DYNAMIC ANALYSIS OF ELECTRIC EQUIPMENT FOR NUCLEAR POWER STATIONS UNDER SEISMIC LOADS

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SUMMARY

The response spectrum method, generally accepted as the most practical method for linear seismic analysis of power station components, is here applied in conjunction with the finite element method to electric components.

The dynamic response of complex structures is governed by a number of natural modes of the multi-degree-of-freedom system, where for seismic analyses the low-frequency end of the spectrum of natural frequencies usually is significant.

The accuracy of the response spectrum method for a given response spectrum hinges on the accuracy by which the natural frequencies and modes of the structure are known. In contrast to published work on electric equipment, frequencies and modes are here determined using finite element idealisations. These mathematical models of switchgear and other electric equipment may involve several hundreds of degrees-of-freedom. The numerical effort is justified by the ability to accurately model complicated structures as early as in the design phase.

The fully dynamic analysis based on the superposition of the natural modes as carried out for an electronics cabinet and for transmitter supports is outlined and selected results are presented. Several different methods are in use for the superposition of the contributions of the different modes. Here addition of absolute values, the square-root of the sum of squares, and a mixed method taking account of closely spaced modes is applied. For different structures, the degree of conservativity is thus demonstrated, the largest difference in the stresses computed by the different methods being approximately 30 percent.

For structures whose natural frequencies are in the spectrum range with zero period response, a simplified response analysis using static loads is often carried out. This is demonstrated for the electronics cabinet and transmitter mountings, and the results are compared with the fully dynamic analyses. It is seen that this "pseudo-dynamic" analysis yields useful approximations for the distributions of stresses.

Practical details of the structural models as well as results are presented for several switchgear and electronics cabinets.

The practicability of the response spectrum method based on detailed finite element modelling is thus demonstrated, and the data on computer run times for different types of structures allow a reliable estimate of the effort required.

1. Introduction

Seismic loads on components of power plants, be they nuclear or conventional, arise due to the shaking motion of the foundation or the supporting structure in the case of a seismic event. For seismic ground motion, the dominant frequency content usually is in the range of 0.1 to 20 Hertz. This randomly vibrating motion is amplified and filtered by the resonance characteristics of the building to produce the motion at the location of the component. From a structural point of view, switchgear and related electrical equipment react similarly to a forced motion, amplifying and filtering according to its own resonance characteristics. Essentially the same vibratory response of equipment may be due to aircraft crash or other (hypothetical) emergency conditions; however, in these cases higher frequencies may become significant.

In nuclear power plants, critical electrical equipment is required to withstand seismic loads, see eg. [1] , [2] . Typically, qualification requires proof of no-loss-of-function during or after a seismic event. In many cases of electrical equipment, such as metal-clad switchgear, and control and instrumentation systems, these requirements may be interpreted as consisting of

a) structural integrity of the mechanical system consisting of equipment supporting structure and electric device(s),
b) no loss of electrical function of device(s).

Both requirements a) and b) may be proved by dynamic testing. Aspects of dynamic testing are discussed in [1] , [3] , [4] , e. g.. The advantages of full-scale tests, i. e. realistic model, may be offset by problems such as the difficulty of limitations of shaker table size and performance. Therefore, combinations of experimental with analytical techniques have been proposed, [1] , [4] . Whereas experiments must be used to prove requirement b), requirement a) may be examined both by experiment and analytically. This paper deals with an analytical qualification procedure for a); it should also be helpful in selecting the test environment for b).

In this paper, the mechanical behaviour of the equipment ("aspect a)") is analysed by application of the Finite Element Method. This method, which is well documented in the open literature, allows the realistic modelling of complex structures from the information contained in technical drawings. The natural frequencies and corresponding modes may be determined to a high degree of accuracy, depending of course on the degree of fineness of the discretisation and the number of required frequencies. Corresponding computer programs have been on the market for years and lend themselves to routine application.

Once the natural frequencies and modes are known in the range of interest for the respective seismic motion, the response of the component may be calculated, provided the motion history of its base is supplied.

