THE UNIVERSITY OF ASTON IN BIRMINGHAM

Measurement of Acoustic Parameters in Room at Low Frequency

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SUMMARY

One of the requirements of the transmission suite and sound studios is that the sound in them should be diffuse, but since in the low frequency range the problem of poor diffusion arises, that consequently leads to discrepancies in the transmission loss measurement at various facilities. This thesis attempts to show how to achieve adequate experimental accuracy without the need for a very large room. Considerable attention has been given to the problem of understanding the performance of sound on the low frequencies range.

As far as the sound and recording studios are concerned, with only a few room modes in a measurement band at low frequencies, there tends to be an irregular room response. This can change the original tonal qualities of music and also the intelligibility of speech. To avoid these problems it is necessary to treat the surfaces of the studios acoustically.

The modal parameters including the natural frequencies, damping constants of a resonant mode are measurable quantities and can therefore be employed as parameters for judging the acoustic performance of a room where a particular absorbing surface treatment is applied to its boundary.

In the thesis a small room is considered and the behaviour of sound in the low frequency range for various source and receiver positions has been obtained experimentally and is then compared with the theoretical analysis. The agreement and the disagreements between the coupling of the modes is applied to predict anamolies in transmission loss measurements, the design of small acoustic studios and the measurement of sound absorption coefficient.

Low, Frequency, Measurement, Parameter, Room.

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CHAPTER 1

Introduction

There always seem to be problems as far as measurements at low frequency are concerned and this has lead to part of the research in acoustics being concentrated in this field.

The aim of this project is to study the behaviour of sound at low frequency within a transmission suite, and the characteristics of the transmission suite and small studios.

The measured transmission losses of single panels at low frequency range sometimes differ from the values obtained when the mass law is applied. The discrepancy is shown to depend on some properties of the individual room in which the measurements of the transmission loss are made rather than the properties of the panel itself(1), since it has been known for some time that the transmission loss of a single wall depends on its mass(2): in fact, within a certain frequency range the transmission loss follows the mass law and depends at a particular frequency only on the mass of the wall where the high order resonances in the panels are not considered to be important. The size of the discrepancy implies that care should be taken when comparing low frequency results obtained in different measuring facilities. The cause of the discrepancy was first attributed to some property of the wall, but the results obtained later showed that this discrepancy was not entirely due to the wall properties, but to some extent was dependent on some properties of the transmission suite where the measurements were taken. Therefore, the characteristics of the transmission suite influence the measured values of the transmission loss at low frequency range. It is also evident that when the reverberation times of the transmission suite are changed, this did not alter the measured transmission loss(1). It has been shown that when the transmission losses of single panels are measured, in some facilities there is a difference from the mass law form which depends on the particular transmission loss facility used, and it has not been possible to determine which room factor is causing this discrepancy.

As far as the small studio is concerned, the aim of this project and also the areas of investigation will be as follows:-

i) The behaviour of sound at low frequencies will be investigated with the purpose of identifying the modes and related coupling and damping.

ii) The experimental results will be compared with the theoretical results.

iii) The combination of the experimental results and theoretical analysis will be used to predict the modal frequencies, coupling and damping for any position of the microphone.

One of the requirements of the transmission suite is that the sound field in it should be diffuse, since there are certain limitations for analysing real physical situations. The mathematics can only be applied to certain ideal conditions exemplified in mathematical models, and this limitation affects most of the acoustical test procedures.

i) Absorption measurement in a reverbertion room requires a diffuse sound field (3), both before and during sound decay, so that the mean free path that appears in the term that governs the decay rate can be defined. It is also true when designing studios that a diffuse sound field is required.

ii) Transmission loss measurement, both in the laboratory(4) where special devices are provided to enhance the diffusion and in field measurements(5), where the conditions found in the field are acceptable. In both cases the transmission loss is defined in terms of the ratio of energy incident upon a partition to that transmitted through and radiated away on the other side: diffuse field is needed to calculate the immeasurable energy incident on the test specimen from the results of a measurement of the space/average sound pressure level in the source room and also to calculate the energy radiated away from the partition from the measured space/average sound pressure level in the receiving room.

iii) In determining the power output of a sound source from a steady-state measurement of space/average pressure throughout a room, and also for a steady-state measurement of room absorption, a diffuse sound field is required to relate the power level of the source of the total absorption of the room and the average sound pressure level in the room (9,10).

In most cases, an attempt has been made to make the test environment resemble an ideal physical situation corresponding to a mathematical model that one's analytical abilities can cope with. This permits an immeasurable quantity, usually power or energy flow, to be deduced from relatively straightforward measurements of pressure.

The most straightforward way of achieving diffusion in a room is to make the room large compared to the wavelength of the sound interest. As frequency increases, the number of room-resonance modes within a measurement bandwidth increases rapidly and tends to fill both the spectrum of frequency and of angle of incidence. The more modes there are, the higher the frequency in a given room, the more nearly diffuse the sound field. It follows, therefore, that the problem of poor diffusion is most troublesome at low frequencies. With only a few room modes in a measurement bandwidth, there tends to be an irregular room response, with only a few angles of incidence for energy impinging on each room boundary.

An alternative criterion of adequate diffusion may be formed in terms of the least permissible number of room modes in the measurement bandwidth: values of 10-20 modes have been suggested. In a statistical sense, the number of modes in a bandwidth is dependent only on the volume of the room, the frequency and the width of the band.

It is possible to achieve adequate diffusion at very low frequencies according to this criterion by building an enormous room. The drawbacks to this approach, however, are the prohibitive cost and the increasing contribution of air absorption to the total of room absorption as the room volume substantially increases. The latter condition tends to violate Sabine's necessary assumption of negligible energy loss during a mean free path transit for the validity of the reverberation equation. In order to achieve adequate diffusion without the need for a very large room therefore, considerable experimental attention has been given to the problem of enhancing diffusion at low frequencies.

Both Bolt (6,7) and Sepmeyer (8) have calculated the modal frequencies of various rectangular rooms, regarding the room modes as uncoupled. Their studies show that certain room proportions favour a uniform distribution of resonances along the frequency axis - which is said to lead to a more uniform transfer function in the room - and a uniform distribution of angles of incidence of energy upon each boundary.

These studies, however, are appropriate only for empty, perfectly rectangular rooms, or for rooms with, at most, only slight perturbations of the rectangular shape (7). In practice, the inaccuracies of room construction and the addition of furniture and of special diffusing devices modify these calculated distributions of frequency and incidence angle sufficiently to mean that the probable benefits of trying to optimize room proportions by theoretical means do not justify the great deal of calculated effort, even with high speed computers. Moreover, there are experimental and theoretical reasons to suspect that real rooms, in which substantial coupling exists between the room modes, do not behave in the way that the Bolt and Sepmayer analyses predict for uncoupled rooms.

Until the turn of the 19th century, the design of rooms and halls for public speaking and music did not have any scientific basis. In fact, the problems involved had not even been clearly defined.

W.C. Sabine among others did his pioneer work on the practice of acoustic design when he was consulted about the Fogg Art Museum of Harvard University.

The theory of absorption by porous materials had been discussed by Rayleigh, but it was Sabine who introduced measuring techniques for determination of the absorption.

The relation $RT = \frac{KV}{A}$, expressing the reverberation time in terms of the room volume and the total absorption of its bounding surfaces, was first deduced by him and it was he who adopted the definition for the reverberation time as the time taken for the energy density to fall to the minimum audible value from an initial value of a million times greater, i.e. a range of 60dB. The relative British Standard (B.S.3638) specifies that R.T. should be measured over a range of only 30dB, starting at a level 5dB lower than the initial steady state. In deriving this formula, Sabine assumed that the sound field was diffuse, i.e.

i) There is uniform total (potential plus kinetic) energy density at all points in the room and each volume element radiates equally in all directions.

ii) There is equal probability of energy flow in all directions and random angle of incidence of energy upon the boundaries of the room.

iii) A diffuse sound field compromises a superposition of an infinite number of plane progressive waves, such that all directions of propagation are equally probable and the phase relations of the waves are random at any given point in the space.

Sabine's empirical formula exhibits limitations when applied to 'live' rooms and it was Eyring (1930) who was prompted to bring modification to it. His result was:

$$RT = \frac{KV}{-S\, Ln(1-\bar{a})}$$

where a is the average absorption coefficient of all the surfaces He assumed the acoustic behaviour of the room to be determined by the average absorption coefficient and considered a wave front travelling a fixed distance (the mean free path) between successive reflections at which it lost the same fraction of its incident energy.

A further modification of Eyring's formula was suggested by Millington, who considered the room to be characterised by the behaviour of a specific sound ray after successive reflections at surfaces whose absorption coefficients varied. The term of his formula is:

$$RT = \frac{KV}{\Sigma S_i \ln(1-\alpha_i)}$$

These formulae were the basis for what is termed the classical statistical and gometrical theory which treats the sound in an enclosure as a field through every point, of which a large number of waves reflected from bounding surfaces are passing in a manner similar to light rays.

An alternative approach (22), that of the wave theory, is based on the fact that the air space contained within an enclosure acts as a vibratory system which is excited by a sound signal.

The wave theory (22) of room acoustics was developed by P.M. Morse and R. Bolt in the 1940s from one dimensional experiment first suggested by Rayleigh in 1878.

Kuundsen (23) first showed experimentally that the reverberant sound has the characteristic frequencies of normal modes of vibration of the room and not necessarily the the frequency of the sound source.

Wente (23) studied the steady state response of a room as a function of the frequency of the source and pointed out that the sharp response peaks occurred at the resonant frequencies of the room model.

Several other theoretical and experimental workers have studied other aspects of wave acoustics. With the application of the statistic approach, extended to wave acoustics by Schroeder (24), many of the properties of frequency and space response in rooms can be studied.

The reverberation time, as a parameter of judging the acoustical qualities of rooms, has been used extensively in the design of enclosures. Beranek (25) questioned the application of such a criterion to modern auditoria and studios because of the differences in shape between the rooms which Sabine studied for the modern design of rooms. he enlists other quantities which can be measured and used as indices for judgement for the acoustic properties of enclosures. Among these are the fluctuations of sound during decay, the variations of sound pressure with position of source and receiver, the variations with frequency of the source, the ratio of direct to reverberant sound and modification of wave shape during transmission from one point to another. He concludes that each gives some useful information about the performance of the enclosure.

In the course of searching for physical measurable indices of the acoustic qualtiy of a room, in addition to the reverberation time, a number of studies have been made of the fluctuations occurring in the sound pressure level when the frequency of a simple harmonic source is varied or when the position of the source or receiver is varied.

The theory of frequency irregularity originally given by Bolt and Roop (25) has been revised and extended by Schroeder (25). Bolt, Doak and Westervelt (26) commented that the reverberation time is not a completely adequate index of the acoustical quality of a room; implicit in the concept of reverberation time is the assumption that the room reaches a steady state diffuse condition. In practice, most sounds of speech and music can be classified as pulsed wave trains whose amplitude and frequency components fluctuate rapidly so that the room seldom reaches steady state.

E. Meyer and H. Kuttrf (26) point out that the reverberation time, which is based on the statistical room acoustics, involves only the volume of the room and its sound absorption but not its shape, and the number of echoes, the sequence in time and intensity are dependent on the dimensions and shape of the room and the position of the sound source and the receiver.

Knusden and Harris (28) emphasize the importance of the principles of physical acoustics in determining the location of absorptive materials and studio dimensions. They stress that physical acoustics seem to provide a logical and practical guide to the problems of studio design, but conclude that due to the complexity of the approach it is not yet feasible to present a complete set of practical design formulae based on physical acoustics.

According to H. Kuttruff (29), the wave theory of room acoustics can only be applied exactly in highly idealized cases. The method cannot be applied to such types of room as concert halls or theatres. Since they are irregular in shape, largely because of the furniture, the boundary conditions cannot be formulated in a satisfactory way, and so the theory can only yield approximate or qualitative results. The immediate practical application of the wave theory to room acoustics is therefore very limited.

1.1 Design of Studios

General Considerations

It is most important that from the start of the design of a studio centre or wherever the insulation or acoustics are likely to be of importance, the advice of an acoustic consultant be sought. Too many mistakes are still being made in the basic design of buildings which result in serious shortcomings in the sound insulation or the internal acoustics. Backus (30) quotes an example, as recent as 1967, of an auditorium built with an elliptic plan form against the advice of the acoustic consultant, which had to be heavily treated with sound absorbers to suppress the inevitable echoes and which, on completion, was quite unsuitable for music.

In the design of all rooms for speech and music, the basic requirements are (28):

1) To preserve the original qualities of a musical tone such that the relative magnitude of all harmonics remains unaltered as they are transmitted.
2) To ensure the intelligibility of speech.

The steps that must be taken in planning a new studio centre, or a studio in an existing building, are as follows (31):

a) The exclusion of intruding noise and vibrations by a high degree of insulation
b) The studio centres should be planned, as far as possible, to

separate the technical and studio areas acoustically from public or service areas, and from offices, plant rooms, stores or work shops.

c) Within a studio building, music studios should be separated by less sensitive areas or by corridors and if this is not possible provision should be made between the areas for fairly massive multi-layer walls that will give up to 70dB mean sound level reduction.

d) Space must be allowed for floating the studio against the structure-borne noise transmitted from other parts of the building and, where necessary, from traffic vibrations. The inter-area sound insulation must also be considered. In general, each studio is separated from its neighbours by a structural wall forming part of the main solid shell of the building and has an independent inner wall integral with the floated floor. Space must be allowed within the storey height for a false ceiling containing low frequency sound absorbers and services.

e) Space must be allowed for quiet ventilation systems serving all technical areas.

1.2 Use of Scale Models in Planning and Design of Studios

The use of scale models for acoustic investigations originated more than half a century ago. The development of recording opened up the possibility of using scale models as actual auditoria to add reverberation to a non-reverberant programme. When a scale model of a studio is constructed using the same materials and all dimensions are exactly one nth of the original, the conditions for acoustic similarity are fulfilled when the time scale is reduced in the same ratio as the linear dimensions. This means that all frequencies must be increased in the ratio of n:1, then all wave lengths are in the same relationship to the model interior, and the reverberation times are unchanged relative to the altered time scale.

The absorption coefficients of all the materials must have the same values at the scaled up frequencies as those of the studio materials at the original frequencies. By measurement in a scaled down reverberation room it is possible to build up a collection of scaled absorbers equivalent to those in actual studio use.

The only important property that is not correctly scaled is the absorption of the air. It has been proved that the air absorption index changes rapidly with frequency, and becomes excessive in relation to the scheme of scaling. To overcome this, it is necessary to dry the air so as to reduce its absorption. Objective tests have been carried out on models and it has bee proved that it is possible to measure the reverberation characteristics, impulse response, transmission characteristics etc. and, in fact, make all the normal acoustic measurements.

The first use of a scale model for subjective listening tests was by Spandok in 1934, who used a microphone with variable speed to effect the necessary frequency transformation.

The most intensive work has been carried out by a team at the Technische Hochschule at Munich, led by Spandok (1965). Most of the major problems were solved where differences in the acoustic treatment in a model could be detected. The loudspeakers were electrostatic, of 25mm diameter. They gave a usable characteristic from 400 to 500Hz, and up to 90 of them were used to produce a high enough sound level and to stimulate an extended source such as an orchestra.

1.3 Wave Modal Theory

The behaviour of the sound formed by the vibration of the air inside an enclosure can be analysed by considering the physical properties of the wave phenomena in an enclosure. Comparing this method with the statistical and geometrical analyses reveals the basic characteristic of the sound phenomena in the enclosure.

The volume of air inside an enclosure is a complex vibratory system, consisting of a combination of a number of simple systems. When a sound source is switched on inside an enclosure, it sets up a vibratory process which has a steady state vibration, having the frequency of the source together with a transitory free vibration composed of the various normal modes of vibration, and as this fades away a steady state situation is reached in the enclosure. The vibration of this steady state situation can be expressed as the sum of a large number of standing waves whose amplitude depends on the frequency of the source, the impedance of the standing wave with respect to the frequency and position of the source in the room.

When the source is switched off, the system is not in balance any more, and is carrying out only its normal mode of vibration, which has the form of standing waves, which gradually decay according to the exponential law which is common to all forms of vibration. When the source is operated continuously, each component frequency of the transient dies out exponentially at its own particular rate, leaving only the steady-state vibration.

As these waves damp out exponentially according to their individual free-vibration properties, they often interfere with one another and thus produce beat notes.

Experience shows that in normal enclosures the phenomenon of reverberation is observed at any frequency of the exciting sound, which is clearly only possible if these enclosures have a great many characteristic frequencies close to one another, and an infinite number of characteristic frequencies is associated with an equal number of vibratory systems with evenly distributed constants.

The volume of air in an enclosure is a complex vibratory system with distributed parameters, a system that when excited by a sound impulse carries out its own gradually decaying vibration mode.

The characteristic frequencies of vibration of the standing waves in a room depend primarily on the shape and size of the room, whereas their rates of damping depend chiefly on the boundary conditions.

Therefore, there are three areas that are open for investigation:

1) The behaviour of the sound wave in relation to the boundary conditions of the enclosure (in the case of a room, these are the surfaces of the room).

2) The nature of the steady-state response of the room to a source of sound.

3) The nature of the transient response, in particular reverberation.

To study the behaviour of a sound wave at the surfaces of the room, the simplest boundary conditions, i.e. perfectly rigid walls and no damping, are considered in deriving the expression for the characteristic frequencies.

1.4 Behaviour of Sound in a Rectangular Enclosure

Consider a rectangular enclosure (a room) in which the sound field is non-uniform and in which the absorption of energy is negligible.

With a sound source situated in one corner of the walls, sound waves are formed in the enclosure and, when they reach the walls of the room, they are reflected. The source sets into motion one or more of the normal modes of vibration of the air inside the enclosure, and intensity is not inversely proportional to the square of the distance from the source; in some rooms the intensity at some point far from the source is greater than the intermediate points. The intensity of sound in a room is not simply related to the power radiated by the sound generator. The mechanical behaviour of a loudspeaker, its mechanical and electrical impedance and total power radiated is practically unaltered by the properties of the room, but the intensity of the sound produced and the distribution of this intensity will be greatly altered.

The Sound Field in Rectangular Rooms with Rigid Boundaries

The Wave Equation

During the passage of acoustic waves, there is no rotational motion of the particles and so the velocity vector of a particle is an irrotational vector which may be represented as the gradient of some scalar potential function (Lord Rayleigh: The Theory of Sound Vol.2 (10) DOVER pub.).

i.e. $V = \nabla\phi$ (i.e. $U = \frac{\partial\phi}{\partial x}$, $V = \frac{\partial\phi}{\partial y}$, $w = \frac{\partial\phi}{\partial z}$ in component terms)

where $\phi(x,y,z,t)$ is the velocity potential

The propagation of the acoustical waves is governed by the wave equation:

$$\frac{\partial^2\phi}{\partial t^2} = c^{-2}\nabla^2\phi \qquad 1.1$$

where c = speed of sound, ∇ the laplacian operator which represents the propagation of the velocity potential (ϕ). The wave equation expressed in terms of the pressure in cartesian co-ordinates is:

$$\frac{\partial^2\rho}{\partial x^2} + \frac{\partial^2\rho}{\partial y^2} + \frac{\partial^2\rho}{\partial z^2} = \frac{1}{c^2}\frac{\partial^2\rho}{\partial t^2} \qquad 1.2$$

For a rectangular room whose sides have the lengths Lx, Ly, Lz, the general expression for a plane wave is:

$$\rho = Ae^{-j(wt - k_x X - k_z Z)} \qquad 1.3$$

If this equation is to satisfy the general wave equation, the constant kx, ky and kz must satisfy:

$$K = \frac{w}{c} = (K_x^2 + k_y^2 + k^2 z)^{\frac{1}{2}} \qquad 1.4$$

as can be shown by direct substitution of equation (1.3) into equation (1.2). By replacing any one, or two, or all three of the negative signs in equation (1.3) by positive signs, we obtain seven additional expressions similar to equation (1.3), all of which have identical values of kx, ky and kz. This array of eight expressions represents the family of plane waves generated by the original wave as successive reflections of the six boundary surfaces of the room.

The Boundary Conditions

The origin of the co-ordinate system is chosen at the corner of the room Fig.(1.0) and the room extends from

$x = 0$ to $x = Lx$ in the x direction

$y = 0$ to $y = Ly$ in the y direction

$z = 0$ to $z = Lz$ in the z direction

Since the room surfaces are assumed to be rigid, the velocity of the air particles near any surface must be parallel to that surface, i.e. the normal component of the particle velocity must vanish.
This is equivalent to:

$u = 0$ for $x = 0$ and $x = Lx$

$v = 0$ for $y = 0$ and $y = Ly$

$w = 0$ for $z = 0$ and $z = Lz$

where u,v and w are the components of the particles velocity vector in the x,y,z directions respectively.

From the equation of motion (Rayleigh: The Theory of Sound Vol.2), the pressure is:

$$P = - e \frac{\partial \phi}{\partial t} = -jw\rho\Phi \qquad 1.5$$

where is the density of the medium

The component particle velocity is by definition:

$$U = \frac{\partial \phi}{\partial n} = - jK_n \phi \qquad 1.6$$

where n denotes a direction normal to the surface

The particle velocity in the x direction can be obtained from equations (1.5) and (1.6) as:

$$U = - \frac{1}{jw\rho} \frac{\partial \rho}{\partial x} \qquad 1.7$$

with similar expressions for v in the y direction and w in the z direction.

1.5 Characteristic Values and Characteristic Functions

Application of the boundary conditions at x=0, y=0 and z=0 to the respective equations for particle velocities obtained from the array of the eight expressions represented by equation (1.3) results in the standing wave equation:

$$P = P(\cos k_x x \cos k_y y \cos k_z z) \qquad 1.8$$

Substituting equation (1.8) into equation (1.7), the particle velocity in the x-direction becomes

$$U = -\frac{k_x p}{jwp} (\sin k_x X \cos k_y Y \cos k_z Z) e^{-jwt} \qquad 1.9$$

which is zero for x = 0.

If the additional condition of u=0 at x=Lx is applied to equation 1.9, then:

$$\sin k L = 0$$

or

$$k_x = \frac{n_x \pi}{L_x} \quad (n_x = 0, 1, 2, 3) \qquad 1.10$$

with similar expressions for k_y and k_z.

$$K_y = \frac{n_y \pi}{L_y} \quad n_y = 0, 1, 2, 3$$

$$K_z = \frac{n_z \pi}{L_z} \quad n_z = 0, 1, 2, 3$$

On substitution of these allowed values for k_x, k_y and k_z in equation 1.8

$$Pn(x, y, z, t) = P \cos\left(\frac{n_x \pi x}{L_x}\right) \cos\left(\frac{n_y \pi y}{L_y}\right) \cos\left(\frac{n_z \pi z}{L_z}\right) e^{-jwt} \qquad 1.11$$

i.e. a general expression for the characteristic function describing the pressure in any of the standing waves (as denoted by the subscript: n) inside a rigid walled rectangular enclosure.

Considering the decaying characteristic of the vibration which is caused by absorption at the boundaries of the enclosure:

$$Pn(x, y, z, t) = P \cos(\frac{n_x \pi X}{L_x}) \cos(\frac{n_y \pi Y}{L_y}) \cos(\frac{n_z \pi Z}{L_z}) e^{-jwt-\delta nt} \qquad 1.11a$$

The component characteristic values k ,k and k in this expression are limited to the values given by the equation 1.10. This limitation in turn restricts the characteristic frequencies corresponding to the allowed normal modes of vibration as obtained by substituting in equation 1.4:

$$fn = \frac{W_n}{2\pi} = \frac{c}{2} \left((\frac{n_x}{L_x})^2 + (\frac{n_y}{L_y})^2 + (\frac{n_z}{L_z})^2 \right)^{\frac{1}{2}} \qquad 1.12$$

From equation 1.12 it can be deduced that the natural frequencies of the enclosure are defined by its parameters L_x, L_y, L_z and by the number of standing waves which arise in direction x,y,z. Considering equation 1.12, it can be seen that if two or three dimensions of the enclosure are equal, or if their lengths are multiples of one another, coincidence of characteristic frequencies may result. Thus, if the dimensions of the enclosure are identical ($L_x=L_y=L_z$), identical values of characteristic frequency are obtained where $n_x=1$, $n_y=1$, $n_z=1$, while if these dimensions are all different there are three of these frequencies.

A particular normal mode of vibration corresponding to any set of values of n ,n and n can be produced by starting a wave in a direction given by any combination of the direction cosines $\pm \frac{k_x}{k}$, $\pm \frac{k_y}{k}$

and $\pm \frac{k_z}{z}$ and letting it be reflected from the various room surfaces until it becomes a standing wave, and when the wave produced is not in the path of the direction cosines mentioned and frequency, the reflected waves interfere with each other in a non periodic manner, and consequently no regular pattern of standing wave is set up.

Various types of waves can occur in a rectangular room.

The waves with none of the n's being zero are termed oblique waves.

The waves with one n being zero are tangential waves.

The waves with two n's being zero are axial waves.

Axial waves are made up of two travelling waves propagated parallel to an axis and striking only two walls.

Tangential waves are built up of four travelling waves, reflecting from four walls and moving parallel to two walls.

Oblique waves are built up of eight travelling waves reflecting from all six walls.

Therefore the standing waves can be separated into three categories or seven classes:

Axial waves (for which two n's are zero)

x-axial waves parallel to the x-axis ($n_y, n_z = 0$)

y-axial waves parallel to the y-axis ($n_x, n_z = 0$)

z-axial waves parallel to the z-axis ($n_x, n_y = 0$)

Tangential waves (for which one n=0)

y,z-tangential waves, parallel to the y-z plane ($n_x = 0$)

x,z-tangential waves, parallel to the x-z plane ($n_y = 0$)

x,y-tangential waves, parallel to the x-y plane ($n_z = 0$)

Oblique waves (for which no n is zero)

Each of these three types of waves has different properties; even to the first approximation, waves of different classes have different decay rates and accordingly different reverberation times.

From equation (1.11a) it follows that normal modes of vibration of the air space of an enclosure are characterised by a complex interweaving of standing waves and decay of the various waves which comprise these vibrations can proceed with varying speeds defined by the index of decay .

Oblique waves decay faster than the others because they have the shortest mean free path and undergo the largest number of reflections in each second. The next to dècay are the tangential and, finally, the axial waves. The sound field, which is fairly uniform and diffuse at the first instant after the source has been switched off, becomes progressively less uniform as the oblique and tangential waves decay. This is because after the decay of oblique waves, the randomly distributed waves are increasingly dominated by the more orderly tangential or axial waves distributed parallel to surfaces and to the edges of the surfaces respectively.

A knowledge of the characteristic frequencies of a room is essential to a complete understanding of its acoustic properties, for the room will act as a resonator and respond strongly to those impressed sounds having frequencies in the immediate vicinity of any of its characteristic frequencies, as might arise in a room situation where two different modes of vibration have the identical characteristic frequency, which would result in the room responding strongly to the particular characteristic frequency. It is just the characteristic frequency that affects the output of a loudspeaker, as measured in a reverberant room, and causes the results to have limited significance as a true criterion of the speaker's properties.

Every room or enclosure superimposes its own characteristics on any sound source present and this results in concealing the true output characteristics of the source, since the fluctuation of the sound pressure level occurs when the microphone position is moved from one position to another, or the frequency of the source is varied, and it is for this reason that measurements of the response curves of loudspeakers should be carried out either in open air or in a specially constructed dead room, known as an anechoic chamber.

Each normal mode of vibration of an enclosure can only be excited to its fullest extent when the sound source is located at the position where the particular standing wave pattern has a maximum pressure amplitude.

Equation (1.8) indicates that the pressure amplitude of all patterns of standing waves in a rectangular enclosure are at maximum in the corner of the room, so positioning the source at the corner of a room results in exciting all the allowed mode of vibration to its fullest extent. Correspondingly, if the microphone is located in the corner of the room, it measures the peak sound pressure for every normal mode of vibration that is excited.

If the loudspeaker is positioned in the centre of a rectangular room, only the modes of vibration with simultaneous even numbers for n_x, n_y, n_z are excited.

1.6 The Sound Field in Rectangular Rooms with non-Rigid Boundaries

1.6.1 The Boundary Condition

When the surfaces of the walls are hard, to derive the standing wave equation, an index of decay that is expressed as an exponential factor $(e^{-\delta t})$ and is the δ of absorption at the boundaries of the enclosure is taken into considertion, but it is assumed that the wave shape is not changed.

Where the walls are not completely rigid, the normal components of the particle velocity at the wall surfaces is not zero, and this affects the shapes and natural frequencies of the standing waves and therefore it must be taken into consideration.

The boundary conditions for the rigid walls that result in velocity components normal to the walls being equal to zero must be replaced in the case of the non rigid walls by the more general condition of the complex ratio of the pressure to the nromal component of the particle velocity at the wall surface which is equal to the acoustic impedance of the wall.

1.7 Characteristic Waves and Characteristic Functions

Assuming as before a harmonic law for the acoustic variables, the wave equation (1.1), when expressed in terms of the pressure wave, can be written as:

$$\nabla^2 P + K^2 P = 0 \qquad 1.15$$

This equation gives non-zero solutions fulfilling the new boundary conditions only for the particular discrete values of k_n known as the characteristic values and denoted by the symbol k_n, and it is contained in the acoustic impedance since the impedance depends, in general, on the frequency. It also depends on the size of the room and the shape and absorptive properties of the walls.

The pressure distribution in a single standing wave is expressed by:

$$p(x, y, z, t) = \psi_n e^{-jwt} \qquad 1.16$$

where

$w = \frac{K}{c}$ is the angular driving frequency in forced vibrations, or

$w = w_n = \frac{k_n}{c}$ is the angular characteristic in free vibrations

and

ψ_n is the characteristic function of the standing wave, being the solution of the wave equation which satisfies the proper boundary conditions.

For the rigid-walled condition, the characteristic values - K_n - are real and the characteristic function (ψ_n) are expressed from equation (1.8) as:

$$\psi_n = \cos k_x x \cos k_y y \cos k_z z \qquad 1.17$$

where the component characteristic values k_x, k_y and k_z are given by equation (1.10).

If the walls are not rigid, the characteristic values are complex ($k_n = \frac{w_n + \delta_n}{c}$) and so are their co-characteristic components:

$k_x = \frac{w_x + j\delta_x}{c}$ with similar expressions for k_y and k_z

Also some phase constants should be included in the arguments of the cosine functions of equation (1.17), which, together with the new characteristic components, will be determined by solving for the boundary conditions.

Equation (1.17) is, thus, equivalent to:

$$\psi_n = \cos (k_x x + \phi_x) \cos (k_y y + \phi_y) \cos (k_z z + \phi_z) \qquad 1.18$$

where

$\emptyset_x$, $\emptyset_y$ and $\emptyset_z$ are some phase constants.

1.8 Resonance Frequencies and Damping Constants

It is possible to compute the component characteristic values k_x, k_y, k_z in terms of the specific acoustic admittances β_x, β_y and β_z of the room surfaces.

Since these component characteristic values can be represented as:

$$k_x = \frac{w_x + j\delta_x}{c} \qquad k_y = \frac{w_y + j\delta_y}{c} \quad \text{and } k_z = \frac{w_z + j\delta_z}{c}$$

and are related as

$$k_n = (k_x^2 + k_y^2 + k_z^2)^{\frac{1}{2}}$$

therefore

$$\left(\frac{w_n + j\delta_n}{c}\right)^2 = \left(\frac{w_x + j\delta_x}{c}\right)^2 + \left(\frac{w_z + j\delta_z}{c}\right)^2 \qquad 1.27$$

and equating the real and imaginary parts:

$$w_n^2 - \delta_n^2 = (w_x^2 + w_y^2 + w_z^2) - (\delta_x^2 + \delta_y^2 + \delta_z^2) \qquad 1.28$$

and

$$\delta_n = \frac{\delta_x w_x}{w_n} + \frac{\delta_y w_y}{w_n} + \frac{\delta_z w_y}{w_n} \qquad 1.29$$

In most cases of practical interest (P.M. Morse & R.H. Bott (12) Sound waves in rooms, Rev.Modern Phys. Vol.16 (1944)), the damping constant δ_n is much smaller than the resonant frequency w_n, and the term (δ_n^2) in equation (1.28) can be ignored.

$$w_n = (w_n^2 - \delta_x^2) + (w_y^2 - \delta_y^2) + (w_z^2 - \delta_z^2) \qquad 1.30$$

Equation (1.29) can be written as:

$$\delta_n = \frac{1}{2w_n} \left(2\delta_x w_x + 2\delta_y w_y + 2\delta_z w_z\right) \qquad 1.31$$

The characteristic frequency (w_n) and the damping constant (δ_n) are the quantities usually determined experimentally rather than their components w_x, w_y, w_z and δ_x, δ_y, δ_z and then for their computation in terms of the boundary conditions it is more suitable to obtain values for the quantities $(w_x^2-\delta_x^2)$, $(w_y^2-\delta_y^2)$, $(w_z^2-\delta_z^2)$ and $(2\delta_x w_x)$, $(2\delta_y w_y)$, $(2\delta_z w_z)$, and these can be obtained from the transformation of k^2_x, k^2_y and k^2_z in terms of the specific acoustic admittance of the respective pairs of walls.

$$k^2_x = \left(\frac{w_x + j\delta_x}{c}\right)^2 = \frac{(w^2_x - \delta^2_x)}{c} + 2j\,\frac{\delta_x w_x}{c^2} \qquad 1.32$$

$$\text{i.e.} \quad c^2k^2_x = (w^2_x - \delta^2_x) + j^2\delta_x w_x \qquad 1.33$$

the quantities $(w^2_x - \delta^2_x)$ and $2\delta_x w_x$ are the real and imaginary parts of $(c^2k^2_x)$

To the first approximation, these results can be expressed in terms of the real and imaginary parts of the specific acoustic admittance (β), the specific acoustic conductance being denoted by (γ) and the specific suseptance as (α), so that $\beta(w)=y+j\gamma$.

1.9 The Steady-State Response of the Room

So far the characteristic values and functions have been considered and methods have been derived to compute their values in terms of acoustic admittance of the boundary surfaces and the driving frequency of the impressed sound in the steady-state sound field for individual modes of vibration. The analysis can be extended further to study the steady-state response of the room.

The sound field in the room is represented by the wave equation, which is expressed in terms of the velocity potential as:

$$\nabla^2 \phi - \frac{1}{c^2} \frac{\delta^2 \phi}{\delta t^2} = o \qquad 1.34$$

Assuming a harmonic law for the pressure variation,

$$\nabla^2 po + k^2 po = o \qquad k = \frac{w}{c}$$

The solution to the wave equation that satisfies the boundary condition for particular discrete values of (k) that are the characteristic values k_n of the room have been discussed earlier.

Each characteristic value (k_n) is associated with a solution ($\psi_n(x, y, z)$ which is the characteristic function of the room, so that

$$p = A \psi_n$$

Since the specific wall impedance depends in general on the frequency $w=k_n$, it can therefore be identified with k_n.

Applications of Resonance Testing Techniques to Room Acoustic Measurements

Generally speaking, a resonance test may be carried out to determine the normal modes of the system, the associated natural frequencies and the damping. The procedure is, basically, to excite the system and measure its response.

In sound propagation, the acoustical variables are governed by the wave equation, which, for the pressure wave in the presence of a source distribution of density (q_r), is:

$$\nabla^2 p + k^2 p = -jw\rho q_r \qquad 1.40$$

The solution has been discussed earlier, and for a single mode is:

$$P_{(r_1 t)} = P_{(r)} e^{jwt} = Ae^{j(wt-\alpha)} = \frac{jw\rho Q_o}{V\Lambda_n} \cdot \frac{\psi_n(r)\ \psi_n(ro)}{k^2 - k_n^2} \cdot e^{jwt} \qquad 1.41$$

where

$$\iiint \psi_n \psi_m dv = \begin{cases} o & n <> m \\ V\Lambda_n & n = m \end{cases}$$

Λ_n being the average over the mean volume

$r=(x,y,z)$ the receiver's position

$r=(x_o,y_o,z_o)$ the source position

The transfer function of the system is obtained by dividing the response by some reference function (e^{jwt}) proportional and in phase with the input function, thus:

$$P_{(r)} = A_{(r)} e^{-jx} = {}^{jw} = \frac{jw\rho Q_o}{V\Lambda_n} \cdot \frac{\psi_n(r)\ \psi_n(ro)}{k^2 - k_n^2}$$

$P_{(r)}$ is the transfer function of the system, representing the complex pressure amplitude A(r): the amplification (on gain) representing the magnitude of the pressure amplitude α : the phase angle, the response lag behind a reference signal in phase with the input signal.

It will be necessary, experimentally, to keep the amplitude of the input signal constant so that the effect of variation in its magnitude will be eliminated. This is achieved by considering the receptance of the system under investigation.

By non-dimensionalising the response in terms of the frequency and the source strength, the following expression results for the response per unit amplitude of the strength of the source, and independent of the driving frequency:

$$P = \frac{P_{(r)}}{WQ_o} = Ane^{-jx} = \frac{A_{(r)} e^{-jx}}{WQ_o} = \frac{j\rho}{V\Lambda_n} \cdot \frac{\psi_n(r)\ \psi_n(ro)}{k^2 - k_n^2} \qquad 1.42$$

If δ_n is small compared to w_n, Equation (1.42) reduces to

$$P = A_n e^{-jx} = \frac{jc^2\rho}{V\Lambda_n} \cdot \frac{\psi_n(r)\ \psi_n(ro)}{(w^2 - w_n^2) - 2jw_n\delta_n}$$

If the position of the source and receiver are kept fixed, then:

$$P = A_n e^{-j\alpha} = \frac{jR_n}{(w^2 - w_n^2) - 2jw_n \delta_n}$$

where:

$$R_n = \pm \frac{\rho c^2 \psi_n(r)\ \psi_n(ro)}{V\Lambda_n} = \text{a constant} \qquad 1.43$$

or:

$$P = (x_n + jy_n) = \frac{-2R_n w_n \delta_n}{(w^2 - w_n^2)^2 + 4w_n^2 \delta_n^2} + j\,\frac{R_n (w^2 - w_n^2)}{(w^2 - w_n^2)^2 + 4w_n^2 \delta_n^2} \qquad 1.44$$

in rectangular co-ordinates.

x_n and y_n being in phase and inquadrative components of the receptance.

And:

$$A_n = \frac{R_n}{(w^2 - w_n^2)^2 + 4w_n^2 \delta_n^{2\ \frac{1}{2}}} \qquad 1.45$$

$$\alpha = \tan^{-1} - \frac{y_n}{x_n} = \tan^{-1} \frac{(w^2 - w_n^2)}{2w_n \delta n} \qquad 1.46$$

In polar co-ordinates

Equation (1.44) is the theoretical basis for the Kennedy and Pancu Technique, while equations (1.45), (1.46) are the basis for the peak amplitude and phase angle methods respectively.

The Peak Amplitude Method

In this technique the system is excited harmonically and the response amplitde at a particualr point measured over a range of frequency. The amplitude of the response depends not only on the dynamic characteristics of the system investigated, but also on the amplitude as well as the location and driving frequency of the exciting force. In order to eliminate the effect of variation of the magnitude of the exciting force, the recepatnce of the system is considered instead.

The source and receiver are both located at antinodes for the mode under investigation, so that the maximum response is attained.

Display of Results

The absolute value of the receptance is plotted against the driving frequency (Fig.2.70). It can be seen that the curve consists of a number of peaks, each in the vicinity of a natural frequency associated with a particular ormal mode of vibration.

Determination of Natural Frequencies

From equation (1.45), the absolute value of the receptance is given by:

$$A_n = \frac{R_n}{\left((w^2 - w_n^2)^2 + 4w_n^2 \delta_n^2\right)^{\frac{1}{2}}}$$

The minimum value of A occurs when:

$$w^2 \simeq w_n^2 \quad \text{i.e.} \quad w \simeq w_n$$

This defines the natural frequency of the mode as the value of the frequency at which the peak of the receptance is attained.

Analytically, some approximations are made in the above definition as follows:

a) the peaks do not occur exactly at the natural frequency, but at frequencies displaced one way or another from them,

b) the receptance is made up from contribution.

With the existence of heavy damping and closely spaced natural frequencies, experimental results become difficult to analyze, since some of the peaks may be missed altogether from the plot due to the fact that off-resonant contribution from other modes may be comparable in magnitude to the vibration in the resonant mode: this will tend to flatten the curve at these frequencies.

The method is therefore unreliable for modes which are heavily damped or with close natural frequencies, but it may still provide a rough guide to their properties.

Determination of Damping Constants

On the assumption that each peak represents response of only one mode, then the damping constant may be calculated from the sharpness of the peak and considering this assumption, the receptance in the mode is represented by:

$$A = \frac{R_n}{\left((w^2-w_n^2)^2 + 4w_n^2\delta_n^2\right)^{\frac{1}{2}}} \qquad 1.45$$

The maximum value of A occurs at $w \simeq w_n$

$$A \text{ peak} = \frac{R_n}{2w_n\delta_n} \qquad 1.47$$

Consider the particular value of (w), one above and one below (w_n) for which:

$$2w_n\delta_n = \pm \ (w^2 - w_n^2) \qquad 1.48$$

Substituting in Equation (1.45) shows that the amplitude of these values of (w) is $\frac{1}{V_z}$ of its peak value.

The difference between these frequencies and the natural frequency is small, and if it is assumed that there is no off-resonant contribution from neighbouring modes, then they are equi-distant from the natural frequency, Fig. (2.71).

Let

$$w_2 = w_1 = \Delta w$$

then

$$w_1 = w_n - \frac{\Delta w}{2}$$

and

$$w_2 = w_n + \frac{\Delta w}{2}$$

Substituting in Equation (2.78)

$$2w_n\delta_n = - \ w_n - \frac{\Delta w}{2} - w_n \ = w_n\Delta w + \frac{\Delta w^2}{4}$$

and

$$2w_n\delta_n = + \left(w_n + \frac{\Delta w}{2}\right)^2 - w_n^2 = w_n\Delta w + \frac{\Delta w^2}{4}$$

Add together

$$4w_n\delta_n = 2w_n\Delta w$$

Therefore

$$\delta_n = \frac{\Delta w}{2} = \frac{w_2 - w_1}{2}$$

1.49

The damping constant δ_n is then found by measuring the frequency band between points on an amplitude resonance curve corresponding to the amplitude being equal to $\frac{1}{\sqrt{2}}$ of its peak value at these frequencies, i.e. to half-power width on an energy resonance curve.

Since, by application of peak amplitude method, the part of the peak which is associated to the resonant mode cannot be identified precisely, the contribution from the off-resonant vibrations which are also included within the peak produces some error which must certainly be considered when this method is applied.

The contribution to the amplitude from the off-resonant vibration is not constant. Because of the marked change in phase of the response in the resonant mode when passing through resonance, the effect of the off-resonant vibration on the amplitude above the resonant frequency will differ from the effect below, and the resonance curve will not be symmetrical.

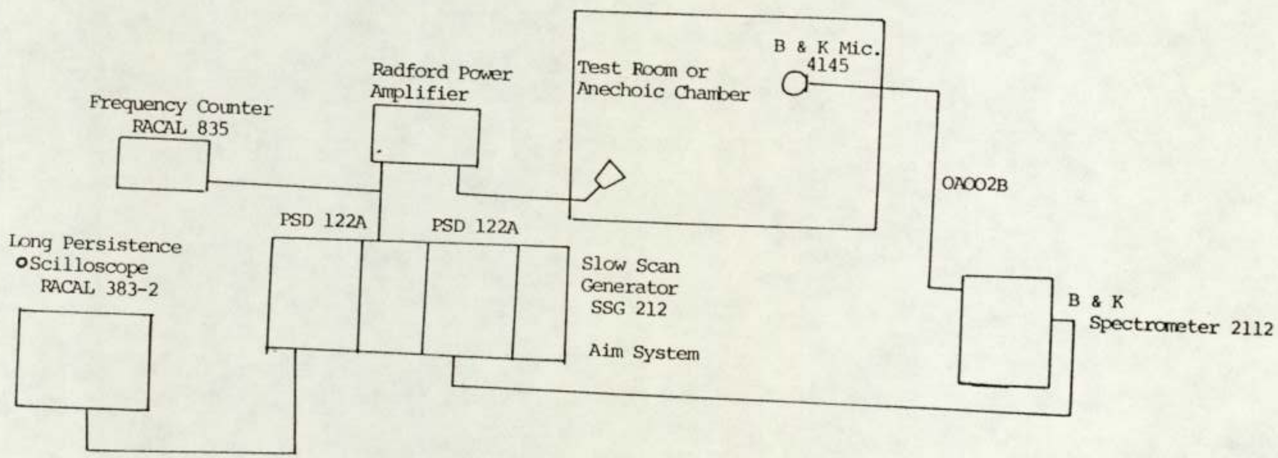
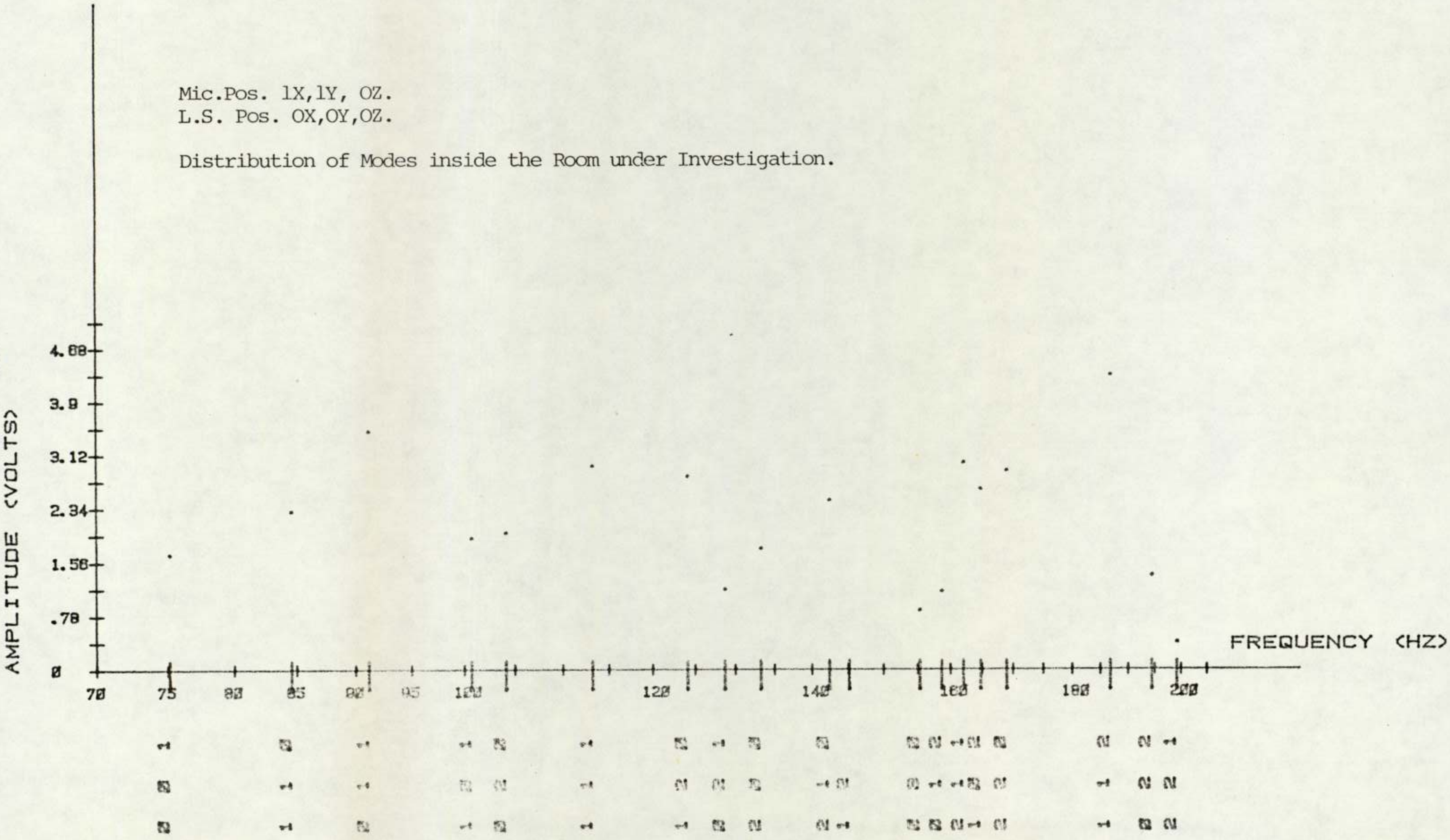


FIG. 1.45 Instrumentation Set Up for Complex Frequency Response Measurements.



Mic.Pos. lX,lY, OZ.
L.S. Pos. OX,OY,OZ.
Distribution of Modes inside the Room under Investigation.
AMPLITUDE (VOLTS)
FREQUENCY (HZ)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200

CHAPTER 2

2.1 Resonant Technique Applied to Room Acoustics

The resonance frequencies and their amplitude and the damping constants are the quantities that are determined experimentally, rather than the characteristic values of an isolated normal mode of vibration. Techniques are developed whereby the normal modes of a room are identified and their associated resonance frequencies and damping constants are measured.

In electrical terms the width of the resonance curve is used as a measure of the effective dissipation of a resonant circuit. Its application in the acoustical case to determine the acoustical damping of a resonant mode of vibration was first suggested by Hunt (13) (Investigation of Room Acoustics by steady-state transmission Measurements) J.Acous.Soc.Am.10, 216.

