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The Transmission of Vibration by the Human Body with Special Reference to the Problems of Measurement and Analysis.

by

GEOFFREY FRANCIS ROWLANDS

A Doctoral Thesis
Submitted in partial fulfilment of the requirements for the award of
Doctor of Philosophy of the Loughborough University of Technology
December 1974

Department of Human Sciences

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The research described in this Thesis was initiated by the author, and the responsibility for the work submitted lies solely with the author.

In order to complete parts of the research programme, the author was sometimes assisted by his colleagues at the Royal Aircraft Establishment, Farnborough.
ABSTRACT

Modern forms of transportation can impose high levels of vibration upon their occupants. If one is concerned with the welfare and comfort of the passengers, or the ability of the pilots or drivers to perform their respective tasks in such an environment, then clearly one needs to be able to define the vibration levels experienced by such people. This definition must not only quantify the input levels but also include the levels measured at that part of the human body which will be most affected by the vibration.

This Thesis therefore covers work aimed at determining the frequency response of the human body under vibration conditions and presents amplitude ratio and phase angle plots of head, and shoulder, accelerations/seat acceleration against frequency for various postures and limb positions. Attempts are made to model these response curves so that by simply monitoring floor vibration in vehicles one can predict the range of vibrations present at the head and shoulder. The importance of measuring acceleration at these positions is stressed as these are assumed to be the areas which hold the function most likely to be affected by vibration, i.e. the visual and manual loop.

Emphasis is placed on the problems of presenting subjects with the required vibration in the laboratory and of calibrating the accelerometers used to measure the vibration. One Chapter is devoted to the drawing up of a specification, procuring and commissioning of a new two-axis vibration facility which is used in the final experiment to determine the required frequency response curves; another Chapter details a new method of dynamically calibrating the accelerometers in the range used for the experiment.

Summing up, frequency response curves are given for the shoulder and head responses for various posture and limb positions. This basic knowledge will enable designers to ensure there is no conflict between the vibration produced in the vehicle and the human frequency response, thus resulting in a possible degradation in performance or comfort.
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INTRODUCTION

Vibration is a phenomenon which accompanies man from the cradle to the grave. Vibration may be defined as any form of energy input into a system which results in an oscillation or repetitive type motion of any part of that system. It is generally defined by its frequency and its amplitude. The frequency is the number of complete cycles of motion per unit time and the amplitude is the actual displacement of the system from its datum position and will obviously vary, in a sinusoidal manner throughout the cycle. The amplitude usually quoted is the maximum value in the cycle - the cycle being of displacement velocity or acceleration. From early childhood man is conditioned to vibration from the rocking of a pram or in more primitive times from being carried on its mother's back. The ultimate experience of vibration has been quoted as the vibration experienced in the hearse!

During the course of one's daily life, vibration is met during such basic functions as walking and running, as well as from most forms of transport such as buses, trains and cars. In fact, one is so acclimatized to the vibration self-induced during walking that subjects adjudge frequencies around 2 Hz (the so called cradle rocking/walking frequencies) to be pleasant if met in a vehicle. In general the levels of vibration met in transport are simply background levels and at worst interfere with reading or drinking a cup of tea.

Man however has a habit of inflicting upon himself levels of stress, including vibration, which if met outside the confines of his hobby, he would deem intolerable in any form of transport. Vibration levels measured in speedboats, or when driving an old car, or a motor cycle over rough ground, would elicit howls of protest if transferred to a local bus or railway carriage, yet are cheerfully tolerated in the name of sport. Thus clearly if one wishes to quantify the effects of vibration then there exists a psychological problem concerning the degree of tolerance. Again, this time
restricting ourselves to "non-hobby" forms of travel, passengers are readily prepared to tolerate high degrees of vibration if some other saving is involved. If a journey can be done in a third of the time but under three times the vibration environment then many travellers are prepared to accept the penalty.

In the scientific field vibration is normally present as an undesirable part of the environment. Clearly performance can be degraded by the vibration sometimes to such a degree that the environment has to be modified to reduce the vibration - by reducing speed perhaps - or the vehicle has to be redesigned. In aviation, vibration can be induced from operation off rough runways, although as take-off speeds increase, what would appear to be a smooth runway can produce high levels of vibration in the aircraft. As an example defining the orders of magnitude involved, a take-off speed of 200 mph will induce an input to the wheels at 3 Hz for a wavelength of 100 ft, a factor which can easily arise in the natural geography. Also at this frequency a displacement amplitude of only ±0.50 in is equivalent to an acceleration amplitude of ±0.50 g (where g is the earth's gravitational constant). Low altitude high speed flight can impose a high vibration on the pilot, as can helicopter flight where vibrations due to imbalance of the motors can be felt in the cockpit, although normally at a higher frequency than for fixed wing aircraft.

So far all of our examples have centred around a comfort reaction, but clearly one could equally well discuss the effects of vibration in terms of performance - or, as is usual, a loss in performance, this loss normally being associated with vision or limb movements (see section 2). Over the frequency range normally met in vehicles, the human body behaves in a dynamic manner i.e. it possesses mass (or inertia), elasticity and damping, three essential properties for a system to respond to forcing input.
vibrations, and thus exhibits phenomena such as resonance, transmission amplification, or less, at different frequencies etc. (for a definition of these terms see Appendix A). If one therefore wished to quantify the effects of vibration then, as the body behaves dynamically, specification of the input vibration conditions - on the floor of the vehicle perhaps - is insufficient to specify the levels and relative displacements present at the positions where the vibration is felt - the head and limbs. Thus this Thesis sets out to define the dynamic characteristics of the body, in terms of frequency response of transmission functions, or in terms of the mass, elasticity and damping factors which make up such a frequency response. Variations in response curves are given due to changes in posture, limb positions and acceleration levels, so that designers of vehicles will be able to ensure that there is no conflict between the vibration produced in a vehicle and the human frequency response.
The effect of vibration on the human body is basically threefold. Firstly, the vibration can affect the visual sense resulting in a loss of visual acuity, and consequent degradation of performance. Secondly, the vibration can induce involuntary hand/arm motions which may then result in movement of various controls - steering wheels, control columns, rudder pedals etc. - and again degradation of performance. The third factor is that the vibration may affect the subjects psychological state in terms of lack (or possibly gain) in motivation or arousal, or some other motor or conscious response.

In order to be able to define the effects of vibration on the human organism, it is necessary to know what accelerations are present at specific parts of the body, these parts being directly related to the threefold effect listed above. Measurement of factors such as body impedance or mobility (for a definition of terms used such as resonance and impedance see the Appendix of this Chapter) are inadequate in attempting to define the effects of vibration, as they merely give a measure of the 'lumped' characteristics of a system 'loading' the source of vibration. Unless the circuit diagram of the body is known, then impedance measures cannot be extrapolated to quantify the actual vibrations at specific parts of the body. This is demonstrated in the Appendix, at the end of this Chapter, where the impedance of a two degree of freedom system is computed. The second system has a much smaller mass than the base system and is more lightly damped. The major points to be made here are that the important system as far as the body is concerned is the second system, and the effects of this system are overwhelmed in the impedance values by the large, less important system. This is shown in Figs.3 and 4, where Fig.3 shows the overall impedance and Fig.4 shows the impedance calculated without the upper system. The only difference between the curves is at the low frequency end, where the addition of the second system lowers the peak frequency. This is explainable in terms of an increase in the overall mass, but clearly does not help in any analysis aimed at
detecting effects on performance at a frequency twice the peak frequency.

Fig.5 shows the overall system frequency response - between the top mass and the base - and the presence of the second system is evident. The other advantage of transmission measurements is that the system under test may be broken down, i.e. measures may be made between shoulder/seat and head/shoulder for example.

As an example, Figs.6 and 7 show the individual responses of the two systems used in the previous figures.

Also, as will be seen later, if it is found that the man is not a simple series network as described in Appendix B, but rather a combination of parallel and series sprung-mass systems, then the impedance analysis and the frequency response analysis will both be unrepresentative of the total system. However again, if one is interested in the actual accelerations transmitted to and present at a particular part of the body, then the fact that the parallel loop prevents any exact simulation being conducted does not really matter, as the required absolute quantities have been measured.

Fig.8 schematically demonstrates a situation where a transmission response measure can be made, but exact simulation cannot easily be performed as the unmeasured parallel network (whose output may not be known or in fact measurable) must in some way effect the answers. An impedance measure will of course include the complete system. So here is a case where transmission response results and impedance results are completely different.

Clearly then, if one wants to quantify the effects of vibration then it is necessary to know what the vibration levels are on the head, shoulders, legs etc. Having decided on the positions of measurement, it must be realised that various changes in body parameters for each subject might affect the frequency response curves. Parameters such as hand/arm position, leg position and posture should be included as major variables in any investigation. As will be seen later these parameters have been notable by their absence in past reports. Potemkin sums this up in saying "it should be noted that hitherto (1971) there
have been no papers on the investigation of the effect of vibration on the
dynamic reactions of man dependent upon the change in his working posture".

This thesis therefore covers work conducted at the RAE over a number of
years dealing with the measurement of vibration transmissibility to various parts
of the human body. RAE first became interested in vibration and its effect on
humans when it was predicted that the acceleration levels present in a proposed
supersonic aircraft design would be a great deal higher than those measured in
aircraft then in service. These predictions also included the presence of a
high amount of lateral or sway vibration, a phenomenon which had been largely
absent from most previous forms of transport. A number of experiments were
conducted aimed mainly at the ability of pilots to perform their requisite tasks
under the vibration conditions. The findings of these experiments included the
proposition that pilots may have possible visual acuity problems during the
rotation phase of the take off. As a result, following the main body of these
experiments, the author conducted an experiment aimed at defining the transmiss-
ibility of vibration across the body for vertical and lateral inputs. Part of
this trial, reported in Chapter 2 was an attempt to quantify the non-linear
behaviour of the body. Following these trials interest in the vibration field
grew as it was appreciated that many forms of modern transport exhibit vibration
environments which are undesirably high, e.g. hovercraft, helicopters, military
aircraft at high speed and low level transport and passenger aircraft encountering
turbulence or during take off, some trains and cars etc. In all cases examined
the need to know what was happening at specific body parts became more and more
evident. It was also realised that in order to test for the required levels and
frequencies (and at a lower harmonic distortion content) a better vibration
facility was necessary. Chapter 3 therefore presents a short history of the
acquiring and commissioning of such a rig, giving vibration in two axes at levels
compatible with those found in practice, with very small amounts of harmonic
distortion. The inclusion of such a programme in this thesis is valid in that
without such a rig, further experimentation would have been impracticable.
During the installation and commissioning of this new rig, two subsidiary experiments were conducted on another rig, of direct interest to the thesis. The first was aimed at defining the transmissibility of vertical vibration to the shoulder of the seated man and is reported in Chapter 5. The second experiment dealt with the dynamic calibration of the accelerometers used to measure the vibration for all the experiments detailed in this work. Chapter 7 then gives the final definitive experiment in this series, using the new vibrator and driven by a swept sine wave oscillator, instead of the more usual sinusoidal input. This experiment gives the frequency response curves between the input and the shoulder, and the input and the head, for sitting man, for the vertical axes only. Variations in the response curves are shown for four possible postures, three leg positions and three arm positions. The duration of the input swept sine pulse was only about 25 s. Whilst providing a more realistic input to the subject (compared with the more usual hours when sinusoids were used), it was also possible for the subject to maintain the required postures and limb positions throughout the duration of the tests (again compared to the sinusoidal tests). Three input levels were used ranging approximately (as far as general subjective reaction was concerned) from "definitely aware of the vibration" to "barely aware of the vibration". For these final tests, the subjects sat on a rigid hard seat, thus ensuring proper control of the input vibration and repeatability. Clearly the presence of any cushioning material between the subject and the input will affect the amount of vibration reaching the subject. So any investigation which attempts to quantify the manner in which vibration present on the floor of a vehicle is modified to arrive at specific parts of the body, must include discussion of the vibration transmission characteristics of seats and cushions. Chapter 6 therefore reports some ad hoc work done by the author on the frequency responses of seats and cushions.

