AT THE IMPACTED SUPPORT FOR VARIOUS EXCITATION LEVEL

(radial clearance : 0.25 mm) (Time is in seconds, displacement in mm)

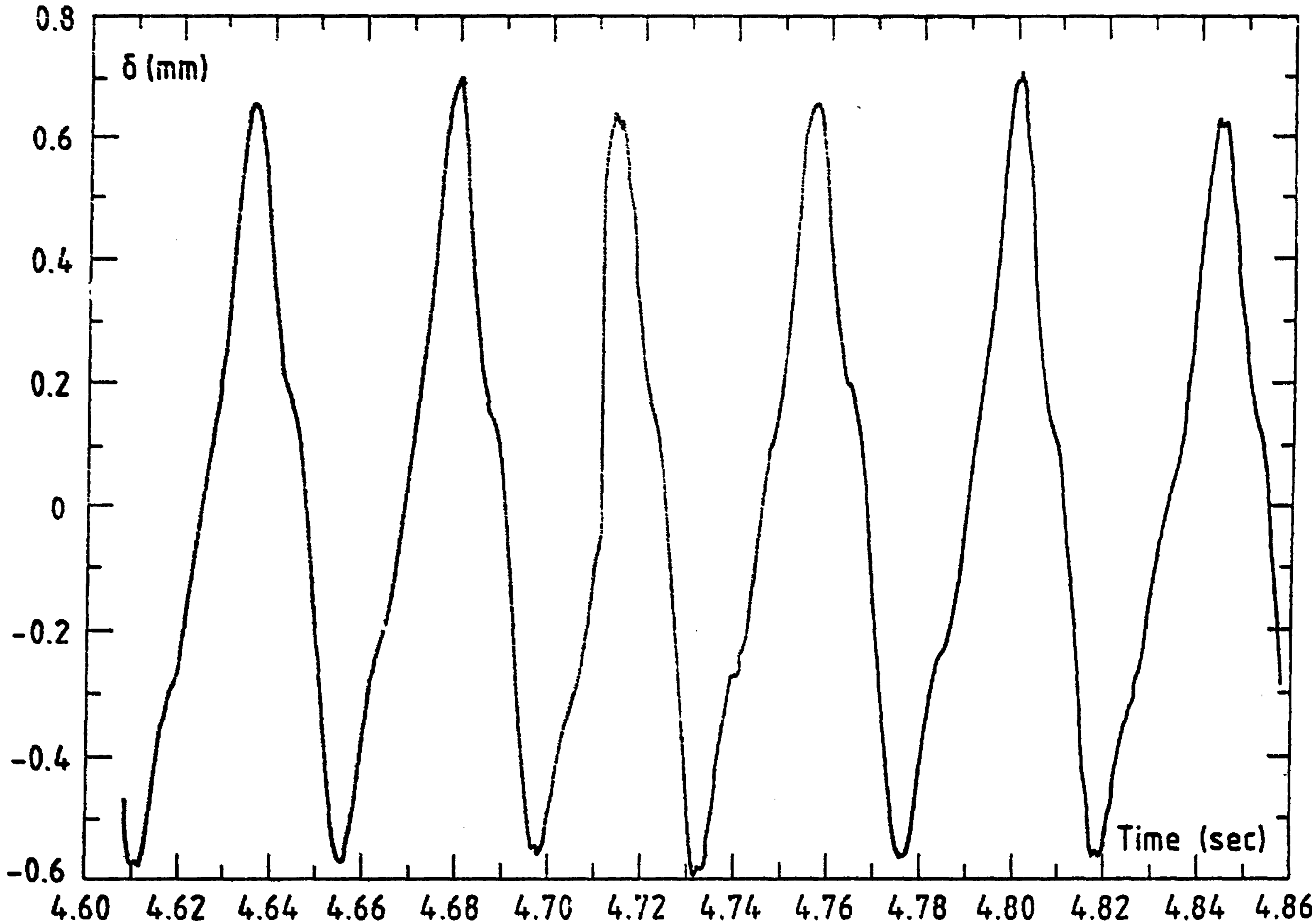


FIGURE 6 - VERTICAL TUBE MIDSPAN DISPLACEMENT, EXCITATION 10 Nrms

