# A STUDY OF THE MECHANICAL IMPEDANCE OF THE HUMAN BODY AT LOW FREQUENCY BY CONTINUOUS MONITORING TECHNIQUES

by

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A DOCTORAL THESIS submitted in partial fulfilment of the requirements for the award of DOCTOR OF PHILOSOPHY of the LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY

OCTOBER, 1974

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## ACKNOWLEDGEMENTS

The author wishes to acknowedge the assistance given to him by the staff and students of Loughborough University of Technology, Department of Ergonomics and Cybernetics, also the help and patience shown to him by his family and friends during the writing of this thesis.

CONTENTS

# PART B

## MECHANICAL IMPEDANCE STUDIES

# SECTION 1

## INTRODUCTION

## I - Introduction

Since the advent of the technological age, man has found that he has had to endure increasing physical and mental stresses. The effect of vibrational forces on man is just one such problem which has been brought about by the increasing use of road, air and space vehicles. These vibrational forces may result from slight imbalances in rotating or reciprocating machines, discrete energy inputs or by the environment in which the vehicle is travelling. The vibrations, as well as having an adverse effect on the machine itself may also affect its user.

All the vibrational inputs first have a mechanical effect on the body, the resultant response of the subject being dependent on the precise nature of the vibration input. In considering the effects of steady-state vibration on the body the frequency range of 1-20 Hz has received particular attention, below 1 Hz the effect to the subject is predominately that of motion sickness, while above 20 Hz the protection of the body by mechanical damping systems is relatively easy. Within the frequency range 1-20 Hz the response of the human body to vibration is very complex and may be dependent on a large number of variables, (See Figure No.1) which are in themselves time dependent.

OTHER INFLUENCES

| Physical | Psychological | Physiological |
|---|---|---|
| sound | motivation | health |
| light | experience | biorhythms |
| etc | etc | etc |

Vibration Stimulus. → receptors → BRAIN → Response

Figure No.1

If the human body is viewed from a mechanical viewpoint only, the body still appears a complex system. This complex system being studied in various ways, depending on the researchers particular interest. The study of the mechanical properties of the body tissue, bones and organs is often associated with the vibration of a specific part of the body; whilst whole body vibration studies have often been concerned with the postulation of mechanical analogues to simulate the mechanical response of the human body to vibrational inputs. In the latter study the question of the linearity of the bodies response has received particular attention.

Mechanically the body may be regarded as a complicated system of masses connected by visco-elastic elements, whose mechanical properties are non-linear, and due to the "active" nature of the body, are also time dependent. It would therefore appear to be a very complex if not impossible task to accurately simulate the response of the body; previous research has shown however that under certain vibrational inputs the body can be approximated by a linear mechanical system. Thus a mechanical analogue can be devised to simulate the response of the body but only for a specified vibrational input.

A method of assessing the mechanical properties of an unknown system, such as the human body is that of mechanical impedance measurement. In impedance studies the body is regarded as a "black box", the force input to the black box is then measured together with the motion it produces. The impedance is then defined as the complex ratio of the force to the motion produced.

Motion however can be specified in one of three ways:- displacement, velocity or acceleration, this means therefore that "impedance" can be computed in terms of displacement, velocity or acceleration. Many names have been given to these "impedances" in the literature. In this thêsis however all three will be referred to as impedance, with the motion being specified; i.e. they will be referred to as displacement, velocity or acceleration based impedance.

In most previous studies the velocity based mechanical impedance has been measured e.g. Coermann R.R. (1963), Krause H.E. and Lange K.O. (1963) Edwards R.G. and Lange K.O. (1964), and Suggs C.W., Abrams C.F. and Stikeleather L.F. (1969). The impedance computed in the studies reported here was the acceleration based impedance, as the force cell used in the research measured the force and acceleration inputs to the body. The acceleration impedance was computed due to the difficulty in accurately converting the acceleration signal to the velocity signal required to obtain the velocity based impedance. The acceleration based impedance can be easily converted by a digital computer to the velocity or displacement based impedances.

The mechanical acceleration based impedance was measured primarily in the seated posture using a sweep of the sinusoidal input motion from 3 to 30 Hz, the subject's legs being supported. This posture was adopted as it was thought that it best simulated the seated passenger in a vehicle. The input force being in the direction of the longitudal axis of the body, i.e. the measured force and acceleration being in the direction as specified in the I.S.O. Recommendations (1969). To complete the study measurements were made on a small number of subjects in:-

a) performing a tracking task in the seated posture while being vibrated.

b) three different standing postures, to try to assess the vibration isolation qualities of the legs.

Previous researchers have in their determination of whole-body mechanical impedance, used a steady frequency and have plotted the force and velocity input to the body, digitised these plots, and computed the impedance modulus and phase angle with a digital computer. This procedure being repeated at different frequencies in steps of $\frac{1}{4}$ or $\frac{1}{2}$ Hz. The object of this research programme was to obtain a continuous plot of the impedance modulus against the input frequency of the applied vibration during the experiment.

Prior to undertaking the impedance monitoring experiment a study was made into the safety of subjects undertaking vibration experiments. Although this study was made with particular reference to the Loughborough University actuator, the research did however provide some useful information for other institutions engaged or wishing to engage in this particular field of human research. A systems analysis approach was used on the subject - actuator - operator system to detect any potential failure situations and test, if possible, the effects they would have on a subject. The analysis also enabled modifications to be made to the actuator before its use in a major experimental programme.

As the experimental programme of impedance measurements was not undertaken until after the safety study the research reported in the thesis is divided into two distinct and separate parts, A. and B.

Part A describes the safety tests carried out on the actuator, and recommendations for modifications to improve its safety; while Part B reports on the experimental programme for the continuous monitoring of the mechanical impedance of the human body during steady state sinusoidal vibration within the frequency range 3 Hz to 30 Hz.

# SECTION 2

## LITERATURE REVIEW

# 2 - Literature Review

2 - 1 Introduction This review of the literature is not intended to be a full bibliography (as for instance is that of Guignard and Guignard (1970) on the effects of vibrational forces on man, but refers to literature which is relevant to the topics of research detailed in the introduction to this thesis.

The literature reviewed in the field of vibrational effects on man's performance, although not directly relevant to the research topics of this thesis, have been included as they contain useful information on equipment design - e.g. with regard to actuator type and performance, and subject safety.

The literature review has, for convenience, been divided into three parts. The first part being the following section detailing general papers and reviews. The other two parts are given separately.

## 2 - 2 General Papers and Reviews

A fair proportion of the research concerned with the effects of sinusoidal vibration on the human body has been carried out in the United States. However the majority of this research has been sponsored by the Armed Forces or N.A.S.A. and the test equipment used and the research techniques employed have, in many cases, been applicable only to the particular problems being studied. However, several general papers and reviews from the United States have proved useful to the author.

A review by Hornick R.J. (1962) gave summaries of the effect of sinusoidal vibration on compensatory tracking, visual acuity and reaction times. With regard to compensatory tracking he stated that while there were discrepancies in the literature, there was general agreement that compensatory tracking ability showed a decrement, which was related to amplitude and acceleration increases, and vibration duration in the range 1 - 30 Hz. Hornick's views on compensatory tracking decrement were shared by Bevis and Hughes (1970) and they also commented - "certain tasks, for instance:- compensatory tracking,do not show complete recovery from interference immediately following vibration."

Hornick also listed a number of basic problems in the execution and reporting of human vibration research. He stated that not enough attention was paid to:- (i) subject types - i.e. age, sex, intellect, anatomy, motivation and experience levels; (ii) the influence of restraints used in experiments designed to measure body dynamics.

He also noted that to obtain meaningful results in this research field a faithful transmission of the input motion to the subject was important and this required the subject to be seated on a rigid support attached to the particular shaker system. This did not apply if the objective was specifically to study seat characteristics The input motion should also adequately describe:- Does it have constant amplitude, or constant acceleration? Is it measured as a peak value or peak to peak or is r.m.s. units?

The paper by Hornick was of general interest, but is particularly useful to the research student as it gives an insight into the pitfalls into which he can easily fall in this particular research field.

Hornick also referred to the terminology for motion direction used by the National Academy of Sciences - National Research Council, and how it can be carried over to the field of vibration research:- For acceleration the Council has designated motion from chest to spine with the subscript:-

"x" (+ $g_x$ = eyeballs in)

for motion from shoulder to shoulder:-

"y" (+ $g_y$ = eyeballs left)

and for motion from head to toe:-

"z" (+ $g_z$ = eyeballs down)

Hence vibration inputs could be described with reference to the human body. This is the same terminology as is adopted by the International Standards Organisation recommendation "Guide for the Evaluation of Human Exposure to whole body vibration" (1970). This terminology is also used for acceleration measurements quoted in this thesis.

A general paper by Guignard, J.C. and Irving, A. (1960) outlined the various physiological effects of vibration and indicated the frequency ranges in which these effects occur. Guignard and Irving gave details of a transmissibility study and a study of the effects of vibration on visual performance. The studies were carried out on ten male subjects, five "large" and five "small". Posture was standardised by a simple horizontal sighting tube, the tube being adjusted for each subject when he was seated on the vibrator platform. However, it might be argued that this rigid control of posture was not as representative of the "field" situation as a posture adopted voluntarily.

A summary of results and the equipment used by Guignard and Irving are shown in Fig . No 2

One of the most comprehensive literature reviews to be undertaken in this country was that of J.C. Guignard and Elsa Guignard (1970). The following extract from the preface of the review gives an indication of the scope and nature of the review:-

"Its purpose was to provide guidance to the diffuse and conflicting literature on human response to vibration, with particular reference to low-frequency vibration affecting the efficiency of aircrew. Such guidance was needed in the formulation and appraisal of criteria and limits of human exposure to vibration in flight and in ground operations, and in planning of future research. Work related to the problems of control and ride in modern transport in general was also examined during the course of the survey."

The survey covered about six hundred and fifty papers published before September 1969. Each paper was assigned to one of the following six categories:-

(1) Description of environment.

(2) Definition of criteria and limits of exposure to vibration.

(3) Research reports on the human response to vibration.

(4) Development or testing of anti-vibration procedures

or devices.

(5) Development of experimental facilities or techniques.

(6) Information or education (reviews and bibliographies).

The survey then reviewed papers generally under the headings given above and detailed the percentage of reviewed papers that fell into that particular category as well as giving details of subject breakdown under these headings. The survey contained sections on "Ethical and Safety aspects of Human Vibration Experiments", and "Recommendations for the reporting and carrying out of vibration experiments". The former section will be referred to again later. (Section A - 2).

The review also contained a comprehensive glossary of terms commonly used in the vibration. The terminology laid out in this glossary has been adhered to as closely as possible in this thesis.

Guignard and Guignard pointed out several areas where they felt reporting had been very sparse in the past, and that these should be reported in future papers on vibrational effects.

The first point to be made at the beginning of the review is that the field covers such a large number of disciplines, this often leading to the ambiguous usage of words. One of the most common ambiguities being in the reporting of experiments using sinusoidal vibration, authors referring to the amplitude of the vibration, "sometimes the quotation of the half-wave or vector amplitude appears to be intended: at others the peak to peak or double amplitude. Lack of clarity on this point alone can - and occasionally does - introduce a factor of uncertainty of 2 into any conclusions to be drawn from the experiment."

Guignard and Guignard (1970) made many comments throughout their review on the quality of reporting, and the deficiencies which often occurred in the surveyed papers. The review has been of great value in obtaining relevant references on mechanical impedance and vibrator safety and has also been of considerable help in considering the format and contents of this thesis.

Many of the comments on reporting and discrepancies in the literature cited by Guignard and Guignard (1969) were echoed by Allen (1969, 1971 and 1971a). In his paper "Human Reaction to Vibration (1971) he reviewed the present knowledge in the field and also commented on the ISO proposals (ISO(1969). The report concluded with some suggestions for future research, and the format it should take.

Allen in his conclusions and recommendations commented that:-

a) A lot of the confusion had arisen in the past because experimentors had not clearly defined the vibration environment used. Allen suggested that experimentation should be "standardised" on a national and international basis.

b) More "field research" should be undertaken as this type of experimentation can provide realistic evidence as to the effects of vibration on the human being.

c) Allen also called for designers and research workers to use cautiously the present literature on the human reaction to vibration, and commented on the transfer of laboratory obtained results to "real-life" situations.

d) He concluded:- "Fourthly, in research into the effects of vibration, let us always strive to find out 'why', not just 'what'."

Another comprehensive review of the literature although not on the scale of Guignard and Guignard (1969) was that of Beevis D. and Hughes, J. (1970).

The report made many interesting comments on vibration research as well as giving the research worker a larger number of useful references. Some of the comments are outlined in the later sections of this literature review.

The report also contains a section on the background history of the ISO. TC/108 Recommendations, and commented upon its use and validity in practice.

Other reviews of interest are those of Von Gierke, H.E. (1964) (1966). Von Gierke's reports contain considerable information on the mechanical impedance of the whole body and also body organs. This information will be detailed in sections three and four of this review.

A review by Matthews, J. (1964) contained much useful information about "off the road" vehicle seats and suspensions, and some interesting data from many researchers' experiments on seat attenuation and transmissibility. A graph by Simons, A.K. (1952) compared the transmissibility of two seats to that of the human legs. This is shown in Fig. No.39.

A comprehensive review by Sandover (1969) dealt with the reaction of the human body under impact and aircraft ejection conditions, and the design of a suitable analogue to represent the body under these conditions. The review also contained many relevant comments and references, in the field of vibration research, and especially with regard to the Mechanical Impedance of the human body under impact and sinusoidal conditions.

Shurmer (1967), Synder, F.W., Beaupeurt, J.E., Brumaghin, S.H. and Knapp, R.K. (1968), and Harris and Shoenberger, R.W. (1965), all contained useful references and offered good general reading in the field of human reaction to vibration. The latter report, as well as reviewing existing information on human performance also detailed an experiment carried out by Harris and Shoenberger into human performance under vibration conditions.

There are also several text books available dealing with vibration, in general terms, and some physiology and engineering text books containing sections referring to vibration effects on the human body, mechanical impedance and vibration isolation theory, the following were consulted in the course of this research.

Harris and Crede (1961) "Shock and Vibration handbook" (Volumes 1, 2 and 3) contains chapters on vibration isolation and mechanical impedance. The handbook does contain some information on the physiological and biodynamic effects of vibration on the human body. There is also detailed information on measuring vibrations and on the types of transducers used. Another textbook giving useful information on the physiological aspects of vibration research is that edited by Alt, F. (1966) "Advances in Bioengineering and Instrumentation." The text tells which physiological quantities need to be measured, and how to measure them. "Anatomy and Physiology for Nurses" by E. Pearce (1962) was found by the author to be a useful reference text on human physiology, for engineers as well as nurses.

As the research project was more of a bioengineering nature, a textbook, dealing with mechanical vibrations, "Vibration Theory and Applications by Thomson, W.T. (1965) was consulted. The text dealt with single and multi-degree-of-freedom systems, in free and forced vibration situations. It also covered shock loading, sinusoidal and random vibrations and sections on analogue and digital computing techniques.

Texts useful in regard to analogue computation and analogue elements are referred to in Section B - 5 of the thesis.

# EXPERIMENTAL PROGRAMME

# PART A

## VIBRATOR SAFETY

# PART A - Vibrator Safety

## A - 0 Introduction

The initial aim of the research was to conduct mechanical impedance measurements during steady state sinusoidal vibration on a number of subjects in defined seated and standing postures. The research also involved the installation and testing of a suitable vibrator, before any impedance measurements could be made. During consultations with the manufacturers of the vibrator and during its installation, it was realised that a comprehensive programme of safety check tests would need to be carried out before the actuator could be used with large numbers of subjects in a full experimental programme. This part of the thesis deals with the installation, performance, testing and safety appraisal of the actuator before its use in major experimental programmes.

When the actuator and its power supply had been installed, initial performance tests were carried out, with regard to actuator performance and noise pollution of the surrounding environment. These are reported in Section A - 3. After the installation of the actuator a careful analysis of the operator-actuator-subject system was undertaken, and a range of safety check tests were devised from the results of the analysis. (Section A - 4). These tests were to ensure that in any potential failure, the resulting acceleration (or impulsive) waveforms of the actuator piston would be within the draft limits laid down by a National committee investigating subject safety in vibration experiments.

The actuators performance during these safety check tests proved satisfactory, but using a collection of comments from operators, subjects and observers during subsequent experimental programmes, a set of recommendations to modify the Loughborough University actuator were drawn up. (Section A - 5)

During the design and development of the electro-hydraulic actuator, it was realised that the actuator would be unsuitable for use in undergraduate practicals. Thus whilst the electro-hydraulic actuator was in production, a small mechanical portable vibrator, for use in undergraduate vibration studies was designed and developed. (See Appendix 2).

## A - 1 Risk

### A - 1.0 Introduction

The safety of the human subjects must be of paramount importance to any experimenter working in the field of human research. This is especially so when the experimental equipment could cause considerable injury or even prove to be fatal to the subjects used. The use of an electro-hydraulic actuator constitutes this sort of risk, thus the safety of human subjects used has to be borne in mind from the initial design of the vibration equipment through to the completion of the experimental programme.

The risks to a subject involved in vibration experimentation can be classified into two types, the Inherent Risk and the Extraneous Risk.

### A - 1.1 Inherent Risk

Inherent risk is the risk involved as part of the experiment, that is the risk associated with the vibration level used, or any other procedure that might be adopted in the course of the experiment.

The most likely causes of inherent risk in this context would be:- a) use of too severe a level of vibration or

b) failure to exclude a subject who is medically unfit to participate in the experiment.

The level of vibration and method of subject selection used by the author were adopted with reference to the "Draft Guide on the safety aspects of Human vibration experiments". (Draft 3 1969). This document will be superseded by a British Standard "Draft for Development".

A - 1.2. Extraneous Risk

Extraneous risk is that incidental to the nature of the experiment. This type of risk involves the failure of the equipment or operator; also any accident that might occur to the subject in the laboratory; before or after the actual experimental programme.

The following section is mainly concerned with the extraneous risks to the subjects. The section deals with the design and selection of the actuator and this has a considerable bearing on the safety of the vibration equipment. Simulated failures are also reported as these provide useful data on the dangers of the equipment to experimental subjects.

## A - 2 Literature Review

### A - 2.1 Safety in Human Vibration Experimentation.

In reviewing the literature on vibration experimentation in this country and abroad, one notices that one aspect commonly neglected is the reporting of the safety precautions taken to protect subjects from equipment or operator failure. This fact was noted by Guignard and Guignard (1970) "Less than 5% of the papers in Category 3 - which report human experiments using whole-body vibration machines or allied motion devices, give any details of safety measures or other provisions for the well-being of the subjects used in the experiment."

Considered below are a number of papers presented at an informal meeting of research organisations interested in, or already engaged in, vibration research. This meeting was called by J. Matthews at N.I.A.E., Silsoe on 7th November, 1968. The author has also found the document; "vibrator safety" , Matthews J (Ed) (1968) drawn up by a committee formed at this meeting, to have been of considerable help in the design of the safety systems used on the actuator.

Chisholm (1968) "Subject Safety in Hydraulic vibrator experiments" gave the results - although only a visual appraisal of acceleration was used - of some simulated faults on a particular test rig. Five simulated faults were used. From the visual assessment of the resulting ram accelerations, the failure giving the most severe accelerations was that of over-loading the unit with a "square wave input signal whose magnitude corresponded to a displacement of the ram which was one and half times its physical maximum stroke".

The failures were simulated with the ram unloaded. Chisholm did however retest with the square wave, loading the ram to 150 lbs with bags of lead shot, and measured the resulting acceleration - "The peak acceleration was of the order of 40g using 150% maximum stroke square wave input at 10 Hz." He concluded his paper by listing possible safety devices and some disadvantages in their use. He noted that "A safety cut-out should be as far downstream in the system as possible to cater for the greatest variety of failure conditions."

He started by suggesting a mechanical link between the subject and actuator ram designed to fail at an injurous acceleration level, (i.e. with regard to the subject). He also pointed out that this could be difficult due to variation in the plastic behaviour of the failure link and that "different mechanical impedancies of the rig/seat/subject system would give wide variation in the ratio of subject acceleration/fail-safe load."

He also postulated the use of an electronic control of the servo-valve - this depending on the type of servo-valve used. The control would be in the form of a pre-set inertia switch placed on the rig which could interrupt, via a relay, the servo-valve control line. Mechanical failures, although almost impossible to prevent, could be minimised by regular checking of the equipment - actuator, subject support rig, and electrical leads etc.

Two other papers presented at this meeting dealing more with the design of actuators for use in human experimentation were Clarke, M.J.(1968) "Methods of Subject Protection" and Ashley, C.(1968) "Special requirements in Specification of Vibrators for Human Vibration." A third paper presented at this meeting was by Guignard, J.C.(1968) "Some notes and recommendations on the use of human subjects in vibration experiments." These papers formed the basis of the "Guide on the safety aspects of Human Vibration Experiments", (1968 - 1969). The earlier draft copies of this guide were very useful to the author in his work with regard to the safety of the Loughborough University actuator.

The papers by Clarke (1968) and Ashley (1968) dealt with the mechanical and electrical aspects of equipment used in human vibration experiments. The papers discussed the features that needed to be designed into an actuator when it is to be used for experiments with human subjects. The authors recommended the use of hydraulic snubbers, electric control on the servo-valve input signal, electric limit switches which come into action at the limits of travel of the actuator ram, and a well designed operator control panel.

Other papers in the specific field of human vibration experiments consulted by the author were:- Guignard and Guignard (1970) Section 7 "Ethical and safety aspects of human vibration experiments." This section is a precis of the Guignard J.C. (1968) paper, and refers the reader to the Code of practice -

"Guide on the safety aspects of Human Vibration experiments." Allen (1970) and Allen (1971) "Techniques for body vibration experiments", suggested that a code of practice for all aspects of human vibration experiments should be drafted. The two papers contained Allen's views on the format this code of practice should take. The papers also made recommendations to experimenters working in this field, as to how they should conduct their investigations, and also what they should report, and the format their report should take. This was an attempt to standardise the research in this particular field which had been carried out on a very ad-hoc basis in this country and abroad.

A - 2.2. Industrial Safety

Several textbooks on industrial safety were also consulted and found to be of general interest with regard to the safety studies.

DeReamer, R.(1961) "Modern safety Practices". the text, although dealing with industrial safety, did however contain three useful sections on safety theory, and the practice of this theory in industrial situations. The sections referred to Chapter 2 (pp 15 - 28) dealing with basic accident prevention; Chapter 3 (pp 28 - 45) "Accident Proneness fact and fiction" and Part IV of the text which dealt with "Appraising Analysing and measuring safety".

DeReamer did however detail a useful industrial safety technique - that of reporting "near accidents". This method involves the operator in reporting any situations which he considered were "near accident" situations. The safety officer can then modify the process or equipment to reduce the possibility of an accident, in the light of these reports.

Johnson (1970) "New approaches to Industrial Safety" and Chapanis A. (1958) "Research techniques in Human Engineering" provided useful additional reading on the subject of safety in human experimentation. Chapanis in Chapter 3 "Methods for the study of accidents and near accidents" outlined some of the difficulties and dangers of this technique as a means of reducing accidents. The accurate reporting of any accident or near accident is essential if the situation is to be reconstructed under test conditions. The reconstruction may however be difficult if the situation was a result of operator error, for the operator may have been unaware of the error he made, and when and how he made it. Chapanis also noted that this study may not in fact help in the reduction of accident or near accident situations as many occur in conditions which would appear ideal for safe operation.

## A - 3 Vibrator Selection.

### A - 3.1 Introduction

To investigate the response of the human body to sinusoidal or random vibration, requires an actuator capable of vibrating the whole body. The actuator must perform with maximum safety to the subject, and accurately with respect to the required output waveform at the low frequencies required for this type of study.

There are four main types of actuator in use for industrial and research purposes. These are:-

1) Mechanical
2) Hydro-Mechanical
3) Electro-Hydraulic
4) Electro-Magnetic

The four types of actuator have certain advantages and disadvantages with regard to use for human whole-body vibration experimentation which would affect the selection of an actuator for a particular research area.

### A - 3.2 Appraisal of the four actuator types

#### A - 3.2.1 Mechanical Actuators - Introduction

This actuator system, as the title implies, is entirely mechanical. It relies on mechanical linkages or cams to obtain reciprocating motion. There are three common types of mechanism and these are described briefly below.

i) Slider-crank mechanism
ii) Scotch-yoke Mechanism
iii) Cam-follower mechanism

A - 3.2.1 (i) Slider-crank Mechanism

This is probably one of the best known mechanisms for converting rotary motion into a receiprocating motion.

The slider-crank chain consists of three turning pairs and one sliding pair, see Fig. No. 3. The mechanism usually appears in the form of the reciprocating engine mechanism.

This type of mechanism will not, however, produce true sinusoidal motion, but the larger the ratio C/r becomes, the closer the motion of the mechanism approximates to sinusoidal motion. A table showing the deviation from simple harmonic motion of this type of mechanism for two values of C/r was calculated by Edwards and Harding (1963).

True sinusoidal motion is only achieved when the ratio C/r becomes infinite.

A - 3.2.1 (ii) Scotch-yoke Mechanism

This mechanism is a double slider-crank chain consisting of two turning and two sliding pairs, see Fig No. 4.
This type of mechanism will produce a true sinusoidal motion but as can be seen from the figure the mechanism has an extra sliding bearing to the slider-crank mechanism. This additional sliding bearing could well cause a considerable amount of "Mechanical noise" on the acceleration waveform of such an actuator. Mechanical filters can be used to improve the acceleration waveform, but they are expensive, especially in the frequency range required for human whole-body vibration investigations.

A - 3.2.1 (iii) Cam-follower Mechanism

The cam-follower mechanism is known as a higher kinematic pair. Elements in higher kinematic pairs generally have line or point contact and the pair must be force-closed in order to provide completely constrained motion, see Fig. No. 5.

This mechanism will produce a true sinusoidal motion. The easiest way of obtaining such motion is to use a circular cam rotated eccentrically, using a circular follower, (see also Appendix 2). Although the stroke cannot be altered so easily as that on a slider-crank or scotch-yoke mechanism, a cam-follower type actuator has the advantage that other cyclic waveforms can be obtained by reprofiling the cam.

The cam-follower mechanism can only be used up to acceleration levels of one g (single peak amplitude) without spring loading the cam. If the cam is spring loaded higher acceleration levels can be obtained. When, however, acceleration levels of more than one g (single peak amplitude) are used the subject has to be restrained to move with the actuator table.

The mechanical actuator can only be used to produce cyclic waveforms and some are restricted to sinusoidal only. Mechanical actuators cannot produce random waveforms, or reproduce recorded "in the field" vibrations for laboratory simulation experiments. Theoretically the cyclic waveform should not be distorted by increased load, the load increase does however affect the frictional forces which cause waveform distortion. The waveform distortion can be eliminated by low-pass mechanical filtration of the output motion, This, however, is expensive.

A mechanical actuator is easy to construct and operate, and is inherently safer than the other types of actuator. Mechanical vibrators are commonly used in the United States where they are often referred to as "shake-tables". The research literature from the United States, however, gives little detail of construction or performance, and very few acceleration waveforms are available in the research reports.

As part of this study a small, portable mechanical vibrator was designed, and built at Loughborough University of Technology. The design of the actuator is described in detail in Appendix 2.

A - 3.2.2 Hydro-Mechanical Actuators

This type of actuator incorporates a cam-follower mechanism to generate the desired cyclic waveform, with hydraulic transmission of the motion to the actuator table, see Fig. No. 6. This type of actuator could well be described as a mechanical actuator using hydraulic transmission to give flexibility to the drive. The hydraulic transmission also has the added advantage of acting as a low-pass mechanical filter.

As with the cam-follower mechanism, the hydraulic transmission of motion to the table is single-acting, and has to be spring returned if acceleration levels of more than one "g" (single peak amplitude) are to be used.

This type of actuator is not commonly used in human whole-body vibration research field. One was, however, in use at the N.I.A.E. and is detailed by Matthews J. (1964).

A - 3.2.3 Electro-hydraulic Actuators

This type of actuator utilises hydraulic power which operates a double sided piston. The flow of oil to the actuator is controlled by a servo-valve which responds to electrical signals from an electronic control unit.

The high pressure oil is usually supplied by an electrically driven oil pump, the total hydraulic power pack also contains an oil cooler (air or water), a pressure relief valve, a hydraulic accumulator and an hydraulic "dump" valve, see Fig. No. 7.

The hydraulic power pack as well as being an expensive piece of equipment, is often noisy. The high noise level often entails the building of a separate housing for the power pack to eliminate the pump noise from the laboratory and surrounding environment.

The high pressure oil from the hydraulic power pack is piped to the actuator manifold, upon which the servo-valve is mounted. The oil flow to the cylinder and exhaust flow is then controlled by the servo-valve, see Fig. No. 8. A detailed description of the actuator used for the author's experimental programme is given in Appendix 1.

The control of the electro-hydraulic system is usually accomplished by position feed-back or sometimes pressure feed-back to the electronic control unit. The electronic control unit follows that of a conventional servo-system and can be set to give the actuator constant displacement or acceleration.

The control unit can respond to a sinusoidal input, or any other electrical signal (within the limits of the total system response). This means that recordings of "field" vibrations suitably processed can be reproduced in the laboratory.

The electronic servo-control and the low-friction of the actuator give the total system a high accuracy of reproduction. The actuator can also be built with a large stroke which enables this type of actuator to be used for low frequency vibration.

The electro-hydraulic actuators are, however, much more dangerous than the types of actuator already described, because they are capable of producing a much larger thrust. Due to the complex construction of electro-hydraulic actuators they are also more prone to failure than one which is purely mechanical.

Electro-hydraulic actuators have therefore to be designed with safety as the primary consideration, and this results in a need for more safety devices than are usually present on a conventional mechanical actuator.

The electro-hydraulic actuator is the type most commonly used in this country for human experimentation. Many of the actuators in service, however, have been designed for the testing of mechanical components, and are therefore not as suitable for human experimentation as actuators specifically designed to vibrate human subjects.