Initially it was hoped to include the standing man, and the lateral and fore-and-aft axes. However the results from the experiments dealing with the
seated man subjected to vertical vibration, proved so interesting and the results so definitive that it was decided to limit the experiments. It is stressed however in the final conclusions that future work should include these parameters so that a complete picture may be constructed.

Thus summing up, this thesis details work aimed at defining the transmission of vibration across the human body for various body parameter variations, and gives special reference to the problems of measurement, simulation and analysis of vibrational data.
3 LITERATURE SURVEY

3.1 General

Section 2 gives a statement of the basic requirement that this Thesis sets out to satisfy - the definition of the vibration transmission characteristics of the human body - followed by a list of some of the experiments performed to provide the necessary basic data. Before the experimental programme commenced the past literature was examined to ensure that the required knowledge was not already available. This section gives an account of this literature survey. In order to facilitate the survey the following abstract sources were used:

(a) Human Response to Vibration, A Critical Survey of Published Work, 1970. ISVR Memorandum No. 373, published by the Human Factors Unit, Institute of Sound and Vibration Research, University of Southampton. This survey is continually being updated and additional memoranda published when necessary. As well as providing an abstract service, all the listed reports are held at the University for inspection and/or photocopying.

The catalogue contains a 'Topic' index together with a breakdown of the appropriate reports under the headings

- Essential
- Recommended
- Relevant.

The topic used was,

Biodynamics (Body resonance; mechanical impedance)

(b) Aerospace Medicine and Biology - a continuing bibliography published monthly by the National Aeronautics and Space Administration.

A selection of annotated references to unclassified reports and journal articles that were introduced into the NASA Scientific and Technical information system and announced in

- Scientific and Technical Aerospace Reports (STAR)
- International Aerospace Abstracts (IAA)
In its subject coverage, 'Aerospace Medicine and Biology' (AMBB) concentrates on the biological, physiological, psychological and environmental effects to which man is subjected during and following simulated or actual flight in the earth's atmosphere or in interplanetary space. Such related topics as sanitary problems, pharmacology, toxicology, safety and survival, life support systems, exobiology and personnel factors receive appropriate attention. In general, emphasis is placed on applied research, but references to fundamental studies and theoretical principles related to experimental development also qualify for inclusion.

Initially both the STAR and the IAA abstracts were examined for relevant papers, but when it was realized that the Aerospace Medicine and Biology Bibliography (AMBB) was effectively extracting any such relevant papers from the other abstracts, work was continued using the AMBB only. The first issue of the AMBB was published in July 1964, for any papers prior to this date the STAR and IAA were used.

AMBB also publish an alphabetical Thesaurus listing the headings used to sub-divide the reports. The headings used were

- Biodynamics
- Bioengineering
- Cushions
- Dynamic Models
- Human Factors Engineering
- Mechanical Impedance
- Vibration Effects
- Vibration Isolators
- Vibration Perception.

It is interesting to note that, after a while, when most of the relevant papers had been collected and the references contained in these papers examined, the situation arose that the numbers of papers converged, i.e. of
the relevant fifty papers say, any one paper will contain as its own references only reports which are in our original list of fifty. This situation served to show that the survey was complete and no vital papers had been overlooked.

3.2 Required report

Having discussed the requirements of the Thesis, the literature was searched (as per section 3.1) in order to find the following hypothetical report or reports. A report defining the transmission characteristics of the human body covering (a) vertical and lateral excitation, (b) frequency range approximately 1 Hz to 30 Hz, covering the main whole-body resonances and the frequency range where the eyes are reported to be affected, (c) measured parameters to cover head/seat and/or shoulder/seat for manual loop and visual loop responses, (d) effect of various input levels ranging from positively aware to barely aware of the vibration and (e) effect of variations in subjective posture, arm position and leg position. Clearly it was not anticipated that one report would be found covering all the above topics.

Before going into the details of the reports found some general comments are in order. Almost all the reports found covered the vertical axis only, and used sinusoidal inputs. Some of the vibrators used were electrodynamic which are limited in displacement, thereby only being able to give low vibration levels at the lower frequencies (below 5 Hz). As mentioned above most papers used sinusoidal motion and little reference is made to the possibility let alone the measurement of the harmonic distortion. Presumably, because of the difficulty in attaching accelerometers to the subjects, a lot of the reports found measured the input impedance characteristics of the body, whereas we require the vibration levels at, and therefore the frequency response to, the head and limbs. Very few papers mentioned subject posture as a major variable and fewer still attempted to vary it, to determine its effect. A lot of the reports conducted their tests, with the subjects sitting on just a base, i.e. no back rest.
3.3 Survey

A list of references used is given at the end of this Chapter. It contains 43 papers and reference to any paper will be made by its number only. Clearly there are many more papers in existence dealing with the biodynamic response of the human body, but it was felt that the papers listed are the most pertinent.

Some of the listed reports (or books) provided an overview of the field of biodynamics and in a sense aided the compiling of this survey (12, 13, 14, 17, 28 and 37). (14 and 37) in particular provided an excellent picture of the field of biodynamic response. Several bibliographies of reports dealing with biodynamics were also found (21 and 43) and when they were taken, in conjunction with the overviews listed above, a general survey could be easily collated.

Several reports on the measurement of impedance of the body have been included, as it is felt that although they do not measure the required quantities, the values of the peak frequencies will be applicable and any trends regarding non-linearity, postures etc., should read across to the transmissibility results. These reports are (5, 6, 7, 10, 25, 40 and 41).

Compared with the measurement of other vibration effects, such as comfort or performance, the measurement of the biodynamic response to vibration has produced more agreement in research results. Various attempts have been made to model such responses, but a definitive model does not yet exist due primarily to the wide range of environmental parameters of importance - such as frequency, level, duration etc. as well as individual parameters such as body size, weight, posture, age, limb positions etc. Unfortunately even though most of the reports found agreed moderately well as regards frequency and amplitude of whole body resonances, little information was found regarding the individual and personal parameters mentioned above. There is general agreement that vertically the body has a whole body resonance around 5 Hz (2, 6, 8, 10, 16, 22, 25, 31, 35, 36 and 42). As far as the lateral axes are concerned there are fewer reports but the agreement here is for a resonance at around 1.5 Hz (7, 24, 42). The transmissibility ratios to the shoulder have
been investigated by reports (4 and 42). The presences of resonant peaks higher than the 4-5Hz one has conflicting evidence (6 versus 4 and 7 etc.). The conflict is probably due to unreported factors such as seating posture etc. Most of the reports listed above used sinusoidal inputs. Two reports (16,22) used real life vibration, one (16) a helicopter and the other (22) a simulated high speed low level environment.

Three papers (6, 11 and 30) were found which attempted to investigate the effects of varying posture. Paper (6) unfortunately did not use a back rest for the tests and his slumped posture was produced by bending the spine considerably with the head falling forward onto the chest - completely unrepresentative of the real life. Papers (11 and 30) measured the transmission of vibration to the head for three postures none of which could be said to apply to real-life performance postures - but at least an attempt had been made to find the effects of the extremes of posture variations. Unfortunately the frequency range chosen was very large (2-200 Hz) and the resulting analysis provided a frequency increment of 6 Hz, providing little low frequency information. It is interesting to note that (6) demonstrates a trend in the response for slumped posture of reducing the peak frequency, due to an increase on the body's elasticity. "Therefore in order to protect the head against vibrations at frequencies below 5 Hz, it is better to assume the erect posture, and at frequencies above 5 Hz to relax the muscles and to bend the spine." Clearly there is a need to prove the above quotation in more representation seats and postures.

Paper (7) covers some results for standing and sitting subjects and show a vertical resonance at the head and shoulder (about 1.15 and 1.70 amplitude ratios respectively) at around 5 Hz. The head results show slight evidence of a higher resonance but only as a point of inflexion in the curve - the amplitude ratio being less than unity. No definitions or variations in posture are given and the input was sinusoidal. Two graphs are included which show the elliptic head motions induced by input linear sinusoids. This clearly
could be of great importance in any experiment which attempts to define the ratio of linear vibration at the head to that at the seat.

Paper (23) is interesting in that although the experimental design was limited in number of subjects and frequencies used, results are quoted for the variation in transmissibility to the head with time. They show that for standing subjects transmissibility at 2 Hz can increase from 68% to 93% and at 5 Hz from 25% to 39% in about 2 minutes. Whilst results for sitting subjects would clearly be different (the variations given above being due to the loss in effectiveness of the legs as vibration absorbers), the possibility of the human frequency response being time dependent must be faced.

Several papers were found dealing with the frequency response characteristics of seats (1, 2, 3, 9, 27, 35 and 39). Few of the authors appreciated that the peaks around 4 Hz were basically a reflection of the body's response and that these curves could be reduced by modifying the cushions. Also little work had been done on the lateral characteristics of seats, especially with respect to the seat structure itself.

It is interesting to note that very few of the papers listed gave any details of the calibration facilities available or used during their experiments.

Of the reports which used single sinusoidal inputs, only papers (4, 18, 19 and 20) appeared to appreciate the possible inaccuracies due to ignoring the presence of harmonic distortion. Only paper (4) actually measured distortion.

Following the literature survey described here, it must be concluded that our mythical required report is not available. Clearly there still exists the need for research into the human frequency response when seated in a representative manner under varying conditions of posture, harness and limb positions. This information is required for all three linear axes and possible non-linearity effects, as the input vibration level varies, must be investigated. In order to be able to use such response curves in the real-life situation they must be used in conjunction with similar response curves measured for
seats actually used. Response curves for such seats are not readily available and thus must be measured. There is also a need for a method of calibration for the accelerometers used, in the amplitude and frequency ranges used in the experiments.
Appendix A

DEFINITION OF TERMS USED IN BIODYNAMIC RESEARCH

In order to be able to define the terms used in this Thesis and in other research papers, let us assume that the human body can be represented by a mechanical system such as that shown schematically in Fig. 1a.

$x$ represents the motion of some part of the body and is regarded as an output quantity. It can be thought of as a displacement $(x)$ or a velocity $(dx/dt$ or $\dot{x}$) or an acceleration $(d^2x/dt^2$ or $\ddot{x}$). Similarly $z$ is our input quantity (or $dz/dt$ or $d^2z/dt^2$).

Frequency response or transmission function is defined as the complex ratio $x/z$, or $\dot{x}/\dot{z}$ or $\ddot{x}/\ddot{z}$. This function, being complex, is defined (for any one frequency) by its amplitude and its phase lag.