In aeronautical engineering, in particular, most of the techniques of resonance testing were developed and used to provide essential information for flutter calculations in aircraft structural variations (1.4).

Various techniques for performing such measurements have been developed which are distinguished by:

1) the physical quantities which are measured
2) the ways in which the experimental data are displayed

3) the ways in which the data are analysed.

The relationship

between the damping factor of a simple oscillator and the fractional loss of energy of free vibration, that expresses the effect of damping in terms of an exponential decay of the total energy, is assumed (11) to hold for any vibration, and can be used to study the damping of standing waves in rooms. Since the air in a room is considered as an assemblage of resonator-standing waves that can be set into vibration by a sound source.

In theory, the techniques of resonance testing are applied in the study of room response to acoustic excitation.

In most basic forms of resonance testing, a system is excited harmonically and the driving frequency adjusted until a resonant state is found.

The resonance frequency is assumed to be a natural frequency of the system, the resonant vibration being assumed to take place accurately in the corresponding principal mode that can be identified and measured, so that the variation of response with driving frequency around the resonant frequency is a measure of damping associated with that mode.

The system may possess close natural frequencies and as a result the violent vibrations may not be separately excited in their corresponding principal modes, and the pressure of heavy damping can lead to the obscure detection of some of the modes.

2.1.2 Resonance Testing Techniques

The commonest method of resonance testing is to excite the system concerned and to measure the total forced vibration at enough points to be sure that all the modes display their resonant characteristics in at least one of their responses.

A response curve of the total amplitude against the driving frequency is then plotted and the required information is extracted from the plotted curve. This is known as the peak-amplitude method. From the theoretical point of view it is inadequate and seems to be misleading. Its main limitation lies in the fact that no account is taken of the changes in the phase lag of the response of the system behind the exciting force. Since this change in phase is most marked when the system is passing through resonance, information can be derived from the analysis of a simple phase-angle plot. Subsequent analysis shows that phase-angle plots yield as much information as those of peak-amplitude.

This was recognised by Kennedy and Pancu (42) in 1947; they proposed an alternative and theoretically more attractive method which was claimed to be superior (15) to the conventional peak-amplitude method of detection of natural frequencies and damping constants.

In the Kennedy and Pancu method, the amplitude and phase are measured, but instead of plotting two separate graphs, the amplitude and phase are plotted together on an Argand plane. This was claimed to be able to isolate modal frequencies and damping parameters of a system with relatively close natural frequencies by easily eliminating the contribution to overall vibration level of off-resonant motion. The method, as pointed out by Bishop and Gladwell (41) was based entirely on theoretical grounds, backed by numerical examples: nevertheless, it has found wide application in practice (15).

EXPERIMENT

2.1.3 Instruments

Microphone

The microphone used in the experiment was a one-inch condenser microphone type 4145 B&K, which was connected to the pre-amp input socket of the Hetrodyne Analyser 2010 through a 16m extension lead.

Loudspeaker

The loudspeaker unit was the KEF B110, which was mounted inside a wooden enclosure constructed according to the manufacturer's required specifications and dimensions and was connected to the beat frequency oscillator socket of the Hetrodyne Analyser 2010.

Hetrodyne Analyser 2010

Hetrodyne Analyser 2010 was used to generate pure tone sound and to sweep the required frequency range, and also to measure the frequency response of the sound at a selected bandwidth and consequently store it on a Digital Event Recorder.

Digital Event Recorder Type 7502

This was used to record the frequency response of the sound, which was then displayed on the oscilloscope. The Digital Event Recorder was connected to the x,y plotter (Hewlett Packard) and the frequency response was plotted. The Digital Event recorder was also connected to the computer and the recorded data as digital was transferred to the computer.

Function Generator FG1

This was used to generate the required sinusoidal wave, the exact frequency or the average period of it was measured by the frequency counter.

Level Recorder Type 2305

The level recorder was used to adjust the speed with which the frequency is swept. This was achieved by altering the drive shaft speed and the paper speed.

Stabilizer Power Supply

This was used to trigger the start of the recording of the frequency response at the required frequency and consequently terminated the recording at the required frequency.

2.1.4 Experiment

To study the behaviour of low frequency absorption in a small enclosure, a small rectangular room within Room 209 in the Main Building of the

University was prepared for the purpose of the experiment. The room was emptied, with no materials on the walls, floor or ceiling. Figure 1.0 shows the small room where the experiment was carried on.

The KEF B110 loudspeaker was placed in one corner of the room. As the equation

indicates, the pressure amplitudes of all patterns of standing waves in a rectangular enclosure are at the maximum in the corner of the room.

The KEF B110 loudspeaker was placed in one corner of the room, so that every mode of vibration was fully excited, and also the value Ψ_n (S) (which is the value of Ψ_n at position S of the source) was made unity for all values of n (n_x, n_y, n_z) and variation in energy dependent only on the impedance functions for the normal modes and on the position of the observer. The loudspeaker was connected to the BFO output socket of the Hetrodyne Analyzer 2010.

The microphone was placed on six positions along the x,y and z axis of the room (three different corner positions and three respective mid-point positions). The positions are shown in Figure 2.0.

The microphone was connected to the preamp input socket of the Hetrrodyne Analyzer 2010 by a 16 meter extension lead, so that the sound received through the microphone was amplified.

Mechanical mean (i.e. level recorder) was used to synchronise the speed with which the frequency was swept through by the Hetrodyne. The scanning system of the oscillator was mechanically connected to the drive of the level recorder by a flexible shaft.

The paper speed of the level recorder was chosen so that a relatively slow frequency sweep was obtaained. This was done so that the peaks could be identified more accurately, and a certain time was allowed to elapse between each frequency interval.

By connecting the stabilizer power supply to the EXT trigger of the Digital event Recorder, the recording was triggered at a set frequency (in this experiment 70.0Hz) and also recording ceased when a certain frequency was reached. This allowed suitable and accurate recording of the frequency response into the Digitaal Event Recorder to be carried out for each position of the microphone inside the room in the frequency range of 70.0 to 200.0Hz.

The 'Input Sample Rate' selector of the Digital Event Recorder indicates the 'sweep time per 2k' (1k=2 =1024). This is the recording time available if the instrument had a 2k word memory.

The time taken for the range of frequency to be covered by the Hetrodyne Analyzer 2010 was 76 secs, therefore the frequency of the Function Generator was adjusted as near as possible to $\frac{4096}{76}$ = 53.895Hz, and this was measured more accurately when the frequency counter was employed. The average period was also measured and was checked against the frequency.

The actual frequency range was also plotted by the x,y plotter as a logarithmic scale, and consequently the variation of the amplitude with frequency for each position inside the room was plotted by the x,y plotter and the peaks were identified.

The frequency response of the room was also displayed on the oscilloscope and by adjusting the input and output attenuation of the Hetrodyne, peaks with reasonable amplitude were obtained.

The Digital Event Recorder was connected to the PET computer and by using program 'AMPDATA' the amplitudes of 4096 points on the whole frequency response curve were transferred to the computer as a number that could be changed to voltage and also to the sound pressure level.

The computer program 'ANALYSIS-PRINT' was used to obtain the resonant frequency, the amplitude of the resonant frequency (coupling) and the frequency bandwidth (damping).

The theoretical frequencies of the normal modes of vibration of the room on the x,y,z plane were obtained from computer program 'ROOMODES'.

The experimental and theoretical results of the resonant frequencies of the normal modes of vibration were compared and the parities were identified.

Consideration must be given to the way the frequencies of the nromal modes were calculated theoretically.

In theory it was assumed that all the surfaces of the room or enclosure were rigid walls and therefore the minimum of sound energy was passed through them and there was no interference from outside. The nodal points of the sound waves were found by considering that the pressure at the points of reflection of the sound wave is maximal.

The amplitudes of the frequency response curve for each position of microphone that are transferred to the PET from the Digital Event Recorder are then stored on the floppy disk and for each position the Data file is given as follows, SRPSTN1 - where SR represents the small room and PSTN1 represents the position of the microphone inside the room.

2.1.5 Frequency Response of the Loudspeaker

The loudspeaker used was the KEF B110 type and its frequency response was measured inside the anechoic chamber of the Department of Physics. To investigate the variation of its sound pressure level over the frequency range required for the experiment (i.e. 70 to 200Hz), the response obtained was compared with the graph provided by the manufacturer.

The loudspeaker was placed nearly in the middle of the anechoic chamber and the B&K condenser microphone type 4145 was placed at 1.0m from the loudspeaker. The dimensions of the box covering the loudspeaker are 330mmx240mmx150mm.

The set up for the frequency response of the loudspeaker is shown in Figure 1.1. The frequency response of the loudspeaker was obtained before and after the experiment so that any possible variation of the response could be investigated, and the two curves were compared. As can be seen from the curves, the frequency response of the loudspeaker is reasonably constant over the experimental frequency range, and it also matches the frequency response curve provided by the manufacturer. The graph of frequency response is shown in Figures 2.2, 2.3, 2.4, 2.5.

The program 'Analysis-Print' calculates the peak of the modes, the resonant frequency and the bandwidth of the modes.

The peak of the modes is found by comparing the amplitude of the 4096 points on the curve. The peak is the point where the amplitudes of the point before and after are both smaller than the point. When the peak is found then the frequency related to the peak is calculated.

To find the bandwidth, the r.m.s. value of the peak is calculated and a horizontal line is then passed through this point till it intersects the two sides of the curve: the length of this line represents the bandwidth of the mode. Difficulty arises when one side of the curve before reaching the r.m.s. level starts to move in an upwards direction.

Two different approaches are adopted, as follows:

i) The intersection of the horizontal line with one side of the curve is taken and the length of the line is multiplied by two.

ii) Since the total curve is divided into 4096 equal points, the number of the point before the curve starts its upward motion is noted and two numbers, one related to the peak, are taken and then from symmetry the approximate number of the point intersecting the horizontal line with the curve is calculated, then the related frequency is found and consequently the bandwidth is found.

For the waves that are distributed along the x,y and z plane, the wave equation is:

$$\frac{\delta^2 p}{\delta x^2} + \frac{\delta^2 p}{\delta y^2} + \frac{\delta^2 p}{\delta z^2} = \frac{1}{c_o^2} = \frac{\delta^2 p}{\delta t^2}$$

Frequencies of the nromal modes of vibration on the x,y and z planes

$$f_n = \frac{c_o}{2} \left[\left(\frac{n_a}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2 \right]^{1/2}$$

where

$n_x = 0,1,2,3.....$

$n_y = 0,1,2,3.....$

$n_z = 0,1,2,3.....$

As can be seen from Figure 2.7, the sound pressure is either maximum or zero depending on the value of n , and as can be seen for the microphone position at the end of the principal axes, the peak of the sound pressure is related to the odd mode and the minimum of the sound pressure is related to the even mode, while the situation reverses when the microphone position is on the middle of any of the principal axes of the room. Therefore, by comparing the modes occurring when the microphone is positioned at a corner of the room and in the middle of the principal axes of the room, the effect of the position of the microphone relative to the loudspeaker is studied. The co-ordinate of the plane of the microphone relative to the loudspeaker is considered. The co-ordinate on the plane x,y or z of the loudspeaker position is considered as the reference plane, and the co-ordinate position of the microphone is taken relative to the plane of the loudspeaker. The intrumentation set up is shown in Figure 2.8.

The frequency response curve of the room with bare surfaces was obtained for each of the two positions of the microphone on the middle and the corner of the principal axes of the room with the loudspeaker placed at the origin of the co-ordinate system. The frequency response curve was then measured by transforming from analogue to digital form and transferred to the CBM computer. A computer program 'PLOT8' was written and by using this program the frequency response curve was plotted by the 7225A Hewlett Packard plotter.

The frequency response curve of the untreated surfaces of the room for the six positions of the microphone along the three principal axes are shown shown in Figures 2.9, 2.10, 2.11, 2.12, 2.13, 2.14.

The values of the resonant frequencies obtained for each mode from the frequency response curve of the room with untreated surfaces and the values obtained from the theoretical analysis by applying the wave theory (i.e. the computer program 'ROOMODES') were different when compared. This variation could be due to

i) In deriving the theoretical formulae for the resonant modes, all the boundary surfaces were considered to be rigid, but in the room under consideration not all the surfaces were rigid and the admittance of them was not uniform.

ii) The diffusitivity of the sound in the room can also contribute towards the resonant modes.

iii) The asymmetry of the geometrical shape of the room is also one of the factors that influences the resonant modes and, as was shown, the room under consideration was not quite a perfect rectangle.

iv) The variation of admittance along surfaces of the room also alters the behaviour of sound in a room.

v) The method and also the materials used in the construction of the room can influence the results of the resonant frequency.

It must be considered that in deriving the expression for the characteristic frequencies, rigid surfaces and were assumed. However, the truth is that the normal modes of vibration of the air space of an enclosure are characterised by a complex interweaving of standing waves.

It is shown that the sound field in a room is generally far from statistically uniform and that decay curves are usually characterised by fluctuations and that the placing of absorption samples in reverberation rooms must be governed by certain rules if the results from different laboratories are to agree. A perfectly rectangular unfluctuating decay curve, far from indicating isotropic conditions in a room, shows the existence of a single overriding room mode in which all particle velocities are in the same direction.

Any improvement in the state of diffusion of a room therefore implies the breaking up of standing wave systems, the diversions of energy from strongly excited modes into weaker ones and the reduction of points of high intensity caused by focusing sound into particular planes of direction.

It is generally accepted that two studios that are similar in their reverberation characteristics may be different in their sound, one having less acceptable sound than the other, and it is found that it can be improved and even changed to be less coloured, by adding to the walls and ceiling scattered shapes, such as coffering, hemispheres or hemicylindrical pilasters. These additions need not be absorbers, and may not change the reverberation time by an amount corresponding to the change in the acoustics. It is reasonable to assume that the initial differences between the studios are due to different states of diffusion, since differences in absorption or reverberation time have been eliminated. The same changes can be produced by changes in the distribution of absorbers, or by the modification of surfaces that cause focusing.

The frequency of fluctuations in the steady-state transmission characteristics between two points in the room, the depth of fluctuation in decay curves, the variation of intensity or reverberation time from point to point in the field have been investigated so that the statistical properties of a room of a kind that is expected to be affected by the state of diffusion can be quantified. However, most of the investigation proved to be mainly dependent on the reverberation time and even independent of the diffusion. For instance, the number of fluctuations per cycle bandwidth in the steady-state transmission characterisitc is related only to the reverbertion time and the distance between the points between which the characteristic is measured. Generally speaking, these methods have not been particularly successful, largely because there was not a prior reason for assuming any definite relationship between the characteristic under investigation and the state of diffusion.

Meyer and Kuttruff (33) (1958), using scale models, showed that the effective absorption of material fixed to the walls of an enclosure can be used as a measure of state of diffusion.

Randall and Ward (329 (1960) carried out systematic experiments on the mutual effects of patches of absorbent material and of rectangular or short hemicylindrical projections on the walls and floor of a reverberation room. They found that patches of absorbent material were as effective in promoting good absorption as perturbation of the walls. Consequently, two patches of absorber on two mutually perpendicular walls of a room were very much more effective than the sum of the two measured singly, unless the field had already been made diffuse by wall perturbations. They accordingly proposed and evaluated a method of measurement of the state of diffusion by comparing the slopes of the top and bottom halves of decay curves.

A continuing problem in the design of small studios is the variation in the absorbing cross-section of a sample of material placed in different rooms or in different positions in the same room. These variations are greatest with small rooms and cause errors in design. Moreover, small rooms are characterised by colourations due to isolated strong modes and it is necessary to understand how these are influenced by measures to improve diffusion and how to assess independently the effect of such measures.

It was noted (31) that the division of an area of absorber into several patches on mutually perpendicular surfaces yields highly effective absorption coefficients. It was also noticed that the subdivided sample had, at certain frequency, a peak of absorption considerably higher than that reached by the single large sample even in conditions of good diffusion. This excess absorption is attributable to the effect of diffraction at the edges of the sample, which was mineral wool. Since the sample was softer than the surrounding wall surfaces, the lines of flow in the immediate neighbourhood tended to converge to the sample.

Kuhl (1960) measured the random incidence absorption coefficient of a sample of rockwool in several successive states of subdivision. As the individual areas became smaller, the peak absorption increased, eventually reaching a maximum value of 70% higher than that of an infinite area sample. Kuhl showed that this effect increases as the impedance of the material decreases and that the excess absorption due to the subdivision is a steadily rising function of the ratio of the perimeter of the sample to its area. The subdivision of an absorber therefore has two important effects: it introduces diffusion, and is effective in this respect at all frequencies at which absorption takes place, and it increases the absorption of a given area of material, particularly in the lower middle frequencies, where common types of shallow porous absorbers are inefficient.

Since at the low frequency range the room modes are well separated, it is desirable to make the mean modal bandwidth at least equal to the mean modal spacing, in order that the few available modes cover the frequency range most uniformly. In the absence of the room absorption, the modal bandwidth is small and each mode is, therefore, poorly excited by sound at frequencies only a small distance away on the frequency axis. As absorption is added to the room, the bandwidth of individual modes spreads, so that the skirt of each modal response curve tends to overlap those of adjacent modes. Excitation frequencies lying between modes can then involve both upper and lower neighbouring modes in the excitation of the room and, thus, added absorptive treatment helps the few available modes to cover the frequency range uniformly, and because of this absorptive materials were placed on the surfaces of the room and the effect of them on the spread and spacing of the modes was studied.

At the upper region of the low frequency range, the number of modes that exist within a certain frequency range is much more than the number of modes within the same frequency band covering the lower region of the low frequency range. When absorptive materials are added to the room, it is inevitable that where the concentrtion of room modes is high, because of the additional absorption, the frequency band of the mode increases and since prior to the addition of absorptives, two or more modes are closely spaced, the extra added absorptive causes the modes to overlap and sometimes this develops to the appearance of one mode with a very wide frequency band or even in some cases the shift of the mode occurs and this causes the problem of identifying the exact mode and the mode number associated with it.

It is also shown that the location of the absorptive materials and also the amount of the absorptive materials which cover the surfaces of the room are the dominant factor in determining the spread of the modes within a frequency range and because of this the amount of covered surfaces were varied each time and the effect of this on the spread and its influence on the resonant frequency, damping and the coupling of the modes was observed.

The frequency response curve when three surfaces of the room (i.e. surfaces ABCD, ADEH, DCGH) were covered with 25mm thick fibre-glass was obtained for each position of the microphone and the values of resonant frequency, damping and coupling of each mode obtained from the curve were then compared with the values obtained for untreated surfaces of the room.

For the purpose of this experiment, the two adjacent walls and ceiling which are mentioned above were covered with fibre-glass. The frequency response curve was plotted for each position of the microphone.

This experiment was carried out to investigate if possible the relationship, if any, between the amount of damping and the resonant frequency and coupling of each mode.

The frequency response curves for each position of the microphone are shown in Figures 2.15,2.16,2.17,2.18,2.19,2.20.

The values of resonant frequency, damping and coupling of each mode obtained from the frequency response curve were compared for treated and untreated surfaces of the room and as a consequence the following points were observed:

1) The values obtained for the resonant frequencies varied.

   This could be due to:

   a) the change in dimension of the room that could have occurred by fixing the absorbing materials on the surfaces,

   b) fixing of the absorbing materials tht causes the admittance and the rigidity of the surfaces to vary,

   c) the shift of the axis of the resonant mode; this was caused by the shift of the point of maxima and minima pressure,

   d) the unpredictability of the behaviour of the sound when absorrbing materials were added, particularly at low frequency,

   e) the uneveness of the absorbing material,

   f) the method of fixing absorbing materials to the surfaces,

   g) since the absorbing material was cut to sections prior to fixing, when the sections were fixed to the surfaces, narrow gaps were created between each section, and this could change the normal pattern of the sound waves.

Comparing the results for the resonant frequencies of treated and untreated surfaces showed that there were modes that only appeared in one of the cases and not in others. Sometimes there were modes that seemed to be apparently new modes, but this was not true at all. As can be seen from the frequency response curve when three surfaces were covered with absorbing materials, close resonant modes that appeared on the upper region of the low frequency range merged and appeared as a new single mode, and the coupling was expected to be more evenly distributed.

The introduction of absorbing materials to the surfaces of the room was expected to increase the amount of damping in the room and consequently widen the frequency bandwidth (damping) of the modes.

As can be seen from the results obtained, the frequency bandwidth (damping) of some of the modes widened since the presence of the absorbing materials which covered the surfaces increased the amount of absorbed sound energy.

The two close modes could combine and appear as a single mode, which is often difficult to separate into two separate modes. Therefore, it is difficult to identify the frequency bandwidth (damping) related with each mode, and in some cases where it is difficult to identify them, this has lead to misjudgment and errors in calculating the exact value of damping related to each mode.

The presence of the absorbing materials tends to couple the normal modes of an unperturbed room, so that each eigen function is a mixture of many of the simple waves. The variation in admittance of the walls couples the modes together, which results in a much more randomly distributed standing wave, with the regularity in its nodal surfaces.

The values of the amplitude (coupling) of each mode obtained from the frequency response curve when compared for treated and untreated surfaces of the room varied. For treated rooms, the values of amplitude (coupling) of some of the modes decreased, but for some modes this value actually increased, and this could be due to the fact that:

1) In a certain condition, when the loudspeaker and microphone are placed in the points of maxima pressure of one mode, and there is only one single mode in the room, the values of amplitude (coupling) should decrease for treated surfaces of the room when compared with the similar values obtained for untreated surrfaces of the room.

   The introduction of absorbing materials does not necessarily lead to an increase in the amplitude values, since the degree of coupling of each mode is dependent on the location of the loudspeaker and microphone along the path of pressure curve. The presence of absorbing materials could well shift the position of maxima and minima of the modal pressure curve and therefore this results in the position of the microphone being shifted towards the maxima and minima. The modal pressure curve is then shifted, thus altering the relative recording position. The introduction of absorbing materials should decrease the average spatial coupling.

2) Apparent modes that were not exactly identifiable were formed as two or more close modes that existed in the case of untreated surfaces, were combined and appeared as a single mode when absorbing materials were added to the room. Therefore the value for the amplitude (coupling) of this apparent single mode is the coupling between the number of modes forming this apparent mode, and this could increase or decrease the value of amplitude (coupling) substantially, since the apparent mode could easily be formed by a combination of odd and even modes.

The location of the loudspeaker was changed and the loudspeaker was moved into another corner of the room, to investigate the variation, if any, and the amount of influence that the change of location of the loudspeaker had on the values of resonant frequency, damping and coupling of each mode.

The position of the loudspeaker was moved to a new origin, i.e. at the extreme end of the y-axis co-ordinate along the floor, and the frequency response curves were obtained for each position of the microphone by placing the microphone at positions so that its co-ordinate relative to the loudspeaker remained the same as before.

The values obtained for the resonant frequency, damping and coupling of each mode were then compared with:

i) The similar values obtained when the loudspeaker was placed at the original co-ordinate.

ii) The value of resonant frequency, damping and coupling obtained for each mode when the surfaces of the room were untreated.

The frequency response curve for each position of the microphone is shown in Figures 2.15(A), 2.16(A), 2.17(A), 2.18(A), 2.19(A), 2.20(A), 2.21, 2.22, 2.23.

The positions M1,M3,M2 are the mid-points on the floor, ceiling and middle of the room respectively.

Comparing the value of resonant frequency damping and coupling obtained for each mode with the related values, when the loudspeaker was placed at the original co-ordinate of the room the following could be deduced:

i) The values of resonant frequency, damping and coupling of most of the modes were in good agreement, and the contrary was true for some of the other modes.

ii) Some of the modes seemed to have been split and appeared as two modes very close to each other, and were in good agreement with the modes obtained when the loudspeaker was placed at its original position.

iii) The variation in the modes when compared, showed that their values did not follow a pattern, and so the variation in the values could not be generalized.

The reason for the discrepancies when the values of resonant frequency, damping and coupling of each mode were compared for each position of the loudspeaker could be explined as:

i) The slight variation in the diffusity of the room due to the variation in value of admittance of the surfaces where the loudspeaker is facing them,

ii) The asymmetrical nature of the shape of the room and the non-uniformity of the surfaces of the room, which can influence the reflection and the angle of reflection considerably,

iii) The behaviour of the sound wave near the source when the loudspeaker is moved to the new position,

iv) Since three surfaces of the room were covered with fibre-glass and the loudspeaker in this case is placed at the corner on the floor of the room so that it is facing the intersection of the two surfaces with one of them covered with fibre-glass, this could affect the behaviour of sound. This could be compared with the previous case, when the source was facing the intersection of two untreated surfaces.

The fibre-glass that is used to cover the surfaces of the room was cut as far as possible into equal sections, but becaause of the geometrical shape of the room and the size of the fibre glass that was available, some of the fibre glass had to be cut into unequal sections in order to cover the whole of the surfaces.

To be able to investigate whether the order with which the fibre glass was placed on the three surfaces affects the values of resonant frequency, damping and coupling of each mode, the frequency response curve for each position of the microphone was obtained when the same three surfaces were covered, but the order in which the sections of fibre glass were placed changed. The loudspeaker was placed at the previous position.

The values obtained for the resonant frequency, damping and coupling were compared with the similar values obtained when the order of placing absorbing materials on the surfaces was different.

The two sets of results were then compared and this showed, as expected, that they varied. This discrepancy in the results could be due to the:

i) When the order of placing the fibre glass changed, the non-uniformity of the fibre glass surfaces therefore also changed, and this could influence the behaviour of sound.

ii) The size of the gap that was created between the boundary of each section varied as the order in which the fibre glass was placed changed.

iii) The amount of diffusitivity of the sound in the room changed.

The decision to cover three surfaces of the room with fibre glass was taken on the basis that by covering three surfaces, the room could be sufficiently damped and the behaviour of sound in a damped system could be studied.

The presence of absorbing materials on three surfaces of the room was thought to make the admittances of the three surfaces more uniform and this could be used as a guideline when studying the damping of the room and the behaviour of sound with more surfaces covered with fiibre glass. This idea could be extended and five surfaces of the room could be covered with fibre glass, which could enable study of the behaviour of sound at the untreated surface.

Since there are always complications with the low frequency absorption and the results obtained at the low frequency absorption varies with the characteristics of the transmission suite and cannot be used as a guideline, it was thought that all the surfaces of the room should be covered with fibre glass, so that the maximum possible damping could be achieved. The results of the resonant frequency, damping and coupling of each mode can then be compared, it also can be investigated whether the results follow a pattern and, if possible, the pattern that the results follow be generalized.

All the surfaces of the room were then covered with the 25mm thick fibre glass, to make the room more damped. The frequency response curve was then obtained for each position of the microphone as before.

The values of resonant frequency, damping and coupling obtained for each mode were obtained from the freuency response curve for each position of the microphone and the loudspeaker at the same position and were compared with:

i) the values obtained for untreated surfaces of the room,

ii) the values obtained when three surfaces of the room were covered with fibre glass.

The frequency response curves for each position of the microphone are shown in Figures 2.25,2.26,2.27,2.28,2.29,2.30.

By covering all the surfaces of the room, more damping was introduced to the room, and therefore the amount of sound energy that was absorbed should have increased. The sharpness of each resonant curve was expected to decrease and the resonant frequency curve to become more flat.

The results obtained for the resonant freequency compared with the values obtained when three surfaces of the room were covered showed that there was not much variation and it could even be said that they were in good agreement. The slight variation in the values of the resonant frequencies could be due to:

i) The change in the dimensions of the room that could have occurred by fixing absorbing materials to the surfaces,

ii) The extra absorbing materials that could have caused the admittance and the rigidity of the surfaces to vary,

iii) The shift of the axis of the resonant mode; this was caused by the shift of the point of maxima and minima pressure,

iv) The non-uniformity of the property of the absorbing material,

There were modes that seemed to be new, but as it is explained, this could not be true, since by introducing more absorption close resonant modes could merge and appear as a single mode or in some cases a mode splits into two very close distinguishable modes. The damping of most of the modes increased as extra absorbing materials were fixed to the wall.

The amplitudes (coupling) of each mode obtained from the frequency response curve were compared with the values obtained for the case of absorbing materials covering three surfaces of the room; the amplitudes of some of the modes showed a considerable increase contrary to expectation, since increasing the amount of absorbing material should decrease the amplitude as far as a single mode is concerned. This discrepancy in the results is explained earlier.

The frequency response curves were then obtained for each position of the microphone when the loudspeaker was moved to a new co-ordinate (i.e. 0X,1Y,0Z). All the surfaces of the room were still covered with the 25mm thick fibre glass and the positions of the microphone relative to the loudspeaker were kept the same as before. This experiment was carried out to compare the results as it was done when three surfaces were covered and to investigate whether the results follow a pattern.

The values of resonant frequency, damping and coupling were then compared with the values obtained respectively for :

i) the untreated surfaces of the room,

ii) when three surfaces of the room were covered with fibre glass,

iii) when the loudspeaker was moved to a new origin, and

a) the surfaces of the room were untreated,

b) three surfaces of the room were treated.

The results of the resonant frequency, damping and coupling when the respective values were obtained were compared with the similar value and the loudspeaker was placed at its original position.

The reason for this discrepancy in the values could be due to the reasons given when three surfaces were covered with fibre glass and the position of the loudspeaker was changed.

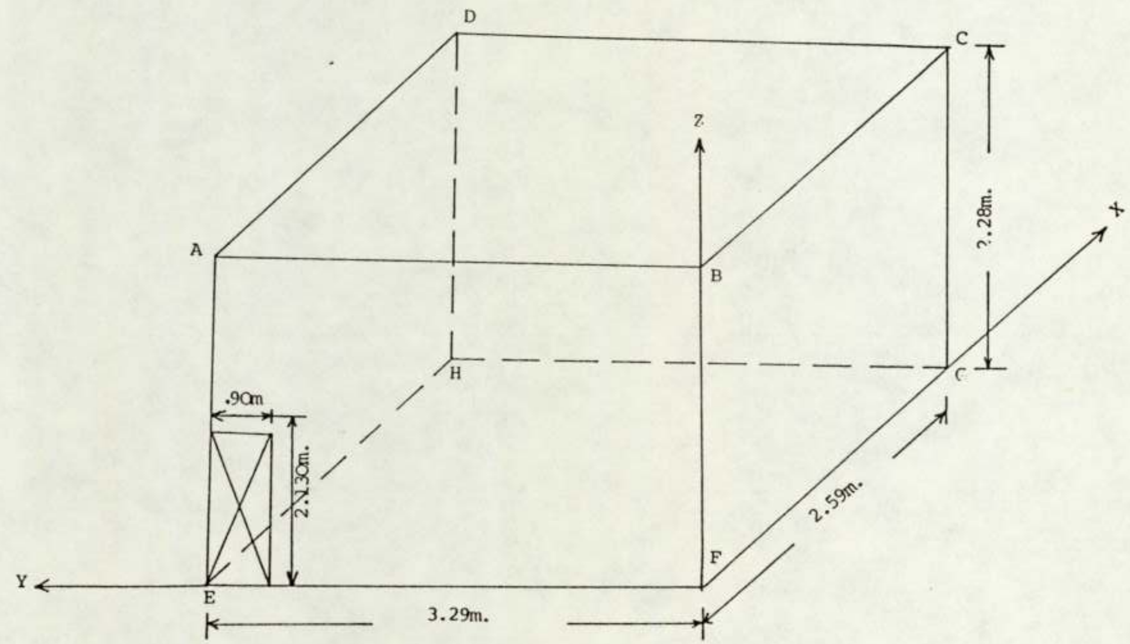
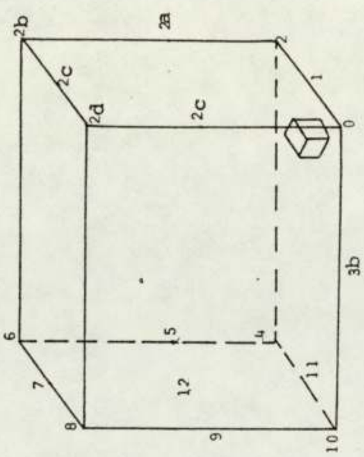


FIG. 2.0 The Small Room.


2b
2a
2c
2d
2c
2
1
0
3b
6
5
4
7
12
11
8
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10

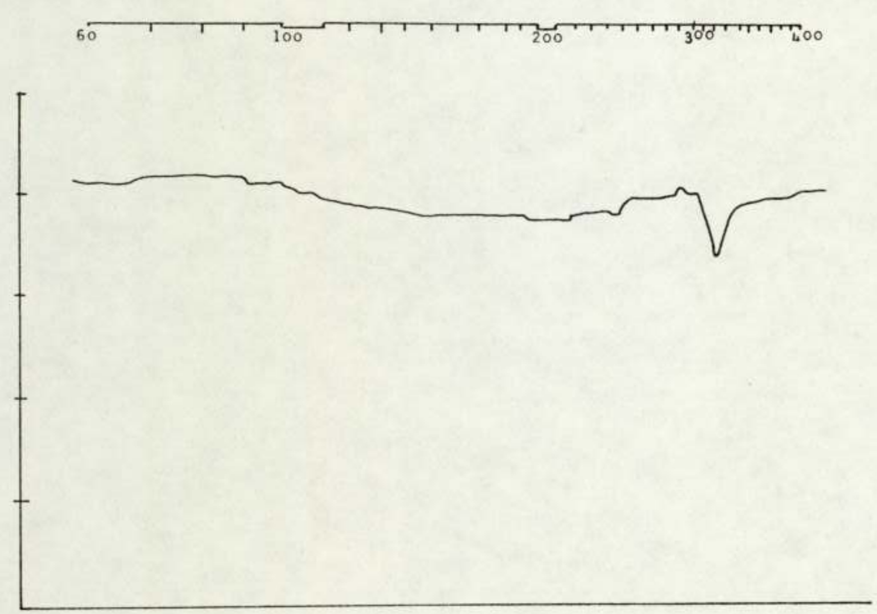


FIG. 2.5 Frequency Response Curve for the KEF B110 Loudspeaker.

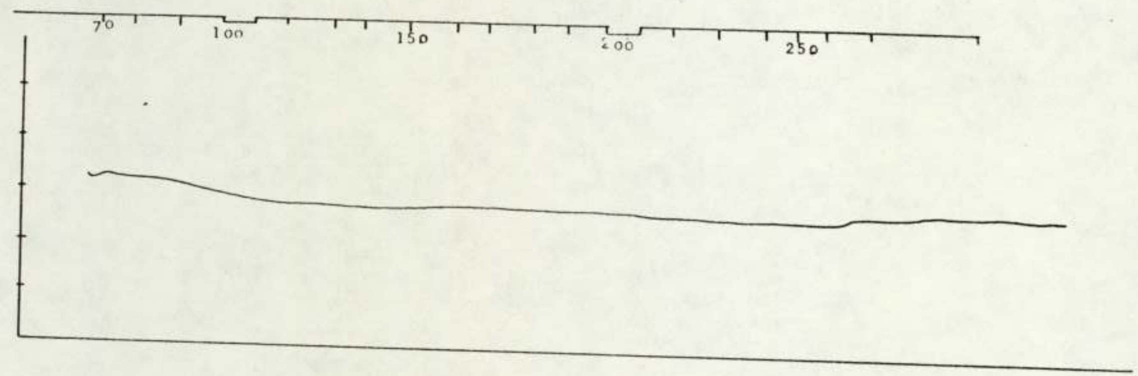
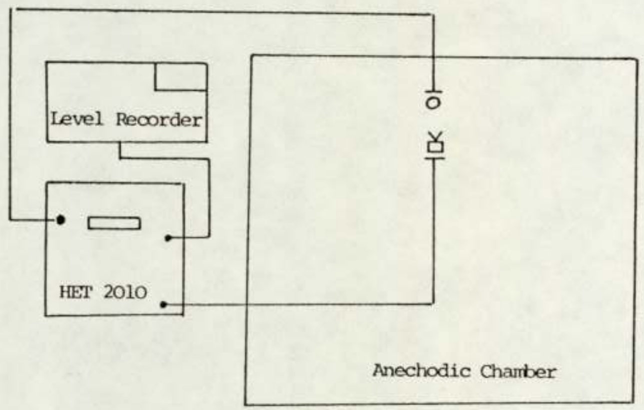
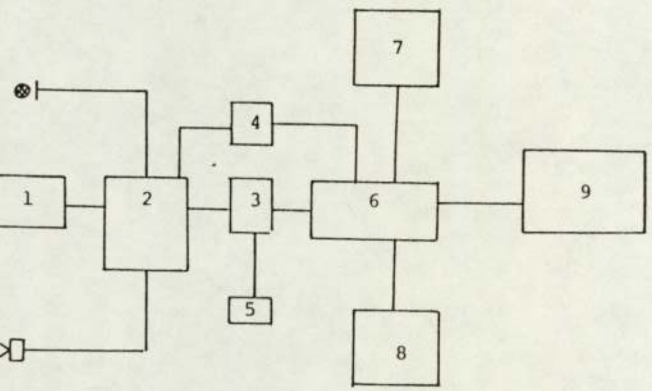


FIG.2.3 Frequency Response to the KEF B110 Loudspeaker after Experiment.

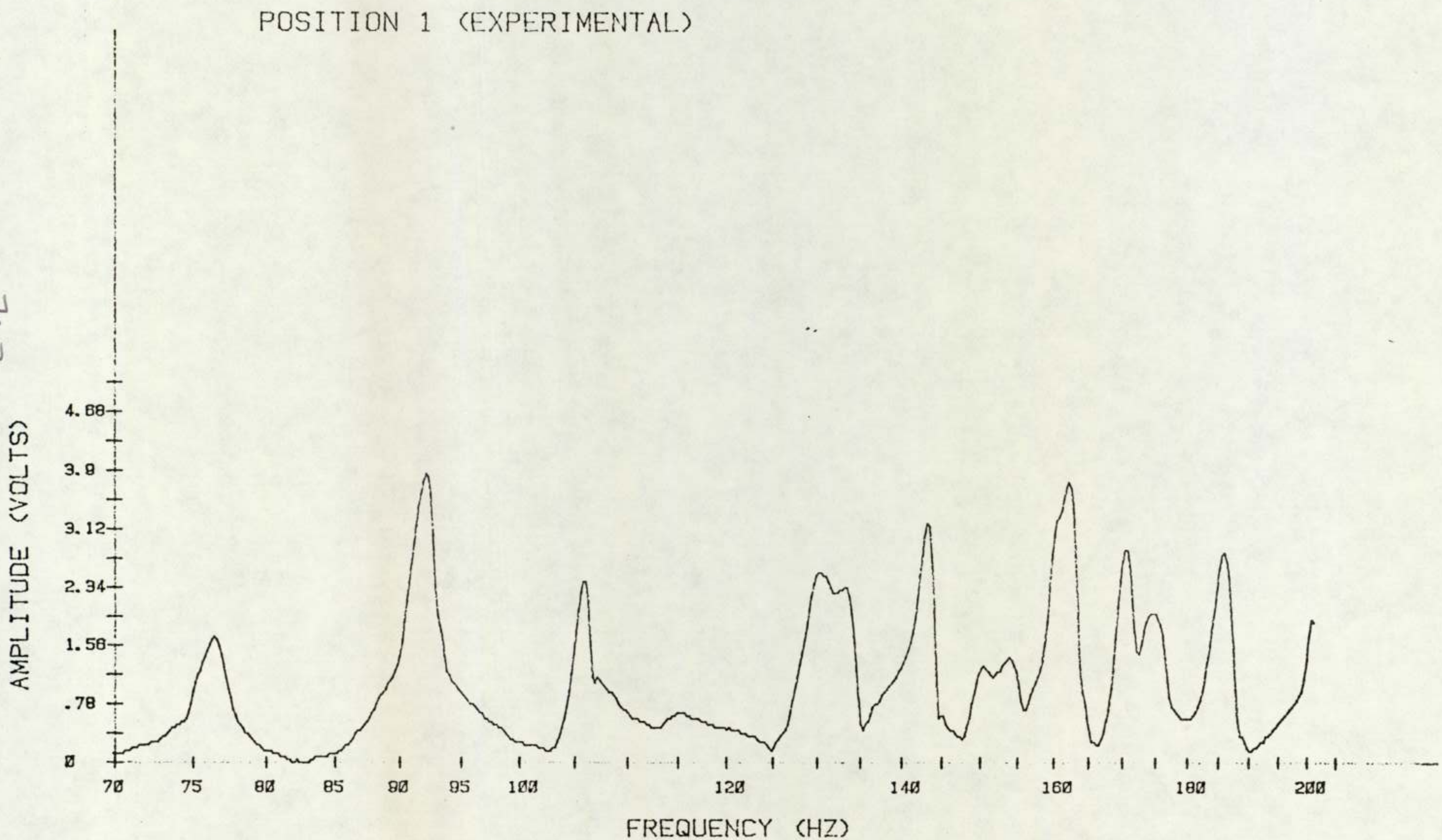


Instrumentation set up for Measuring Frequency Response of the Loudspeaker.



1. Level Recorder
2. Hetrodyne 2010
3. Function Generator
4. Stabilizer Power Supply
5. Frequency Counter
6. Digital Event Recorder
7. Oscilloscope
8. PET Computer
9. H.P. Plotter

FIG. 2.8 Instrumentation Set Up for Measuring the Frequency Response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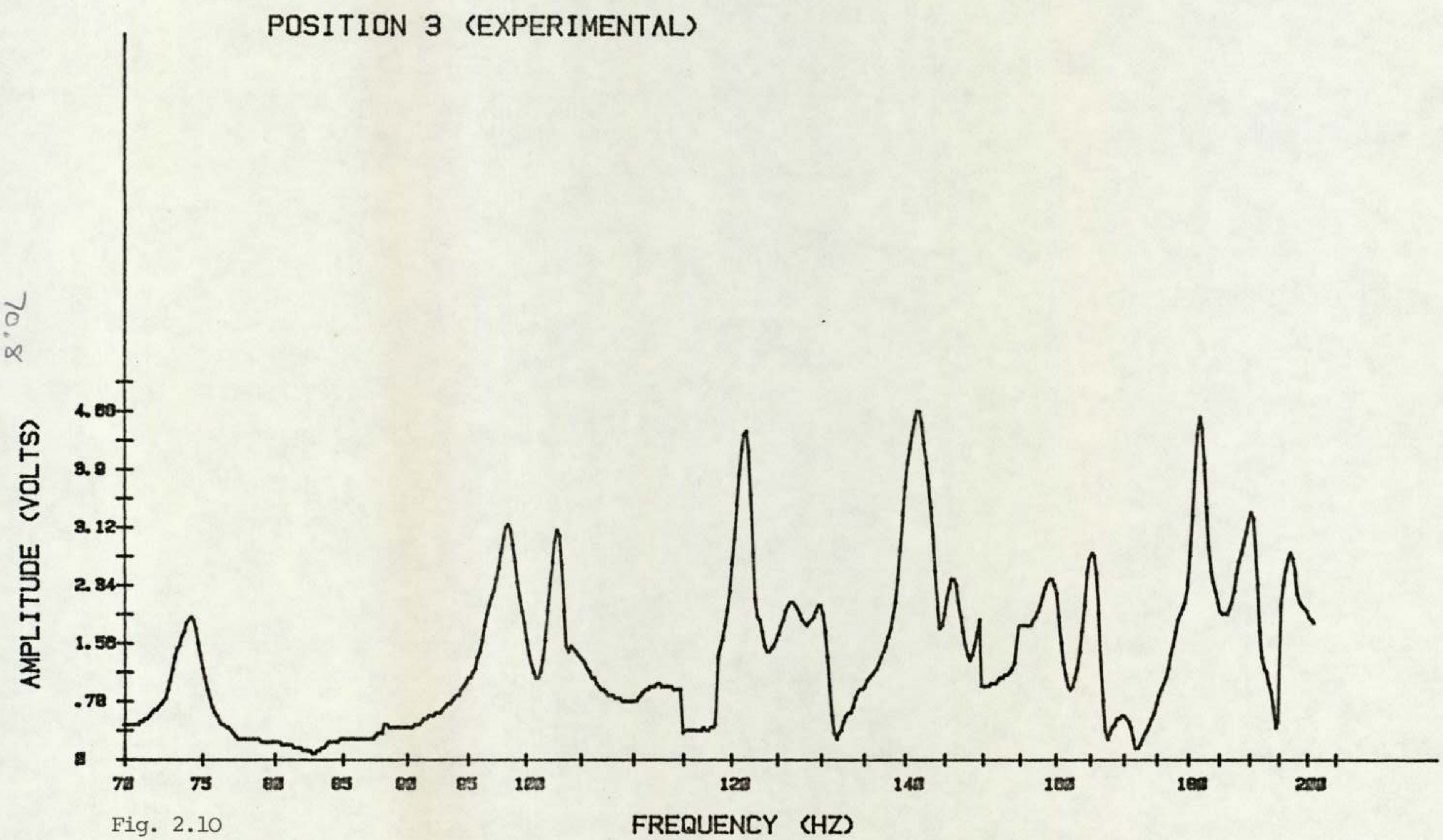
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.9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AMPLITUDE (VOLTS)
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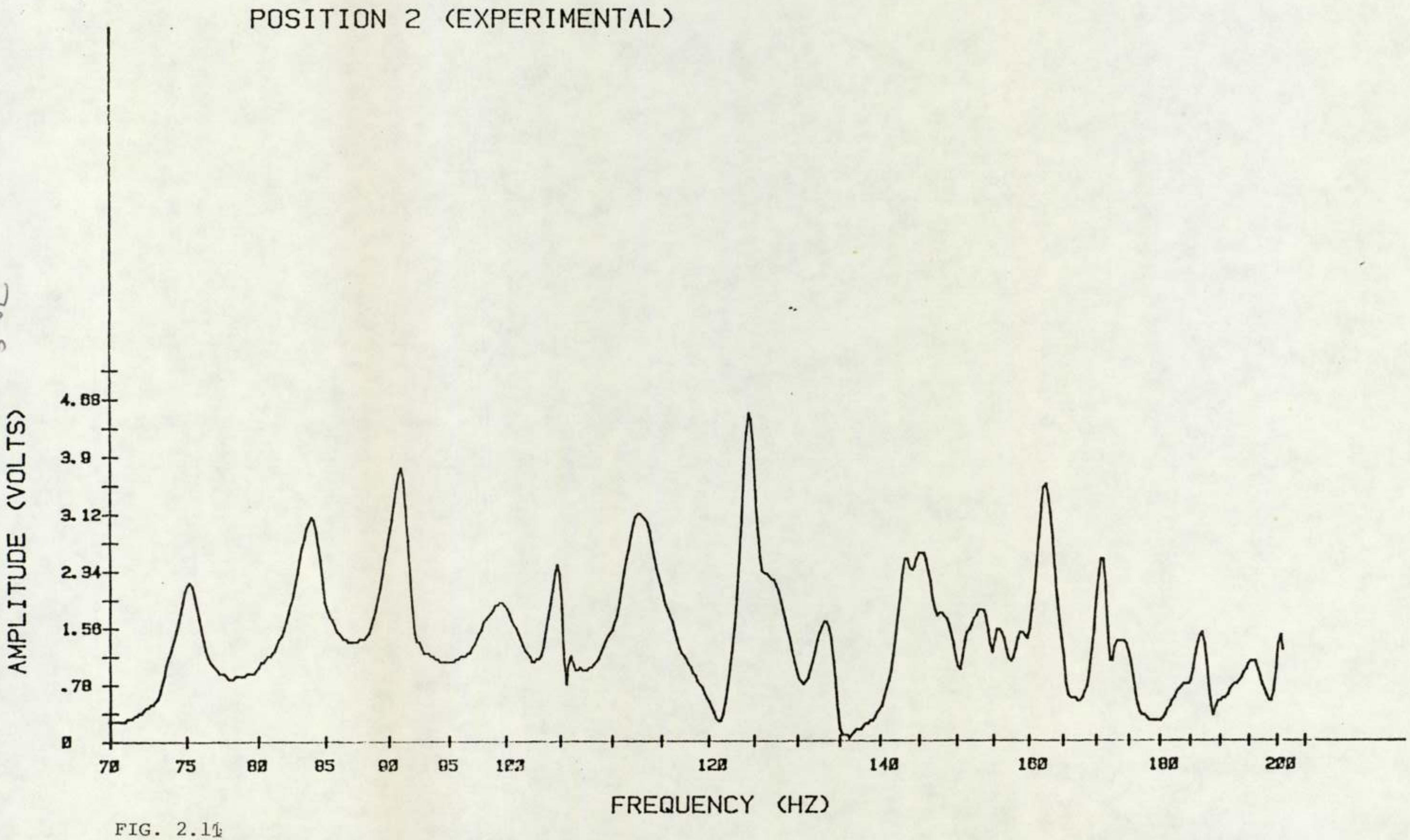
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.10