A - 3.2.4 Electro-Magnetic Actuators

Electro-magnetic actuators have been previously only used for vibrating parts of the body or small mechanical components. Modern developments, however, now make possible the use of an electro-magnetic whole-body actuator. This operates on a similar principle to that used in a loudspeaker, which enables sinusoidal, random and "field" recordings, to be reproduced, see Fig. No. 9

The actuator's moving parts have a low inertia, and this means the system has a very fast response time, faster than the other types of actuator previously mentioned. This fast response time and low inertia of moving parts, means that these actuators are often used for the assessment of vibration threshold levels. Electro-magnetic actuators are, however, only capable of small displacements, and this together with their low static load capacity means that they are unsuitable for low frequency whole-body vibrations experiments.

## A - 3.3 Vibrator Specification

### A - 3.3.1 Actuator Physical Requirements

The nature of the research programme into the vibration environment that was to be carried out at Loughborough University of Technology had a considerable influence on the vibrator specification. The results of previous studies carried out by the filming of subjects, walking and running, had shown that the head and other parts of the body were exposed to high acceleration levels. (That is compared to the limits proposed for the ISO Exposure Limits (1972) ). The acceleration levels were obtained by displacement measurements on successive film frames.

As the research programme involved the use of acceleration levels close to the ISO recommended maxima, at frequencies comparable to those the body is exposed to during running and walking, the actuator would therefore require a large stroke, at frequencies between ·5 Hz and 5 Hz. The values obtained from the subjects running and walking, and the proposed ISO limit are shown on Fig. No. 10. this figure also illustrates the performance envelope of the Loughborough University actuator.

The research programme would also involve simulating field conditions in the laboratory. This meant the actuator would have to be capable of reproducing recorded signals, and also have sufficient thrust to enable a small experimental rig to be vibrated with the subject.

These requirements meant that the actuator would need a total stroke of 25 to 30 cms (10 to 12 inches) with a thrust of 450 N (1000 lbf) to enable a subject and equipment of 450 kgs (1,000 lbs) to be accelerated to a level of one "g" (single peak amplitude). The actuator would also have to be capable of reproducing " in the field" conditions accurately and then change to sinusoidal waveforms quickly, to enable comparative studies to be carried out easily. The only type of actuator to fulfil these specifications economically would be an electro-hydraulic vibrator.

A - 3.3.2 Actuator Safety Requirements

The physical requirements of the actuator were set by research requirements and the safety requirements had to be specified before the final actuator design could be drawn up. This type of actuator, as previously stated, is potentially the most dangerous, and many safety features have to be incorporated in the actuator at the design stage as they are difficult to incorporate afterwards.

The safety requirements for the actuator were formulated from a flow diagram of the different types of equipment and for operator failure. The diagram consists of a number of basic units.

Basic unit

| | | |
|---|---|---|
| Cause of fault (Equipment and/or Operator) → | Safety Device if involved → | Effect of fault on subject and/or equipment. |

The final complete flow diagram is given in Appendix No 5 and consists of a number of such basic units together with feeds forward and interconnecting loops. The main use of the diagram is to ensure that the safety devices envisaged for the actuator would be sufficient but not replicated. The diagram can also show the most efficient locations within the total actuator system to incorporate safety devices, as failure of one particular component may lead to a number of possible paths, some of which could be dangerous with regard to the subject.

The use of this flow diagram technique enabled the designers of the actuator to incorporate many features which had not previously been built into conventional electro-hydraulic servo-systems.

The incorporation of hydraulic snubbers on the actuators was to eliminate the metal to metal impact of the piston into the actuator body. The hydraulic snubbers are governed by relief valves which are preset to cater for different actuator pay-loads.

The electronic control unit incorporated an emergency shut down device to bring the piston gently to rest, a ten turn frequency setting control (instead of the normal decade frequency switch) and the servo-valve was selected to limit the maximum linear velocity of the actuator piston to 125 cms/sec.(50 ins/sec).

The electronic control unit also contains a relay circuit which prevents the operator switching on the internal oscillator until the amplitude potentiometer was set to zero. Thus the operator cannot set the actuator in motion instantly, he has to set the amplitude to zero and then increase the amplitude to the desired level. The subject can also stop the actuator if he becomes apprehensive of the increasing acceleration level.

The action of these safety devices is shown on the flow diagram. The testing and evaluation of these safety devices is reported in Section B - 4 (Vibrator Safety Tests).

A - 3.4 Vibrator Installation

The actuator was mounted on a concrete block - 1.55m(5ft) by 1.85m(6ft) and 1.25m(4ft) deep, the block being isolated from laboratory floor by polystyrene. As the overall actuator height is 1.15m(45ins) the concrete block was set 0.40m(15ins) below the laboratory floor level. The setting of the actuator below the floor level meant that sitting on the actuator was equivalent to sitting on a table; it was felt that subjects would feel safer on the vibrator if they thought they could jump off it easily and safely. Positioning the actuator below floor level in this way also enabled experiments to be carried out on standing, as well as seated subjects.

The actuator was fixed to two cast steel channels set into the concrete block. The actuator could be fixed in one of five positions on the channels, the hydraulic power was supplied to the actuator through flexible pipes to facilitate this movement.

The laboratory site was such that the operator could look at the subject during the course of the experiments. The operator was also provided with an emergency stop button, and a stop button for the hydraulic power pack. The power pack control console was visible through a triple glazed window, but a warning light for the oil filter (to indicate when the filter was blocked) was also fitted near the operator's control station.

The subject was also provided with an emergency stop button, which was mounted on a hand grip. Another emergency stop button was installed at the observer's station. These were to enable the subject or the observer to stop the experiment at any time they thought fit. A view of the complete installation is given in Fig No. 11.

A - 3.5 Hydraulic Power Pack Installation

The hydraulic power pack was installed in a small brick built annex adjoining the laboratory site. The annex was designed in such a way as to minimise noise to the neighbourhood and in the laboratory area.

The noise emission from the hydraulic power pack was measured with a Bruel and Kjær noise meter and the results are tabulated below:-

| Condition | Noise Level |
|---|---|
| Noise 1 metre away from hydraulic power pack - no sound insulation. | 86 DbA |
| Noise outside annex. | 70 DbA |
| Inside laboratory with actuator functioning normally. | 54 DbA |

The noise meansurements show that although the hydraulic power pack had not been completely silenced, its noise output was now of such a level as to cause little inconvenience to subjects in the laboratory or local residents in the neighbourhood of the laboratory.

The main difficulty in reducing the noise level of the hydraulic power pack was the air-cooling of the oil. This required ducts to be provided to enable air to flow through the annex and these provided a noise path which was difficult to treat.

## A - 4 Vibrator Safety Tests

### A - 4.1. Introduction

Although the initial actuator design had been biased towards safety, the measures taken at the design stage now had to be evaluated by means of tests. These safety tests and the reporting by subject, operator or observer of any potentially dangerous situation they could envisage, were used to modify the safety flow diagram. The flow diagram had been drawn up on a board in the laboratory area so modifications could be made as soon as reports from those participating in experiments were received. Reports that could not be added directly to the flow diagram were filed by the author. The results of these reports and modifications to the diagram are shown in Section A - 5.

### A - 4.2 Simulated Failure Modes

The testing of the actuator safety features and the simulation of faults that could prove potentially dangerous to the subject were undertaken in a programme of nine tests. The testing of all potentially dangerous situations was not possible, as carrying them out may well have resulted in mechanical damage to the actuator - or its control unit; these situations were therefore noted but no results given for them.

The series of tests was arrived at by studying the complete flow diagram, and selecting the potentially dangerous end-points. The diagram was then simplified to the nine basic units which were to be involved in the tests. The units are shown in Table No. 1 the units are labelled alphabetically, the tests numerically. The breaking down of the complete flow diagram to the basic units was effected to enable the test programme to cover as many different failure modes as possible with the minimum number of tests. The reduction of the diagram to the nine basic units also enabled the simulated faults to be as close as possible to the actual failure situations.

A - 4.3 Test Procedure.

The vibrator was set up for the tests as close to its normal operating condition as possible. The precise settings are given in the description of each test.

The vibrator for all the tests was left unloaded apart from the table. (Approx. weight 15 Kg). If the table had been loaded, solid cast iron weights would have been used, and these would have required bolting down to the actuator table to avoid damage to the actuator or test personnel, at acceleration levels greater than 2g (pk to pk). Any measured acceleration levels with the actuator not loaded would be the highest attainable levels and any loading, whether by a subject or weights would reduce the acceleration levels attained by the actuator in a failure mode. The vibrator was then tested with no load, set at a frequency of 5 Hz with an acceleration level of 1.5g (peak to peak) (except for Test No. 3.) and with an operating oil pressure of $20.7 \times 10^6$ N/$m^2$ (3000 p.s.i.). The accelerometer (see Appendix 3 "Experimental Instrumentation") used to monitor the piston acceleration was mounted on the underside of the aluminium table as close to the centre of the table as possible. The displacement and oscillator outputs were obtained directly from the control unit. The four signals were then fed via amplifiers to a Honeywell ultra-violet recorder. The calibration sheets for two signals are shown in Figure No. 12 together with amplifier gain settings, quantity measured and resultant calibration. The oscillator output signal was not calibrated as it was only required to show the operating frequency. The output from the control panel is the maximum signal output of the internal oscillator and is thus not proportional to the displacement of the piston.

## A - 4.4 Safety tests- Results

Test No. 1

The switching off of the power supply to the vibrator electronic control unit (Fig. No. 13.).

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 5g |
| Duration | 10 milli-secs |
| Velocity Change | .5 metres/sec. |

Test No. 2

The switching on of the power supply to the vibrator electronic control unit (Fig. No. 14.).

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 2g |
| Duration | 20 milli-secs |
| Velocity Change | .4 metres/sec. |

Test No. 3

The vibrator was set such that the piston was moved into the snubber travel, allowed to operated within the snubber and then moved out of the snubber back to normal operation.

The operating frequency for this test had to be changed to a higher frequency (20 Hz) and a lower acceleration level (.5g peak to peak) to allow the piston to operate within the snubber travel. The test showed that the highest acceleration level occured on leaving the snubber travel, both results were, however, recorded and are shown in Figs.No. 15 and 16.

a) On entry to the snubber travel

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 1.35g |
| Duration | 10 milli-secs. |
| Velocity Change | .13 metres/sec. |

b) On leaving snubber travel

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 6g |
| Duration | 12 milli-secs. |
| Velocity Change | .7 metres/sec. |

Test No. 4

Switching off the power supply to the hydraulic power pack, but leaving the control unit to function normally (Fig. No. 17 ).

System returns gently to rest with the piston at the bottom of its travel. The time taken for this to occur varied slightly between 3.5 seconds to 4.0 seconds. The time would depend on the control settings and the load on the piston, with heavy piston loads the shut-down time would be less.

Test No. 5

The testing of the subject operated "emergency stop". The operation of the emergency stop causes the piston to be brought to rest at the mean value setting, by causing the oscillator to decay away exponentially (Fig. No. 18.).

The piston is brought gently to rest in a shut-down time of about 1 to 1.5 seconds, depending on operating frequency and acceleration level.

Test No. 6

Switching off of the command signal, leaving the feed-back amplifier and displacement transducer operating. This was achieved by operating the internal external input switch which has a central off position (Fig. No. 19 ).

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 1.8g |
| Duration | 40 milli-secs |
| Velocity Change | .7 metres/sec. |

Test No. 7

This test was to simulate what would happen if the connectors or the cable failed joining the actuator to the control unit. To avoid breaking individual wires in the cable the simulation was achieved by unplugging the cable from the control unit. The cable connects the moog valve, displacement and pressure transducers to the control unit (Fig No.20.).

Unplugging cable from control unit

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 9.5g |
| Duration | 15 milli-secs |
| Velocity Change | 1.4 metres/sec. |

Test No. 8

Plugging in the cable between the control unit and the actuator. This test, together with test No. 7, would simulate an intermittent connector or cable failure (Fig No. 21. ).

| | |
|---|---|
| Maximum Acceleration (Peak to peak) | 5.6g |
| Duration | 14 milli-secs. |
| Velocity Change | .76 metres/sec. |

Test No. 9

The setting of system gain control to an excessive level, such as to cause the system to become unstable. The system on becoming unstable oscillates at its own natural frequency. (This frequency is independent of the setting of the internal oscillator) (Fig. No. 22).

| | |
|---|---|
| Acceleration level (peak to peak) | 18g |
| Frequency | 65 Hz |

The tests which would involve the simulation of mechanical failure; or tests which, if simulated, might result in mechanical failure to the actuator or its control system, were not carried out due to the risk of them causing premature actuator failure. The probable result of some of the failures can be assessed from the tests carried out, and similar failures that have occured to actuators operating in the fatigue testing of components in industry can provide useful data as to the potential danger of a particular type of failure.

A - 4.5 Discussion of Results

The results of the safety tests were compared with available human tolerance levels to shock and vibration and to the recommendations in "Draft Guide to the Safety of Human Vibration Experiments (Draft 3)".

The Guide gives recommendations as to the levels of shock and vibration to which a subject should be expected to endure in the case of equipment failure. The recommendations for vibration levels are based on the ISO proposals "ISO/TC 108/WG7 (1969). The recommendations for shock levels are obtained from combining the results of many researchers in the field of human tolerance to shock. The draft guide (1969) recommendations are as follows:- From Annexe 2 "Limits of acceptable shock and vibration". Page 16.

"(b) Shock

In the case of equipment failure, any shock transmitted to the subject should be less than the values given below:-

Duration 0.02 seconds. The total velocity change should not exceed 2m/sec.

Duration 0.02 seconds. The mean acceleration/retardation level should not exceed $50 m/sec^2$ (5g) and the rate of rise should be less than 1,000 $m/sec.^2/sec.$ (100g/sec.)

The tolerance levels indicated are approximately half the values obtained in experiments where healthy male subjects were adequately restrained in their seats."

The results from the nine simulated failure modes are tabulated in Table No.2. These results show that for all but one of the tests the simulated failures resulted in ram waveforms being of the impulsive type. These shock waveforms were thus compared with the recommendations of the draft guide (1969).

Comparing the tests results with these limits shows that test numbers 1, 3a, 3b, 7 and 8 have durations of less than 20 milli-seconds and are thus dependent on the total velocity change which in all these cases is less than the "acceptable" 2m/sec. recommended in the draft guide. Test numbers 2 and 6 have durations greater than 20 milli-seconds, so the rate of rise of acceleration was computed for each test. The rate of rise of acceleration for test number 2 was 100g/second and for test number 6 the rate of rise was 45g/second. The "acceptable" rate of rise as recommended in the draft guide was 100g/second, test number 6 gave a rate of rise of less than half this value, for test number 2 the rate of rise was, however, the same as the recommended maximum.

Test number 2 involved switching on of the power supply to the actuator unit, this situation could be avoided if a switch were fitted to the signal amplitude of the control unit such that the amplitude had to be set to zero before the signal could be applied to the moog valve of the actuator. The switch fitted to the amplitude potentiometer would operate a relay, so that a power failure, or accidental switching on of the control-unit by the operator would not cause a large amplitude signal to be fed to the moog valve. This modification for the internal oascillator was fitted after the results of the simulated failure modes had been assessed, and was later modified to function with the external input as well. (See section A - 5).

The only simulated failure mode to result in a sinusoidal output was that of test number 9 - the system going into unstable operation. The frequency obtained was the natural frequency of the entire system, the electronic control and the mechanical actuator. The frequency and the acceleration level obtained during this test are shown plotted on the I.S.O. curves in Fig. No. 23.

The curves show that the level obtained when this failure occured would be in excess of the I.S.O. 1 min. exposure limit, the system operated shut-down or switching off the hydraulic power results in a shut-down time of less than ten seconds. This short time could, however, result in injury to certain subjects, especially where no restraint harness is used.

The limits to shock, quoted earlier in this section, apply to subjects using restraint harnesses, this is not, however, practical in some experiments and is impossible in impedance or transmissability studies, as the harness would greatly influence the results, and in many cases invalidate them. The tests were carried out with the actuator unloaded, but even with a subject loading the actuator piston the acceleration levels would not be reduced to less than 2g (peak to peak) thus the subject would still be in danger of leaving the actuator platform in its subsequent motion. In experiments where restraint harness cannot be used without invalidating the result, the experimental apparatus should be designed in such a way that the subject is free to leave the actuator platform in case of an accident or severe vibration levels. The use of a safety harness has to be evaulated by the researcher for individual experiments. Subject safety should be an integral part of all experimental design (see Conclusions - Section A - 6.)

## A - 5 Recommendations for modifications to improve safety.

### A - 5.1 Introduction

The following list of modifications was drawn up using information from three sources. The three sources were:-

a) Information obtained from the safety tests carried out on the vibrator unit (Section A - 4)

b) Reports from operators and subjects working with the actuator of accident or near-accident situations.

c) Consultation with experimenters in other institutions and industry using similar equipment.

The list of modifications given is a full list and contains some modifications which are at present technologically difficult or very expensive; these have been included for the sake of completeness. After each potential failure are suggestions as to possible modifications which would improve subject safety with regard to that failure. The section also notes why these failures are potentially dangerous, and in some cases how these potentially dangerous situations were noticed.

A glossary of terms used in connection with the actuator are given in Section A - 5.2. This is to avoid any confusion between the use of the terms in this context and in the context of control theory.

### A - 5.2 Glossary of terms used in conjection with electro-hydraulic actuators

Automatic Control - Allows the actuator to operate as a constant displacement or acceleration vibrator. The switch has three positions "displacement-off-acceleration" to allow for manual operation of the amplitude control if required.

Dump Valve - An electrically operated valve which is situated across the inlet and outlet parts of the actuator, when operated it short-circuits the actuator and "dumps" the high pressure oil supply back to the hydraulic power pack.

Emergency Stop - A set of stop switches which when operated cause the control signal to decay away slowly which brings the actuator piston gently to rest.

External Input - The actuator operating from an external signal source, such as a tape-recording of field conditions

Internal Input - The actuator operating by means of its own internal oscillator.

Internal-External Switch - Enables the operator to switch from the internal oscillator driving the actuator, to an external control signal.

Mean Value - The operation of the mean value control adjusts the height of the piston above the main actuator body, and allows the operator to compensate for differing piston loads.

Snubbers - These are pockets of oil at the top and bottom of the pistons overall stroke which prevent the piston crashing into the main actuator body. The piston on entering the snubber compresses the trapped oil, which is allowed to leak away at a given rate through a pre-set relief valve to the oil return line.

System Gain - Controls the overall gain of system. Two small pre-set gain controls are also situated inside the control unit to set the individual gains when the system is operating on acceleration or displacement feedback.

A - 5.3. Modifications - Suggested List

a) An "emergency stop" provision similar to that fitted to the"internal" input should be fitted to the "external" input.

This was one of the most potentially dangerous parts of the system and could be easily modified to afford to external input signals, the same shut-down facility as that fitted to the internal oscillator.

b) The switching arrangements on the "int-ext" switch should be such that on switching from one signal input to another the amplitude potentiometer has to be reset to zero before the new signal will actuate the vibrator. This would eliminate the danger of the operator switching from a small "internal" signal to a large "external" signal or vice versa.

A modification which could be carried out easily and at low cost would be the fitting of a relay. The unit could also be fitted with an override switch at the back of the control consul to enable the operator to switch directly from "internal" to "external" and back if required. This switching between the two inputs may be required in an experiment comparing subjectively random and sinusoidal type vibrations, or any other similar comparative experiments.

c) The fitting of limit switches on the actuator piston. This could be achieved by electronic switches incorporated into the present displacement transducer system. The use of some form of limit switches would have several advantages. The switches could be set to stop the actuator if its displacement amplitude became too large for the given operating frequency. The limit switches would also stop the actuator before the piston could travel from end to end and prevent the piston from running in and out of the snubbers. Another way to stop this type of failure would be to install pressure sensing switches on the snubbers, to stop the actuator if the piston entered the snubber travel.

d) Mains failure - the fitting of an electronic device to protect the subject against mains failure or an accidental switch off of the supply to the control console.

This modication can be achieved and would also need a power storage unit or battery to allow the actuator to run down smoothly. This was found to be necessary during the safety test carried out on the actuator after its installation.

e) Cable failure - this would be the breakage of the cable or failure of the connectors at either end of the cable.

This cannot be completely protected against but the selection of high quality cable and connectors, together with regular inspection of the cable for cuts or squashing, can minimise the risk of cable failure.

f) A means of restricting the rate at which the frequency can be increased on the internal signal generator.

This can be achieved easily by fitting a braking device to the existing frequency change potentiometer, thus restricting the speed at which the frequency can be increased, and allowing the subject to stop the experiment before the vibration level becomes intolerable.

g) The pre-set gain control should be made such that it cannot be operated as easily as it can be in its present position.

This could be achieved in several ways which would mean that the actuator could not be set into unstable operation by an inexperienced operator or some-one moving the controls accidentally. The gain potentiometer could be moved to the side of the panel or could be set behind a perspex cover, which would only allow movement of the potentiometer by screwdriver.

h) The incorporation of the hydraulic stop button on the main control consul and the hydraulic power being switched together with the power supply to the electronic control unit, by a key switch (on the actuator control panel).

These would also be easily incorporated, the hydaulic stop button would be better sited on the main control panel so that the operator has full control of the hydraulic power as well as the electronic control unit. The hydraulic stop provides an effective way of bringing the unit to rest smoothly, and would have to be used if the electronic emergency stop failed.

i) The fitting of a foot-operated emergency stop button for the subject, for use when he is performing a task which requires the use of both hands (e.g. tracking) or where the arms have to be fixed to maintain a set posture.

This modification could be easily fitted as the emergency stop only requires a simple on-off switch of the push button type. The switch operates a relay so once it has been pressed the control unit brings the actuator gently to rest. The need for this modification was noted during a series of experiments by a post-graduate using a tracking situation. During these experiments the existing hand held push-button stop was fixed to the tracking task joy-stick box, which proved satisfactory for these experiments but would not be for a more complex tracking task.

j) The author also recommended a re-design of the main control panel, see Fig No. 24

The design of the panel was to be submitted to the manufacturers as a possible suggestion which their designers could use as a basis for the new control panel.

k) An indication on the mean value potentiometer. (This is the potentiometer which sets the actuator piston position about which it then oscillates).

This would just require the etching onto the panel of two arrows with the words "up" and "down". The indication of movement direction is required especially if one is working close to one end of the total piston travel, where an incorrect movement of the control by the operator would send the piston into the snubber. This control has to be operated with the subject on the actuator as subject weight affects the mean-value.

i) Mechanical failure of actuator, servo-valve or other associated parts.

This type of failure cannot be completely guarded against, only the chances of it occuring minimised. The use of generous factors of safety in the design of the actuator and the use of high quality components in its construction, together with careful operation, regular maintenance and inspection of the mechanical parts of the actuator, all contribute to the system's overall safety.

The use of an hydraulic "dump-valve" can in certain cases save the subject from serious injury due to mechanical failure of the servo-valve, this does, however, depend on the response time of the dump-valve from when it is set into operation. The response time of the dump-valve system is a critical factor here since any mechanical failure of the servo-valve is likely to set the piston in motion at a high velocity towards one end of its travel. It could be found that stopping the hydraulic pump would bring the system to rest in a comparable time to a dump-valve system, with the added advantage that the oil pressure drops steadily and not suddenly as it would if a dump-valve were used. This sudden drop in pressure would cause the ram to move suddenly and shock the subject. The sudden drop in pressure caused by a dump-valve system was one of the main reasons why the Loughborough University actuator was fitted with an electronic emergency stop device, which brings the actuator piston gently and smoothly to rest. The subject thus feels confident to use the stop whenever he feels unsure about the vibration level or any other part of the experiment.

### A - 5.4. Modifications to the vibrator

The following modifications were carried out after this study, as a result of the list of suggested modifications (Section A - 5.3.)

a) Mains failure protection. This device causes the actuator to run down smoothly and does not lead to a sudden stop.

b) Displacement cut outs. These can be set to bring the actuator to rest if it exceeds a preset maximum displacement.

c) The hydraulic power supply is governed by the control unit, so that the hydraulic power cannot be switched on until the control unit is in operation.

d) The external input to the control unit is now protected by all the safety devices which previously only used to operate with the unit's internal oscillator.

e) A pressure sensing switch for hydraulic oil pressure has been fitted so that if the pressure falls off, the system is brought to rest.

f) Although not related directly to subject safety, the control console has been re-designed. A good design for the console will, however, make operation easier for the actuator operator which will in turn make the experiment safer for the subject.

g) An R.M.S. acceleration level meter has been fitted together with a built in accelerometer. This will enable the operator to monitor the acceleration level of vibration experienced by the subject. This is particularly useful for the monitoring of random vibration.

h) The external input can now be used to reproduce a recorded acceleration signal, previously a displacement signal had to be used. This makes it much easier to reproduce "field"conditions in the laboratory.

## A - 6. Conclusions

The safety of a subject in an experiment involving whole body vibration depends not only on the vibration equipment but on the task or tests the subject is performing or having performed upon him during the course of the experiment. The use of electrodes attached to the subject, or tests involving senses (i.e. the effects of vibration on hearing, eyesight and speech) must also be taken into account by the experimenter. The use of an auditory warning of subject discomfort, for example in an experiment investigating the combined effects of noise and vibration close to tolerance levels, may not be heard by the operator (or experimenter) until the subject has suffered temporary or permanent damage to his person.

Thus each individual experiment needs to be analysed in respect to its own inherent safety risks, as well as the risks involved in any general whole body vibration experimentation.

The reporting of safety precautions taken during vibration experiments has in the past been omitted from most research papers, this has made it difficult for experimenters to find data about safety precautions taken by previous researchers, and also what effect considerations for the subject's safety have had on the design of the experiment, and the apparatus used. In a field of research which is potentially dangerous to subjects taking part, the researcher needs all the information he can find with respect to the design of safer apparatus and experiments. The reporting of all safety aspects of an experimental project and its apparatus could therefore be as important to a future researcher as the actual results of the experiment, and it is a great pity that such a gap exists in previous reporting. It is felt, however, that with the current national and international interest in the safety of the individual at work, as well as in experimental situations, this gap in our knowledge of whole body vibration research will soon be filled.

The researcher has a responsibility to other researchers and future subjects to see that the experiment and the vibration apparatus are designed in such a way as to minimise the risk of accident to the subject and that these details are reported. The researcher has therefore to carry out a detailed study of the experiment with respect to safety, by a systems analysis or any other similar technique. He can then, using the data from his analysis, see if the experimental programme he has designed can be used on a large number of subjects, or if not what modifications need to be made to the experiment, or, to the sample population of subjects to be used in order to minimise the risk of injury.

The reporting in the future of the safety aspects of experiments as well as just of their results, together with the increase in the knowledge of man's reaction to the vibration environment should ensure that safer experiments and vibration apparatus can be designed which will advance knowledge of the human body as well as benefitting the subjects taking part in future research experiments.

# FIGURES and TABLES

for

PART A

Guignard J.C. and Irving A.

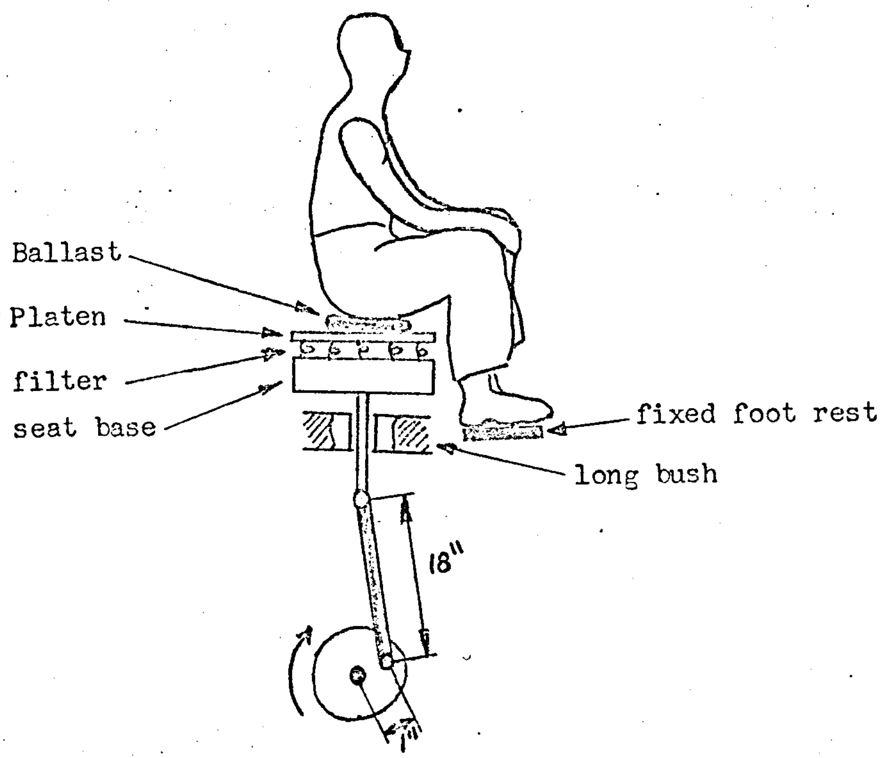



Figure No.