The response of the mounting location of the component is commonly given by floor response spectra. These spectra contain information on the maximum values of the response as encountered by a single-degree-of-freedom system. The response of multi-degree-of-freedom systems may also be calculated from the floor response spectrum. This method is described e. g. in [5] , [6] , [7] , [9] , [10] , [11] and will be reiterated below only with regard to its underlying assumptions. The generation of floor response spectra from ground motion is not discussed here, see e. g. [8] , [9] .

The detailed discretisation of the equipment structure as proposed here allows the prediction of the local maximum response values at the location of individual devices in the equipment. Thus, the maximum acceleration with which the individual device is to be tested may be specified as outlined below. However, it is conceded that for equipment containing many different small devices, this procedure may prove too cumbersome; for such cases a simplified procedure is to apply empirical amplification factors to the overall equipment response, while still making use of the analytically found natural frequencies of the equipment.

Probably the most important feature of the detailed discretisation employed here is the direct translation of actual structural elements into finite elements, and vice versa. This allows modification of a structure as early as in the design phase to achieve prescribed dynamic properties. As an example, switchgear and related equipment may be designed to requirements regarding minimum natural frequencies. This was done for the applications presented below, where a minimum natural frequency of 13 Hertz was achieved, in some cases by relatively simple modifications. For the floor response spectra prescribed in this case (see Fig. 1), this minimum natural frequency ensures that the dominant resonant frequencies of building and equipment resp. are far enough apart to avoid excessive amplification of the seismic acceleration. The required floor response spectra for aircraft impact (see e. g. Fig. 2) or explosion usually are such that a design goal of minimum equipment frequency exceeding the frequency range of significant floor response would impose severe cost penalties. However, raising the minimum natural frequencies and thus the complete spectrum of natural frequencies will usually lead to reduced device response even in these cases.

2. Prerequisites of the Response Spectrum Method

2.1 Single-degree-of-freedom Systems

For a simple system consisting of a concentrated mass connected to its foundation or the floor via a spring and a damper, the motion is governed by the well-known differential equation

$$\ddot{y} + 2\omega_1 \xi \dot{y} + \omega_1^2 y = -\ddot{x}_o \qquad (1)$$

where y is the relative displacement of the mass, x_o is the displacement of the foundation, with dots denoting differentiation with respect to time. The quantity

$$\omega_1 = \sqrt{K/M} \qquad (2)$$

denotes the circular frequency of the undamped system, where K and M are the stiffness and the mass, resp.;

$$\xi = C/2\sqrt{KM} \qquad (3)$$

gives the damping ratio (in relation to the critical damping), where C is the damping constant. ω_1 and ξ are constants for a linear system, for which the damping force is assumed proportional to the relative velocity (viscous damping).

Maximum absolute values of the displacement response,

$$S_d = \max |y(t)| \qquad (4)$$

for different natural frequencies ω_1 and damping ratios ξ due to a specified floor motion history x_0(t) are contained in the displacement response spectrum. For a system governed by eq. (1) with constant coefficients, it can be shown that the velocity response and the total acceleration response are well approximated by the spectral pseudo velocity S_v and acceleration S_a ,

$$\max |\dot{y}(t)| \approx S_v = \omega_1 S_d \qquad (5)$$

$$\max |\ddot{y}(t) + \ddot{x}_0(t)| \approx S_a = \omega_1^2 S_d \qquad (6)$$

provided the damping ratio ξ is small, and the natural frequency ω_1 is not too small. For the equipment analysed here, the damping ratio is in the order of .02 to .07 [2] . Therefore, if the lowest natural frequency of the equipment is not below the frequency range of significant response, eqs. (5) and (6) appear applicable.

In the case an experiment is conducted to prove system qualification to a given required response spectrum, the question arises as to which floor motion history is to be used. If e. g. the acceleration amplitudes as given in the required response spectrum are used as the acceleration amplitudes of a sine sweep shaker motion, this will lead to a response acceleration of the component mass which is the required (maximum) acceleration times the acceleration amplification factor for this type of floor motion. The acceleration amplification factor may take values both smaller and greater than unity. Rather, it would seem appropriate in this case to employ as foundation acceleration amplitude the value given in the required response spectrum for the natural frequency of the system, divided by the corresponding acceleration amplification factor.