POSITION 2 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
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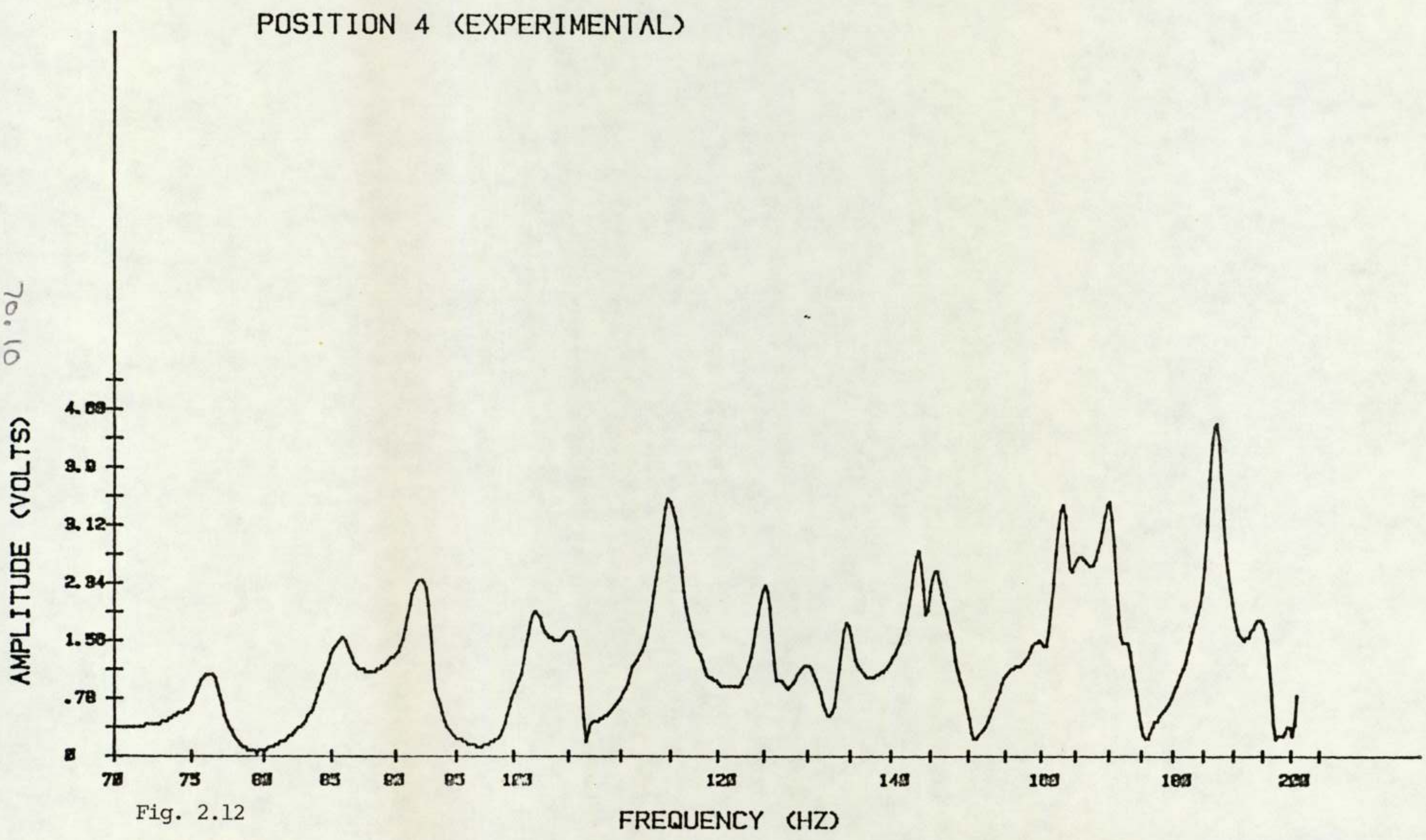
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.11


POSITION 4 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
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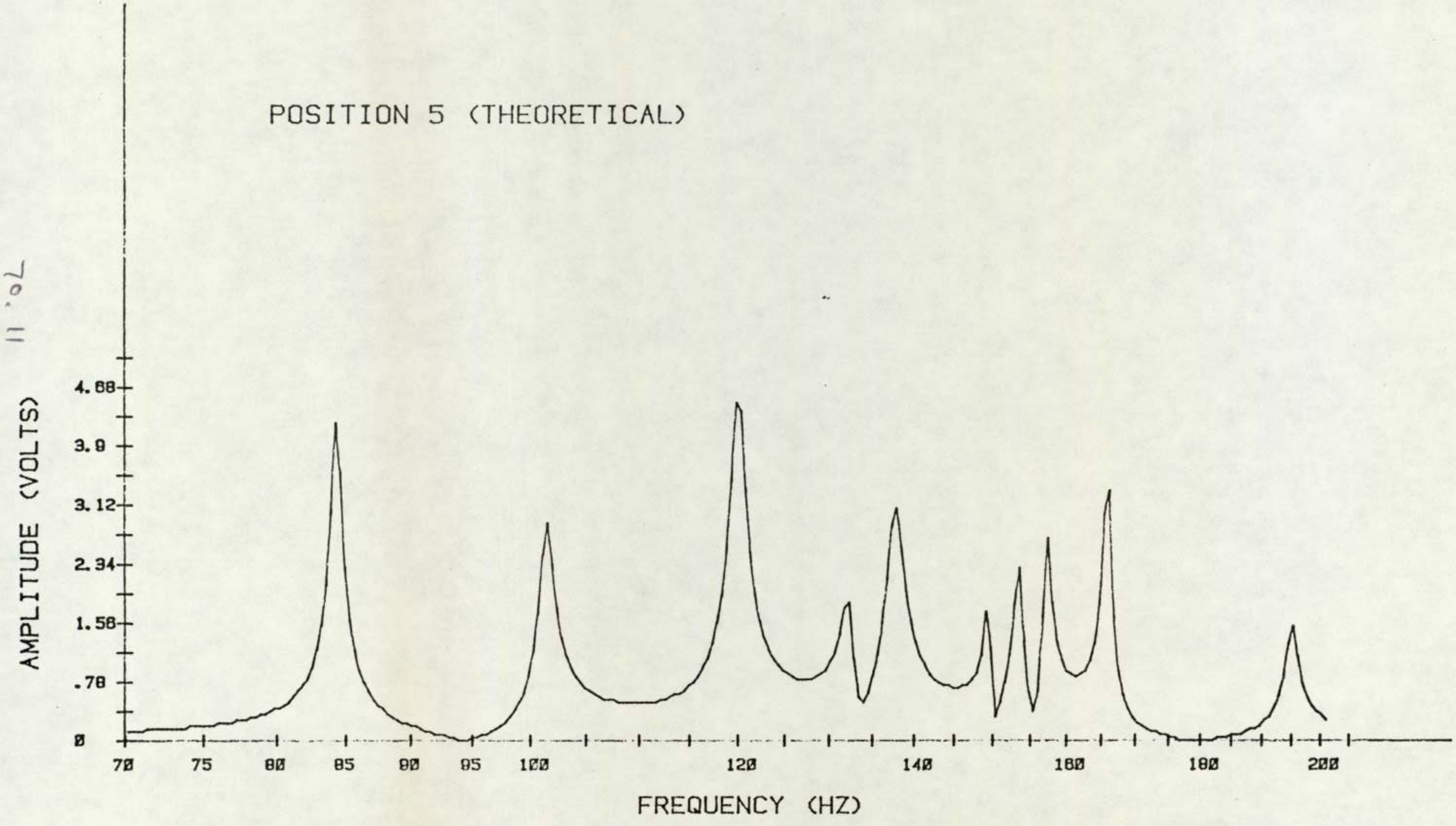
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.12


POSITION 5 (THEORETICAL)
AMPLITUDE (VOLTS)
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3.12
2.34
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.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig.2.13

70.11

POSITION 6 (EXPERIMENTAL)

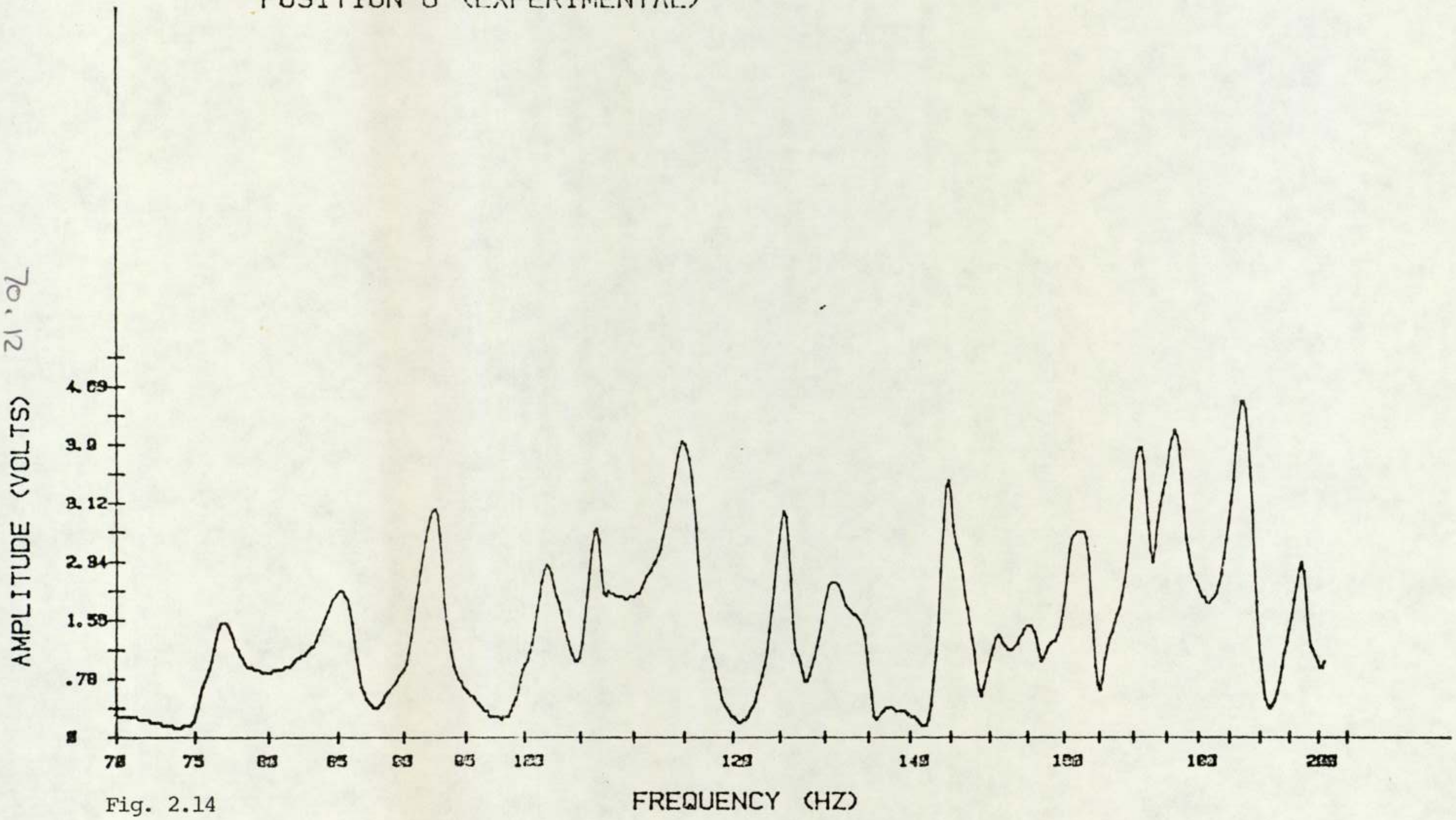
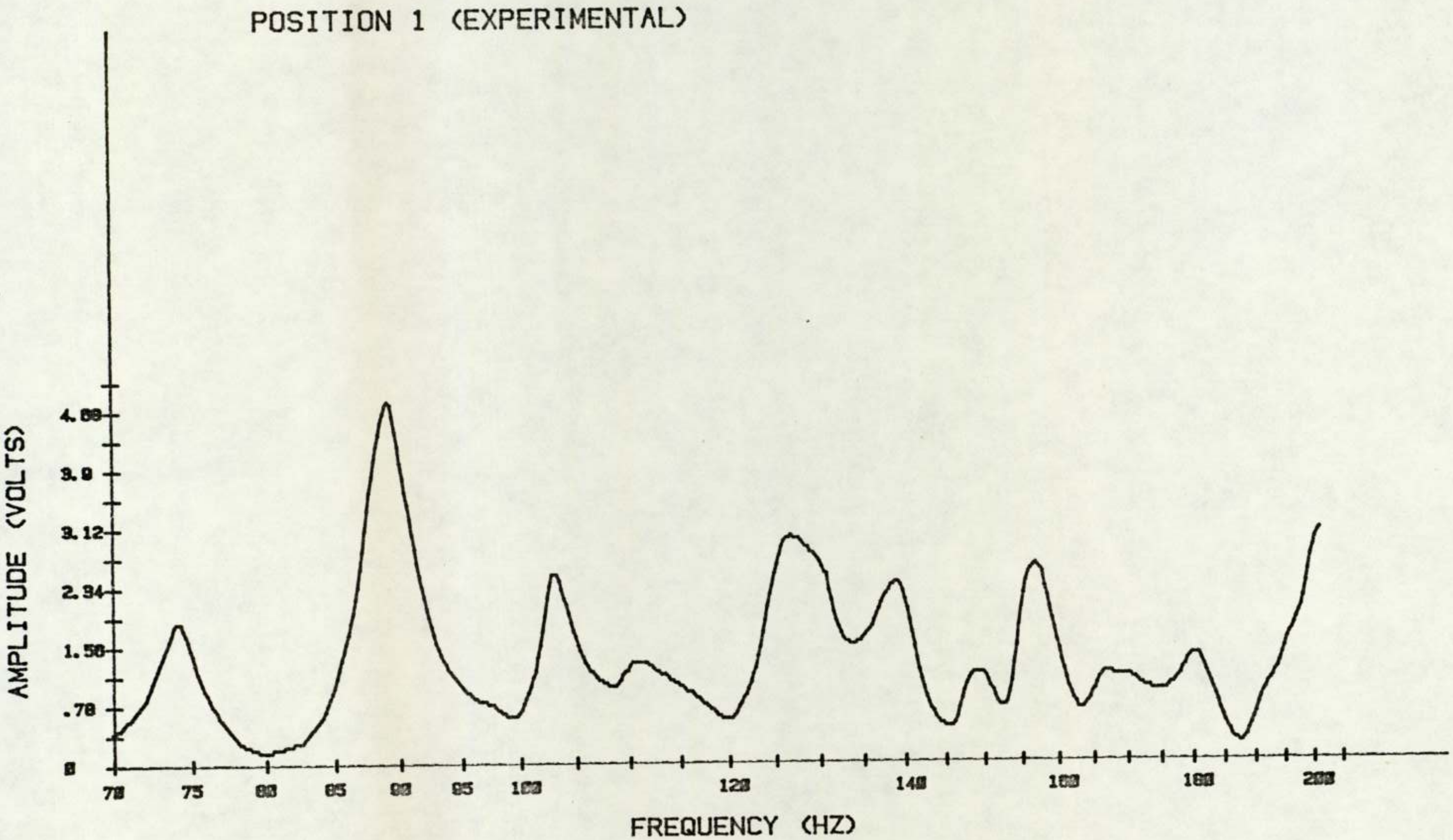


Fig. 2.14


POSITION 1 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
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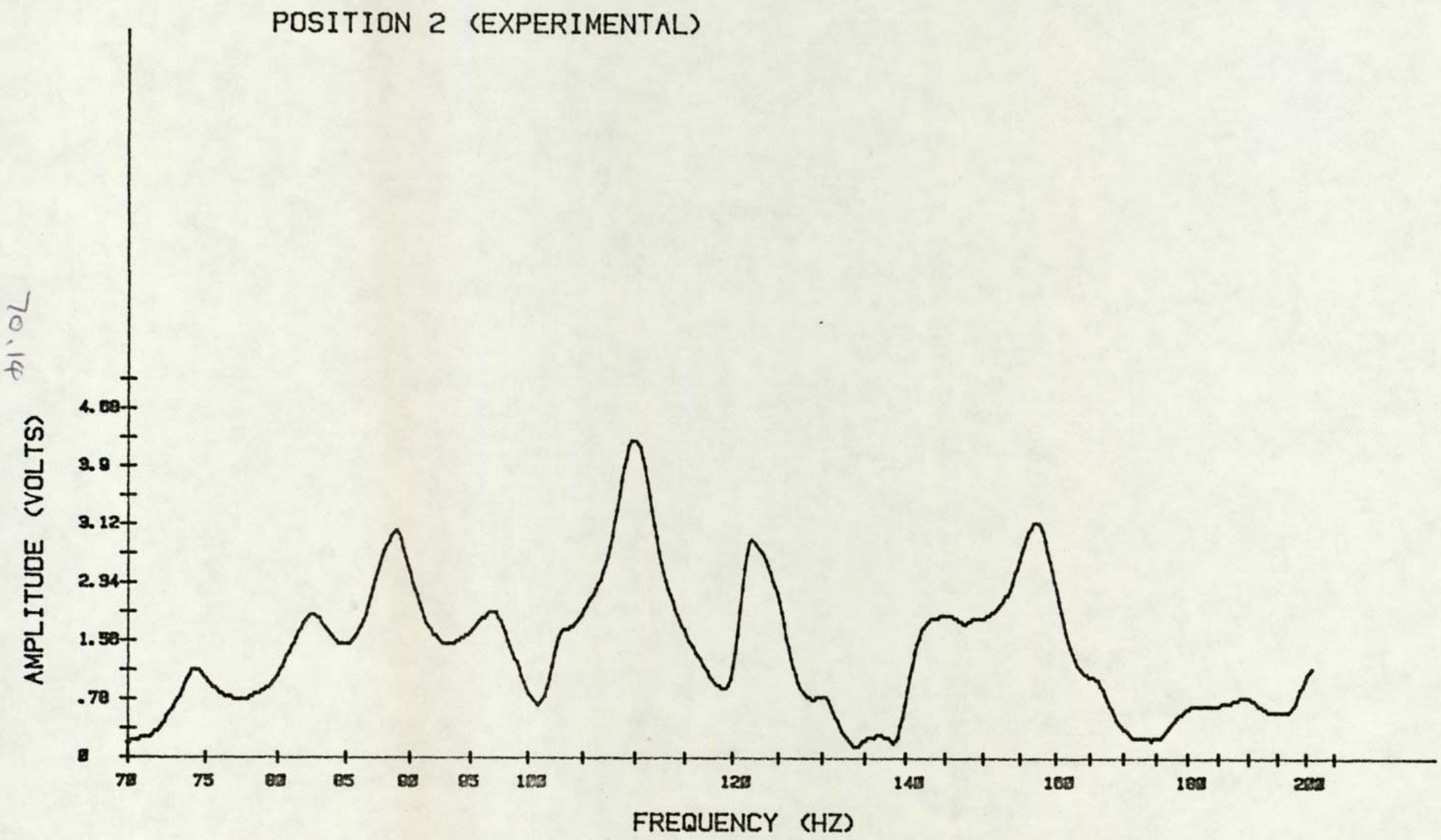
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.15



POSITION 2 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
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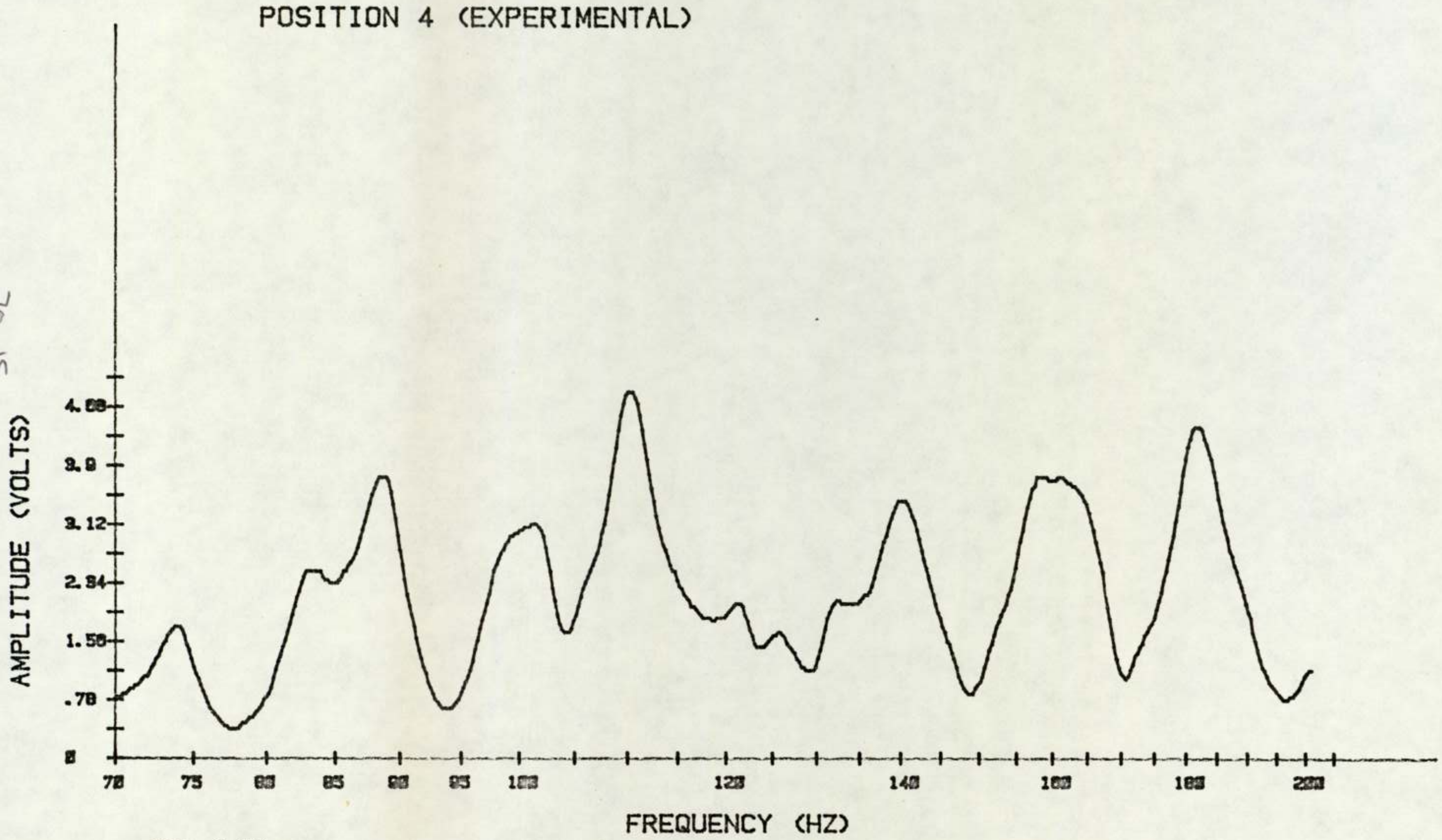
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.16


POSITION 4 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
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3.12
2.34
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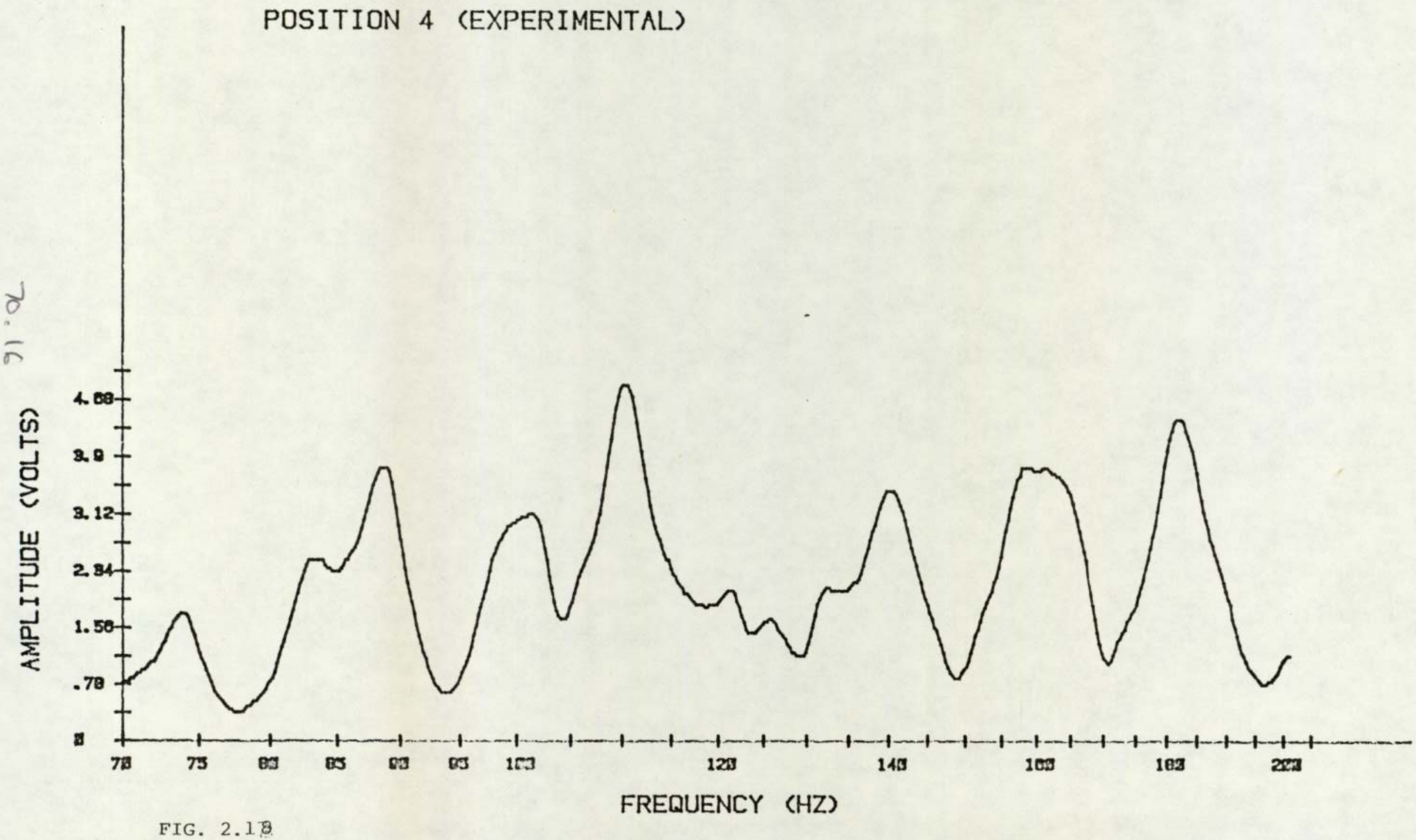
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.17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AMPLITUDE (VOLTS)
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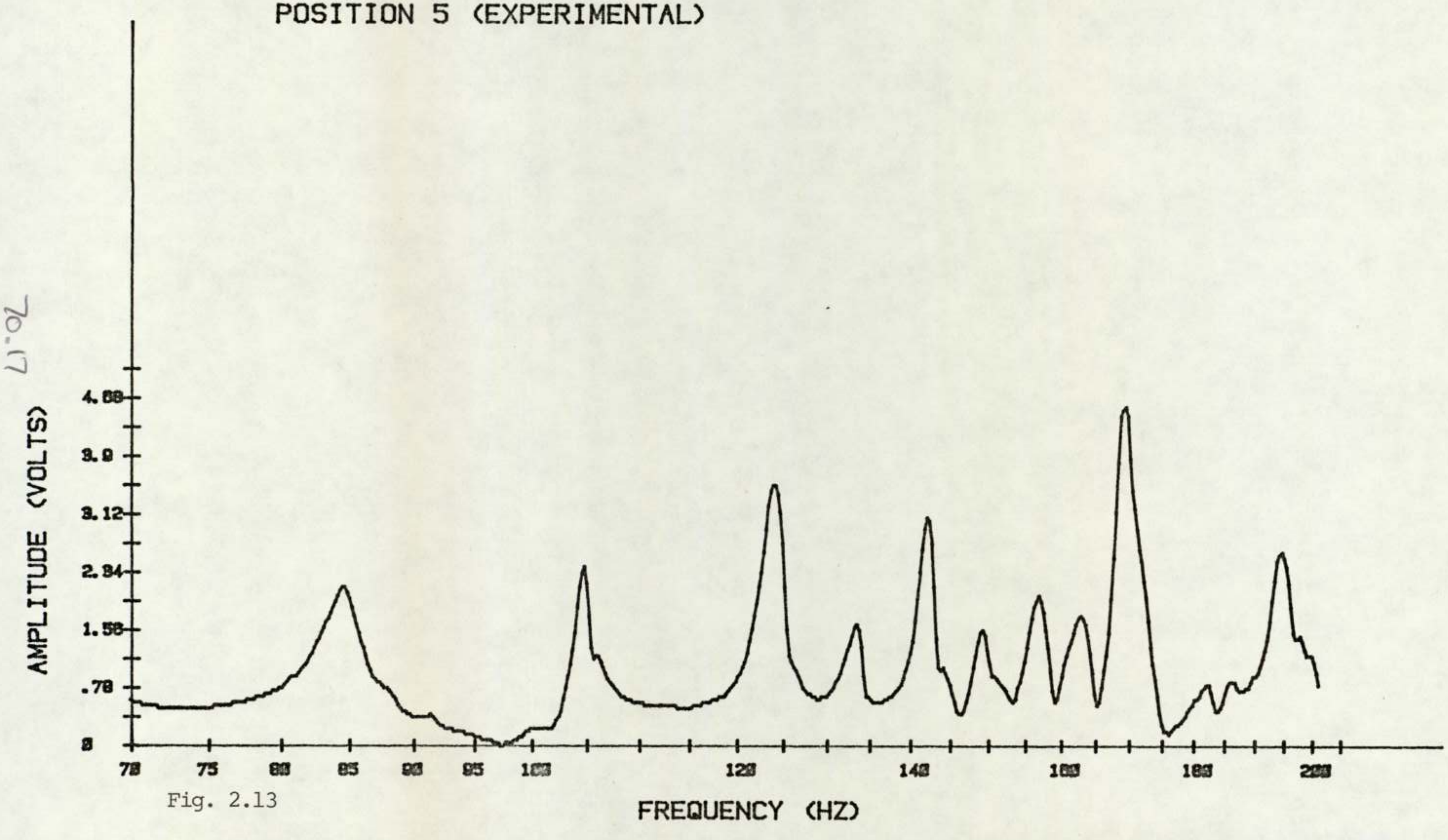
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.18



POSITION 5 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.13

70.17

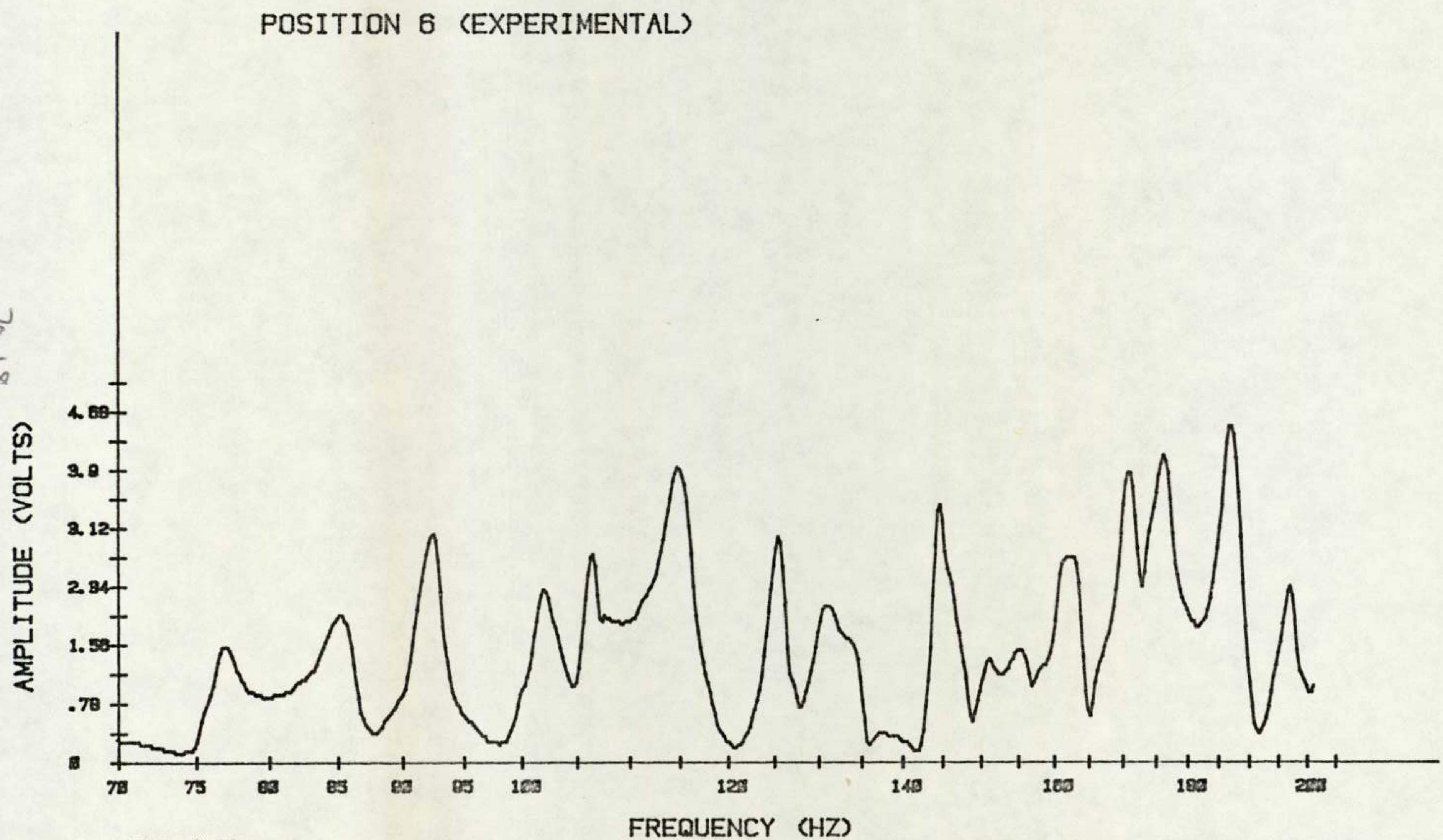
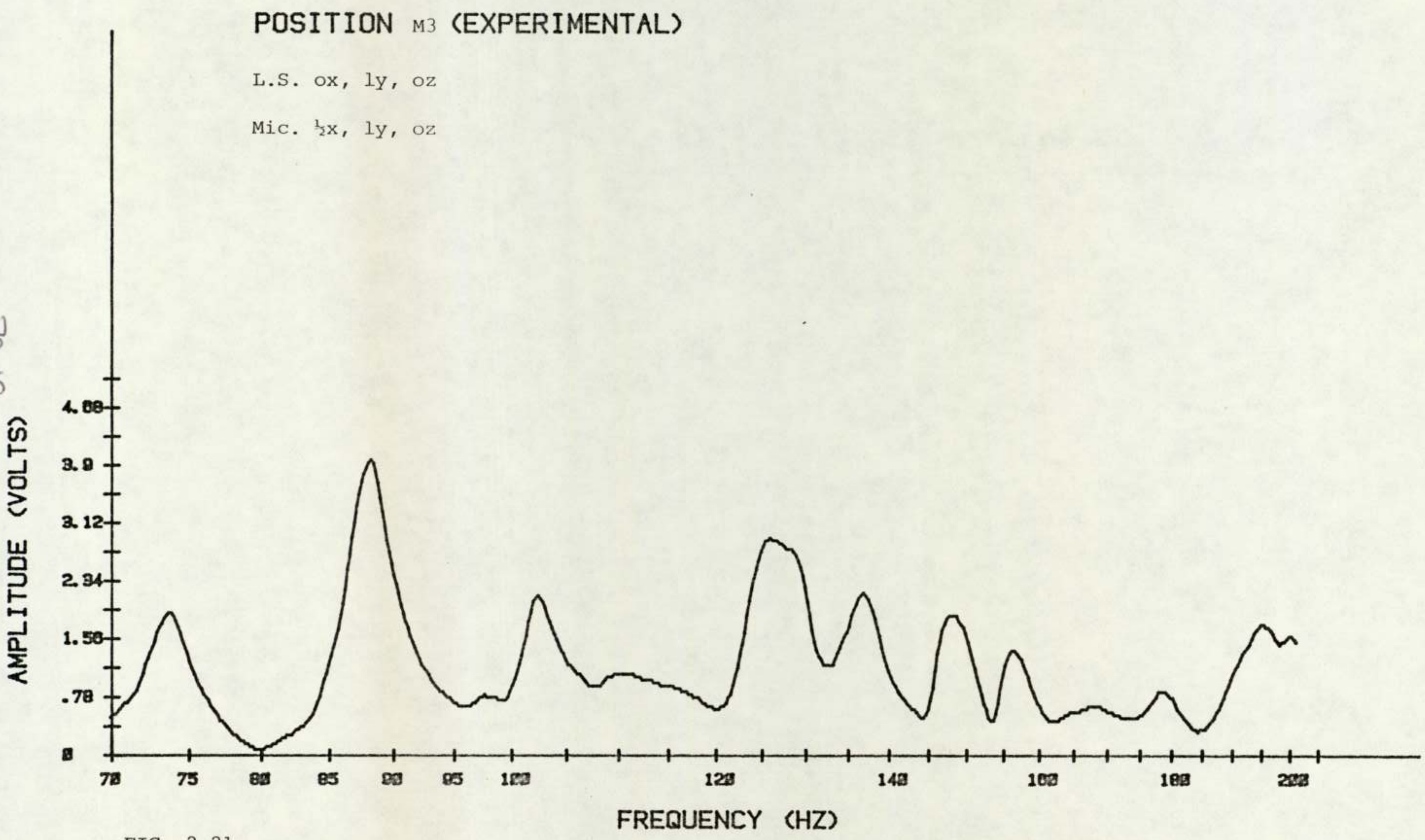

POSITION 6 (EXPERIMENTAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.19



POSITION M3 (EXPERIMENTAL)
L.S. ox, ly, oz
Mic. ½x, ly, oz
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.21

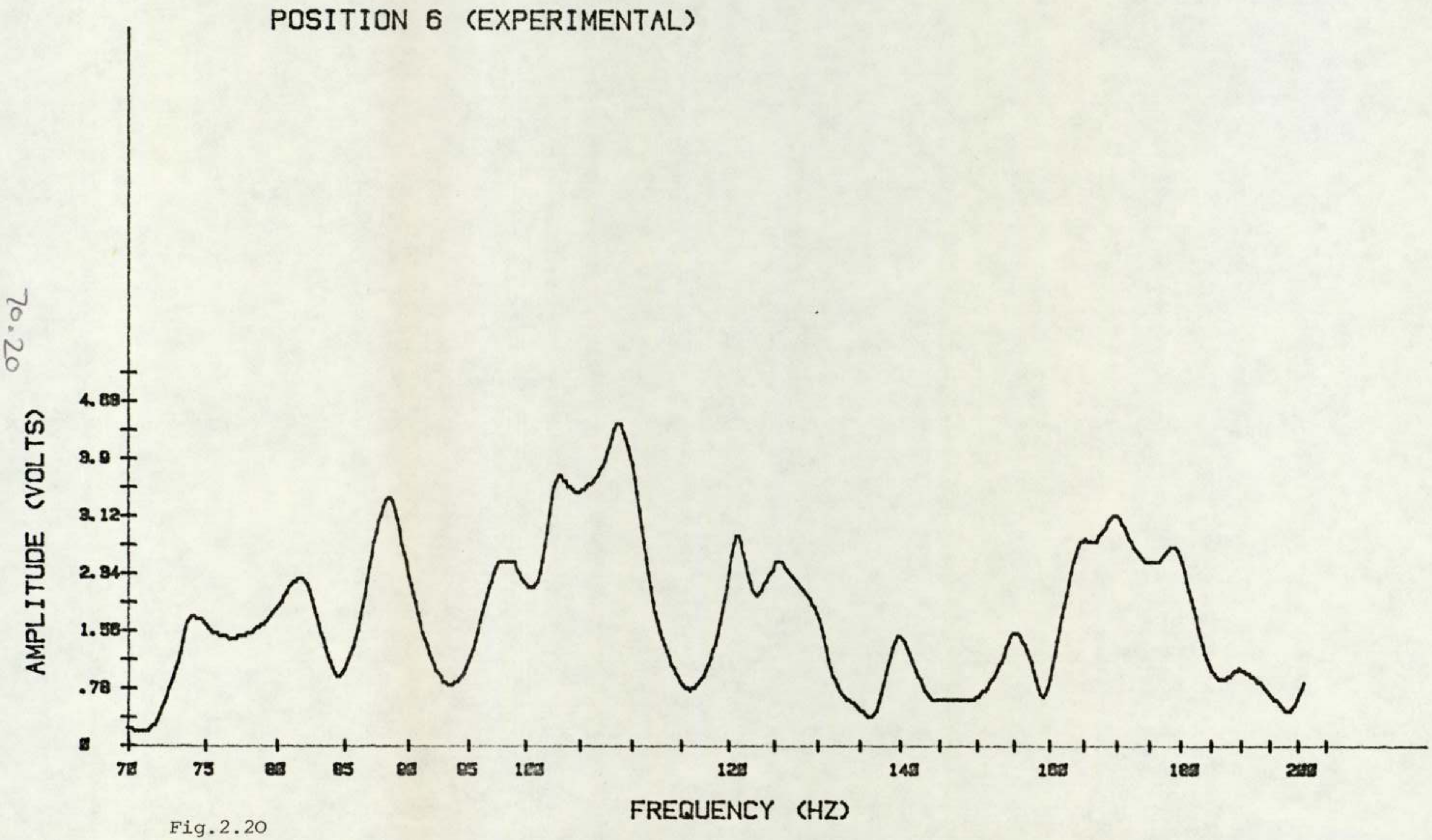


Fig.2.20

70.21

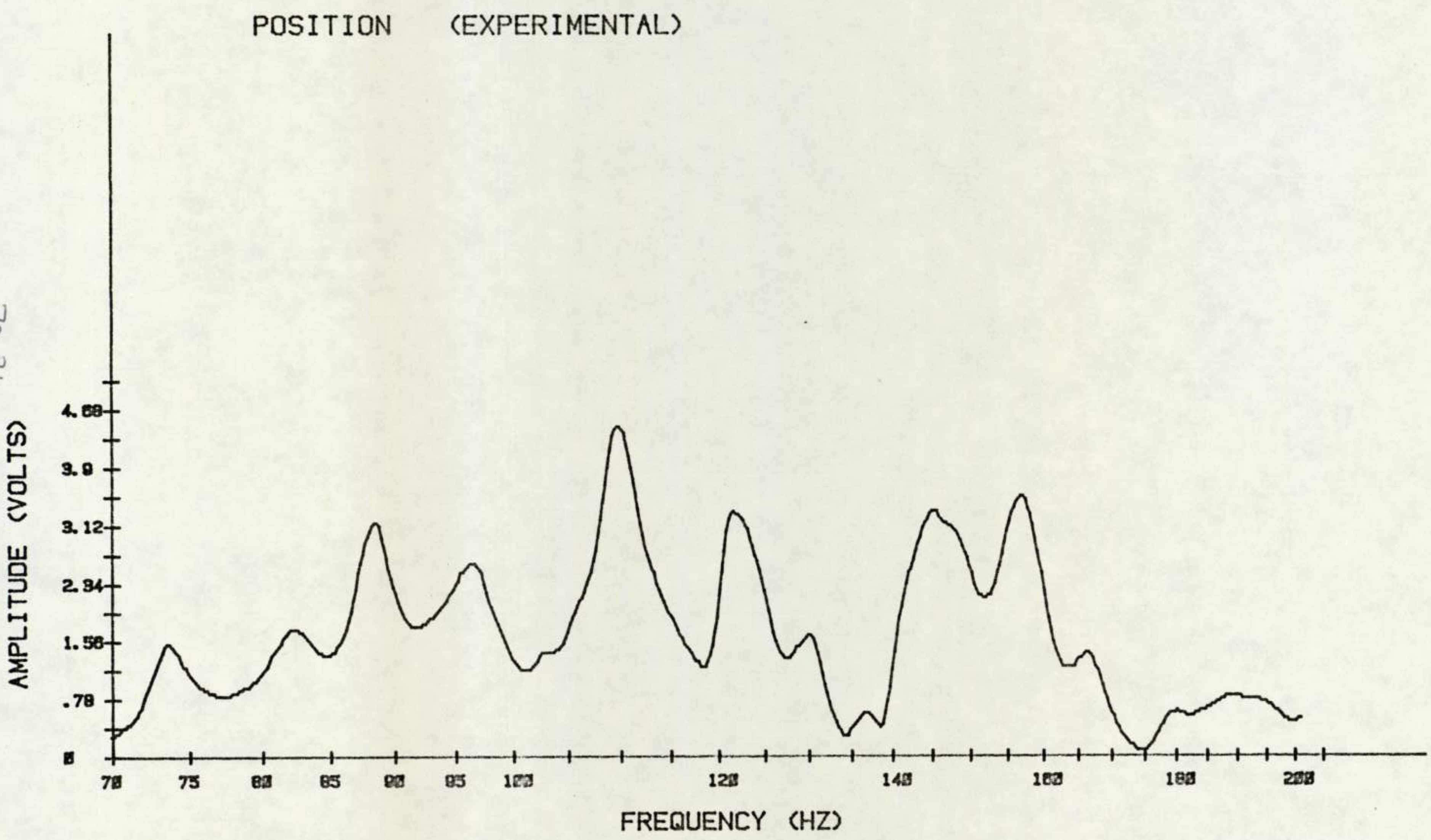


FIG. 2.15(a)

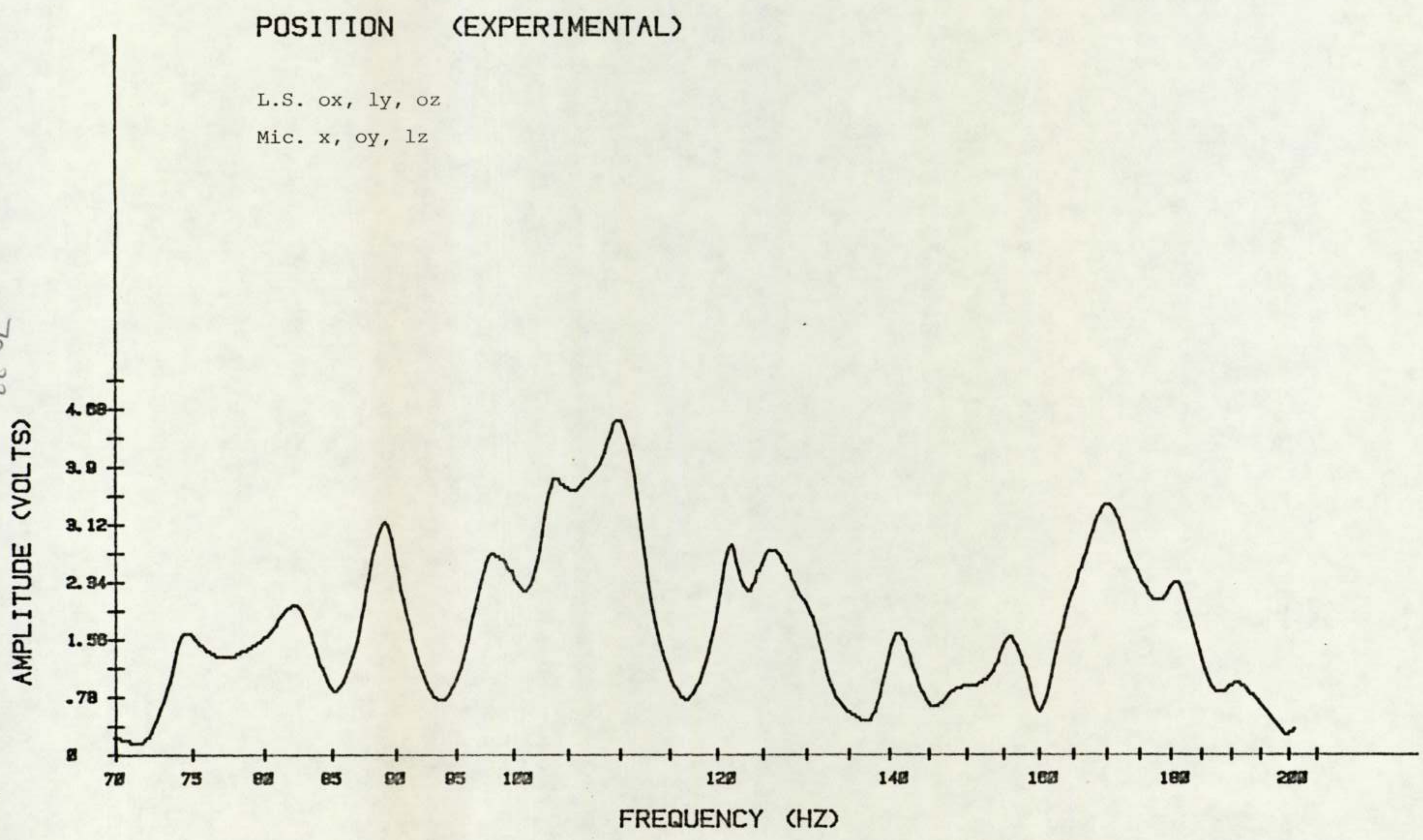

POSITION (EXPERIMENTAL)
L.S. ox, ly, oz
Mic. x, oy, lz
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)
70.22

Fig. 2.16 (b)