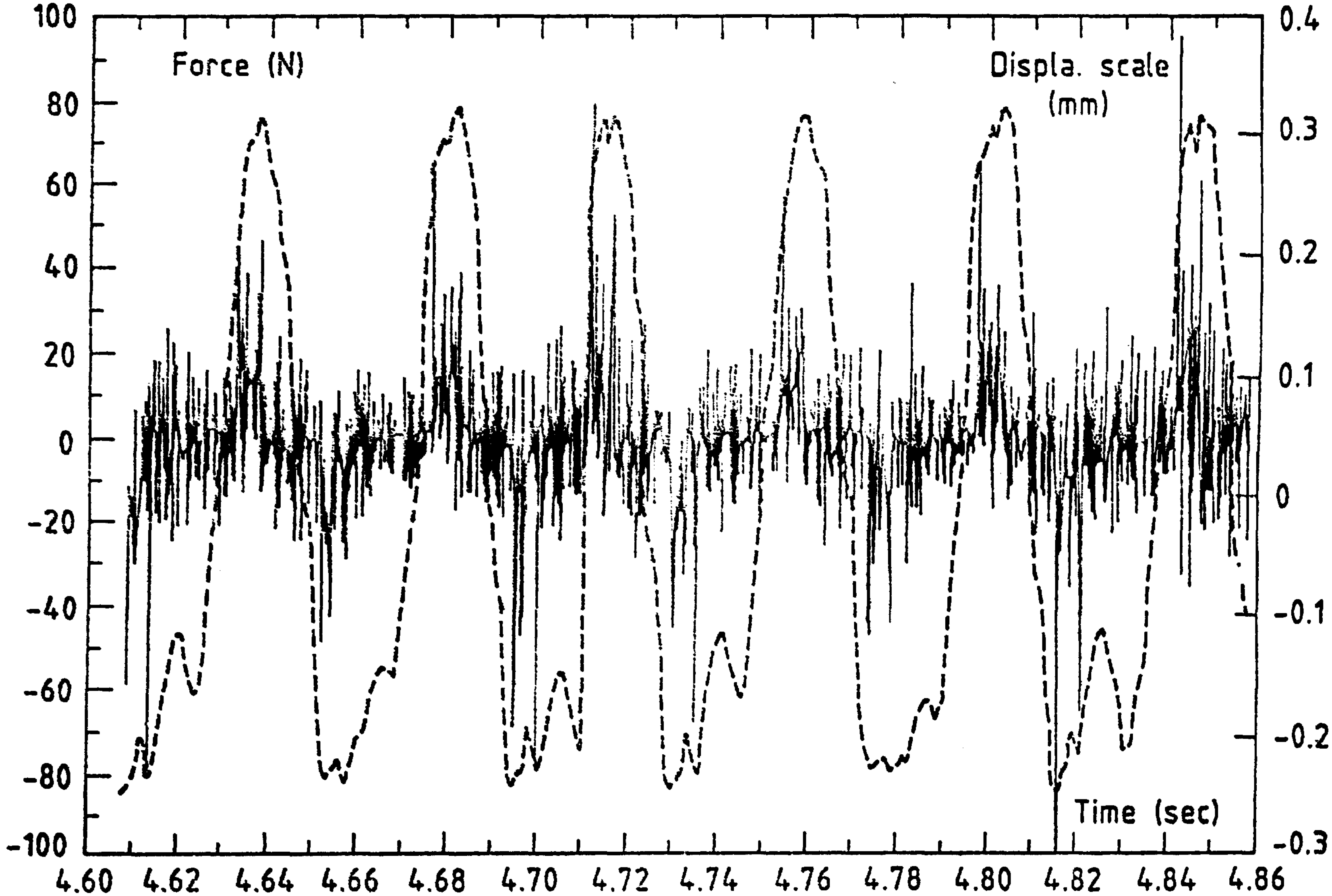


FIGURE 7 - VERTICAL COMPONENT OF IMPACT FORCE (full-line) - EXCITATION 10 Nrms
TUBE VERTICAL DISPLACEMENT NEXT SUPPORT (dashed-line)