2.2 Multi-degree-of-freedom System

A continuous or discrete structural system may be idealised as an assemblage of finite elements. In the current context, it is assumend that the structure is fixed to a rigid foundation or floor, where the foundation may execute rigid movements (displacements and rotations). Without entering into details, we state here that the motion of the system is governed by a set of n_E equations

$$\ddot{y}_i + 2\omega_i \xi_i \dot{y}_i + \omega_i^2 y_i = -\sum_j P_{ij}^* \ddot{x}_{oi} , \quad i = 1, \dots n_E , \; j = 1, \dots n_R \qquad (7)$$

In eq. (7) above, the different symbols denote the following:

$y_i(t)$	generalised coordinate of normal mode i
ω_i	undamped natural frequency of normal mode i
ζ_i	damping ratio of normal mode i
x_{oj}	generalised coordinate of rigid foundation mode j
n_E	number of (elastic) normal modes,
n_R	number of modes of rigid foundation motion,
P_{ij}^*	participation factor of i-th elastic mode with j-th rigid mode, see below eq. (9).

The motion of the individual nodal points is given by the vector of nodal displacements

$$\mathbb{r} = \sum_i \mathbb{Y}_i y_i + \sum_j \mathbb{X}_{oj} x_{oj} \tag{8}$$

where $\mathbb{Y}_i$ denotes the vector of nodal displacements for the i-th normal mode shape, and $\mathbb{X}_{oj}$ denotes the vector of nodal displacements due to rigid mode j. The determination of the normal mode shapes from the mass matrix $\mathbb{M}$ and the stiffness matrix $\mathbb{K}$ of the structure is standard finite element technique. The participation factor mentioned above derives from the mass matrix via

$$P_{ij}^* = (\mathbb{Y}_i^T \mathbb{M} \mathbb{X}_{oj}) / (\mathbb{Y}_i^T \mathbb{M} \mathbb{Y}_i). \tag{9}$$

In the derivation of the above equations, several assumptions are made. First, both material and geometric linearity are assumed, the latter including a linear relation between foundation motion and the nodal accelerations (as pointed out in [11] , this implies neglecting Coriolis forces). Second, it is assumed that damping is viscous and that the damping matrix $\mathbb{C}$ takes the special form

$$\mathbb{C} = \alpha \mathbb{K} + \beta \mathbb{M} \tag{10}$$

with α and β being constant factors (so-called proportional damping). Furthermore, coupling of elastic and rigid modes via damping forces is neglected, which usually implies $\beta = 0$. The simularity of eq. (1) and each of eqs. (7) is evident, especially if we consider the contribution of only one rigid foundation mode ($n_R = 1$). From a given response spectrum (in the sense defined for a single-degree-of-freedom system) corresponding to a specific floor motion $\ddot{x}_{oj}(t)$, the maximum contribution of elastic mode number i to the relative nodal displacements may be found from

$$\mathbb{r}_{Ei} = \mathbb{Y}_i P_{ij}^* S_{dij} \tag{11}$$

where S_{di} is the value of the displacement response appropriate to the frequency ω_i. The reasoning which led to eq. (5) giving the velocity response may again be applied to yield

$$\dot{\mathbb{r}}_{Ei} = \mathbb{Y}_i P_{ij}^* S_{vij} = \mathbb{Y}_i P_{ij}^* \omega_i S_{dij} \tag{12}$$

However the extension of eq. (6) to distributed systems (with non-unit participation factors) in general is not immediately evident. It is usual practice to take the effective acceleration response at the nodal points to be

$$\ddot{r}_{eff\,i} = \gamma_i P^*_{ij} S_{aij} = \gamma_i P^*_{ij} \omega_i^2 S_{dij} \qquad (13)$$

The response spectrum contains only the absolute values of the maximum response and thus gives no information as to how the contributions of the individual normal modes are to be combined. An upper bound to the response is certainly obtained by absolute summing of the response entities,

$$R = \sum_i |R_i| \qquad (14)$$

where R stands for displacement, velocity, acceleration, or any other response entity such as stress or stress resultants, and R_i denotes the contribution to R due to the normal mode i. Other rules for the superposition have been given, e. g.

$$R = \sqrt{\sum_i R_i^2} \qquad (15)$$

$$R = \sqrt{R_1^2 + R_2^2 + \cdots + (|R_n| + |R_{n+1}|)^2 + \cdots} \qquad (16)$$

where for eq. (16) ω_n and ω_{n+1} denote natural frequencies which are closely spaced.