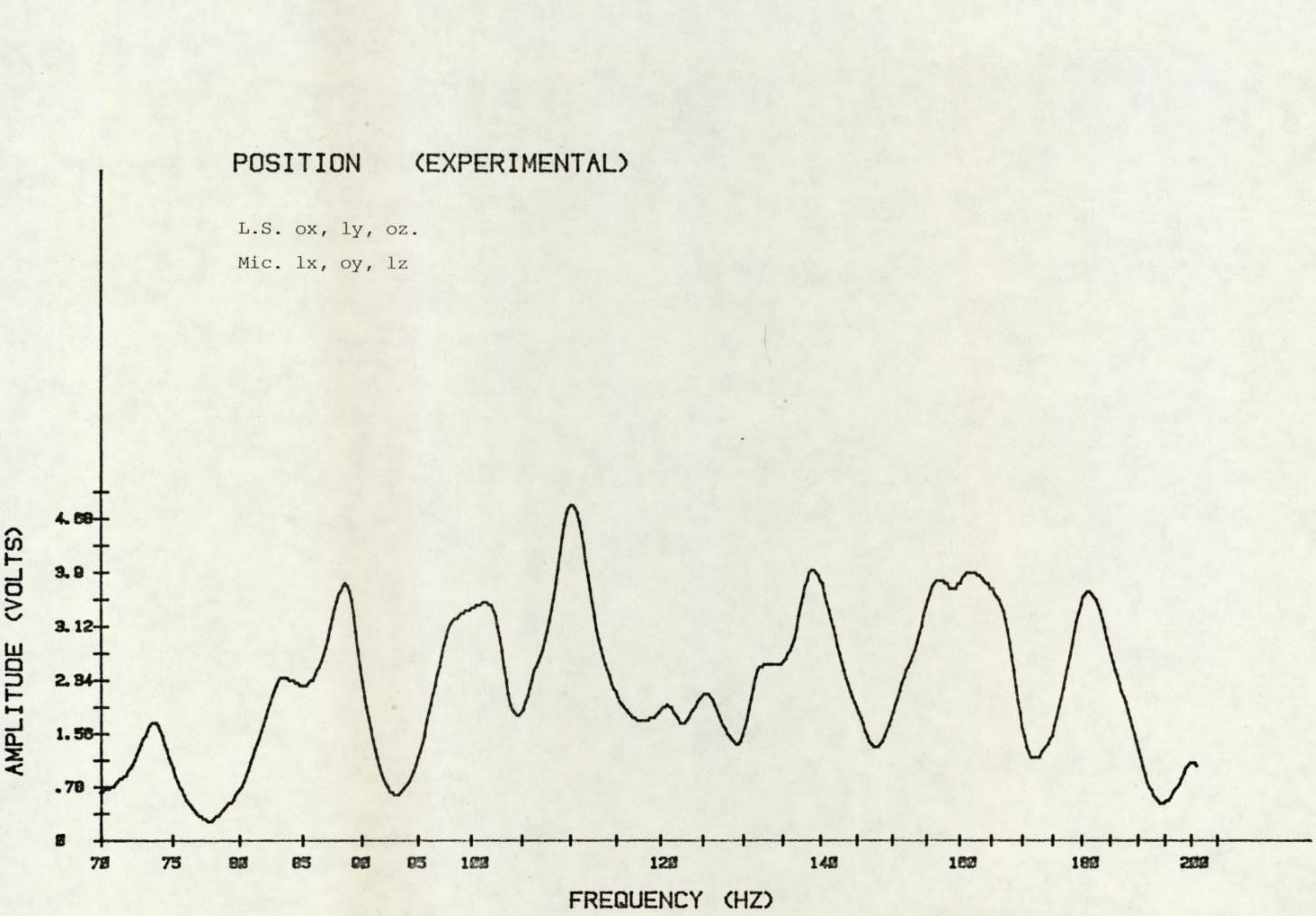
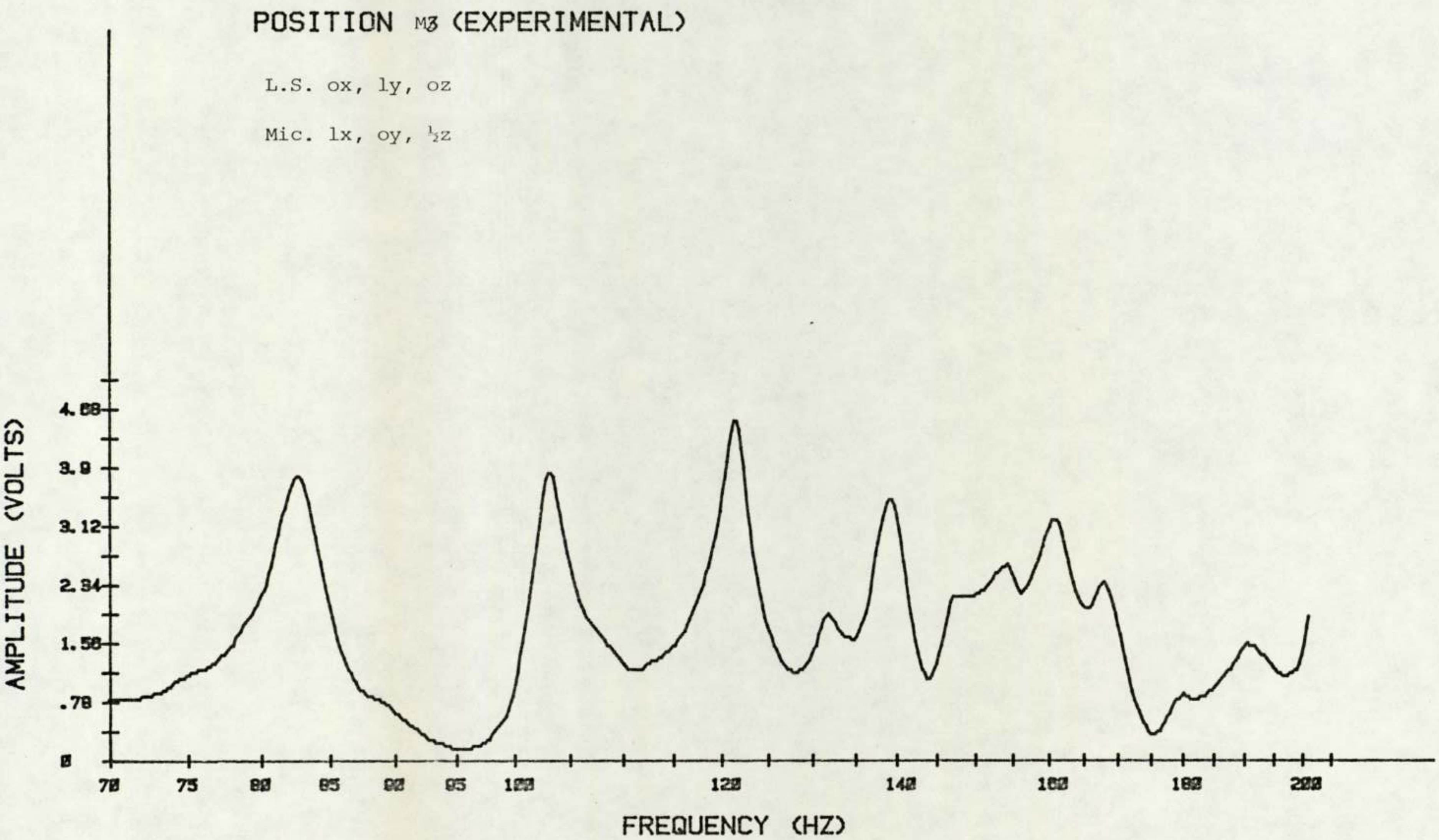


FIG. 2.17(a)

70.23


POSITION M3 (EXPERIMENTAL)
L.S. ox, ly, oz
Mic. lx, oy, ½z
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)
FIG. 2.22

70.24

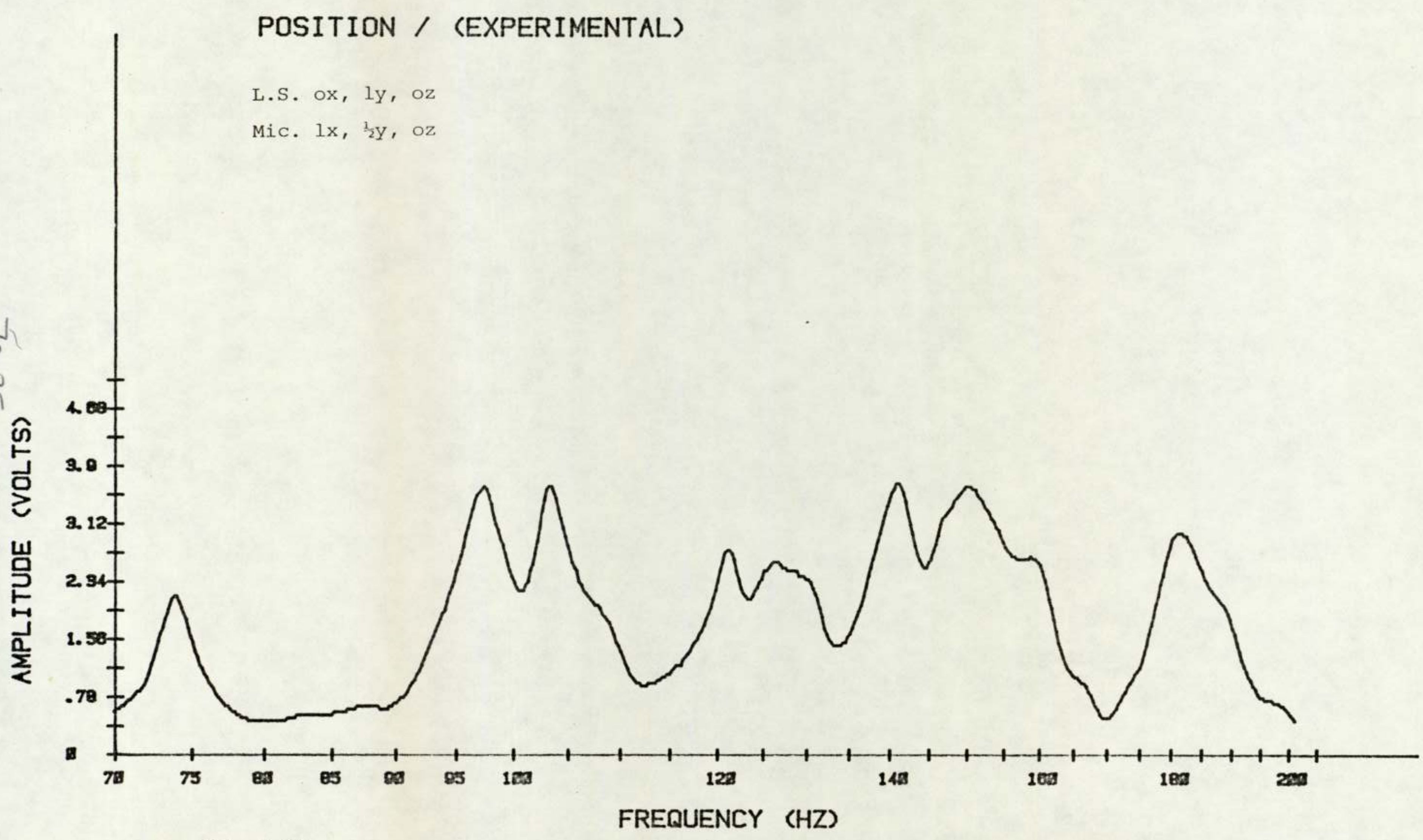

POSITION / (EXPERIMENTAL)
L.S. ox, ly, oz
Mic. lx, ½y, oz
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)
70.25

FIG. 2.23

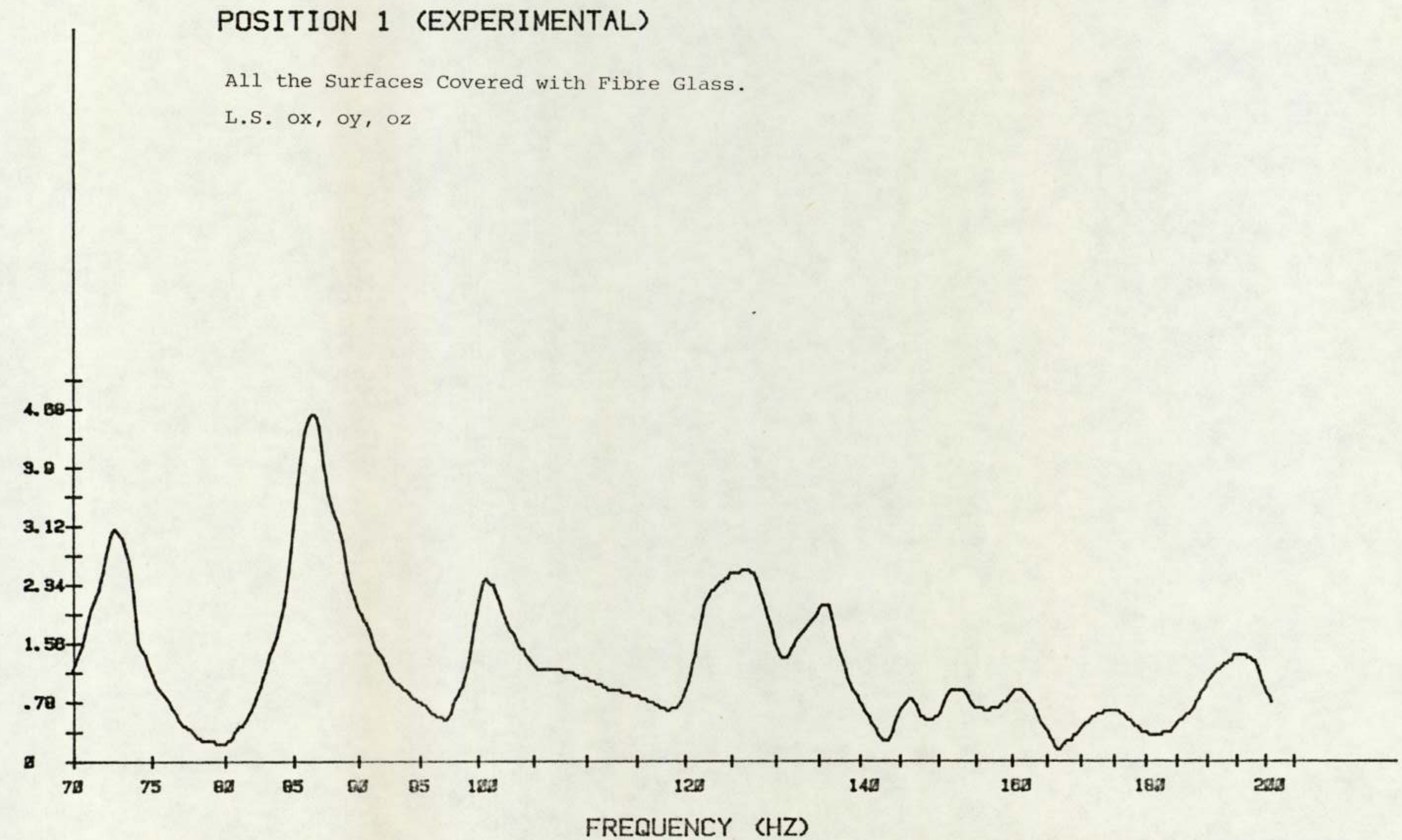

POSITION 1 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass.
L.S. ox, oy, oz
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

Fig. 2.25

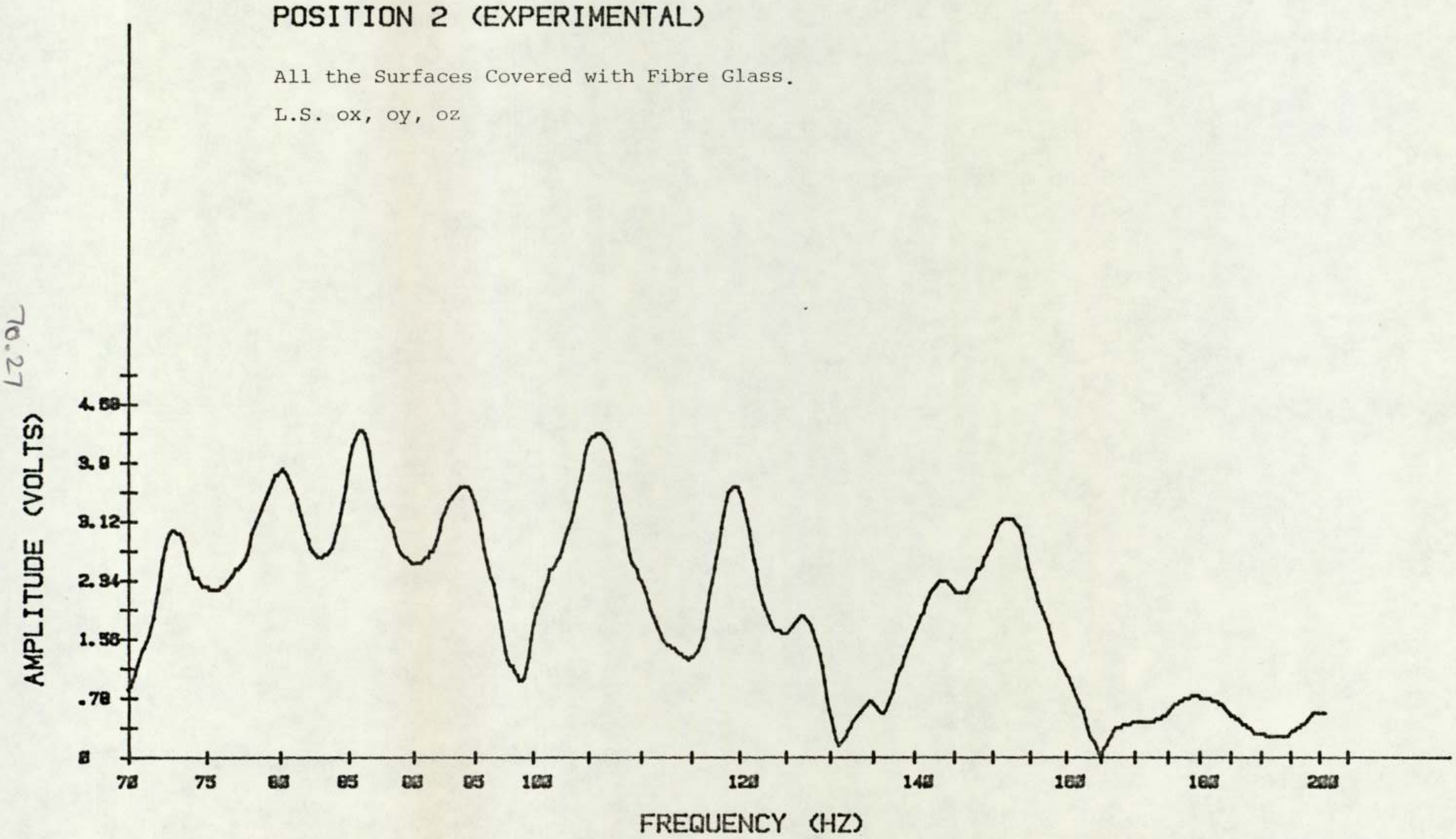

POSITION 2 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass.
L.S. ox, oy, oz
AMPLITUDE (VOLTS)
FREQUENCY (HZ)

Fig. 2.26

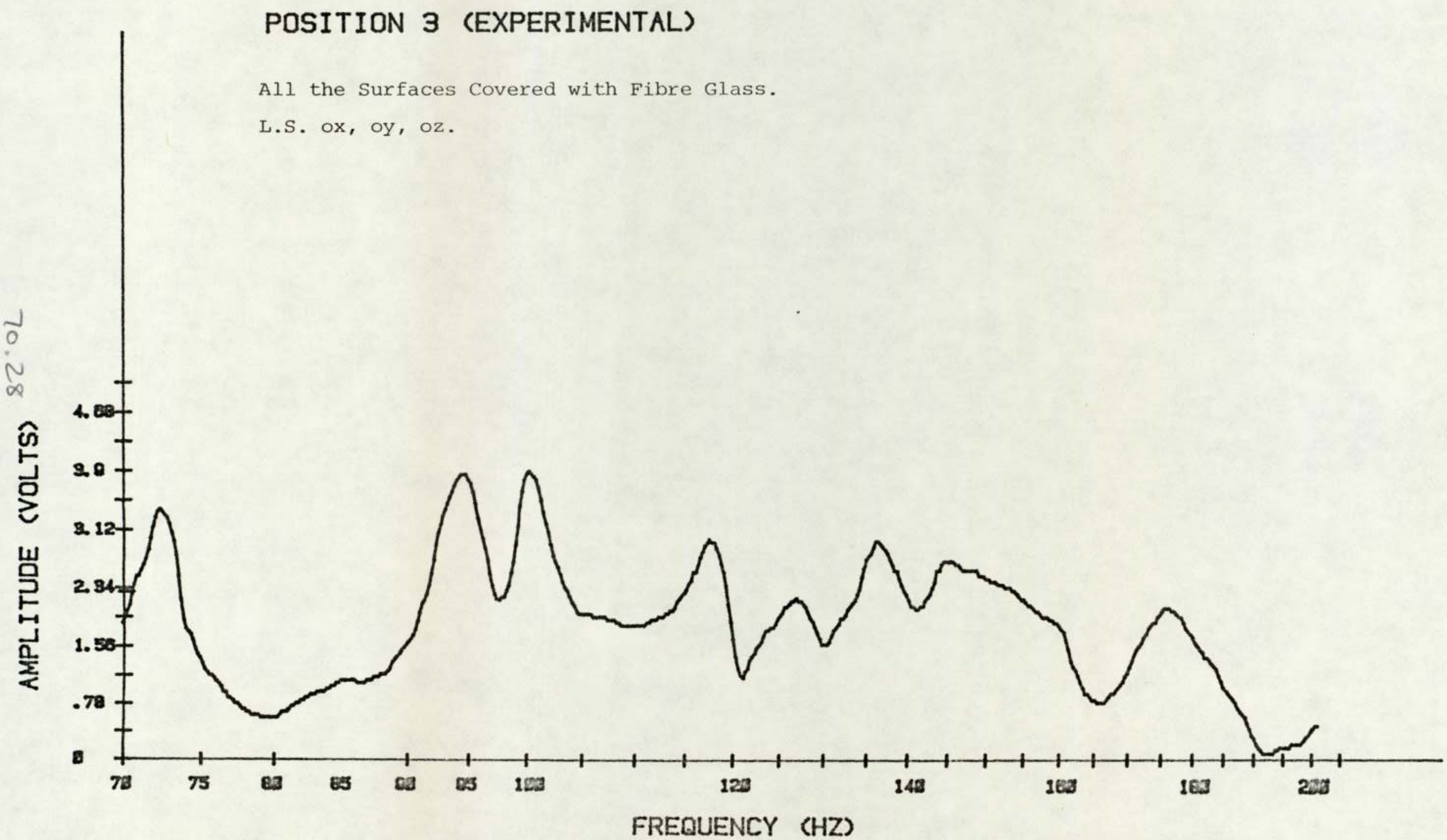

POSITION 3 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass.
L.S. ox, oy, oz.
AMPLITUDE (VOLTS)
FREQUENCY (HZ)
70.28

Fig. 2.27

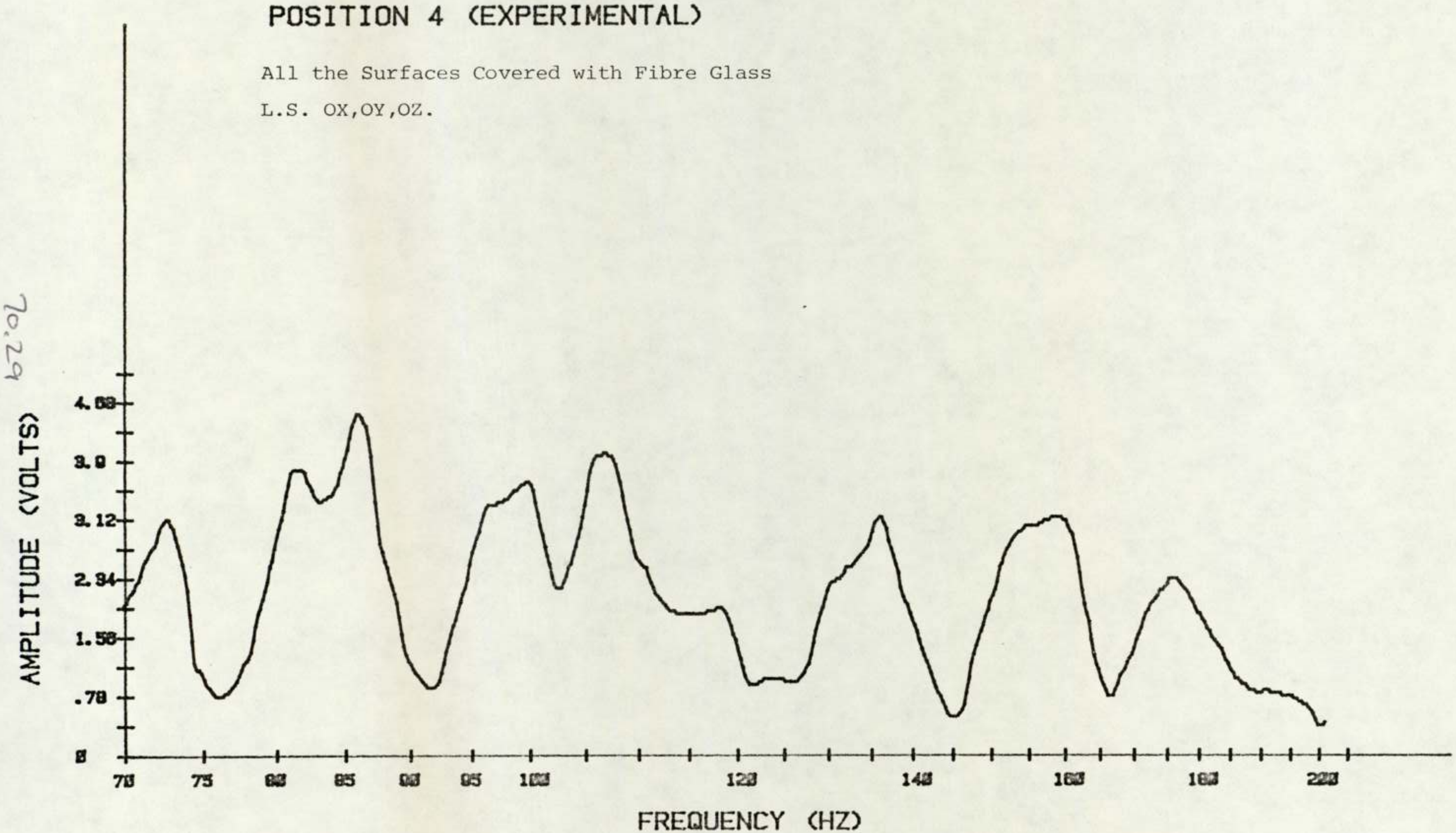

POSITION 4 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass
L.S. OX,OY,OZ.
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.28

70.29

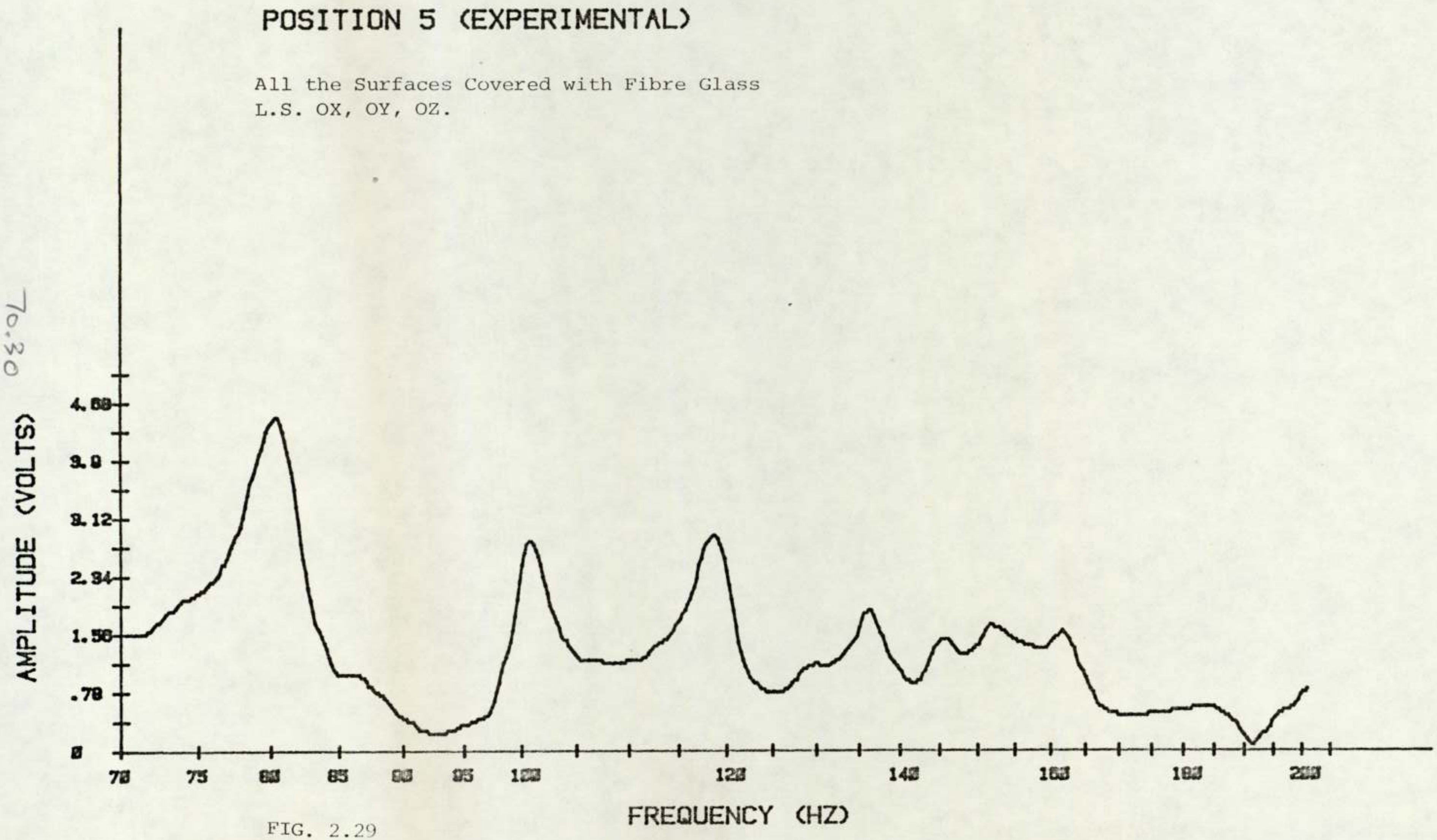

POSITION 5 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass
L.S. OX, OY, OZ.
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.29

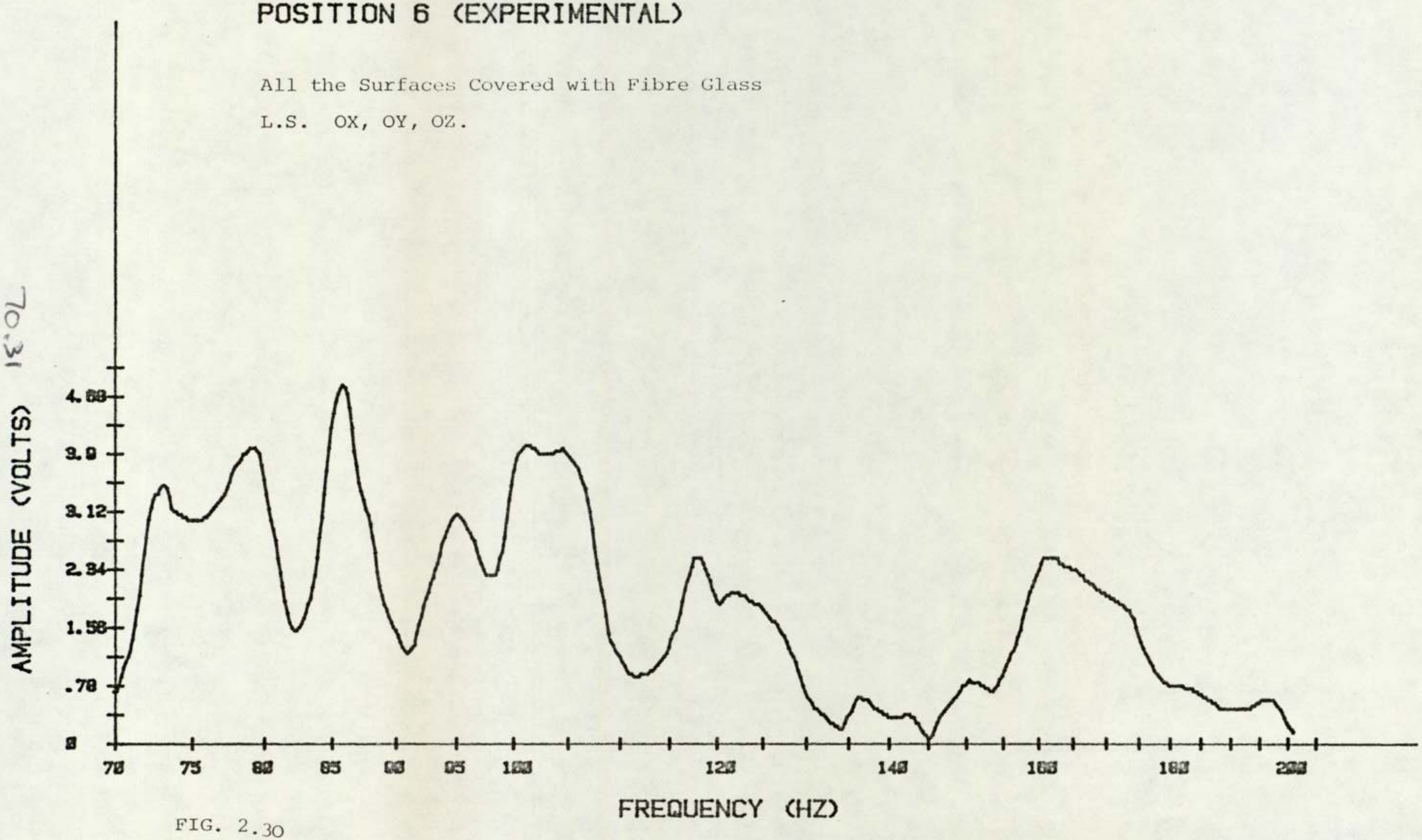

POSITION 6 (EXPERIMENTAL)
All the Surfaces Covered with Fibre Glass
L.S. OX, OY, OZ.
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 2.30

CHAPTER 3

Reverberation Times for Oblique, Tangential and Axial Waves

A rectangular room has walls smooth enough so that some standing waves move parallel to two or more walls and the walls parallel to the motion have less effect on the damping (one half as much, to the first approximation) as they do for oblique waves.

In a rectangular room, when the source is turned off, the various standing waves will take different lengths of time to die out. The oblique waves will damp out most quickly, then the tangential waves, and the axial wave parallel to the most absorbent walls will take nearly twice the time of an oblique wave.

When the source in the room emits sound in a frequency band $d\nu$ (between $\nu-\frac{1}{2}d\nu$ and $\nu+\frac{1}{2}d\nu$) there will be excited on average, dN_{ob} oblique waves, dN_{ta} tangential waves and dN_{ax} axial waves. Each of these wave groups is excited and contributes to the resulting average intensity by an amount proportional to the dN's, and a formula for the average attenuation of sound in a rectangular room can be written as

$$r \simeq r_o D_x(t)\, D_y(t)\, D_z(t)$$

$$D_x(t) \simeq e^{-(c/4v)a_x t} + \left(\frac{c}{2\nu l_x}\right) e^{-\left(\frac{c}{8v}\right) a_x t}$$

where a_x, a_y, a_z are defined as

$$a_x \simeq \sum_{x\,walls} (8k_s)AS$$

where

k = acoustic conductance ratio of the wall material A_s = exposed area of the wall

Therefore, by this approximation, the sound decay curve (the intensity level vs. time) for a rectangular room can be expressed in terms of a sum of individual terms, 10log(Dx) etc. Each of these terms starts at t=0 as a straight line with negative slope proportional to a . A time $(\frac{8v}{ca_x})\ln(\frac{c}{2vl_x})$ later the curve has a break, ending up byond this, as a straight line with a slope proportional to $\frac{1}{2}a_x$, half of the intial slope.

3.1 Reverberation Time at Low Frequency

One of the characteristics of the transmission suite is its reverberation time and it is assumed that if the sound field in a room is diffuse, both in the steady state and in the decay, the decay curves of sound pressure level versus time will be linear. In the papers by Larsen (17) and Bruel (18), two interesting problems connected with reverberation room measurements are pointed out and discussed. The first problem is that the ensemble averaged decay curve reveals a monotonic curvature at low frequencies. The second phenomenon is that often systematically larger sound power output values are reported at low frequencies according to the free field method than according to the reverberation room method. In searching for an explanation of these anomalies some measurements and a classical normal mode theory analysis have been made. It is shown that it is not possible to explain fully the curvature of the low frequency decay curves by means of the normal mode theory. The measured curves are not as linear as their respective theoretical ones. It is possible to explain this lack of agreement by the fact that the absorption characteristics of normal reverberation chambers significantly deviate from the situation of uniform wall admittance which has been assumed in the theoretical deductions. The theoretical analysis and comparison between theory and practice indicate that the damping characteristic of the individual waves varies much more than is predicted for a uniform wall admittance. This reasoning is supported by the observation that the monotonic curve increases when a plane concentrated absorbent is added to one of the walls. One way to decrease the curvature has also been identified. When the room surfaces are provided with randomly placed small samples of low frequency absorbents the resulting decay curves turn out to be almost perfectly linear.

It is shown that the normal mode theory does not imply significantly different sound output values than the ISO 3741 model. This fact has been verified with a comparative test. According to the normal mode theory, the average sound power output as measured in the reverberant room should equal the free field output (19). Therefore, iit is possible to think that the analysis of the classical normal mode theory fails in explaining the anomalies mentioned.

When estimating the damping characteristics of a room, the short time averged mean square pressure can be ensemble averaged in respect both to several noise excitations and several microphone positions to reveal the true nature of the energy dissipation in the room.

Each separate vibrational mode of the sound field decays exponentially and has its own specific damping constant. If the various dominating modes have nearly the same damping constants, the energy dissipation of the sound field is exponential and the corresponding logarithmic reverberation curve is linear. This is what one normally observes in typical measurement situations, at least in the middle and high frequency ranges. However, if the damping constants are significantly different, the decay shows a convex curvature, as theoretically demonstrated in Ref.(20). This fact explains qualitively the monotonic curvature which has recently been independently observed at low frequencies by various investigations (17)(21).

3.2 Measurement of Reverberation Time at the Resonance Frequency

The reverberation time of the empty room at each theoretical value of resonant frequency was measured. The actual decay of sound energy inside the room did not follow a straight line as expected. The decay curve started as a straight line and later on a break and kink appeared along the straight line which followed on.

The pure tone of sound was generated and the reverberation time was measured for the sound at a certain frequency.

As mentioned, the curve for the decay of sound energy did not follow the normal path and often a break and a kink on the path was noticed (as shown in Figures 3.32 and 3.33). This could well be due to the fact that when the room was excited at each theoretical resonance frequency, the adjacent frequencies were also excited, especially when two resonant frequencies were close to each other, and as these waves died out they might have alternatively reinforced and interfered with each other as they were of slightly different frequency. The intensity then fluctuated instead of decreasing uniformly and the sort of fluctuations obtained depended on the position of the source aand the microphone in the room. This also occurs when the two opposite walls are more absorbent than the rest.

To separate the frequency bands from each other the Kennedy and Pancu Method was employed.

Figure 3.49 shows the set up for measuring the reverberation time at each of the resonant frequencies. The reverberation time was measured at different positions inside the room and the average of it is then taken. The decay of sound did not follow a straight line and other modes which were closely associated with the resonant mode were also excited, since the pure tone sound generated by the Hetrodyne 2010 as shown in Figures 3.32 and 3.33.

Two methods were adopted to find the Reverberation Time:

i) In the case where the break occurred at the early stage of the decay curve and the remainder of the curve followed a linear decay, a straight line is plotted through the larger section of the decay curve, ignoring the early stage of the decay as shown in Figure 3.32.

ii) Where the break occurred halfway along the decay curve, the reverberation time was calculated using two of the options as follows:

a) two straight lines were plotted separately for each part of the decay curve that were separated by the break, and the average of the two values of the reverberation time as shown in Figure 3.33.

b) one straight line to fit two sections of the decay curve best was plotted, and this was taken to be the decay line and the reverberation time measurement was based on this line: Ref. Figure 3.33.

When the break occurred in more than two positions along the decay curve, there were always two sections of the decay curve that consisted of the longest path of the decay curve, which were taken and the reverberation time was calculated considering the two paths. Ref. Figure 3.33.

The Kennedy and Pancu Method

The system under investigation is excited harmonically, as in other forms of resonance testing, and the variation of the transfer function of the response with the driving frequency is measured. In this technique, its components in phase with and in quadrature with the driving signal was measured, and from then on their plots information is derived from modes identification and determination of the resonant frequencies and damping.

Some of the assumptions that are made in the Kennedy and Pancu method are as follows:

1) There is no coupling between modes of a system.

2) The shape of each normal mode is fixed for a given system and is independent of the magnitude frequency or location and direction in space of the applied force.

3) For small damping, the mode shapes, phase relations and coupling is assumed to be unaffected as in the case of zero damping.

4) The contribution from off-resonant modes around the natural mode is effectively constant and can be extracted and measured from complex receptance plots.

Display of Results

The vector diagram was obtained by plotting the receptance components in-phase with an in-quadrature to the acoustic operating signal. The receptance of a single, isolated mode, when plotted in this way, gave a pure circle, centred on the phase axis of the complex plane (as shown in Figure 3.40).

Several modes of different natural frequencies and damping combine to give a complex system of loops, curves and knots; a number of them can be identified as circles or parts of circles of varying diameters, and by examining the rate, with respect to frequency, at which the vector moved round the circle, the modes of the system could be identified.

A detailed analysis of a particular loop could yield information on both the natural frequency and damping of the mode.

Vector Presentation of the Receptance

Convention

Complex qualities are represented on the complex plane as rotating vectors according to the sign convention adopted in Figure 3.41.

An input function is represented by the vector $v_e e^{jwt}$, the factor e^{jwt} signifies a relation of the vector v_e, through an angle (wt) with respect to the real axis, in an anti-clockwise direction.

The response function of the system is given by another vector $v_a e^{j(wt-\alpha)}$ also rotating in an anti-clockwise direction and lagging behind the input function by the angle (α).

The transfer function of the system is then:

$$F = \frac{V_a e^{j(wt-\alpha)}}{V_e e^{jwt}} = V_R e^{-j\alpha}$$

which is a vector of magnitude v_R and a phase angle (α) rotating in the clockwise direction. If:

$$F = (x + jy) = V_R e^{-j\alpha}$$

Then $$|F| = V_R = (x^2+y^2)^{\frac{1}{2}}$$

and $$\alpha = \tan^{-1} \frac{y}{x}$$

Considering systems having more than one degree of freedom, like a room, neighbouring modes to the particular one of interest are likewise excited to a degree depending on their closeness in frequency and spatial distribution to the resonating mode.

As the driving source is sweeping through a certain frequency range and in the immediate vicinity of resonance of a mode, the off- resonant contribution is both small and constant compared to the resonant vibration and the tip of the total receptance vector will tend to describe partly a circle corresponding to that mode. As the frequency departs from the resonant range of that particular mode, the off resonant contribution increases and the circular plot becomes distorted.

If the off-resonant vibration is not small but assumed to be constant over the resonant frequency range of the mode, all points will be shifted due to the extraneous vibration, and a new displaced origin has to be assumed.

In the case of the multi-degree of freedom systems, e.g. where room modes are closely spaced in a particular frequency range, the response is characterized by the presence of significant off-resonant contribution from neighbouring modes, and the natural frequency is always taken to be the point at which there is maximum spacing of equal frequency increment points. At the resonant frequency the amplitude is the greatest and this property is used to identify the exact resonance frequency.

Instruments

For complex response measurments, two modules of the plugging units of AIM system (Advanced Instrumentation Modules) type 5.15B were used. A programmable filter oscillator type SSG212 supplied the electrical input to the loudspeaker through a RADFORD power amplifier. The output of the slow scan generator was a triangular wave form, the amplitude of which could be varied from 0 volts to 20 volts, and its sweep rate was manually controlled. The frequency range swept by the programmable filter oscillator was controlled by the voltage output of the generator.

Two reference signals of approximately one volt r.m.s. and having a phase difference of 90 were supplied by the oscillator.

The Receiver

A B&K one inch condenser microphone type 4145 was used as the sound receiver. It had a linear free field frequency response and omni-directional characteristics.

Phase Resolution

The resolved components of the output signal were measured using a pair of phase sensitive detectors type PSD 122A of AIM system. The output signal of the system under investigation (i.e. the room) was amplified by using a B&K Measuring Amplifier which was connected to the input sockets of the phase detectors. Two reference signals from the programmable filter oscillator were connected to the in-phase and quadrative phase-reference input sockets of the two phase detectors.

The resolved signals were amplified and presented at output sockets and also displayed on indicating meters. The output signals provided simultaneous readings of the real and imagined components of the transfer function of the system under investigation, thus allowing a Nyquist diagram to be displayed on the screen of an oscilloscope, and also be plotted by a x,y plotter.

The programmable filter oscillator, which was connected to the amplifier, was also connected to a frequency counter (to enable measurement of the frequency at any time) and an oscilloscope.

A Moseley x,y Recorder (Model 7035B), fed by the resolved components of the signal from the output sockets of the two phase detectors, was used to produce a Nyquist plot of the transfer function of the room response as a function of frequency. In a circuit which, when operated with a square wave form WEST HYDE pulse generator through a relay device, lowers the x,y plotter pen momentarily on the paper at equal intervals, thus making the plot provided information on the rate of change of the vector with frequency. In this fashion a first approximation was obtained on the locations of the natural frequencies of the system under investigation.

To be able to find the rate of change of frequency of the slow scanning Generator by using a programmable filter oscillator, frequency counter and stop watch; the time taken for a certain range of frequency to be covered was measured and the experiment was then repeated for an equal range of frequency, but it was noticed that this method of measuring the rate of change of frequency was inaccurate and not very reliable, as the rate of change of frequency varied between different ranges of frequencies, so another method was adopted. The output from the amplifier was connected to the level recorder, the paper speed of the level recorder was set at (10mm/sec) and the average period on the frequency counter was noted down.

The period of the required frequencies was obtained, the level recorder was run at the set speed and when the required period was displayed on the frequency counter, the pen of the level recorder started to draw a line on the graph paper of the level recorder.

The distance between the initial frequency and each frequency was then measured using

$$S = UT$$

where

S = distance U = speed of the level recorder T = time

(all in one unit)

Knowing S and U, the T (the time taken for a certain range of frequency by plotting the graph of speed against time and measuring the slope of the graph (Figure1.43) to be covered) was measured and it was noted that the rate of change of frequency was linear.

The results were corrected using computer program 'Least Square Method' (provided by Commodore).

The graph of frequency against time was plotted and the slope of it was measured (Figure 3.43).

The speed with which the pen of the x,y plotter was lowered down was adjusted so that each time the pen is lowered and consequently a trace is left by the pen, it represents on Hertz frequency interval.

The speed with which the pen of the x,y plotter was lowered down so that each frequency interval represented 1 Hz frequency range was done by adjusting the frequency on the function generator knowing the rate of change of frequency of the programmable filter oscillator. Therefore, each time the pen of the x,y plotter was lowered and consequently a trace was left by the pen on the paper, the interval between the start of each trace represented the 1Hz frequency.

By plotting the Nyquist diagram, it was expected that the exact resonance frequency could be identified, as the resonant frequency occurred where the amplitude was the greatest and also the curvature that was plotted as part of the circle by the x,y plotter at certain intervals covered the longest curvature as part of the circle. Therefore the length of the curvature increases as the resonant frequency is approached and at the resonant frequency it had the greatest curvature length and afterwards the length of the curvature starts to shorten. Figure 1.45 shows the set up for plotting the Nyquist diagram.

Mic. Position IX,OY,OZ Bare Wall

Frequency	Frequency Bandwidth using Kennedy & Pancu Method
76.23	1.53-1.59-2.4-1.73-1.96-1.58-1.03-1.01-1.03-1.4-1.34-2.1
85.59	2.9-2.9-3.83-3.19
92.08	1.67-2.07-2.25-2.25
101.31	3.15-1.77-
101.49	3.28-2.78
106.2	1.69-1.64-1.59-1.51
115.51	2.3-2.32-2.25-3.4-2.35
126.12	1.37-1.71-1.9
135.5	
145.47	1.53
147.65	2.25-2.0-2.28-2.26
156.16	1.38-1.37
158.5	
161.93	
165.04	2.02-1.48-2.43
173.43	1.47-1.57-1.81
176.03	2.11 1.81
185	

The Phase Angle Method

Introduction

This technique of measurement is similar to that used in the peak-amplitude method, i.e. the system under consideration is excited harmonically, but instead of measuring the amplitude of the response, its angle of phase lag behind the driving signal is measured.

Since a phase change occurs at resonant frequency, by making an analysis of a plot of phase angle with frequency, information on the natural frequency and damping constant of a mode of vibration can be obtained.

Determination of Natural Frequencies

The phase angle of the response is given by:

$$\mathrm{Tan}\ \alpha = \frac{W^2 - W_n^2}{2W_n \delta_n} \qquad 2.76$$

at resonance

$$W = W_n \quad \text{and}$$

$$\tan a = 0, \text{ i.e. } a = 0 \text{ or } \pi \text{ radians.}$$

Thus for a pure mode, the natural frequency may be calculated by the intersection of the phase angle curve with the line $\alpha = 0$, or $\alpha = \pi$.

For a linear system, the phase angle is independent of the amplitude of the driving signal and the response of the system.

There is no discrepancy between the natural frequency of the mode and the frequency at which resonance occurs, as there was in the case of the peak-amplitude method.