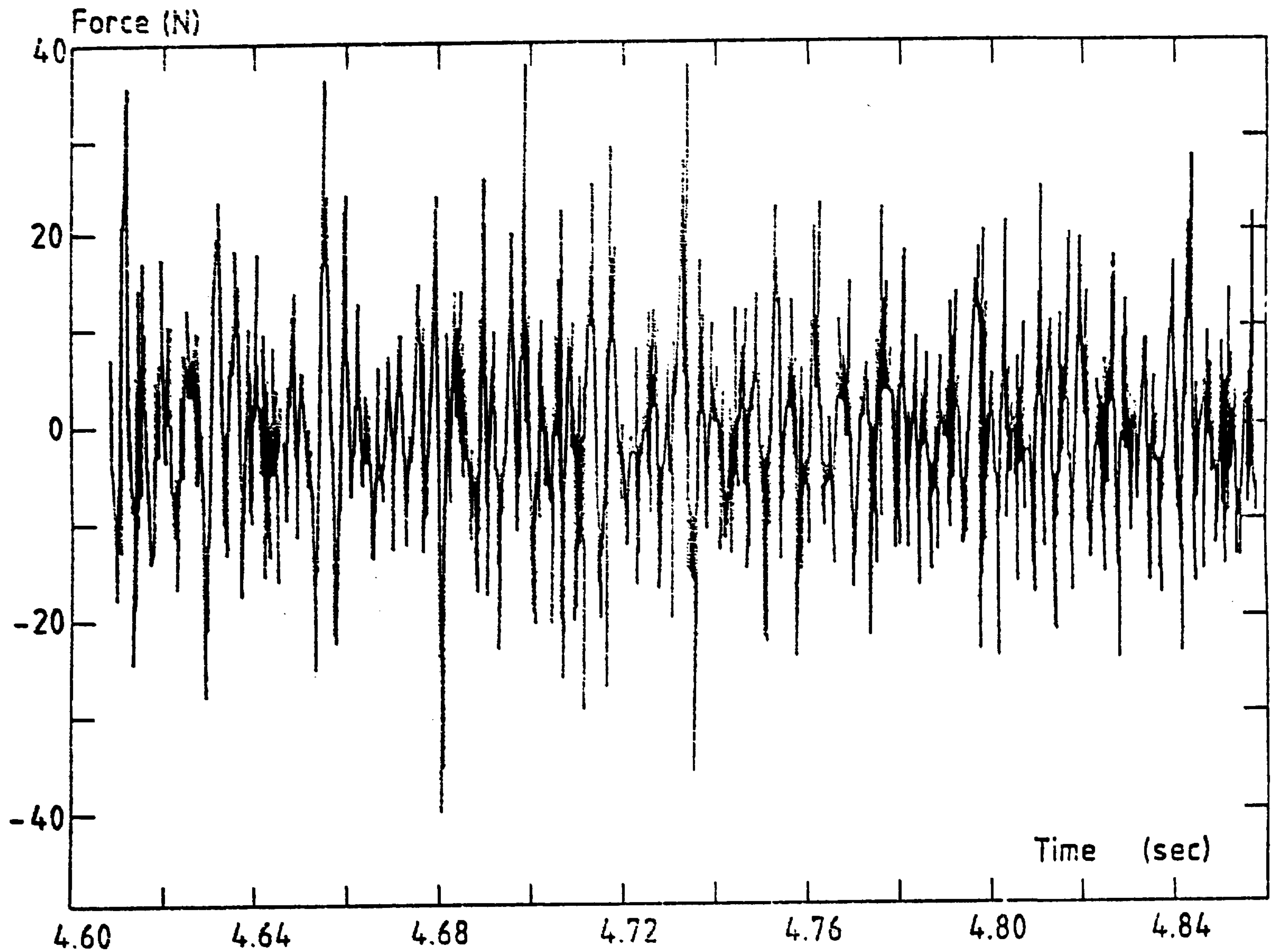


FIGURE 8 - HORIZONTAL IMPACT FORCE HISTORY - EXCITATION 10 Nrms

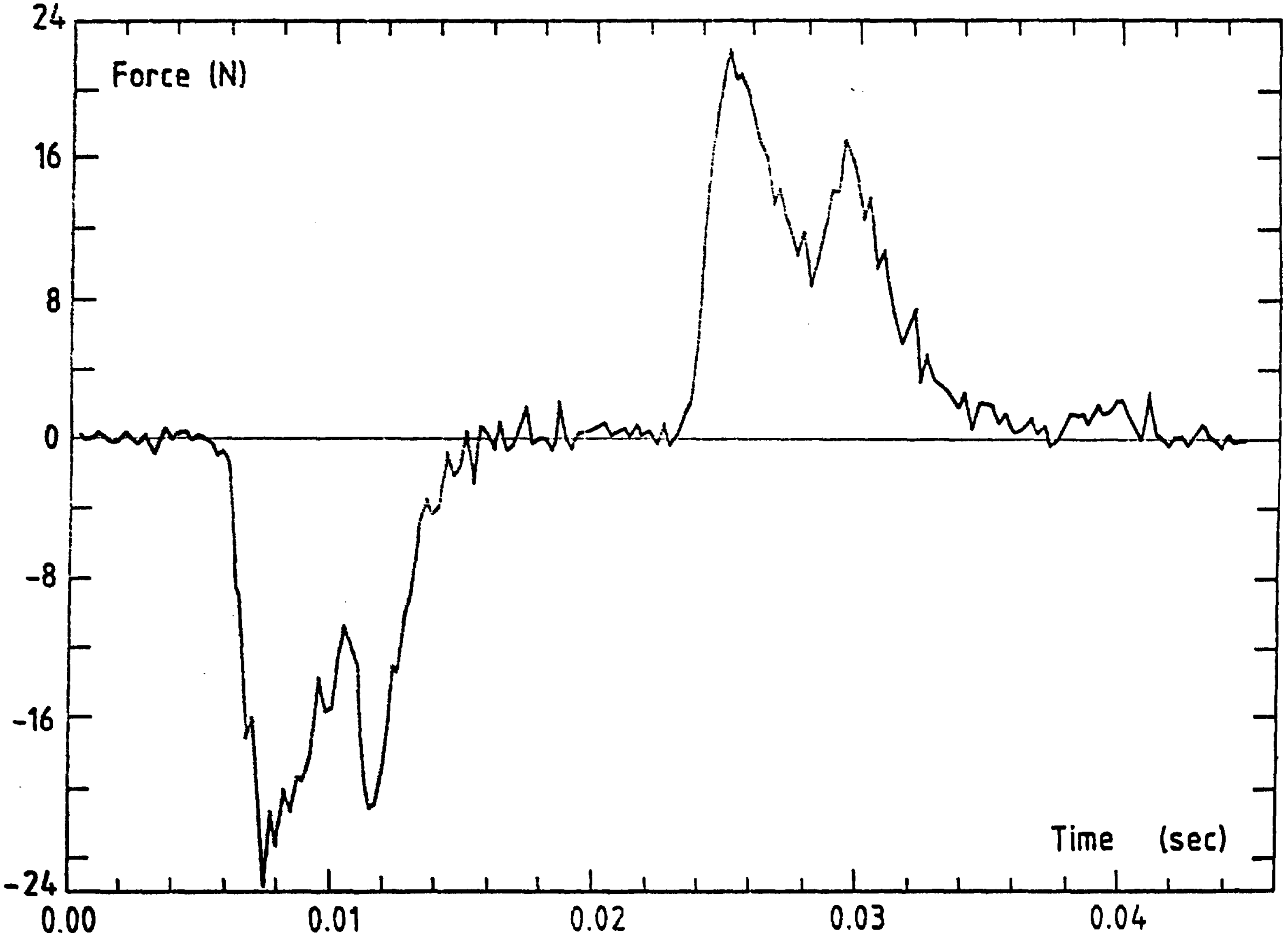