The governing equation of the mechanical system is,

$$m\ddot{x} = C(\ddot{z} - \dot{x}) + K(z - x)$$

or using the operator $D = d/dt$,

$$\frac{x}{z} = \frac{(CD + K)}{(mD^2 + CD + K)}$$

$$= \frac{(DC/m + K/m)}{(D^2 + DC/m + K/m)}$$

$\omega_0$ is defined as the undamped natural frequency of the system, $\omega_0$. $C/m$ is defined as twice the ratio of the damping to critical damping, times the undamped natural frequency, and equals $2h\omega_0$. Thus substituting and rationalizing we have,

$$\frac{x}{z} = \frac{\left[1 + D(2h/\omega_0)\right]}{\left[1 + D^2/\omega_0^2 + D(2h/\omega_0)\right]}$$
using the vector notation, letting \( D = j\omega \),

\[
\frac{x}{z} = \left[ \frac{1 + j \cdot 2\alpha}{(1 - \alpha^2)^2 + j \cdot 2\alpha} \right], \quad (\alpha = \omega/\omega_0)
\]  \( \text{(5)} \)

This then is our complex response function, which for any value of \( \alpha \) has an amplitude ratio and a phase lag between the output and the input. The amplitude ratio is given by the modulus of the above equation and the phase angle by the ratio of the imaginary and real parts.

Thus

\[
R = \sqrt{\frac{1 + 4h^2\alpha^2}{(1 - \alpha^2)^2 + 4h^2\alpha^2}}
\]  \( \text{(6)} \)

and

\[
\phi(\text{lag}) = \tan^{-1} \left[ \frac{2\alpha^3}{1 - \alpha^2(1 - 4h^2)} \right].
\]  \( \text{(7)} \)

Sometimes it is more useful to talk in terms of the relative displacement across the system \( (x - z) \) with reference to the input acceleration \( (\ddot{z}) \).

Manipulation of equation (1), yields

\[
m(\ddot{x} - \ddot{z}) + C(\dot{x} - \dot{z}) + K(x - z) = -m\ddot{z}
\]  \( \text{(8)} \)

Proceeding as before, we arrive at

\[
\frac{(x - z)}{\ddot{z}} = \frac{-[\omega_0^2]}{(1 - \alpha^2)^2 + j \cdot 2\alpha}\]

which has modulus and phase lag given by

\[
|R| = \frac{[\omega_0^2]}{\sqrt{(1 - \alpha^2)^2 + 4h^2\alpha^2}}
\]  \( \text{(10)} \)

and

\[
\phi(\text{lag}) = \tan^{-1} \left( \frac{2\alpha}{(1 - \alpha^2)} \right).
\]  \( \text{(11)} \)
In some instances it is not possible to measure the above quantities on the body (x in particular). However there is another way of obtaining a good insight into the dynamic properties of such a complicated system. Similar to the measurement of a complex resistance, such as the so-called 'impedance' of an electrical circuit which consists of inductances, capacities and resistances, we can measure the mechanical impedance of the human body. According to the electrical impedance, which is the complex ratio of the voltage and the current in the circuit (z = volts/current), we can define the mechanical impedance as the ratio of the applied force to the velocity, at that point where the force is transmitted.

\[ z = \frac{P}{\dot{x}} \]

From equation (1), it can be seen that

\[ P = C(\dot{x} - \ddot{x}) + K(\dot{x} - x) \]

\[ = m\ddot{x} \]  \hspace{1cm} (12)

Thus

\[ z = \frac{m\ddot{x}}{\dot{x}} = m\omega \left( \frac{\ddot{x}}{\dot{x}} \right) \]  \hspace{1cm} (13)

\[ = \frac{m\omega(1 + 2\alpha)}{[1 - \alpha^2] + j \cdot 2\alpha} \]  \hspace{1cm} (14)

Thus for this simple circuit, the impedance \((m\omega)\) (the transfer function).

It is also possible to define a general quantity called acceleration driving point impedance (as opposed to the above, more, normal, velocity driving point impedance), where the quantity is equal to the system mass times the ratio of the motion of the centre of mass and the input motion. An advantage is that most of the measurements taken to define input motion are basically acceleration, so that any computation previously needed to derive the velocity (to define the normal impedance quantity), i.e. integration, which effectively destroys high frequency information, will be eliminated.
Equation (6) defines the transfer function of our system and it can be shown that such an equation has a peak value (resonant value) occurring at a frequency (resonant frequency) near the undamped natural frequency \( \omega_0 \).

It can be shown that, for the system shown, the resonant value is given by

\[
R_{\text{max}} = \sqrt{\left[\frac{8h^4}{8h^4 - 4h^2 + (\sqrt{1 + 8h^2} - 1)}\right]}
\]  

(15)

and the frequency at which this occurs is given by

\[
\alpha_{\text{max}} = \sqrt{\left[\frac{\sqrt{1 + 8h^2} - 1}{4h^2}\right]}
\]  

(16)

which tends to one as \( h \) tends to zero.

Re-arranging equation (15) yields

\[
h = \frac{1}{2} \left[ \sqrt{\frac{R_{\text{max}}}{R_{\text{max}} - 1}} - \sqrt{\frac{R_{\text{max}}}{R_{\text{max}} - 1}} \right]
\]

(17)

Also eliminating \( h^2 \) between (15) and (16), we have

\[
(\alpha_{\text{max}})^4 = \left[1 - \frac{1}{R_{\text{max}}^2}\right].
\]

(18)

This gives a relationship between the maximum amplitude on a response curve and the frequency at which it occurs - independent of the damping involved.

Comparison between equation (5) and (14) indicates that the impedance function also has a peak modulus at the same frequency.

Thus, in this Appendix, we have defined the terms used in most biodynamic experiments. Most of the experiments described in this Thesis provide frequency response curves (giving amplitude ratios and phase lags between input and output motion) rather than impedance measures, for reasons given in Appendix B.
Appendix B

SIMULATION OF FREQUENCY RESPONSE AND IMPEDANCE OF A MULTIPLE SYSTEM

Fig. 1b shows a two-mass mechanical system, whose governing equations are

\[ m_2 \ddot{x}_2 = C_2 (\dot{x}_1 - \dot{x}_2) + K_2 (x_1 - x_2) \]  
\[ m_1 \ddot{x}_1 = C_1 (\dot{z} - \dot{x}_1) + K_1 (z - x_1) + C_2 (\dot{x}_1 - \dot{x}_2) + K_2 (x_1 - x_2) \]  

re-arranging we have

\[ D^2(x_2 - x_1) = -\left[ \frac{C_2}{m_2} D(x_2 - x_1) + \frac{K_2}{m_2} (x_2 - x_1) \right] - D^2 x_1 \]  
\[ D^2(x_1 - z) = -\left[ \frac{C_1}{m_1} D(x_1 - z) + \frac{K_1}{m_1} (x_1 - z) \right] + \frac{m_2}{m_1} \left[ \frac{C_2}{m_2} D(x_2 - x_1) + \frac{K_2}{m_2} (x_2 - x) \right] - D^2 z \]  

Let the upper system have a damping factor of 0.10 and a resonant frequency of 20 rad/s\(^2\) and the lower system a damping factor of 0.30 and a resonant frequency of 10 rad/s\(^2\). Also assume a mass ratio term (\(m_2/m_1\)) of 0.10.

Then the response of our system can be reproduced on an analogue computer using the circuit shown in Fig. 2. Notice that this circuit allows us to monitor all the functions necessary to represent the frequency responses and the impedances.

The results of calculating these parameters from the computer are given in Figs. 3 to 7. Fig. 3 shows the overall impedance (defined as Force/Acceleration rather than the more usual Force/Velocity) and Fig. 4 shows the impedances of the lower system only - the upper system being completely disconnected. Fig. 5 shows the overall frequency response function and Figs. 6 and 7 shows the responses of
the two systems when treated in isolation. It should be noted that simple
addition of Figs. 6 and 7 does not give Fig. 5, as examination of equation (4)
above and Fig. 2 shows that there is a coupling term, qualified by the mass
to term \( \frac{m_2}{m_1} \) which should be included.

As far as the body is concerned it may well be that the upper system is
by far the most important. This may represent the head where the vibration may
produce discomfort, or it may be an internal organ such as the heart or lungs,
where large movements may occur resulting again in discomfort and a probable
loss in performance if a complex task has to be performed. As far as impedance
measures are concerned, for the general case considered (which is reasonably
realistic as far as relative resonant frequencies and damping coefficients are
concerned for the body parts mentioned above) comparison of Figs. 3 and 4 gives
little indication of significant events at 20 rad/s\(^2\). In fact the only
difference between the curves is an increase in the frequency of maximum
impedance from 1.38 Hz to 1.48 Hz and a reduction in the maximum impedance
values from 2.34 to 2.00. The increase in the frequency is explainable in that
the overall mass has been reduced and also the peak value will change as we have
removed a relatively lightly damped system. However at 20 rad/s\(^2\) (3.2 Hz), the
impedance values are very similar. Inspection of Fig. 5 however reveals that
there is a peak in the frequency response curve due to the presence of the upper
system, indicating that if vibration is present at about 3.2 Hz, then performance
may suffer as amplification of vibration will occur. Having therefore 'discovered'
that 3.2 Hz is a critical frequency, using frequency response techniques it is
now possible to attempt to find where this amplification is occurring. Figs. 6
and 7 show a breakdown of the complete system into individual sub-systems (but
still coupled together and governed by equations (3) and (4)) and indicate that
the upper system has a lightly damped resonance at 3.1 Hz.