The presence of off-resonant vibration associated with heavy damping or closely spaced natural frequencies is bound to distort the location of resonance on the phase angle plot, as in the peak-amplitude plot.

Determination of Damping Constants

The rate of change of phase angle with respect to frequency is utilized to measure damping in a manner analogous to the sharpness of peaks in amplitude frequency response.

Consider the particular value of (w), one above and one below w for which:

$$2w_n\delta_n = \pm (w^2 - w_n^2)$$

Substitution of this condition in equation (3.16) shows that the phase angle at these values of (w) is:

$$\tan\alpha = \pm 1 \qquad 3.17$$

i.e. $\alpha = \frac{3\pi}{4}$ and $\frac{5\pi}{4}$ or $\frac{\pi}{4}$ and $\frac{7\pi}{4}$ radians.

Therefore the damping constant (δn) can be found by measuring the width of the frequency band between the points on the phase angle plot corresponding to the vaues of the phase angle given by equation (3.17).

The damping constant is represented by equation

$$\delta n = \frac{\Delta w}{2} = \frac{w_2 - w_1}{2} \qquad 3.18$$

as derived in discussing the peak-amplitude technique.

An alternative method, reported by Pendered and Bishop (35) and claimed to be more exact, is derived from equation

$$\alpha = \tan^{-1} \frac{w^2 - w_n^2}{2w_n\delta_n} \qquad 3.19$$

$$\text{let } \alpha = \tan^{-1} u \text{ where } U = \frac{w^2 - w_n^2}{2w_n\delta_n} \qquad 3.20$$

Therefore $\tan \alpha = u$

The slope of the angle curve is obtained by differentiating both sides of equation (3.20) with respect to (w).

Hence

$$\mathrm{Sec}^2\alpha \cdot \frac{d\alpha}{dw} = \frac{du}{dw}$$

$$\frac{d\alpha}{dw} = \frac{1}{1+\tan^2\alpha} \cdot \frac{du}{dw} = \frac{1}{1+u^2} \cdot \frac{du}{dw}$$

$$\frac{d\alpha}{dw} = \frac{1}{1+\left(\frac{w^2 - w_n^2}{2w_n\delta_n}\right)^2} \cdot \frac{2w}{2w_n\delta_n}$$

$$= \frac{4ww_n\delta_n}{4w_n^2\delta_n^2 + (w^2-w_n^2)^2}$$

at resonance:

$$w = w_n$$

and

$$\frac{d\alpha}{dw} = \frac{1}{\delta_n} \qquad 3.21$$

Thus the damping constant is given by

$$\delta_n = \frac{1}{\text{slope at } w_n} \qquad 3.22$$

If the assumption that, between points satisfying the condition $\tan \alpha = 1$, the phase angle curve is approximately linear, then the slope is given by:

$$\text{Slope } \underline{N} \simeq \frac{\pi}{2} \div \Delta w = \frac{\pi}{2\Delta w}$$

Substituting in equation (3.22)

$$\delta_n = \frac{2\Delta w}{\pi} \qquad 3.23$$

In this way the second method gives a value of (n) higher than that given by the first method by a factor $^{\pi}/_{4}$. Bishop and Pendered (35) report that comparison of practical results confirm a factor of this order and attribute the error to the assumption of linearity, discussed above, and to the approximations inherent in the first method.

The validity of both methods is affected by the presence of off-resonance vibrations.

Determination of Modal Shape

Considering the variation in the location of the receiver along the x-direction, the component phase angle, from the expanded form of equation

$$\tan \alpha_x = \tanh \left(\frac{\delta_x x}{c}\right) \tan \left(\frac{w_x x}{c}\right) \qquad 3.24$$

where

$$\alpha = \alpha_x + \alpha_y + \alpha_z$$

is the total phase angle of lag behind the driving signal.

The trigonometric function in the right -hand side of equation (3.24) assumes alternating values of 0 and depending on the value of (x). Accordingly, the component phase angle (α_x) is equal to ($\frac{n\pi}{2}$) radians; n=(0,1,2,3.....)

Even values of (n) correspond to an in-phase or anti-phase response, while an odd value indicates a quadrative response. Therefore, from the plot of response phase with distance, the nodal and anti-nodal planes can be identified. However, the method is subject to the same limitations discussed under the Peak-Amplitude method for identification of modal shapes.

For the resonant testing experiment to be carried out, it is essential to find the point where the maximum and the minimum of sound pressure level occurs along the three principal axes of the room. In the peak-amplitude method the source and receive are both located at anti-nodes for the mode under investigation, so that the maximum response is attained. Theoretically, it has been proved that when the microphone and the source are placed at the corner of the room all the modes are excited and that the maximum response is obtained. Also when the microphone is placed at the middle of the room, the odd mode related to the axis of the microphone will be minimum.

Since the physical properties of the two parallel walls of the room vary including the admittance value, it is expected that the variation of sound pressure level along the axis of the two parallel walls will be asymmetrical, and as a result the point of maxima and minima sound pressure level on the axis could well become different from the point obtained theoretically.

To study the sound pressure level inside the room and also to be able to investigate its possible variation across the two opposite walls of the room, curtain railing track was placed on the middle of the ceiling along the y-axis, to be able to move the microphone, which is placed on the curtain track along the y-axis. The string of the curtain rail was then connected to a D.C. motor and the speed of this was adjusted to be reasonably slow.

Pure tone sound was generated at the theoretical and experimental resonant frequency through the hetrodyne 2010, which was connected to the level recorder to measure the sound pressure level along the track as the microphone moved very steadily along the curtain rail. The set up for the purpose of this experiment is shown in Figure 3.49.

The background sound and the vibration due to the movement of the microphone along the curtain rail was also plotted by the help of a level recorder. The experiment was carried out with some of the surfaces of the room covered with fibre glass.

The theoretical resonant frequencies were obtained from the program 'ROOMODES' and the experimental resonant frequencies were obtained for each condition of the treated surfaces of the room. As can be seen from Figures (3.50-3.56), when some surfaces of the room were covered with fibre glass, the sound pressure level varied considerably across the two opposite walls on the y-axis, but when all the surfaces of the room were covered by fibre glass considering one mode along the y-axis, the sound pressure levels were reasonably equal at the surfaces of the two opposite walls (Figure 3.60).

i) The anomalies in the sound pressure level perpendicular to the surfaces that are covered with fibre glass were further investigated by altering the position of the loudspeaker from one corner to another corner and studying the degree of influence of source position on the coupling of the mode.

ii) The directionality of the loudspeaker is to be studied.

Further investigation is required to find the position where the maximum sound pressure level occurs, as the impedances of the surfaces of the room vary, and also the existence of the double door has rendered the room asymmetric nd the maximum sound pressure level does not occur as expected at the middle point along each axis of the room: even the axis where the sound pressure is maximum could be an axis which is not necessarily parallel to the original axis of the room.

To study this, the sound pressure level at each point along one axis and perpendicular to the axis at a certain interval of distance and the variation of the sound pressure level across the mentioned axis specially in relation to each wall, was considered to be a very useful approach.

Impedance of the Walls

The surfaces of the small room have varying impedances and this affects the sound energy distribution inside the room, and also the boundary conditions when the equation of the sound wave is considered.

Standing wave ratio and the distance between the surface of the sample and the sound pressure minima (the two parameters required for the determination of the acoustical normal impedance) are obtained by using the B&K standing wave apparatus type 4002.

The complete apparatus is comprised of a measuring tube, with one end of it attached to the loudspeaker and the other end attached to the properly sealed sample which is to be tested. The loudspeaker is connected to the oscillator (H.E.T. 2010) which generates the sound at the required frequencies.

The sound field is explored by means of a probe microphone that moves on a scaled track whereby the exact distance between the probe and the test sample can be read.

The microphone output is fed into the spectrometer B&K 2112, which is used as a selective amplifier and a direct indicator of the value of the absorption coefficient on its meter scale.

Since the behaviour of the room at resonance frequency is important, the impedance of the surfaces is measured at the modal frequencies.

The standing wave tube is directly connected to the surface DCGH and the area of contact between the tube and the surface of the wall is then sealed properly, since the values of impedance vary for various parts of each wall which lead to inaccurate results.

The standing wave tube is normally used when a small piece of the sample is available. Readings for the normal absorption coefficient and the distance of the first and second sound pressure minima from the face of the sample are taken at approximate logarithmic frequency intervals at modal frequency; for large samples, which allow panel vibration, alternative techniques should be used.

The use of the standing wave tube is limited to sounds of frequencies higher than a minimum frequency limit, below which no pressure minimum is formed. The range of the frequency on the spectrometer scale is Frequency $x(2^{\frac{1}{2}})^{1/3}$ = Frequency x 1.122.

The results obtained by this method are not reliable, as the surface area of each wall is not homogeneous and the measured impedance in only related to a small surface of the wall. Therefore, to cover the whole surface area, the experiment should be repeated many times and sometimes it becomes impossible to carry out the experiments on parts of the surface area of each wall.

A computer program 'IMPEDANCE' has been written to calculate the special acoustic impedance of the surface under consideration.

End Correction = 1.55cm

Frequency	% Absorption Coefficent	Distance+End Correction cm	Special Acoustic Impedance ( )	
			Real	Imaginary
100.55	19.3	85.07	0.22	.39
104.26	19.0	82.90	1.43	-2.18
113.26	17.8	75.30	0.39	1.06
123.78	19.0	69.75	0.75	-1.5
128.56	18.9	66.75	0.19	-0.11
133.46	18.7	62.30	0.56	1.34
143.28	19.5	61.00	0.20	0.11
144.85	20.00	59.73	3.07	-2.35
150.44	20.20	56.00	4.64	-1.18
153.20	20.30	55.10	1.43	2.07
156.38	20.30	55.75	0.27	0.54
159.21	20.00	54.45	3.24	2.30
161.83	7.00	52.15	0.47	2.34
164.57	7.30	53.50	0.66	-2.76
169.36	6.50	50.99	0.32	-1.90
170.03	6.90	51.30	0.08	0.47
172.63	6.40	50.73	0.07	-0.20
173.53	6.80	49.94	8.21	7.27
183.03	6.10	48.44	0.18	1.39
185.31	5.80	47.34	0.07	-0.52
185.92	6.50	47.65	0.29	1.83
194.82	6.90	44.75	0.09	0.57
200.19	5.30	42.85	0.05	-0.06
201.11	5.30	42.65	0.05	-0.04
205.59	5.00	41.10	0.06	-0.42

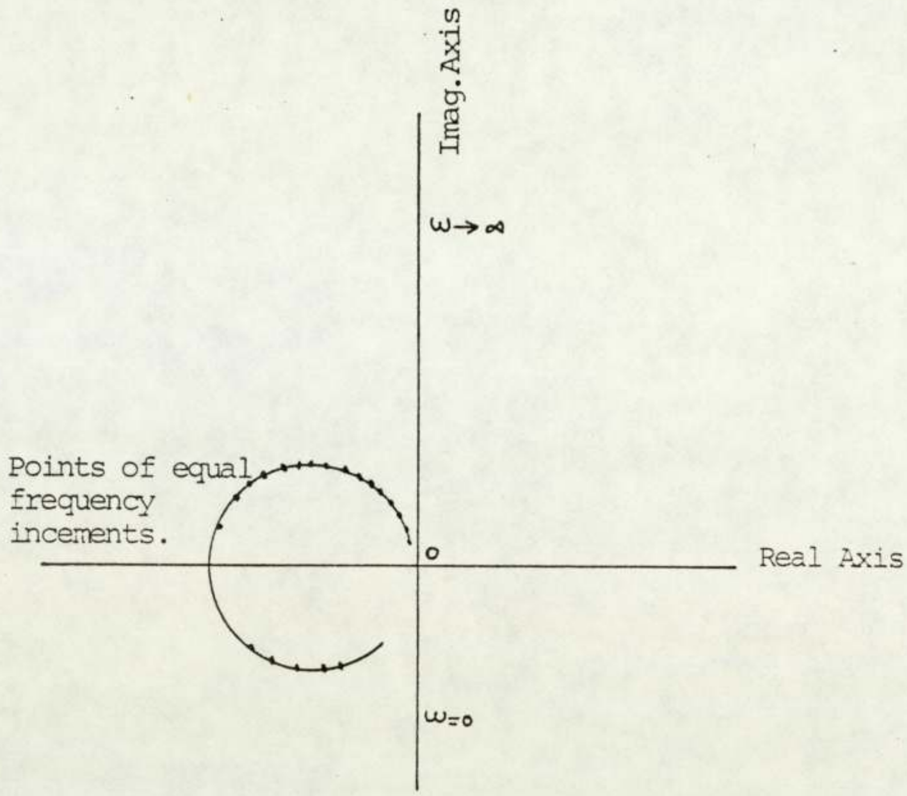


FIG. 3.40 COMPLEX FREQUENCY RESPONSE OF A PURE MODE.

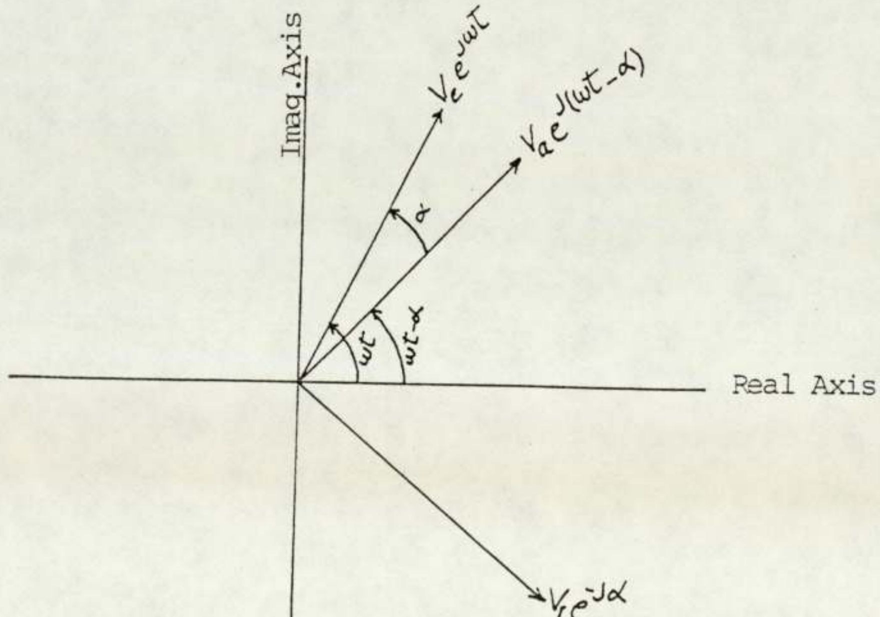
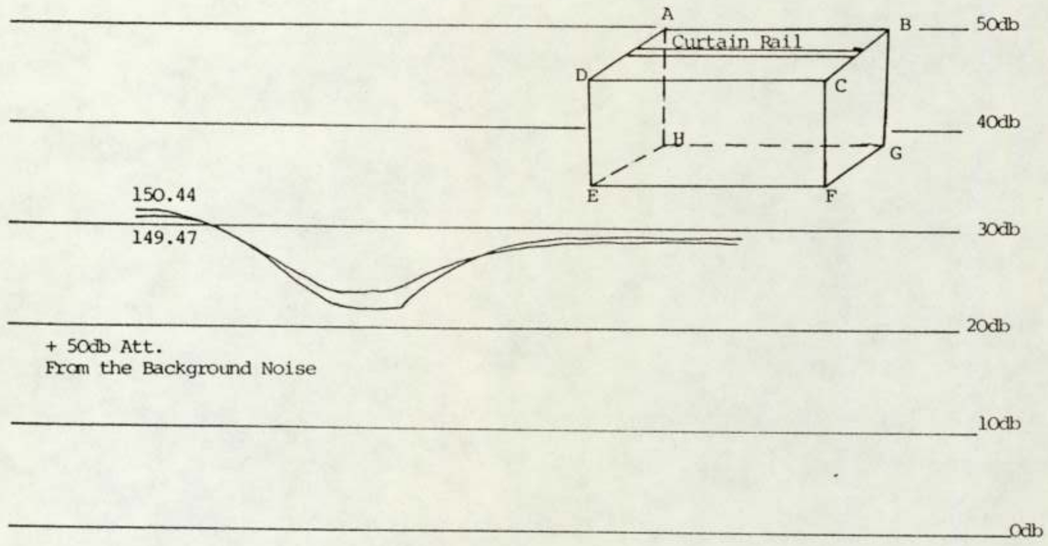


FIG. 3.41 SIGN CONVENTION OF VECTOR REPRESENTATION.



SURFACES COVERED WITH FIBRE GLASS

ABCD - EFGH
CDEF - ABGH

FIG. 3.54

Loud Speaker Position OX, OY, OZ.

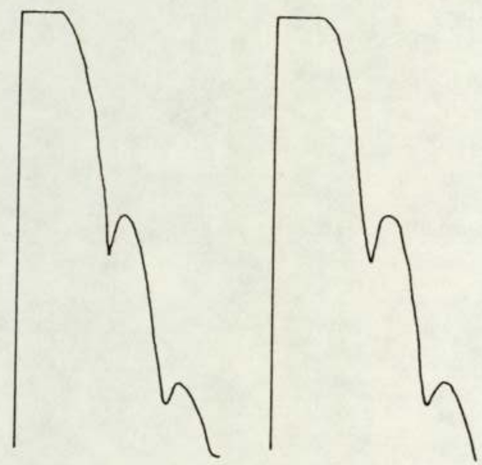


FIG. 3.33 The Decay of Sound at Resonant Frequency 200.19 HZ.

Loud Speaker Position OX, OY, OZ.

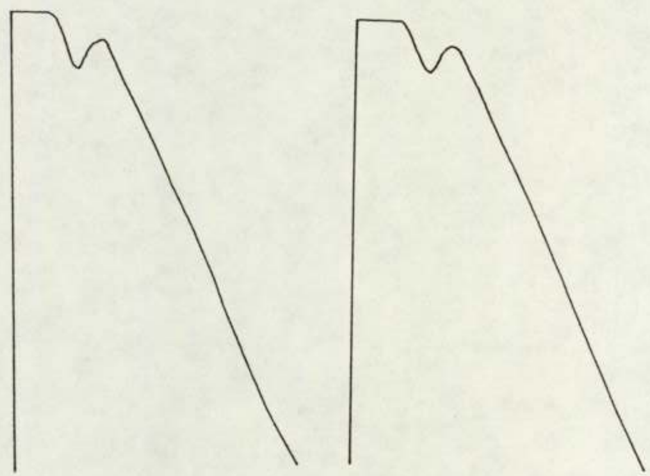


FIG. 3.32 The Decay of Sound at the Resonant Frequency 200.19 HZ Microphone at Different Position.

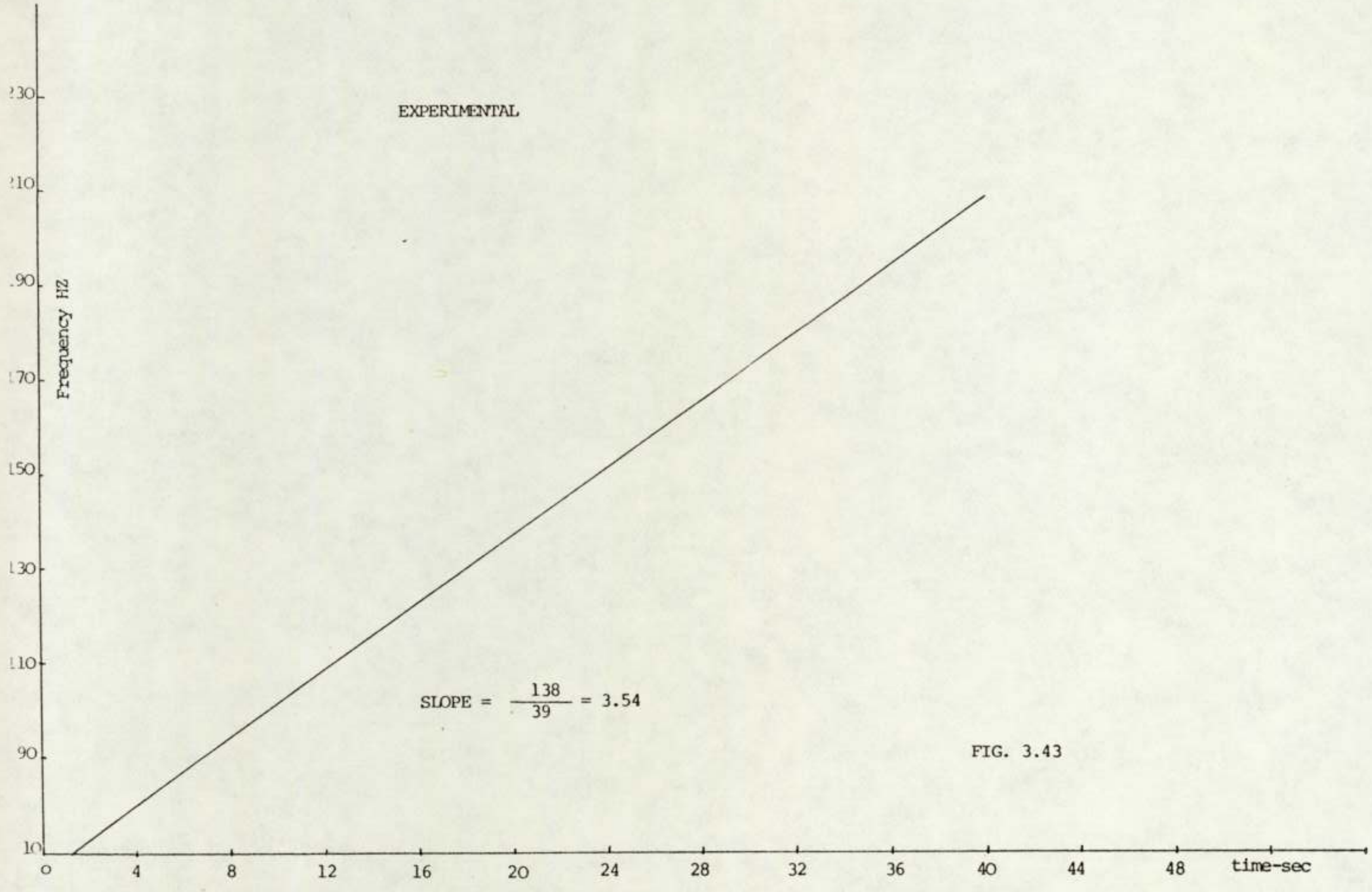
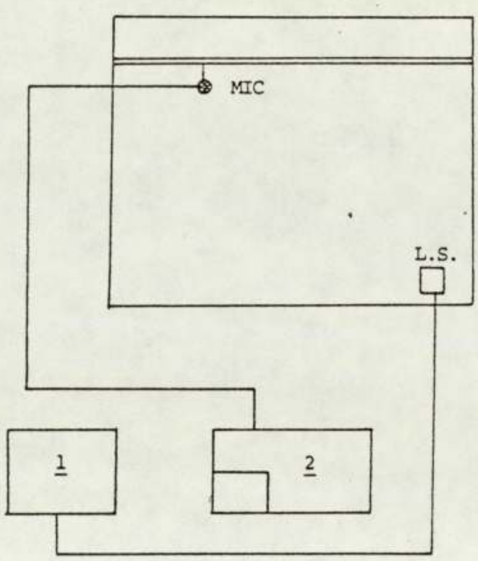

EXPERIMENTAL
Frequency HZ
230
210
190
170
150
130
110
90
10
0
4
8
12
16
20
24
28
32
36
40
44
48
time-sec
SLOPE = 138/39 = 3.54

FIG. 3.43

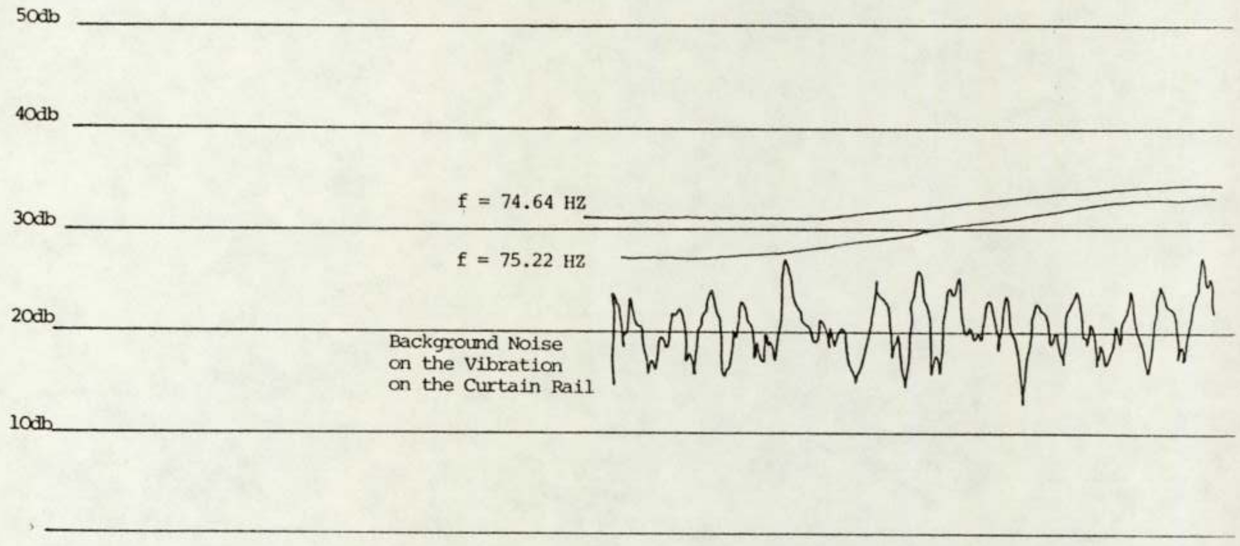


1 - Hetrodyne 2010

2 - Level Recorder

FIG. 3.49

Instrumentation set up for R.T. Measurement and Spatial Variation of the Sound Pressure Level.



All the Surfaces of the Room
Covered with Fibre Glass

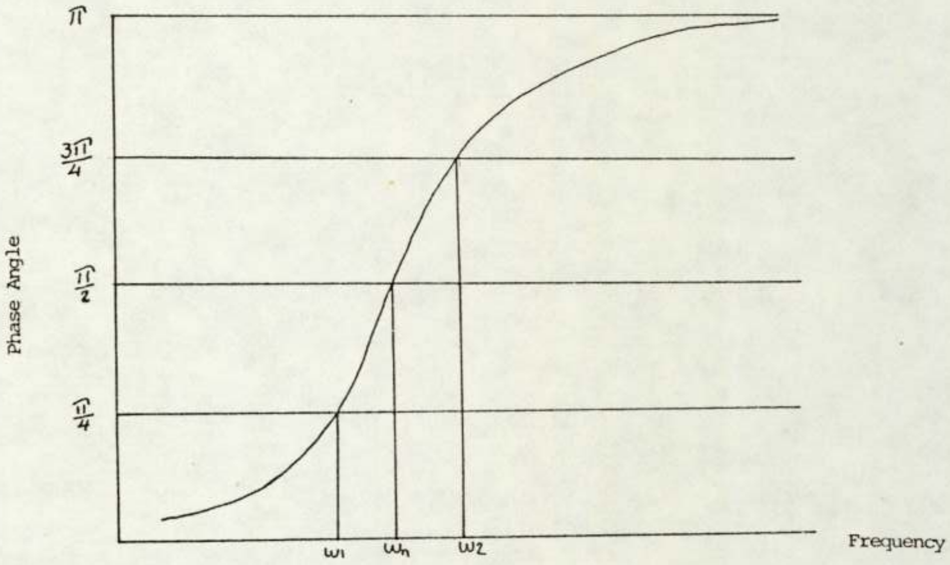
L.S. Pos. OX, IY, OZ.

FIG. 3.51

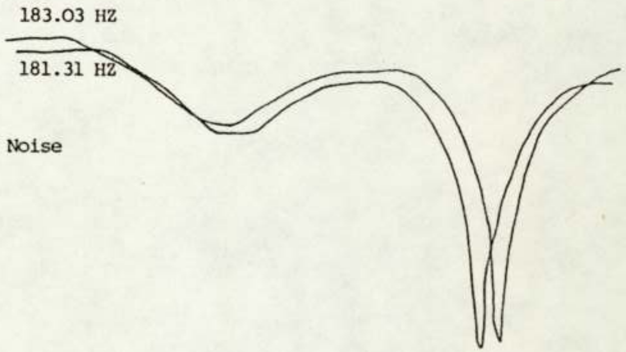
Fig. 3.34a

DETERMINATION OF NATURAL FREQUENCY AND DAMPING CONSTANT

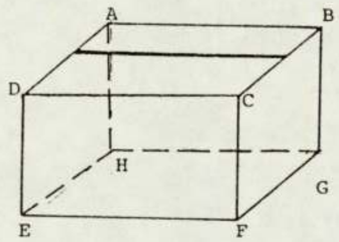
BY PHASE-ANGLE METHOD



L.S. Pos. OX,IY, OZ
MIC.



Att + 40
Background Noise



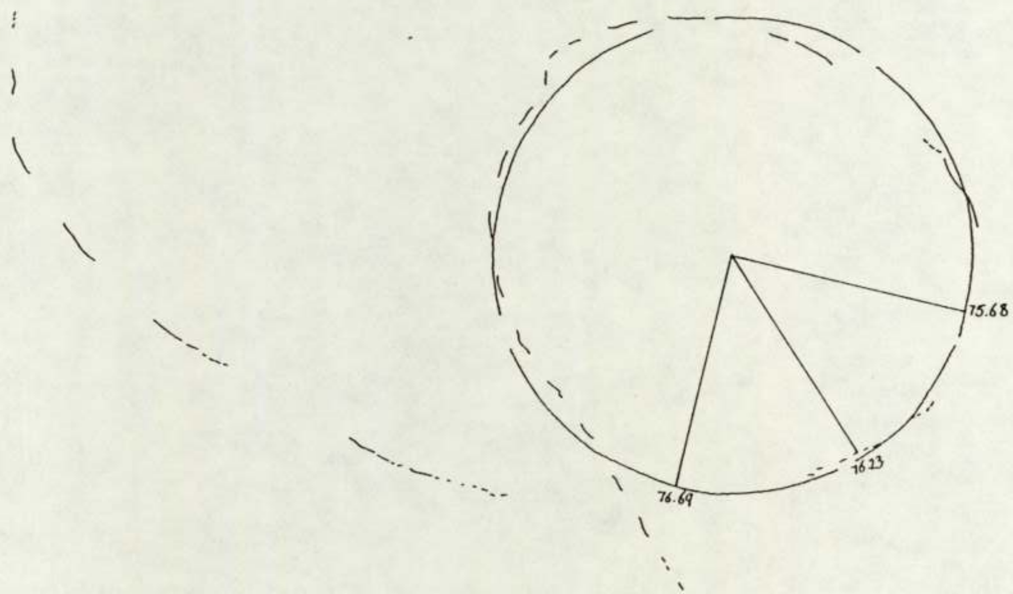
FOUR SURFACES COVERED WITH
FIBRE GLASS

ABCD - EFGH
CDEF - ABGH

FIG. 3.56

NYQUIST DIAGRAM MICROPHONE POSITION 2

Δf = 76.69-75.68=1.01
76.23



CHAPTER 4

Synthesis

Considering a simple linear mechanical system (i.e. one degree of freedom system consisting of a mass, damper and spring, and analyzing the equation of the motion of the system under forced condition, the whole system with the mass, damping and the stiffness of the spring) is analogous to the behaviour of a room which has been excited by a sound mode.

The behaviour of the sound at low frequency inside the room is dependent on various factors. The most important of them, those which are common when compared with the mechanical system, are the damping and the degree of coupling of the whole system of the room, even though the room is considered to be a system with n degree of freedom.

To simplify the mathematics involved in deriving the formulae relating damping and reverberation time and the relationship between various factors, a mechanical approach has been adopted rather than the complicated sound wave formulae.

4.00 Viscous Damping

During vibration, energy is dissipated in one form or another and steady amplitude cannot be maintained without its continuous replacement. The actual description of the damping force associated with the dissipation of energy is difficult. However, ideal damping models can be conceived which will often permit a satisfactory approximation. Considering the viscous damping force to be proportional to the first power of the velocity leads to the simplest mathematical analysis.

Figure 4: Free Vibration with Viscous Damping

Viscous damping can be expressed by the equation:

$$F = - c\dot{x} \qquad 4.1$$

Where c is a constant of proportionality and

x the velocity

The equation of motion when such a system undergoes free vibration:

$$m\ddot{x} = -c\dot{x} - kx \qquad 4.2$$

Rearranging

$$m\ddot{x} + c\dot{x} + kx = 0 \qquad 4.3$$

Equation 4.3 being a homogeneous second-order differential equation, can be solved by assuming a solution of the form

$$x = e^{\delta t} \qquad 4.4$$

where s is a constant to be determined.

Substituting Equation 4.4 into 4.3,

$$\left(s^2 + \frac{c}{m} s + \frac{k}{m}\right) e^{\delta t} \qquad 4.5$$

which is satisfied for all values of t when

$$s^2 + \frac{c}{m} s + \frac{k}{m} = 0 \qquad 4.6$$

Equation 4.6, which is known as the characteristic equation, has two roots

$$s_{1,2} = -\frac{c}{2m} \pm \left(\left(\frac{c}{2m}\right)^2 - \frac{k}{m}\right)^{1/2} \qquad 4.7$$

and hence the general solution for the damped free vibration as described by Equation 4.3

$$X = Ae^{s_1 t} + Be^{s_2 t} \qquad 4.8$$

where A and B are arbitrary constants depending on how the motion is started.

4.2.0 Forced Vibration with Harmonic Excitation

From the free body sketch in Figure 4.2.1 the equation of motion of the system is (37)

Figure 4.2.1

$$m\ddot{x} = \sum (\text{forces})_x \qquad 4.2.1$$

$$m \frac{d^2}{dt^2} (x+d_{st}) = -k(x + \delta_{st})-c\frac{d}{dt}(x + \delta_{st}) + mg + F\sin(wt) \qquad 4.2.2$$

$$m\ddot{x} + c\dot{x} + kx = F\sin(wt) \qquad 4.2.3$$

The gravitational force mg is equal to the static spring force k .
Using d'Alembert's principle, Equation 4.2.1 can be expressed as

$$(\text{Forces})_x - m\ddot{x} = 0 \qquad 4.2.4$$

The quantity $-m\ddot{x}$ is called the inertia force.

The equation of motion for the system in Figure 4.2.1 is a second order linear ordinary differential equation with constant coefficients, Equation 4.2.3. The general solution $x(t)$ is the sum of the complementary function $x_c(t)$ and the particular integral $x_p(t)$.

$$x = x_c + x_p \qquad 4.2.5$$

4.3 Sound Decay

The rate at which sound decays in a room once the source has stopped radiating direct sound energy provides a useful numerical description of the acoustic properties of the room. Once the direct sound is stopped, the whole of the sound is due to reverberant field, which will gradually die out due to absorption at the walls.

The decay rate is found by considering the rate of energy absorption. The rate at which energy arrives at a surface, multiplied by the absorption coefficient of the surface, is the rate at which energy is being removed at the surface.

The rate at which energy arrives is the intensity multiplied by the area. So the rate of energy removal is αIS, or averaged over the whole area, $S\alpha I$, where

S = the whole surface area of the room

α = the averaged absorption coefficient of the room

I = the sound intensity of the source

The total energy in the room is the energy density multiplied by the volume, so the rate of energy removal must also be the product of volume and the time rate of change of energy density, a decrease being indicated by a negative sign for the rate of change. Equating these two expressions for the rate of energy decrease gives (39):

$$-v\frac{d\varepsilon}{dt} = s\alpha I \qquad 4.2.32$$

But $$\varepsilon = \frac{4I}{c}$$

so $$\frac{d\varepsilon}{dt} = \frac{4}{c}\frac{dI}{dt}$$

and

$$-\frac{4}{c}\frac{dI}{dt} = s\alpha I = \alpha I$$

$$\frac{dI}{dt} = -\frac{ac}{4v}I$$

$$\frac{dI}{I} = -\frac{ac}{4v}dt$$

$$\ln I = \frac{-ac}{4v}t + \text{constant}$$

4.2.33

If the initial intensity at time t=0 is I , then the constant of integration is ln I_o and the Equation (4.3.2) can be written as:

$$\ln I - \ln I_o = -(\frac{ac}{4v})t$$

$$\ln(\frac{I}{I_o}) = -(ac/4v)t \qquad 4.2.34$$

$$I/I_o = \exp -(ac/4v)t \qquad 4.2.35$$

The rate at which the sound is decayed in a room depends on both the total absorption and on the volume of the room. The rate of decay of sound intensity is an important property of a room.

Reverberation time is defined as the time taken for the intensity or the sound pressure level to fall to 60dB of its initial value. A reduction of 60dB means the sound has fallen to one millionth of its original value, or from a fairly loud sound to about the threshold of audibility.

Considering Equation 4.2.34:

$$-60 = 10 \log(I/I_o) \qquad 4.2.36$$

$$-60 = 20 \log(\frac{P}{P_o}) \qquad 4.2.37$$

4.3.0 Relationship Between Half-Power Bandwidth and Modal Reverberation Time

4.3.1 4.3.2

Considering the system shown in Figure 4.3.1 under forced conditions, the equation for such a system is :

$$M\ddot{x} + B\dot{x} + Kx = Fe^{-jpt} \qquad 4.3.1$$

The equation of motion of the system under free condition is:

$$\frac{dx}{dt} = -jwae^{-jwt} \qquad \frac{d^2x}{dt^2} = -w^2Ae^{-jwt} \qquad 4.3.2$$

It must be considered that as the system moves faster, the damper becomes more resistant.

Solving Equation 4.3.2, let

Substituting in Equation 4.3.2

$$Ae^{-jwt}(-Mw^2-Bjw+k)=0 \qquad 4.3.3$$

The term Ae cannot be equal to zero, therefore the 2nd term of the Equation 4.3.3 must be equal to zero, i.e.

$$-Mw^2-Bjw+k=0$$

$$Mw^2+Bjw-k=0 \qquad 4.3.4$$

Equation 4.3.4 has two roots

i.e.
$$W = \frac{-jB\pm(-B^2+4KM)^{\frac{1}{2}}}{2M}$$

$$W = \left(\frac{K}{M} - \frac{B^2}{4M^2}\right)^{\frac{1}{2}} - J\frac{B}{2M}$$

4.3.5

For small B
$$W \quad \left(\frac{K}{M}\right)^{\frac{1}{2}} - \frac{JB}{2M} \qquad 4.3.6$$

and substituting for w in the expression x=e

$$x = e^{-jwt} = e^{-j\left(\frac{K}{M}\right)^{\frac{1}{2}}t} . e^{-\left(\frac{B}{2M}\right)t} \qquad 4.3.7$$

let $\delta = \frac{B}{2M}$

$$x = e^{-j\left(\frac{K}{M}\right)^{\frac{1}{2}} t} \quad e^{-\left(\frac{B}{2M}\right) t}$$

$$x = e^{-j\left(\frac{K}{M}\right)^{\frac{1}{2}} t} e^{-\delta t}$$

4.3.8

Referring to the definition of reverberation time in Chapter 4.2 (Applied Acoustics) and also to the equation 4.2.37, since the pressure is proportional to the amplitude and since the time that has elapsed is considered so that the sound pressure level reaches $\frac{1}{60}$-th of its original value,

$$20 \log \frac{P}{P_o} = -60 \qquad 4.2.37$$

$$20 \log \frac{x_{(t)}}{x_{(t_o)}} = -60 \qquad 4.3.9$$

where initial and final time is t and t respectively,

$$\frac{x_{(t)}}{x_{(t_o)}} = 10^{-3}$$

Now

$$\frac{x_{(t)}}{x_{(t_o)}} = \frac{e^{-\delta t}}{e^{-\delta t_o}} = e^{-\delta(t-t_o)}$$

where

$$t - t_o = RT$$

$$e^{-\delta RT} = 10^{-3}$$

$$e^{\delta RT} = 10^{3}$$

$$\log_e 10^3 = \delta RT$$

$$\frac{\log 10^3}{\log_e} = \delta RT$$

$$\frac{3}{\log_e} = \delta RT$$

Therefore $$Rt = \frac{3}{\log_e} \times \frac{1}{\delta}$$

4.3.9a

$$Rt = 6.92 \times \frac{1}{\delta} \qquad 4.3.10$$

Considering the case of the forced vibration system as shown in Figure 4.3.1. Consider the equation of motion of the system under external force, as shown in Figure 4.3.1. The equations of the motion of the system are as follows:

$$M\ddot{x} + B\dot{x} + Kx = Fe^{-jpt}$$

$$\dot{x} = Ae^{-jpt}$$

$$\dot{x} = -AjPe^{-jpt}$$

$$\ddot{x} = -P^2Ae^{-jpt}$$

$$-P^2MAe^{-jpt} - jPBAe^{-jpt} + KAe^{-jpt} = Fe^{-jpt}$$

$$\left(-P^2M - jPB + K\right) A = F$$

$$A = \frac{F}{\left(K-P^2M\right) - jPB}$$

$$P_o = \left(\frac{K}{M}\right)^{\frac{1}{2}}$$

$$A = \frac{F/M}{\left(\frac{K}{M} - P^2\right) - jP\frac{B}{M}} \qquad A = \frac{F/M}{P_o^{\,2} - P^2 - jP\frac{B}{M}} \qquad 4.3.13$$

Amax occurs when $P_o = P$ and $P_o = \left(\frac{K}{M}\right)^{\frac{1}{2}}$

$$Amax = \frac{F/M}{-jP(B/M)} = \frac{F}{-jPB} = \frac{F}{-j(K/M)^{\frac{1}{2}}B}$$

or

$$Amax = \frac{JF}{BP_o}$$

$$Amax = \frac{JF}{B}\left(M/K\right)^{\frac{1}{2}} \text{ or } \frac{JF}{BP_o}$$

Half power points occur when

or when $|x| = x_{max} \,|\, 2^{\frac{1}{2}}$

$$\left| \frac{JF}{BP_o 2^{\frac{1}{2}}} \right| = \left| \frac{F/M}{(P_o^2 - P^2 - jP^B/_M)} \right| \qquad 4.3.15$$

$$\frac{JF}{BP_o 2^{\frac{1}{2}}} = \left| \frac{F}{BP_o 2^{\frac{1}{2}}} \right|^2 = \left| \frac{F}{BP_o 2^{\frac{1}{2}}} \right| \qquad 4.3.16$$

$$\frac{F/_M}{\left(P_o^2 - P^2 - jP\frac{B}{M}\right)} = \frac{\frac{F}{M}\left(P_o^2 - P^2 + jP\frac{B}{M}\right)}{P_o^2 - P^2\,{}^2 + \left(P\frac{B}{M}\right)^2}$$

$$= \frac{F/_M\,(P_o^2 - P^2)}{(P_o^2 - P^2)^2 + \left(P\frac{B}{M}\right)^2} + j\,\frac{F/_M\;P\frac{B}{M}}{\left(P_o^2 - P^2\right)^2 + \left(P\frac{B}{M}\right)^2}$$

$$\left| \frac{\left(\frac{F}{M}\right)^2 \left[P_o^2 - P^2\,{}^2 + \left(P\frac{B}{M}\right)^2 \right]}{\left[\left(P_o^2 - P^2\right)^2 + \left(P\frac{B}{M}\right)^2\right]^2} = \right|^{\frac{1}{2}}$$

$$\left| \frac{\left(F/_M\right)^2}{\left(P_o^2 - P^2\right)^2 + \left(P\frac{B}{M}\right)^2} \right|^{\frac{1}{2}} \qquad 4.3.17$$

Equating Equations 4.3.16 and 4.3.17

$$\frac{F}{BP_o 2^{\frac{1}{2}}} = \left(\frac{\left(F/_M\right)^2}{\left(P_o^2 - P^2\right)^2 + \left(P\frac{B}{M}\right)^2} \right)^{\frac{1}{2}} \qquad 4.3.18$$

$$\left(P_o^2 - P^2\right)^2 + P^2\left(\frac{B}{M}\right)^2 = 2\left(\frac{B}{M}\right)^2 P_o$$

Let $k' = (B/M)$, $w = P^2$ and $w_o = P_o^2$

$$\left(P_o^2 - P^2\right)^2 + P^2\left(\frac{B}{M}\right)^2 = 2\left(\frac{B}{M}\right)^2 P_o^2 \qquad 4.3.19$$

$$\left(w_o - w\right)^2 + k'w - 2k'w_o = 0$$

$$w_o^2 + w^2 - 2w_o w + k'w - 2k'w_o = 0$$

$$w^2 + w(k' - 2w_o) + (w_o^2 - 2k'w_o) = 0 \qquad 4.3.20$$

$$w = \frac{-(k'-2w_o)}{2} \pm \left(\frac{(k'-2w_o)^2 - 4(w_o^2 - 2k'w_o)}{4}\right)^{1/2}$$

$$w = w_o - \frac{k'}{2} \pm \left(\frac{k'^2 + 4w_o^2 - 4k'w_o^2 - 4w_o^2 + 8k'w_o}{4}\right)^{1/2}$$

$$w = w_o - \frac{k'}{2} \pm \left(k'w_o + \frac{k'^2}{4}\right)^{1/2}$$

For small k'

$$w = w_o - \frac{k'}{2} \pm k'w_o \qquad 4.3.21$$

Substituting the values of w, k' and w_o

$$P^2 = P_o^2 \pm \left(\left(\frac{B}{M}\right)^2 P_o^2\right)^{1/2}$$

$$P^2 = P_o^2 \pm \left(\frac{B}{M}\right) P_o \qquad 4.3.22$$

Knowing that $\frac{B}{2M} = \delta \qquad \frac{B}{M} = 2\delta$

Substituting in Equation 4.3.22

$$P^2 = P_o^2 \pm (2\delta) P_o$$

$$P^2 = P_o^2 \; 1 \pm \frac{2\delta}{P_o}$$

$$P = P_o\left(1 \pm \frac{2\delta}{P_o}\right)^{1/2}$$

4.3.23

Considering the binomial expression for the term $\left(1 \pm \frac{2\delta}{P_o}\right)^{\frac{1}{2}}$

$$\left(1+\frac{2\delta}{P_o}\right)^{\frac{1}{2}} = \left(1 + \tfrac{1}{2}\left(\frac{2\delta}{P_o}\right) + (\tfrac{1}{2})(-\tfrac{1}{2})\left(\frac{2\delta}{P_o}\right)^2 + \ldots\ldots\right.$$

For small value of δ

$$P = P_o\left(1 \pm \frac{\delta}{P_o}\right) \qquad 4.3.25$$

$$P_{HP1} = P_o\left(1 + \frac{\delta}{P_o}\right) \qquad P_{HPZ} = P_o\left(1 - \frac{\delta}{P_o}\right)$$

$$\Delta P = 2\delta \qquad 4.3.26$$

or

$$2\pi\Delta f = 2\delta$$

$$\pi\Delta f = \delta$$

From Equations 4.3.9a and 4.3.10 when substituting for the following expressions are obtained:

$$R.T. = \frac{3}{\log_e} \times \frac{1}{\delta} \qquad 4.3.9a$$

$$R.T. = 6.92 \times \frac{1}{\delta} \qquad 4.3.10$$

$$\delta = \frac{3}{(\log_e)R.T.}$$

$$\Delta P_o = \frac{13.82}{R.T} \qquad 4.3.27$$

$$\Delta P_o = \frac{6}{(\log_e R.T} \qquad 4.3.28$$

$$2\pi\Delta f = \frac{13.82}{R.T} \qquad \Delta f = \frac{13.82}{2\pi} \times \frac{1}{RT}$$

$$\Delta f = \frac{2.2}{R.T}$$

4.3.30

4.4 The Steady-State Response of the Room

The sound field in the room is represented by the wave equation; this, expressed in terms of the velocity potential, is:

$$\nabla^2 \phi = -\frac{1}{c^2} \frac{\partial^2 \phi}{\partial t^2} = 0 \qquad 4.4.1$$

Assuming the harmonic law for the pressure variation,

$$\nabla^2 P_o + k^2 P_o = 0 \qquad K = \frac{W}{C} \qquad 4.4.2$$

The wave equation yields non-zero solutions fulfilling the boundary condition, only for the particular discrete values of k, called characteristic values k_n.

Each characteristic value (k_n) is associated with a solution ($\psi_{n(x,y,z)}$) which is the characteristic function of the room, in such a way that

$$P = A\psi_n \qquad 4.4.3$$

As the characteristic function forms a complete set of orthogonal functions (23,20), the steady state distribution of sound can be expressed as a series (ψ_n), i.e.

$$P = P_n e^{-jwt} = \sum_n A_n \psi_n e^{-jwt} \qquad 4.4.4$$

The Source Function

Assuming that the effect of the source and the room boundaries could be replaced by a distribution of simple source such that the element of volume (dv') at (x ,y ,z)cm^3/sec, where ( ) is the density function, and that any source of sound may be considered as an assemblage of simple source, then:

$$Q_{(t)} = \iiint_{room} q \, dV' \qquad 4.4.5$$

$Q_{(t)}$ being the equivalent volume velocity of the source.

If the source is simple harmonic, then:

$$q_{(t)} = qe^{-jwt} \text{ and } Q_{(t)} = Qe^{-jwt} \qquad 4.4.6$$

Assuming the orthogonality property (23), the source distribution can, likewise, be expanded in a series of ψ_n , i.e.

$$q_{(x,y,z)} e^{-jwt} = \sum_n c_n \psi_n e^{-jwt} \qquad 4.4.7$$

where

$$C_n = \frac{1}{V\Lambda_n} \iiint_{room} q\psi_n dV \qquad 4.4.8$$

and $\iiint \psi_n \psi_m dv = \begin{cases} \int & n<>m \\ V\Lambda_n & n=m \end{cases}$

4.4.9

Λ_n being the average of the integral over the room volume.