FIGURE 9 - VERTICAL EQUIVALENT IMPACT FORCE HISTORY - EXCITATION 10 Nrms

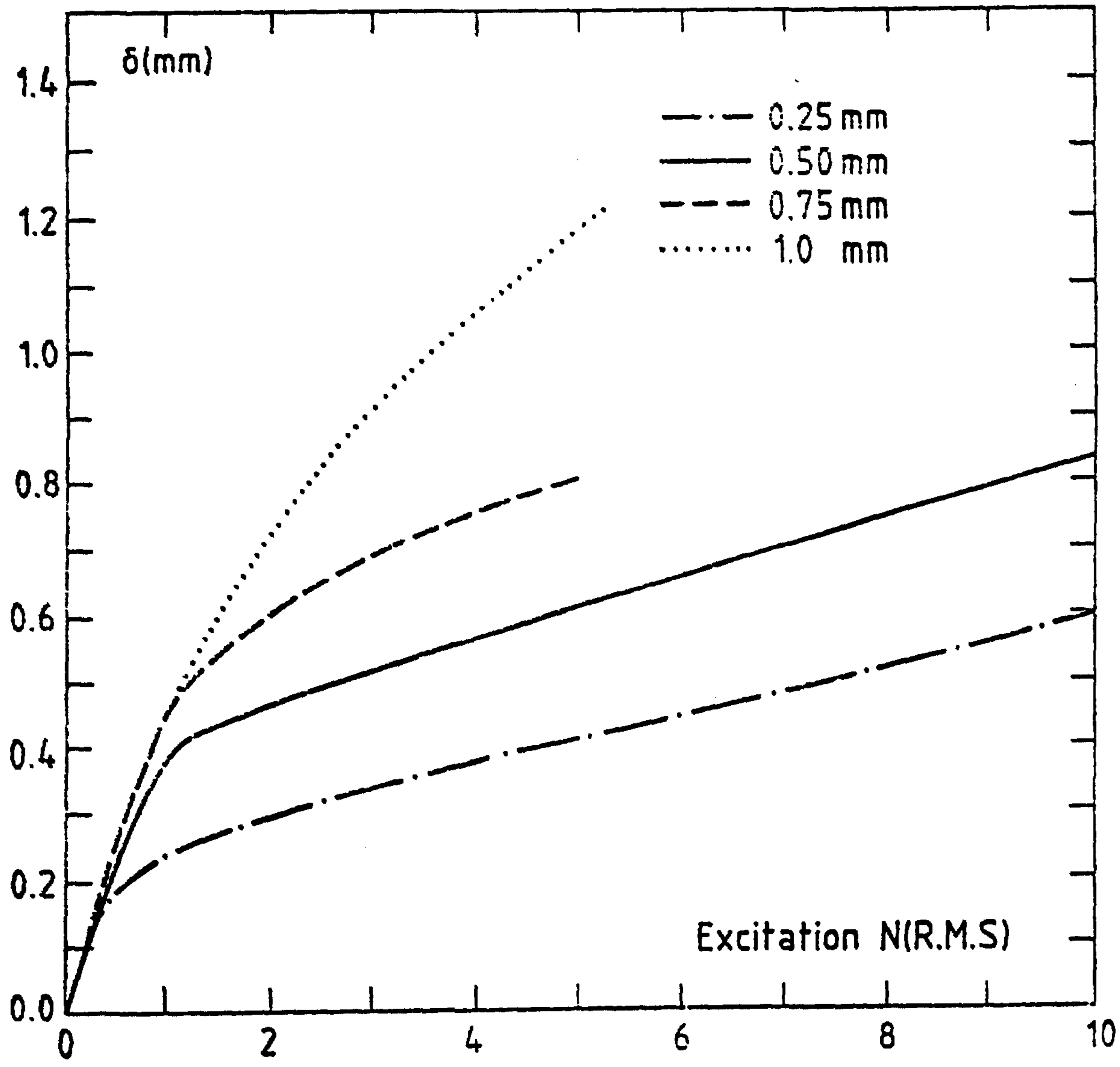


FIGURE 10 - MIDSPAN TUBE DISPLACEMENT FOR SEVERAL SUPPORT CLEARANCES