Seat - Head Transmissibility

Phase Angle

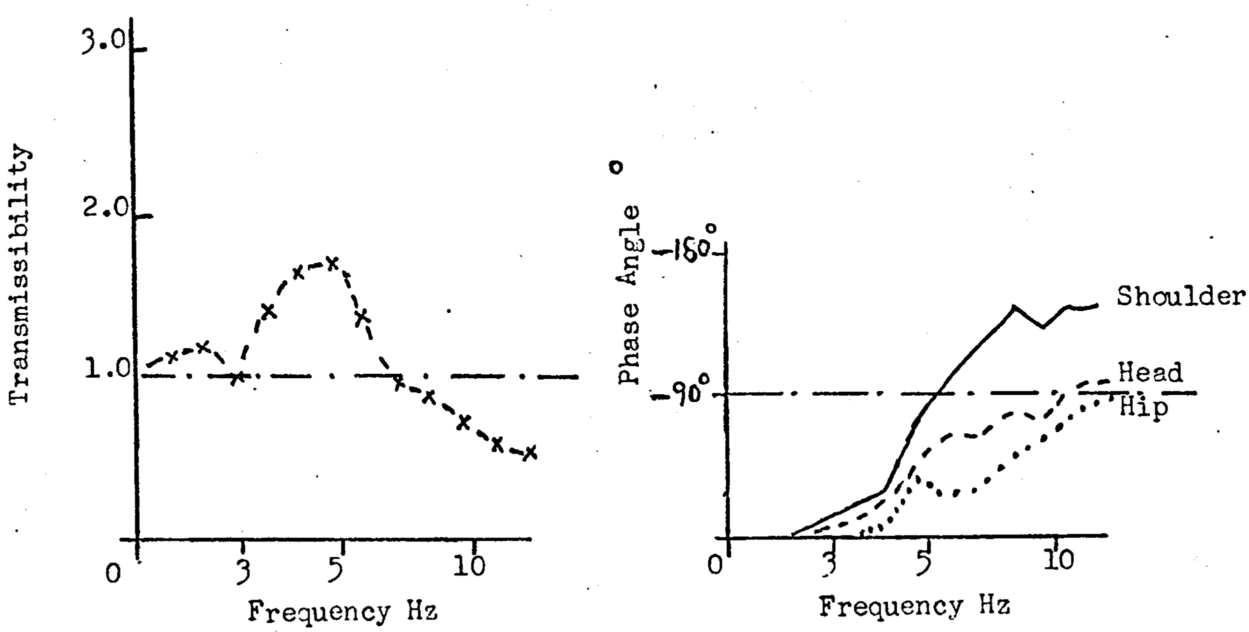



Figure No. 2

Slider-Crank Mechanism

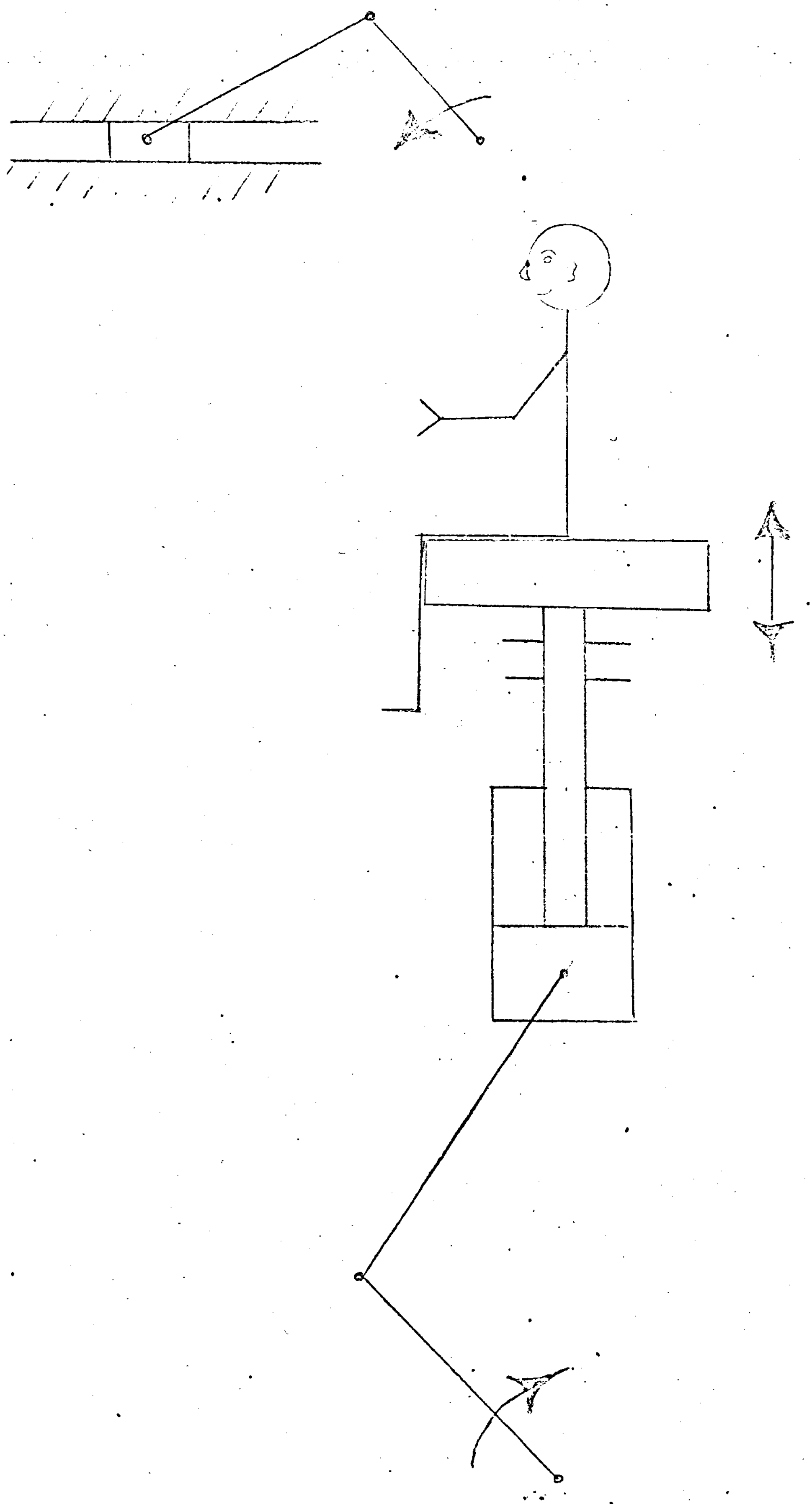


Figure No 3

Scotch Yoke Mechanism

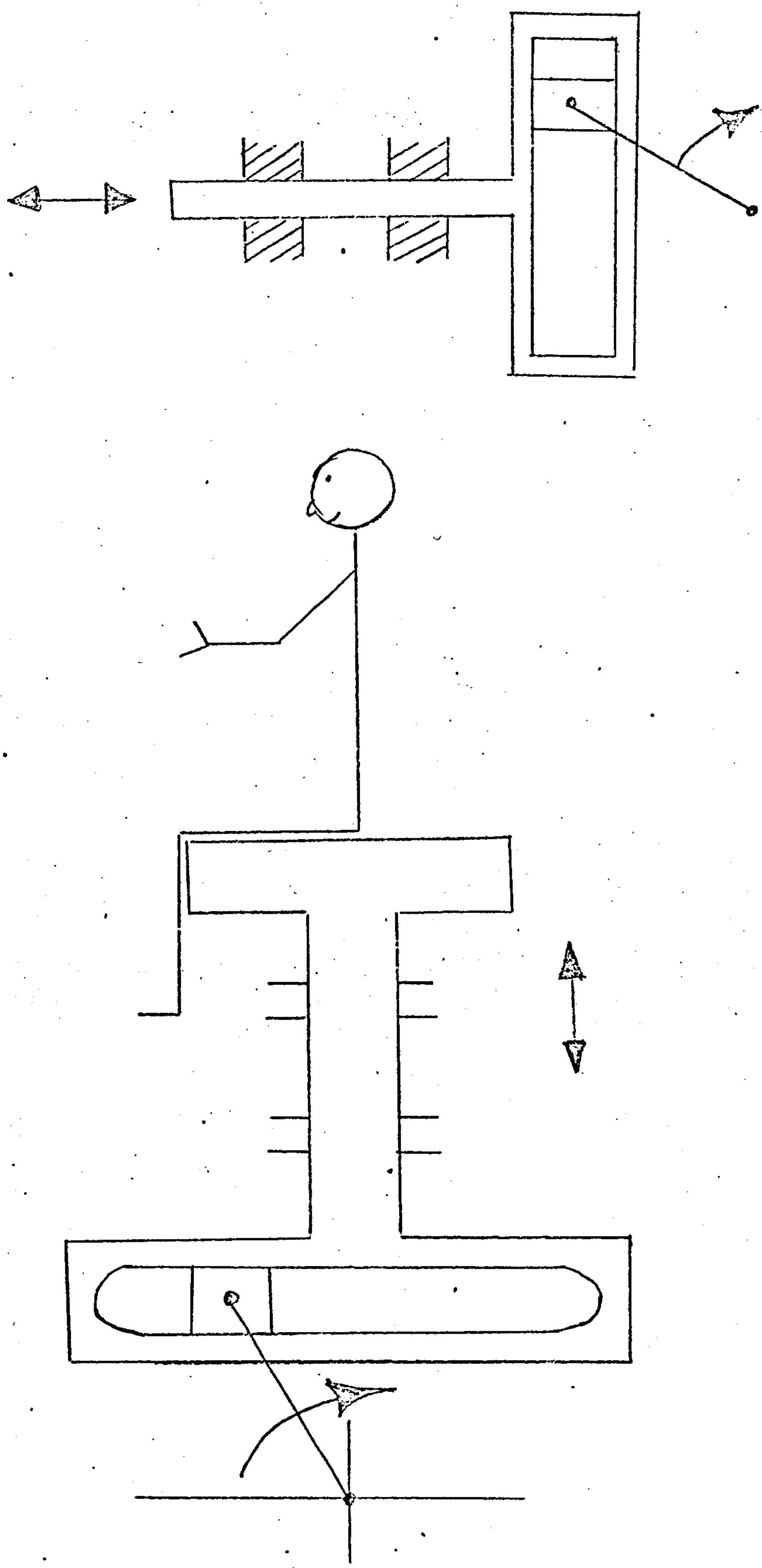


Figure No 4

Cam-follower Mechanism

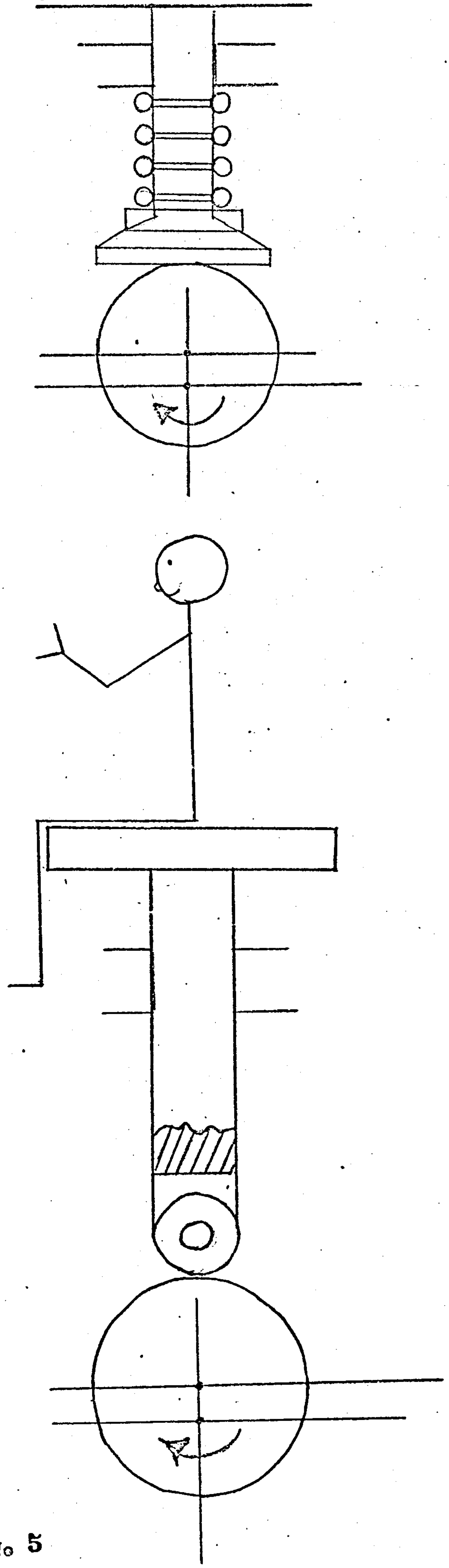


Figure No 5

Hydro-Mechanical Actuator

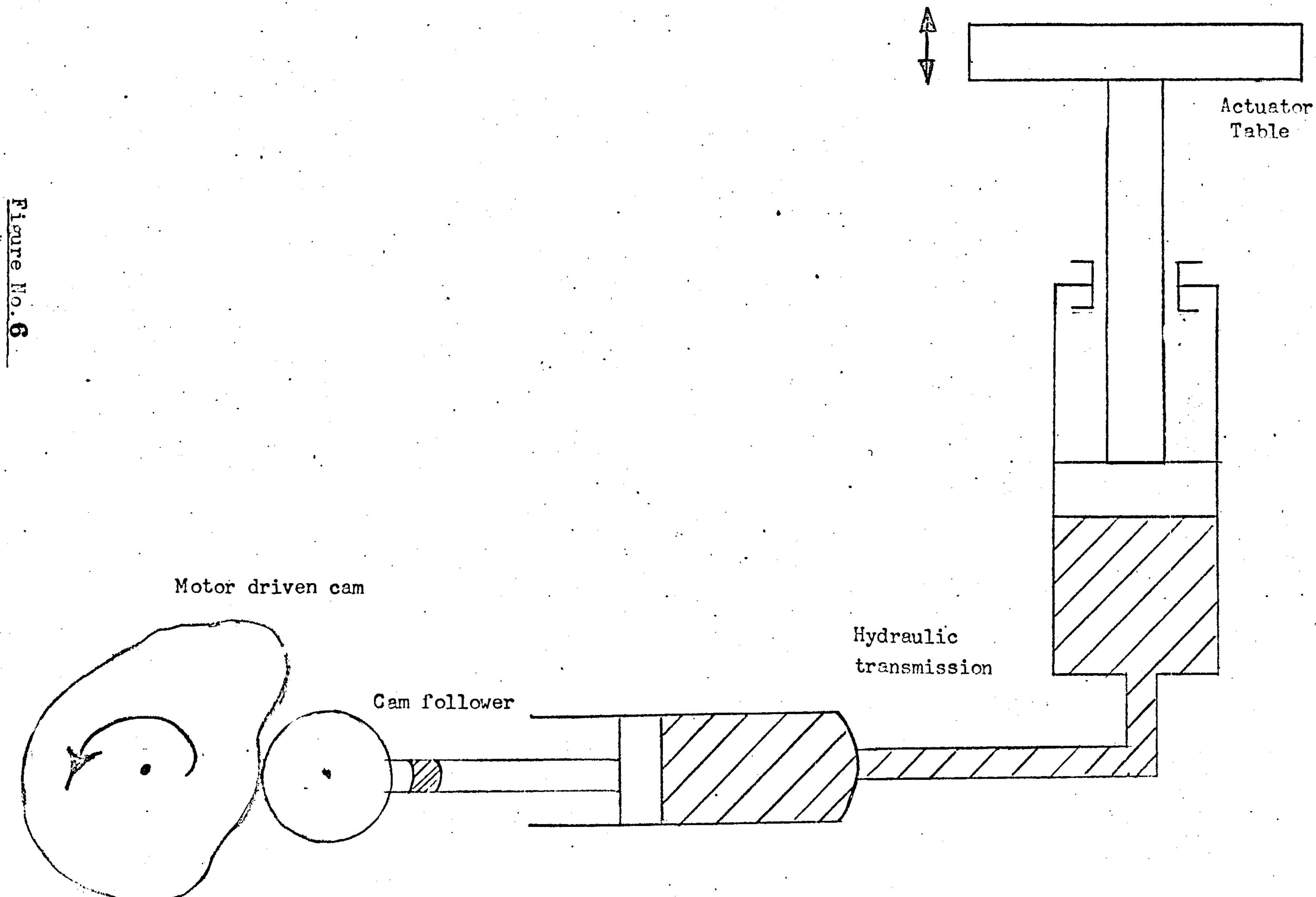
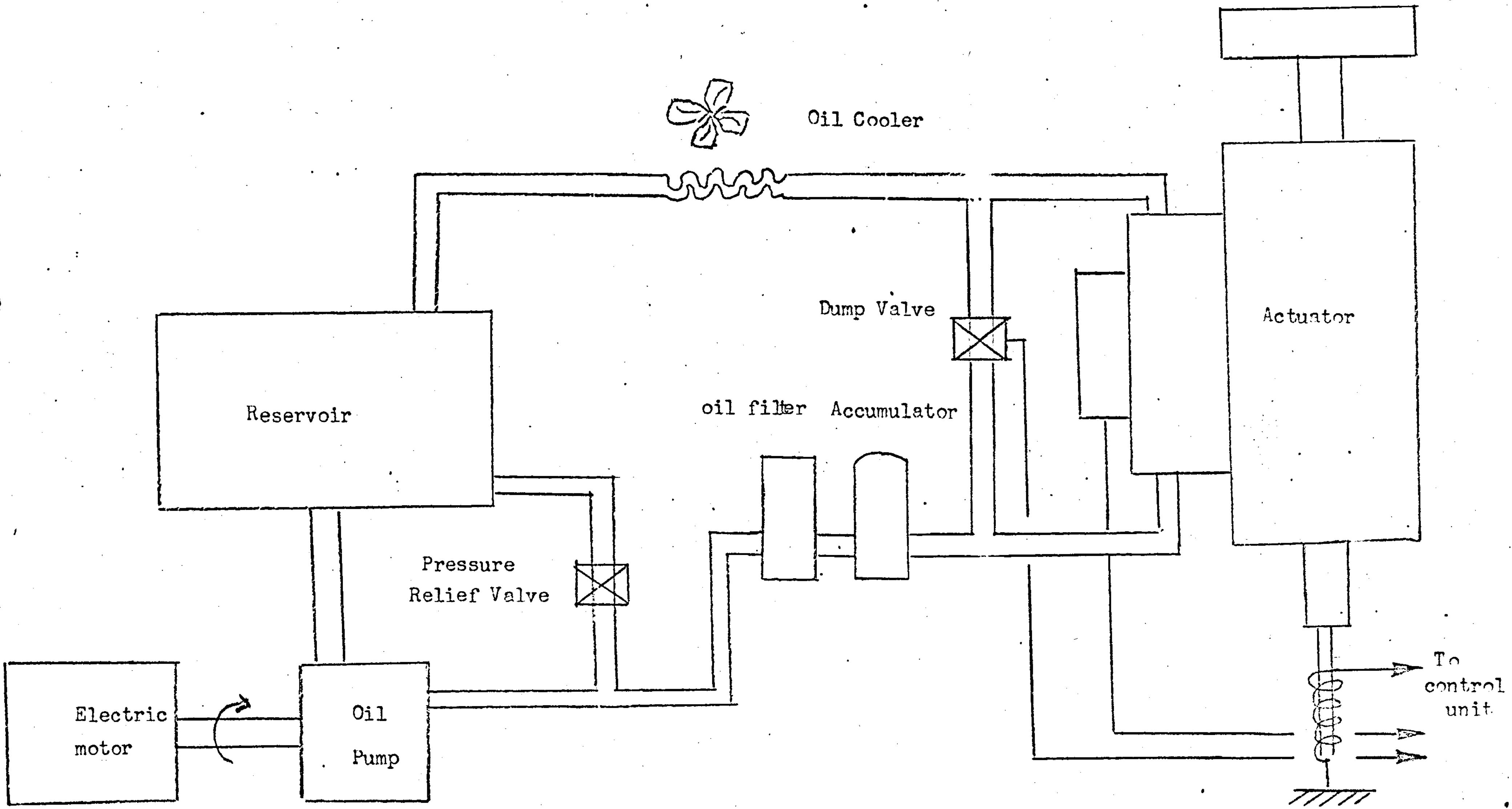