The choice of superposition method should take into account the type of motion (random, such as earthquake, or deterministic, such as well defined shaker motion) as well as the dynamic characteristics of the structure (natural frequency and mode shape).

If the floor motion has several components (as is the case in seismic motion, where both horizontal and vertical motions are induced independently), these response contributions also have to be superimposed. Again, the choice of method of superposition is somewhat subject to individual judgement (or qualification standards).

3. Structural Idealisation Based on Finite Elements

The first step in the structural analysis consists of the idealisation of the actual load-carrying structure by finite elements. The load-carrying structure of switchgear and related electrical equipment usually consists of a rather light-weight steel skeleton, possibly reinforced by thin-walled plates. The direct translation of the actual structural elements to strut, beam and plate finite elements is rather straight forward finite element technique.

In the context of seismic and related qualification produre, the analysis may be considered as conservative if the stiffness of the actual structure is underestimated while retaining the mass both with regard to amount and to distribution. With this in mind, only those structural elements should enter in the idealisation whose loadsustaining capabilities are beyond doubt.

From the great numberof different types of equipment which outfit even a single power plant, representative "worst case" types were selected for seismic analysis. Compartments mounted in a row were individually analysed, the contention being that neglect of the strengthening effect due to the interconnection of compartments is again conservative.

The damping of the structure depends on the type of construction (welded or bolted); modal damping was assumed in the limits prescribed by the regulatory guide [2].

Figs. 3, 4, 5 and 7 show computer plots of the idealised structure for a 6kV switchgear compartment, a low voltage compartment, an electronics compartment and a transmitter rack, resp.. The dotted arrows indicate suppressed degrees of freedom, either to simulate actual supports or to avoid singularity of the idealised structure. The numbers in Fig. 7 denote nodal point numbers.

4. Selected Results

4.1 Electronics Compartment

An electronics compartment the idealised structure of which is shown in Fig. 5, was analysed for seismic response. The required acceleration response spectrum of Fig. 1 was used to calculate the response according to eqs. (7) to (9) and (11) to (13). The normal modes of the eight lowest natural frequencies were employed; all of these frequencies lie in the range of the zero-period acceleration in the required response spectrum, i. e. above 12 Hertz. For each mode, local stress resultants and stresses were calculated and superimposed according to the rules of eqs. (15), (14) and (16), resp.. The maximum values are given in Table I together with the results of a simplified analysis which is based on the static response of the same structure to the corresponding zero-period acceleration. Fig. 6 depicts the maximum stresses due to horizontal and vertical motion for the two main struts, plotted as dotted lines (the length of the dotted lines in the oblique direction corresponds to the magnitude of the stress).

Inspection of Table I demonstrates the degree of coincidence which may be achieved for the different superposition rules, and also between dynamic and simplified static analysis. It is also evident, that in this case the choice of participating normal modes is insufficient for the vertical response; actually, none of the eight modes included contributes significantly to the vertical response, so that for vertical motion, the dynamic results seem doubtful. The results for the two horizontal motions underline the agreement between static and dynamic solution, as long as the dominant natural frequencies come to lie in the zero-period-acceleration range.

4.2 Transmitter Rack

A dynamic analysis was carried out for the transmitter supporting structure of Fig. 7 under aircraft impact loads. The response spectrum of Fig. 2 applies. 21 natural frequencies were found in the range up to 100 Hertz. The corresponding modes were taken as generalised coordinates for the response analysis, which again follows eqs. (7) to (9) and (11) to (15). Fig. 8 and 9 give plots of modes no. 5 and 20.