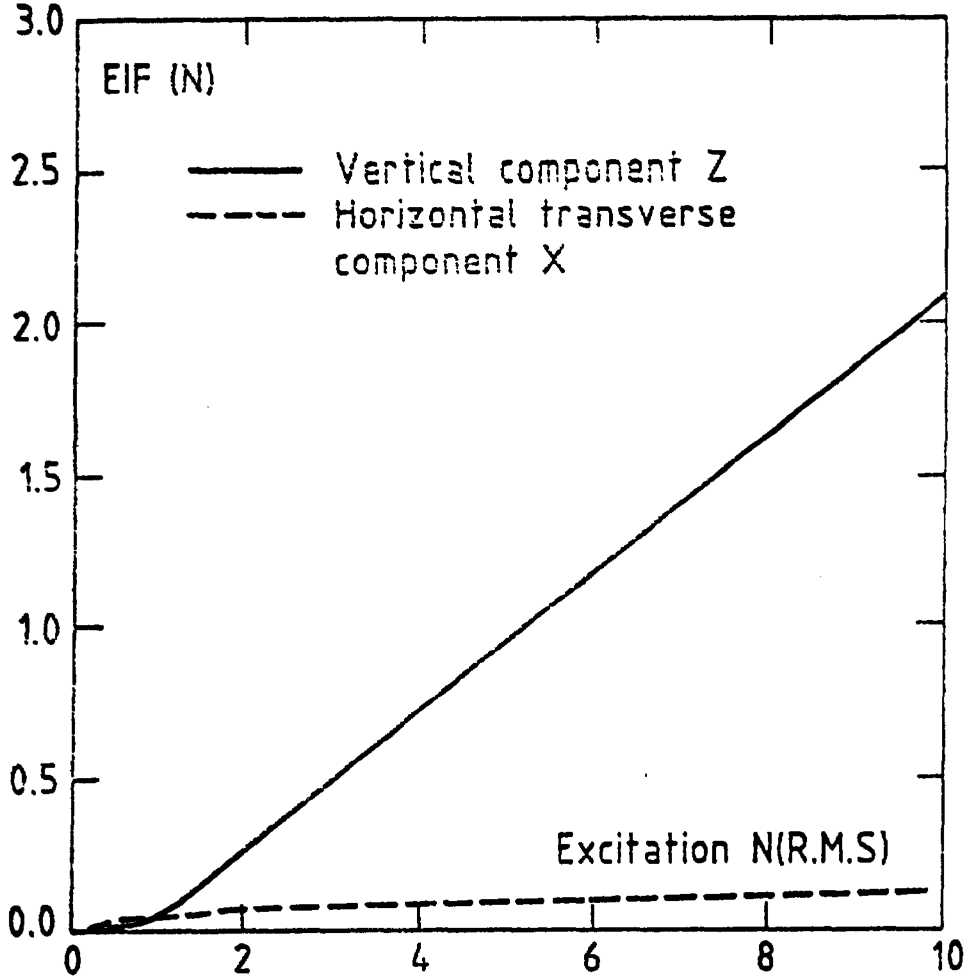


FIGURE 11 - VARIATION OF EQUIVALENT IMPACT FORCE WITH TUBE EXCITATION (clearance = 0.25 mm)