Figure No. 6

Figure No. 7

Hydraulic Power-Pack



Electro-Hydraulic Actuator

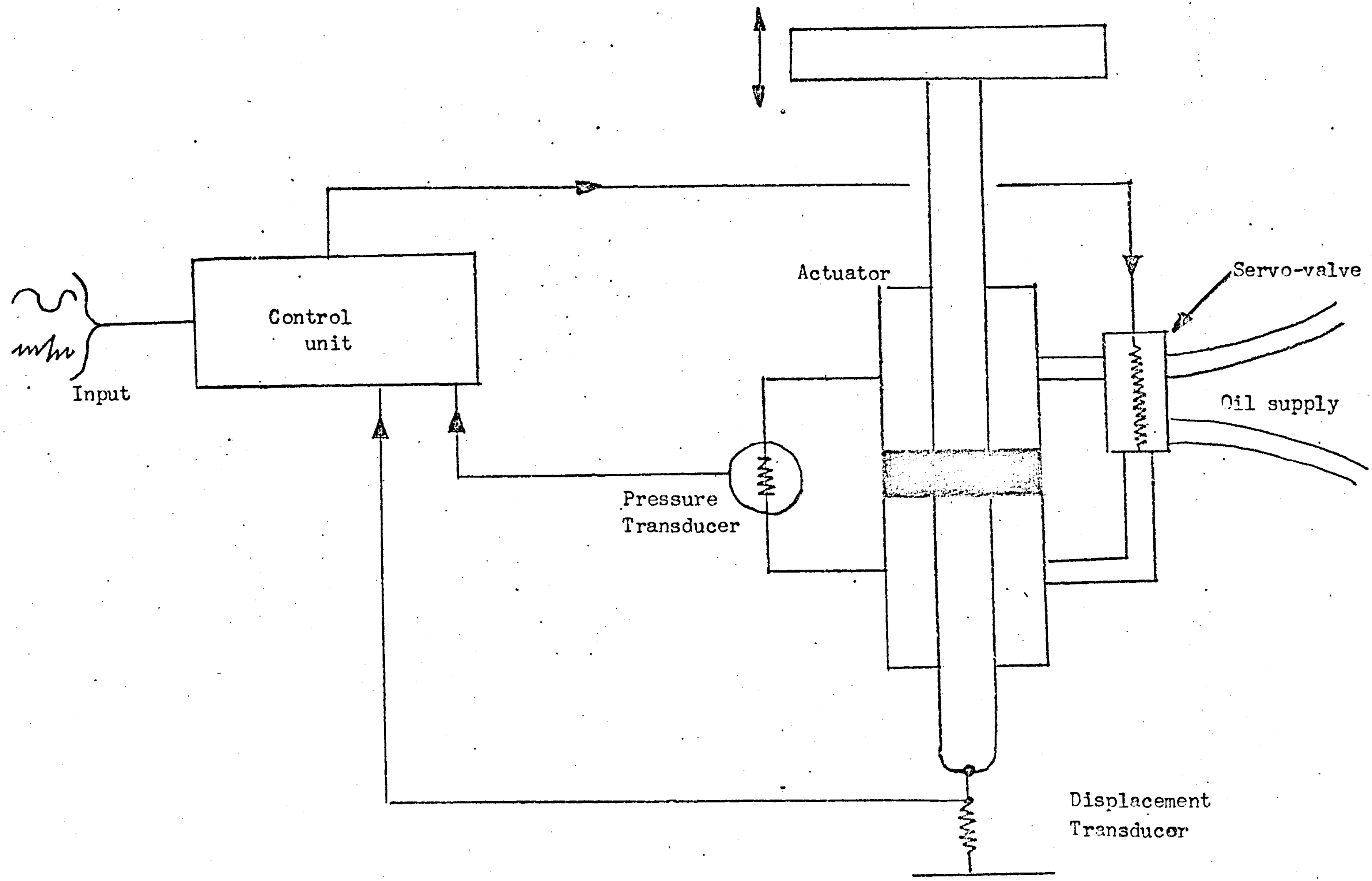



Figure No. 8

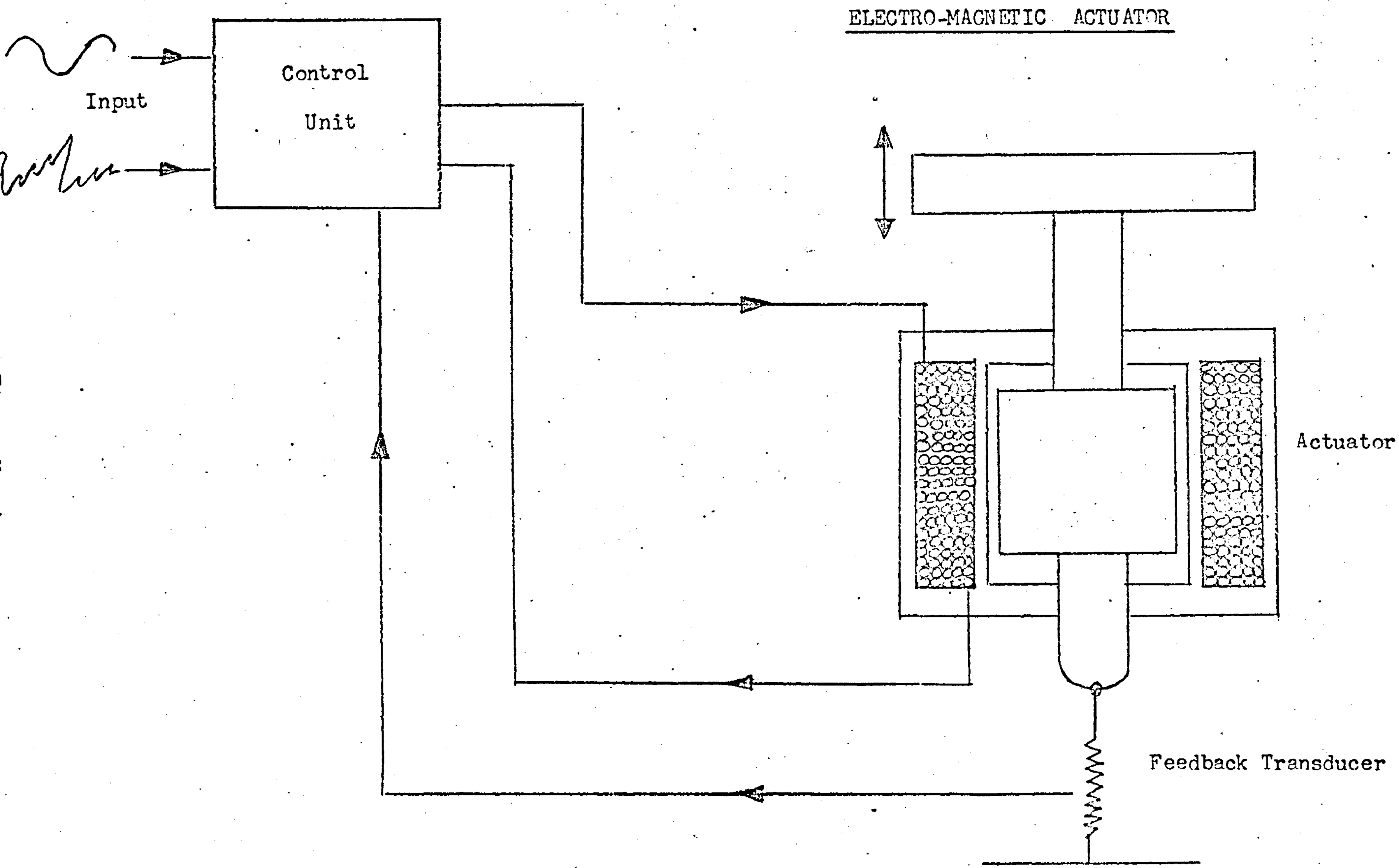

ELECTRO-MAGNETIC ACTUATOR
Input
Control
Unit
Actuator
Feedback Transducer

Figure No. 9

Vibrator Envelope

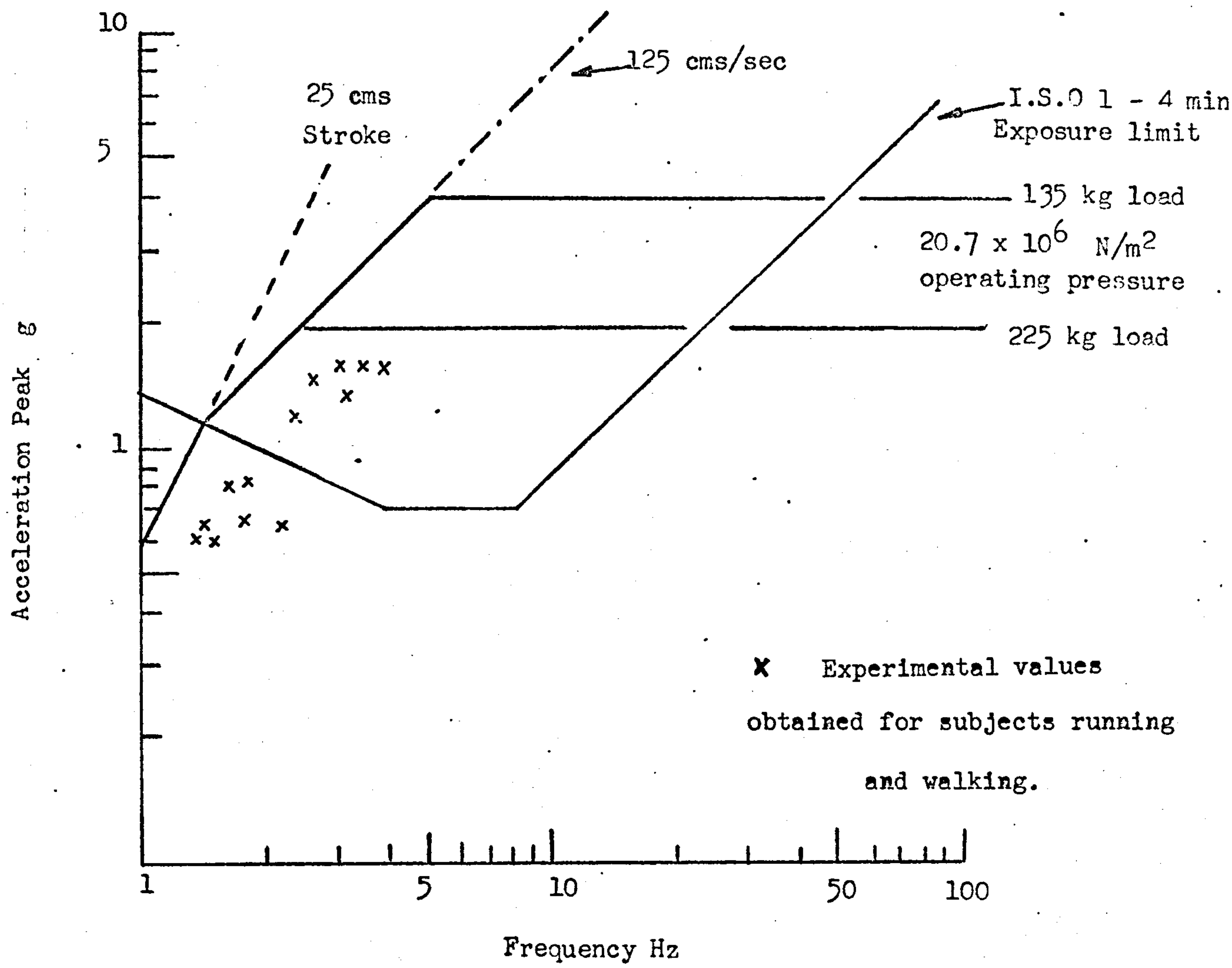



Figure No. 10

Laboratory Plan

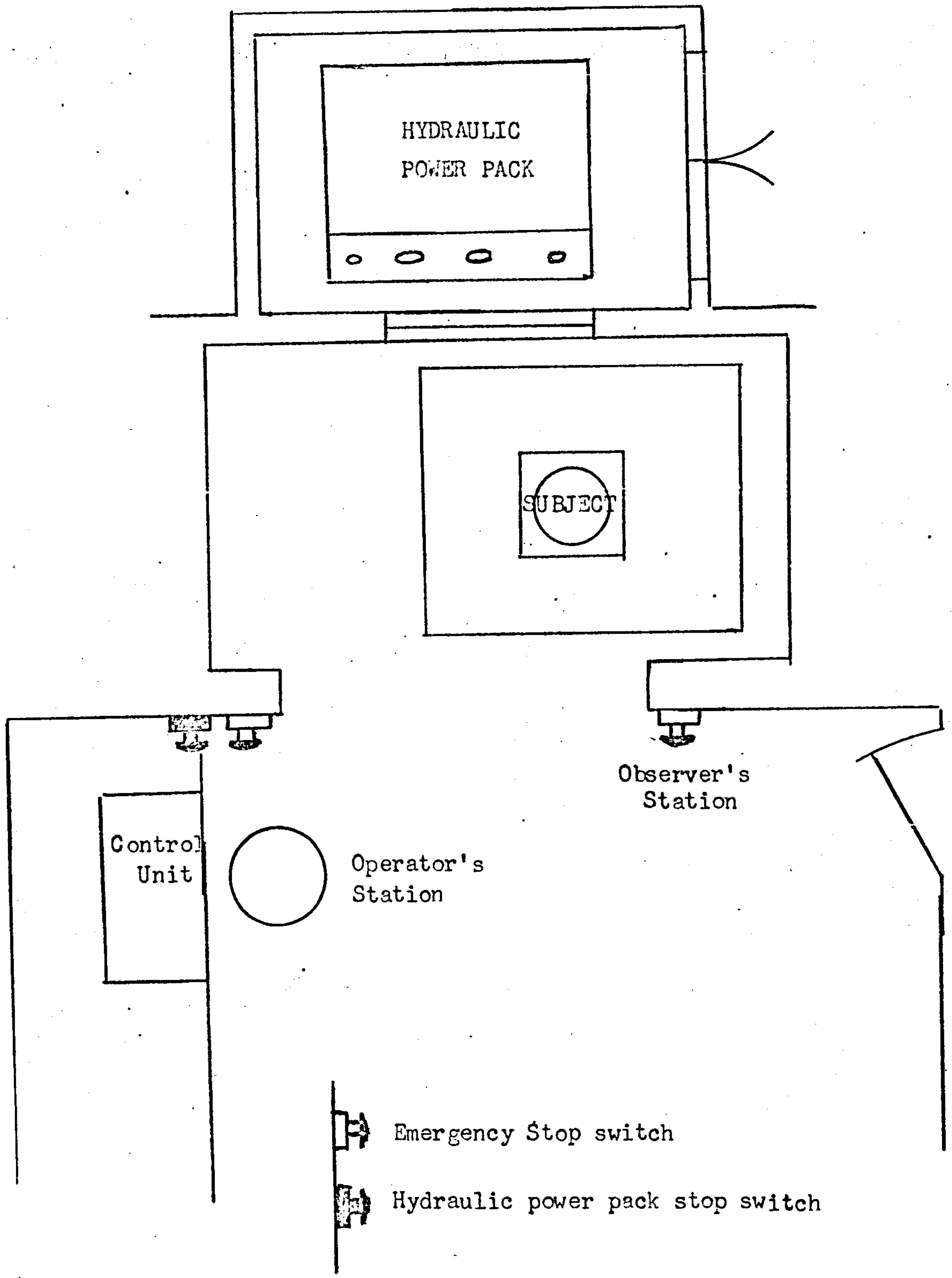
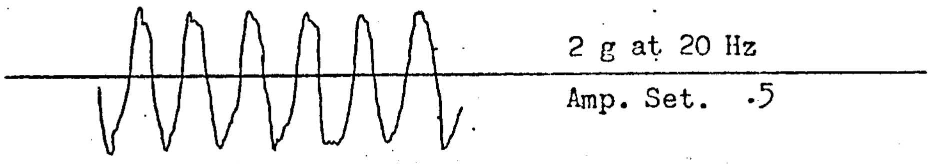
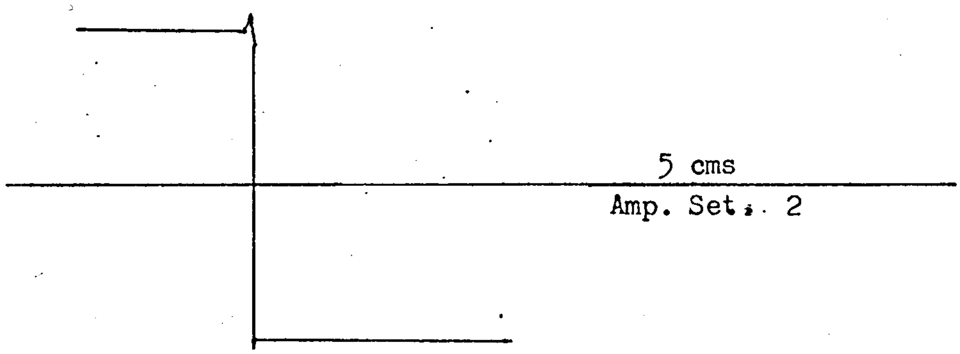



Figure No. 11

Calibration




Calibration of 1g/cm




Calibration of 1cm/1·2 cms

Figure No. 12

Switching "OFF"

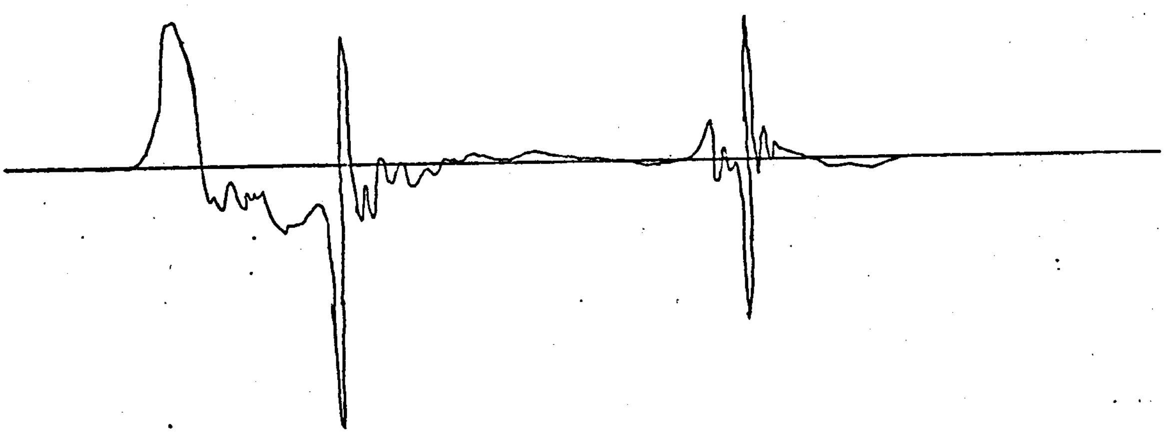


Figure No. 13

Switching "ON"

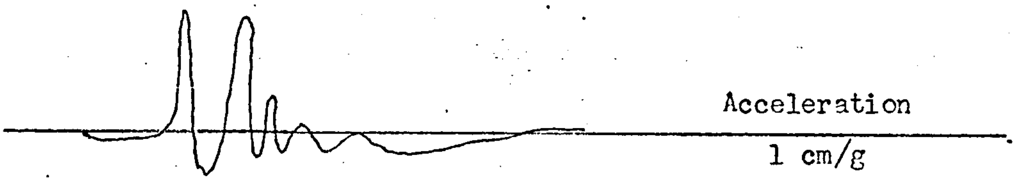



Figure No. 14

Piston entering snubber travel

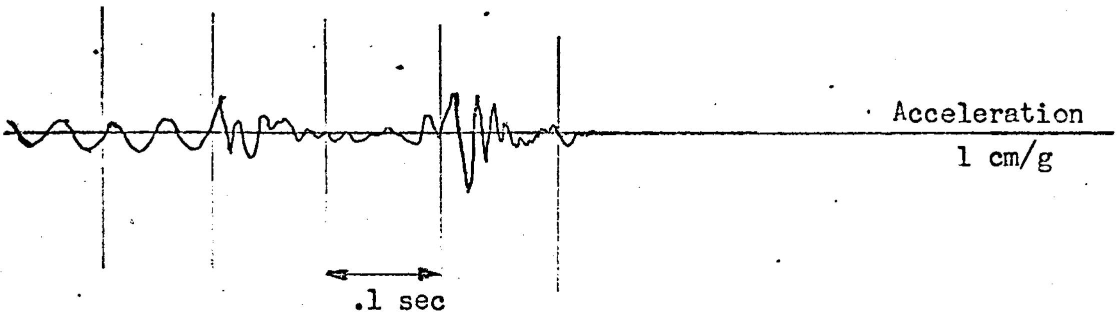



Figure No. 15

Piston leaving snubber travel

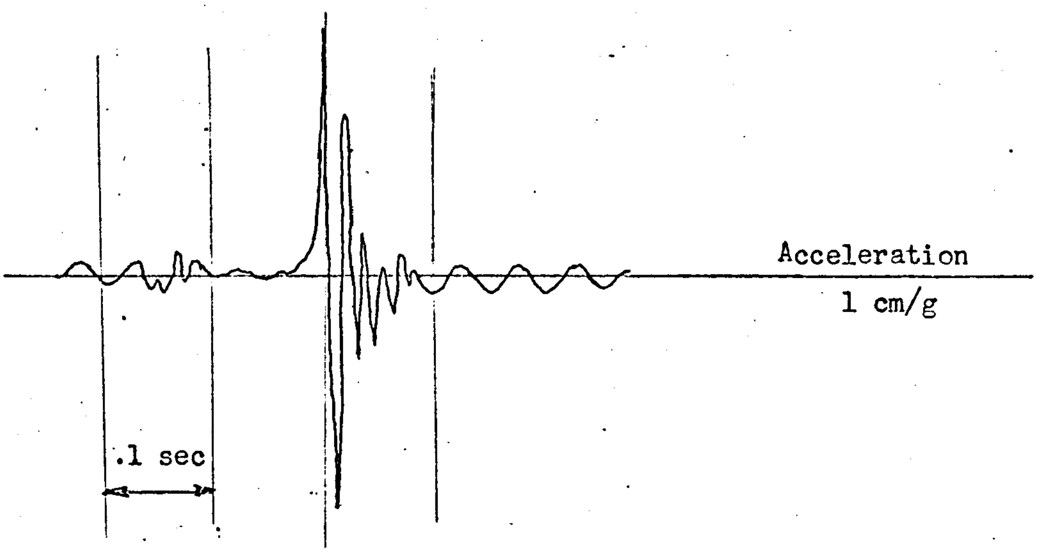



Figure No. 16

Hydraulic Stop

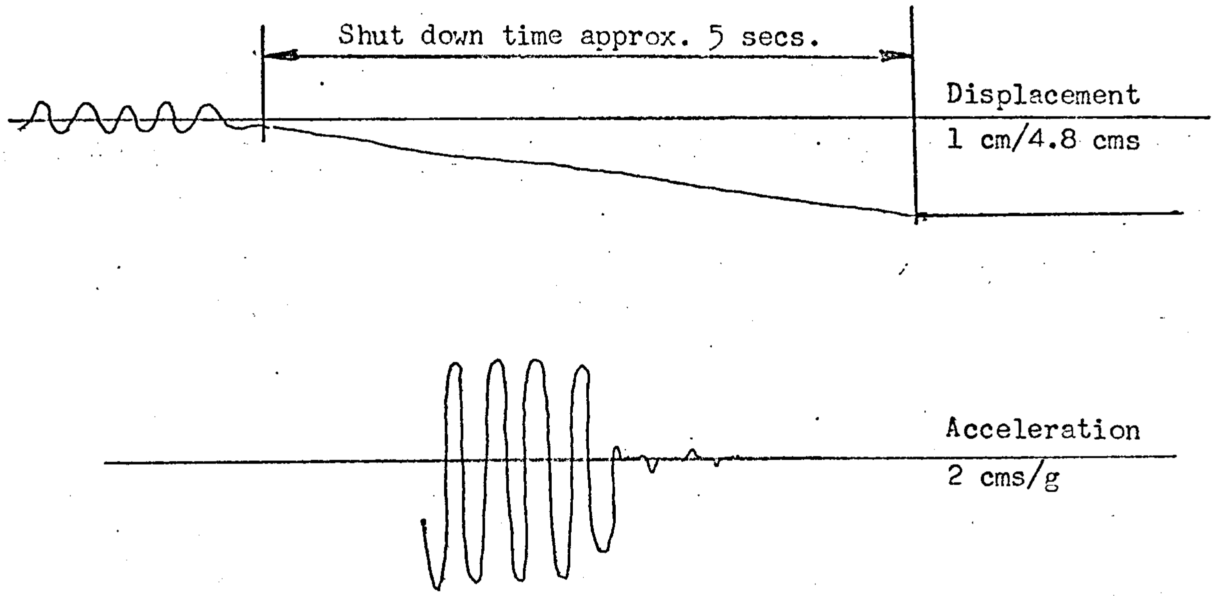



Figure No. 17

Emergency Stop

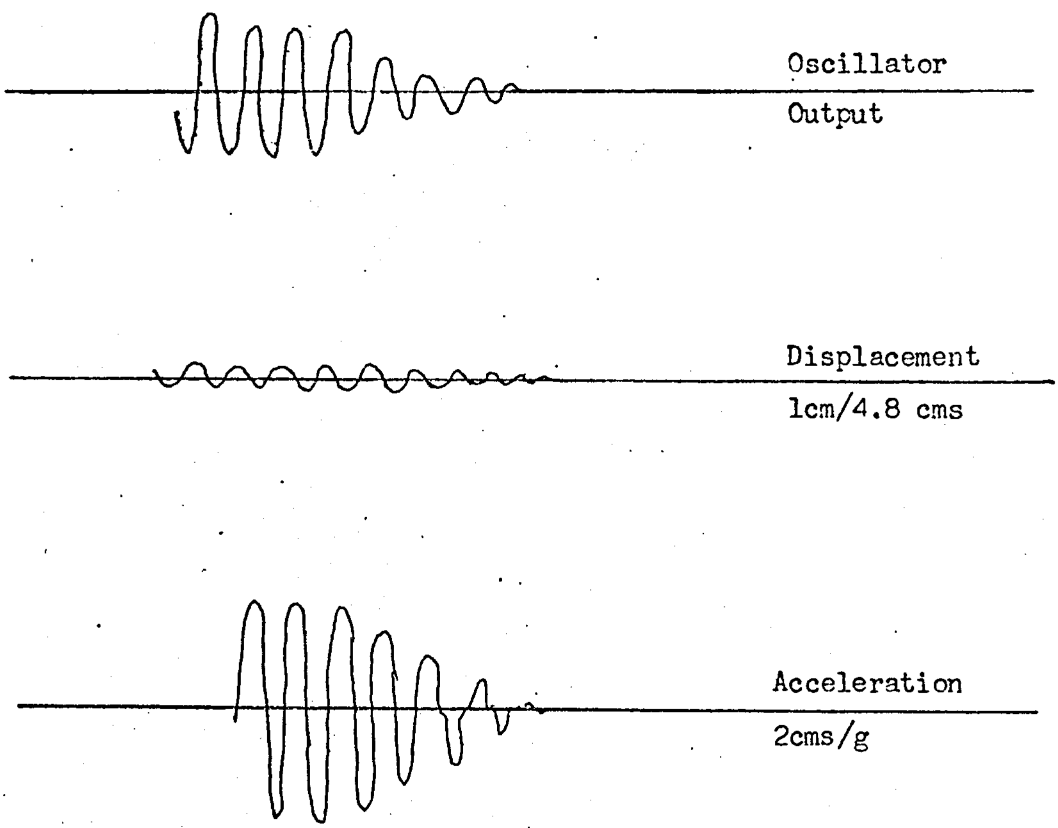



Figure No. 18

Switching from internal input to "off"

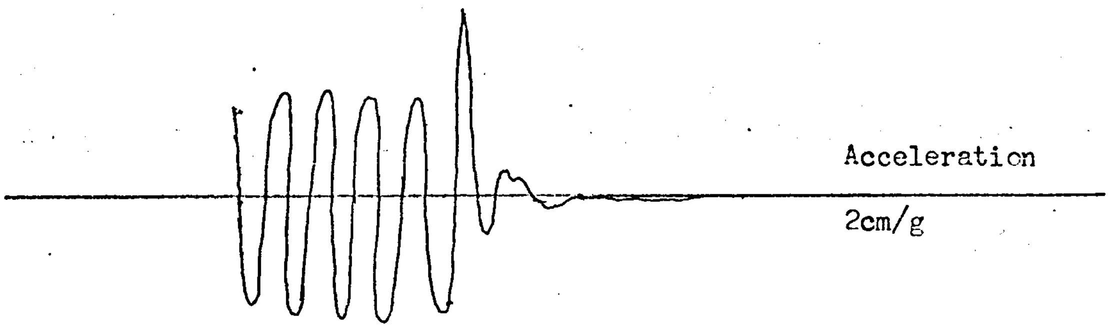



Figure No. 19

Unplugging cable from control unit to actuator

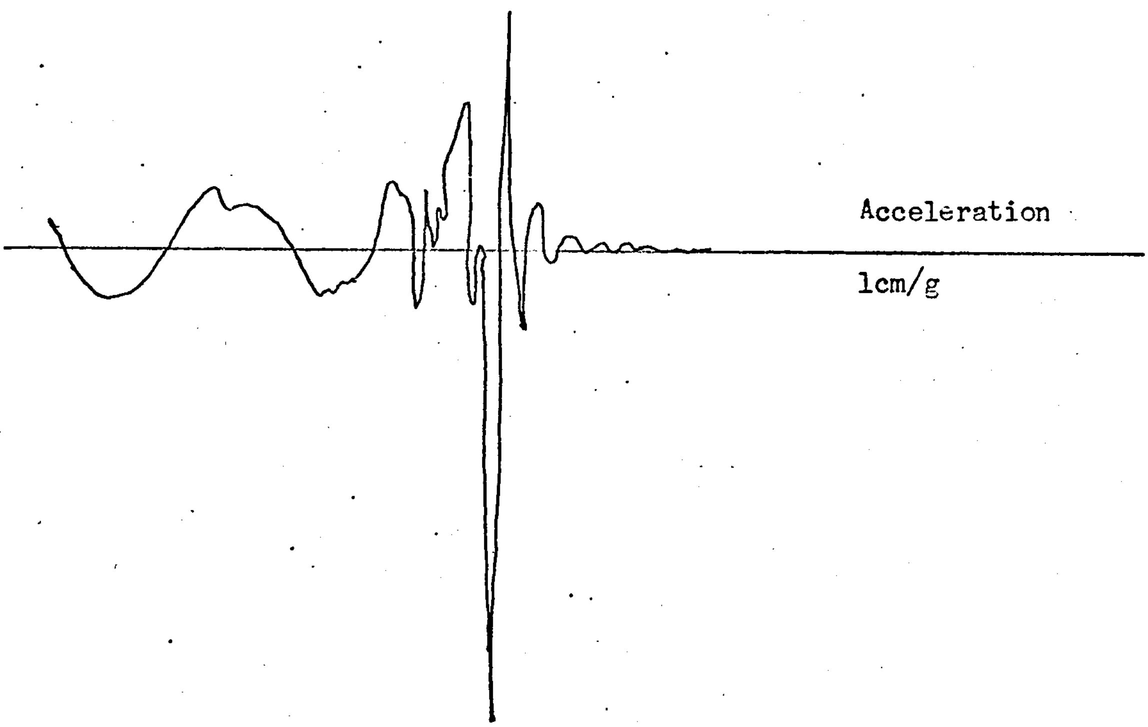



Figure No. 20

Plugging in cable from control unit to actuator

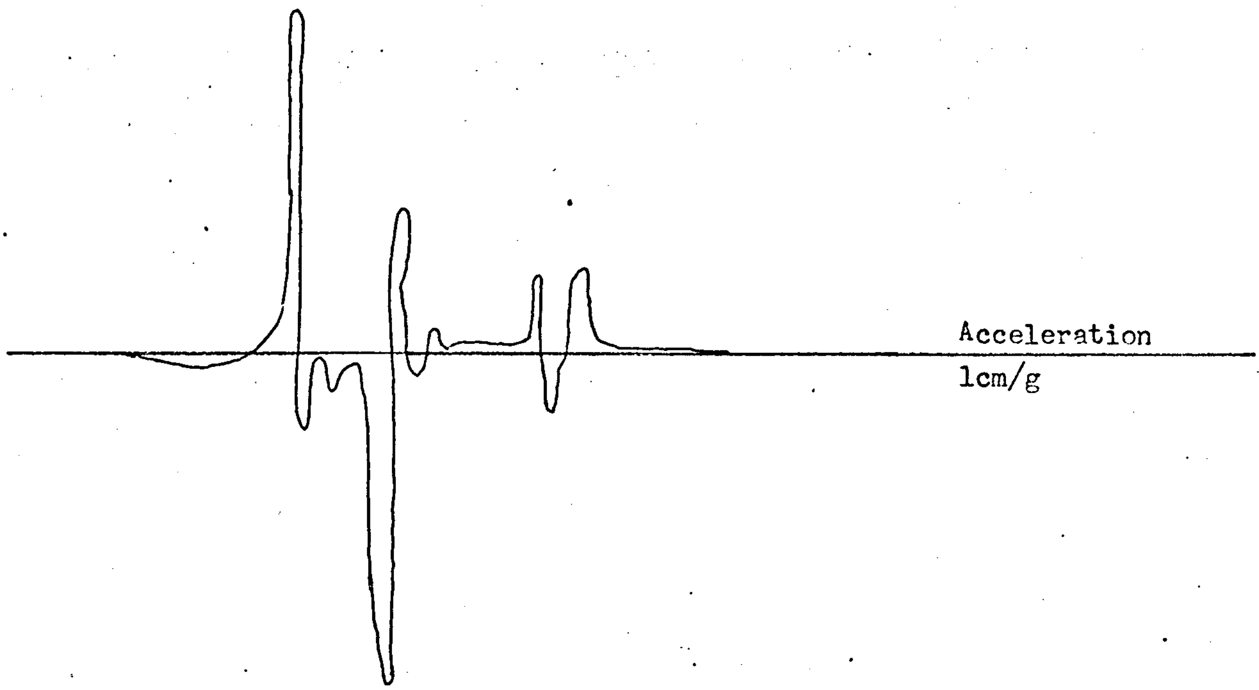



Figure No. 21

System Unstable

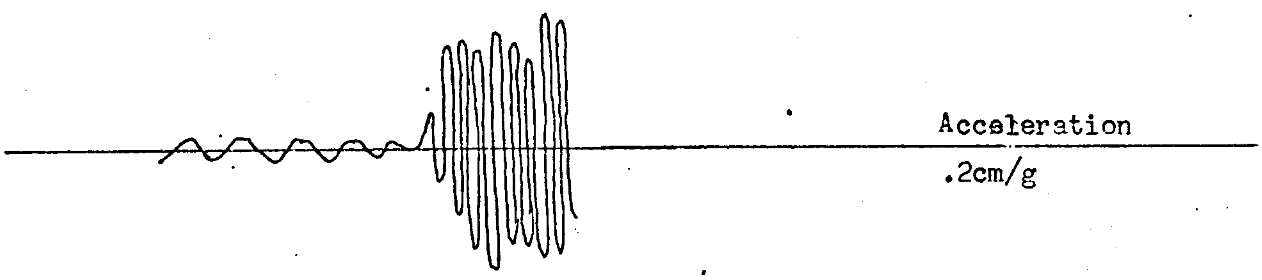



Figure No. 22

System Unstable (Test No. 9)

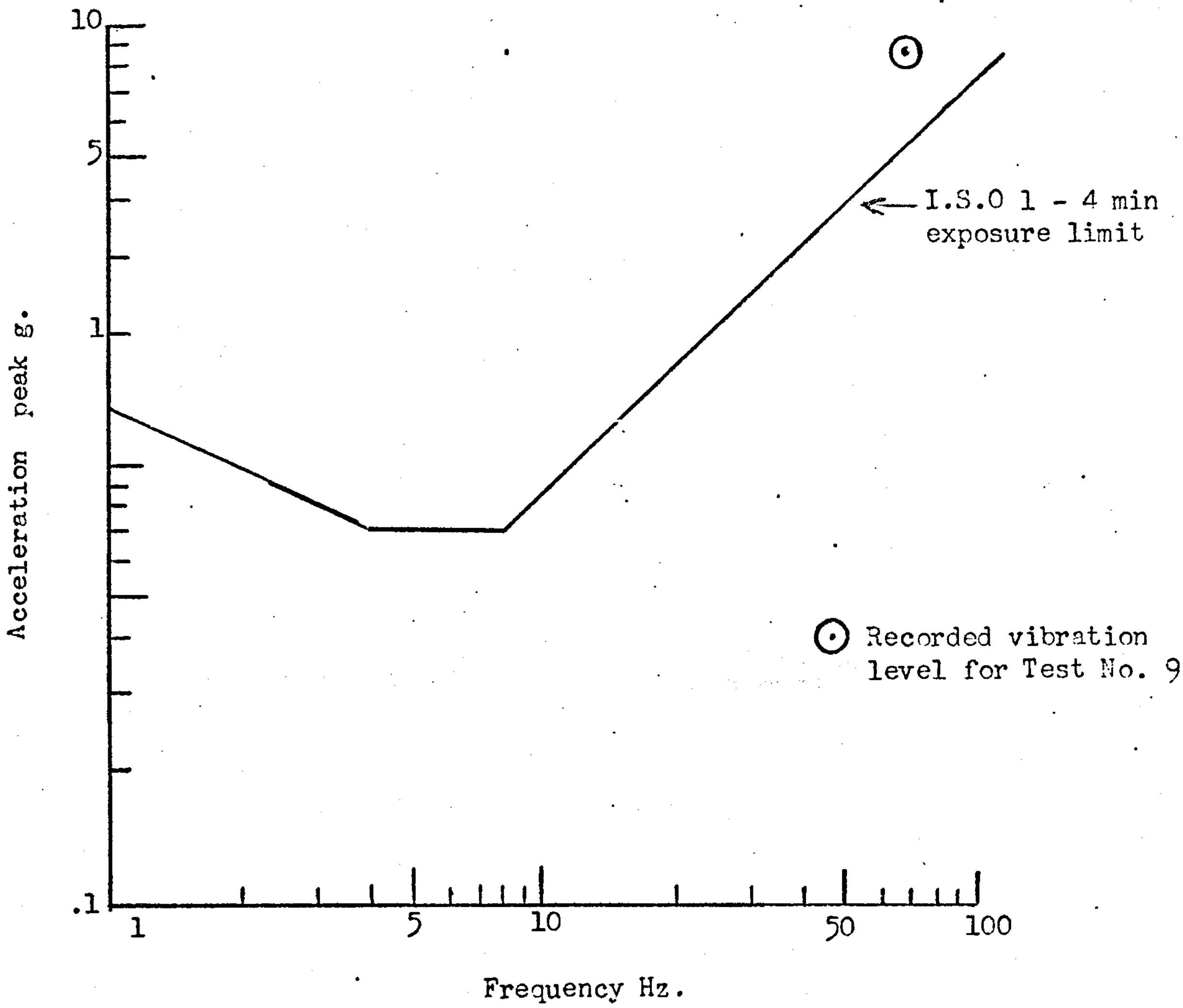
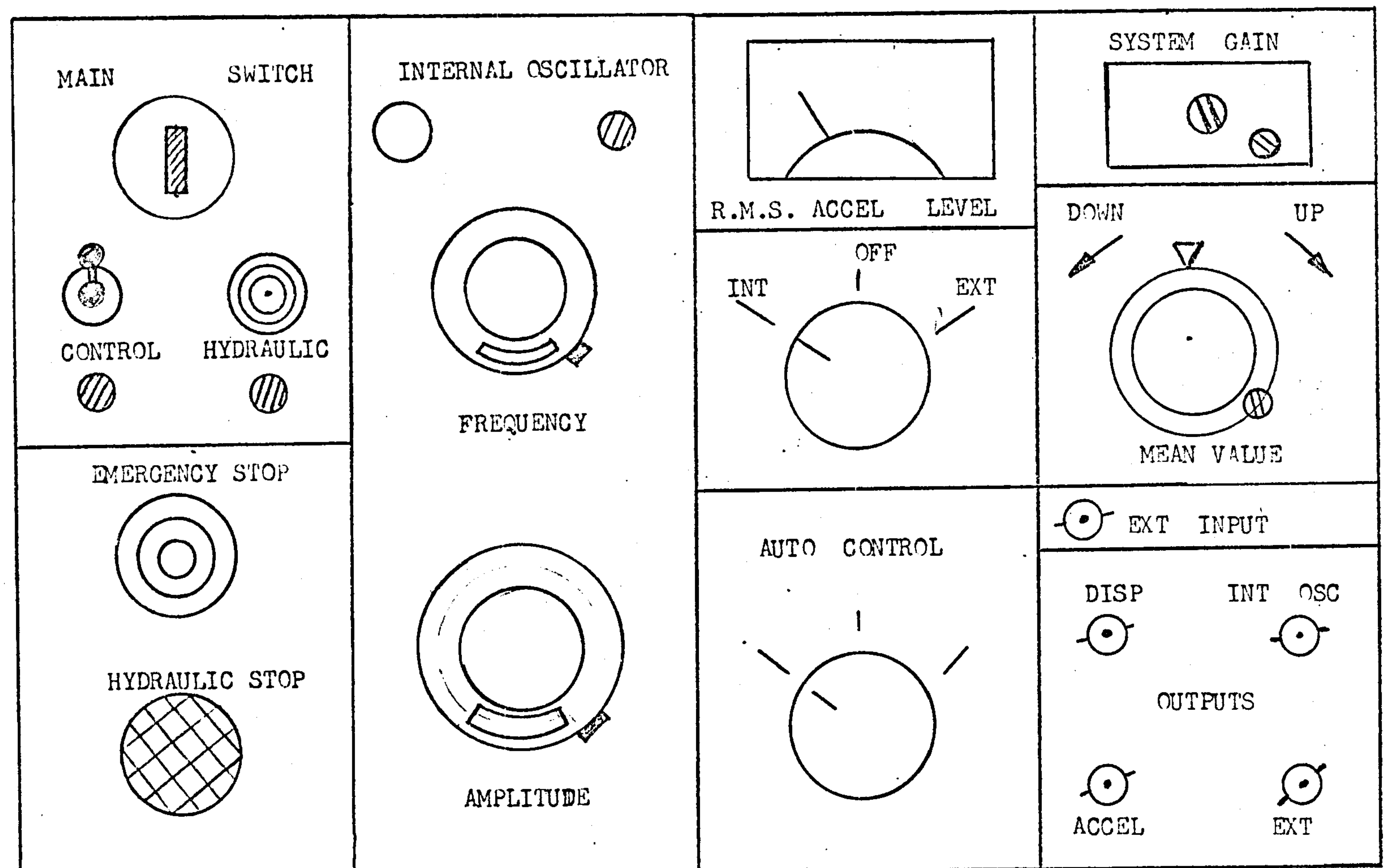



Figure No. 23




Suggested Design for Control Unit Console

Figure No. 24

Table No. 1

| Unit | | | Simulated Test No. |
|---|---|---|---|
| A | Operator - System gain set too high → | System goes unstable → | Test No. 1 |
| B | Operator - switches off or power failure → | (i) To actuator control unit → | Tests Nos. 2 and 3 |
| | | (ii) To control unit and hydraulic power pack → | Tests Nos. 2 and 3 |
| | | (iii) To hydraulic power pack only → | Test No. 4 |
| C | Operator - Switches from internal oscillator to "OFF" → | | Test No. 5 |
| D | Operator - Frequency changed too quickly → | Sudden rise in acceleration level of actuator → | Subject presses "Emergency Stop Button". (See unit J) |
| E | Operator makes incorrect setting or failure of electronic control unit → | Excessive command signal - within operating limits of actuator → | Simulated test Ram travels between or into snubbers (See unit H) |
| F | Head from control unit to actuator fails → | Large transient signal → | Tests Nos. 6 and 7 |
| G | Mechanical failure of Moog valve → | Sends piston hard to one end → | Ram travels into snubbers (See unit H) |
| H | Ram travels into hydraulic snubbers → | Snubber relief valve fails → | Piston crashes into actuator casing |
| | | Snubber relief valve set incorrectly: too low → | Piston crashes into actuator casing |
| | | Snubber relief valve set incorrectly: too high → | High retardation (See Test No. 8) |
| | | Snubber valves set correctly → | Test No. 8 |
| | | Snubber valves set correctly → | Limit switch - S.Op. S.D. (See unit J) |
| J | Subject, operator or observer presses "Emergency Stop Button" → | S.Op. S.D. (System Operated Shut Down) → | Test No. 9 |

Results from the nine simulated failure modes

| Test No. | Max. Accel. change 'g' | Velocity Change (total) metres/ sec. | Duration milli-secs. |
|---|---|---|---|
| 1 | 5 | .5 | 10 |
| 2 | 2 | .4 | 20 |
| 3 a | 1.35 | .13 | 10 |
| 3 b | 6 | .7 | 12 |
| 4 | shut down time | 3.5 - 4 seconds | |
| 5 | shut down time | 1 - 1.5 seconds | |
| 6 | 1.8 | .7 | 40 |
| 7 | 9.5 | 1.4 | 15 |
| 8 | 5.6 | .76 | 14 |
| 9 | 18(pk to pk) frequency | 65 Hz | |

Table No. 2    Tabulated results from the nine simulated failure modes.