If all the characteristic values and characteristic functions, which in general are functions of the frequency, are known, any acoustical property of the room can be evaluated, one of these acoustical properties being steady state response to arbitrary sound sources. Suppose the sound sources are distributed continuously over the room according to a density function $q_{(x,y,z)}$, where $q_{(x,y,z)}dv$ is the volume velocity of a volume element dv at x,y,z. $q_{(x,y,z)}$ may be a complex function taking acount of possible phase differences between the various infinitesimal sound sources.

The wave equation for the velocity potential in the presence of a source distribution of density q is:

$$\nabla^2 \phi - \frac{1}{c_2} \frac{d^2\phi}{d_t^{\,2}} = -q_{(x,y,z)} e^{-jwt} \qquad 4.4.10$$

and

$$P = \rho \frac{d\phi}{dt} \quad P = jw\rho\phi$$

Equation 4.4.10 expressed in terms of the pressure is:-

$$\nabla^2 P - \frac{1}{c^2} \frac{\partial^2 P}{\partial t^2} = -\rho \frac{\partial q_{(t)}}{\partial t} = -jw\rho q_{(t)}$$

4.4.11

The wave equation is modified as

$$\nabla^2 P_o + k^2 P = -jw\rho q \qquad 4.4.11a$$

with the same boundary condition. Since the characteristic functions form a complete and orthogonal set of functions, the source function can be expanded in a series of ψ_n:

$$q_{(x,y,z)} = \sum_n C_n \psi_n \text{ with } C_n = \frac{1}{k_n} \iiint_V \psi_{n(x,y,z)} q_{(x,y,z)} dV \qquad 4.4.12$$

where the summation is extended over all possible combinations of subscripts. In the same way, the solution $P_{o(x,y,z)}$ can be expanded in characteristic functions:

$$P_{o(x,y,z)} = \sum_n A\psi_{n(x,y.z)} \qquad 4.4.13$$

since P_n are the solutions of the equation:

$$\nabla^2 \psi_n + k^2_n \psi_n = 0$$

$$\nabla 2\psi_n = -k^2 \psi_n \qquad 4.4.14$$

Substituting the values of P_o and q in Equation 4.4.11a and also the value of $\nabla^2\psi_n = -k^2_n \psi_n$:

i.e. $$-k^2_n A_n \psi_n + k^2 A_n \psi_n = -jw\rho c_n \psi_n$$

$$A_n = j\omega\rho \frac{c_n}{k^2-k_n^2} \qquad 4.4.15$$

Substituting for c_n:

$$A_n = \frac{j\omega\rho\frac{1}{V\Lambda_n}\iiint_{room} q\psi_n dv}{k^2-k_n^2} \qquad 4.4.16$$

If the source dimensions are small, compared to the wavelength of the radiated sound, it approximates to a simple source and the integral $\iiint q\psi_n dv$ (in Equation 4.4.16) reduces to $Q\ \psi_{n(x_o, y_o z_o)}$ where:

Q_o is the equivalent magnitude of source strength $\psi_{(x_o, y_o, z_o)}$ is the value of ψ_n at the source position.

Equation 4.4.16 then becomes:

$$A_n = j\omega\rho \frac{1/_V\ \Lambda n Q_o\ n\cdot(X_o Y_o Z_o)}{(k^2-k_n^2)} \qquad 4.4.17$$

Substituting for A_n in Equation (4.4.4), the expression for the instantaneous pressure at the point (x,y,z) in a room excited by a simple source of strength (Q) and frequency (w) is*

$$P = \frac{Q_o\rho}{V} \frac{\sum \frac{jw}{\Lambda_n} \psi_{n(x,y,z)}}{k^2-k_n^2} \psi_{n(x_o, y_o, z_o)} \qquad 4.4.18$$

or

$$P = \frac{Q_o\rho c^2}{V} \frac{\sum \frac{jw}{\Lambda_n} \Psi_{n(x,y,z)} \Psi_{n(x_o, y_o, z_o)}}{w^2-w_n^2} e^{-jwt} \qquad 4.4.19$$

but the term w is a complex quantity. The real part, w , corresponds to the resonance frequency, and the imaginary part corresponds to the damp- ing constant. Therefore:

$$P = \frac{Q_o^{\,2}}{V} e^{-jwt} \sum_N \frac{jw}{\Lambda_n} \frac{\Psi_n(x,y,z)\Psi_n(x_o,y_oz_o)}{w_n - j\delta_n^{\,2} - w^2} \qquad 4.4.20$$

which is the Green's function of the room.

When the source is placed at the corner of the room, the value of $\Psi_{n(x_o,y_o,z_o)}$ is equal to one.

4.5 Normalising Factors

These are weighting factors applied to each mode of vibration and determined by consideration of the location of the source and receiver in the room and the steady-state pressure amplitude to which each mode is excited (40).

The normalization factors for the function are the mean value of $(\ _n^2)$ averaged over the room volume.

$$V\Lambda_n = \iiint_{room} \Psi_n^2 dV \qquad 4.5.1$$

If the walls of the room are rigid, the function $_n$ is the product of the three cosines, and since the cosine functions have a mean-squared value of one for a n argument equal to zero and 1/2 for an argument different from zero, the constant $_n$ in the above equation can be computed as follows:

$$\Lambda_n = \Lambda_{nx} \Lambda_{ny} \Lambda_{nz}$$

4.5.2

where $\Lambda_{nx} = 1$ for $n_x = 0$

$\Lambda_{nx} = \frac{1}{2}$ for $n_x \quad 0$

and similarly for Λ_{ny} and Λ_{nz}.

If the walls have some admittance, the functions (Ψ_n) are hyperbolic functions with complex argument and their normalizing factors are given (41):

$$\Lambda_{nx} = 1 - \frac{jwl_x}{3c} \quad \frac{B_{x1}^3 + B_{x2}^3}{B_{x1} + B_{x2}}^2$$

and

for $n > 0$

and similarly for Λ_{ny} and Λ_{nz}

B_{x1} and B_{x2} being the admittance of the opposite walls in the room.

Substituting the value of $\Lambda_n = \frac{1}{8}$ and $Q = Q_o e^{-jwt}$ and considering the source position to be at the origin of the co-ordinate

$$P = \frac{8\rho c^2 Q}{V} \sum_{N} jw \frac{\Psi_{n(x,y,z)}}{w_n - j\delta_n{}^2 - w^2} \qquad 4.5.3$$

where $\Psi_n = \cos \frac{n\pi x}{L_x} \cos \frac{n\pi y}{L_y} \cos \frac{n\pi z}{L_z}$

When a room is excited by a corner loudspeaker, the relation between the volume velocity V_s and the response of the NML room mode is (from Morse)

$$\frac{PNML}{V_s} = Z_\phi = \frac{-jw\rho 8c^2}{L_1 L_2 L_3} \quad \frac{(-1)^L cf_{NML}}{W_{NML} - j\delta_{NML}{}^2 - w^2} \qquad 4.5.4$$

L_1, L_2 and L_3 - Room dimensions.

The sound pressure in a room excited by a point source of frequency (Room Acoustics, H.Kuttruff)

$$P_w(r) = iQw\rho_o \sum \frac{\Psi_{n(r)} \Psi_{n(ro)}}{k_n - (k^2 - k_n^2)} \qquad 4.5.5$$

This is called the 'Green's function' of the room under consideration, and it is symmetric in the co-ordinates r_o of the sound source and of the point of observation. By placing the sound source at r the same sound pressure should be observed at the point r_o as in r.

Since the boundary conditions are usually complex equations containing k_n, the latter are in general complex quantities.

$$K_n = \frac{w_n}{c} + i\,\frac{\delta_n}{c}$$

Assuming that $\delta_n \ll w$

$$P_{w(r)} = iQc^2 w\rho_o \sum_n \frac{\Psi_{n(r)} \Psi_{n(ro)}}{(w^2 - w_n^2 - 2i\delta_n w_n)k_n} \qquad 4.5.5a$$

considered as a function of the frequency. This expression is the transmission function of the room between the two points r and r_o. At the angular frequencies $w=w_n$, the associated term of this series assumes a particularly high absolute value. The corresponding frequencies are called 'eigen frequencies' of the room, and referred to as the 'resonance frequencies' because of some sort of resonance occurring in the vicinity of those frequencies. The δ_n term is called the damping constant.

The steady state acoustical behaviour of a room, when it is excited by a sinusoidal signal with angular frequency w, is described by a series of the form:

$$P_w = \sum_n \frac{A_n}{w^2 - w_n^2 - 2j\delta_n w_n} \qquad 4.5.5.6$$

where it is assumed that $\delta_n \lll w_n$.

Comparing this with the Equation 4.5.6a, it is noticed that the coefficient A_n are functions of the source position, of the receiving position and the angular frequency w.

Since the constants δ_n are small compared with the corresponding w_n, the absolute value of one series term changes so rapidly in the vicinity of $w=w_n$. If the frequency is altered, the frequency dependent can be neglected except that of dominator.

Since it is the term $w^2-w_n^2$ which is responsible for the strong frequency dependence, w_n can be replaced by w in the last term of the dominator without any series error. The absolute value of the nth series term then becomes:

$$\frac{|A_n|}{\left(w^2-w_n^2 \; ^2+4w^2\delta_n^2\right)^{\frac{1}{2}}}$$

and thus agrees with the amplitude frequency characterisitc of a resonance system. Therefore the stationary sound pressure in a room at one single exciting frequency proves to be the combined effect of numerous resonance systems with resonance (angular) frequencies w_n and damping constants δ_n.

The amplitude (coupling) of each resonant mode is considered to be:

$$\sum_n \frac{c_n w \Psi_n(x_o y_o z_o) \Psi_n(x,y,z)}{(w_n - j\,\delta_n)^2 - w^2} \qquad 4.5.6$$

where $N = m_1, m_2, m_3$

m_1, m_2, m_3 = mode numbers

c_n = coupling of the m ,m ,m room modes

w = angular frequency

w_n = natural frequency of m ,m ,m room modes

$_n$ = decay factor of m ,m ,m room modes

x,y,z = the co-ordinate of the receiver position

x_o, y_o, z_o = the co-ordinate of the source position

When the source is placed at the corner $\Psi_n(x_o, y_o, z_o) = 1$ and the Equation 4.5.6 is written as:

$$\sum \frac{c_n w \Psi_{n(x,y,z)}}{(w_n - j\delta_n)^2 - w^2} \qquad 4.5.6a$$

Now

$$c_{oBS} = C_n \frac{w \Psi_{n(x,y,z)}}{(w_n - j\delta_n)^2 - w^2}$$

means that this value is the observed level at the resonant frequency. Considering the absolute value of the nth series term and also knowing that at resonance $w_n = w$,

$$C_{oBS} = \frac{c_n w}{-2j\delta_n w - (\delta_n)^2} \simeq \frac{c_n}{2\delta_n}$$

Therefore

$$C_n = C_{OBS} \times 2_n$$

Consider the equation

$$\sum_n \frac{c_n w \psi_{n(x,y,z)}}{(w_n - j\delta_n)^2 - w^2} \qquad 4.5.6a$$

To simplify this expression and be able to separate the real and imaginary part so that it can be used in a computer program, the following steps are taken.

Let

$$\psi_{n(x,y,z)} = \psi_{n(x)}$$

$$\frac{c_n w \psi_{n(r)}}{(w^2_n - \delta^2_n - w^2)^2 + (2w_n \delta_n)^2}$$

$$= \frac{c_n w \psi_{n(r)} \left(\left(w^2_n - w^2 - \delta^2_n \right) + 2jw_n \delta_n \right)}{\left(w^2_n - \delta^2_n - w^2 \right)^2 + \left(2w_n \delta_n \right)^2}$$

$$= \frac{c_n w \psi_{n(r)} \left(w_n^2 - w^2 - \delta^2_n \right)}{w^2_n - {}^2_n - w^2 \, {}^2 + 2w_{n\,n} \, {}^2} + j \frac{c_n w \psi_{n(r)} (2w_n \delta_n)}{(w^2_n - \delta^2_n - w^2)^2 + (2w_n \delta_n)^2}$$

To find the magnitude of this expression, let

$$Z1 = \left(w^2_n - \delta^2_n - w^2 \right)^2 + (2w_n \delta_n)^2 \qquad 4.5.7$$

$$S2 = \sum_n \frac{c_n w \psi_{n(r)} \left(W^2_n - w^2 - \delta^2_n \right)}{Z1} \qquad 4.5.8$$

$$T2 = \sum_{n} \frac{C_n w \Psi_{n(r)} \left(2w_n \delta_n\right)}{Z t} \qquad 4.5.9$$

$$SI = \left(S_2^2 + T_2^2\right)^{1/2} \qquad 4.5.10$$

4.6.0 The Synthesis Computer Program

The relative position of the microphone to the source in 3-dimensional co-ordinates is given,

The number of peak frequencies in the range of the frequencies that have been swept is noted,

The dimensions of the room are also given.

The following values related to each resonant mode are also given:

1) resonant frequency

2) The mode number of each resonant frequency

3) The amplitude (coupling) of each mode

4) The damping which is a function of the bandwidth

The values of resonant frequency, the amplitude and the damping of each mode is obtained from the computer program Analysis-print and the mode number is obtained when the theoretical and experimental values of the resonant mode are compared. For each theoretical value of the resonant mode there is the respective mode number.

The value of $\Psi_n(x,y,z)$ for each resonant mode is obtained from the expression:

$$\Psi_n = RC = \cos\frac{m_1 \pi x}{L_x} \cos\frac{m_2 \pi y}{L_y} \cos\frac{m_3 \pi z}{L_z}$$

From Equation 4.5.11

$$C_n = 2C_{OBS}\,\delta_n$$

The values of Ψ_n and C_n for each resonant mode are obtained. The Equation 4.5.6, i.e.

$$= \sum \frac{c_n w \Psi_{n(r)}}{(w_n - j\delta_n)^2 - w^2}$$

where $C_n = C_{obs} X2\, \delta_n$

is applied and for each value of w the summation of the expression which leads to Equation 4.5.10 is obtained.

Then the values of amplitude for each interval of the frequency which is swept are stored on the floppy disk.

The synthesis computer program with its full explanations is listed on the following pages.

The applicability and the validity of the synthesis program was tested when the theoretical curve of amplitude v frequency was plotted for the range of frequency under consideration and particular microphone position. The amplitude and bandwidth values were both taken from the analysis of the frequency curve. Then for each microphone position the experimental and theoretical graphs were produced by applying program 'PLOTEXP'. The theoretical and experimental curves were produced by inputting the data obtained from experimental results to program Synth Disk , where is the microphone position.

The 4096 experimental points of amplitude and frequency were reduced to 512 point for plotting purposes and then the program 'PLOTEXP' was applied to plot the curve. When compared it showed that they matched reasonably well and only when the modes were close was an increase in coupling of the modes observed. (This is studied in more detail later.)

It must be mentioned that no correction factor concerning the frequency response of the loudspeaker was applied since the frequency response of the loudspeaker did not vary considerably over the frequency at which the experiment was carried out.

The program Data Big to Small is used to reduce the 4096 points of amplitude to 512 points, so that the curves could be plotted on the HP plotter.

The experimental and theoretical frequency response curves are shown in Figures 4.6.1 to 4.6.6.10. It must be noted that the frequency bandwidth values that have been obtained from the frequency response curve are the combination of the resonant modes bandwidth for two very close modes and also the contribution from the off-resonant mode, and when this is computed to the synthesis program in the case of two close resonant modes they tend to merge and appear as a single frequency with a much larger bandwidth.

The plot of the synthesis program when the damping and coupling values from the frequency response curve was applied showed that when the two resonant modes were close to each other the two modes combined and appeared as a single mode with different characteristics as far as bandwidth and coupling were concerned. Therefore the behaviour of the two close modec was studied further.

The synthesis program was plotted for each of the modes separately (Figurs 4.6A, 4.6B) and then was compared with the curve when both of the close modes were considered. As has been mentioned, the two modes combined and appeared as a single mode (Figure 4.6.1 to 4.6.10). Another set of graphs was then obtained when the bandwidth of each mode was reduced by the same amount. The procedure was repeated until the experimental and theoretical graph were in good agreement; this was achieved when the values of the bandwidth were reduced as shown in Figures 4.6.1A to 4.6.10A. The contribution from the two close resonant bandwidths, especially when one or both are heavily damped, can distort the modal shape considerably.

The damping values that are obtained from the frequency response curve are not necessarily the damping values of a pure mode. There are off-resonant contributions from adjacent modes and this often contributes towards the higher damping values, as can be proved when two modes are very close to each other and one of the modes is dominant.

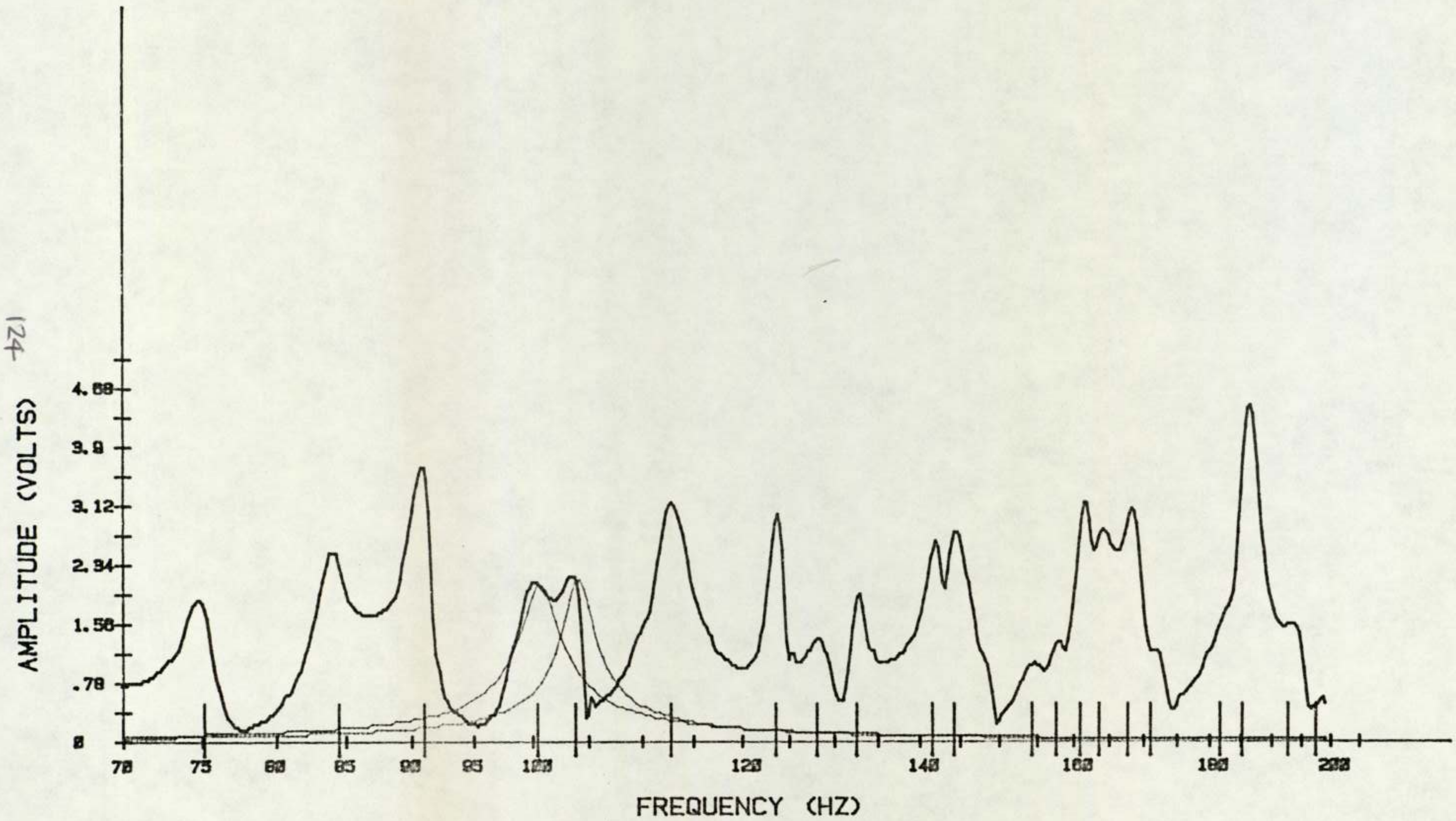
Further study of the resonant modes and their damping, coupling and mode number of each mode and the relationship between these characterisitcs when two modes are very close to each other needs to be studied in detail.

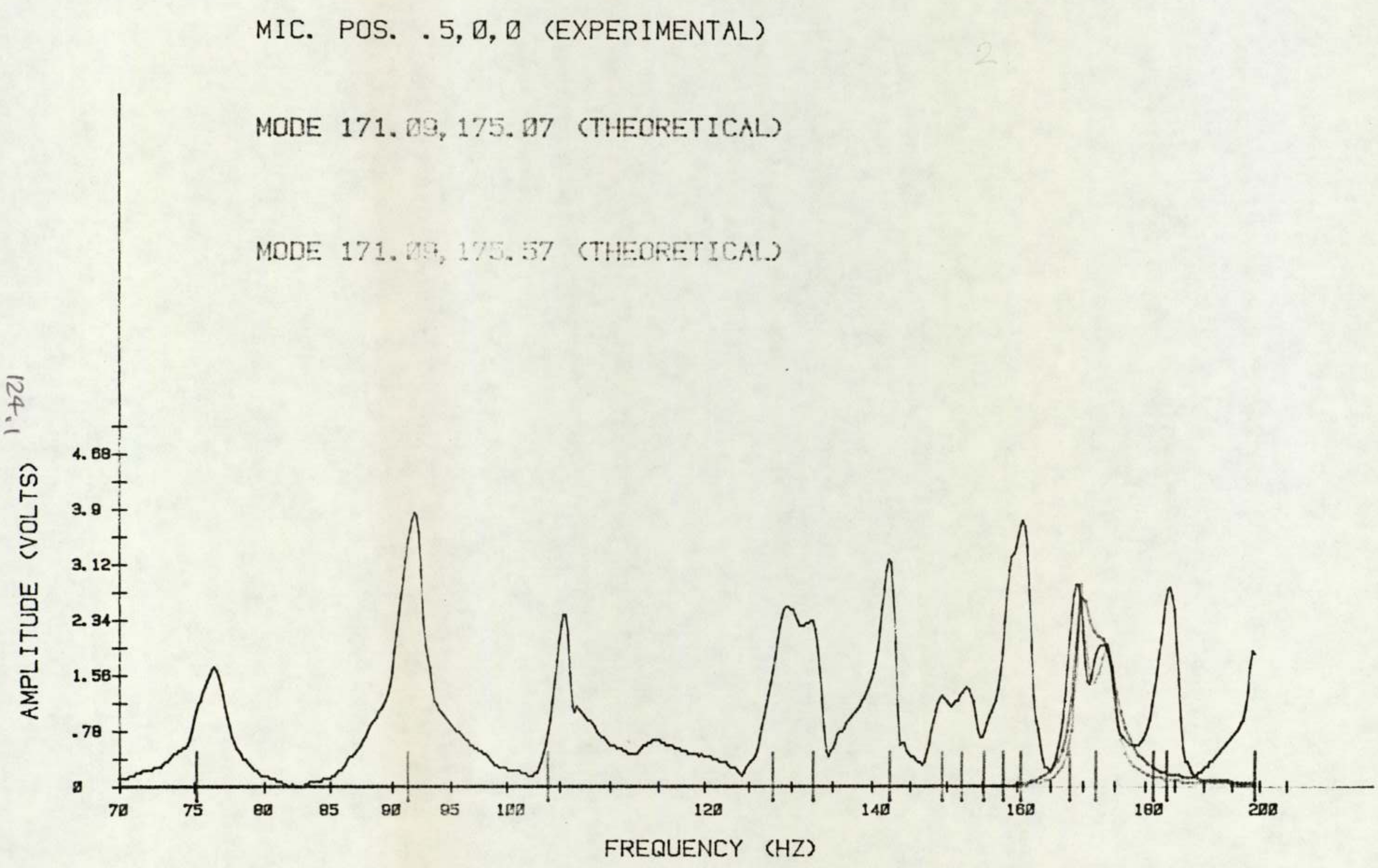
It was noted that placing the microphone at the various corners of the room did not lead to the full excitation of all the resonant modes, as in some corners some of the modes disappeared and in other corners some modes appeared again. This has led to the strong belief that a particular corner position of the microphone does not necessarily excite all the modes to their full capacity and the same is true for the loudspeaker position.

The microphone was placed at various corners of the room (1,0,0), (1,1,0), (1,1,1), (0,1,0), (0,1,1) and the result of the resonant mode for each microphone position was compared with the results obtained from the theoretical analysis. It must be emphasised again that the theoretical calculations were based on the assumption that all the surfaces are rigid. It must also be pointed out that the low frequency range is covered and therefore this can cause problems as far as the impedances of the walls are concerned. To study this it is necessary that the impedances of the surfaces at the range of frequency be studied in full detail, since it seems that at low frequency the behaviour of the impedances of the surfaces of the room can influence the results of resonant frequency amplitude and bandwidth of the mode to a high degree.

The source position can also influence the degree of coupling and the frequency bandwidth of each mode. To show the degree of influence of sound source position on the amount of coupling inside the enclosure, the source was placed at various corners of the room and the receiver position was moved to positions so that the relative position remained

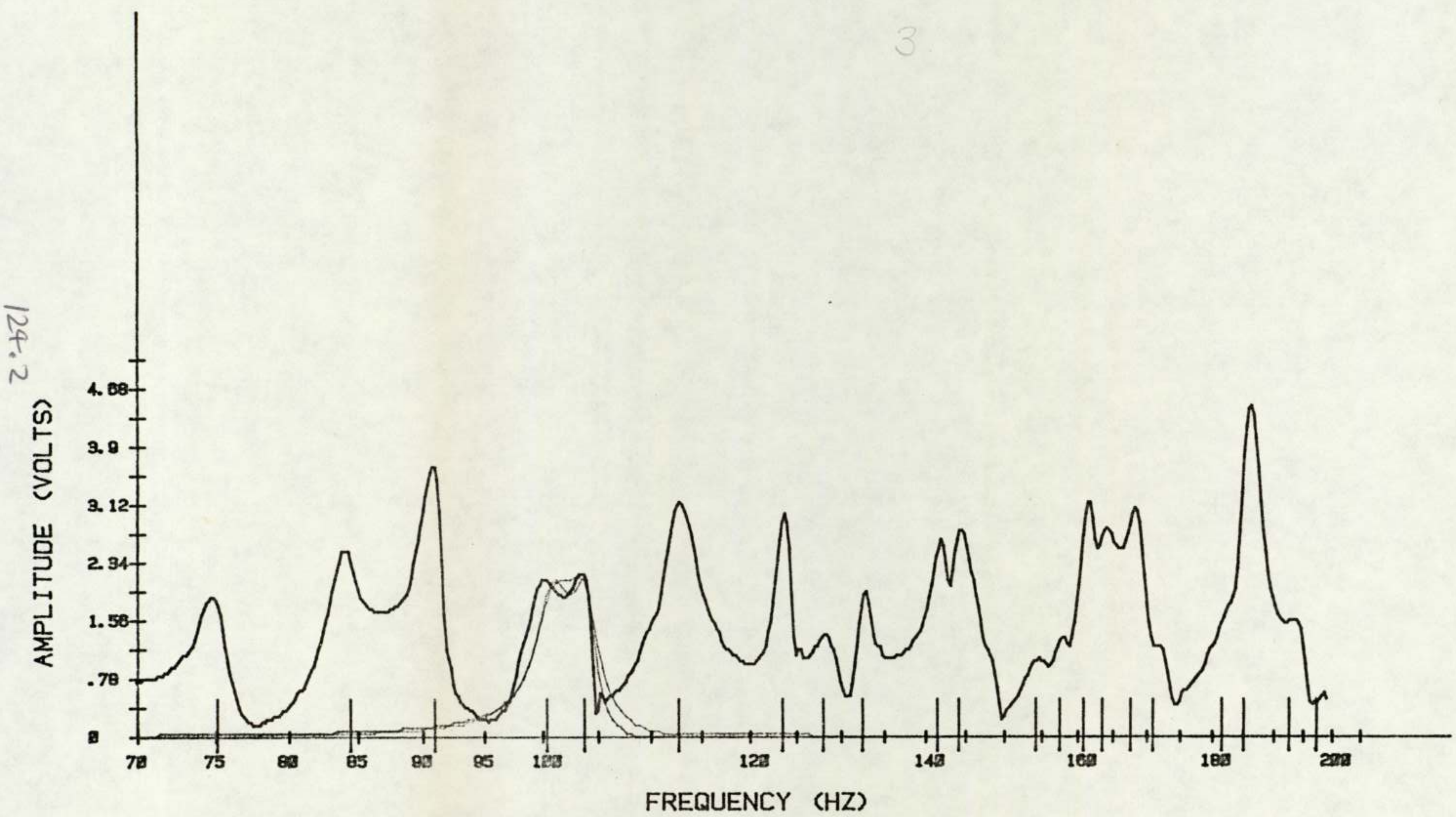
MIC. POS. 1, 1, Ø (EXPERIMENTAL)




MIC. POS. .5,Ø,Ø (EXPERIMENTAL)
MODE 171.Ø9, 175.Ø7 (THEORETICAL)
MODE 171.Ø9, 175.57 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
Ø
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

124.1

MIC. POS. 1, 1, Ø (EXPERIMENTAL)



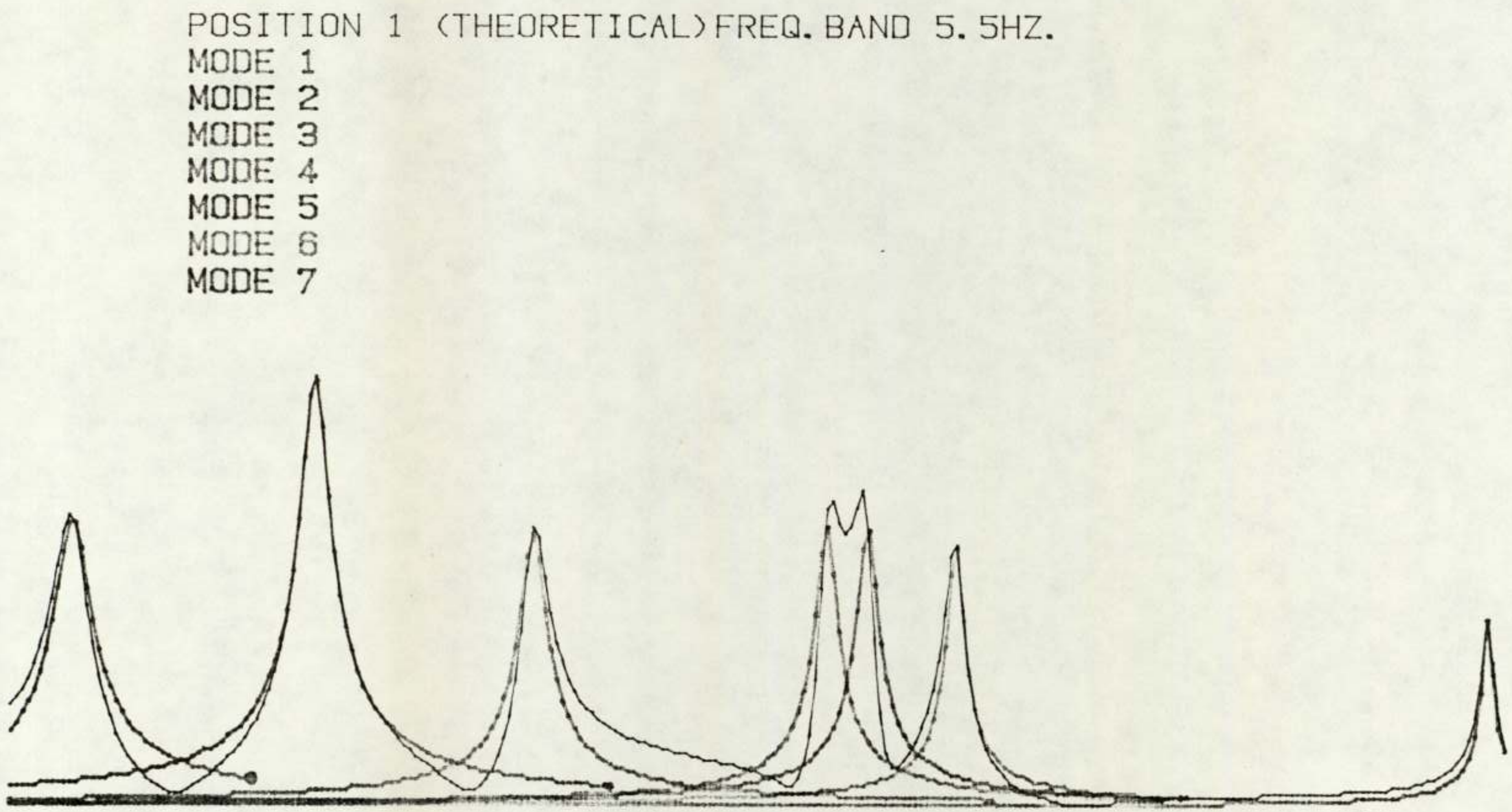

POSITION 1 (THEORETICAL)FREQ. BAND 5.5HZ.
MODE 1
MODE 2
MODE 3
MODE 4
MODE 5
MODE 6
MODE 7

FIG. 4.6.A

POSITION 1 (THEORETICAL) FREQ. BAND 5.5HZ.
MODE 1
MODE 1

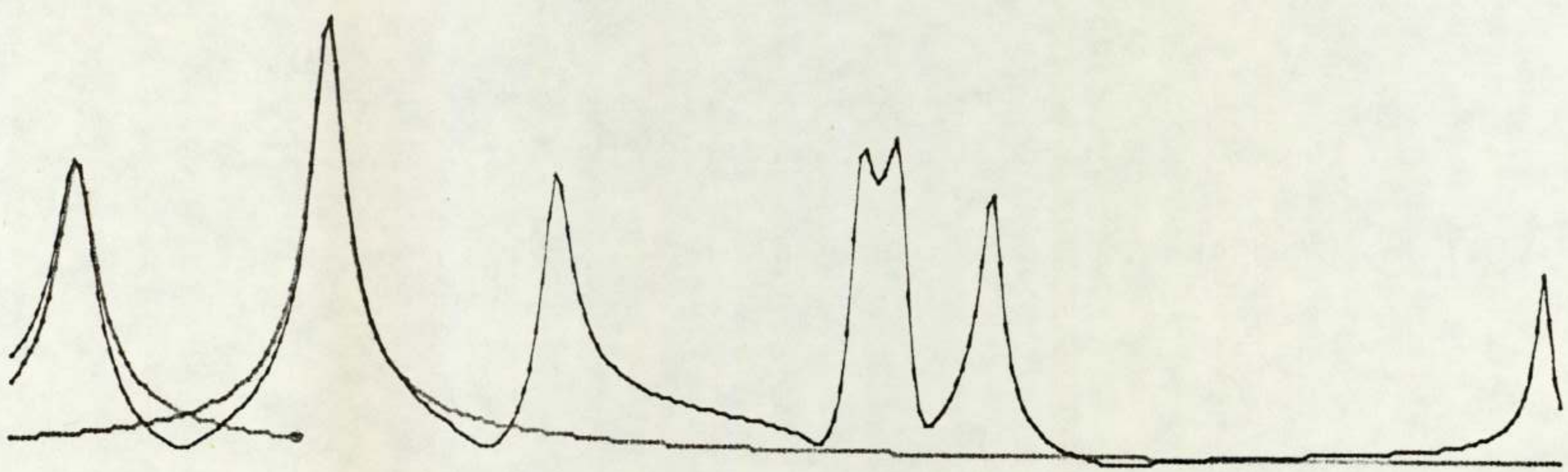


FIG. 4.6.B

the same for each source and receiver position and a comparative study of the results was then carried out.

Further investigations were carried out by obtaining the frequency response curve and altering the damping of two close modes, since when the two modes are close there is always some contribution from the off-resonant mode and this can contribute to change the shape and the damping of the resonant mode. The results are shown in Figures 4.6.20, 4.6.21, 4.6.22, 4.6.23. Sound pressure level perpendicular to the axis of the double leaf door varied considerably and this was proved by considering wave number related to the particular resonant mode. Although this was not true when the minima pressure on the x-axis while the pressure on the y and z axes of the room were maxima. (This will be investigated further later.)

The variation in the bandwidth of the resonant modes is the most complex phenomenon, and that it does not seem easy to explain may be due to the fact that

a) when both doors of the room are closed, the double door actually acts as a laminated structure and therefore the law concerning a single uniform rigid surface does not hold any more and this causes a sudden change in the sound pressure level inside the room. This will be investigated in more detail later.

b) the position of the door influences the coupling of the modes. The relationship between the position of the door and source and the microphone position relating to the degree of coupling of the modes will be investigated further.

This idea can be expanded to the surface boundaries of the room under investigation and the influence of the boundary surfaces made of different materials on the level of coupling in relation to the source and receiver position is one area that must be looked at in detail. This will result in the introduction of new terms to the equation of the wave number and, subsequently, consideration of the complex part of the surfaces impedance.

As has already been mentioned, the boundary of the enclosure influences the behaviour of the sound considerably and as a result the degree of coupling and also the frequency bandwidth of the mode are affected. For the purpose of simplicity the derivation of the formula for the wave number in the synthesis program was based on the assumption that all the boundary surfaces are rigid, but in practice this is not true, as the surfaces of the room under investigation were not rigid and in particular the existence of a double leaf entrance door to the room has already proved to be one of the main causes of the anomalies in coupling and the bandwidth of the modes.

The location of the door and its influence on the degree of coupling and bandwidth is the subject that has already been investigated. The frequency response curve obtained showed that there is a marked difference in the level of coupling of the modes perpendicular to the plane of the door when the results of both doors being shut are compared with the results of one inner door being kept shut. Detailed examination of the curves showed that

a) the variation in the level of coupling varies considerably for different positions of the receiver inside the room,

b) the variation in the level of coupling is true for the odd modes perpendicular to the plane of the door. This can almost be generalised.

c) the degree of coupling between the modes has always been increased when the outer door was kept open,

d) for some modes considerable frequency shift was observed. This could be due to the location of receiver, the wave number of the mode, the location of the door or the positioning of the source relative to the door. Although this part is not investigated in detail here, it is essential that this fact be taken into consideration when the theoretical curve is not in good agreement with the experimental curve.

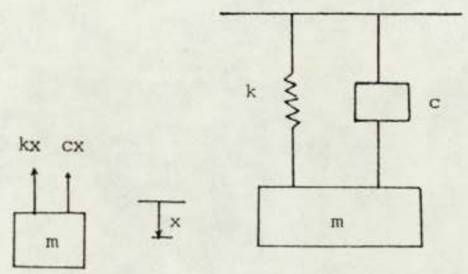


FIG. 4.1 Free Vibration with Viscous Damping

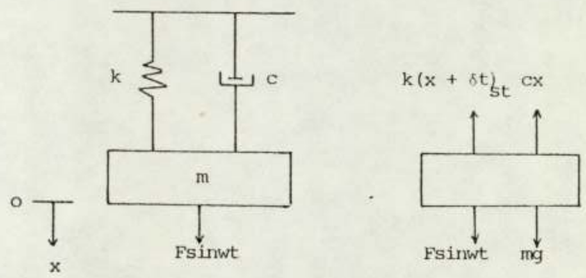
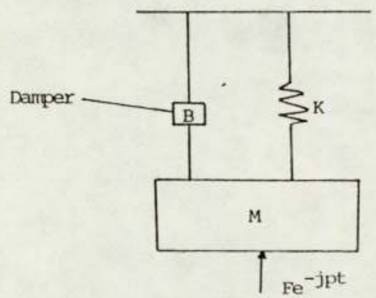
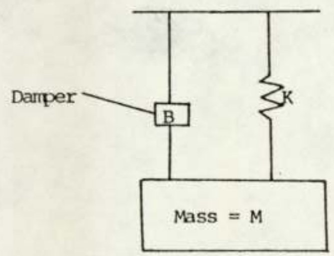


FIG. 4.2.1 Forced Vibration with Harmonic Excitation.



431



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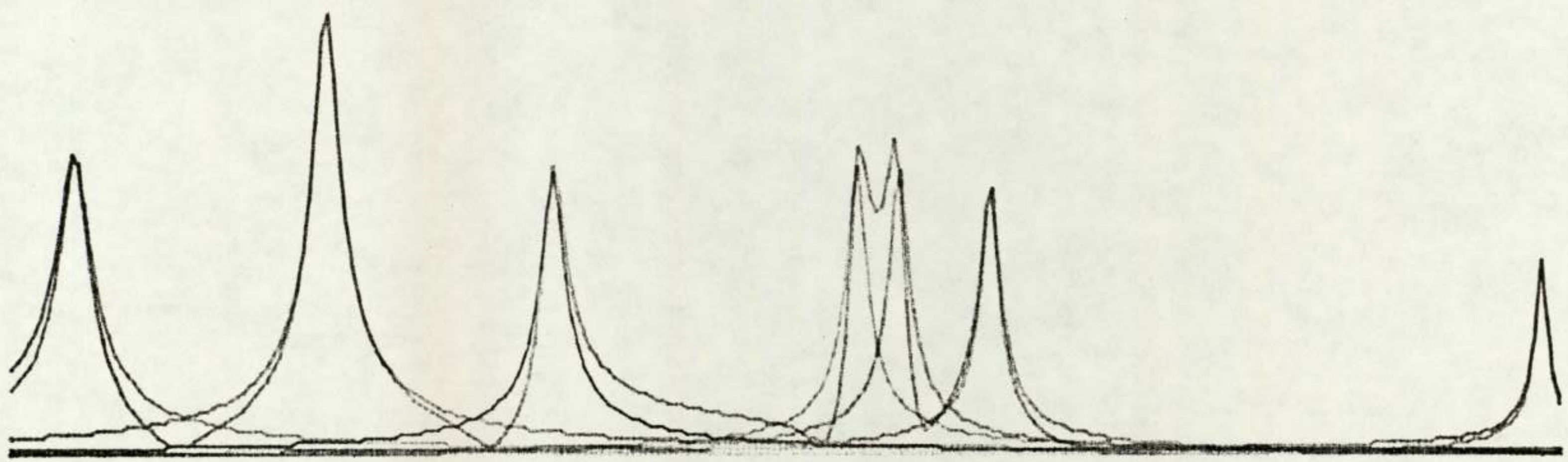
Comparing Each Mode Individually

MODE1M (THEORETICAL)



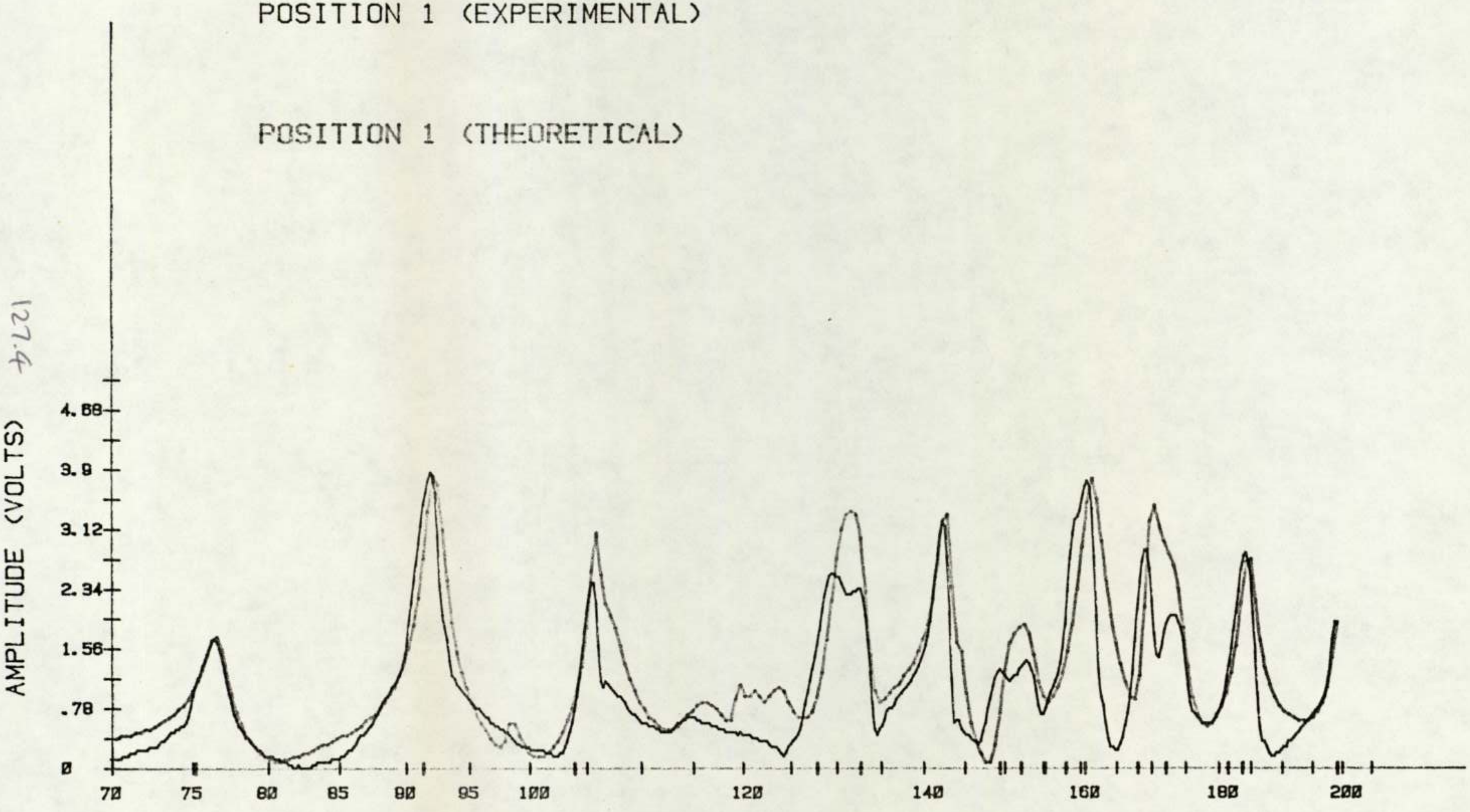

POSITION 1 (EXPERIMENTAL)
POSITION 1 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.1

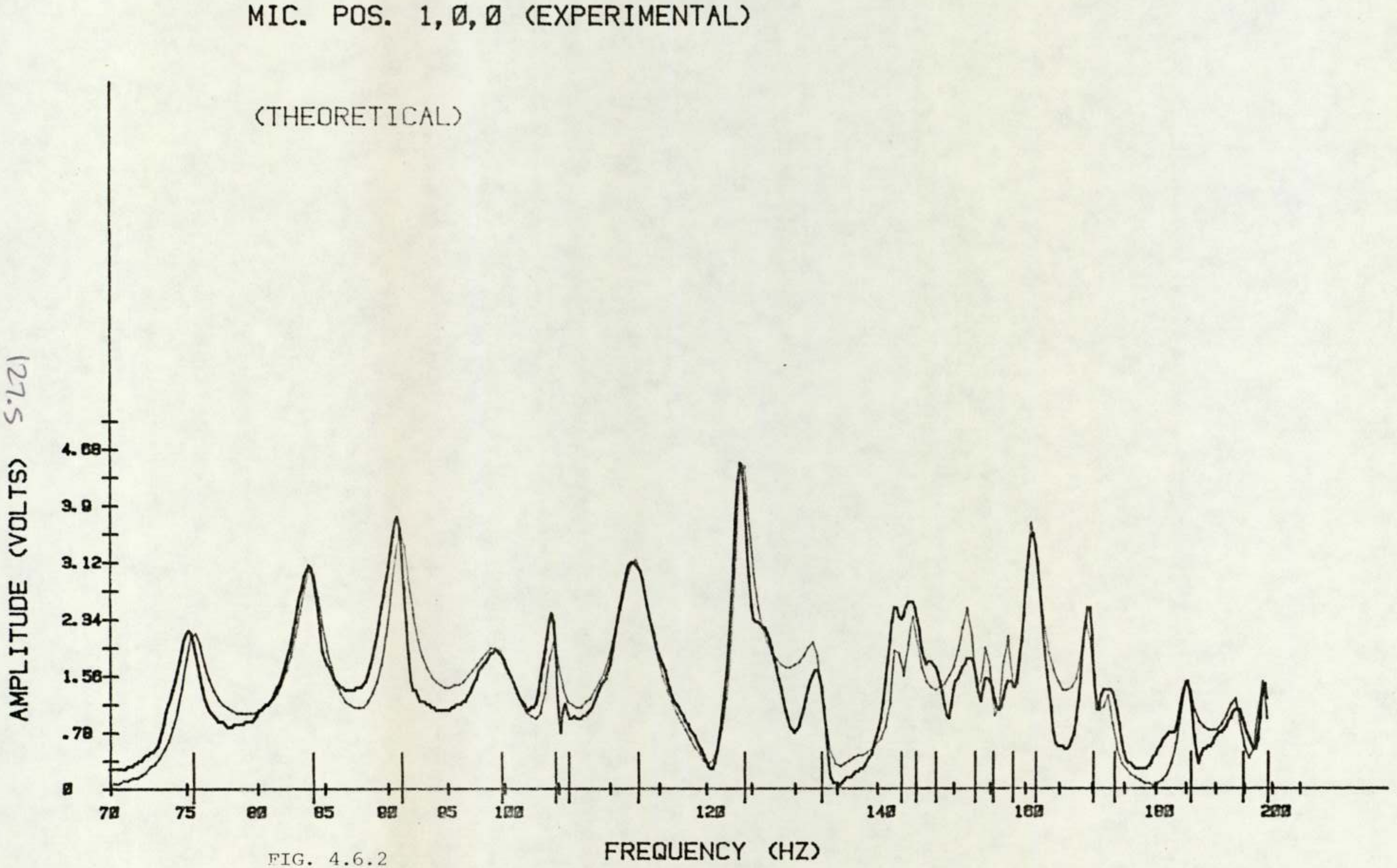

MIC. POS. 1,Ø,Ø (EXPERIMENTAL)
(THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
Ø
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.2