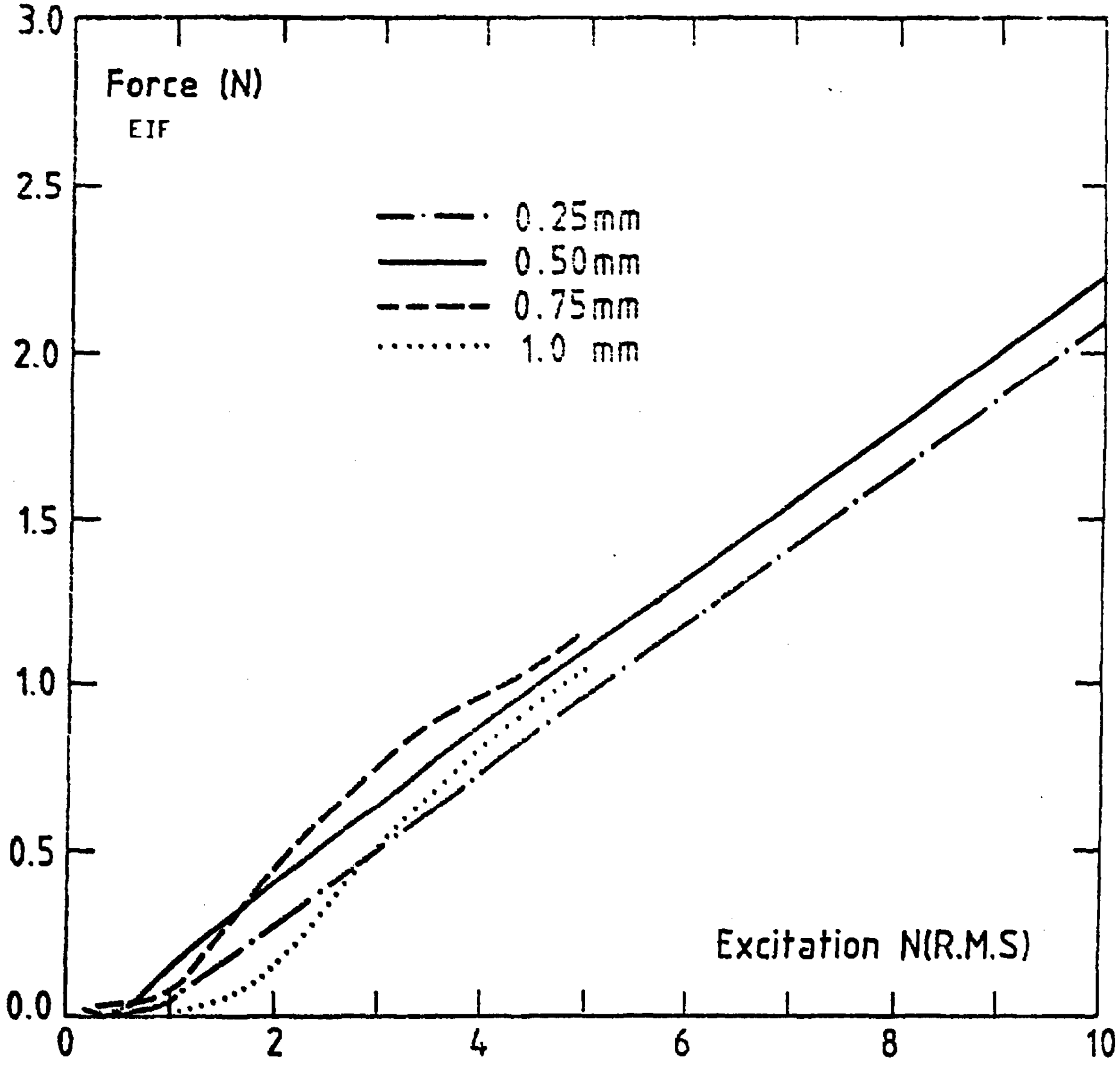


FIGURE 12 - VERTICAL EQUIVALENT IMPACT FORCE FOR SEVERAL CLEARANCES

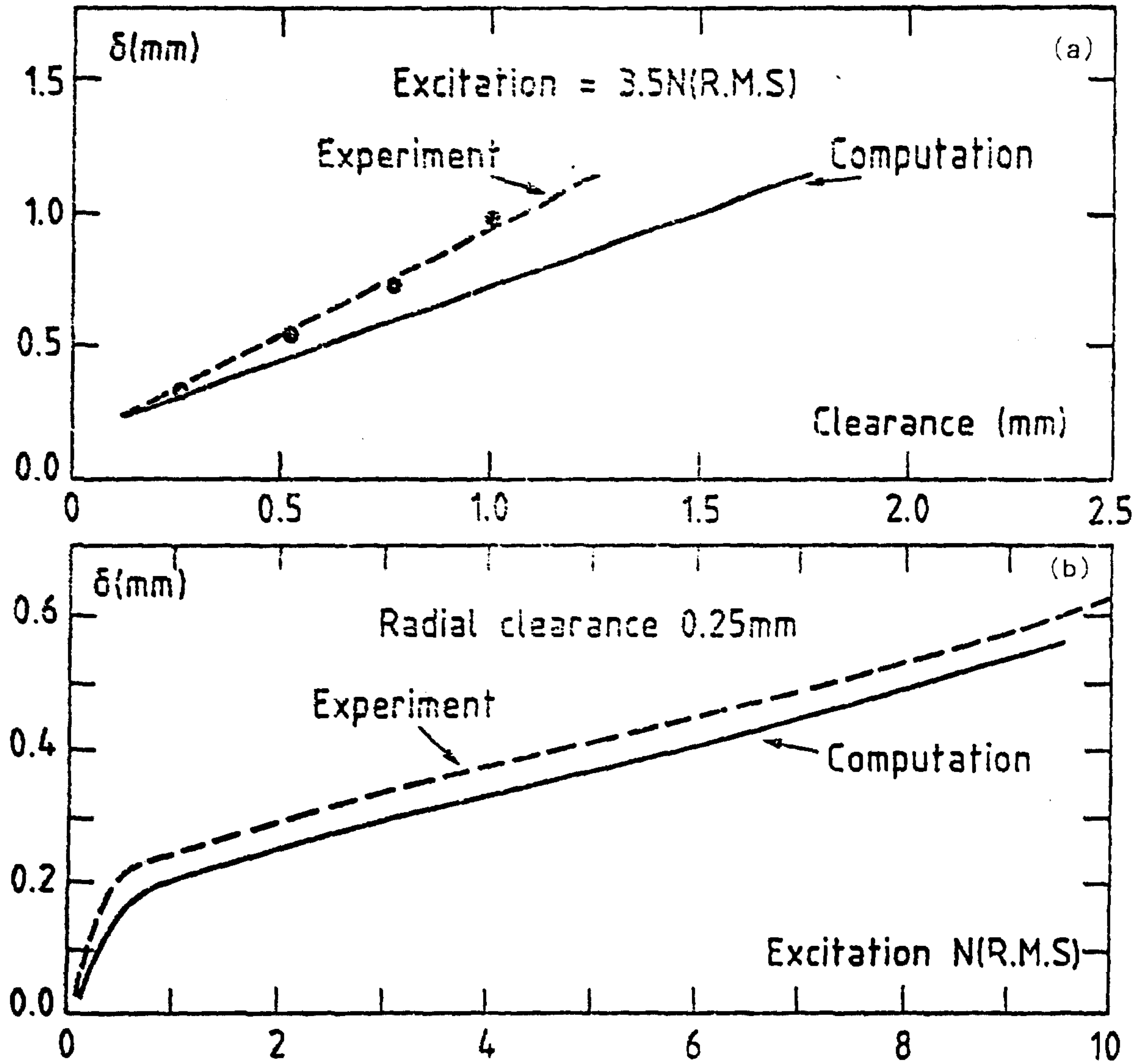