# PART B

## MECHANICAL IMPEDANCE STUDIES

## B - Mechanical Impedance Studies

### B - 1. Introduction

The use of mechanical impedance as a technique for assessing the behaviour of a mechanical structure under dynamic conditions, has increased in importance in the last ten years. Mechanical impedance has therefore been increasingly used in bio-engineering as a method of describing the response of the human body to various dynamic inputs. Coermann's (1959) and (1963), and Dierckmann's (1958) measurements of mechancial impedance showed the value of the technique as a means of determining the response of the body in various postures to steady-state sinusoidal vibration.

The following part of the thesis details the main experimental programme using the electro-hydraulic vibrator. The main aim of the programme was to continuously monitor the mechanical impedance of the human body in various seated and standing postures. The continuous measurement of the mechanical impedance of the body started with a small pilot study using one subject, a number of weights of known impedance, and a weight placed on a foam rubber cushion. The pilot study (Section B - 6) was used to check and modify if necessary, the analogue computer processing circuits and the experimental procedure ready for the main experimental programme.

The main experimental programme involved the use of sixteen subjects, impedance measurements being made in various seated and standing postures. The modulus of impedance results were obtained using a small analogue computer as a direct readout during the experimental run, and tape recordings were also made to enable the phase angle plots to be obtained later by analysis on a large analogue computer. Previous investigations using mechanical actuators had built up impedance plots by means of measurements taken at fixed frequencies; their results had shown that the posture adopted by the subject had a considerable influence on the resultant whole-body impedance. Thus during the research programme particular attention was paid to the posture adopted by the subjects. It was hoped that if the posture adopted by subjects could be standardised, and then maintained for the short period of time required for the experimental run, individual differences between subjects could be studied.

Subject data was also recorded at the time of the experiment, the subjects having first been asked to complete a confidential form to assess their suitability as subjects, full details of this procedure are given in Section B - 7. Impedance measurements were also taken for female subjects, these have not been reported in previous studies, probably because most of the previous research has been sponsored or undertaken by military establishments. As impedance measurements are used by seat designers for transport systems, data referring to both sexes is necessary if the system is to be used by the general public.

The final part of the experimental programme involved a statistical analysis, and conversion, of the data on a digital computer. The conversion carried out was that of changing the acceleration based impedance measured in this research to the velocity-based impedance used by previous researchers, the conversion being necessary to enable direct comparison of this research work with existing data.

## B - 2 Literature Review.

### B - 2.1 Introduction

The following section reviews the methods used by previous experimenters in their investigations of the mechanical impedance of the human body, as well as their results. However, due to the short length of papers in most journals, accurate reporting of all the details necessary for a concise and comprehensive report on a human vibration experiment, is often difficult or impossible. Most of the work of this nature has been carried out in the United States, often in military establishments. This has meant that most of the subjects have been service personnel. The main emphasis in the investigations has been that of the vibration environment encountered in military and space vehicles.

This section, as well as reviewing methodology and results, also indicates the instrumentation and analysis techniques used, together with notes on the type and number of subjects taking part. The details given of the type of apparatus used and its performance varies considerably. In some reports the details are given in the odd paragraph, whilst in others several pages of description together with photographs is considered necessary. Guignard and Guignard (1970) note that "although many different kinds of vibration machine or motion simulators have been used for human experimentation, detailed specification of the machine used, its type of drive, means of programming or control, range of operation, fidelity of motion, and so on are regrettably uncommon in the published literature on human experimentation in this field.

B - 2.2 Experimental Facilities and techniques used in the measurement of whole body Mechanical Impedance

Probably the best known of papers on whole body mechanical impedance is that of Coermann, R.R. (1963). He used a mechanical shake table in his investigation, a picture of which is shown in the report. There is little other information about the shake table. Coermann does note however "The mechanical shake table could not be expected to produce an absolutely pure sinusoidal motion". The subjects were placed on top of a very stiff plate, connected to the force table by three force transducers. The resonant frequency of this system was over 120 Hz. The force transducers to obtain this high resonant frequency were stiff, and this meant that a very small deflection had to be measured. Coermann used a special variable reluctance transducer which was connected via a band-pass filter to the carrier amplifier and recorder. The table displacement was also recorded by means of a variable reluctance type displacement meter, the velocity being computed from the displacement recordings.

The equipment was calibrated in two ways. The displacement transducer was calibrated statically, while the force cells were dynamically calibrated. To calibrate the force transducers Coermann bolted two 75 lb lead weights to the table and these were shaken at various amplitudes from 1 Hz to 20 Hz. As well as enabling Coermann to calibrate the force transducers, he could also calculate the phase shift introduced by the band-pass filters.

Coermann used several posture types, erect and relaxed postures of the sitting subject, and an erect posture with stiff knees.

The sitting posture was defined by Coermann in the following way: "The man's legs hung down freely over the side of the shake table not touching the ground. The hands lay on the upper thigh and all belting was removed from the waist. There was no padding between the force cell table and the man except for clothing". He also notes that "At the moment before the records were taken the subject breathed out completely".

In order to check the influence of the mobility of the soft body parts on the human body impedance, Coermann repeated these postures with a semi-rigid envelope applied to the abdomen. He also made an investigation as to the influence of a U.S.A.F. Type MC3 pressure suit. At the end of his test programme Coermann also carried out some experiments with regard to the transmissibility of the human body, and he was particularly interested in discovering which part of the skeleton undergoes the most stretching at the resonant frequency.

The acceleration levels used for the majority of his experiments were "kept at a comfortable level to the subject" but actual figures are not given as to the precise levels used. Coermann also used levels of up to .5g over the specified frequency range. He reports he could not use higher levels due to limited subjective tolerance to vibration of his subjects. These higher levels were used to study the linearity of impedance with increasing acceleration levels.

The records were taken at $\frac{1}{2}$ Hz intervals from 1 Hz to 14 Hz and then 1 Hz intervals to 20 Hz. After reaching 20 Hz a further twenty records were taken between the frequencies used for the first series, (with the frequency decreasing). These records acted as a check on the first set of records. Coermann noted that he needed about 2000 records to complete his programme.

The results were analyzed by means of a digital computer, from the experimental records. The results were presented graphically in the paper. Coermann also quotes the accuracy of his measurements, the modulus of impedance accuracy being better than 2% and phase angle measurements to within .1% - these accuracies do however assume a perfect sinusoidal input.

Krause and Lang (1963) report on experiments carried out at the Wenner-Gren research Laboratory, to investigate the non-linear behaviour of the human body, this being in respect of increasing acceleration levels under steady-state vibration conditions.

The investigation was under-taken by computing the mechanical impedance of the human body to acceleration levels of .5g (peak). The investigations were continued to higher acceleration levels by using a 70 lb pig. Levels of up to 3.0g (peak) acceleration were used with the pig strapped onto the force measuring table.

The vibration machine used was an electrohydraulic shake table, which was capable of producing reasonably pure harmonic motion with a double amplitude of up to 10 inches. Force was measured over a frequency range of 2 - 20 Hz by means of specially developed high-frequency load cells which have negligible deflections. The force readings were converted into "mechanical impedances" by computing the ratios of force amplitude to the respective velocity amplitude of the shake table. The authors also reported that "subsequent techniques were developed which furnishes live-load impedance as a direct readout value" - this apparatus being reported by Sharp and Lange (1964).

The paper described a series of tests undertaken with the following postures:- standing erect and relaxed, sitting erect and relaxed, lying prone on the side, supine and semi-supine. The authors remark that the last two posture types were the two of prime interest, and the reported results only show graphs for these postures.

Although it was reported that a small group of male subjects was used no numbers were given. As well as mechanical impedance measurements, abdominal strain and internal pressure were also measured on the human subjects, and abdominal deformation was measured on the 70 lb pig.

Edwards and Lange (1964) reported a further series of experiments undertaken at the Wenner-Gren Aeronautical research laboratory.

The authors gave details of the shake table used (no doubt the same as that mentioned above). It was reported as "being capable of producing continuously variable motion within the limits of 10 inches of amplitude, frequencies to 100 Hz and velocities up to 80 inches per second. These limits correspond to accelerations up to 12 g and vector force outputs up to 3000 lbs."

The impedance measurements were made in the same way as those of Krause and Lange (1963) and were undertaken with two male subjects, the postures adopted being supine, lateral and standing. The variations of impedance due to change in muscle tone and due to the padding of the subject support were also investigated.

The frequency range studied was 1 - 20 Hz at peak shake table accelerations of 0.2, 0.35 and 0.5g.

The report contained photographs of the apparatus used.

A study by Pradko et al (1965) however contains little information about the subjects, and no information about the vibration apparatus and associated instrumentation. The report described several tests which were carried out as part of a large programme of research. Most of the tests were associated with investigating a parameter "absorbed power" first postulated by R. A. Lee. Absorbed power was used to form a scale of vibration intensity which was compared to subjective assessment of intensity by use of military and civilian subjects.

Pradko et al gave a curve for mechanical impedance but say of their data:- "The experimental data is the mean of different acceleration levels using human test subjects". Since there was no data on the subjects, i.e. age, weight and height, this leaves a considerable gap in the reporting and detracts from the value of their report.

The vibration used was both random and sinusoidal and a number of levels was reported. The acceleration levels for both random and sinusoidal were measured in r.m.s. "g". The report concluded that "whole body response to random and sinusoidal vibration displays linear characteristics". A conclusion not shared by other researchers:- Wittmann and Philips (1969), Edwards and Lange (1964) and Krause and Lange (1963).

A study of mechanical impedance of a seated subject under sustained acceleration was carried out by H. L. Vogt et al (1968).

The subject was seated on a shake table mounted on the arm of a centrifuge. The shake table was electro hydraulic and capable of producing a constant acceleration amplitude of .5g (vector) through the frequency range 2 - 20 Hz. The subjects were "ten young and healthy male subjects", who had volunteered to be subjects.

This report was one of the few reviewed by the author to note many of the deficiencies of previous papers.

Suggs et al (1969) used mechanical impedance measurements as an aid in the design of a dynamic simulator for testing cross country vehicle seats. Suggs makes an interesting point about the existing data on the mechanical impedance of seated subjects which deserves attention. "Because existing data was for seated subjects with unsupported feet it was necessary to re-evaluate the parameters for subjects with their feet resting on a platform as is the case for vehicle operators. The authors found that only about 75% of the subject's weight was supported by the seat".

A mechanical actuator was used - "The platform was mechanically driven at a peak to peak amplitude of 0.10 inches by a scotch-yoke mechanism which produced a sinusoidal vertical motion from 1.75 to 10 Hz." (See Appendix I - Actuator. ). The measurements were taken by means of a velocity transducer, and a strain gauged beam to measure the force input to the subject. There were no pictures of the apparatus used but a description was given in the text. Suggs et al used a strain gauged cantilever, they did not however give any indication as to the natural frequency range of study. The tests were carried out in a frequency range of 1.75 Hz to 10 Hz. This value was chosen by Suggs et al because the major whole body resonances occur within this frequency range. Also of importance was the fact that most of the terrain induced factor in "off-road" vehicle vibrations occurs in this frequency range. Suspension tractor seats also have natural frequencies in the lower half of this range.

The experiment used eleven subjects ranging in weight from 118 lbs to 195 lbs, and Suggs et al noted "The impedance magnitude is much less (165 lb.sec/ft) than the 350 lb.sec/ft measured by Coermann (1963) for the whole body system". This difference was thought to be due to the support given to the subject's legs. (The legs were unsupported in Coermann's measurements).

The results were analysed from recordings taken at discrete frequencies. The frequency interval used was $\frac{1}{2}$ Hz with $\frac{1}{4}$ Hz steps used near resonance. The force strain gauges were calibrated directly by using known rigid weights. The accuracy of the system was evaluated by comparing the observed impedance of each weight to its calculated impedance.

The work of Coermann has for some time now been regarded as "a classic work" in this particular field of study. The review points out the meticulous care which Coermann exercised in his experimental programme, and analysis of results. In the design of his apparatus - i.e. his impedance measuring platform, Coermann took great care to ensure its natural frequency was sufficiently high, so as not to affect his results. The results of Coermann and the other investigators are shown in the figures at the end of Part B.

The use by Suggs et al of a seated posture, with the legs supported was a point noted by the author, who used a similar posture in his own experiments. The experiments by Pradko et al from which they concluded "The human body displayed linear characteristics" is a view which was not shared by other rsearchers. Most researchers do however agree that the body displays linear characteristics at low vibration levels (i.e. those below .5g (peak to peak) in the frequency range 2 - 20 Hz). Thus the acceleration level used to determine the value of the mechanical impedance of the human body under steady state sinusoidal vibration can be an important factor in experimental design, and in the interpretation of previous experimental results.

### B - 2.3 Results and Conclusions of previous researchers of whole body Mechanical Impedance

Mechanical impedance and phase-angle plots for the response of the human body to steady-state sinusoidal vibrations, have been obtained by various researchers in the past in attempts to obtain a correlation between mechanical impedance measurements and subjective response to vibration and/or to enable mechanical and electrical analogues of the human body to be postulated. These analogues could then be used to determine the response of the human body to high acceleration levels, (assuming the response of the body to be linear) which could not be obtained experimentally.

Dieckmann (1958) was one of the first to study the dynamic properties of the human body by impedance techniques. Fig No.26 shows an example of the modulus of impedance curves obtained. Dieckmann used these curves to postulate mechanical analogs of the human body, He also used mechanical impedance curves to formulate a series of vibration tolerance curves. (Dieckmann (1957 ) .

Coermann (1963) using a group of eight male subjects which were "all in a healthy condition with no abnormalities" obtained a series of now well known modulus of impedance (velocity based) and phase angle curves. The data for the subject panel used by Coermann is given in Table No.3 and his Impedance curves for one subject (RC) in three given postures:- sitting erect, sitting relaxed and standing erect, in Figure No.28 and for all eight subjects in the sitting erect posture in Figure No.27

Also shown in Figure No. 29 are the results of a transmissibility study on one subject, undertaken during the impedance studies.

Coermann noted that at frequencies up to 2 Hz the body responded at any posture as would a pure mass, as the frequency increased above 2 Hz, however, the impedance deviated from that of a pure mass, The impedance becoming higher as the frequency increased above 2 Hz, culminating in a peak which was dependent for magnitude and frequency on the subject's posture. Coermann found that the peak was highest in magnitude in the sitting erect posture, the frequency of this peak being about 7 c/s with a second peak at 10 Hz, and a third peak at 15 Hz.

Coermann's results for the sitting relaxed and standing erect posture were obtained using one subject. In the sitting relaxed posture, Coermann stated that the first resonant peak occurred at 5.2 Hz with a more pronounced second peak at 11.5 Hz, the third peak which was visible at about 15 Hz in the sitting erect posture, was almost completely flattened when a relaxed posutre was adopted. Coermann used these results to formulate a mechanical analog of the human-body in the sitting erect posture. Sandover (1969) noted that Coermann's was a parallel mechanical system, unlike the series mechanical systems postulated by Dieckmann (1958).

In commenting on his obtained phase angle plots, Coermann noted that in the sitting erect posture the phase angle becomes negative above 14 Hz indicating that the impedance of the elasticity dominates; in the sitting relaxed posture, however, the plot although similar to 15 Hz, turned downwards to give a positive phase angle, indicating the predominance of the mass impedance.

Edwards and Lange (1964) carried out mechanical impedance studies (velocity based) using a steady state sinusoidal input in the frequency range 1 - 20 Hz. Although they used various body postures, no results are given for subjects in seated postures. Edwards and Lange were primarily concerned with investigating the linearity or otherwise of the human body, and Figure Nos. 32 and 33 show the whole body impedance at three specific acceleration levels. Edwards and Lange noted that a degree of non-linearity did exist, they had however to keep acceleration levels below .5g for safety reasons.

Krause and Lange (1963) also investigated the problem of linearity, by a series of experiments carried out on an anaesthetized pig. In summarizing the results of their mechanical impedance and circumferential deformation experiements on the pig they stated that their experiments confirmed the existence of definite natural frequencies within the body. The impedance data, (see Figure No. 34. No phase angle plot however, was given to complete the impedance data) stated Krause and Lange, showed that the whole body was not a linear system, and their data indicated that the damping elements in particular seemed to be non-linear. Tests they carried out on abdominal strain showed that it increased with the intensity of the forcing motion, and that the strain could increase in a roughly linear manner in one place, and exhibit non-linear properties in a place right next to the one displaying linear characteristics.

They also noted of the impedance plots, that the shape of the 1.08g curve at 5 Hz and the 0.65g curve at 8.4 Hz were characteristic of impedance curves of human subjects with their muscles tensed. They assumed that the anaesthesia was beginning to wear off when these points were measured and that the pig was tightening up against the vibration. As the pig was anaesthetized and evidently held in some form of harness to facilitate the high acceleration used, both these factors would considerably influence any impedance plots they obtained, and thus any conclusions which can be drawn from them.

Weis E.B., Clarke N.P., Brinkley J.W., and Martin P.J. (1964) showed impedance plots (velocity based) for subjects under steady state vibration conditions, and also under impact conditions. The steady state vibration curves are from Coermann's experiments, the curves obtained under impact conditions were evaluated by the authors; they measured the force and velocity inputs to the subject and calculating the impedance using Fourier transforms.

The results of Weis, Clarke, Brinkley and Martin, are shown in Figure Nos. 35 and 36. These curves show the similarity of response that occurred in the two mechanical environments.

Weis, Clarke, Brinkley and Martin were also interested in obtaining a connection between the mechanical impedance curves and subjective tolerance curves, by means of the energy transfer from the environment to the body. Coermann (see Fig No. 31) had investigated various ways of linking an impedance plot to a subjective tolerance curve, these being the transmitted force, the dissipated energy (i.e. within the body) or the relative displacement of the effective body masses. (The effective body masses Coermann defined as the two masses which he had evaluated for his mechanical analog of the human body). The subjective tolerance curve used by Coermann was that of Magid and Zeigenruecker (1959), (the indication for tolerance used by Magid and Zeigenruecker was not moderate discomfort but the maximum endurable stress and pain) and he found that with a simple extrapolation the constant relative displacement of the effective body masses best fitted the subjective tolerance curve.

Weis, Clarke, Brinkley and Martin however considered the energy transferred to the body, they commented that under 5 Hz energy was transferred to the subject, but that this was stored and returned to the environment; from 5 Hz or 15 Hz a considerable amount of transferred energy began to be dissipated within the body and could have gone into stretching the elastic components. Above 15 Hz the relationship between dissipation and/or storage and return to the environment of the transferred energy was very dependent on the posture adopted.

Weis E.B., Clarke N.P. and H.E. Von Gierke (1966) commented on the similarity between the impedance plots obtained under steady-state and transient conditions, the semi-supine posture showing the best correlation, with the standing and then seating postures showing correlation, but to a lesser degree. They used the same impedance plots as Weis, Clarke, Brinkley and Martin (1964).

The authors concluded that the variability that existed between the steady-state and transient impedance plots was probably due in part to the variability of the test situation including such things as different restraint systems; and the fact that the biological system was not really passive. They inferred that the human body, particularly in steady-state impedance determinations at low frequencies could anticipate the motion and protect against it. (Compare to comments of Krause and Lange (1963).

Weis, Clarke and Van Gierke also, while commenting on the general shape of the impedance curves during steady-state sinusoidal vibration conditions, noted that the impedance was tending towards lower values at higher frequencies. This tendency they thought could have been due to the fact that most of the steady state measurements of impedance were made at acceleration levels below 1g (single peak) and with no subject restraint system to couple the subject positively to the table. It could, however they stated have been due to inherent non-linearities of the system (i.e. the human body) such that the impedance was amplitude and/or velocity direction dependent.

The extent of non-linearity in the response of the human body to steady-state sinusoidal vibrations was investigated by Wittmann and Phillips (1969). Most researchers accept that a degree of non-linearity does exist in the response of the body, the main discussion has been on whether the non-linearity that exists is sufficient to make it impractical to assume that the system is linear.

Wittmann and Phillips in their paper demonstrated that non-linearities did exist, and that they could be of sufficient magnitude to make the linearity assumption impractical. During a vibrational study in which the input force and acceleration to the body were measured, as the input acceleration was sinusoidal the force input should also have been for all frequencies, at some frequencies however it was not sinusoidal. Wittmann and Phillips took this to mean that although the response was linear at certain frequencies and acceleration levels, where the force input was sinusoidal, at the other frequencies the body exhibited considerable non-linear characteristics. (See Figure Nos. 37 and 38 ) They also commented on the peak value and time durations of the "loading" and "unloading" cycles, during the non-linear phases.

Wittman and Phillips also compared the response of a non-linear mechanical system (using non-linear spring characteristics) to that of a linear series two degree of freedom system. They concluded that the response of the non-linear system could be confused with that of the linear two degree of freedom system if the phase angle were not documented, as it was the fact that the phase angle exceeded minus ninety degrees for the non-linear system, (this is a condition that cannot be achieved by a linear system) which enabled the two systems to be distinquished from each other. The need to show the phase angle plot as well as the modulus of impedance plot was also stressed by Murfin (1969) - "It is mandatory however, that phase be measured for impedance measurements to be useful."

In a paper by Pradko F., Lee R.A., and Greene J.D. (1969) investigating the development of transfer-function of the human body under steady-state sinusoidal and random vibrations, they stated that in the sitting erect posture the response of the body was essentially linear. The frequency range being 1 - 30 Hz with acceleration levels up to 1g (r.m.s.) or 1.4g pk. to pk. Coermann (1963) also noted that in the frequency range o - 20 Hz and at acceleration levels of 0.2g., 0.6g. and 1.0g. (pk. to pk.) the impedance and phase angle plots stayed within the 10 per cent range (which was the accuracy with which his phase angle and impedance measurements could be taken). The variation Coermann suggested could easily have been due to slight posture changes during the experimental runs. The subject chosen was a heavy man (99.5 kg) as Coermann expected the soft tissue to have more influence on the linearity than the bone elasticity. From these tests, Coermann concluded that within the frequency range and acceleration limits used, the body did behave as a linear system.

Having established the linearity of the body within the limits of frequency and acceleration they set, Pradko, Lee and Greene proceeded to develop a transfer function for the body in the sitting erect posture; they used the asymptatic approximation method, the resultant factors are listed in their paper. The transfer function for acceleration was simulated on an analogue computer and compared to experimental data; the force-motion transfer function was converted to give a theoretical impedance curve, this was compared with experimental impedance data. (The experimental data was - "the mean of different acceleration levels using human test subjects.") The authors concluded that whole body human response to random and sinusoidal vibration displayed linear characteristics, and that transfer-function statements could be found which accurately described human dynamic response.

During their investigations into "tolerance" and "linear response" experiments the authors noted that quite probably the term "ride comfort" was non linearily related to the amplitude of acceleration. Pradko, Lee and Greene then examined "absorbed power" as a means of predicting subject response to sinusoidal and random vibrations. Absorbed power was first posulated by R.A. Lee. Lee defined it as the time rate at which the vibration energy imparted to the body was completely dissipated. After a series of tests in which subjects were asked to assess the ride comfort of the vibration, the results were then compared to the computed absorbed power, (the previously calculated transfer functions were used to compute the absorbed power). The results showed that absorbed power identified and discriminated ride comfort over large and small differences of the input characteristics. The authors however felt that further experimental work was required, before they could confidently say that absorbed power correlated with subjective response.

Ride comfort has been a subject of particular interest in the agricultural engineering field, with special reference to tractor drivers, Matthews J. (1964) in a comprehensive series of articles entitled "Ride comfort for Tractor Operators". The articles although not specifically related to mechanical impedance studies does contain in the review a graph (Figure No.39) from the experiments of Simons (1952). Matthews commented that even if an optimum design of tractor seat was postulated, it would still probably be inferior to the human legs as a vibration absorber.

Sugg C.W., Abrams C.F., and Stikeleather L.F. (1969) conducted a series of experiments to measure the impedance (velocity based) of 11 subjects whose weights ranged from 53.5kg to 88.5kg in the frequency range 1.75 Hz to 10 Hz using a fixed vibration displacement of .25cms (pk. to pk.) The experiments were thus conducted with the acceleration level varying from 0.03g at 1.75 Hz to 1.0g at 10 Hz. (acceleration pk. to pk.) The subjects adopted a seated posture with the legs supported on a platform to simulate vehicle operators. The authors found that this reduced by 25% the sitting weight of subject.

Sugg, Abrams and Stikeleather reported that the general shapes of the modulus of impedance and phase angle plots were as Coermann's (1963); they observed a primary resonance of about 165 lb. sec/ft (2460.0 x $10^3$ dynes x sec/cm) and a lower amplitude secondary resonance at approximately 8 Hz. The curves obtained by Suggs, Abrams and Stikeleather are shown in Figures No. 40 and 41. The impedance magnitude observed by Coermann for a subject sitting in a relaxed posture was 5050.0 x $10^3$ dynes x sec/cm. The authors do not offer any reasons for the discrepancy between the two sets of results, but several factors could have influenced the results:-

a) the fact that Coermann results (and the results of the other studies reported in this section) have been for sitting subjects with their feet unsupported;

b) all other tests have involved measuring the impedance over a fixed frequency range at a constant acceleration level

c) Suggs, Abrams and Stikeleather do not give sufficient data on their subject panel - were they tractor drivers used to a vibration environment? were they young or old? This however is a comment which could be levelled against several other papers reviewed in this section.