Thus frequency response techniques have isolated a possible problem area
which impedance measures cannot yield.
This foregoing analysis assumes that the body behaves as a simple series of interconnected sub-systems. Fig.8 shows a possible series/parallel model of the body where \( m_2 \) is at some measurable position and \( m_3 \) may be completely unknown. Frequency response curves covering \((x/z)\) will clearly include the effects of the unknown branch, but any attempt at analysis of \((x/z)\) will be exceedingly different. Measurement of input impedance will however include all the sub-systems - so here we have a special case where the impedance and the selected frequency response functions are completely different. However the point still remains, the body probably senses vibration, in terms of discomfort or loss in performance, through a relatively small mass, i.e. the head, or an internal organ, and the measurement of input impedance, compared with a frequency or transmission response, may not show the presence of such a sub-system.
<table>
<thead>
<tr>
<th>No.</th>
<th>Author</th>
<th>Title, etc.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>C. Ashley, B.K.N. Rao</td>
<td>Equal sensation study of differential vibration between seat and feet. Birmingham University</td>
</tr>
<tr>
<td>2</td>
<td>B. Caiger</td>
<td>Some effects of vertical vibration on a pilot's head motion and his instrument reading ability. NRC, Ottawa (1966) Aero Rep. LR-463</td>
</tr>
<tr>
<td>3</td>
<td>P.C. Calcatessa, D.W. Schubert</td>
<td>Active isolation of human subjects from severe dynamic environments. ASME 69-VIBR-65</td>
</tr>
<tr>
<td>4</td>
<td>R.E. Chaney</td>
<td>Whole body vibration of standing subjects. Mil. Aero Division, Boeing D3-6779 (1965)</td>
</tr>
<tr>
<td>11</td>
<td>K.V. Frolov</td>
<td>Dependence on position of the dynamic characteristics of a human operator subjected to vibration. ASME Conf. Proc. 'Dynamic Response of Biomechanical Systems' (1972)</td>
</tr>
<tr>
<td>No.</td>
<td>Author</td>
<td>Title, etc.</td>
</tr>
<tr>
<td>-----</td>
<td>-------------------------------</td>
<td>-----------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>12</td>
<td>H. von Gienke</td>
<td>Dynamic characteristics of the human body. AMRL communication</td>
</tr>
<tr>
<td>13</td>
<td>D.E. Goldman</td>
<td>Biological effects of vibration.</td>
</tr>
<tr>
<td>14</td>
<td>J.A. Gillies (Ed.)</td>
<td>Textbook of Aviation Physiology. Pergamon Press</td>
</tr>
<tr>
<td>15</td>
<td>M. Griffin</td>
<td>Whole body vibration and human vision. ISVR Ph.D Thesis</td>
</tr>
<tr>
<td>16</td>
<td>M. Griffin</td>
<td>Transmission of tri-axial vibration to pilots in the Scout helicopter. ISVR Report 58</td>
</tr>
<tr>
<td>17</td>
<td>J.C. Guignard, P.F. King</td>
<td>Aeromedical aspects of vibration and noise. AGARDograph 151</td>
</tr>
<tr>
<td>19</td>
<td>J.C. Guignard</td>
<td>Physiological effects of mechanical vibration, a selected bibliography Part I, body resonance phenomena. RAF/IAM Report 124 (1959)</td>
</tr>
<tr>
<td>No.</td>
<td>Author</td>
<td>Title, etc.</td>
</tr>
<tr>
<td>-----</td>
<td>---------------------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>28</td>
<td>J.F. Parker, V.R. West</td>
<td>Bio-astronautics data bank. NASA SP-3006</td>
</tr>
<tr>
<td>29</td>
<td>P.R. Payne</td>
<td>Dynamics of human restraint systems. Nat. Acad. of Scis. NRC/Publ. 977, pp.195-258</td>
</tr>
<tr>
<td>30</td>
<td>B.A. Potemkus, K.V. Frolov</td>
<td>Representation by models of the biomechanical system man-operator under the action of random vibrations. RAE Library Translation 1651</td>
</tr>
<tr>
<td>33</td>
<td>G.F. Rowlands</td>
<td>Transmission of harmonically distorted low frequency vibration to the head of the seated man. RAE Technical Report 72080 (1972)</td>
</tr>
<tr>
<td>34</td>
<td>G.F. Rowlands</td>
<td>Transmission of vertical vibration to the shoulder of the seated man. RAE Technical Memorandum EP 540 (1973)</td>
</tr>
<tr>
<td>35</td>
<td>I. Schmid</td>
<td>The seat as an element connecting the motor vehicle and man. Automob. Zeit, 74, p.4 (1972)</td>
</tr>
<tr>
<td>No.</td>
<td>Author</td>
<td>Title, etc.</td>
</tr>
<tr>
<td>-----</td>
<td>-----------------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>37</td>
<td>C. Shurmer</td>
<td>Review of the effects of low frequency vibration on man and on his tracking performance. BAC (OP) Ltd., Hum. Fact. Study Note, Series 4, No.7</td>
</tr>
<tr>
<td>38</td>
<td>E.L. Stech</td>
<td>Dynamic models of the human body.</td>
</tr>
<tr>
<td></td>
<td>P.R. Payne</td>
<td>AMRL-TDR-WPAPB</td>
</tr>
<tr>
<td></td>
<td>C.F. Abrams</td>
<td></td>
</tr>
<tr>
<td></td>
<td>L.F. Stikeleather</td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>E.B. Weiss</td>
<td>Mechanical impedance as a tool in biomechanics.</td>
</tr>
<tr>
<td></td>
<td>W.S. Clarke</td>
<td>USAF, AMRL-TR-66-84</td>
</tr>
<tr>
<td></td>
<td>H. von Gierke</td>
<td></td>
</tr>
<tr>
<td>41</td>
<td>T.J. Whittman</td>
<td>Human body non-linearity and mechanical impedance analysis.</td>
</tr>
<tr>
<td></td>
<td>N.S. Phillips</td>
<td>J. Biomechanics</td>
</tr>
<tr>
<td>43</td>
<td>R.J. Zember</td>
<td>Bibliography of research reports and publications issued by the Biodynamics and Bionics Division. AMRL covering 1963-1970.</td>
</tr>
</tbody>
</table>
a) Single mass system.

b) Double mass system.

Fig 1: Schematic representation of the human body.
Impedance defined as $(F/z)$ differs from usual in that acceleration used instead of velocity.
FIG 3: Impedance of complete system. (F/Hz)
Fig 4: Impedance of lower system only. (F/2)
Fig 5: Frequency response of complete system ($x_2/z$).
Fig 6: Frequency response of upper system. (x_2/x_1)
Fig 7: Frequency response of lower system, plus coupling effect of upper system. ($x_1/z$)
Fig 8: Possible series/parallel combination.
CHAPTER TWO

A preliminary experiment to determine the transmission of harmonically distorted low frequency vibration to the head of the seated man.
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INTRODUCTION

The work described in this Chapter originated in an attempt to compare and correlate the performance of subject on a tracking task under sinusoidal vibration, with that under random vibration. Initial experiments however indicated that the simulator then available was incapable of producing a pure sinusoidal acceleration input, thereby precluding any real comparison between responses to sinusoidal and random inputs. Inspection of the literature revealed that few research workers have apparently been aware that the acceleration output from a vibrator, although powered from a sine-wave electrical oscillator, usually contains harmonics. It is suspected that such investigators either studied the displacement traces, for which distortion would be considerably lower than for the corresponding acceleration traces, or the traces were so compressed that the distortion was not obvious. A few papers have been found which mention distortion in the output; in one a mechanical filter was constructed to eliminate the relatively high frequency noise induced by bearing rumble. It is doubtful whether this eliminated any lower frequency harmonic distortion present.

In practise, harmonic distortion may amount to as much as 50% (in some cases values over 100% have been calculated) of the fundamental. As an example of how important this can be, consider a fundamental of 0.50g rms at 1 Hz, which has 40% harmonic distortion in the fifth harmonic, i.e. 0.20g rms at 5 Hz. The subject, due to his major body resonance, will probably be more affected by the harmonic than by the fundamental, although all his reactions, transmissibilities etc., may well be recorded as occurring at 1 Hz.

In this investigation it was intended to explore the effects of distortion, by Fourier series analysis of the signals, and to judge whether or not the body acts as a linear mechanical system, under the conditions of test...
A repetitive acceleration wave-form of the type used in this investigation consists of a basic sinusoidal component at the fundamental frequency with superimposed sinusoidal harmonics, and can be represented by a Fourier series.

Thus for any given input and output wave-form, it is possible to calculate acceleration transmissibility ratios, across the body, not only for the fundamental frequency but also for the higher frequency harmonics, even though the latter will generally be considerably less than that of the fundamental. By varying the input amplitude and the fundamental frequency, a number of transmissibility curves can therefore be obtained over a given frequency range. If all the results from the analysis at all frequencies are now plotted together, it should be possible to decide whether or not the body is effectively linear. If all the points fall together to form one overall curve then this will indicate that the body behaves in a linear manner. If however the superimposition results in wide scatter and no clear shape, it seems probable that the mechanical systems involved are non-linear, and nothing can be inferred directly from the transmissibility ratios produced by the harmonics.

An alternative explanation of any large scatter found might be that the body, behaving in some psycho/physiological manner, responds differently to the high frequency overtones of a low frequency fundamental, form the way it would respond to the high frequency alone, that is, the body may have different response to the higher frequencies if there is low frequency present. The present results, however, will enable us at a later date, when a more suitable vibrator is available, to compare the ratios produced at high frequencies - of the order of 10-20 Hz - with and without low frequency harmonics.

The analysis method available at the time was laborious and time-consuming and hence only a fraction of the data available could be analysed.
In future, digital analysis techniques and a vibrator capable of providing higher fundamental frequencies should provide more comprehensive measures of transmissibility and of the degree of linearity.
2 EXPERIMENTAL METHOD

2.1 The vibrator

The tests were carried out at the British Aircraft Corporation's factory at Weybridge on their five-degree-of-freedom Dynamic Manned Vehicle Simulator, shown schematically in Fig.1. The rig was fitted with a full-scale simulation of a fighter cockpit (TSR 2), complete with ejection seat. The seat was a modified Martin-Baker Mk.8 ejection seat, fitted with a three-point harness and head support yoke, as was installed in the TSR 2. A normal complement of instruments, with control stick and rudder pedals, was included, so that the subject sat in a simulated fighter aircraft environment. To maintain correct posture and eye position, the subject sat watching a fixed spot on the head-up display mounted on the simulator coaming. The effect of mental or physical actions on the transmission of the vibration was not explored.

2.2 Acceleration measurement

The miniature accelerometers (6 mm x 8 mm x 14 mm) used were of the gauged cantilever type, model number BLA2 made by the Ether Engineering Co. Ltd. The strain gauges formed one half of a bridge network, the output of which was fed into an SEL ac carrier amplifier, which contained the other half of the bridge together with a balancing potentiometer which was used as a calibration facility (see section 2.4.2). The demodulated output from the amplifier was then fed into a double parallel T-filter (giving a middle cut-out frequency of 180 Hz, corresponding to the resonant frequency of the accelerometer cantilever) and finally into a pen recorder.

Two accelerometers measuring vertical ('heave') and lateral ('sway') vibration (±g\textsubscript{z} and ±g\textsubscript{y}) were rigidly mounted in a duraluminium bar (250 mm x 50 mm x 13 mm), and attached to the ejection seat cushion by a strip of 'Velcro', so that the subject sat directly on it (on his 'ischial tuberosities').
Three similar accelerometers measuring heave (on the top of the head), sway (on the temple) and fore-and-aft, or 'shunt' g, were mounted on the head using a tightly tensioned harness made of 1 inch commercial elastic. Figs. 2 and 3 show the dural bar and head harness. The calibration and setting up of the accelerometers, together with a discussion of the problems of using such a harness to measure head acceleration are discussed in more detail in section 2.4.

2.3 Subjects

Seven men, with ages ranging from 19 to 30 were used for this experiment. The anthropometric statistics of five subjects are given in Table 1, which also included the run number applicable to each subject. Unfortunately, due to an oversight, body statistics for two subjects are not available.

2.4 Test procedure

2.4.1 Calibration

Before any testing or calibration started, the accelerometers, amplifiers and filters were allowed to warm up for at least half an hour. The amplifier gains were then selected to give the appropriate amplitude on the recorder and the outputs balanced for zero voltage output for a 'zero g' configuration. The accelerometers were then positioned to give +1 g and then inverted to give -1 g, with the pen recorder operating. This gave a check calibration on site.

The settings of the balance potentiometer, for zero output voltage, on the face of the amplifiers for conditions of +1 g, 0 g and -1 g were also carefully noted, as these results would be used later to check the correct orientation of the accelerometers on the head and in the seat bar. The dynamic calibration of the accelerometer is covered in Chapter 4.

2.4.2 Fitting of seat bar and harness adjustment

The bar was placed in position on the seat cushion on its strip of 'adhesive' Velcro, and the subject sat on the bar. If the resultant output of the amplifier was not zero for the 1 g or 0 g setting of the balance potentiometer (according to whether the accelerometer was to be used for
vertical or horizontal measurements), this was an indication that the bar was not accurately angled, and its position was adjusted until the output for both accelerometers was zero.

After positioning the bar the subject was given a short ride on the vibrator to familiarise himself with the vibration conditions to which he would be exposed and, for those runs during which harness was used, to allow him to adjust its tightness. All subjects stated that initially they were very much aware of the presence of the bar beneath the ischial tuberosities, but after a while they became fully accustomed to it and felt no discomfort.