Selected results are listed in Table II separately for the three directions of floor motion. Stresses resulting from modal superposition according to eqs. (15) and (14) resp. differ by no more than 15 percent, whereas the maximum accelerations differ by as much as 39 percent. (For aircraft impact, absolute summation of the modal contributions is prescribed in [2]).

5. Computational Aspects

For a number of different analyses, Table III lists information on problem size, finite element program, computer type and central processor time used. (Note: on the IBM/370-158 and -168, ASKA uses double-precision programming for essential steps, whereas for the CDC 6600, single precision is sufficient).

As is the case with any finite element analysis, the efficiency by which such analyses are performed depends not only on the reliability and efficiency of the program, but also on the availability of flexible pre- and postprocessors for data generation and presentation of results. Both finite element programs used are large general purpose programs which have been well tested. General purpose pre- and postprocessors developed by the authors helped reduce man-time, and the overhead thus invoked (less than 20 percent. of the computer cost for the finite element runs) is more than offset by the gain.

6. Conclusion

For electric equipment of power plants, the mechanical response to seismic and other vibratory loading may be calculated, using existing methods and software. The insight into the mechanical behaviour of the equipment gained through the analysis allows modifications in the design stage, and may help reduce the mechanical requirement on the individual electrical device. The analysis cannot replace experimental qualification completely. In many cases, however, in conjunction with the analysis, tests may be limited to the experimental proof of no-loss-of-function of the individual devices. The costs of such a combined qualification procedure are usually well below the costs of full-scale tests.

References

[1] SKREINER, K. M, FITZGERALD, E. M., MARTZ, J. W., SHERLOCK, J. E.: New Seismic Qualification Methods for Class IE Switchgear and Equipment. IEEE Trans. Power App. Systems PAS-95 (1976) pp. 335-343.

[2] Anon.: Auslegung von Kernkraftwerken gegen seismische Einwirkungen Teil 1: Grundsätze. Sicherheitstechnische Regel des KTA 2201, 1975. Teil 4: Bemessung der maschinen- und elektrotechnischen Anlagen. Vorbericht an den KTA, 1976.

[3] HAMMOND, G. W., CHAPMAN, F. X., HUNNEWELL, R. E.: Qualification Testing of Class 1E Switchgear per 1974 Seismic Standards. IEEE Trans. Power App. Systems PAS-94 (1974) pp. 2002-2007.

[4] KUGLER, R.: Erdbebenfestigkeit von Schaltgeräten.
etz-a 98 (1977) 2 pp. 122-128.

[5] NEWMARK, N. M, ROSENBLUETH, E.: Fundamentals of Earthquake Engineering.
Englewood Cliffs, N. J., 1971, Prentice-Hall.

[6] NEWMARK, N. M.: Earthquake Response Analysis of Reactor Structures. Nucl. Eng. Design
20 (1972) pp. 303-322.

[7] CLOUGH, R. W.: Earthquake Response of Structures.
Chapter 12 in: Earthquake Engineering, Englewood Cliffs,
N. J., 1970, Prentice-Hall.

[8] BIGGS, J. M.: Seismic Response Spectra for Equipment
Design in Nuclear Power Plants. Contribution K 4/7
2nd Int. Conf. Structural Mechanics in Reactor Technology,
Berlin 1973.

[9] MARGUERRE, K., SCHALK, M., WÖLFEL, H.: Berechnung der Erdbeben-
schwingungen von Strukturen mit der Finite-Elemente-Methode -
Mechanische Modelle von Kernkraftwerken mit Einbauten.
Proc. 2nd Int. Conf. Structural Mechanics in Reaktor Technology,
Berlin 1973.

[10] KALNINS, A., GODFREY, D. A.: Seismic Analysis of Thin Shell Structures.
Contribution K4/5, 2nd Int. Conf. Structural Mechanics in Reactor Technology,
Berlin 1973.

[11] SCAVUZZO, R. J., RAFTOPOULOS, D. D., BAILEY, J. L.: Application of
Three-dimensional Normal Mode Theory to Seismic Analysis of Nuclear
Power Plants. Contribution K 3/4, 2nd Int. Conf. Structural Mechanics
in Reactor Technology, Berlin 1974.