As the impedance curves obtained by Suggs, Abrams and Stikeleather, had the same general shape as those obtained by Coermann, Suggs, Abrams and Stikeleather concluded their simulator could be of a similar type to that proposed by Coermann - i.e. a parallel two "mass-spring damper" system. A simulator was designed using parameters obtained by a digital computer analysis of the impedance curves, these however were modified to make the simulator static weight the same as that of the average static load imposed by the subjects. The modulus of impedance plot for the simulator followed closely the curve obtained from the 11 subjects, no phase angle plot was given however for the simulator. To assess the degree of accuracy of the simulation, particularly of the simulator, as Suggs, Abrams and Stikeleather said was to be used for testing tractor seats with linear and non-linear suspensions, the phase angle plot of the simulator must correspond to that of the 11 test subjects; otherwise any results obtained using it in the testing (or assessment) of tractor seats could be misleading.

The problem of "tolerance" -"comfort" and "fatigue decreased performance" (F.D.P.)limits has been of interest to researchers for some years, interest being renewed when the International Standards Organisation (I.S.O.) published a working group proposal in 1968, (the proposal was revised in 1970). This proposal was taken from a background of experimental work showing very large differences in the shape of tolerance-comfort and F.D.P. limits as well as in the obtained acceleration levels. In an interesting paper by Ashley (1970) entitled "Equal annoyance contours for the Effect of Sinusoidal Vibration on Man", he proposed a new technique for obtaining "equal annoyance contours" . The method employed is similar to that used to obtain equal loudness contours for hearing. Ashley used a random vibration spectrum as a datum, and a cross matching procedure with a sinusoidal vibration was employed to obtain the constant annoyance contours. The random vibration spectrum was first related to the I.S.O. proposed F.D.P. limit at 6 Hz by using the cross matching technique, and panel of 27 subjects in the age range nineteen to fifty-seven. Using the random levels obtained the sinusoidal vibration equal annoyance contours were evaluated using six subjects. Ashley compared his equal annoyance contour to a comfort curve computor using the method of absorbed power proposed by Pradko and Lee. Pradko and Lee's published curve for equal comfort in the seated posture was normalised to the I.S.O. 4 hr. F.D.P. limit. The two curves (see Figure No. 42. ) showed noticeable divergence at low frequencies, Ashley commented that this was to be expected as the body moves as a rigid mass at low frequencies (below about 3 Hz) and little energy is thus absorbed. Ashley proposed therefore that comfort was a function of vascular disturbance at low frequencies and absorbed power at high frequencies. He also commented that "safe exposure limits" may well be based on different criteria such as acceptable relative motion of internal organs, (the overall absorbed power would then be low) minor body sub-systems may be experiencing large amplitude displacements, leading to localized discomfort or the imparing of performance. (e.g. the eyeball, which has a resonant frequency of approximately 25 Hz).

Although the last paper reviewed was not directly concerned with the Meohanical Impedance of a the human body, it does however illustrate the vast number of factors which can influence man's subjective assessment of a vibration environment. It also shows that even if the objectives of an experimental programme are clearly set down, the way the researcher sets about achieving these objectives will influence the results obtained, and the way the researcher's conclusions can be used in the future.

## B - 3 Experimental Rationale

As the studies to be carried out were to measure the mechanical impedance of the human body in various seated and standing postures, several decisions had to be made as to the method of measurement, postures to be adopted, and the vibration environment to be used for the tests, before the pilot study and main experimental programme could be carried out. Mechanical impedance measurements computed previously have been those of velocity based impedance. These results have been obtained from records of the force and displacement (or velocity) taken at a fixed frequency. The force and displacement* traces were then digitised and the modulus of impedance and phase angle for the velocity based impedance were computed by digital computers.
*(The measured displacement was converted to velocity by digital computers, or some researchers measured velocity directly using a suitable transducer).

The main aim of these studies was to monitor continuously the mechanical impedance while the subject was on the actuator. A continuous monitoring technique had not been used previously to measure whole body impedance, but the method has the advantage that posture changes and individual differences between subjects can be noticed during the plotting; also any posture changes or items of clothing likely to influence the results, could be modified during the test, and new plots made if necessary.

A continuous monitor of whole body mechanical impedance could have been achieved in two ways, firstly by digitising the force and acceleration traces and processing them by means of a digital computer fitted with a plotter, or alternatively by using an analogue computer and plotting the results directly on an x-y plotter. The first alternative was not however feasible as the digital computing facility of the University was a considerable distance from the laboratory and a suitable digitiser was not available to process the force and acceleration signals. An analogue computer could however be obtained economically and an x-y plotter was available to enable direct readout of the modulus of impedance while the subject was on the actuator.

The conversion of the acceleration signal obtained from the force cell, using the analogue computer elements however proved to be unsuccessful without the use of a special integrator unit which was unavailable. Acceleration based mechanical impedance could however be obtained directly from the analogue computer processor, this modulus of impedance curve could then be converted to a velocity base by a digital computer after the tests if required.

A small analogue computer was available for use at the vibrator, to monitor directly the modulus of impedance, to have obtained simultaneously a direct readout of the phase angle (between the force and the acceleration) would however have required more amplifiers than were available on the small computer. The force and acceleration signals, therefore, had to be tape recorded to enable processing on a larger analogue computer. (Situated some distance from the vibrator). It was originally intended that the recordings would be made simultaneously with the modulus of impedance plot-out. During the course of the pilot study however it was found that the tape recorder loaded down the force and acceleration signals, this interfering with the modulus of impedance monitoring circuit. Two test runs at each posture had to be made, the first to obtain the modulus of impedance plot, the second to enable the force and acceleration signals to be recorded for future processing.

The time taken to obtain an impedance plot and tape recording should be as short as possible, but compatible with the response of the monitoring systems as the results were to be of a direct read-out nature. A frequency sweep was used over the frequency range of 3 - 30 Hz. The impedance was monitored first with the frequency increasing from 3 - 30 Hz, the sweep time being two minutes and then with the frequency decreasing from 30 - 3 Hz also in two minutes. This meant that the subject would only have to assume one of the set postures for a maximum time of ten minutes, in an attempt to reduce the affects of postural changes on the results obtained.

The postures to be adopted by the subjects were selected as being typical of postures which would be adopted by passengers and operators of transport systems. Two seated postures were used for the majority of subjects, with three standing postures, and a seated posture set by the subject whilst undertaking a tracking task, was used on a small number of subjects. For all the seated postures adopted the subject's feet were supported on an adjustable foot rest, which was attached to the actuator table. The foot rest was adjusted so that the subject's thighs were horizontal before the test runs were carried out. This was a similiar procedure to that used by Sandover. The similarity was to enable a future comparison to be made between the impedance results obtained under steady-state vibration conditions to those using impact and random vibrations as the dynamic inputs to the body.

The acceleration level used for the experiments was .5g (pk to pk), this level was chosen because previous researchers have reported marked non-linearity in the response of the body at high acceleration levels. Also this level corresponds to the proposed ISO 30 minute Fatigue Decreased Proficiency (FDP) Limit for human beings in the gz direction. Although this level is higher than that normally experienced by passengers and apparatus in transport systems this slightly higher acceleration level was required to enable the monitoring unit to function accurately.

The actuator was set in such a way that the set acceleration level of .5g (pk to pk) would be maintained to within 5% of the level set over the frequency range used for the tests - 3 to 30 Hz. (Figure No. 78 )

This enabled the analogue-divider to be set up so that the quoitent was always less than unity. The use of a fixed acceleration over the swept frequency range also meant that the computed impedance magnitude (acceleration based) at a given frequency would be equatable to the "apparent mass" of the subject at that frequency.

## B - 4 Mechanical Impedance Theory

### B. - 4.1 Definition of Mechanical Impedance

Initially mechanical impedance will be considered with a velocity base. Later these principles will be extended to include displacement and acceleration. The majority of the work has been carried out on a velocity base as this is the base which gives a direct analogy to Electical Impedance.

The human body can be considered as a complex combination of the mechanical elements of mass, elasticity and damping. As with complex electrical circuits consisting of inductances, capacities and resistances, a good insight into the properties of a complex system can be obtained by measuring the impedance of the electrical circuit.

The input impedance of an electrical circuit is defined as the complex ratio of the voltage to the current going through the circuit.

$$Z_E = \frac{E}{I}$$

The mechanical impedance of a mechanical network is defined as the ratio of the transmitted force to the motion of the point where the force is transmitted.

$$Z_M = \frac{F}{\text{motion}}$$

The impedance is first considered for electrical components and then for mechanical elements on a velocity basis and finally for mechanical elements on an acceleration basis.

B - 4.2 Impedance of Electrical Elements

Resistive Element

From Ohm's Law the voltage and current vectors are in phase. (Figure No. 43 )

Inductive Element

$$E = L\frac{dI}{dt}$$

$$i = I_{max} \quad \text{Sin } wt$$

$$\frac{dI}{dt} = I_{max} \quad w \text{ Cos } wt$$

$$E = L\, I_{max} \quad w \text{ Cos } wt$$

$$= L\, I_{max} \quad w \text{ Sin } (wt + \frac{\pi}{2})$$

From the above expression it can be seen that the voltage vector leads the current vector by $\frac{\pi}{2}$ ($90^{\circ}$) in an inductive element. (Figure No. 44 )

Capacitive Element

$$E = \frac{Q}{C}$$

$$Q = \int I dt$$

$$E = \frac{1}{C} \int I dt$$

$$I = I_{max} \ \text{Sin}\ wt$$

$$\int I dt = - \frac{I_{max}}{w} \ \text{Cos}\ wt$$

$$E = - \frac{I_{max}}{wC} \ \text{Cos}\ wt$$

$$= \frac{I_{max}}{wC} \ \text{Sin}\ (wt - \frac{\pi}{2})$$

From the above expression it can be seen that the voltage vector lags the current vector by $\frac{\pi}{2}$ ($90^\circ$) in a capacitive circuit. (Figure No. 45 )

The symbol $\phi_E$ is given to the phase angle between the Applied voltage and the current through the circuit, and $Z_E$ for the modulus of impedance. (Figure No. 46 )

B - 4.3 Impedance of Mechanical Elements (Velocity Base)

Pure Damper

For a pure damper of coefficient C by definition the damping force equals the damping coefficient multiplied by the relative velocity of the two ends of the damper.

$$F_c = C.\dot{x}$$

$$Z_{mc} = \frac{F_c}{\dot{x}}$$

$$Z_{mc} = C$$

$$\phi_{mc} = 0^o$$

Pure Mass

From Newton's Laws of Motion:-

$$F_m = M.\ddot{x}$$

$$\dot{x} = \dot{x}_{max} \text{ Sin wt}$$

$$\ddot{x} = \dot{x}_{max} \text{ w Cos wt}$$

$$F_m = M\dot{x}_{max} \text{ w Cos wt}$$

$$= M\dot{x}_{max} \text{ w Sin } (wt + \frac{\pi}{2})$$

$$Z_{mm} = \frac{F_m}{\dot{x}}$$

$$Z_{mm} = Mw \text{ Sin } (wt + \frac{\pi}{2})/\text{Sin wt} = jMw$$

$$\phi_{mm} = + \frac{\pi}{2}$$

Pure Spring

From the definition of a pure spring:-

$$F_k = x K$$

$$\dot{x} = \dot{x}_{max} \text{ Sin } wt$$

$$x = -\dot{x}_{max} \frac{\text{Cos } wt}{w}$$

$$= \frac{\dot{x}_{max}}{w} \text{ Sin } (wt - \frac{\pi}{2})$$

$$Fk = \frac{K\dot{x}_{max}}{w} \text{ Sin } (wt - \frac{\pi}{2})$$

$$Z_{mk} = \frac{K}{w} \frac{\text{Sin } wt - \frac{\pi}{2}}{\text{Sin } (wt)} = -\frac{jk}{w}$$

$$\phi_{mk} = -\frac{\pi}{2}$$

Nomenclature

The symbol $Z_v$ will be used for Mechanical Impedance (Velocity Base) in the ensuing text.

$$\text{eg.} \quad Z_{vk} = \frac{-jk}{w}$$

The general shape of the velocity based modulus of imdedance and phase angle plots for basic mechanical elements are shown in figure no. 47 .

## B-4.4 Impedance of Mechanical Elements (Displacement Base)

### Pure Damper

For a pure damper of coefficient C, by definition, the damping force acting across the dashpot the damping coefficient multiplied by the relative velocity of the two ends of the damper

$$F_c = C\dot{x}$$

$$Z_{mc} = \frac{F_c}{x}$$

$$x = x_{max} \ \text{Sin}\ wt$$

$$\dot{x} = x_{max}\ w\ \text{Cos}\ wt$$

$$Z_{mc} = C\ w\ \text{Cos}\ wt$$

$$= Cw$$

$$\phi_{mc} = +\frac{\pi}{2}$$

### Pure Mass

From Newton's Laws of Motion:-

$$F_m = M\ddot{x}$$

$$x = x_{max}\ \text{Sin}\ wt$$

$$\dot{x} = x_{max}\ w\ \text{Cos}\ wt$$

$$\ddot{x} = -x_{max}\ w^2\ \text{Sin}\ wt$$

$$Z_{mm} = Mw^2$$

$$\phi_{mm} = +\pi$$

Pure Spring

From the definition of a pure spring:-

$$F_k = x . K$$

$$Z_{mk} = \frac{F_k}{x}$$

$$\therefore Z_{mk} = K$$

$$\phi_{mk} = 0^{\circ}$$

Nomenclature

The symbol $Z_d$ will be used for Mechanical Impedance (Displacement Base) in the ensuing text.

$$\text{eg} \quad Z_{dk} = K$$

The general shape of the displacement based modulus of impedance and phase angle plots for the basic mechanical elements are shown in figure no. 48 .

## B - 4.5 Impedance of Mechanical Elements (Acceleration Base)

### Pure Damper

For a pure damper of coefficient C, by definition, the damping force equals the damping coefficient multiplied by the relative velocity of the two ends of the damper.

$$F_c = C\dot{x}$$

$$Z_{mc} = \frac{F_c}{\ddot{x}}$$

$$\ddot{x} = \ddot{x}_{max} \text{ Sin } wt$$

$$\dot{x} = -\ddot{x}_{max} \frac{1}{w} \text{ Cos } wt$$

$$Z_{mc} = +\frac{C}{w} \frac{\text{Sin } (wt - \frac{\pi}{2})}{\text{Sin } wt} = -\frac{jc}{w}$$

$$\phi_{mc} = -\frac{\pi}{2} = -90^\circ$$

### Pure Mass

From Newton's Laws of Motion:-

$$F_m = M\ddot{x}$$

$$Z_{mm} = \frac{F_m}{\ddot{x}}$$

$$Z_{mm} = M$$

$$\phi = 0^\circ$$

Pure Spring

By definition of a spring:-

$$F_k = x . K$$

$$Z_{mk} = \frac{F_K}{\ddot{x}_{max}}$$

$$\ddot{x} = \ddot{x}_{max} \ \text{Sin wt}$$

$$\dot{x} = -\ddot{x}_{na} \ \frac{1}{w} \ \text{Cos wt}$$

$$x = -\ddot{x}_{max} \ \frac{1}{w^2} \ \text{Sin wt}$$

$$Z_{mk} = - \ \frac{K}{w^2}$$

$$\phi_{mk} = -\pi = -180^{\circ}$$

Nomenclature

The symbol $Z_{\alpha}$ will be used for Mechanical Impedance (Acceleration Base) in the ensuing text.

$$\text{eg} \quad Z_{\alpha k} = \frac{-k}{w}$$

The general shape of the acceleration based modulus of impedance and phase angle plots for the basic mechanical elements are shown in figure no. 49 .

The mechanical impedance of a body thus describes a body's resistance to motion. In the equation to determine the mechanical impedance of a body, the quantities are stated as vectors, as in the majority of cases there will be a phase difference between the applied force and the resulting velocity (acceleration or displacement).

Mechanical impedance can be calculated in two forms:- (a) Point Impedance and (b) Transfer Impedance. (Figure No. 50 )

(a) Point Impedance

The concept of Point Mechanical Impedance describes the ability of a structure (or body) to withstand or absorb vibration. The applied force and velocity vectors are both measured at the input to the body whose mechanical impedance is being computed.

(b) Transfer Impedance

The concept of Transfer Mechanical Impedance describes the ability of a structure (or body) to transmit or isolate vibration. The applied force vector is measured at the input to the body, while the velocity vector is measured at the output of the body whose mechanical impedance is computed.

## B - 4.6 Mechanical - Electrical Analogies

The preceding section detailing the behaviour of electrical elements shows that two analogies can be drawn between mechanical and electrical elements. The first is the Mass-Inductance analogy, in which Force is analogous to voltage, and velocity to current. In the second analogy - the Mass-Capacitance analogy, force is analogous to current, and velocity to voltage.

| Mechanical | | Electrical Equivalents | | | |
|---|---|---|---|---|---|
| | | Mass-Inductance | | Mass-Capacitance | |
| Force | F | Voltage | E | Current | i |
| Damping | c | Resistance | R | Conductance | 1/R |
| Mass | m | Inductance | L | Capacitance | C |
| Compliance | 1/k | Capacitance | C | Inductance | L |
| Acceleration | $\ddot{x}$ | Rate of change of current | $\frac{di}{dt}$ | Rate of change of voltage | $\frac{dE}{dt}$ |
| Velocity | $\dot{x}$ | Current | i | Voltage | E |
| Displacement | x | Charge | q | Integral of Voltage | $\int E\,dt$ |

### Mechanical System

For the mechanical system shown in figure 51 the sum of the forces at any point in the system is zero. The equation governing the response of the system is:-

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + Kx = F(t)$$

### Electrical System (1) (Mass-Inductance analogy)

For the analogous system shown in Fig. 52 the sum of the voltage around the loop is zero. The equation governing the response of the system is:-

$$L\frac{di}{dt} + Ri + \frac{1}{C}\int i\,dt = E(t)$$

or $$L\ddot{q} + R\dot{q} + \frac{q}{C} = E\,(t)$$

Electrical System (2) (Mass-Capacitance analogy)

For the electrical system shown in Fig.53 the sum of current at any point in the system is zero. The equation governing the response of the system is:-

$$C\frac{dE}{dt} + \frac{E}{R} + \frac{1}{L}\int E dt = i(t)$$

or $$C\ddot{E} + \frac{1}{R}\dot{E} + \frac{E}{L} = \frac{di(t)}{dt}$$

The analogy used in this section is the Mass-Capacitance analogy. This follows the principles outlined in the preceding section on the combination of mechanical elements. Mechanical elements that are attached to a fixed member are shown "grounded". The ground connection also applies to mass elements that develop inertia forces. These forces have magnitudes proportional to the acceleration with respect to a fixed datum. Existing forces are usually placed at the left side of the diagram and elements to the right.

## B - 4.7 Combination of Mechanical Elements

As with electrical elements, one first has to determine whether the elements are coupled in series or parallel.

a) Parallel system (Single Degree of Freedom)

A parallel system Fig.55 has the same response for all components. The total force acting on the system is the sum of the forces acting on the individual components.

$$Z_A = \frac{F}{\ddot{x}}$$

$$Z_A \text{ for system} = \frac{F_m + F_{k1} + F_{k2}}{\ddot{x}}$$

$$Z_{AS} = Z_{AM} + Z_{k1} + Z_{k2}$$

b) Series System (Single Degree of freedom)

A series system Fig.56 has the same force acting on all components. The total response of the system is the sum of the individual system's component responses.

$$Z_A = \frac{F}{\ddot{x}}$$

For a series system however it is much easier to use Mechanical Mobility - $M_A$ (acceleration based) Mechanical Mobility is the reciprocal of Mechanical Impedance.

$$M_A = \frac{\ddot{x}}{F}$$

$$M_A \text{ for system} = \frac{\ddot{x}_m + \ddot{x}_k + \ddot{x}_c}{F}$$

$$M_{AS} = M_{AM} + M_{AK} + M_{AC}$$

$$Z_{AS} = \frac{1}{Z_{AM}} + \frac{1}{Z_{AK}} + \frac{1}{Z_{AC}}$$

The table shown in Fig.54 shows several schematic impedance diagrams. Mechanical symbols are used rather than the analogous electrical symbols. In these figures the point of excitation determines partly, if the combination of elements are in series or parallel.

B - 4.8 Response of Mechanical Systems

A) Single Degree of Freedom

The analysis given below will be for acceleration based Mechanical Impedance $Z_a$ , but a similar approach can be used for velocity or displacement based impedances.

The acceleration based impedances can be calculated for the three components in Fig.57 as a function of the forcing frequency. These impedances can be shown as curved and straight lines on the normal plot, or all straight lines on a log-log scaled impedance chart.

The system as shown in the figure is a parallel system, thus all the elements have the same velocity and the force impressed on the system equals the sum of the component forces.

$$Z_{AS} = Z_{AM} + Z_{AK} + Z_{AC}$$

The equation is a complex number.

$$Z_{AM} = m$$

$$Z_{AK} = \frac{-K}{w^2}$$

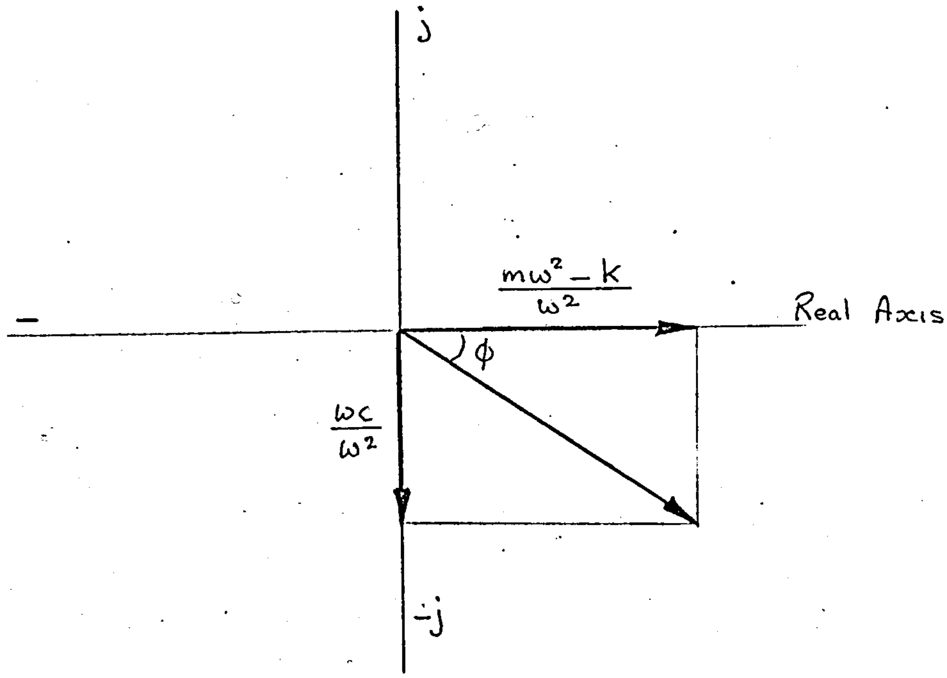
$$Z_{AC} = \frac{-jc}{w}$$

$$Z_{AS} = m - \frac{K}{w^2} - \frac{jc}{w}$$

$$Z_{AS} = \frac{mw^2 - K - jwc}{w^2}$$

$$Z_{AS} = \frac{F_s}{a_s}$$

The modulus of impedance, which is the quantity usually plotted, can be found by taking the square root of the sum of the squares of the real and imaginary parts.




$$|Z_{AS}| = \frac{\{w^2c^2 + (mw^2 - K)^2\}^{\frac{1}{2}}}{w^2}$$

$$\tan \phi = \frac{wc}{mw^2 - K}$$

The intersection of the spring and mass impedance lines occurs at the undamped natural frequency of the system. At this frequency a finite force causes infinite motion, thus the impedance at the natural undamped frequency is zero.

Thus solving this equation for w gives the familiar expression for natural frequency $w_n^2 = \frac{K}{m}$

System resonance occurs when the system impedance has a zero real part. Phase angle $\phi$ is then $-\pi/2$ (-90°), the system is entirely controlled by the damper force. At this point the system modulus of impedance curve is tangent to the damper impedance line.

The modulus of impedance curve shown in Fig. 58 is asymptotic to the spring impedance line at low forcing frequencies. This is also shown by the phase angle diagram (Fig. 58 ). At low frequencies $\phi$ lies between -180 and -90° ($-\pi$ ). Thus for this system we can say that for frequencies below the natural frequency the system is "spring controlled". The modulus of impedance curve is also asymptotic to the mass impedance curve at high forcing frequencies, this is also shown by the phase angle diagram (Fig. 58 ) at high frequencies $\phi$ lies between -90° and 0° . Thus for this system we can say that for frequencies above the natural frequency the system is "mass-controlled".

B) Single Degree of Freedom (System with base excitation)

The previous analysis dealt with a system in which the mass was being excited and the spring damper elements were grounded. The human body however in a seated posture in which the driving point for the impedance measurement is the subject's seat, is more analogous to a two-degree of freedom system with base excitation. The system is shown in Fig. No. 59 which also shows the impedance diagram.

This system is a combined series-parallel system; the spring and damper are in parallel, and the spring-damper system is then in series with the mass.

The impedance of the spring-damper system is found first:-

$$Z_{ACK} = Z_{AC} + Z_{AK}$$

$$= \frac{-cj}{w} - \frac{k}{w^2}$$

$$= \frac{(-cwj + k)}{w^2}$$

This can then be combined with the mass - as the two are in series the mobility is used to simplify the calculations.

$$M_{ACK} = \frac{-w^2}{cwj + k}$$

$$M_{AM} = \frac{1}{m}$$

$$M_{AS} = M_{ACK} + M_{AM}$$

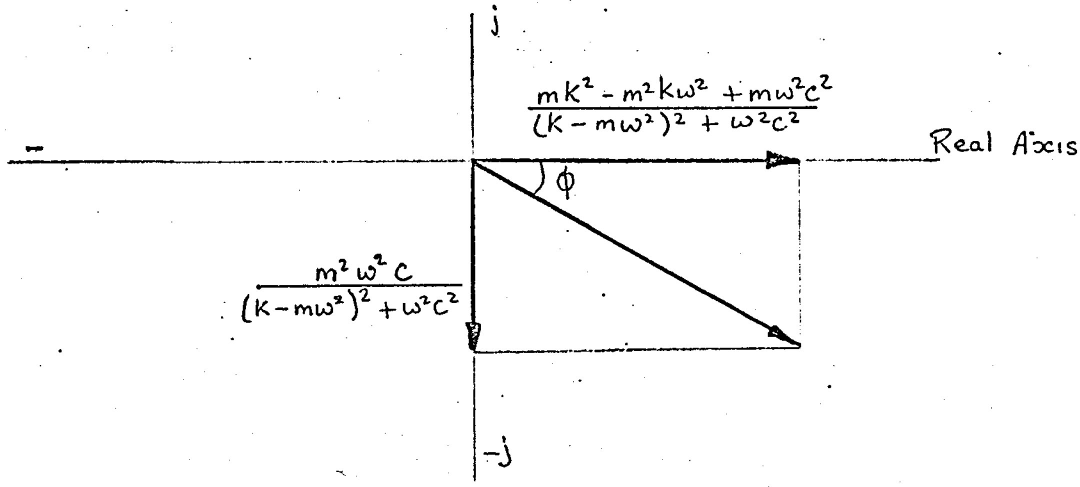
$$= \frac{-w^2}{cwj+k} + \frac{1}{m}$$

$$= \frac{cwj+k - mw^2}{m(cwj+k)}$$

$$Z_{AS} = \frac{m(cwj+k)}{cwj+k - mw^2}$$

$$= \frac{mk^2 - m^2kw^2 + mw^2c^2 - m^2w^2c\,j}{(k-mw^2)^2 + w^2c^2}$$

The modulus of impedance is found by the square root of the sum of the real and imaginary parts.




$$Z_{AS} = \frac{\left[(mk^2 - m^2kw^2 + mw^2c^2)^2 + m^4w^4c^2\right]^{\frac{1}{2}}}{(k - mw^2)^2 + w^2c^2}$$

$$\tan \phi = \frac{-m^2w^2c}{(mk^2 - m^2kw^2 + mw^2c^2)}$$

The intersection of the spring and mass impedance lines occurs at the undamped natural frequency of the system. At this frequency an infinite force will be impressed on the driving point for a finite motion of the point. When this occurs the impedance of the undamped system would be infinite.

$Z_{AU}$ = Impedance of undamped system

$M_{AU}$ = Mobility of undamped system

$$M_{AU} = M_{AK} + M_{AM}$$

$$= \frac{-w^2}{k} + \frac{1}{m}$$

$$= \frac{k - mw^2}{km}$$

$$Z_{AU} = \frac{km}{k - mw^2}$$

$Z_{AU}$ will be infinite when $k - mw_n^2 = 0$

i.e. when $k = mw_n^2$

$$w_n^2 = \frac{k}{m}$$

$$w_n = \sqrt{\frac{k}{m}}$$

The damped system resonance occurs when the system impedance has a zero real part. The phase angle $\phi$ is equal to $-90^\circ$ the system response - as was the case with the former system, is controlled entirely by the damper force, and this gives the modulus of impedance curve a finite value at the resonant frequency. The modulus of impedance and phase angle curves are shown in Fig. No. 60.

The modulus of impedance curve is asymptotic to the mass impedance line at low forcing frequencies; this is also shown by the phase angle diagram as $\phi$ lies between $0^\circ$ and $-90^\circ$ $(-\pi/2)$. The system is "mass-controlled" below the resonant frequency, above this frequency however the phase angle lies between $-90^\circ$ and $-180^\circ$ $(-\pi)$ and the modulus of impedance curve becomes asymptotic to the spring impedance line at high forcing frequencies - the system is "spring-controlled" at frequencies above the system's natural frequency.

When the modulus of impedance plots for the two systems are studied they show that the resonance of the first system (A) causes a "valley" in the modulus of impedance. The second system (B) at its resonant frequency has a "peak" in the modulus of impedance curve which corresponds to an anti-resonance of the system. This analysis of the two curves however can help in the analysis of a complex mechanical structure - such as the human body - whose modulus of impedance curve contains several "valleys" and "peaks" corresponding to "resonances" and "anti-resonances".

C) <u>Single Degree of Freedom</u> (Displaying resonance and anti resonance).

The two previous systems described display either a resonance or anti-resonance, however, by adding an extra mass to system B, will result in a system which displays resonance and anti-resonance characteristics.

This system is shown in Fig. No. 61 together with the impedance diagram of the system.