After adjustment, the harness was marked with a crayon (a different colour for each subject) so that the tightness was repeatable. Similarly the position of the rudder pedals was marked and checked for each subject.

2.4.3 Fitting of head harness

The head harness was positioned on the head and the straps tightened until the subject stated it was on "very tightly and the experimenter noted that the tension was adequate. By placing a finger under the elastic it was possible to assess the tendency of the accelerometers to rise under vibration. However the harness could not be so tight that the subject felt uncomfortable, as this might well have introduced a comfort artifact, an uncontrollable variable. It was explained to the subject that for meaningful results the harness had to be tight, but not unduly uncomfortable, and the subject was allowed to determine the point at which in his judgement, wearing the harness for a period of approximately 30 minutes might prove too uncomfortable. As soon as possible after the harness had been positioned, and the subject had positioned himself correctly so that he could watch the head-up display, the accelerometers' zero balances were checked. If different form the calibration values, the accelerometers were inched along the harness until the output was zero for the original balance setting. The accelerometers were then known to be measuring true heave, sway and shunt.
It should be noted that the accelerometers were all of the linear type and any rotation of the head would effectively add an angular acceleration term to the output, which is inseparable from the linear acceleration unless a rotational accelerometer is also used and the radius of gyration can be measured. This raises the much larger problem of how many accelerometers are needed to describe the motion of the head, when that motion is composed of both linear and rotational acceleration. This question is not covered in the present investigation.

2.4.4 Tests

Following his initial ride, the subject was instructed to watch the head-up display and to try to maintain this head/eye position throughout the experiment. It was strongly emphasised, however, that on no account should the subject tense his muscles unnaturally in order to maintain this position. It was thus hoped to obtain results for a normal sitting position and to eliminate unusual effects due to slumping or tensing.

The vibration was increased to the required level at 1 Hz fundamental and after approximately 20 seconds of vibration to enable the subject to become accustomed to the motion, a trace of accelerometer outputs was taken. The vibration was stopped, and the procedure repeated at 0.5 Hz intervals up to 4 Hz. This was termed a set of results. Sets of results were taken with each of the seven subjects, at 0.40 g rms and 0.20 g rms for the following conditions:

(a) Heave motion, with harness,
(b) Heave motion, without harness,
(c) Sway motion, with harness,
(d) Sway motion, without harness.

2.5 Analysis

In order to maintain transmissibility ratios of fundamentals and harmonics in the type of wave-forms used in this experiment, it is necessary to determine the Fourier series corresponding to each wave, that is the constants in the formula,
\[ F(t) = A_0 + \sum_{j=1}^{\infty} \left[ A_j \cos(\omega J t) + B_j \sin(\omega J t) \right] \]  

where \( \omega \) is the circular frequency of the fundamental, and equals \( 2\pi f \), where \( f \) is the frequency in Hz, and \( J \) is the number of the harmonic.

The total harmonic distortion in the wave corresponding to equation (2), is the ratio of the square root of the sum of the squares of the individual amplitudes of all the harmonics, to the amplitude of the fundamental.

\[ \text{Distortion} = \left[ \sum_{j=2}^{\infty} C_j^2 \right]^{1/2} / C_1. \]  

The only feasible method of analysing the records was to measure a number of ordinates in a cycle, and perform a numerical calculation. The analysis was done by enlarging the trace using an epidiascope, reading off forty ordinates per cycle, and feeding the results into a digital computer programmed to perform a Fourier analysis.

Due to the length of time needed to analyse the traces using the epidiascope (the estimated time to evaluate the ordinates of only one cycle being 30 minutes and there being approximately 4000 traces) it was decided only to analyse the results from the high g acceleration tests, (0.40 g rms); in any case there was little possibility of using the harmonics from the lower level because there absolute level was so small. It was also decided to confine the analysis to the ratio between like conditions, that is, heave at the head to heave at the seat, and sway at the head to sway at the seat.

The results of a typical analysis of one wave-form are given in Table:

Supposing for the input of the seat we have,
\[ F_s(t) = A_0 + \sum_{j=1}^{20} C_j \sin(\omega_j t + \psi_j) \]

and for the corresponding output at the head we have,

\[ F_h(t) = A'_0 + \sum_{j=1}^{20} C'_j \sin(\omega_j t + \psi'_j) \]

then the amplitude of the transfer function is given by \( \frac{C'_j}{C_j} \), and the phase by \( (\psi'_j - \psi_j) \).

This calculation was performed for the conditions listed in 2.4.4, and the individual results and means are given in Tables 3 to 6. The gaps in the tables occur where the values of \( C_j \) or \( C'_j \) were less than 5% of \( C_1 \) or \( C'_1 \); since it was thought that such results were not significant. The mean values are plotted in Figs. 4 to 7, and some of the individual results plotted in Figs. 8 to 11.

It is worth noting that the same answers would be obtained, in theory, if displacement or velocity were analysed, instead of acceleration, but the amplitudes of the higher harmonics would be reduced proportionately by the number of the harmonic and the square of the number of the harmonic respectively.

If the system under test is non-linear than the values of the transmissibility ratio obtained from the harmonics could be misleading, and care must be taken in any interpretation (see section 4.5 and Appendix A).

Appendices B and C show the theoretical responses of a single-degree-of-freedom system (governed by a second order, linear differential equation), and a two-degrees-of-freedom system, to an acceleration input. Comparison between the results obtained and Appendices B and C will be made later.
3 RESULTS

The mean results of the analysis for the high g tests (0.40 g rms) are shown in Figs. 4 to 7, and all the individual results, including the means are listed in Tables 3 to 6. The results were limited to the sixth harmonic, as higher harmonics generally contained less than 5% of the fundamental amplitude.

Figs. 4 and 5 include results from Guignard for head-to-seat transmissibility in heave, the two lines representing a spread of ±1 standard deviation. Guignard used an acceleration level of 0.18 g rms, and the subjects sat upright on a backless platform, with no harness or back support.

Figs. 6 and 7 include results published by Hornick of the sway transmissibility of a seated, unharnessed, unsupported subject. The upper dotted curve represents the mean results for an input acceleration level of 0.15 g rms, and the lower dotted curve represents the mean results for 0.35 g rms. The difference between these two curves suggests a degree of non-linearity.

Figs. 4 and 5 for the heave axis, and Figs. 6 and 7 for the sway axis, also show curves for the transmissibility ratio produced by the fundamentals only, extrapolated to 3 Hz. This information is needed later to check for linearity of the body's response.

Figs. 8 and 9 are intended to show whether or not the body behaves as a linear system in the heave mode. The transmissibility ratios produced by different input levels for the same frequency, generated by several harmonics are shown.

Figs. 10 and 11 show corresponding graphs for the sway axis.
4 DISCUSSION

4.1 General

The validity of the results is subject to the limitations of the experiment and of the analysis. Only seven subjects were used, four subjects doing each run twice, and three only once, making eleven results in all for each case. The analysis was performed only for like vibration conditions and for one input acceleration level, that is, 0.40 g rms input in heave compared with the resulting heave output at the head, and 0.40 g rms input in sway compared with the resulting sway output at the head. The highest fundamental frequency available from the rig was 4.0 Hz, so it was impossible to cover all the major body resonances with a fundamental excitation. The initial intention was to cover the body resonances below 10 Hz by using the harmonics present in the input waveform, but this can only be justified if the body is a linear system where any frequency in the input be it fundamental or harmonic, can only produce the same frequency in the output and hence the transmissibility ratio for 6 Hz for example may be calculated from the sixth harmonic of 1 Hz, the fourth harmonic of 1.5 Hz or the third harmonic of 2 Hz. If the body behaves in a non-linear fashion, then the transmissibility ratios produced by the harmonics are not acceptable (see Appendix A) and the highest frequency possible is 4 Hz. However the calculations were still considered since the various values obtained for a particular frequency may give an indication of the existence, and possibly of the type, of non-linearity. If non-linearity were shown to exist, the validity of using Fourier transforms of shock inputs for assessing transmissibility and body impedance would be in doubt.

4.2 Transmissibility in heave

The results, plotted in Fig. 4 for the condition of no harness, appear to show a resonance at about 4 Hz giving a transmissibility factor of about 2.0. The resonant frequency, when harnessed, see Fig. 5 is about 6 Hz giving a transmissibility factor of about 1.8. The result for no harness agree
fairly well with Guignard's results. Comparison of the phase angle plots on Figs. 4 and 5 with Fig. 13 suggests that the body behaves as a single-degree-of-freedom system in the heave axis. However, the scatter in the phase angle points means that no firm conclusion can be drawn. Similarly, the theoretical approach laid out in Appendices B and C could not be used with any degree of certainty as the $R_0$ (see Appendix B) point is indeterminate.

Fig. 5 also shows at 5 Hz (the second harmonic of 2.5 Hz) a point well below the other results. Inspection of the actual data in Table 4 shows values varying from 1.09 to 0.09—all below the expected values. It can only be concluded that this is a 'rogue' result and should be discounted.

4.3 Transmissibility in sway

Figs. 6 and 7 show the transmissibility ratio and phase lag between the head and the seat when no harness is worn and when harnessed respectively. The results obtained are generally higher than Hornick's, consistently so when harness is worn. Both experiments are not inconsistent with the existence of a resonance at about 1 Hz, as has been reported by Dieckmann. The phase lag results (showing lags greater than 180°) and to some extent the transmissibility curves appear to indicate at least a two-degree-of-freedom system (Appendix B).

4.4 Effect of harness

With regard to the effect of harness, comparison of Figs. 4 and 5 indicates that under heave vibration, as might be expected, harness slightly reduces the amplitude at resonance and increases the frequency of resonance. Comparison of Figs. 6 and 7 suggests that in the sway mode, harness makes conditions worse as far as head acceleration is concerned, since in general the ratios in Fig. 6 are 15-20% higher than the corresponding points in Fig. 7. Observing the subjects confirmed that head motion was least when no harness was used, and the body could be seen to flex, acting as a vibration absorber. The presence of the harness appeared to reduce this adaptation.
4.5 Linearity

Figs. 8, 9, 10 and 11 show plots of transmissibility ratio against the input acceleration level of each harmonic including the fundamental, for all seven subjects. If the body acts as a linear mechanical system, then any sinusoidal input acceleration level of a given frequency, whether a fundamental or harmonic, should give the same ratio. Inspection of Figs. 8 and 9 indicates that whilst the spread of the ratios produced by the various harmonic contents, in heave, to a large degree overlap, indicating linearity (Figs. 8c and 9c), the ratios produced by the fundamentals both measured and extrapolated are slightly higher than those for the harmonics, (Figs. 8a and b, 9a and b). From Figs. 8 and 9 it would appear that the ratio produced by a fundamental of 0.40 g rms is 10% higher than the ratio produced by a harmonic of 0.03 g rms, that is an increase in ratio of 10% for an increase in input acceleration of nearly 140%. Hence the argument for and against linearity is to some degree conflicting. The conclusion must be that within the range of heave accelerations used, the body approximates to a linear system, but there is some slight evidence of non-linearity which might repay further investigation.