TABLE I. ELECTRONICS COMPARTMENT - SEISMIC RESPONSE
MAXIMUM VALUES OF STRESS AND STRESS RESULTANTS

Seismic motion	Method of analysis	Normal force [N]	shear force [N]	Bending moment [Nmm]	Torsion moment [Nmm]	Normal stress [N/mm²]
horiz. x	root sum	173.6	115.5	43547	26330	7.8
	abs. sum	299.8	146.9	49727	30077	9.5
	block sum	199.0	116.7	43625	26389	8.1
	static	204.4	128.0	43819	26546	7.5
horiz. y	root sum	597.8	193.3	56849	30038	6.7
	abs. sum	608.4	196.4	57771	41830	7.0
	block sum	597.9	193.3	56849	30038	6.7
	static	591.2	230.4	67750	3449	7.2
vertical	root sum	12.8	4.1	1216	64	0.15
	abs. sum	14.0	4.5	1342	110	0.16
	block sum	12.8	4.1	1218	66	0.15
	static	181.0	133.5	18742	544	2.26

TABLE II. TRANSMITTER RACK - AIRCRAFT IMPACT RESPONSE
MAXIMUM VALUES OF STRESSES AND ACCELERATION

Foundation motion	Modal Super-pos.	Normal force [N]	Bending moment [Nmm]	Normal stress [N/mm^2]	Acceleration [mm/s^2] horiz. x	horiz. y	vertical
horiz. x	root sum	1535.4	318958	108.3	9336	1178	1170
	abs. sum	1549.4	321979	110.3	11479	1852	1213
horiz. y	root sum	1147.3	995136	52.9	1579	29366	586
	abs. sum	1294.4	1146082	61.0	4420	48109	1215
vertical	root sum	11.2	9752	0.1	9864	7353	26794
	abs. sum	14.7	12550	0.1	14877	16186	33810

TABLE III. COMPUTATIONAL DETAILS OF TYPICAL EQUIPMENT TYPES

Type of equipment	Finite element idealisation no. dof.	no. elem.	Type of analysis	Program	Computer	Central processor time [s]
6kV switchgear	1172	404	static	ASKA	370-158	516
Low voltage compartment	658	219	8 normal modes	ASKA	370-158	785
			8 normal modes	NASTRAN	6600	496 (ARU's)
Electronics compartment	730	218	8 normal modes	ASKA	370-158	1193
			8 normal modes	ASKA	370-168	340
			8 normal modes	NASTRAN	6600	564 (ARU's)
			part. factors and superpos.	ASKA	370-158	390
Transmitter rack	636	140	21 normal modes	ASKA	370-168	549
			part. factors and superpos.	ASKA	370-168	44