The initial system response, that is at very low frequencies (V.L.F.) is mass like (m), as in system B. An anti-resonance (A) occurs with the spring-damper system and $m_2$. The very high frequency (V.H.F.) response is also mass like as in system A. This change from the spring-damper control to the mass-control of m causes the resonance (R ) of the system. The modulus of impedance and phase angle plots for the system are shown in Fig. No. 62. The complete analysis for this system can be obtained by combining $m_2$ with system B to obtain:-

$$Z_{AS} = \frac{m_1 (cwj+k)}{cwj + k - m_1 w^2} + m_2$$

$$= \frac{m_1cwj + m_1k + m_2cwj + m_2k - m_1m_2w^2}{cwj + k - m_1w^2}$$

$$= \frac{(m_1 + m_2)\ cwj + (m_1 + m_2)k - m_1m_2w^2}{cwj + k - m_1w^2}$$

$$m' = m_1 + m_2$$

$$= \frac{m'cwj + m'k - m_1m_2w^2}{cwj + k - m_1w^2}$$

The modulus of impedance can be found by taking the square root of the sum of the squares of the numerical values of the real and imaginary parts of the impedance equation. The tangent of the phase angle can be obtained by dividing the numerical value of the imaginary part, by the real part of the impedance.

i.e. $$Z_{AS} = \sqrt{(\text{Imag})^2 + (\text{Real})^2}$$

$$\tan \phi = \frac{(\text{Imag})}{(\text{Real})}$$

$$\phi = \tan^{-1} \frac{(\text{Imag})}{(\text{Real})}$$

The similarity of the impedance to that of system B can be seen but the $m_1m_2w^2$ term however makes the V.H.F. response "mass-controlled" ($m_2$) and this causes the resonance R

D) Two Degrees of Freedom system (A parallel system)

The system is shown in Fig. No. 63 together with the impedance diagram. The algebraic analysis for the complete system impedance can be obtained by adding an additional spring-damper-mass system to system C.

$$Z_{AS} = \frac{m_1(c_1wj + k_1)}{(c_1wj + k_1 - m_1w^2)} + \frac{m_2(c_2wj + k_2)}{(c_2wj + k_2 - m_2w^2)} + m_3$$

The calculations to obtain the modulus of impedance and phase angle are somewhat lengthy, and have been ommitted as they can be found in most classic texts on vibration theory. The main thing of interest is the modulus of impedance and phase angle plots, which can be seen in Fig. No. 64.

This type of system (D) is of particular interest as it was used by Coermann (without $m_3$) and by Suggs, Abrams and Stikeleather as a dynamic simulator for the human body during steady state sinusoidal vibrations.

The curves indicate that the system possesses two anti-resonance "peaks" and two resonance "valleys". The V.L.F. and V.H.F. response of system D will be mass-controlled. The change from the V.L.F. mass-control to spring-control causes the first anti-resonance; as the frequency increases further the masses again control the response and a resonance R occurs. As the frequency continues to increase the springs response becomes more important and a second anti-resonance occurs as the system changes to being mass-controlled.

E) <u>Two degree of Freedom System</u> (A series system)

The system is shown in Fig. No. 65 together with the impedance diagram. The complete algebraic analysis has to be done using the mobility of each spring-damper-mass sub system, as they are combined in series.

$$M_{AS} = M_{A1} + M_{A2}$$

$$M_{A1} = \frac{c_1wj + k_1 - m_1w^2}{m_1(c_1wj + k_1)}$$

$$M_{A2} = \frac{c_2wj + k_2 - m_2w^2}{m_2(c_2wj + k_2)}$$

$$M_{AS} = \frac{(c_1wj + k_1 - m_1w^2)}{m_1 (c_1wj + k_1)} + \frac{(c_2wj + k_2 - m_2w^2)}{m_2(c_2wj + k_2)}$$

$$M_{AS} = \frac{(m_1 + m_2)(k_1k_2 - c_1c_2w^2 + c_2k_1wj + c_1k_2wj) - m_1m_2w^2(k_1 + k_2 + jc_1 + jc_2)}{m_1m_2(k_1k_2 - c_1c_2w^2 + c_2k_1wj + c_1k_2wj)}$$

Let $W = (k_1k_2 - c_1c_2w^2 + c_2k_1wj + c_1k_2wj)$

Then $$Z_{AS} = \frac{W}{\left(\frac{(m_1 + m_2)}{(m_1m_2)}\right) W - w^2 (k_1 + k_2 + jc_1 + jc_2)}$$

The modulus of impedance and tangent of the phase angle can be found from the numerical values of the real and imaginary parts as outlined for system C. The modulus of impedance and phase angle plots are again of particular interest, for this type of system (E) was postulated by Dieckmann as a dynamic simulator for the human body during steady state sinusoidal vibration conditions.

The modulus of impedance and phase angle curves for the system are shown in Fig. No. 66. The curves are similar to those shown for system D, and contain two anti-resonance "peaks" and two resonance "valleys" (if $m_3$ is not zero). The similarity of the curves between systems D and E is to be expected for if system E had been driven from $m_2$ and not $m_3$ it would be a parallel system like system D.

## B - 4.9 Discussion of the system impedance plots

The modulus of impedance curves and phase angle curves shown are only general curves to indicate the basic characteristics of the five systems analysed. The actual shape of the plots for any given system would depend on the system parameters and the degree of linearity of the system. A parameter which has a considerable influence on the shape of the impedance plots obtained is the degree of damping present, for the relative value of the damping present to that of the mass and elasticity can determine whether or not the phase angle curve crosses the 90° phase line. The systems studied have been simple linear systems, and from the algebraic analysis of systems D and E it can be seen that a similar analysis on a many degree of freedom system would be a difficult and complicated exercise.

The modulus of impedance and phase angle plots can however lead to an understanding of the mechanical response of the system under a given vibrational input, and can also help to produce an accurate indication of the systems response to a different input condition. More comprehensive text on the impedance technique for the analysis of mechanical systems can be found in Slater (1969) and Church (1963)

## B - 5 Analogue Computer Processing Units

### B - 5.0 Introduction

The analogue computer processing units were based on those suggested by Nathan (1969). Nathan had carried out a theoretical study of a method of processing force and velocity signals on an analogue computer to obtain modulus of impedance and phase angle. Nathan had not however tested these circuits in practice. The other texts referred to by the author were Key (1965), Blum (1968) and Jackson (1960).

The processing units were patched into the analogue computer as three separate units. The first unit was a frequency-voltage conversion unit, the second unit was to obtain the modulus of impedance; the third was the phase angle monitoring unit. The units were first patched into a large analogue computer - a Pace T.R.48 (Plate No. 2. ) to enable development of the circuits before their use in the experimental programme. The small analogue used to monitor the modulus of impedance during the experiment was a "logi-kit" computer with a Tranchant T.B.O.6 analogue multiplier unit. A diagrammitic view (Fig. No. 67 ) of the laboratory set up, shows the position of the "logi-kit" analogue computer relative to the rest of the apparatus used in the studies.

## B - 5.1 Frequency-Voltage Conversion Unit

The frequency-voltage conversion unit is shown in Fig. No. 68 the input to the unit is Signal No.1, the signal from the actuator's control unit.

The three signals which are shown entering the computer are Signal No. 1 from the actuator control unit, Signal No. 2 and No. 3 are the Force and Acceleration signals respectively from the force cell. Signal No.1 is a square wave whose frequency is the same as the actuator's operational frequency. The amplitude of this square wave is also directly proportional to the operating frequency of the actuator, thus on full-wave rectification, the resultant voltage will be a D.C. voltage directly proportional to the operational frequency of the actuator. This voltage can then be used to drive the x - plot of the x - y plotter, enabling Modulus of Impedance and Phase Angle to be plotted out directly against the actuator's operating frequency.

The full-wave rectification circuit employed is a conventional one, a description of the circuit is given in Blum (1968) Page 148.

This circuit was used for the Pilot Study, and for the phase angle plots of the main experimental programme. This square wave (Signal No.1) was rectified using a diode bridge network during the main experimental programme for the Modulus of Impedance results. This was because the logi-kit analogue computer had not enough available amplifiers to facilitate the patching of a full-wave rectification unit, as was possible on the Pace Computer.

B - 5.2. Modulus of Impedance Monitoring Circuit.

The modulus of impedance monitor was first patched onto the Pace analogue computer in the pilot study. The circuit diagram is shown in Fig. No.73. In order to facilitate a direct readout of the modulus of impedance during the experimental programme, however, the circuit was also patched up on "logi-kit" analogue computer, circuit diagram Fig. 70. The circuit consists of three basic units, two maximum postive voltage detectors, and a dividing unit. As the modulus value of the impedance is required, the peak values of the force and acceleration sinusoids have to be monitored. The Modulus of Impedance is then computed by dividing these two peak values.

The peak voltage detector is a modified form of the simple lag transfer function circuit - Stewart and Atkinson. (1967) pp 80 - 88. The inclusion of the diode between the input resistor and the summing junction of the amplifier, enables the circuit to discriminate between increasing positive voltages and decreasing voltages, both positive and negative.

The circuit acts as an integrator for the positive increasing pulses with a time constant of .027 seconds. For decreasing, and negative pulses, the circuit acts as a leaky integrator with a longer time constant of 5 seconds.

The circuit will therefore detect the maximum postive amplitude of the sine wave with an element of decay. The decay will enable the detector to follow a sine wave whose amplitude is increasing or decreasing. A typical waveform obtained from the circuit is shown in Fig. 71.

The rate at which the circuit's output voltage follows the changes in the input sine wave amplitude depend on the time constant of the leaky integrator, and the frequency of the incoming sine wave. Thus for the author's experimental programme which involved an increase in frequency from 3Hz to 30 Hz the value of the time constants had to be a compromise between fluctuations in output voltage (shown as $\delta a$ in Fig. 71. ) for low frequency input sinusoids and the response to amplitude changes in the input voltage for high frequency input sinusoids.

The values used were found by experimenting with a series of values for the circuit components and inspecting the resultant output waveform on a display oscilloscope. This procedure is more fully reported in the pilot study section B - 6.

The resultant waveforms from the peak voltage detectors were then divided to form the Modulus of Impedance ratio - Force/ Acceleration.

This division is achieved by patching a multiplier into the feedback of a high-gain amplifier. The circuit set-up on the Pace computer used one of the computers quarter-square multipliers, the "logi-kit" computer used a separate analogue multiplying unit - a (Tranchant TB02, )this unit was such that it could operate from the "logi-kit" computer power unit.

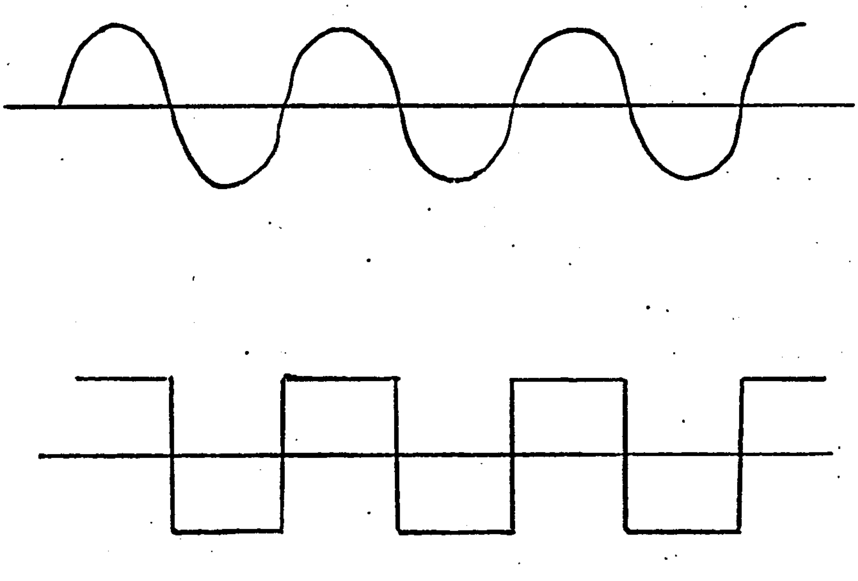
The gains of the input amplifiers on the force and acceleration are adjusted such that the accelerating signal voltage is always larger than that of the force, this is because the divider will not give an output greater than one machine unit.

B - 5.3 Phase Angle Monitoring Unit

The phase angle monitoring unit consists of two precision sign function circuits (Blum (1968) pp 150 - 2) a summer, and a leaky integrator.

The precision sign function circuit operates in such a way as to give a positive output of half a machine unit for any positive input voltage, and a negative output of half a machine unit for any negative input voltage. Thus the units will convert the incoming sine waves into square waves of the same frequency. (See Fig. No.25).

The resultant square waves are then added by the summer circuit. The comparator circuits are set to give a square wave whose output is half a machine unit, (single peak amplitude). The voltage output of the summer has a maximum output of one machine unit. The resultant output from the summer will depend on the phase difference between the two input waves. The resultant waveform will be of the nature shown in Fig. No. 72.



Input sinusoid.

Output waveform of the precision sign function circuit.

Figure No. 25

The figure shows that the waveform pulse length is directly proportional to the phase angle difference of the input sinusoids. The amplitude of the waveforms is constant at one machine unit (single peak amplitude). Thus the area of a single pulse is also proportional to the phase angle of the input sinusoids. If a modified peak voltage detector circuit is used, the circuit will give an output proportional to the pulse areas of its input waveform. This is achieved by modifying the time constants used in the circuit from those used in the modulus of Impedance monitoring circuit. The circuit again acts like an integrator for positive pulse butthe time constant is much longer, the value being .6 seconds. The leaky integrator has to have a longer time constant, as the pulse width is less than that in the modulus of impedance signals. The leaky integrator time constant was 132 seconds.

The circuit used is shown in Fig. No. 69. A rectifying circuit was added during development, between the summer and the leaky integrator circuits. This, however, was not satisfactory because with no phase difference between the input sinusoids the integrator could not "leak" and its output voltage just continued to rise.

The output voltage from the monitoring circuit is independent of frequency (provided the comparators can still operate). Althoughh the frequency of the input sinusoids affects the pulse width it also changes the number of pulses per unit time. This means that for a given phase difference the pulse area of the positive pulses is the same, irrespective of frequency.

## B - 6 The Pilot Study and Calibration

### B. - 6.1 Introduction

The pilot study was carried out with two aims in view. The first being to test and modify if necessary the analogue computer processing circuits before the main test programme. The second being to devise a suitable calibration technique, and calibrate the equipment before the first set of experimental runs using subjects.

The pilot study was carried out by means of recording the force and acceleration signals from the force cell on an F.M. tape recorder. The signals were recorded to enable them to be used repeatedly without necessitating the repeated use of the actuator. This meant that the signals could be processed on the Pace Analogue Computer which was situated in a different building to the actuator. The force and acceleration signals also had to be reproduced quickly and accurately to enable changes to be made in the analogue computer processing unit parameters.

The force cell was loaded first with known masses, whose impedance could be easily calculated, and then with a 40 kg mass on top of a foam rubber cushion 50mm thick. (The foam rubber cushion was used to introduce a phase difference the force and acceleration signals, no phase difference is present when a pure mass is used. Theory Section B - 4). The signals from the force cell for the above loading conditions were recorded and used for the pilot study.

A signal from the actuator electronic control unit proportional to the actuator frequency was also recorded. This was to enable the Modulus of Impedance and phase angle traces to be plotted against actuator frequency. The signal obtained was a square wave whose amplitude was proportional to the actuator frequency. The square wave was rectified using a full-wave rectifier circuit on the Pace analogue computer, (Fig. No. 75 ). This circuit could not, however, be used during the main experimental programme as not enough amplifiers were available on the small "logi-kit" analogue computer. The square wave was therefore rectified using a diode bridge network with a small capacitor placed across its output to give a smooth trace on the x-y plotter, during the modulus of impedance tests.

B. - 6.2. <u>Analogue Computer Processing Unit - Development and Calibration.</u>

The first unit to be patched up and calibrated was the full-wave rectifer unit (see Fig. No. 73 ) which was required to drive the x ordinate of the x-y plotter. The amplifier acted as an inverter to the signal, enabling the connection of the computer and x-y plotter "earth" without reversing the normal direction of plotting. The plotter gain was then set so that the frequency sweep of 3 to 30 Hz occupied most of the axis. The calibration was achieved by recording about 15 seconds of the square-wave from the actuator control unit, with the actuator set at a fixed frequency, the frequency set being recorded on the voice track.

The actuator was then re-set to the next frequency, and so on at 1 Hz intervals between 3 - 10 Hz and 2 Hz intervals from 10 to 30 Hz. The resultant calibration of the frequency axis is shown in Fig. No. 79.

The frequency axis was calibrated in a similar manner in the test laboratory, when the diode-bridge rectifier was used. The calibration was checked by marking the 3, 15 and 30 Hz frequencies at the beginning of each subject text run.

As a result of calibrating the frequency axis, the modulus of impedance monitoring circuit could be developed. The development involved the selecting of suitable parameters for the leaky feedback resistor and capacitor, setting the gains for the input amplifier and the x - y plotter.

The cast iron weights were used as they have a known modulus of impedance, and phase angle plots. The feedback parameters were adjusted during the pilot study to make the modulus of impedance plots as linear as possible, when the two values shown in Figure No. 73 were selected (i.e. 5 MΩ and 1 MF) the modulus of impedance plots for the cast iron weights were plotted out against vibration frequency, the plot obtained was similar to that shown in Fig. No. 80. Also shown is a modulus of impedance plot for a 40 kg weight placed on a foam rubber cushion, Fig No. 76. To avoid any confusion between the effects of hysteresis due to the foam rubber cushion, and the changing of the system parameters the recording for the 40 kg weight on the cushion was made between 3 - 30 Hz only.

The calibration of the modulus of impedance axis was carried out in the laboratory with the "logi-kit" computer in a similar manner to that described above for testing the monitoring circuit. The charge amplifiers were set up as detailed in Appendix 3, the analogue multiplier was set up with reference to the manufacturers instructions, and checked using known voltages. The actuator was then loaded with cast iron weights and plots of their impedance were made for the experimental frequency range 3 - 30 Hz and then 30 - 3 Hz. The first calibration shown in Fig. No. 80. was carried out and used for the majority of the experiments. This calibration was however not as linear as was hoped, and each test run required tabulation (using the calibration graph) before further analysis could be made. The two input amplifiers were then replaced with amplifiers whose characteristics were linear in nature over the experimental frequency range, a second calibration was carried out and is shown in Fig. No. 81.

The phase angle monitoring unit was developed and calibrated on the Pace analogue computer first using a phase lock function generator, and then the force and acceleration signals recorded in the vibration laboratory for the cast iron weights and the 40 kg weight on a foam cushion.

The circuit for phase angle monitoring had originally been designed with bistable units converting the input sinusoids to square waves, this did not however prove successful due to triggering problems. The bistables were then replaced by precision sign function circuits, descriptions of the circuit are given in Section B - 5. The development of the circuits involved the setting of the "leaky" integrator feedback components i.e. the $6\mu F$ capacitor and the $22 M\Omega$ resistor, these values were used after tests with sinusoids of known phase differences. The main problem encountered in fixing the feedback parameters was to get the circuit to respond quickly to the changes in phase of the input sine waves and to maintain a smooth trace over the experimental frequency range (3 - 30 Hz) The values chosen proved a successful compromise between these two requirements. (Fig. No. 74.)

A calibration chart for the phase angle monitor is shown in Fig No.82 also shown is the phase angle plot obtained for the 40 kg weight on a foam cushion Fig. No.77. The actual plot for system was the curve in the 0 to 90° part of the graph, the lower curve was obtained by inverting the force signal. This was done to check the linearity of the response of the phase angle monitor, the two curves should be, and in fact are symmetrical about the 90° line.

B - 6.3. Accuracy of Impedance monitoring circuits

The overall accuracy of the measurement of the modulus of impedance and associated phase angle depends not only on the analogue monitoring units, but also on the subject's ability to maintain his posture, and repeat a given posture if necessary. Coermann (1963) noted that:- "The impedance obtained and phase angle curves remained in the range of ± 10 percent (which is about the accuracy with which such impedance and phase angle measurements can be taken). This minor variation was probably due to the fact that the subject changes his posture slightly"

The accuracy of the results obtained from the analogue computer units can be computed from the impedance curves for the solid weight and the phase angle calibration using the phase lock function generator. The accuracy being dependent on frequency and magnitude. The accuracies for the modulus of impedance monitoring unit at various impedances and frequencies are shown in Table No. 4. The phase angle monitor was such that the error did not exceed ± 2% of the true phase difference for the frequency range 3 - 30 Hz. and over the phase difference range of $0^{\circ}$ - $180^{\circ}$

The total vibration time for any given posture did not exceed ten minutes and it was hoped that this would minimise the effects of posture change. Coermann noted that his subjects had to maintain the same posture for a period of about an hour. The differences in the postures adopted by subjects can be seen when looking through the modulus of impedance and phase angle graphs (Section B - 8). The erect, and relaxed postures adopted by some subjects only differ slightly, where as the plots for other subjects show considerable variation.

The computation of a figure for the accuracy of subject ability to assume and maintain a given posture is almost impossible, as there is no absolute reference posture which can be adopted by each subject to be used as a datum for other posture types. The effects of subject physique on the impedance recording are also difficult to assess and would require a comprehensive anthropometric study of each subject prior to the impedance analysis.

## B - 7 Experimental Method

### B - 7.1. Subject Selection and Instruction

The potential subject was first asked to fill in a confidential form, (see pages 540 to 543 ) the information from this was then assessed by the experimenter, together with the research associate in charge of the subject's files, to determine the subject's suitability for the experimental programme. The suitability of the individual subject was assessed, using the proposals laid down in the "Draft Guide on the Safety aspects of Human Vibration Experiments". (Draft 4, 1970) referring to Section 3 - "Selection of Subjects for Vibration Experiments". The subject was told briefly how the actuator worked:-

"The actuator is an electro-hydraulic type. The hydraulic power pack in the small room next door supplies high pressure hydraulic oil to the actuator through flexible pipes. The oil is controlled by a valve which responses to electric signals received from the operator's control console. There are three emergency stop buttons in the laboratory, one of which you will be holding during the experiments."

Any questions asked about the apparatus by the subject were answered and the actuator was set into motion, and stopped by the emergency stop button to show the subject how gently the piston was brought to rest by the "emergency stop" system.

The subject was then told about the object of the experimental programme:-

"The aim of the experiment is to monitor continuously your mechanical impedance. Mechanical impedance being the ratio of the maximum force input to the maximum acceleration input to the body; and can be thought of as being analogous to the electrical impedance of an electronic device.

The hand-held button you will be given during the experiments will gently stop the actuator, use it if you feel any discomfort or become apprehensive during the experiment."

The subject was then asked if he had any questions about the vibration apparatus or the experiment before starting the tests. Before any tests were carried out the subject was asked to sit on the actuator which was then gently brought up to the acceleration level used in the tests. When the required acceleration level was reached the frequency was varied as in the tests, the subject was then asked to stop the actuator, using his stop button - to show him how gently the vibration ceased.

After a few minutes on the actuator, and before the experiments were carried out, the foot rest was adjusted for individual subjects, this being necessary to standardise the seated posture adopted by each subject (Section .B-7.2)

While the foot rest was being adjusted the subject was given a brief description of the impedance measuring apparatus and some of the possible uses of the data:-

"The aluminium block you are sitting on contains three force cells, and an accelerometer to measure the force input to your body, and the associated acceleration. The maximum values of the force and acceleration signals are measured and divided by a small analogue computer, the resultant signal, which is proportional to your mechanical impedance is fed to the y axis of an x-y plotter, the x axis is driven from the actuator control console. The resultant plot will be that of your mechanical impedance between 3 and 30 Hz. The results will be compared to those obtained during impact tests at the University, and could also be used by designers of seating systems for off-the-road vehicles, where the protection of the operator from vibration is of particular interest."

The subject was then asked to adopt an erect or relaxed seating posture, for the first of the test runs. The instructions given to the subject with regard to posture are contained in the following section.

B. - 7.2 Posture.

The results of previous investigations show what a marked effect subject posture has on the mechanical impedance results obtained, thus a standardisation of the posture types is necessary to reduce variation in the impedance curves due to the effects of subjects adopting difference postures. Two seating postures were used - an erect posture and a relaxed posture, for both postures the subject was sitting with his feet supported on an adjustable foot rest.

The subject was first asked to sit so that his ischial tuberosities were centrally placed, or slightly to the rear of the force cell. (This being dependent on subject height). An inclinometer was then placed on top of the subject's thigh, the foot rest being adjusted so that the subject's thigh was horizontal. (See Plate No. 5 ). The position of the subject's legs and the setting of the foot rest was the same for both seated postures. The subject was also asked to place his hands on his thighs as shown in the photographs, for the seated postures, and with his arms down by his side for the standing postures.

B - 7.2.1 <u>Sitting Erect Posture</u>

Following the adjustment of the foot rest, the subject was asked to sit upright, looking straight ahead. (See Plates No. 4 and 5 ). The subject assumes this posture with the vibrator stationary, and when the motion starts it could easily distract the subject and cause his to change his posture slightly. To minimise this effect the subject, when seated on the stationary actuator, was asked to focus his eyes on a particular point on the piece of apparatus directly in front of him and about

3 - 4 metres away. This distance was used so that the subject could still concentrate on this point even during the vibration, without moving his head).

B - 7.2.2. Sitting Relaxed

The angle of the subject's thigh was checked using the inclinometer and the foot rest adjusted if necessary. When the foot rest was set correctly the subject was asked to adopt a relaxed sitting posture. The subject was asked to imagine he was a passenger in a vehicle, and could look around the laboratory area in front of him if he wished.

A typical relaxed posture is shown in Plates No. 6 and 7 This meant that the first set of recordings were made with the subject in a sitting erect posture, which was defined for him, the second set of recordings were then made with the subject sitting in a relaxed posture, which was not so specifically defined, and was thus left more to the discretion of the individual subject.

B - 7.2.3 Tracking Posture

The subject was asked to sit with the control joystick .n his thighs, hold the control box with one hand and control the joystick with his other hand. The posture is shown in Plate No 8- the back rest shown in the photograph was not fitted during the impedance measuring tests.

The tracking task used was of a compensatory nature, the subject having to keep a spot on a display oscilloscope moving as horizontally as possible, a reference trace is also present to assist the subject. The subject was asked to concentrate on the tracking task during the entire experimental programme of impedance recordings.

The subject held emergency stop button, which for previous experiments had been held by the subject in one hand, was incorporated into the joystick control box to enable the subject to carry out the tracking task easily, and still be able to stop the actuator if he wished.

B - 7.2.4 Standing Postures

The impedance measurements made for subjects in the standing posture involved a small number of subjects familiar with the experimental aims and procedure. The subjects were thus told to adopt the required standing postures with the following instructions:-

a) Standing Erect Posture

The subject was asked to stand centrally on the force cell, with his feet about 5 cms apart; and then to stand upright with his legs straight and also to look straight ahead. The legs have to be kept straight, as, if they are allowed to bend to any degree at the knees most of the vibration is "absorbed" by the legs before reaching the subject's trunk.

b) Standing Relaxed Posture

The subject again stood centrally on the force cell with his feet about 5 cms apart. He was asked to stand as if on a train, or on an escalator, his legs were not to be as rigid as in the erect posture. The subject was however asked not to deliberately bend his legs at the knees, but to stand in a normal relaxed posture.

c) Standing Knees Bent Posture

The subject again stood centrally on the force cell with his feet about 5 cms apart. He was then asked to stand with his legs bending at the knees, to an angle of about 135° .

B - 7.3. Modulus of Impedance Recordings

When the subject had adopted the required posture the actuator was set in motion, and brought up to the required acceleration level of .5g (pk to pk). The actuator frequency was then varied manually from 3 to 30 Hz and then back to 30 to 3 Hz in four minutes.

The modulus of impedance curve was obtained directly for the full frequency sweep 3 to 30 to 3 Hz on the x-y plotter, the ordinates having been put on the graph before the test run. The plotter was not however started until the actuator had reached the required acceleration level. While the subject was still in the first posture the frequency was again swept manually from 3 to 30 to 3 Hz in four minutes, to enable recordings of the force and acceleration signals from the force cell to be made on an F.M. tape-recorder.

The subject was then asked to adopt a second posture for two more frequency sweeps, to obtain the modulus of impedance and tape-recording for the subject in the second posture. The tape recordings were identified using the voice track to give the subject number, the posture adopted and the tracks used for recording. This enabled the phase angle plots to be labelled correctly after processing.

The experimental programme for each subject was restricted during any one hour period to two different posture types, so that no subject was on the actuator for longer than 30 minutes. If more posture types were required, the subject was weighed and measured before the second experimental programme was undertaken.

B - 7.4 Subject Measurement

When the test runs had been completed and the tape recordings checked the subject was weight and measured. The subject was first weighed standing on the scales, and also measured to obtain his total weight and height. The subject was then weighed and measured in a sitting erect posture, to obtain his sitting weight and height. The sitting erect posture was the same as that adopted during the tests, the subject's thighs being set horizontal using the clinometer and an adjustable foot rest.

This data was recorded with the subject's identification, and then filed with two modulus of impedance plots, and two phase angle plots after processing. The table of data for the subjects used in the experiments is shown in Section B - 8, together with all the modulus of impedance and phase angle plots.

## B - 8 Experimental Results

The experimental results are shown at the end of Part B with the other figures and tables.
The table containing the data on the subjects used is Table No. 5

### B - 8.1. Explanation of the Impedance graphs (Figures No. 83 to No. 160

The graphs contained in the section are the plots actually obtained from the x - y plotter, during the experimental programme. The modulus of impedance values given on the impedance axis are, however, only correct at 15 Hz and not over the entire frequency range due to the non-linearity in the response of the monitoring system. The tabulated results (Tables No. 6 to No. 79 ) however represent the subjects true impedance at the given frequencies, and it is these tables which have been used in all additional calculations. The tables were computed using the calibration charts shown in Section B - 6. (Fig. No. 80 and 81) The tabulated results are also shown in graphical form in Fig. No. 161 to No. 195 . The sitting erect and relaxed postures are plotted on the same graph to enable comparison to be made for the individual subjects between the two seated postures.