The graphs in Figs. 10 and 11 for sway input, indicate that within the range of acceleration used the body is, to some degree non-linear, in this axis. In Fig. 10a the mean of the ratios produced by 0.02 g rms is about 2.5 times the mean of the ratios produced at the same frequency by an input of 0.40 g rms. Fig. 11a shows a similar plot except that the subject is harnessed and the factor is reduced to 1.90. Figs. 10a and c and 11b and c show plots of harmonics only and at 6 Hz also show a degree of non-linearity, where the comparison is between about 0.20 g rms and 0.03 g rms inputs. Thus it would appear that at the higher input levels the body behaves non-linearly in the sway axis. This is borne out by Hornick's results as shown in Figs. 3 and 4. Theoretically as discussed earlier, for a non-linear system,
we should ignore all the results in the sway axis where the transmissibility ratio is not produced by dividing the fundamental of the output by the fundamental of the input, but as the degree of non-linearity would not appear to be too great (a factor of 2.5 on the ratio for a variation of 16 in the input), the results may be taken to indicate a trend, rather than a definite pattern.

It must be emphasised that the shape of the 'curves' in Figs.8 to 11 apply only for one specific input wave-form at the particular frequency in question. Any other input, varied by a change in fundamental amplitude or higher harmonic amplitude could, and probably would, give a different set of ratios and an entirely different picture - even if the same non-linear equation applied.

4.6 Subject scatter

The results obtained for each subject are detailed in Tables 3-6, and some of these individual results plotted in Figs.8 to 11. The breakdown of the run numbers per subject is given in Table 1.

For heave excitation, the scatter of the results for the fundamental frequencies between subjects and within subjects was surprisingly small, for example from Table 4 at 1.5 Hz the variation in transmissibility ratio between the subjects is 1.21 to 1.32 with a mean of 1.27; also for subject R.L., the figures for two runs were 1.21 and 1.24. These figures are typical for other subjects and frequencies. At the higher frequencies (from the harmonics) the scatter becomes increasingly great, for example, Fig.9c shows a scatter of 3 to 1 on the fourth harmonic of 1.5 Hz. This may be explained on two counts, the low acceleration levels involved (about 0.02 g rms) and the fact that more scatter is to be expected near resonance.

For sway excitation there was no similarly explicable pattern. The results show a large amount of inter- and intra-subject scatter. For example, again taking subject R.L. (see Table 5), for 2.0 Hz his two fundamental amplitude ratio figures are 1.07 and 0.45. The point made above for
heave excitation regarding scatter near a resonance is however supported by the sway results. Comparing Figs. 10b and c, and 11b and c, it is seen that the scatter decreases as the frequency increases away from the resonance. The greater scatter found for the sway axis can probably be explained by the fact that whereas in the heave axis, one has a main body structural member (the spine) which acts in the line of action of the vibration, absorbing or modifying the vibration before it reaches the head, in the sway axis the structure of the spine enables it to flex in many modes, either voluntarily or involuntarily. This theory was supported by the subjects, several of whom noted that in sway the point of inflexion on the spine (the part of the spine they thought the vibration 'pivoted' around as the head and seat tended to vibrate in anti-phase) varied during a test.

4.7 Sources of error

In common with many experiments involving human subjects, the results as shown in Tables 2-5, show an appreciable amount of inter-subjects scatter. Inspection of Figs. 8 to 11, where some of the individual results are plotted, shows that the maximum scatter occurs for the low input g levels which are encountered in the higher harmonics away from the resonant frequency (see section 4.6). Apart from this low input signal explanation, other factors clearly decided the degree of scatter present. The most obvious factor is the variation in body weight, height etc. However, inspection of Table 1 shows that the height and weight variations were very low, the range being ±2 in and ±16 lb. For half the tests the subjects were harnessed and the tightness of the harness could well have resulted in scatter, but inspection of Figs. 8 and 9, and 10 and 11 show little difference between the harness and no harness conditions. Posture is another variable, but apart from instructing the subjects to maintain a normal posture, or attempting to use electromyographical (EMG) techniques, there is no way of controlling it. Also the variations in posture would be expected to be greater without harness, so the above statement that no harness and harness conditions
produce roughly the same scatter could be explained by stating that under no harness conditions the scatter is produced by posture variations and under harness conditions the scatter is produced by variations in harness tightness (inter-subject rather than intra-subject). As detailed in section 2, harness tightness was controlled by marking each subject's preferred position, and posture was controlled by instructing the subject to sit with his back firmly against the back of the seat - but not to achieve this by pushing with his feet against the pedals. As it would be very difficult to produce meaningful EMG traces under vibration conditions it was felt that the above method went some way to establishing at least some uniformity in posture. It is proposed, in future work, to explore the effects of posture and muscle tensing on the transmissibility ratio, by selecting a frequency and amplitude and asking the subject to tense and relax his muscles - whilst maintaining the same posture, and to alter his posture - by slightly leaning forward - not necessarily slumping, whilst maintaining his muscular effort.

The accelerometer system used in this experiment may have introduced errors and scatter in the results for the following reasons.

(i) The linear accelerometers used may have been affected by any rotational acceleration having a component in the plane of action of the accelerometer, that is their outputs may be a function of linear acceleration and radius of rotation times rotational acceleration \((\ddot{x} + r\dot{\theta})\).

(ii) The attachment of the accelerometers to the head could either be too loose, with the result that one is measuring the response of the head-harness rather than the head, or be too tight in which case one is adding another variable, discomfort, to the experiment. It was felt that by far the lesser of the two evils was the discomfort so the harness was tightened onto the subject's head to a degree just short of actual pain.

(iii) The seat accelerometer pack used simply measures the acceleration input at the seat-surface/buttocks interface. The amount of vibration
actually reaching the man's main load-bearing structure - the skeleton - is clearly affected by the amount of tissue between his pelvic structure and the seat cushion. It is hoped that eventually it will be possible to measure the acceleration levels more meaningfully by affixing an accelerometer to the base of the spine or hips. Also the seat of the pants may not be the sole vibration input to the man. When seated in an ejection seat, and more so when harnessed, the subject was receiving a vibration input from the back of the seat, the stick and the rudder pedals, none of which was measured, so the results of this experiment can only apply to the particular seat and harness configuration used.
CONCLUSIONS

The main findings of this Report can be summarised as follows:-

(1) The results indicate that, for the particular conditions and subjects used, the maximum transmissibility ratios and the corresponding resonant frequencies are as shown in Table 7.

Table 7

<table>
<thead>
<tr>
<th></th>
<th>Heave</th>
<th>Sway</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>No harness</td>
<td>Harness</td>
</tr>
<tr>
<td>Freq. (Hz)</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Trans. ratio</td>
<td>2</td>
<td>1.8</td>
</tr>
</tbody>
</table>

(2) In heave, the transmissibility ratios over the frequency range 1 to 10 Hz are about 10% lower when harness is worn than when not. This implies that the particular harness used reduced the heave vibration transmitted to the head.

(3) In sway, the transmissibility ratios are 10 to 20% greater when harness is worn than when not. This implies that the particular harness used increased the transmission of lateral vibration to the head.

(4) The transmissibility ratios obtained in heave agree fairly well with other published results; but those obtained in sway are considerably higher than some previously reported.

(5) Over the range of input accelerations involved, the body would appear to behave approximately as a linear system in heave. Under sway excitation the body appears to be non-linear in a similar manner to that described by Hornick. However, more work is needed at higher acceleration levels at fundamental frequencies between 4 and 10 Hz before the non-linearity can be accurately quantified.
In considering the application of the above conclusions it must be remembered that the input acceleration contained fairly severe harmonic distortion (50% on simple addition), the subjects were sitting in an ejection seat and the g level used was at most 0.40 g rms. Also the accelerometers used were linear translational devices, so that the measured acceleration may have included a proportion due to rotational acceleration.

It is interesting to note that, following the accelerometer calibration procedure (2.4.2), the apparent g level when the accelerometer is in use, is an indication of the angle of the accelerometer to the vertical. For example, if the vertical reading accelerometer is apparently subjected to 0.9 g instead of 1.0 g, this implies a tilt of 6°. Thus the accelerometer bar could be used as a remote reading inclinometer, or if nulling is possible, as a remote reading spirit level.
Appendix A

GENERATION OF HARMONICS IN A NON-LINEAR SYSTEM, SUBJECTED TO SIMPLE HARMONIC INPUTS

Fig.12a shows a simple single-degree-of-freedom system with linear damping and spring stiffness. The analysis of such a system is shown in Appendix B.

If, however, the spring stiffness term is non-linear, as is suspected for the human body, then the analytical solution of the equation is only possible by methods of successive approximation or other methods involving laborious algebra (reference McLachlan, pp.52-56 et seq.), if solution is at all possible.

The standard equation for the system shown in Fig.12a is given by,

\[ m(\ddot{y} - \dot{x}) + c(\dot{y} - \dot{x}) + k(y - x) = -mx \]  \( \text{(A-1)} \)

As shown in Appendix B, this is readily soluble and if \( x \) is a simple harmonic input then \( y \) is also simple harmonic of the same frequency.

The assumed equation for the non-linear spring case is of the form,

\[ m(\ddot{y} - \dot{x}) + c(\dot{y} - \dot{x}) + k(y - x)[1 + \gamma k(y - \dot{x})] = -mx \]  \( \text{(A-2)} \)

where \( \gamma \) is some constant.

That is, one which has a square law spring which hardens up for increasing deflection and softens for decreasing deflection.

In order to demonstrate that such a system can generate harmonics in the output which were not present in the input the following equation was set up on an analogue computer, using circuit shown in Fig.13.

\[ \ddot{y} - \dot{x} + 1.9(\dot{y} - \dot{x}) + 10(y - x)(1 + 81\overline{y-x}) = -\ddot{x} \]  \( \text{(A-3)} \)

where \( x \) is our forcing acceleration term.

Fig.13 also shows the results from the computer study. Three frequencies were chosen representing 2, 1 and 0.5 times the undamped resonant frequency of the linear equivalent equation, i.e. from equation \( (A-2) \) if \( \gamma = 0 \).
Four input levels were chosen to demonstrate that a non-linear system, as well as generating harmonics, will generate varying percentages of the same harmonic for varying simple harmonic inputs.

The results show that the worst case appears to be when the forcing frequency is half the resonant frequency, when over 20% may be generated as a second harmonic. This harmonic could clearly introduce severe errors if one were measuring the frequency response as the overall ratio of input to output.
Appendix B

THEORETICAL RESPONSE OF A SINGLE-DEGREE-OF-FREEDOM SPING/MASS/DAMPER SYSTEM

Referring to Fig. 12a, the force acting on the mass \( m \) when displacements \( x \) and \( y \) occur is

\[
c(x \cdot y) + k(x - y)
\]

This equals the mass acceleration term \( m\ddot{y} \). Therefore

\[
m\ddot{y} = c(x \cdot y) + k(x - y) \quad (B-1)
\]

employing the operator \( D = \frac{d}{dt} \), assuming sinusoidal inputs and letting \( c/m = 2\omega_0 \) and \( k/m = \omega_0^2 \)

\[
\frac{\ddot{y}}{\dot{x}} = \frac{(\omega_0^2 + j2\omega \omega_0)}{(\omega_0^2 + j2\omega \omega_0 + (j\omega)^2)} \quad (B-2)
\]

dividing by \( \omega_0^2 \) and re-arranging

\[
\frac{\ddot{y}}{\dot{x}} = \frac{(1 + j2\omega \alpha)}{[1 - \alpha^2 + j2\omega \alpha]} \]

where \( \alpha = \omega/\omega_0 \)

thus

\[
|\frac{\ddot{y}}{\dot{x}}| = |\frac{\ddot{y}}{\dot{x}}| = |\frac{\ddot{y}}{\dot{x}}| = \text{transmissibility ratio } R = \sqrt{\frac{1 + 4\alpha^2 \alpha^2}{(1 - \alpha^2)^2 + 4\alpha^2 \alpha^2}} \quad (B-3)
\]

\( \Phi = \text{phase lag angle} \)

\[
\Theta = \left[ \tan^{-1} \frac{2\omega \alpha}{(1 - \alpha^2)} - \tan^{-1} 2\omega \alpha \right] \quad (B-4)
\]

\[
\Theta = \tan^{-1} \frac{2\omega \alpha^3}{1 - \alpha^2(1 - 4\alpha^2)} \quad (B-5)
\]
Fig. 14 shows plots of $R$ vs. $\alpha$ and $\phi$ vs. $\alpha$ for various values of $h$.