FIGURE 13 - TUBE MIDSPAN DISPLACEMENT : COMPARISON BETWEEN EXPERIMENT AND NUMERICAL SIMULATION

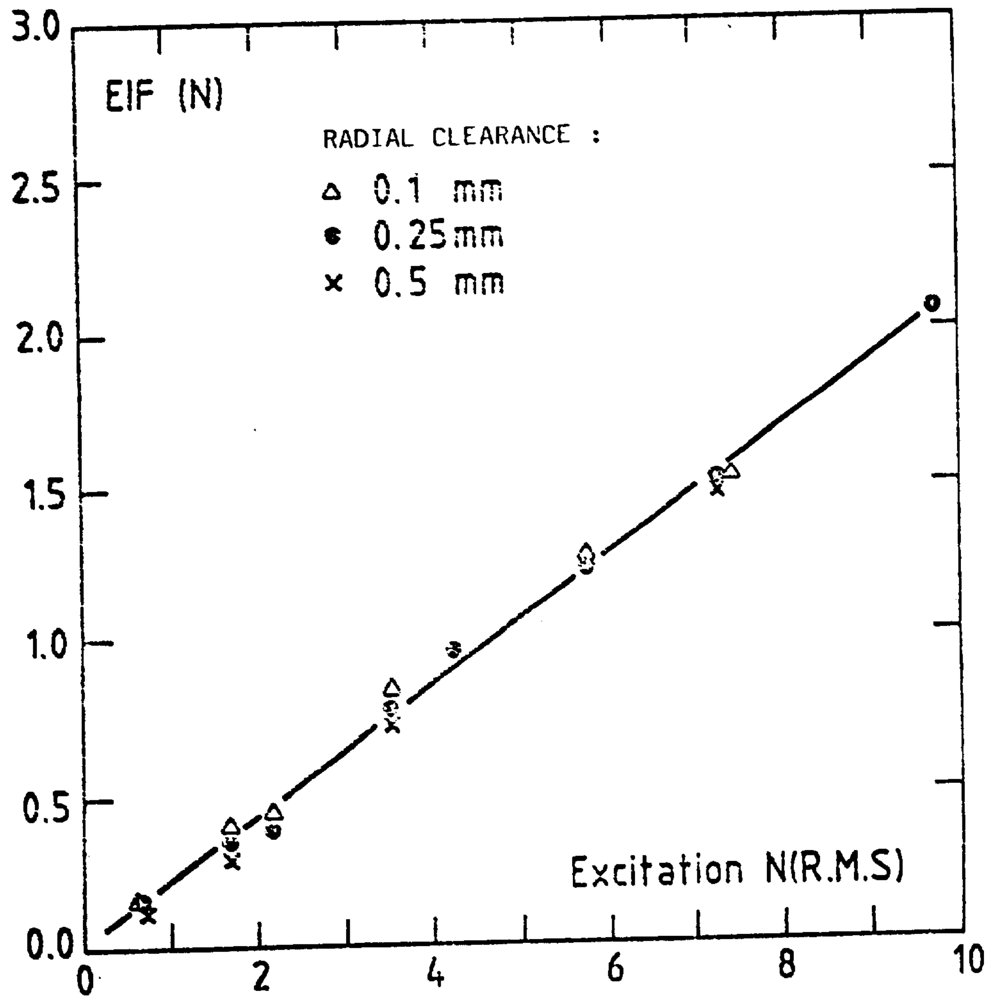


FIGURE 14 - NUMERICAL SIMULATION : EQUIVALENT IMPACT