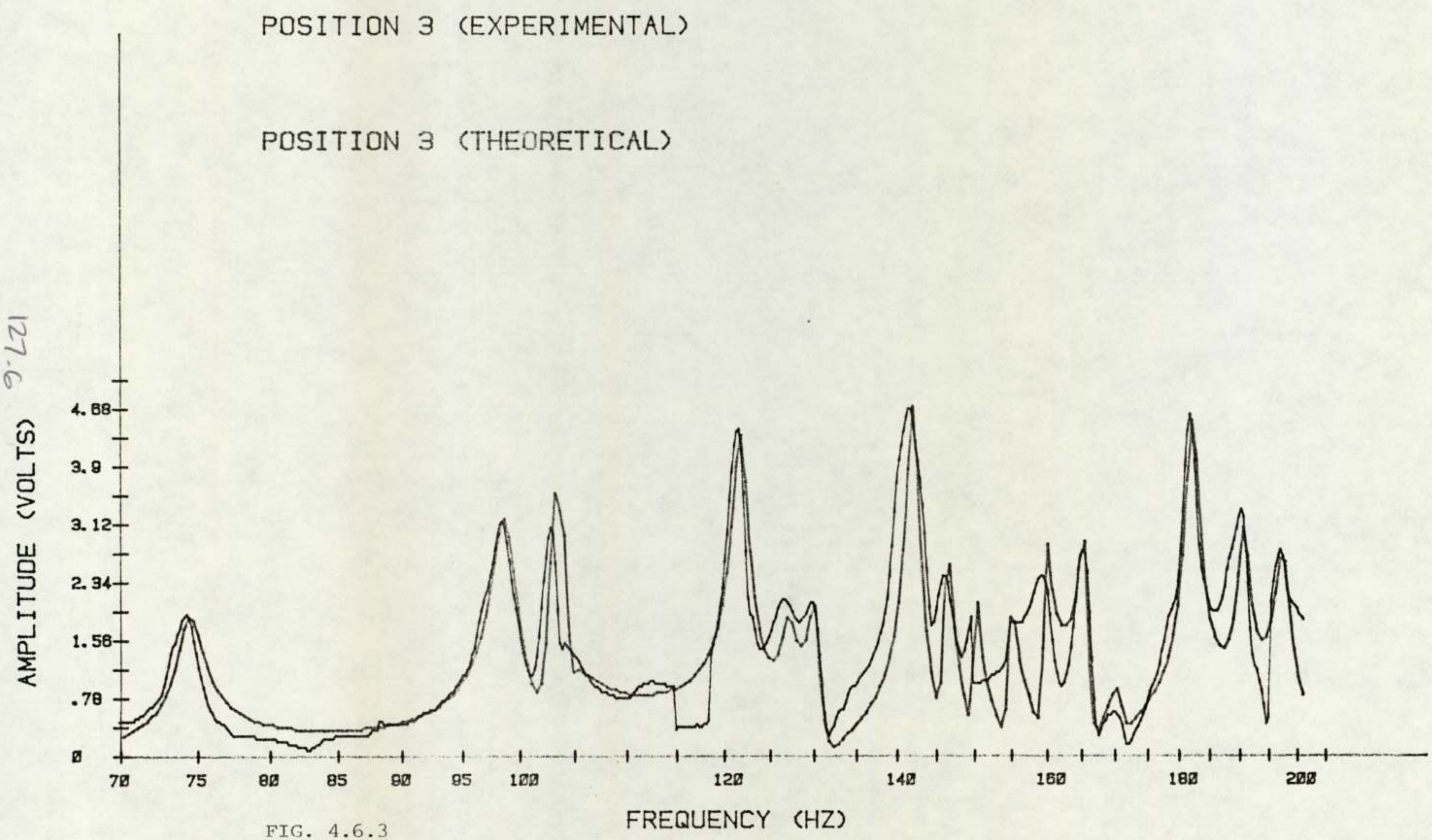

POSITION 3 (EXPERIMENTAL)
POSITION 3 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.3

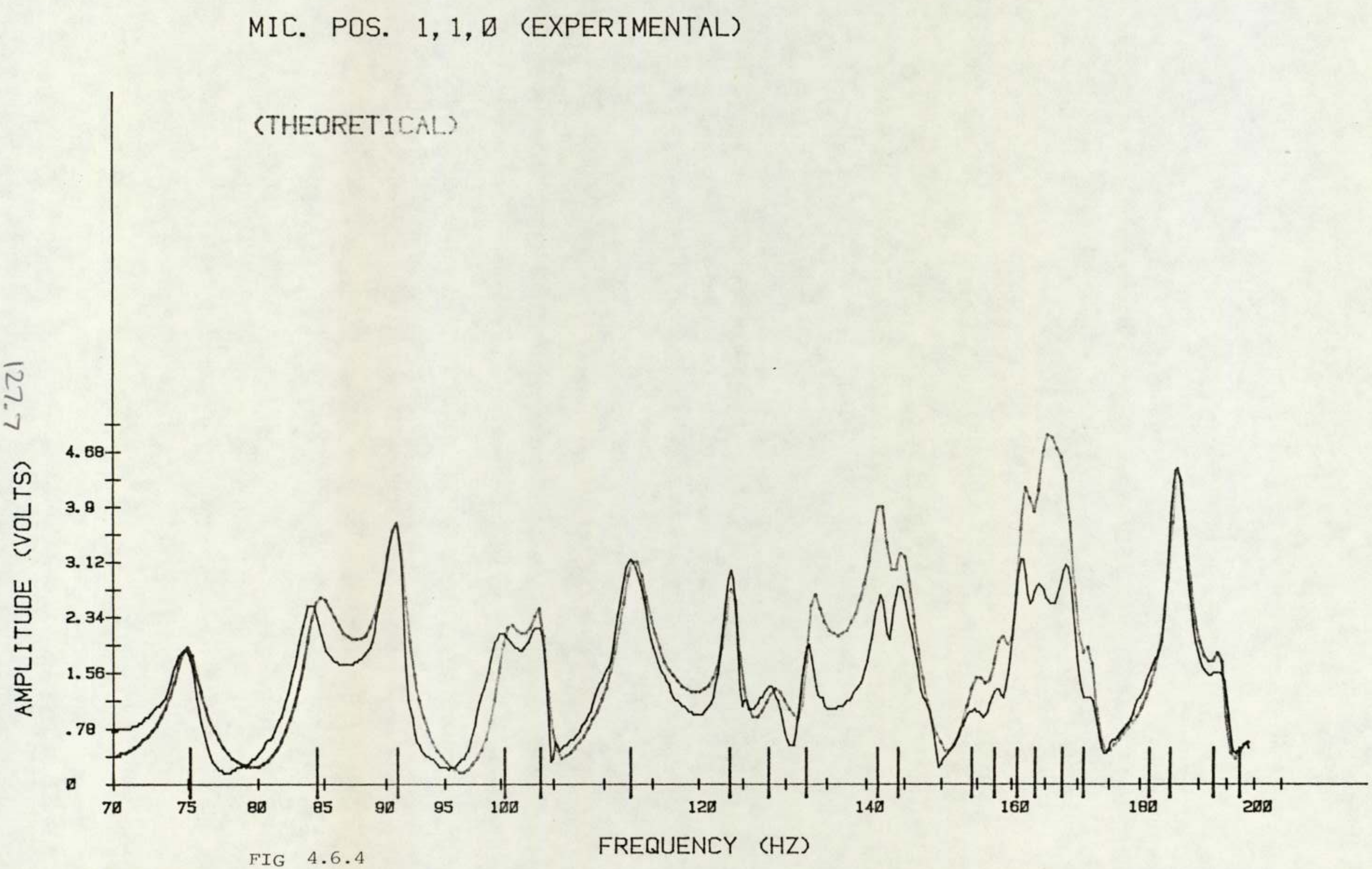

MIC. POS. 1, 1, Ø (EXPERIMENTAL)
(THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
Ø
70
75
8Ø
85
9Ø
95
1ØØ
12Ø
14Ø
16Ø
18Ø
2ØØ
FREQUENCY (HZ)

FIG 4.6.4

127.7

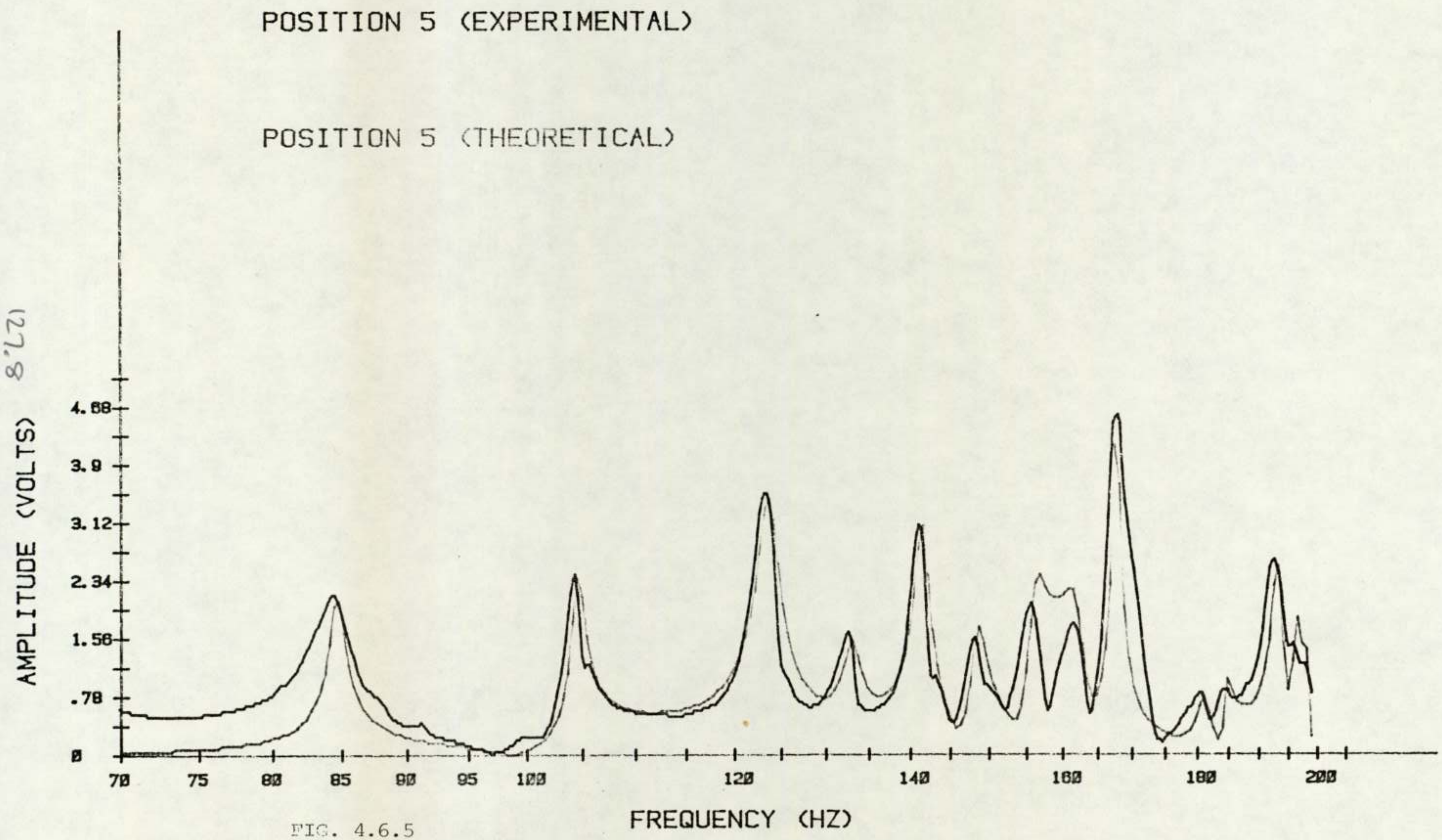

POSITION 5 (EXPERIMENTAL)
POSITION 5 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.5

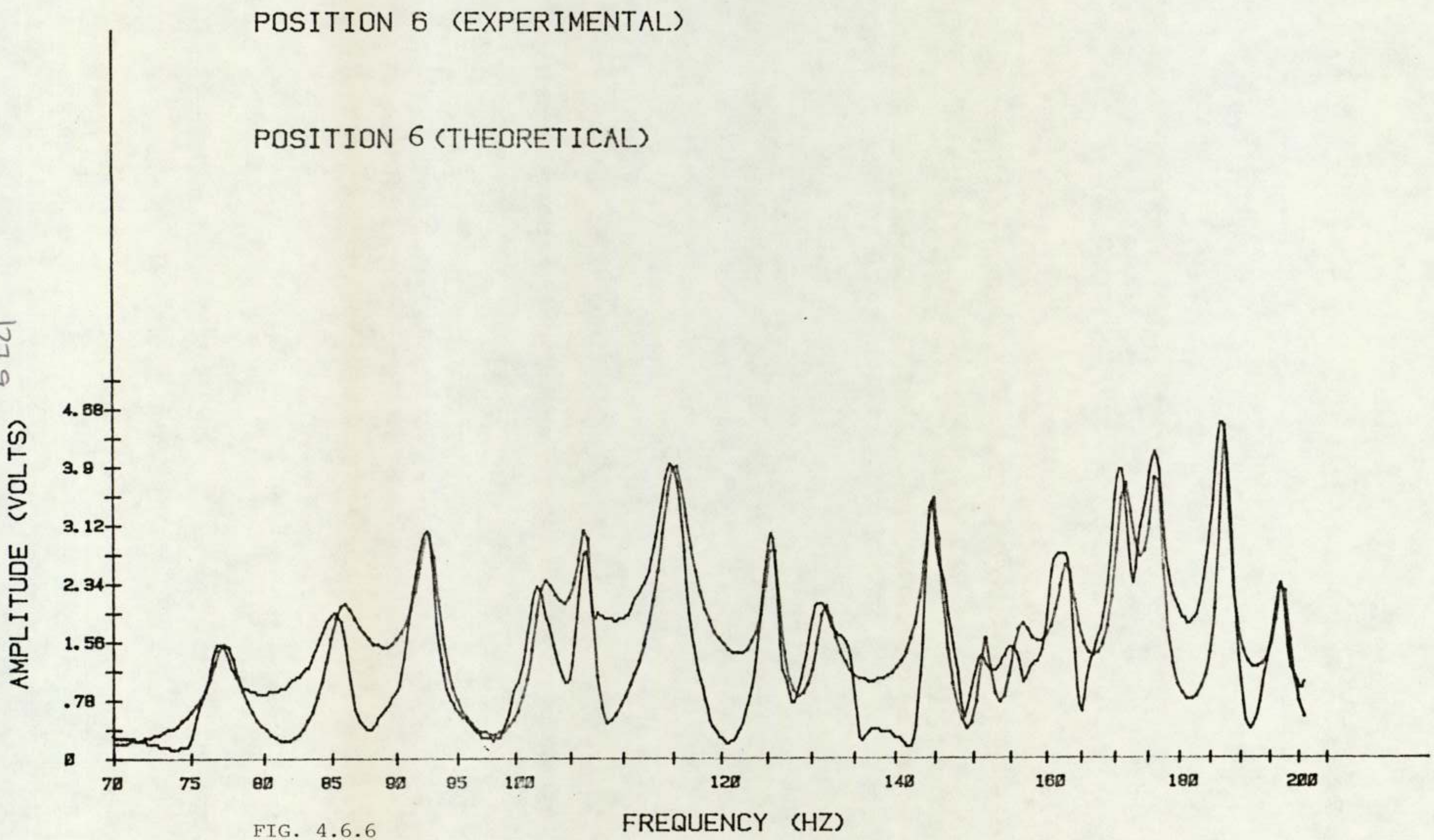

POSITION 6 (EXPERIMENTAL)
POSITION 6 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.6

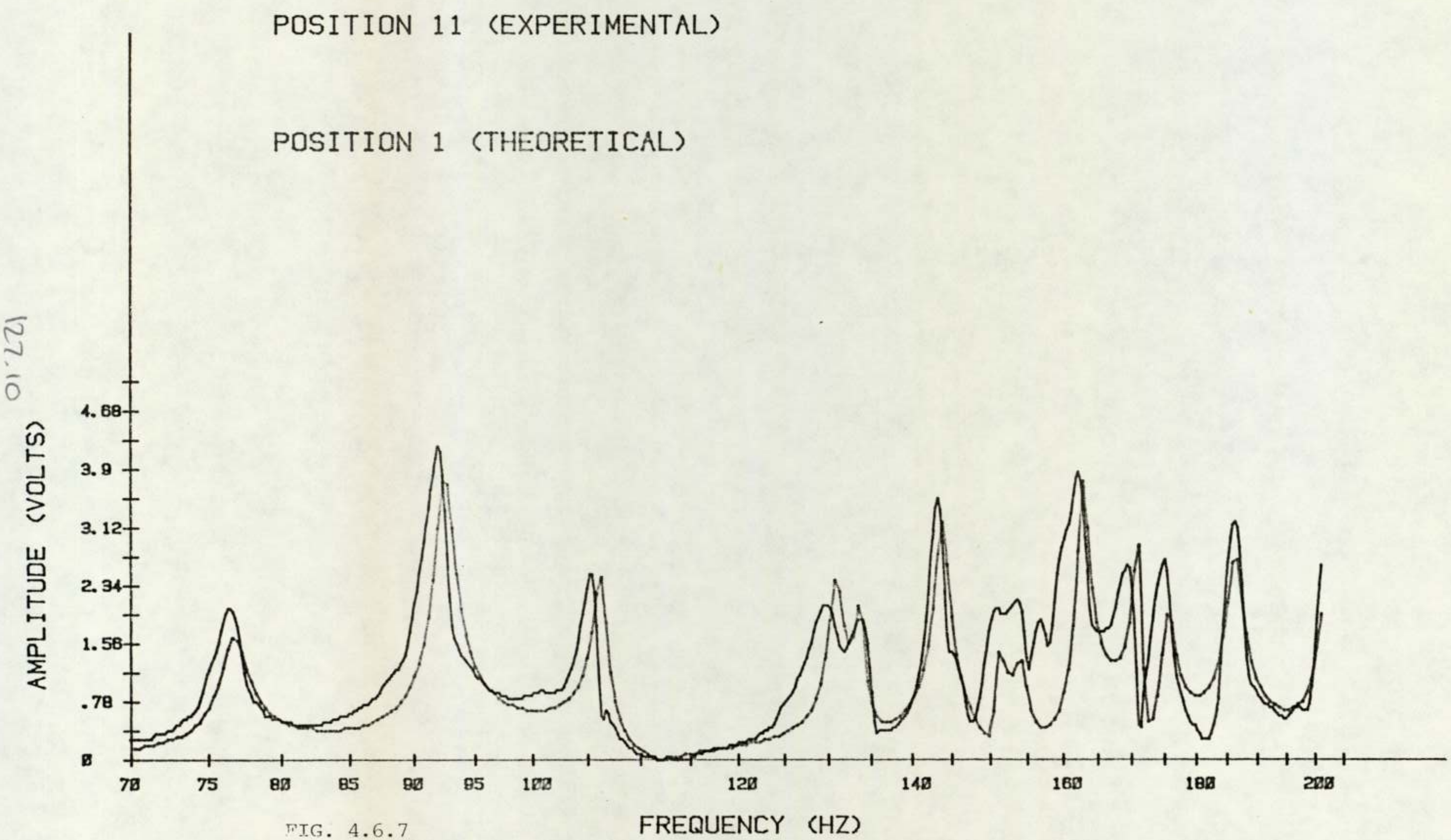

POSITION 11 (EXPERIMENTAL)
POSITION 1 (THEORETICAL)
AMPLITUDE (VOLTS)
4.68
3.9
3.12
2.34
1.56
.78
0
70
75
80
85
90
95
100
120
140
160
180
200
FREQUENCY (HZ)

FIG. 4.6.7

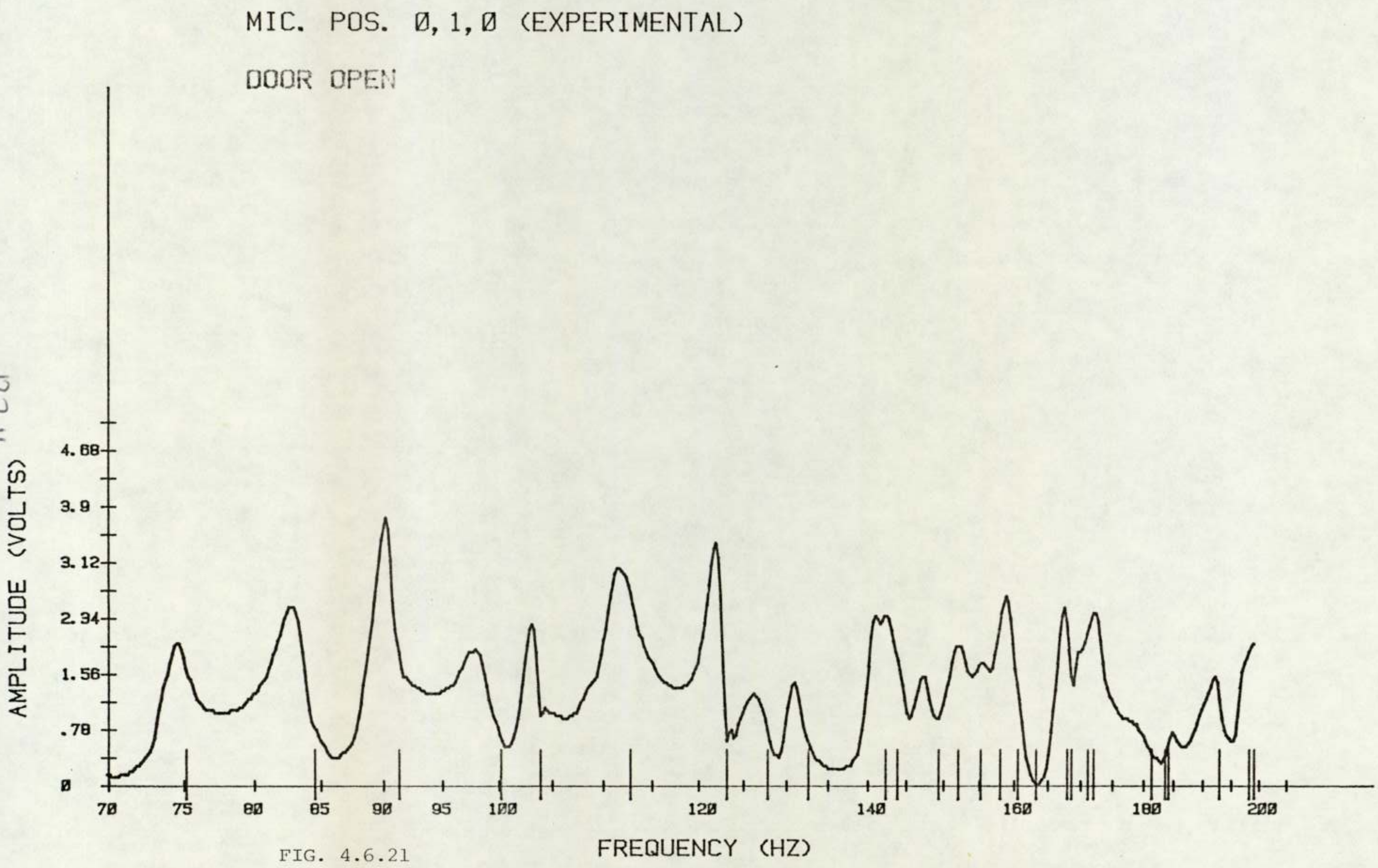


FIG. 4.6.21

127.11

MIC. POS. Ø, 1, .5 (EXPERIMENTAL)

DOOR OPEN

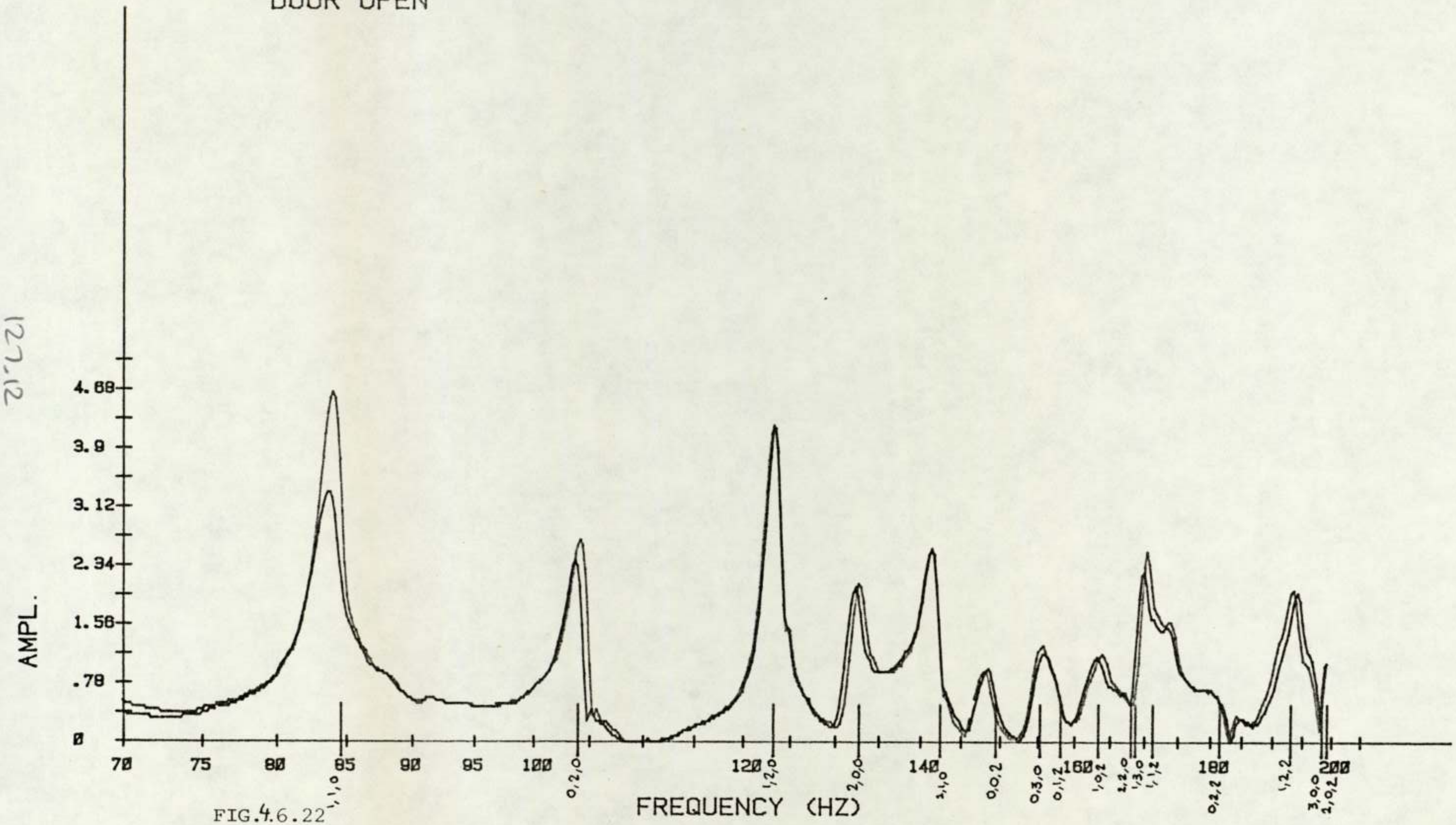


FIG.4.6.22

127.12

MIC. POS. Ø, 1, 1 (EXPERIMENTAL)

DOOR OPEN

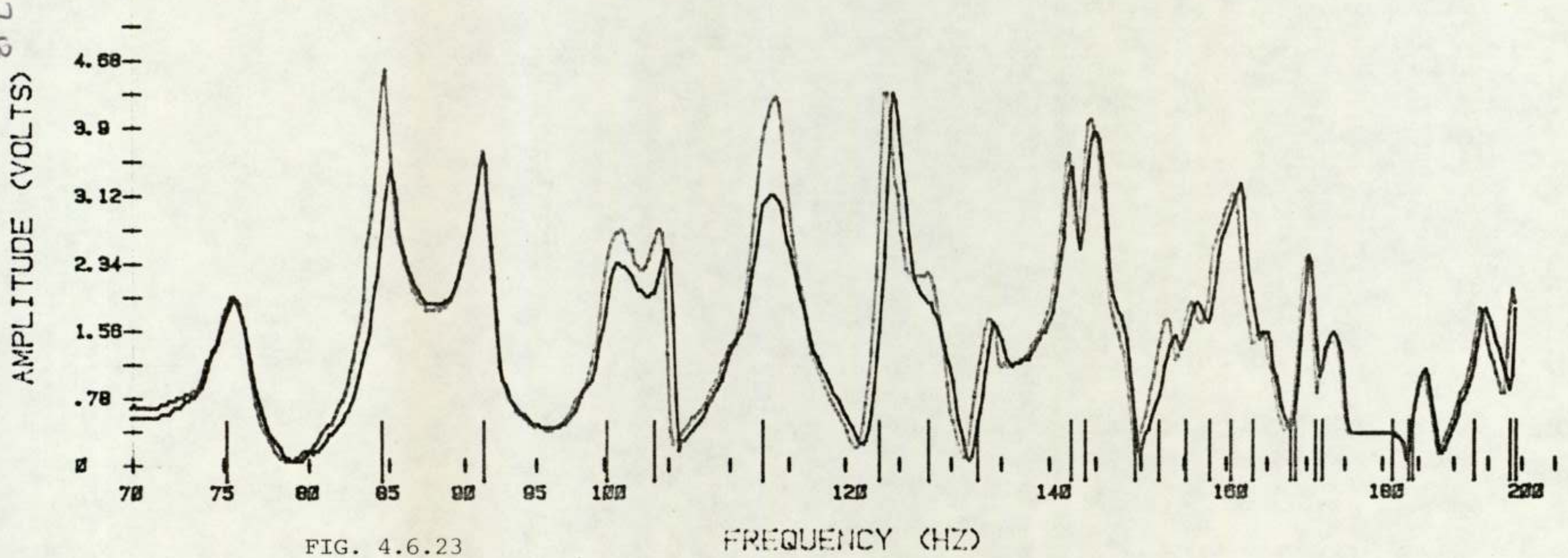


FIG. 4.6.23

127.13

CHAPTER 5

So far in this work, the prime concern has been devoted to deriving theoretical predictions and establishing experimental measurements of modal parameters characterising the physical properties of sound fields and relating the behaviour of sound at low frequency range with the theoretical analysis when the room is excited sinusoidally under steady state conditions.

The acoustic studios that are primarily used for microphone pick up include radio-broadcasting, television and sound-recording studios. Microphone pick up is subject to about the same peculiarities as mono aural hearing, and therefore a studio should have about the same acoustical properties as would be required for the most favourable conditions for listening with one ear.

However, the introduction of stereophonic systems in sound transmission and recording is an attempt to overcome this limitation. The use of multi-channel systems fed from several pick-up microphones leads to a change in the ratio of reflected and direct sound energy in reproduced sounds that helps in the location of the apparent sound source correctly and gives naturalness and fullness to the change of sound (22). The reproduced sounds have specific time shifts which change with changes in the position of the performer relative to the microphone system during recording and this helps to enhance the spatial character of the sound.

The Subjective Qualities of Small Studios

In view of differences between speech and music, the requirements for speech studios and for music studios are not identical and, accordingly, the qualities for subjective judgement of them are different.

In studios intended for speech, the listener expects the room to render speech intelligible and therefore the prime objective in the design of such studios would be the realization of the conditions that will provide intelligibility of speech. In studios used for music, the room is expected to support the music being performed and the objective is the creation of the most favourable conditions for the preservation and enrichment of the formal quality of the music.

The rating of music is more elusive and therefore some suitable criteria must be obtained for the most preferred listening conditions in music rooms.

However, there are certain broad features that apply to both types of room, and the emphasis on these common features is justifiable since it is not uncommon for the studios to be used on a programme sharing basis, especially television studios.

The problem is aggravated by the numerous factors which influence the making of subjective value judgements on the acoustical qualities of rooms used for speech or music presentation. In judging the acoustical qualities of a studio, one would then expect a difference of opinion not only on which qualities play an important role in the overall assessment, but also on the manner these very individual qualities are judged.

The Conditions for 'Good Acoustics' in Small Studios

The most important requirements that should be considered in the design of studios are:

1) Spatial diffusion and uniformity of the distribution of sound energy throughout the room. Diffusion improves the liveliness of the room and enhances the natural qualities of speech and music. In practical considerations it reduces the hazards of improper microphone locations. Alternative techniques for measurement of diffusion have been worked out and reported (41,42).
2) Optimum ratio of direct to reverberant sound energy: Reverberation is considered to continue to the loudness of direct sound in speech rooms. It blends musical sounds and masks their imperfections.
3) Lateral reflections: the provision of a spatial impression of the sound field is affected by the directional distribution of the sound components. In this respect, lateral reflections are more significant than horizontal ones; and the ratio of the lateral to the non-lateral energy is a measure of the spaciousness of the room.
4) Early reflections: the ratio of the early sound, which is a combination of direct sound and first reflections to the reverberant sound, defines the room liveness (43). It is the sound signal to reach the listener first which subjectively determines the direction from which the sound comes. This fact is called - according to

Cramer - the law of the first wave front (20). The condition under which this rule is valid has been investigated by Hass (20).

5) Freedom from acoustical defects: the superposition of a strong isolated component on the sound can cause an undesirable effect, especially with music, namely a colouration. This is represented by a characteristic change of the signal spectrum, which takes the form of an unnatural and often monotonous emphasis of a certain frequency of sound (31). This occurs at low frequencies.

The phenomenon of flutter echo, by contrast, occurs at mid and high frequencies. The listener hears a harmonic series of sound components which remain separate and distinct up to high frequencies and are frequently repeated.

Both of these defects require proper treatment.

The Physical Properties of Sound Fields in Small Rooms

In order that the objective requirements for 'good acoustics' in small studios can be related to some physical properties of sound fields in small rooms, these physical properties must be derived in terms of the measurable modal parameters, i.e. the natural frequencies of the normal modes, their damping constants and modal shapes.

It is worth mentioning that, in considering small rooms, there are some basic differences in their acoustical behaviour between large and small rooms. Large rooms, by virtue of their size, are characterized by reverberation and echoes with delays of recognizable time scales, whereas in small studios reverberation is no longer appreciable as a time extension of the original sound. Instead, small studios are characterized by phenomena recognizable as frequency effects (31). And whereas the wavelength of sounds in the low frequency end of the available spectrum is of an even greater order than the dimensions of the room, a large room has dimensions that are large compared to wavelengths of sound. Consequently a small room will have well defined resonances. This can be seen from the predicted and measured modal frequencies.

The physical properties of sound fields in small rooms that will have an influence on the basic requirements for good acoustics, discussed above, can be summarized as follows:

1) The relative importance of different classes of modes is signified by special properties of each type of mode. This point has been dealt with by Mayo (44) who showed that axial modes alone are likely to have sufficient amplitudes and durations and their occurrence in low frequency regions is responsible for such acoustical effects as colourations. At higher frequencies other tangential and oblique modes occur, which, though virtually absent at low frequencies, are far more numerous than the axial modes and their presence reduces the intensity of the axial modes.

   The measurements of shifts in frequencies and damping are evidence of this fact.

2) The effect of the spacing of the modes on the acoustical quality has been widely investigated by Mayo (44), who considered that 'it is the absence of modes rather than their presence, that spoils a small studio'. In his view, a prominent single mode is annoying because it lacks the necessary 'help' of other modes.

The effect of mode spacing on diffusion has been investigated by Bolt (45). Equal spacing is considered favourable as it reproduces the sound faithfully and with time delays that extend the original sound (44). The occurrence of clusters of closely-spaced modes with wide gaps in frequency should be avoided.

3) The damping of individual modes affects both the steady-state amplitudes of sound as well as the decay rate of modes in the transient stage. Hence, equal damping constants will improve the uniformity of the spatial diffusion of sound and contribute to the useful early reverberation.

4) The directional distribution of sound energy in a modal field is affected by the directional pattern of the standing waves excited. In this respect lateral axial modes help in enhancing the lateral reflections and increasing the spaciousness in the room.

5) The spacial distributions of modal fields affects the diffusion of sound in the room. Measurements of modal shapes in a bare-walled and a treated-walled room demonstrate the presence of absorption. The spatial irregularity in small rooms has been investigated by Doak (46) and an index based on the ratio of pressure maxima and minima is introduced by him.

Studios Design and Construction Data

The most important elements in the construction and design of small rooms which have direct implications on the physical properties of sounds reproduced in them should be considered.

The energy spectrum of sound for which the room is used as a studio for broadcasting or recording must first be considered. For speech, the maximum energy is in the neighbourhood of 300-400Hz and the distribution of its fundamentals is limited to the frequency range 124-141Hz(31). It is the coincidence of widely-spaced isolated modes within these ranges that produces colouration in speech.

The choice of the studio size is governed by the function and the number of performers on the one hand and by economical considerations on the other. The effect of the size is to cause audible sounds in the lower frequency spectrum to occur in the form of isolated modes.

The shape of the studio is another factor which comes into consideration. Rectangular shapes are most common, for constructional reasons, but non-rectangular shapes such as triangular or pentagonal prisms have been used.

The dimensional ratio of the sides of a studio is the most important factor that governs the frequency distribution and spacing of the modes. Dimensions that bear simple ratios to one another have specially bad response. A ratio of length to width to height of 1.6:1.3:1 is one of the most favourable in the design of small studios.

The types of sound absorptive material employed in the construction or finishes of the room boundaries influence the frequency distribution of modes, their decay rates and spacial patterns. The complex acoustic impedance describes the physical properties of the material. The magnitudes and the signs of the complex components of acoustic impedance and their variation with frequency affect the way the modes are shifted and their amplitudes damped.

The pattern of distribution of absorptive materials is another factor in controlling modal shaping and spatial variations. Only the effect of uniform coverage of room surfaces has been considered, but the application of patchy asymmetrical arrangements of absorbents has been studied (47) and recommended for the provision of diffusion.

The phenomenon of flutter echoes has been attributed to strong undamped axial modes formed between a pair of untreated surfaces.

The use of projections in wall surfaces for breaking up the standing wave patterns and providing diffusion is rendered of little value in small rooms if consideration is given to the fact that projections are only useful if their size is comparable to the wavelength of sound.

CHAPTER 6

Conclusions and Suggestions for Further Work

Conclusions

In this research work the problem of the acoustics of a small room at low frequency range has been investigated.

In Chapters One and Two the validity of the application of a statistical geometrical approach to the study of sound fields in small rooms has been examined. An alternative method, based on the wave analysis theory, has been proposed.

For experimental investigations, steady-state methods have been chosen in preference to transient-state methods, as the former are considered to describe better the physical properties of sound fields in small rooms.

The wave theory of room acoustics has been presented in Chapter Three. A modal analysis has been formulated in the form of characteristic parameters that can be predicted in terms of the geometry of the room and the properties of its bounding surfaces. The predicted characteristic parameters can be used to evaluate both the steady-state and transient response of the room to acoustic excitation. The Kennedy and Pancu method has been applied to measure experimentally the modal parameters, (i.e. natural frequencies, damping constants and modal shapes). The excitation and response extraction of pure modes and the spatial pressure variation of the sound across the axes of the room were investigated when various boundaries were covered with absorbent materials, and this was compared with the theoretical analysis.

Chapter Four deals specifically with the synthesis. The modal parameters obtained experimentally are applied to the steady-state equation of the room when it is excited by a sinusoidal signal and the results over a frequency range are compared with the experimental results.

From the synthesis analysis it is possible that for any source and receiver position the pressure variation of the sound over a frequency range be predicted by considering the boundary conditions. The synthesis analysis has shown that the physical properties of the boundary influence the degree of coupling and damping to a large extent. The main findings in this research work can be summarized as follows:

1) A modal approach based on the wave theory can be successfully applied in the field of investigating the acoustic properties of small rooms. This results in defined modal parameters which are both predictable and measurable quantities and which describe both the steady-state and transient performance of small rooms.
2) Steady-state measurements, based on resonance testing techniques, are invaluable tools in experimental investigations of room response to acoustic excitation. However, complex response measurements, containing information on phase as well as amplitude, are most useful in this respect.
3) The use of digital computer techniques in room acoustics helps to overcome the complexity of theoretical analysis and their employment in conjunction with digital recording facilitates the analysis of

experimental results and their presentation in tabulated or graphical forms.

The conditions for good acoustics in small studios and the subjective qualities on which they are based, can be explained in terms of physical properties of sound in small rooms as described by the modal parameters. These, in turn, can be interpreted in the form of design and constructional data based on the room geometry and the properties of its bounding surfaces.

Introduction of the absorbing materials to all the surfaces of the room can influence the formation of the standing waves and therefore causes a significant pressure variation along the axes of the room and consequently results in anomalies as far as coupling between the modes is concerned.

Throughout this work, the implementation of digital computing techniques and methods of numerical analysis have been conducted in an effort to overcome the complexity of the mathematics of the wave theory, and in the analysis and presentation of the experimental results.

Suggestions for Further Work

The main discrepancies between theoretical predictions and experimental measurements of the modal parameters describing the room performance have been attributed to errors involved in impedance determination.

Measurements of impedance by the standing wave tube method are only meaningful for locally reacting materials. However, the discrepancies between the theory and measurements of modal parameters suggest the need for an investigation of the possible variation of impedance with angle of incidence of sound.

The scope of this work has been confined to considering rooms with surfaces that have uniform, but different, coverage of absorbing material. The investigation of the performance of rooms with asymmetrical patchy treatment has been studied theoretically (47). An experimental investigation on a parallel basis is a field to be considered for further work.

The variation in sound pressure level near the two opposite walls covered with the absorbing materials is a topic to be investigated.

A 1/4 size scale model of the enclosure is to be constructed and the sound is to be generated outside the room and passed through the room to even out the possible coupling between the loudspeaker and the mode.

The model is to be used to investigate the variation in coupling and damping when different materials are introduced to the surfaces and this is to be compared with the results obtained from the real room.

The scale model is to be used to investigate the variation in transmission loss measurements with various dimensions and surface absorbing materials at low frequency range and the results are to be compared with the real model test: applying the results obtained from the scale model to the real model, the anomalies in transmission loss can then be corrected.

Methods and techniques are to be found to eliminate the difference in coupling between experimental and theoretical work, (e.g. the introduction of patches of absorbents, diffusers).

Experimental work has shown that in the case of the double door of a room, when the outer door is left open, the coupling between the modes perpendicular to the surface of the door is higher than when both doors are shut. Further investigation of this is to be carried out by introducing damping to the door and varying the location of the door.

Theoretical analysis of the synthesis is to be extended for non-rigid surfaces can be taken into account. This analysis is to be extended to find the relationship between reverberation time and bandwidth by considering the decay of modes.

A more accurate technique is to be applied in reverberation time measurements at low frequency.

The H.P. Frequency Spectrometer that the Department is about to buy is to be employed in measurements of modal parameters by phase angle method, since it has a built in Fast Fourier Transformer chip.

Since preliminary investigation has shown that the modal parameters can be measured by generating white noise, other possibilities of the employment of the H.P. Frequency Spectrometer will be explored.

The spatial variation of sound pressure level along the axes ofthe room is to be investigated in detail, when absorbent materials are introduced to the surfaces.