It is interesting to note that all the $R/\alpha$ plots pass through the point $R = 1$ at the same value of $\alpha$. Inspection of equation (B-3) shows that this value of $\alpha$ is $\sqrt{2}$, thus the frequency is $\sqrt{2f_0}$ where $f_0$ is the undamped natural frequency.

Further, the amplitude of the curve at $f_0$ designated $R_0$ is given by

\[
R_0 = \frac{1 + 4h^2}{4h^2}^{1/2}
\]

or

\[
h = \frac{1}{2} \frac{1}{\sqrt{R_0^2 - 1}} \quad (B-6)
\]

Thus, if one were presented with an experimental curve of transmissibility ratio $R$ vs. frequency, inspection of two factors, the frequency at which the curve crosses the $R = 1$ line and the value of $R$ at the calculated $f_0$, will yield the resonant frequency and the damping ratio of the assumed single-degree-of-freedom system.
Appendix C

THEORETICAL RESPONSE OF A TWO-DEGREE-OF-FREEDOM SPRING/MASS/DAMPER SYSTEM

Referring to Fig. 12b, the equations of motion for the two masses may be written:

\[ m_1 \ddot{y}_1 = K_1 (x - y_1) + c_1 (\dot{x} - \dot{y}_1) + K_2 (y - y_1) + c_2 (\dot{y} - \dot{y}_1) \]  (C-1)

and

\[ m_2 \ddot{y} = K_2 (y - y_1) + c_2 (\dot{y} - \dot{y}) \]  (C-2)

Using the same notations as for Appendix B, with suffices 2 for upper system and 1 for the lower system, we have

\[
\left( \frac{\dot{y}}{\dot{x}} \right) = \frac{(1 + j2\omega_1 \alpha_2)(1 + j2\omega_2 \alpha_2)}{\left[ (1 - \alpha_1^2) + j2\omega_1 \omega_2 \right] \left[ (1 - \alpha_2^2) + j2\omega_2 \omega_2 \right] - \frac{m_2}{m_1} \alpha_1^2 (1 + j2\omega_2 \alpha_2)}
\]  (C-3)

where \( \alpha_1 = \omega_1 / \omega_1 \) and \( \alpha_2 = \omega_2 / \omega_2 \).

If the ratio \( m_2 / m_1 \) is small, then the last term in the denominator may be ignored and the equation becomes

\[
\left( \frac{\dot{y}}{\dot{x}} \right) = \frac{(1 + j2\omega_1 \alpha_2)(1 + j2\omega_2 \alpha_2)}{\left[ (1 - \alpha_1^2) + j2\omega_1 \omega_2 \right] \left[ (1 - \alpha_2^2) + j2\omega_2 \alpha_2 \right]}
\]  (C-4)

i.e. the product of the responses of each system when treated separately, with no coupling effect.

For the human body where \( m_2 \) may be taken as the head and \( m_1 \) the trunk, the ratio \( m_2 / m_1 \) is about 1:10, so that this approximation is probably justified.

If the body is assumed to be a two-degree-of-freedom system then the various damping factors and resonant frequencies involved can be found theoretically by multiplying two response curves, chosen from Fig. 14 to give the final required experimental curve.
Table 1

SUBJECTIVE DATA

<table>
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<tr>
<th>Subject</th>
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<th>Height (in)</th>
<th>Weight (lb)</th>
<th>Age</th>
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<td>I.H.</td>
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<td>E.L.</td>
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<td>R.L.</td>
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<td>C.R.</td>
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<td>Anon 2</td>
<td>11</td>
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Table 2

TYPICAL COMPUTER OUTPUT FOR SEAT ACCELERATION TRACE —
EXCITATION: HEAVE 2.5 Hz, 0.40 g rms, WEARING HARNESS

Case No. 5

<table>
<thead>
<tr>
<th>J</th>
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<th>B_J</th>
<th>C_J</th>
<th>( \psi_J )</th>
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\( J = \) harmonic number

\[ A_J \cos (\omega_J t) + B_J \sin (\omega_J t) = C_J \sin (\omega_J t + \psi_J) \]

\( J = 0 \) corresponds to the dc level of the oscillation and has no physical significance.
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**TRANSMISSIBILITY RATIOS - SHAY - NO HARNESS**
<table>
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<tr>
<th>Fundamental frequency</th>
<th>Run No.</th>
<th>Input (rms g)</th>
<th>Ratio</th>
<th>Phase lag</th>
<th>Input (rms g)</th>
<th>Ratio</th>
<th>Phase lag</th>
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<td></td>
<td></td>
<td>First harmonic</td>
<td></td>
<td></td>
<td>Second harmonic</td>
<td></td>
<td></td>
<td>Third harmonic</td>
<td></td>
<td></td>
<td>Fourth harmonic</td>
<td></td>
<td></td>
<td>Fifth harmonic</td>
<td></td>
<td></td>
<td>Sixth harmonic</td>
<td></td>
<td></td>
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</table>

The table provides data for various frequencies and run numbers, detailing input values, phase lags, and ratios across multiple harmonics.
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<td>7</td>
<td>D. Dieckmann</td>
<td>Intern Z angew Physiol einschl Arbeitsphysiol 16, 519 (1957)</td>
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(Ref Fig. 12a)
CHAPTER THREE

Specification and commissioning of new two-axis vibration rig
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Appendix

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INTRODUCTION

Following the tests detailed in Chapter 2, it became clear that if future work was planned on the measurement of human vibration transmission characteristics (and possibly on the measurement of loss in performance under vibration), then an improved vibration facility was required. The main areas where improvement was needed and would be sought were (i) an increased frequency response to enable tests to be conducted up to about 40 Hz (including the requirement that it be possible to play real-life tape recordings of acceleration into the rig and reproduce on the table the vibration (met in aircraft, cars, hovercraft, etc.) and (ii) a sharp decrease in the amount of harmonic distortion present during sinusoidal testing.

Regarding part (ii), in the past it had been necessary to contend with distortions greater than 50%, resulting in the possibility that any measurements taken, be it transmission or loss in performance, were probably more affected by the harmonics of the fundamental frequency, than by the fundamental itself. With a new rig, with distortions hopefully less than 15% on the acceleration wave-form, up to frequencies around 40 Hz (vertically, laterally or in combination), then accurate experimentation could be performed yielding dependable results.

One major decision was to decide whether to select an electrohydraulic system or an electrodynamic one, the first having possible strokes of up to tens of inches, but suffering perhaps when asked to oscillate at low amplitudes, the second being limited to strokes of no more than a few inches, but capable of oscillating accurately at very small amplitudes.

Measurements taken in aircraft at the time the specification was being drawn up, indicated that problems were being met during take-offs from rough runways and when encountering turbulent conditions. The frequencies measured were normally below 8 Hz and accelerations amplitudes well above ±1 g were measured.
If one were to meet these accelerations at about 2 Hz or under, then the displacement amplitudes involved are around ±4 in - outside the range of electrodynamic systems. Thus an electrohydraulic system was chosen as being more able to generate the low frequency vibrations than being met in aircraft. Accepting that such a rig would not need to generate the motions necessary to produce sea-sickness (roughly ±4 ft at 0.3 Hz), an arbitrary figure of ±10 in was selected as being the required maximum stroke. This will give us ±1 g at 1 Hz and higher frequencies.

The size and payload of the table were dictated by the need to accommodate on the table one subject dressed in full flying equipment, sitting in an ejection seat, with representative control column and rudder pedals and some instruments. The decided figures were 6 ft × 4 ft with payload of 600 lb.

A copy of the specification drawn up by the author and Technical Facilities Department, RAE is attached as an Appendix to this Chapter.
GENERAL DESCRIPTION OF THE RIG AND ITS OPERATION

The rig consists basically of a flat aluminium table (6 ft x 4 ft weighing about 450 lb) supported by three hydraulic actuators, attached to the table by trunnions. Two jacks support the table in the vertical axis and the third is attached horizontally to the side of the table, in the same plane as the vertical jacks, to prevent the table parallelogramming (Figs. 1 to 4). Any tendency for the table to move in the plane at right angles to the jacks is prevented by the trunnions. Each vertical actuator has a piston area of 1.75 sq in and a maximum stroke of ±10 in controlled by three servo valves (Fig. 3). The horizontal actuator has a piston area of 3.5 sq in and a stroke of ±10 in controlled by six servo valves (Fig. 4). Each actuator forms part of a closed loop servo system. The command signal applied to the electronic control gear is compared with two feedback signals derived from table position and differential pressure across the piston. The difference is electronically amplified and applied to the servo valves which control the flow of oil into or out of the cylinder. The servo valves are connected hydraulically in parallel and electrically in series. Each servo valve has a maximum flow of 40 cu in/s which gives a velocity limit in either axis of approximately 60 in/s.

The acceleration and frequency ranges possible from the rig, in both axes, separately or simultaneously, are 0 to ±2 g and 0.5 Hz to about 50 Hz* with a working payload of 600 lb. Large excursions of the load C of G are permitted. The specified limits are 12 in lateral offset from a line joining the two vertical jacks and a height above the table of 24 in.

2.1 Input Parameters

The rig is equipped with its own oscillators connected as displacement inputs. The frequency applied to the two axes can be the same (with adjustable phase angle) or different in the range 0.5 Hz to about 50 Hz. The

* ±1 g at 1 Hz and ±2 g at higher frequencies
control console has input sockets where external displacement or acceleration generators can be connected to the rig. The acceleration input socket is connected internally to the displacement socket via two integrators.

2.2 Feedback

Two feedback loops are used. A signal proportional to differential pressure across the piston is fed back - this reduces stiction effects at the crest of a sine wave and improves acceleration waveforms. In order to provide displacement signal feedback, the outputs from resolvers fitted to the bearing trunnions and the displacement transducers are combined to give a signal proportional to the true horizontal or vertical displacements. To eliminate the tendency for the table to rock due to high C of G of the load when the horizontal actuator is working at high g levels, two further commands are fed to the vertical actuator servo loops. A signal proportional to horizontal acceleration is phase adjusted and fed, one inverted and one non-inverted, to the two error amplifiers on the vertical actuators.

2.3 Safety features

The rig is fitted with various safety features such that under any foreseen emergency situation the resulting acceleration or deceleration imposed on the subject is insufficient to injure him. The pistons are fitted with end snubbers and buffers so that it is virtually impossible for the actuator to contact the end of the cylinder. Electronic limits for acceleration and displacement are provided. Emergency stop buttons are provided which are also connected to oil level, pressure and temperature units etc. A battery mains protection unit is also fitted.
COMMISSIONING

This section covers the performance aspect of the vibration rig and details tests carried out to ascertain whether or not the rig met the specification. No detailed graphs, records of tests, or acceleration traces are given due to their multiplicity.