On the phase angle graphs the values given on the phase angle axis are correct for the entire frequency range of 3 - 30 Hz. The graphs are also shown in tabular form at the given frequencies to enable additional processing of the results by a digital computer.

B - 8.2. Subject Identification

The experimental results shown in this thesis are indexed by means of a letter-number system, the key to which is given below.

Key:-

| | |
|---|---|
| Mx | refers to male subject, number x. |
| Fy | refers to female subject, number y. |
| - E | refers to a sitting erect posture |
| - R | refers to a sitting relaxed posture. |
| - T | refers to a sitting posture adopted by the subject while performing a tracking task. |

Standing postures are specified directly on the figures concerned.

e.g. M6 - R refers to male subject number six in a sitting relaxed posture.

## B - 9 Processing of Experimental Results

### B - 9.1. Statistically Processed Results

The results analysed statistically were those obtained from the fourteen male subjects in the sitting erect and relaxed postures. The other sets of results could not be fully analysed due to the limited number of experiments. The mean value and the standard deviation were calculated using the conventional formula for these parameters, on a P.D.P.12 digital computer. The results obtained are tabulated in Tables No. 82 to No. 119

The tabulated modulus of impedance results were also divided by the subjects weight to obtain a dimensionless parameter, which was also processed on the digital computer; the mean and standard deviation for this parameter are shown in Tables No. 95 and No. 100.

#### B - 9.1.1 Additional processing of results

The parameter obtained by dividing the modulus of impedance by the subject's weight was computed in an attempt to find a parameter which would more closely describe the behaviour of the human body to steady-state sinusoidal vibrations. If this parameter did perform this function, the dispersion of the parameter about its mean value would be less than that obtained with the modulus of impedance results, and thus the ratio standard deviation/mean value would be smaller for the parameter.

The ratio of the standard deviation/mean value was computed for the parameter and the modulus of impedance results, the ratio being expressed as a percentage. (Tables No. 120 and 121). These tabulated results were also shown in the form of histograms

(Fig. No. 196 and 197) an analysis of which is given in the discussion of the experimental results Section 3 - 10.

B - 9.2 Conversion of Acceleration based Impedance to Velocity based Impedance.

B - 9.2.1 Modulus of Impedance

The acceleration based modulus of impedance used in this study were converted to the velocity based modulus of impedance used by Coermann (1963) - dynes x sec/cm x $10^3$ and the lb - sec/ft units used by Suggs et al (1969).

If Za = Modulus of Impedance acceleration base

Zv = Modulus of Impedance velocity base

f = frequency of vibration

g = acceleration due to gravity

w = angular velocity.

$$Zv = \frac{Za \times w}{g}$$

$$= \frac{Za.\ 2\pi f}{g} \text{ Dynes x sec/cm x } 10^3$$

The value of $(2\pi f/g)$ was computed for the frequencies used in the experiments (Table No. 81.). The converted results are given in Tables No. 88 to No. 119.

B - 9.2.2 Phase Angle

The shape of the phase angle curves for the two types of impedance are the same, but as the acceleration vector leads the velocity vector by $90^o$, the pure mass phase line which is $0^o$

on the Za phase plot becomes 90° on the Zv phase plot. Thus the velocity based phase angle plots can be obtained from the acceleration based plots by adding 90° to the scale values on the phase angle axis of the acceleration based plots, the shape of the curve remaining the same.

## B - 10 Discussion of Results

### B - 10.1 Subject Panel

The subject panel used in the experiments consisted of fourteen male subjects and two female subjects from staff and students of the University. A table of subject data is contained in Section B - 8, and a summary of this data together with available subject data from previous investigations is shown in table No. 122 One of the main criteria for a subject panel is that it should represent the "general population" or a specified sector of the population (i.e. tractor drivers, typists etc.) The subject panel used could be described as a panel of young adults of the general population, and could in respect to age be called biased. Apart from Coermann (1963) very little subject data (or none at all!) is given in other research reports and this makes it difficult to assess the bias in the subject panels used by previous investigators.

The tabulated parameters show that the mean height and weight of the subject panels used by Coermann and Suggs et al are greater than the panel used for this study. Coermann's subjects were an average of 20% heavier but only 1.2% taller and those of Suggs about 10% heavier. These differences could be attributable to two factors - Coermann's subjects were about 9 years older (no data is available for the subjects of Suggs et al) than the Loughborough University subjects, and both Coermann's and Suggs' subjects were

American.

The weight and physique of the subjects may influence the shape of the modulus of impedance and phase angle plots; subject sex could however also be a factor, and female subjects were used in these studies to see if any significant differences existed between the sexes in their response to steady-state sinusoidal vibration. To obtain comprehensive impedance data, both sexes must be included and assessed separately if significant differences in their responses are found to exist.

B - 10.2 Sitting Erect Posture

The modulus of impedance results for the 14 male subjects are shown in figures No. 201, 202 and 203, 204 The plot of the mean shows a first anti-resonance (peak) just above 5 Hz the peak is 77 kg (Za) or 2.4 Dynes x sec/cm x $10^6$ ($Z_V$). A second anti-resonance is visible on both the acceleration based impedance plot and the velocity based curve at the frequency of 11 Hz, the peak is not however very pronounced. A third anti-resonance peak is also just visible at 14.0 Hz on the velocity-based impedance plot, but is not really perceivable on the acceleration based impedance plots. The resonance valleys are also just perceivable on the velocity based modulus of impedance curve at frequencies of 8.5 Hz and 12.5 Hz.

The phase angle curves for the male subjects, are shown in figures No. 207 and 208 . Although the phase angle plots do not cross the 90° line as would be expected for a true resonance or anti-resonance condition of a mechanical system. (When the phase angle is 90° the system is entirely "damper controlled" and this gives rise to a resonance or anti-resonance condition. Section B. - 4). The phase angle curve rises to a maximum phase difference (between force and acceleration) of 55° at 8.5 Hz. From this frequency to 30 Hz the curve drops steadily to 45° indicating that visco-elastic effects are still substantially effecting the whole-body response in the frequency range 15 - 30 Hz.

When considering the individual plots for the subjects, two subjects of above average build - M7 and M8 and two subjects of below average build M11 and M12 were compared to the mean value (for the 14 subjects) to ascertain what influence subject physique has on the modulus of impedance and associated phase angle. (Figures No. 225, 226, 231 and 232.)

The modulus of impedance plots for M8 and M7 show a much more pronounced first peak than do the curves for subjects M11 and M12. Considering the plots for M7 and M8, the plot for M7 shows a very flat peak for the first anti-resonance which extends over the frequency at which the second peak would occur, another peak is however visible just below 16 Hz. The plot for M8 shows the three anti-resonant peaks (the third however is not pronounced.) The phase angle plots for M7 and M8 are similar up to 7 Hz, then the curve for M7 turns upwards to a frequency of 12 Hz, indicating the influence of a large secondary mass which turns the phase plot towards the mass line. The two anti-resonances frequencies being so close appear on the modulus of impedance plot as a large and flattened peak.

The modulus of impedance plot for M11 is very similar in shape to that of M7; the plot for M12 however shows a very pronounced second anti-resonance which is not shown on the plots for M7 and M11. These differences in shape of the impedance curves do not however produce large differences in the associated phase angle plots. The modulus of impedance curves for subjects M11 and M12 lie well below the mean curve, indicating that the effects of physique are more marked for subjects of lighter build compared to subjects of above average build. This could be that the skeletal mass has less influence on the shape of the impedance curves than the major body organs which are suspended by visco-elastic tissues within the skeletal frame, or of the flesh and fatty tissues covering the frame, (this is perhaps particularly true of the erect posture where the spinal column is stiffer and provides a more rigid frame capable of transmitting the vibrational force).

The modulus of impedance curves obtained for the two female subjects (Figures No. 198, 199 and 200. ) are very similar to those obtained for the male subjects M11 and M12 - the male subjects of lighter build.* A first anti-resonant peak occurs at a frequency of 5 Hz and has a value of 61 kg (Za) or 1.65 Dynes x sec/cm x $10^6$ ($Z_v$). This first anti-resonance peak is flatter than that obtained for the 14 male subjects, (see figure No. 203) and the first anti-resonant frequency cannot be so easily identified. A second anti-resonance peak is visible at 10 Hz but not a third as was just visible for the male subjects.

Two resonance valleys are visible on the modulus of impedance plot for the two female subjects at just above 8 Hz and the second at 14 Hz. The two curves for the male and female subjects come together at about 24 Hz and the modulus of impedance $Z_a$ is then constant to 30 Hz. This meaning that above 24 Hz the amplitude of the input force sinusoid to the body to maintain the fixed amplitude acceleration sinusoid is constant.

* Figures No. 227 and 233.

When comparing the phase angle plots for the two female subjects (Figure No. 205 ) to the mean curve for the male subjects, considerable differences were found to exist between the curves for the two sexes. The phase angle had not reached $90^{o}$ for any of the male subjects in the sitting erect posture, with the exception of M2 and M10. (M10 - the phase angle only became $90^{o}$ for the frequency increasing from 3 - 30 Hz). The phase angle plots for both the female subjects showed phase angles greater than $90^{o}$. The phase angle for the two female subjects is greater than $90^{o}$ between 7 and 10 Hz, indicating that between these frequencies the influence of the elasticities (springs) within the body was greater than that of the body masses. At 10 Hz the female phase line rises sharply to a phase difference of $60^{o}$ and then follows closely the mean line for the male subjects to 20 Hz. At 20 Hz however the male phase line continues at an almost constant phase difference of $45^{o}$, while the female phase line turns towards the $90^{o}$ line again and has reached $80^{o}$ by 30 Hz. This difference was exhibited by both female subjects, and is shown clearly in figures No. 180 and 181.

B - 10.3. Sitting Relaxed Posture

The modulus of impedance results for the fourteen male subjects are shown in Fig. No. 201, 202 and 203, 204. The plot of the mean for the male subjects shows a first anti-resonance peak just above 4 Hz, the value of the modulus of impedance at the anti-resonance frequency is 74 kg (Za) or 2.0 Dynes x sec/cm x $10^6$ (Zv). A second anti-resonance peak is clearly on both the velocity and acceleration based impedance plots at 8.5 Hz and a thirdpeak just below 12 Hz (The third peak is not very pronounced on the acceleration based plot). These secondary peaks are much more pronounced than those visible on the erect posture curve. Two resonance valleys are also clearly visible on the velocity based modulus of impedance plot at frequencies of 7 Hz and 10.5 Hz.

The relaxed posture phase angle plot for the male subjects shows a steeper rise to an angle of 50° (at 8 Hz) but for the frequency range 8 - 12 Hz, follows the same curve as that for the erect posture. Above 12 Hz the curve turns slowly upwards to 30° (at 30 Hz) indicating that in a sitting relaxed posture the response of the body tends to become "mass-controlled" as the vibration input frequency increases.

The modulus of impedance plots for M7 and M8 are very similar in shape for the sitting relaxed posture, the phase angle also follows the same pattern rising sharply to about $60^{o}$ at 8 Hz and then remaining at approximately 8 - 14 Hz and falling from 14 - 30 Hz to a phase difference of about $30^{o}$. The phase angle plots for the male subjects of above average build are closer to the $90^{o}$ phase line than those for the sitting erect posture, indicating the greater influence of the "damper" elements within the body, to its response to steady-state sinusoidal vibrations. The "damper" influences being more prominent in subjects of above average build. (Figures No. 228, 229, 234 and 235.)

In considering the impedance plots for the female subjects, the modulus of impedance curve is similiar to that for the male subjects, the velocity based plot is more step like than the male plot but a first anti-resonance peak is clearly visible at a frequency of 4.5 Hz with an impedance of 62 Kg (Za) or 1.7 Dynes x sec/cm x $10^{6}$ (Zv). A second anti-resonance peak occurs at 7.5 Hz and a third at 10 Hz. (These peaks are much flatter than those obtained for the male subjects) The resonance valleys are thus not very pronounced but occur at 6 Hz and just over 8 Hz. The modulus of impedance plot for the female subjects is however lower than that for the male subjects between 7 and 16 Hz, this is because the second anti-resonance peak is larger in magnitude than that of the male subjects, and at a lower frequency ( 7.5 Hz - female, 8.5 Hz - male).

In comparing the mean phase angle curves for the two sexes (Fig No. 211 ) a considerable difference is found to exist, the maximum phase difference for the male subjects being nearly $60^\circ$ at a frequency of 9 Hz, whereas the curve for the female subjects has a maximum phase difference of $105^\circ$ at a frequency of 8 Hz. The curves then come together with a phase difference of about $10^\circ$ at 12 Hz, this phase difference is maintained to 30 Hz. The phase angle plots again indicate a much higher degree of body elasticity influencing the phase angle plot for the female subjects. (The difference between 2 - 12 Hz is too large to be attributed to a difference in mass, in fact the male subjects whose phase angle plots were the closest to those for the female subjects were males of above average build. This degree of elasticity is greater than that present in the sitting erect posture where the maximum phase difference was only just over $90^\circ$ for the female subjects. (Figure No. 236 )

B - 10.4 Tracking Posture

The use of a compensatory tracking task was to measure the subject's impedance in a stressed condition, to simulate the operator control situation. Comparing the mean curve for the three subjects, M1, M2 and M3 in the tracking posture (Figs No.215, 216) with the mean curves for the three subjects in the sitting relaxed and erect postures (Fig No212 and 213) show that the modulus of impedance curve for the tracking posture closely resembles that for the sitting erect posture.

The first anti-resonance peak is the same value as that for the sitting erect posture, but is at a frequency of just below 4.5 Hz. Second and third anti-resonance peaks are visible at 12 Hz and 18 Hz, with the impedance rising steadily from 22 Hz to 30 Hz as in the sitting erect posture. The phase angle plot too is very similar to the sitting erect posture curve. The differences shown in the modulus of impedance and phase angle curves would be wholly, or partly, due to postural differences, or to the level of stress experienced by the subjects in performing the tracking task.

The posture adopted by the subjects is shown in Plates No.1 and 8 (The back rest shown in Plate No. 8 was not present during the impedance studies). The subject held the control box in one hand, using the other hand to control the joy-stick, the subject also had to look down slightly to view the tracking task on the display oscilloscope. The changes in position of the second and third resonance peaks are probably due to these postural changes outlined above, but the influence of stress on the subjects impedance is much more difficult to assess. The level of stress experienced by the subject during the impedance measuring experimants depends on the subjects familiarity with the actuator and the seriousness with which he undertakes the tracking task.

The curves do show that during a stressed condition, such as a tracking task the body response to sinusoidal vibration is similar to that in a sitting erect posture. In the sitting erect posture the subjects muscles would be tenser, this has the effect of stiffening the visco-elastic elements within the body, a similar effect appears to be precipitated by stress.

B - 10.5 Standing Postures

i). The first two standing postures were assessed using two subjects - M1 of average build and M2 of above average build. The subject of above average build was used, as it was thought that his extra weight would have an influence on his impedance curves, compared to the subject of average build.

ii). Standing Erect

The standing erect curve for the modulus of impedance of M1* (the subject of average build) shows a first anti-resonance peak at 5 Hz with a value of 84 kg (Za) or 2·69 Dynes * sec/cm * $10^6$ ($Z_v$). A second anti-resonance peak is just perceivable at 8 Hz, after this frequency the modulus of impedance curve (Za) continues a smooth steady drop to a value of 7 kg at 8 Hz, the velocity based curve between 10 and 30 Hz is almost constant, the value being 1·7 Dynes * sec/cm * $10^6$. The phase angle curve for M1 shows that in the frequency range 8 Hz to 30 Hz the whole body response is "damper controlled".

The modulus of impedance curve for M2* (the subject of above average build) shows a very pronounced first anti-resonance peak at 5 Hz with a value of 115 kg (Za) or 3·68 Dynes * sec/cm * $10^6$ ($Z_v$). The second anti-resonance is visible on both Za and Zv plots and occurs at 8 Hz with a value of 56 kg (Za) or 2·88 Dynes * sec/cm * $10^6$. The curve now remains well above that of M1 until 16 Hz, a third anti-resonance peak is then visible at frequency just below 18 Hz (on the $Z_v$ plot), corresponding closely to the frequency at

* Figures Nos, 151, 152, 156, 157, 217, 218 and 219.

which the phase angle curve crosses the 90° phase line. (This third peak is also visible on the Za curve although it is not so pronounced). A very slight fourth peak is also visible on both curves at a frequency of 27 Hz, this corresponds closely to the frequency at which the phase angle curve crosses back over the 90° phase line. Thus during the frequencies 16 and 26 Hz the elasticities within the body control the whole-body response of subject M2. Resonance valleys are also clearly visible at approximately 7 Hz, just below 16 Hz and about 24 Hz. The valleys are not very pronounced for the second, third and fourth peaks, only really represent local maxima on the acceleration (or velocity) based modulus of impedance plot.

These differences in the whole body response of the two subjects M1 and M2 are most probably due to difference in physique between the two subjects, the influence of slight posture changes however cannot be ruled out completely. The effects of posture changes are very difficult to assimulate, and the experimenter can only instruct the subject and check that he appears to be adopting the correct posture, for it is the subject who has control of his internal muscle tensions. (It may however be that the body can consciously or by some reflex action become an "active" system capable of influencing its response to the vibrational input to minimise the effects on certain important body organs. Thus the tension of some of the muscles within the body may be only partially in the conscious control of the subject).

ii). Standing Relaxed

The modulus of impedance curve for M1* rises to a first anti-resonance peak at just below 6 Hz with a value of 86 kg (Za) or 2·8 Dynes * sec/cm * $10^6$. The curve does not then really possess any more prominent peaks or valleys, and the velocity based modulus of impedance plot is almost constant between 10 and 16 Hz when it dips slightly to a shallow valley at 18 Hz, rises to a flat peak at about 22 Hz, and another shallow valley at 26 Hz. After 26 Hz however the modulus of impedance ($Z_V$) rises to the final test frequency of 30 Hz. These shallow valleys and flat peaks in the frequency range 16 to 26 Hz are reflected in the phase angle plot, by slight changes in the phase difference between these frequencies. The phase angle difference decreases between 26 - 30 Hz indicating a move towards mass control at higher frequencies ( i.e. above 30 Hz), this is also shown on the modulus of impedance (Za) plot by an almost constant value of the modulus of impedance between 26 and 30 Hz (The modulus of impedance Za for a mass is a horizontal line on a Za plot).

The plots for M1 shows the differences expected between the two postures, when the results for the standing postures are compared to those for the sitting postures previously discussed, and results obtained by other researchers in the past. The plot for M2* however does not fit into this expected pattern.

The first anti-resonance peak is supressed well below its expected level, the peak occurs at a frequency of just over

* Figures No. 153, 154, 158, 159, 220, 221 and 222.

5 Hz and has a value of 88 kg (Za) or 2·78 dynes * sec/cm * $10^6$ ($Z_v$) the second peak occurs at a frequency of 9 Hz and has a value of 55 kg (Za) or 3·17 dynes * sec/cm * $10^6$($Z_v$). The curve ($Z_v$) then falls slowly to 18 Hz where it then follows the same curve as the standing erect posture.

The phase angle plots for M1 and M2* also show considerable difference, the phase angle plot for M1 (with the frequency increasing) rises to an almost constant phase difference of about 50° between 9 Hz and 30 Hz; with the frequency decreasing. However the phase difference remains at 50° from 30 Hz to 7 Hz. The phase angle plot for M2 with the frequency increasing rises steadily to a phase difference of about 100° at a frequency of 12 Hz and remains constant to a frequency of 24 Hz, but then falls steadily to a phase difference 55° at 30 Hz. This first part of the phase angle plot for M2 is similar in shape to that of M1, the plot with the frequency decreasing (from 30 Hz to 3 Hz) however follows a similar shape to that for M1 down to a frequency of 9 Hz. The phase difference was decreasing down to 9 Hz, at 9 Hz however the phase difference begins to increase again, to a value of 40° at a frequency of 7 Hz; the phase angle then falls quickly from 7 Hz to 3 Hz.

The phase angle plot for M2 with this "figure of eight" shape between 3 Hz and 10 Hz is similar to that for M1 in a standing with knees bent posture. The "figure of eight" shape is not so exaggerated for M2 in the standing relaxed posture,

* Figures No. 158, 159, 220 and 221.

indicating that M2 was allowing his knees to bend more than subject M1. The bending of the knees by M2 caused the suppression of the first anti-resonance peak in his modulus of impedance plot, as can be seen from the modulus of impedance plot for M1 standing with knees bent where the whole of the plot lies below the Za = M (or $Z_v$ = Mw) line.

iii). Standing knees bent

This posture was assessed using one subject of average build M1, the exact posture adopted by M1 is detailed in Section B - 7.

As detailed above the whole of the modulus of impedance plot lies below the Za = M (or Zv = Mw) line. A first anti-resonance peak is visible at a frequency just above 4 Hz on the velocity based plot. (This is also visible as a flattening out on the acceleration based impedance plot at 4 Hz). The value of the modulus of impedance at 4 Hz is 46 kg (Za) or 1·2 Dynes * sec/cm * $10^6$. A resonance valley is visible on both the acceleration, and velocity based plots at 6 Hz. The second peak has flattened out and spans the frequency range 6 Hz to 18 Hz, a third peak is visible at 22 Hz, the acceleration based impedance (Za) then remaining constant at about 10 Kg to 30 Hz. (Figs No. 155, 160, 223 and 224.)

The phase angle curve is also considerably different from any of the other postures. With the frequency increasing from 3 Hz to 30 Hz, the phase angle rises sharply to a peak of 75° at a frequency of just over 6 Hz, and then falls back to a phase difference of 30° at 14 Hz. From 14 Hz the curve rises gently to a phase difference of 40° at a frequency of about 22 Hz.

(corresponding to the third anti-resonance peak on the modulus of impedance plot). With the frequency decreasing however the phase difference follows the same line as for the frequency increasing up to 22 Hz, then the phase difference falls slowly to a value of 26° at a frequency of 9 Hz. In the frequency range 9 Hz to 3 Hz the phase angle rises sharply to a value of 112° at 5 Hz, and then falls quickly to a value of 10° at 3 Hz.

When the frequency is decreasing the phase angle line cuts the 90° line at 4 Hz and 6 Hz corresponding to the anti-resonance and resonance. The difference in the plots with the frequency increasing to the frequency decreasing is much more pronounced on the phase angle plots than for the modulus of impedance, for the standing postures, and with subjects M1 and M2.

B - 10.6 Individual Differences.

The differences existing between the impedance plots for individual subjects can be attributed to five main parameters:-

i) Physiological differences

ii) Postural differences

iii) The subject's sex

iv) Psychological differences

v) Differences in the subject's "active" response to vibration.

The changes in subject response to an increasing frequency, to that when the frequency is decreasing are difficult to attribute to one or more of the above parameters. The response of certain subjects is such that little difference (i.e. within the accuracy of the experimental measurements) exists in their response to an increasing or decreasing vibration frequency; while that of others shows a significant difference in response on the modulus of impedance, phase angle plot, or both.

These differences in the whole body response to sinusoidal vibration of increasing and decreasing frequency indicate a non-linearity in the body's response, the degree and nature of the non-linearity depending on the individual subject. The most likely parameters to influence the linearity of the whole-body response in this way are differences in the "active" response of the body, trying to minim ise the effects of the input vibration on the principal body organs. The extent to which the body can actively influence its response to the input vibration could well be governed by such things as the subjects previous experiences of similar vibration environments, and his level of anxiety about the vibration and the experiment.

## B - 10.7 Comparison with Previous Investigations

The direct comparison of these results with those obtained in the past is difficult, but if the differences in the postures adopted, the subject panels, and the measuring techniques used are taken into account, comparisons can be made with previous researchers results.

The results of Coermann, and Suggs et al are tabulated in Tables No. 123 , No. 124 and No. 125. The modulus of impedance and phase angle plots obtained by these researchers are shown in Section B - 2.

The table of results for the sitting erect posture shows that the frequencies of the principal resonances and anti-resonances obtained at Loughborough University are the same as those found by Coermann; the values of the impedance at these frequencies are however different. The difference in these recorded values could be due to the difference in the posture adopted (Coermann's subjects did not have their legs supported), and because Coermann's subjects were on average heavier and older than those used here.

When comparing the results for the sitting relaxed posture the principal frequencies found by Coermann are higher than those measured in this study. Coermann's results are however only based on one subject (R.C.) Coermann does show results (modulus of impedance only) for another subject (W.B.) obtained at two different times, and considerable difference exists between these two plots (shown in Fig. No. 30 one curve shows a primary anti-resonance at about 5.5. Hz while the other shows an anti-resonance peak at about 3 Hz.)

The results obtained during this study for the sitting relaxed posture show principal frequencies lower than those obtained by Coermann, and lower than those tabulated for the sitting erect posture. This difference in the principal frequencies is to be expected as the tenser muscles of the subject in the sitting posture would lead to higher resonant and anti-resonant frequencies, and measured values of Impedance.

The results for the standing erect posture also show that the principal frequencies obtained by Coermann are higher than those obtained in these experiments. The results also show differences in the magnitudes of the anti-resonance peaks and resonance valleys. The results tabulated are for one subject and due to the large variation that exists between subjects even for a carefully defined posture, the results should therefore be compared cautiously.

## B - 11 Conclusions

### B - 11.1 Conclusions

1) The impedance of the whole body was measured using fourteen male subjects and two female subjects in various seated and standing postures. The following resonances were established for the various postures.

14 Male Subjects:-

| Posture | Primary Anti-Resonance | Primary Resonance | Secondary Anti-resonance |
|---|---|---|---|
| Sitting Erect | 5.1. Hz<br>73.2. kg | 8.5 Hz<br>38 kg | 11.0 Hz<br>31 kg |
| Sitting Relaxed | 4.2. Hz<br>75.0. kg | 7.0 Hz<br>36.9 kg | 9.0 Hz<br>30.2 kg |

1 Male Subject of Average Build:-

| Posture | Primary Anti-Resonance | Primary Resonance | Secondary Anti-Resonance |
|---|---|---|---|
| Standing Erect | 5 Hz<br>84 kg | 6.9 Hz<br>46.2 kg | 8 Hz<br>40 kg |
| Standing Relaxed | 5.8 Hz<br>86 kg | Not visible on either the Za or Zv plots | |

2 Female Subjects:-

| Posture | Primary Anti-resonance | Primary Resonance | Secondary Anti-Resonance |
|---|---|---|---|
| Sitting Erect | 5.0 Hz<br>61 kg | 8.1 Hz<br>28 kg | 10 Hz<br>21.3 kg |
| Sitting Relaxed | 4.5 Hz<br>62 kg | 6 Hz<br>40.5 kg | 7.5 Hz<br>30.8 kg |

(The values of impedance given are for acceleration based impedance Za).

2) The values shown above are similar to those given in the literature,

if . the differences in the postures adopted, and the subject panels used are taken into account. The results for the sitting erect posture showing the best correspondence to those of previous investigators.

The histograms detailed in Section B - 9 indicate that up to 8 Hz the subjects weight is a parameter which has considerable influence on the modulus of impedance curves above 8 Hz, however, the subject's weight appears to have little influence on the impedance curves obtained, and other parameters control the subject's whole body response.

4) The impedance measurement undertaken with female subjects show that a significant difference could exist in the whole-body mechanical impedance between the two sexes.

As only two female subjects were used, there was insufficient data for a statistical analysis of significance to be carried out on the experimental results obtained for the two sexes.

5) The whole body impedance curves of some subjects showed that considerable differences existed in the body's response to an imput vibration increasing in frequency than when the input vibration was decreasing in frequency. These differences were exhibited mostly in the phase angle plots, but significant differences (differences outside the accuracy with which the measurements were made) were also noted in some modulus of impedance plots.
These differences in response could be attributable to a hysteresis effect, such that more or less energy is absorbed into the body

depending on the precise characteristics of the input vibration; or that impedance of the body varies with exposure time of the subject to the vibration.

6) The measurements undertaken during this study were made in a period of about ten minutes for each posture, compared to durations of several hours used by previous investigators. This probably meant that the results obtained were less susceptible to postural changes than those obtained in the past.

7) Previous research has shown that the body cannot be regarded as a "passive" system, although the research reported in this thesis does not directly indicate that the body is an "active" system with regard to modifying its response to steady-state sinusoidal vibration. The differences exhibited in the body's response to increasing and decreasing input vibration frequencies, which are most prominent in the phase angle plots, are often reflected in the modulus of impedance plots, indicating that parameters (of damping and elasticity) are changed by the body depending on the nature, and length of exposure to the vibration environment.

## B - 11.2 Suggestions for Future Work

1) The measurements were made on a young subject panel of fourteen male and two female subjects, and the measurements suggested that a significant difference existed in the impedance of male and female subjects. This research suggests that further investigations need to be carried out with female subjects.

2) Modern computing techniques enable direct modulus of impedance and phase angle plots from sinusoidal or random vibration so that different sweep times could be used to see what influence the sweep time has on the linearity of the body's response.

3) The results of these experiments show that the physique of the subjects has a considerable influence on the impedance results. To assess more accurately the effect of physique on the results, more detailed measurements of the subject's physique need to be made prior to the experimental runs.

4) As impedance measurements very considerably between subjects, a research programme using a test panel and measuring their impedance at various times during the day would show if the impedance of the whole body changed significantly with the time of day at which the measurements were taken. The programme could also be extended to cover longer periods of time - i.e. a week, month. This data could then be compared with available data on biorhythms.

5) Impedance measurements were taken with the subject undertaking a tracking task to simulate a stress situation, with improved impedance measuring techniques. Measurements of impedance

could be made in driving simulators, or in vehicles themselves. Thus impedance measurements would be made under true stress situations and impedance measurements may provide an indication as to the level of stress being experienced by the operator.

6) These experiments and those of Krause and Lange (1964) indicate that the body cannot be regarded as a "passive"system. If the body is an "active" system with regard to input vibrations, the degree of "activity" of the body would change with the fatigue level of the subject.

Thus if a series of impedance measurements were made on subjects at various fatigue levels, and significant differences were found to exist, an assessment as to the degree of "activity" of the body with respect to input vibrations could be made. The results of experiments of this type could be compared with the proposed fatigue decreased proficiency limit of the I.S.O and equal annoyance contours of Ashley (1971).