Program to Transfer Amplitude Level from Digital Event Recorder to PET Computer.

```
 5 DIM A%(4096)
 10 POKE 59459,0
 20 GOSUB 100
 30 IF A<>128 THEN 20
 40 FOR I=1TO4096
 50 GOSUB100
 60 A%(I)=A
 70 NEXT
 72 POKE 59468,12
 75 STOP
 80 INPUT "DISC FILE NAME";A$
 81 A$="@0:"+A$+",U,W"
 82 OPEN 15,8,15,"I"
 85 OPEN 1,8,2,A$
 90 FORI=1TO4096:PRINT#1,A%(I);CHR$(13);:PRINTI;"=      1":NEXT:POKE 59468
 95 CLOSE 1,15:STOP
 100 A=PEEK(59457)
 110 POKE 59468,204
 120 POKE 59468,236
 125 PRINTI;"1"
 130 RETURN
READY.
```

```
 80 REM ROOMODES DATE 1-APRIL-1980
 90 DIM FS(100),NS(100,3):NF=0
 95 INPUT"AUTOMATIC";A$:OPEN 1,8,15,"I"
 96 IFA$="Y"THEN125
 100 INPUT"ROOM DIMENSIONS (M)";LX,LY,LZ
 110 INPUT"FREQUENCY LIMITS";FL,FH
 120 INPUT"MODE NUMBER LIMIT";NM
 124 GOTO130
 125 READ LX,LY,LZ,FL,FH,NM
 130 C=343.0:C$=CHR$(13)
 140 FOR I=0TONM
 150 FOR J=0TONM
 160 FOR K=0TONM
 170 F=INT((C/2*SQR((I/LX)^2+(J/LY)^2+(K/LZ)^2))*100+.5)/100
 180 IF F<FL OR F>FHTHEN 250
 190 PRINT I;J;K;F
 195 NF=NF+1:FS(NF)=F
 196 NS(NF,1)=I
 197 NS(NF,2)=J
 198 NS(NF,3)=K
 250 NEXT K,J,I
 260 INPUT "DATA FILE";N$
 265 N$="@0:"+N$+",SEQ,WRITE"
 270 OPEN 2,8,2,N$:PRINT#2,NF;C$;
 300 FORI=1TONF
 310 FORJ=2TONF
 330 IFFS(J)>FS(J-1)THEN 400
 340 FZ=FS(J):FS(J)=FS(J-1):FS(J-1)=FZ
 350 FORK=1TO3:NZ=NS(J,K):NS(J,K)=NS(J-1,K):NS(J-1,K)=NZ:NEXT
 400 NEXTJ,I
 410 FORI=1TONF:PRINTNS(I,1);NS(I,2);NS(I,3);FS(I):NEXT
 420 FORI=1TONF:PRINT#2,NS(I,1)","NS(I,2)","NS(I,3)","FS(I);C$;:NEXT
 450 CLOSE 2:CLOSE 1
 1000 DATA 2.57,3.29,2.28,70,218
READY.
```

```
READY.
 50 REM PROGRAMME ANALYSIS-PRINT TO CALCULATE FREQ. BANDWIDTH AMPL.
 70 DEF FNA(X)=INT(X*100+.5)/100
 100 DIM NS(100,3),FS(100)
 102 INPUT"POSITION";B$
 110 INPUT"MODEFILENAME";A$
 120 A$="0:"+A$+",S,R":C$=CHR$(13)
 130 OPEN 1,8,15,"I":OPEN 2,8,2,A$
 135 OPEN4,4
 140 INPUT#2,NF
 150 FORI=1TONF:INPUT#2,NS(I,1),NS(I,2),NS(I,3),FS(I)
 152 PRINTNS(I,1),NS(I,2),NS(I,3),FS(I)
 156 NEXT
 160 CLOSE 2
 170 INPUT"DATAFILENAME";A$
 180 A$="0:"+A$+",U,R"
 190 DIM A%(4096)
 195 N=4096
 200 OPEN 2,8,2,A$
 210 FOR I=1TON
 220 INPUT#2,A%(I):PRINTI"1"
 230 NEXT:CLOSE 2:CLOSE 1:FF=1
 242 PRINT#4,TAB(25),B$
 250 PRINT"A1=AMPLITUDE NO UNIT"
 252 PRINT#4,"A1=AMPLITUDE"
 260 PRINT"F1=FREQUENCY-HZ"
 262 PRINT#4,"F1=FREQ.-HZ"
 270 PRINT"F=FREQUENCY BAND-HZ"
 272 PRINT#4,"F=FREQ.BAND-HZ"
 275 PRINT#4,SPC(24),"AMP.","FREQ.BAND","FREQUENCY"
 280 Z1=200:Z2=70
 290 D3=LOG(Z1/Z2)/LOG(10)/4096
 300 I=1
 320 I=I+1
 330 IFI=NTHEN00730
 340 G1=A%(I+1)-A%(I)
 350 IFG1>=0GOTO00320
 360 G2=A%(I-1)-A%(I)
 370 IFG2<0THEN00320
 390 IFA%(I)<4  THEN 00320
 400 F1=Z2*10^(I*D3)
 410 J=0
 420 A1=A%(I)/SQR(2)
 430 J=J+1
 440 K1=J+I
 450 IFK1=NGOTO00470
READY.
```

```
READY.
 460 IFA%(K1)<A%(K1+1)GOTO00475
 465 IFA%(K1)>A1THEN00430
 470 D1=A%(K1-1)-A1:D2=A1-A%(K1):D=K1-D2/(D1+D2)
 472 GOTO00480
 475 D1=A%(K1-2)-A%(K1-1):D2=A%(K1-1)-A1:D=(K1-1)+D2/(D1+D2)
 480 F3=INT(Z2*10^(D*D3)*100+.5)/100
 490 J=0
 500 J=J-1
 510 K1=J+I
 520 IFK1<.1GOTO00540
 530 IFA%(K1)<A%(K1-1)GOTO00542
 535 IFA%(K1)>A1GOTO00500
 540 D1=A%(K1+1)-A1:D2=A1-A%(K1):D=K1+D2/(D1+D2)
 541 GOTO00550
 542 D1=A%(K1+2)-A%(K1+1):D2=A%(K1+1)-A1:D=(K1+1)-D2/(D1+D2)
 550 F4=INT(Z2*10^(D*D3)*100+.5)/100
 560 F=(F3-F4)
 590 PRINTA%(I);F;F1
 655 PRINT#4,SPC(25),A%(I),FNA(F),FNA(F1)
 660 FF=FF+1
 680 I=I+1:IFI=NTHEN00730
 690 G1=A%(I+1)-A%(I)
 700 IFG1>0THEN00320
 710 GOTO00680
 720 GO TO510
 730 CLOSE4:STOP
READY.
```

Program Respos that Stores the Theoretical Resonant Frequencies.

```
READY.

12 REM IS=NO.OF FREQ.
14 REMX,Y,Z MIKE POSITION
29 REM SYNTHESIS MARK 4.0 20-01-81
30 REM F=FREQUENCY
40 REM C=COUPLING
50 REM D=DAMPING
60 DIM RF(30)
100 DIM F(50),C(50),D(50),M1(50),M2(50),M3(50),D1(50),W1(50),RC(50),S1(350)
103 READA$:PRINTA$
104 READX,Y,Z:PRINT"MIKE POSITION";X;Y;Z
106 READIS:PRINT"NO.OF PEAK FREQ.";IS
115 LX=2.57:LY=3.29:LZ=2.28
118 P=0:PI=
122 INPUT"NO.OF RES.FREQ.";KR
124 FORI=1TOKR:READRF(I):NEXTI
126 PRINT"  "
130 FORI=1TOIS
140 READ F(I),M1(I),M2(I),M3(I),C(I),D(I)
142 PRINT F(I);M1(I);M2(I);M3(I);C(I);D(I)
144 D1(I)=D(I)*
150 C(I)=C(I)*2*D1(I)
152 RC(I)=COS(M1(I)*PI*X/LX)*COS(M2(I)*PI*Y/LY)*COS(M3(I)*PI*Z/LZ)
160 NEXT
180 OPEN1,8,15,"I0"
184 INPUT"FILE RES.FREQ.POS.";B$
188 B$="à0:"+B$+",U,W"
190 OPEN2,8,2,B$
194 FORI=1TOKR
196 PRINT£2,RF(I);CHR$(13);
198 PRINTSPC(10);I;"1="
200 NEXT
202 CLOSE2:CLOSE1
210 INPUT"END OF DATA";G$
212 IFLEFT$(G$,1)="Y"THEN220
224 G1=LOG(70)/LOG(10)
226 G2=LOG(202)/LOG(10)
228 G3=(G2-G1)/256
300 F3=70
301 N=0
303 A$="à0:"+A$+",U,W"
307 FORF1=G1TOG2 STEPG3
308 N=N+1
309 F=10^F1
310 S2=0:T2=0

READY.
```

```
310 S2=0:T2=0
311 PRINTN;" 1="
312 W=2*PI*F
320 FORJ=1TOIS
330 W2=2*PI*F(J)
332 PRINTSPC(18);J;"  1="
340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
360 Z3=C(J)*W*2*W2*D1(J)/Z1
370 S2=S2+Z2*RC(J)
390 T2=T2+Z3*RC(J)
410 NEXTJ
420 S1(N)=SQR(S2*S2+T2*T2)
430 NEXT
440 OPEN1,8,15,"I0"
445 OPEN2,8,2,A$
450 FORI=1TON
470 PRINT£2,S1(I);CHR$(13);
472 PRINTSPC(20);I;"   1="
480 NEXT
490 CLOSE2:CLOSE1
900 DATA"SYNTH-POS1",1.285,0,0,23
910 DATA75.22,84.68,91.52,100.55,104.26,113.26,123.78,128.56,133.46
915 DATA143.28,144.85,150.44,153.2,156.38,159.21
920 DATA 161.83,164.57,169.36,170.03,172.63,173.53,183.03,185.31
930 DATA185.92,194.82,200.19,201.11
1010 DATA76.73,0,0,1,43,1.76
1020 DATA92.54,0,1,1,99,1.75
1030 DATA98.8,0,1,1,12,1.01
1040 DATA106.31,0,2,0,64,1.25
1045 DATA107.61,0,2,0,29,2.34
1050 DATA116.14,0,2,0,17,5.35
1060 DATA120.02,0,2,0,12,.96
1080 DATA120.18,0,2,0,12,.21
1090 DATA120.58,0,2,0,12,.1
2000 DATA121.71,0,2,0,12,1.76
2020 DATA123.52,0,2,1,9,1.44
2030 DATA124.48,0,2,1,7,1.07
2040 DATA130.89,0,2,1,65,3.66
2060 DATA133.84,2,0,0,60,2.5
2070 DATA143.82,2,1,0,82,1.9
2080 DATA145.54,2,1,0,17,1.02
2090 DATA151.1,0,0, 2,33,2.37
3000 DATA154.42,2,0,1,36,3.61
3010 DATA162.84,2,1,1,96,3.46
3020 DATA171.09,2,2,0,74,2.26
3030 DATA175.57,0,3,1,51,4.29
3040 DATA186.33,0,2,2,72,2.77
3050 DATA201.32,2,0,2,50,1.74
READY.
```

Computer Program to Calculate Impedance.

```
 5 REM THE FIRST MODE IS THE DISTANCE OBTAINED FROM EXP. IN CM.
 7 REMTHE SECOND MODE IS THE FREQ. AT WHICH THE IMP. IS CALCULATED
 10 INPUT "SWR";SWR
 15 SWR=100/SWR
 20 INPUT "FIRST AND SECOND NODE";NO,N1
 25 N1=(343/(N1))
 30 THETA=--(4**(-(NO/N1)))
 40 B1=(SWR-1)/(SWR+1)
 45 PRINTB1
 50 X=B1*COS(THETA):Y=B1*SIN(THETA)
 60 P=(((1-X)*(1+X))+((-Y)*(+Y)))/(((1-X)^2)+((-Y)^2))
 70 Q=(-((-Y)*(1+X))+((Y)*(1-X)))/(((1-X)^2)+((-Y)^2))
 80 PRINTP,Q
 100 GOTO10
READY.
```

Synthesis Program for Mic Position lx, ly, lz.

```
 12 REM IS=NO.OF FREQ.
 14 REMX,Y,Z MIKE POSITION
 29 REM SYNTHESIS MARK 4.0 20-01-81
 30 REM F=FREQUENCY
 40 REM C=COUPLING
 50 REM D=DAMPING
 100 DIM F(50),C(50),D(50),M1(50),M2(50),M3(50),D1(50),W1(50),RC(50),S1(350)
 103 READA$:PRINTA$
 104 READX,Y,Z:PRINT"MIKE POSITION";X;Y;Z
 106 READIS:PRINT"NO.OF PEAK FREQ.";IS
 115 LX=2.57:LY=3.29:LZ=2.28
 118 P=0
 120 PI=
 130 FORI=1TOIS
 140 READ F(I),M1(I),M2(I),M3(I),C(I),D(I)
 141 PRINT F(I);M1(I);M2(I);M3(I);C(I);D(I)
 144 D1(I)=D(I)*
 150 C(I)=C(I)*2*D1(I)
 152 RC(I)=COS(M1(I)*PI*X/LX)*COS(M2(I)*PI*Y/LY)*COS(M3(I)*PI*Z/LZ)
 160 NEXT
 224 G1=LOG(70)/LOG(10)
 226 G2=LOG(200)/LOG(10)
 228 G3=(G2-G1)/256
 300 F3=70
 301 N=0
 303 A$="@0:"+A$+",J,W"
 307 FORF1=G1TOG2 STEPG3
 308 N=N+1
 309 F=10^F1
 310 S2=0:T2=0
 311 PRINTN;" 1="
 312 W=2*PI*F
 320 FORJ=1TOIS
 330 W2=2*PI*F(J)
 332 PRINTSPC(10);J;"  1="
 340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
 350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
 360 Z3=C(J)*W*2*W2*D1(J)/Z1
 370 S2=S2+Z2*RC(J)
 390 T2=T2+Z3*RC(J)
 410 NEXTJ
 420 S1(N)=SQR(S2*S2+T2*T2)
 430 NEXT
 432 INPUT"DISK OK";Y$
 434 IFLEFT$(Y$,1)<>"Y"THEN00432
 440 OPEN1,8,15,"I0"
 445 OPEN2,8,2,A$
 450 FORI=1TON
 470 PRINT#2,S1(I);CHR$(13);
 472 PRINTSPC(20);I;"    1="
 480 NEXT
 490 CLOSE2:CLOSE1
READY.
```

Synthesis Program for Mic Position 1x, 1y, 1z.

```
 1000 DATA"SYNTH-POS6",2.57,3.29,2.28,20
 1010 DATA75.30,0,0,1,56,1.92
 1020 DATA83.37,1,1,0,61,2.67
 1030 DATA90.57,0,1,1,98,1.59
 1040 DATA99.52,1,0,1,53,2.45
 1045 DATA103.96,0,2,0,81,1.09
 1050 DATA111.89,1,1,1,77,4.17
 1060 DATA123.12,1,2,0,84,1.45
 1080 DATA128.17,0,2,1,46,4.37
 1090 DATA135.36,2,0,0,7.0,2.31
 2000 DATA141.83,2,1,0,65,2.62
 2020 DATA148.56,0,0,2,33,2.10
 2030 DATA152.26,2,0,1,28,4.04
 2040 DATA155.42,0,3,0,28,2.16
 2060 DATA158.27,0,1,2,50,2.42
 2070 DATA159.29,2,1,1,49,1.82
 2080 DATA167.67,2,2,0,67,2.78
 2090 DATA173.66,0,3,1,72,4.17
 3000 DATA184.20,0,2,2,88,4.23
 3010 DATA194.19,1,2,2,31,2.88
 3020 DATA198.57,3,0,0,21,2.15
READY.
```

Program Synth Disk 6 to Calculate the Coupling and Bandwidth of the Modes Theoretically.

```
12 REM IS=NO.OF FREQ.
14 REMX,Y,Z MIKE POSITION
29 REM SYNTHESIS MARK 4.0 20-01-81
30 REM F=FREQUENCY
40 REM C=COUPLING
50 REM D=DAMPING
60 DIM RF(30)
100 DIM F(50),C(50),D(50),M1(50),M2(50),M3(50),D1(50),W1(50),RC(50),S1(350)
103 READA$:PRINTA$
104 READX,Y,Z:PRINT"MIKE POSITION";X;Y;Z
106 READIS:PRINT"NO.OF PEAK FREQ.";IS
115 LX=2.57:LY=3.29:LZ=2.28
118 P=0:PI=
122 INPUT"NO.OF RES.FREQ.";KR
124 FORI=1TOKR:READRF(I):NEXTI
126 PRINT"  "
130 FORI=1TOIS
140 READ F(I),M1(I),M2(I),M3(I),C(I),D(I)
142 PRINT F(I);M1(I);M2(I);M3(I);C(I);D(I)
144 D1(I)=D(I)*
150 C(I)=C(I)*2*D1(I)
152 RC(I)=COS(M1(I)*PI*X/LX)*COS(M2(I)*PI*Y/LY)*COS(M3(I)*PI*Z/LZ)
160 NEXT
180 OPEN1,8,15,"I0"
184 INPUT"FILE RES.FREQ.POS.";B$
188 B$="@0:"+B$+",U,W"
190 OPEN2,8,2,B$
194 FORI=1TOKR
196 PRINT#2,RF(I);CHR$(13);
198 PRINTSPC(10):I;"1="
200 NEXT
202 CLOSE2:CLOSE1
210 INPUT"END OF DATA";G$
212 IFLEFT$(G$,1)="Y"THEN220
224 G1=LOG(70)/LOG(10)
226 G2=LOG(202)/LOG(10)
228 G3=(G2-G1)/256
300 F3=70
301 N=0
303 A$="@0:"+A$+",U,W"
307 FORF1=G1TOG2 STEPG3
308 N=N+1
309 F=10^F1
310 S2=0:T2=0
311 PRINTN;" 1="
312 W=2*PI*F
320 FORJ=1TOIS
330 W2=2*PI*F(J)
332 PRINTSPC(18);J;"  1="
340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
360 Z3=C(J)*W*2*W2*D1(J)/Z1
370 S2=S2+Z2*RC(J)
390 T2=T2+Z3*RC(J)
410 NEXTJ
READY.
```

```
1000 DATA"SYNTH-POS1",1.285,0,0,9
1010 DATA73.44,0,0,1,3.08,2.6
1020 DATA87.22,0,1,1,4.56,3.73
1030 DATA101.9,0,2,0,2.93,3.36
1040 DATA125.49,0,2,1,3,8.69
1050 DATA129.15,2,0,0,3,8.69
1060 DATA137.49,2,1,0,2.85,5.36
1070 DATA148.84,0,0,2,1.33,12.96
1080 DATA156,0,1,2,1.48,5.58
1090 DATA128.88,2,0,0,1.91,7.44
2000 DATA"SYNTH-POS2",2.57,0,0,9
2010 DATA73.96,0,0,1,1.99,26.4
2020 DATA81.39,1,1,0,3.82,4.29
2030 DATA87.15,0,1,0,3.32,11.06
2040 DATA95.98,1,0,1,2.5,25.55
2050 DATA108.29,0,2,0,3.63,5.68
2060 DATA121.93,1,2,0,2.85,4.57
2070 DATA130.59,2,0,0,1.48,30.25
2080 DATA147,2,1,0,2.5,17.81
2090 DATA156.17,0,3,0,3,15.5
3000 DATA"SYNTH-POS3",2.57,1.645,0,10
3010 DATA72.69,0,0,1,3.63,2.64
3020 DATA95.73,0,0,1,3.94,5.3
3030 DATA101.72,0,2,0,4.52,3.62
3040 DATA119.5,1,2,0,2.77,6.59
3050 DATA130.03,0,2,1,2.5,7.6
3060 DATA138.17,1,2,1,3.2,6.36
3070 DATA149.99,0,0,2,2.89,21.05
3080 DATA156.58,2,0,1,2.61,33.18
3090 DATA164.76,1,0,2,2.22,43.5
3100 DATA181.38,0,2,2,2.57,11.16
READY.
```

```
 4000 DATA"SYNTH-POS4",2.57,3.29,0,8
 4010 DATA73.13,0,0,1,3.35,3.81
 4020 DATA82.49,1,1,0,4.64,3.89
 4030 DATA87.03,0,1,1,4.25,4.08
 4040 DATA102.01,1,0,1,4.21,7.3
 4050 DATA109.88,0,2,0,4.17,5.34
 4060 DATA140.55,2,1,0,3.98,8.89
 4070 DATA166.25,2,2,0,3.82,13.7
 4080 DATA184.78,0,2,2,3,11.56
 5000 DATA"SYNTH-POS5",2.57,3.29,1.14,8
 5010 DATA56.55,1,1,0,1.17,21.24
 5020 DATA101.5,0,2,0,3.16,3.88
 5030 DATA120.11,1,2,0,2.96,4.94
 5040 DATA137.82,2,0,0,1.64,6.35
 5050 DATA142.78,2,1,0,1.01,72.69
 5060 DATA153.65,0,0,2,1.99,14.6
 5070 DATA156.38,0,3,0,1.91,15.16
 5080 DATA166.04,2,2,0,1.25,71.02
 6000 DATA"SYNTH-POS6",2.57,3.29,2.28,13
 6010 DATA75.09,0,0,1,3.24,10.28
 6020 DATA81.06,1,1,0,4.02,9.94
 6030 DATA87.3,0,1,1,3.74,3.4
 6040 DATA97.35,1,0,1,3.2,15.15
 6050 DATA103.53,0,2,0,4.17,8.13
 6060 DATA106.09,1,1,1,4.02,8.35
 6070 DATA121.7,1,2,0,2.54,11.38
 6080 DATA125.11,0,2,1,2.38,12.49
 6090 DATA128.88,2,0,0,1.91,14.89
 6110 DATA152.71,0,0,2,1.05,8.09
 6120 DATA167.47,2,2,0,3.24,11.93
 6130 DATA168.58,1,3,0,3.2,12.34
 6140 DATA187.77,1,3,1,1.09,35.11
READY.
```

Program to plot the Frequency Response Curve Theoretically (Results obtained from the Synthesis Program).

```
 100 REM PLOT PROGRAM
 105 FL=70:FH=202·L=(LOG(FH/FL))/4096
 110 DIMA%(4096),F3(60)
 115 OPEN1,8,15,"I0"
 120 INPUT"FILENAME";Z$
 125 IF LEFT$(Z$,2)="SY"THEN135
 130 Q$=" ":Y1=6000
 131 GOTO140
 135 Q$="(THEORETICAL)":Y1=5700
 140 OPEN2,8,2,Z$+",U,R"
 150 GOSUB20000
 155 NR=512:MP=1:IFY1=5700THENNR=256:MP=2
 160 FORI=1TONR
 180 INPUT#2,A%(I)
 182 PRINTA%(I)
 200 PRINT"3               ==========";I
 220 NEXT I
 222 CLOSE2
 223 INPUT"NO.OF FREQ.";K
 224 INPUT"FILE RES.FREQ.";M$
 226 OPEN2,8,2,M$+",U,R"
 228 FORI=1TOK
 230 INPUT#2,F3(I)
 232 PRINTF3(I);CHR$(13);
 234 NEXT
 240 CLOSE 2,1
 260 INPUT"PLOTTER OK";Y$
 280 IFLEFT$(Y$,1)<>"Y"THEN260
 284 INPUT"PEN CHANGE FOR RES.FREQ.";H$
 286 IFLEFT$(H$,1)<>"Y"THEN00284
 300 OPEN128,5:PRINT#128,"IN"
 305 GOTO00325
 310 GOSUB10000:REM PLOT AXES
 320 GOSUB11000:REM NAME AXES
 325 GOSUB14850:REM MRAK RES.FREQ.
 327 INPUT"PEN CHANGE";E$
 328 IFLEFT$(E$,1)<>"Y"THEN00327
 330 GOSUB30000
 400 PRINT#128,"PU;PA980,980"
 410 FORI=2TONR
 420 X=960+16*I*MP
 440 Y=1000+A%(I)*20
 445 IFY>5800THEN00480
 460 PRINT#128,"PA";STR$(X);",";STR$(Y)
 470 PRINT#128,"PD"
 480 NEXT
 500 PRINT#128,"PU;PA1000,1000"
 600 PRINT#128,"IN":CLOSE5
 3030 END
 9999 END
READY.
```

Program to Plot the Frequency Response Curve Theoretically.

```
10000 PRINT#128,"IN"
10100 PRINT#128,"PA1000,1000"
10200 PRINT#128,"PD;PA10000,1000"
10300 PRINT#128,"PU;PA1000,1000"
10400 PRINT#128,"PD;PA1000,6000"
10500 PRINT#128,"PU;PA1000,1000"
10850 RETURN
11000 PRINT#128,"PU;PA4500,500"
11100 PRINT#128,"LBFREQUENCY (HZ)";CHR$(3)
11200 PRINT#128,"PU;PA400,1500"
11300 PRINT#128,"DI0,1;LBAMPLITUDE (VOLTS)";CHR$(3)
11400 PRINT#128,"DI1,0;PA1000,1000"
11450 PRINT#128,"SI0.12,0.18"
11500 FORI=0TO130STEP10
11600 Y=1000+I*20
11700 PRINT#128,"PA1000,";STR$(Y);";YT"
11720 IF I/20<>INT(I/20)THEN11800
11740 V=INT(I*0.039*100+.05)/100
11760 PRINT#128,"PA600,";STR$(Y-30)
11780 PRINT#128,"LB";V;CHR$(3)
11800 NEXT
11850 X=0
11900 FORI=70TO205STEP5
12000 X=INT(2*LOG(I/FL)/L+0.5)+1000
12100 PRINT#128,"PA";STR$(X);",1000;XT"
12200 Z=85:IFX> 95THENZ=140
12210 IF I<105THEN12300
12220 IF I/20<>INT(I/20)THEN12600
12300 PRINT#128,"PA";STR$(X-Z);",800"
12400 PRINT#128,"LB";I;CHR$(3)
12600 NEXT
12700 RETURN
14850 PRINT#128,"PU;PA1000,1000"
15000 FORI=1TOK
16400 X=INT(2*LOG(F3(I)/FL)/L+.5)+1000
16500 PRINT#128,"PA";STR$(X);",900;TL5   ;XT"
17000 NEXTI
18050 RETURN
20000 INPUT#1,A,A$,B$,C$
20100 IFA$="OK"THENRETURN
20200 PRINT"ERROR";A;A$
20300 STOP
30000 PRINT#128,"PU;PA2000,";STR$(Y1)
30005 PRINT#128,"SI0.2,0.3"
30010 PRINT#128,"LB"Q$
30015 PRINT#128,R$;CHR$(3)
30020 RETURN
READY.
```

```
 12 REM IS=NO.OF FREQ.
 14 REMX,Y,Z MIKE POSITION
 29 REM SYNTHESIS MARK 4.0 20-01-81
 30 REM F=FREQUENCY
 40 REM C=COUPLING
 50 REM D=DAMPING
 100 DIM F(50),C(50),D(50),M1(50),M2(50),M3(50),D1(50),W1(50),RC(50),S1(350)
 103 READA$:PRINTA$
 104 READX,Y,Z:PRINT"MIKE POSITION";X;Y;Z
 106 READIS:PRINT"NO.OF PEAK FREQ.";IS
 115 LX=2.57:LY=3.29:LZ=2.28
 118 P=0
 120 PI=
 130 FORI=1TOIS
 140 READ F(I),M1(I),M2(I),M3(I),C(I),D(I)
 141 PRINT F(I);M1(I);M2(I);M3(I);C(I);D(I)
 144 D1(I)=D(I)*
 150 C(I)=C(I)*2*D1(I)
 152 RC(I)=COS(M1(I)*PI*X/LX)*COS(M2(I)*PI*Y/LY)*COS(M3(I)*PI*Z/LZ)
 160 NEXT
 224 G1=LOG(70)/LOG(10)
 226 G2=LOG(200)/LOG(10)
 228 G3=(G2-G1)/256
 300 F3=70
 301 N=0
 303 A$="@0:"+A$+",U,W"
 307 FORF1=G1TOG2 STEPG3
 308 N=N+1
 309 F=10^F1
 310 S2=0:T2=0
 311 PRINTN;" 1="
 312 W=2*PI*F
 320 FORJ=1TOIS
 330 W2=2*PI*F(J)
 332 PRINTSPC(10);J;"  1="
 340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
 350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
 360 Z3=C(J)*W*2*W2*D1(J)/Z1
 370 S2=S2+Z2*RC(J)
 390 T2=T2+Z3*RC(J)
 410 NEXTJ
 420 S1(N)=SQR(S2*S2+T2*T2)
 430 NEXT
 432 INPUT"DISK OK";Y$
 434 IFLEFT$(Y$,1)<>"Y"THEN00432
 440 OPEN1,8,15,"I0"
 445 OPEN2,8,2,A$
 450 FORI=1TON
 470 PRINT#2,S1(I);CHR$(13);
 472 PRINTSPC(20);I;"   1="
 480 NEXT
 490 CLOSE2:CLOSE1
READY.
```

```
1000 DATA"SYNTH-POS4",2.57,3.29,0,21
1010 DATA75.15,0,0,1,48,1.90
1020 DATA84.58,1,1,0,64,2.72
1030 DATA91.17,0,1,1,94,1.90
1040 DATA100.70,1,0,1,54,3.33
1045 DATA104.12,0,2,0,56,2.67
1050 DATA113.22,1,1,1,81,3.27
1060 DATA124.13,1,2,0,77,1.65
1080 DATA128.70,0,2,1,35,3.81
1090 DATA133.26,2,0,0,50,1.75
2000 DATA142.37,2,1,0,68,2.55
2020 DATA145.14,1,2,1,71,3.37
2030 DATA155.38,0,3,0,27,3.49
2040 DATA158.72,0,1,2,34,2.79
2050 DATA162.09,2,1,1,82,2.62
2060 DATA164.81,1,0,2,72,3.34
2070 DATA169.00,2,2,0,79,3.58
2080 DATA172.37,1,1,2,32,2.14
2090 DATA183.17,0,2,2,45,2.39
3000 DATA186.82,2,2,1,115,3.14
3010 DATA194.44,1,2,2,40,1.94
3020 DATA199.23,3,0,0,15,2.91
READY.
```

```
READY.
 100 REM PLOT PROGRAM
 105 FL=70:FH=202:L=(LOG(FH/FL))/4096
 110 DIMA%(4096),F3(60)
 115 OPEN1,8,15,"I0"
 120 INPUT"FILENAME";Z$
 125 IF LEFT$(Z$,2)="SY"THEN135
 130 Q$="POSITION "+MID$(Z$,7,1)+" (EXPERIMENTAL)":Y1=6000
 131 GOTO140
 135 Q$="POSITION "+MID$(Z$,10,1)+" (THEORETICAL)":Y1=5700
 140 OPEN2,8,2,Z$+",U,R"
 150 GOSUB20000
 155 NR=512:MP=1:IFY1=5700THENNR=256:MP=2
 160 FORI=1TONR
 180 INPUT#2,A%(I)
 182 PRINTA%(I)
 200 PRINT"3           ==========";I
 220 NEXT I
 222 CLOSE2
 223 INPUT"NO.OF FREQ.";K
 224 INPUT"FILE RES.FREQ.";M$
 226 OPEN2,8,2,M$+",U,R"
 228 FORI=1TOK
 230 INPUT#2,F3(I)
 232 PRINTF3(I);CHR$(13);
 234 NEXT
 240 CLOSE 2,1
 260 INPUT"PLOTTER OK";Y$
 280 IFLEFT$(Y$,1)<>"Y"THEN260
 284 INPUT"PEN CHANGE FOR RES.FREQ.";H$
 286 IFLEFT$(H$,1)<>"Y"THEN00284
 300 OPEN5,5:PRINT#5,"IN"
 305 GOTO00325
 310 GOSUB10000:REM PLOT AXES
 320 GOSUB11000:REM NAME AXES
 325 GOSUB14850:REM MRAK RES.FREQ.
 327 INPUT"PEN CHANGE";E$
 328 IFLEFT$(E$,1)<>"Y"THEN00327
 330 GOSUB30000
READY.
```

```
310 S2=0:T2=0
311 PRINTN;" 1="
312 W=2*PI*F
320 FORJ=1TOIS
330 W2=2*PI*F(J)
332 PRINTSPC(18);J;"  1="
340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
360 Z3=C(J)*W*2*W2*D1(J)/Z1
370 S2=S2+Z2*RC(J)
390 T2=T2+Z3*RC(J)
410 NEXTJ
420 S1(N)=SQR(S2*S2+T2*T2)
430 NEXT
440 OPEN1,8,15,"I0"
445 OPEN2,8,2,A$
450 FORI=1TON
470 PRINT£2,S1(I);CHR$(13);
472 PRINTSPC(20);I;"    1="
480 NEXT
490 CLOSE2:CLOSE1
900 DATA"SYNTH-POS1",1.285,0,0,23
910 DATA75.22,84.68,91.52,100.55,104.26,113.26,123.78,128.56,133.46
915 DATA143.28,144.85,150.44,153.2,156.38,159.21
920 DATA 161.83,164.57,169.36,170.03,172.63,173.53,183.03,185.31
930 DATA185.92,194.82,200.19,201.11
1010 DATA76.73,0,0,1,43,1.76
1020 DATA92.54,0,1,1,99,1.75
1030 DATA98.8,0,1,1,12,1.01
1040 DATA106.31,0,2,0,64,1.25
1045 DATA107.61,0,2,0,29,2.34
1050 DATA116.14,0,2,0,17,5.35
1060 DATA120.02,0,2,0,12,.96
1080 DATA120.18,0,2,0,12,.21
1090 DATA120.58,0,2,0,12,.1
2000 DATA121.71,0,2,0,12,1.76
2020 DATA123.52,0,2,1,9,1.44
2030 DATA124.48,0,2,1,7,1.07
2040 DATA130.89,0,2,1,65,3.66
2060 DATA133.84,2,0,0,60,2.5
2070 DATA143.82,2,1,0,82,1.9
2080 DATA145.54,2,1,0,17,1.02
2090 DATA151.1,0,0, 2,33,2.37
3000 DATA154.42,2,0,1,36,3.61
3010 DATA162.84,2,1,1,96,3.46
3020 DATA171.09,2,2,0,74,2.26
3030 DATA175.57,0,3,1,51,4.29
3040 DATA186.33,0,2,2,72,2.77
3050 DATA201.32,2,0,2,50,1.74
READY.
```

```
 100 REM PLOT PROGRAM
 105 FL=70:FH=202:L=(LOG(FH/FL))/4096
 110 DIMA%(4096),F3(60)
 115 OPEN1,8,15,"I0"
 120 INPUT"FILENAME";Z$
 125 IF LEFT$(Z$,2)="SY"THEN135
 130 Q$="POSITION "+MID$(Z$,7,1)+" (EXPERIMENTAL)":Y1=6000
 131 GOTO140
 135 Q$="POSITION "+MID$(Z$,10,1)+" (THEORETICAL)":Y1=5700
 140 OPEN2,8,2,Z$+",U,R"
 150 GOSUB20000
 155 NR=512:MP=1:IFY1=5700THENNR=256:MP=2
 160 FORI=1TONR
 180 INPUT#2,A%(I)
 182 PRINTA%(I)
 200 PRINT"3           ==========";I
 220 NEXT I
 222 CLOSE2
 223 INPUT"NO.OF FREQ.";K
 224 INPUT"FILE RES.FREQ.";M$
 226 OPEN2,8,2,M$+",U,R"
 228 FORI=1TOK
 230 INPUT#2,F3(I)
 232 PRINTF3(I);CHR$(13);
 234 NEXT
 240 CLOSE 2,1
 260 INPUT"PLOTTER OK";Y$
 280 IFLEFT$(Y$,1)<>"Y"THEN260
 284 INPUT"PEN CHANGE FOR RES.FREQ.";H$
 286 IFLEFT$(H$,1)<>"Y"THEN00284
 300 OPEN5,5:PRINT#5,"IN"
 305 GOTO00325
 310 GOSUB10000:REM PLOT AXES
 320 GOSUB11000:REM NAME AXES
 325 GOSUB14850:REM MRAK RES.FREQ.
 327 INPUT"PEN CHANGE";E$
 328 IFLEFT$(E$,1)<>"Y"THEN00327
 330 GOSUB30000
 400 PRINT#5,"PU;PA980,980"
 410 FORI=2TONR
 420 X=960+16*I*MP
 440 Y=1000+A%(I)*20
 445 IFY>5800THEN00480
 450 PRINT#5,"PA";STR$(X);",";STR$(Y)
 470 PRINT#5,"PD"
 480 NEXT
 500 PRINT#5,"PU;PA1000,1000"
 600 PRINT#5,"IN":CLOSE5
 9999 END
READY.
```

```
10000 PRINT#5,"IN"
10100 PRINT#5,"PA1000,1000"
10200 PRINT#5,"PD;PA10000,1000"
10300 PRINT#5,"PU;PA1000,1000"
10400 PRINT#5,"PD;PA1000,6000"
10500 PRINT#5,"PU;PA1000,1000"
10850 RETURN
11000 PRINT#5,"PU;PA4500,500"
11100 PRINT#5,"LBFREQUENCY (HZ)";CHR$(3)
11200 PRINT#5,"PU;PA400,1500"
11300 PRINT#5,"DI0,1;LBAMPLITUDE (VOLTS)";CHR$(3)
11400 PRINT#5,"DI1,0;PA1000,1000"
11450 PRINT#5,"SI0.12,0.18"
11500 FORI=0TO130STEP10
11600 Y=1000+I*20
11700 PRINT#5,"PA1000,";STR$(Y);";YT"
11720 IF I/20<>INT(I/20)THEN11800
11740 V=INT(I*0.039*100+.05)/100
11760 PRINT#5,"PA600,";STR$(Y-30)
11780 PRINT#5,"LB";V;CHR$(3)
11800 NEXT
11850 X=0
11900 FORI=70TO205STEP5
12000 X=INT(2*LOG(I/FL)/L+0.5)+1000
12100 PRINT#5,"PA";STR$(X);",1000;XT"
12200 Z=85:IFX> 95THENZ=140
12210 IF I<105THEN12300
12220 IF I/20<>INT(I/20)THEN12600
12300 PRINT#5,"PA";STR$(X-Z);",800"
12400 PRINT#5,"LB";I;CHR$(3)
12600 NEXT
12700 RETURN
14850 PRINT#5,"PU;PA1000,1000"
15000 FORI=1TOK
16400 X=INT(2*LOG(F3(I)/FL)/L+.5)+1000
16500 PRINT#5,"PA";STR$(X);",900;TL5  ;XT"
17000 NEXTI
18050 RETURN
20000 INPUT#1,A,A$,B$,C$
20100 IFA$="OK"THENRETURN
20200 PRINT"ERROR";A;A$
20300 STOP
30000 PRINT#5,"PU;PA2000,";STR$(Y1)
30005 PRINT#5,"SI0.2,0.3"
30010 PRINT#5,"LB"Q$
30015 PRINT#5,R$;CHR$(3)
30020 RETURN
READY.
```

```
100 REM PLOT PROGRAM TO LOOK AT THE DISTRIBUTION OF MODES
105 FL=70:FH=202:L=(LOG(FH/FL))/4096
110 DIMA%(4096),F3(60),RC(50),M1(50),M2(50),M3(50)
115 OPEN1,8,15,"I0"
117 GOTO00223
120 INPUT"FILENAME";Z$
125 IF LEFT$(Z$,2)="SY"THEN135
130 Q$=" ":Y1=6000
131 GOTO140
135 Q$="(THEORETICAL)":Y1=5700
140 OPEN2,8,2,Z$+",U,R"
150 GOSUB20000
155 NR=512:MP=1:IFY1=5700THENNR=256:MP=2
160 FORI=1TONR
180 INPUT#2,A%(I)
200 PRINT"3              ==========";I
220 NEXT I
222 CLOSE2
223 INPUT"NO.OF FREQ.";K
224 INPUT"FILE RES.FREQ.";M$
226 OPEN2,8,2,M$+",U,R"
228 FORI=1TOK
230 INPUT#2,F3(I),RC(I),M1(I),M2(I),M3(I)
232 PRINTF3(I),RC(I),M1(I),M2(I),M3(I);CHR$(13);
234 NEXT
240 CLOSE 2,1
260 INPUT"PLOTTER OK";Y$
280 IFLEFT$(Y$,1)<>"Y"THEN260
284 INPUT"PEN CHANGE FOR RES.FREQ.";H$
286 IFLEFT$(H$,1)<>"Y"THEN00284
300 OPEN128,5:PRINT#128,"IN"
310 GOSUB10000:REM PLOT AXES
320 GOSUB11000:REM NAME AXES
322 INPUT"PEN CHANGE";H1$
323 IFLEFT$(H1$,1)<>"Y"THEN00322
325 GOSUB14850:REM MRAK RES.FREQ.
327 INPUT"PEN CHANGE";E$
328 IFLEFT$(E$,1)<>"Y"THEN00327
330 GOSUB30000
400 PRINT#128,"PU;PA980,1480"
410 FORI=2TONR
420 X=960+16*I*MP
440 Y=1500+A%(I)*20
445 IFY>5800THEN00480
460 PRINT#128,"PA";STR$(X);",";STR$(Y)
470 PRINT#128,"PD"
480 NEXT
500 PRINT#128,"PU;PA1000,1500"
600 PRINT#128,"IN":CLOSE5
9999 END
READY.
```

```
10000 PRINT#128,"IN"
10100 PRINT#128,"PA1000,1500"
10200 PRINT#128,"PD;PA10000,1500"
10300 PRINT#128,"PU;PA1000,1500"
10400 PRINT#128,"PD;PA1000,6500"
10500 PRINT#128,"PU;PA1000,1500"
10850 RETURN
11000 PRINT#128,"PU;PA9500,1650"
11010 PRINT#128,"SI.2,.2"
11100 PRINT#128,"LBFREQUENCY (HZ)";CHR$(3)
11200 PRINT#128,"PU;PA400,1500"
11300 PRINT#128,"DI0,1;LBAMPLITUDE (VOLTS)";CHR$(3)
11400 PRINT#128,"DI1,0;PA1000,1500"
11450 PRINT#128,"SI0.12,0.18"
11500 FORI=0TO130STEP10
11600 Y=1500+I*20
11700 PRINT#128,"PA1000,";STR$(Y);";YT"
11720 IF I/20<>INT(I/20)THEN11800
11740 V=INT(I*0.039*100+.05)/100
11760 PRINT#128,"PA600,";STR$(Y-30)
11780 PRINT#128,"LB";V;CHR$(3)
11800 NEXT
11850 X=0
11900 FORI=70TO205STEP5
12000 X=INT(2*LOG(I/FL)/L+0.5)+1000
12100 PRINT#128,"PA";STR$(X);",1500;XT"
12200 Z=85:IFX> 95THENZ=140
12210 IF I<105THEN12300
12220 IF I/20<>INT(I/20)THEN12600
12300 PRINT#128,"PA";STR$(X-Z);",1300"
12400 PRINT#128,"LB";I;CHR$(3)
12600 NEXT
12700 RETURN
14850 PRINT#128,"PU;PA1000,1500"
15000 FORI=1TOK
15010 P2 =RC(I)*20 +1400
16400 X=INT(2*LOG(P3(I)/FL)/L+.5)+1000
16500 PRINT#128,"PA";STR$(X);",1350;TL3 ;XT"
16506 PRINT#128,"PA";STR$(X-20);",180"
16507 PRINT#128,"DI0,1"
16508 PRINT#128,"SI.18,.18"
16509 PRINT#128,"LB";M1(I);M2(I);M3(I);CHR$(3)
16510 PRINT#128,"DI1,0"
16520 PRINT#128,"PU;PA";STR$(X);",";STR$(P2)
16530 PRINT#128,"PD"
16540 PRINT#128,"PU"
17000 NEXTI
18050 RETURN
20000 INPUT#1,A,A$,B$,C$
20100 IFA$="OK"THENRETURN
20200 PRINT"ERROR";A;A$
20300 STOP
30000 PRINT#128,"PU;PA2000,";STR$(Y1)
30005 PRINT#128,"SI0.3,0.3"
30010 PRINT#128,"LB"Q$
30015 PRINT#128,R$;CHR$(3)
30020 RETURN
READY.
```

Program Synth Disk 6 to Calculate the Coupling and Bandwidth of the Modes Theoretically.

```
12 REM IS=NO.OF FREQ.
14 REMX,Y,Z MIKE POSITION
29 REM SYNTHESIS MARK 4.0 20-01-81
30 REM F=FREQUENCY
40 REM C=COUPLING
50 REM D=DAMPING
60 DIM RF(30)
100 DIM F(50),C(50),D(50),M1(50),M2(50),M3(50),D1(50),W1(50),RC(50),S1(350)
103 READA$:PRINTA$
104 READX,Y,Z:PRINT"MIKE POSITION";X;Y;Z
106 READIS:PRINT"NO.OF PEAK FREQ.";IS
115 LX=2.57:LY=3.29:LZ=2.28
118 P=0:PI=
122 INPUT"NO.OF RES.FREQ.";KR
124 FORI=1TOKR:READRF(I):NEXTI
126 PRINT"  "
130 FORI=1TOIS
140 READ F(I),M1(I),M2(I),M3(I),C(I),D(I)
142 PRINT F(I);M1(I);M2(I);M3(I);C(I);D(I)
144 D1(I)=D(I)*
150 C(I)=C(I)*2*D1(I)
152 RC(I)=COS(M1(I)*PI*X/LX)*COS(M2(I)*PI*Y/LY)*COS(M3(I)*PI*Z/LZ)
160 NEXT
180 OPEN1,8,15,"I0"
184 INPUT"FILE RES.FREQ.POS.";B$
188 B$="@0:"+B$+",U,W"
190 OPEN2,8,2,B$
194 FORI=1TOKR
196 PRINT#2,RF(I);CHR$(13);
198 PRINTSPC(10):I;"1="
200 NEXT
202 CLOSE2:CLOSE1
210 INPUT"END OF DATA";G$
212 IFLEFT$(G$,1)="Y"THEN220
224 G1=LOG(70)/LOG(10)
226 G2=LOG(202)/LOG(10)
228 G3=(G2-G1)/256
300 F3=70
301 N=0
303 A$="@0:"+A$+",U,W"
307 FORF1=G1TOG2 STEPG3
308 N=N+1
309 F=10^F1
310 S2=0:T2=0
311 PRINTN;" 1="
312 W=2*PI*F
320 FORJ=1TOIS
330 W2=2*PI*F(J)
332 PRINTSPC(18);J;"  1="
340 Z1=(W2^2-D1(J)^2-W^2)^2+(2*W2*D1(J))^2
350 Z2=C(J)*W*(W2^2-D1(J)^2-W^2)/Z1
360 Z3=C(J)*W*2*W2*D1(J)/Z1
370 S2=S2+Z2*RC(J)
390 T2=T2+Z3*RC(J)
410 NEXTJ
READY.
```

```
 1000 DATA"SYNTH-POS4",2.57,3.29,0,21
 1010 DATA75.15,0,0,1,48,1.90
 1020 DATA84.58,1,1,0,64,2.72
 1030 DATA91.17,0,1,1,94,1.90
 1040 DATA100.70,1,0,1,54,3.33
 1045 DATA104.12,0,2,0,56,2.67
 1050 DATA113.22,1,1,1,81,3.27
 1060 DATA124.13,1,2,0,77,1.65
 1080 DATA128.70,0,2,1,35,3.81
 1090 DATA133.26,2,0,0,50,1.75
 2000 DATA142.37,2,1,0,68,2.55
 2020 DATA145.14,1,2,1,71,3.37
 2030 DATA155.38,0,3,0,27,3.49
 2040 DATA158.72,0,1,2,34,2.79
 2050 DATA162.09,2,1,1,82,2.62
 2060 DATA164.81,1,0,2,72,3.34
 2070 DATA169.00,2,2,0,79,3.58
 2080 DATA172.37,1,1,2,32,2.14
 2090 DATA183.17,0,2,2,45,2.39
 3000 DATA186.82,2,2,1,115,3.14
 3010 DATA194.44,1,2,2,40,1.94
 3020 DATA199.23,3,0,0,15,2.91
READY.
```

Program to Calculate s.p.l. at the Boundary of the Enclosure of the Room.

```
 10 N=2
 20 Y=0
 30 K=N*/3.28
 40 OPEN4,4
 50 PRINT#4,"DIST","PRESSURE","DECIBEL"
 60 FORY=0TO3.28STEP.82
 65 S=COS(K*Y)
 70 P=S+.85*S+.85^2*S+.85^3*S+.85^4*S+.85^5*S+.85^6*S+.85^7*S+.85^8*S+.85^9
 72 P=P+.85^10*S+.85^11*S+.85^12*S+.85^13*S+.85^14*S+.85^15*S+.85^16*S+.85^
 74 P=P+.85^18*S+.85^19*S+.85^20*S+.85^21*S+.85^22*S+.85^23*S+.85^24*S
 80 IFP=0GOTO00100
 85 DB=10*LOG(P^2)/LOG(10)
 90 GOTO00110
 100 DB=-1000
 110 PRINTY,P,DB
 120 PRINT#4,Y,INT(P*100+.5)/100,INT(DB*100+.5)/100
 130 NEXT
 135 PRINT"K=",K
 140 PRINT#4,"K=",K
 150 K=K+.01
 160 GOTO00060
 170 CLOSE4
READY.
```

Computer Program to Calculate the Decay of Sound Theoretically.

```
2 REMD-IN DEGREE
5 DEFFNA(X)=INT(X*100+.5)/100
10 R=.85
20 L=3.2[illegible]
30 C=242
40 INPUT"PHASE ANGLE DEGREE";D
50 OPEN4,4
60 PRINT#4,"R=";R;"D=";D;"L=";L;"C=";C
100 FORF=60TO2[illegible]0STEP2
110 W=2**F
115 P=(1+R^2-2*R*COS(W/C*L+D*2*/360))
120 ZR=(1-R*COS(W/C*L+D*2*/360))/P
130 ZI=(-1*R*SIN(W/C*L+D*2*/360))/P
140 Z=SQR(ZR^2+ZI^2)
340 PRINTFNA(F);FNA(Z)
345 G=10*INT(Z*10+.5)/10
347 PRINTG
350 PRINT#4,F;SPC(G);"."
400 NEXT
500 CLOSE4
READY.
```

Program to Plot the Frequency Response Curve Theoretically and Experimentally.

```
100 REM PLOT PROGRAM
105 FL=70:FH=202:L=(LOG(FH/FL))/4096
110 DIMA%(4096),F3(60)
115 OPEN1,8,15,"I0"
120 INPUT"FILENAME";Z$
125 IF LEFT$(Z$,2)="SY"THEN135
130 Q$="POSITION "+MID$(Z$,7,1)+" (EXPERIMENTAL)":Y1=6000
131 GOTO140
135 Q$="POSITION "+MID$(Z$,10,1)+" (THEORETICAL)":Y1=5200
140 OPEN2,8,2,Z$+",U,R"
150 GOSUB20000
155 NR=512:MP=1:IFY1=5200THENNR=256:MP=2
160 FORI=1TONR
180 INPUT#2,A%(I)
182 PRINTA%(I)
200 PRINT"3            ==========";I
220 NEXT I
222 CLOSE2
223 INPUT"NO.OF FREQ.";K
224 INPUT"FILE RES.FREQ.";M$
226 OPEN2,8,2,M$+",U,R"
228 FORI=1TOK
230 INPUT#2,F3(I)
232 PRINTF3(I);CHR$(13);
234 NEXT
240 CLOSE 2,1
260 INPUT"PLOTTER OK";Y$
280 IFLEFT$(Y$,1)<>"Y"THEN260
284 INPUT"PEN CHANGE FOR RES.FREQ.";H$
286 IFLEFT$(H$,1)<>"Y"THEN00284
300 OPEN5,5:PRINT#5,"IN"
305 GOTO00325
310 GOSUB10000:REM PLOT AXES
320 GOSUB11000:REM NAME AXES
325 GOSUB14850:REM MRAK RES.FREQ.
330 GOSUB30000
400 PRINT#5,"PU;PA980,980"
410 FORI=2TONR
420 X=960+16*I*MP
440 Y=1000+A%(I)*20
445 IFY>5800THEN00480
460 PRINT#5,"PA";STR$(X);",";STR$(Y)
470 PRINT#5,"PD"
480 NEXT
500 PRINT#5,"PU;PA1000,1000"
600 PRINT#5,"IN":CLOSE5
3030 END
9999 END
10000 PRINT#5,"IN"
10100 PRINT#5,"PA1000,1000"
10200 PRINT#5,"PD;PA10000,1000"
10300 PRINT#5,"PU;PA1000,1000"
10400 PRINT#5,"PD;PA1000,6000"
10500 PRINT#5,"PU;PA1000,1000"
10850 RETURN
11000 PRINT#5,"PU;PA4500,500"
11100 PRINT#5,"LBFREQUENCY (HZ)";CHR$(3)
11200 PRINT#5,"PU;PA400,1500"
11300 PRINT#5,"DI0,1;LBAMPLITUDE (VOLTS)";CHR$(3)
11400 PRINT#5,"DI1,0;PA1000,1000"
READY.
```

```
 11450 PRINT#5,"SI0.12,0.18"
 11500 FORI=0TO130STEP10
 11600 Y=1000+I*20
 11700 PRINT#5,"PA1000,";STR$(Y);";YT"
 11720 IF I/20<>INT(I/20)THEN11800
 11740 V=INT(I*0.039*100+.05)/100
 11760 PRINT#5,"PA600,";STR$(Y-30)
 11780 PRINT#5,"LB";V;CHR$(3)
 11800 NEXT
 11850 X=0
 11900 FORI=70TO205STEP5
 12000 X=INT(2*LOG(I/FL)/L+0.5)+1000
 12100 PRINT#5,"PA";STR$(X);",1000;XT"
 12200 Z=85:IFX> 95THENZ=140
 12210 IF I<105THEN12300
 12220 IF I/20<>INT(I/20)THEN12600
 12300 PRINT#5,"PA";STR$(X-Z);",800"
 12400 PRINT#5,"LB";I;CHR$(3)
 12600 NEXT
 12700 RETURN
 14850 PRINT#5,"PU;PA1000,1000"
 15000 FORI=1TOK
 16400 X=INT(2*LOG(F3(I)/FL)/L+.5)+1000
 16500 PRINT#5,"PA";STR$(X);",1000;XT"
 17000 NEXTI
 18050 RETURN
 20000 INPUT#1,A,A$,B$,C$
 20100 IFA$="OK"THENRETURN
 20200 PRINT"ERROR";A;A$
 20300 STOP
 30000 PRINT#5,"PU;PA2000,";STR$(Y1)
 30005 PRINT#5,"SI0.2,0.3"
 30010 PRINT#5,"LB"Q$
 30015 PRINT#5,R$;CHR$(3)
 30020 RETURN
READY.
```

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