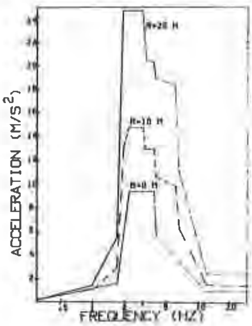

FIG. 1 REQUIRED FLOOR RESPONSE SPECTRUM FOR SEISMIC MOTION

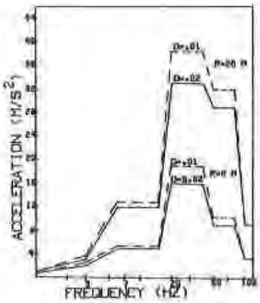

FIG. 2 REQUIRED FLOOR RESPONSE SPECTRUM FOR AIRCRAFT IMPACT

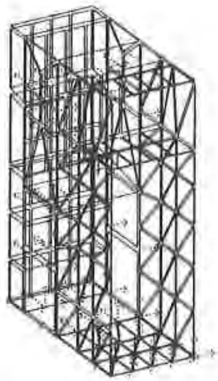

FIG. 3 6KV SWITCHGEAR COMPARTMENT FINITE ELEMENT IDEALISATION

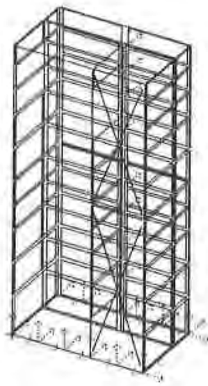

FIG. 4 LOW VOLTAGE COMPARTMENT FINITE ELEMENT IDEALISATION

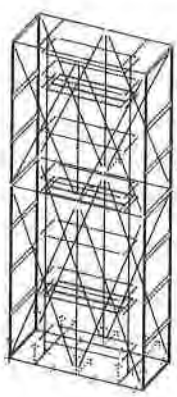

FIG. 5 ELECTRONICS COMPARTMENT FINITE ELEMENT IDEALISATION

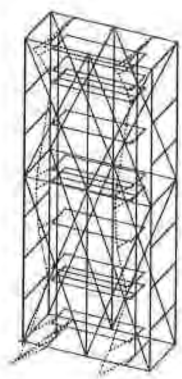

FIG. 6 ELECTRONICS COMPARTMENT CALCULATED MAXIMUM STRESSES

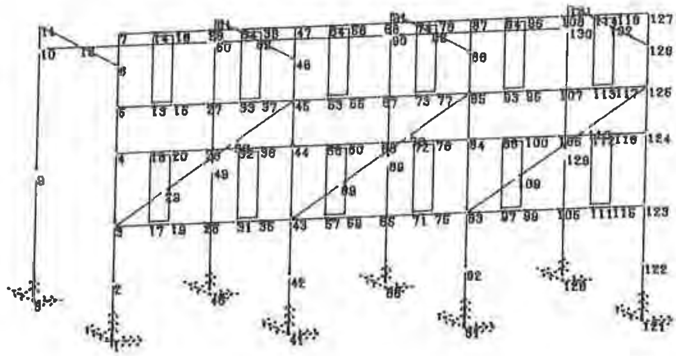

FIG. 7 TRANSMITTER RACK
FINITE ELEMENT IDEALISATION

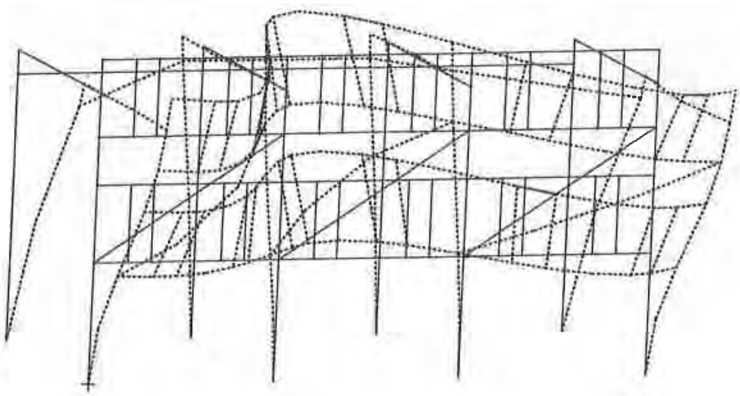

FIG. 8 TRANSMITTER RACK
MODE SHAPE OF 5TH NATURAL FREQUENCY

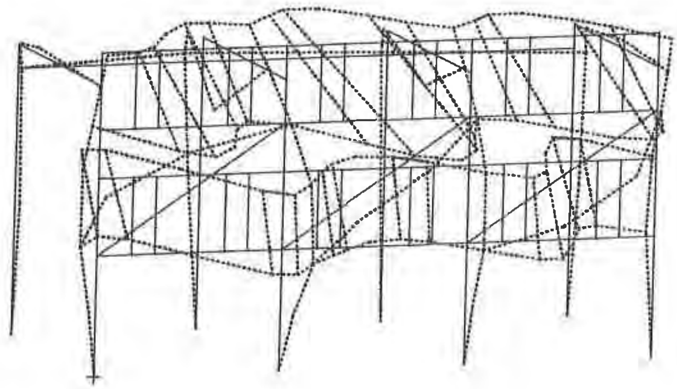

FIG. 9 TRANSMITTER RACK
MODE SHAPE OF 20TH NATURAL FREQUENCY