FORCES FOR DISTINCT CLEARANCES

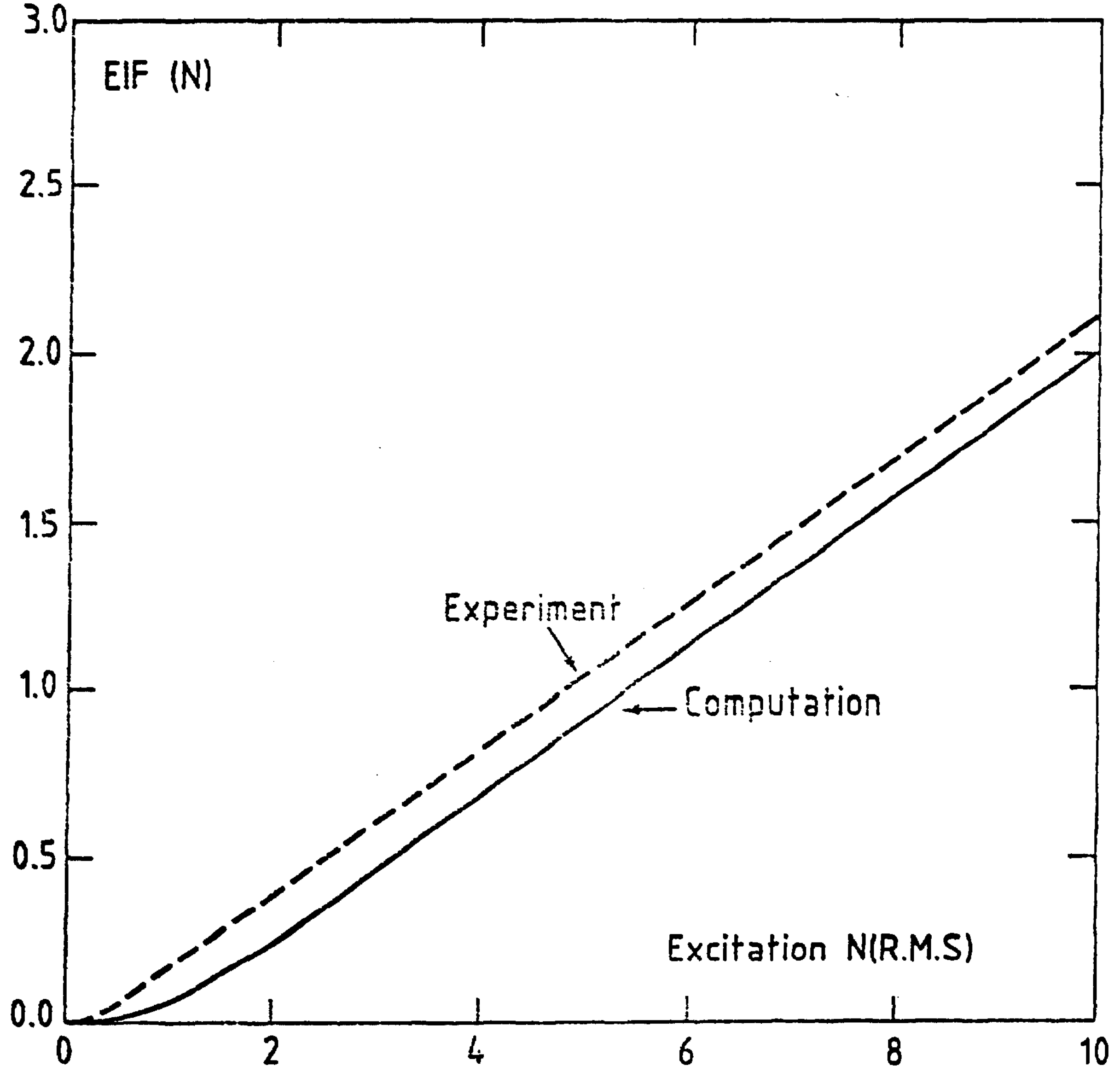


FIGURE 15 - VERTICAL EQUIVALENT IMPACT FORCE : COMPARISON BETWEEN EXPERIMENT AND NUMERICAL SIMULATION