The commissioning trials basically covered three main topics (i) low harmonic distortion, (ii) flat frequency response requirement and (iii) safety factors.

Clearly other tests were conducted to ensure that the system when installed met the specified requirement; these will be briefly referred to at the end of this section by repeating the requirement from the specification and then indicating the results of the tests.

In order to check the characteristics of the rig, a test specimen of the correct weight and C of G (together with the means of altering the position of the C of G within the limits laid down in the specification, section 4.7) was constructed (see Fig.3).

3.1 Harmonic distortion

Referring to the specification, 4.3(d) reads:-

Distortion of the fundamental acceleration sine wave within the frequency range 1 to 10 Hz as measured at the platform when it is fully loaded must not exceed 15% error and may be defined as

\[ \sqrt{\frac{A_1^2 + A_2^2 + A_3^2 \ldots A_0^2}{A}} = 0.15 \]

up to the 20th harmonic.

The amplitude of the harmonic shall lie under a curve bounded by 15% at the first harmonic to 2% at the 10th. It would be acceptable if the above limits are obtained with the aid of mechanical filters.

The distortion figures were checked using the test specimen adjusted to give maximum offset of the C of G - this was considered to be the worst
Distortion was measured using a Solartron Digital Transfer Function Analyser fitted with a harmonic analyser. This performs a Fourier series analysis via a digital filter. This enabled harmonics up to the 10th to be measured. The specification requires the distortion to be measured up to the 20th harmonic, but it was found that the distortion fell off rapidly after the 7th harmonic so that any possible differences between summing up to the 20th harmonic and up to the 10th harmonic were ignored.

The results of the harmonic analysis are shown in the tables below. The results in the brackets are those from tests done some time after the main body and serve two purposes. Firstly they were used to check the rig performance at the conclusion of all the other tests, to see if any wear or other factors had occurred, and secondly they were used to check the distortion at those frequencies and levels where the original figures put the rig out of specification.

### VERTICAL

<table>
<thead>
<tr>
<th>% Total</th>
<th>0.15 g</th>
<th>0.50 g</th>
<th>1.0 g</th>
<th>1.5 g</th>
<th>2.0 g</th>
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<tr>
<td>1 Hz</td>
<td>24.4 (24.8)</td>
<td>26.8 (29.0)</td>
<td>17.7 (18.0)</td>
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<td>N/A</td>
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<tr>
<td>3 Hz</td>
<td>34.4 (35.4)</td>
<td>10.2</td>
<td>12.1</td>
<td>13.3</td>
<td>16.6 (16.1)</td>
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<tr>
<td>5 Hz</td>
<td>33.7 (33.2)</td>
<td>11.0</td>
<td>8.6</td>
<td>11.1</td>
<td>14.7</td>
</tr>
<tr>
<td>7 Hz</td>
<td>29.2 (29.0)</td>
<td>12.0</td>
<td>9.5</td>
<td>8.9</td>
<td>12.6</td>
</tr>
<tr>
<td>10 Hz</td>
<td>32.6 (35.0)</td>
<td>13.4</td>
<td>8.9</td>
<td>8.3</td>
<td>10.6</td>
</tr>
</tbody>
</table>

N/A = Not applicable

### HORIZONTAL

<table>
<thead>
<tr>
<th>% Total</th>
<th>0.15 g</th>
<th>0.50 g</th>
<th>1.0 g</th>
<th>1.5 g</th>
<th>2.0 g</th>
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<tbody>
<tr>
<td>1 Hz</td>
<td>15.8 (19.0)</td>
<td>14.9</td>
<td>19.5 (18.0)</td>
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<td>N/A</td>
</tr>
<tr>
<td>3 Hz</td>
<td>19.0 (17.0)</td>
<td>7.3</td>
<td>8.6</td>
<td>8.3</td>
<td>10.5</td>
</tr>
<tr>
<td>5 Hz</td>
<td>22.6 (23.2)</td>
<td>5.2</td>
<td>6.7</td>
<td>5.6</td>
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<tr>
<td>7 Hz</td>
<td>18.0 (19.0)</td>
<td>9.0</td>
<td>4.8</td>
<td>5.5</td>
<td>6.7</td>
</tr>
<tr>
<td>10 Hz</td>
<td>21.8 (20.0)</td>
<td>12.2</td>
<td>4.7</td>
<td>5.5</td>
<td>7.5</td>
</tr>
</tbody>
</table>

N/A = Not applicable
The above figures were obtained using all six servo valves in each axis. In order to reduce the distortion, especially at the lower frequencies where the specification figures had been exceeded, a system was included whereby the number of hydraulic valves per actuator could be selected depending on the acceleration levels required. In practice this amounted to running the rig with the minimum number of valves selected and then increasing the number until any sign of hydraulic flow limiting present, disappeared. The distortion levels then measured proved to be considerably less than those shown in the Table. Also some special high resolution values were evaluated, but were found to give essentially the same performance as a normal valve.

Typical check values of distortion at much lower acceleration levels than before are given in the following Tables, and are illustrated in Figs. 5, 6 and 7.

### VERTICAL

<table>
<thead>
<tr>
<th>% Total</th>
<th>0.25 g</th>
<th>0.12 g</th>
<th>0.06 g</th>
<th>0.025 g</th>
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<td>3 Hz</td>
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<td>0.50</td>
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<td>5 Hz</td>
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<td>9.1</td>
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<td>8.2</td>
<td>12.8</td>
<td>19.2</td>
<td>1.35</td>
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### HORIZONTAL

<table>
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<tr>
<th>% Total</th>
<th>0.25 g</th>
<th>0.12 g</th>
<th>0.06 g</th>
<th>0.025 g</th>
<th>Limiting g</th>
</tr>
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<tr>
<td>3 Hz</td>
<td>6.5</td>
<td>9.8</td>
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<td>30.2</td>
<td>0.60</td>
</tr>
<tr>
<td>5 Hz</td>
<td>7.8</td>
<td>8.2</td>
<td>14.8</td>
<td>22.3</td>
<td>1.0</td>
</tr>
<tr>
<td>7 Hz</td>
<td>5.6</td>
<td>7.6</td>
<td>11.4</td>
<td>17.9</td>
<td>1.4</td>
</tr>
</tbody>
</table>

Limiting g is defined as the approximate level where flow limiting of the hydraulic servo valve starts to become apparent on the acceleration wave-form.
Thus it would appear that as long as accelerations below ±0.06 g are not required then distortions less than 15% can be achieved (compared with figures of approaching 100%, at these levels, measured on some other vibrators). Part of the original decision to purchase an electrohydraulic machine rather than an electrodynamic was the assumption that the rig would never be used for very low signal work such as long term comfort or threshold, so an arbitrary lower limit of 0.06 g is satisfactory. This is not to say that the rig cannot be used at lower levels, it is just the level at which the acceleration distortion exceeds 15%.

If one looks at the specification (4.3(d)) an error is immediately obvious. The distortion requirement does not stipulate an acceleration level at which the distortion should be measured. Thus the original specification cannot be said to be not met because the distortion is 33% for an input level of ±0.025 g at 3 Hz. However notwithstanding, in terms of distortion, the performance of the rig is highly satisfactory.

In trials following the commissioning, when subjects were sitting on the rig, instead of the test weight, distortion measurements were taken which agreed closely with those quoted above.

3.2 Flat frequency response

The specification states that:

4.3 The response of the complete facility from signal input to platform movement should be as follows:

(a) Frequency 0 to -1 dB over the entire range, i.e. 0/>50 Hz.
(b) Cut-off frequency >40 Hz, above 40 Hz permissible fall 12 dB/octave.
(c) Phase lag not to exceed 20° at 5 Hz, performance at other frequencies within the range 1 to 10 Hz to be stated.

Initial measurement of the rig frequency response (displacement output/voltage applied to the input displacement socket versus frequency, or acceleration output/voltage applied to the input acceleration socket versus
frequency) indicated that the rig had a damped resonance at about 2 Hz and thereafter fell off at 12 dB/octave.

Thus initial indications were that the rig would be unable to reproduce flight vibrations above about 1.5 Hz.

However by altering the feedback characteristics of the integrators which converted any input acceleration voltage into an input displacement signal to the vibrator, it was possible to compensate for the originally unacceptable frequency response. Fig.8 shows a typical feedback circuit having an effective rising characteristic in the required frequency range - thus counteracting the falling characteristic of the rig.

This circuit has a flat characteristic up to 0.15 Hz, then falls at 12 dB/octave to 2.1 Hz, then 6 dB/octave to 2.8 Hz, and finally is flat from 2.8 Hz up to 50 Hz (assuming a 50 Hz low pass filter). Comparing this with standard integrator fall off of 12 dB/octave (for two integrators), this compensation circuit initially climbs at 12 dB/octave, is then flat between 0.15 Hz and about 2.4 Hz and then climbs at 12 dB/octave up to 50 Hz. Fig.9 shows the resulting normalized response curves.

The resulting frequency response - which now applies only for acceleration output/voltage applied to the input acceleration socket versus frequency - was found to be flat up to 40 Hz, within 3 dB, with a linear phase/frequency response, for input signals equivalent to 2 g down to 0.5 g. At lower g values it was found that the frequency response was far from flat, but it was decided that this was a function of the electronics rather than the vibrator, so should not be considered a specification failure. It was accepted that by more careful selection of electronic components a flat frequency response, down to low g levels, could be achieved.

As per the requirement above 40 Hz the frequency response fell off rapidly.
As a check, the weight of the test on the rig was reduced to determine the effect on the frequency response. It was found that the response was significantly different and it was thus decided to stipulate that for any future tests the payload should be kept at about 600 lb - if necessary by adding weights. Clearly it is impedance and not weight that provides the difference - and as this is the unknown that is being determined it has been decided to purchase a set of filtering spectrum shapers to use in conjunction with the feedback circuits to provide a flat response curve.

3.3 Safety features

These were briefly outlined in section 2.3. Section 4.11 of the specification states:

Means must be provided at the platform and control console to enable either the subject or controlling operator to stop all motion immediately with a maximum 10g retardation in the event of an emergency. Means must also be provided to absorb and contain the kinetic and potential energy at the extremities of movements in the event of the control system failing and run-away of the machine taking place.

Each piston rod end is fitted with a buffer plate which makes contact with a rubber buffer when the piston rod is at the extreme end of its travel, e.g. 12 in. The working stroke of the rig is ±10 in and the actuating oil pressure is fed into the cylinder 1 in from the end of the stroke. Thus oil is trapped between the piston and the end of the cylinder when the travel exceeds 11 in. These hydraulic dampers are known as snubbers and the pressure built up in the snubbers is controlled by two adjustable differential piston type relief valves mounted back to back. Since these valves are permanently connected across the piston hydraulically, they will always limit the maximum differential pressure across the piston and hence the maximum acceleration.
The worst failure case possible on a rig of this description is that it should enter the snubber/buffer area at maximum possible velocity (about 62 in/s for this rig). This was arranged by feeding a step signal into the displacement input demanding an output of about 25 in. The resulting retardation was about 22 g followed by a rebound acceleration of about 12 g, then followed a series of decaying oscillations peaking at about 6 g. The initial high retardation and subsequent acceleration took place in only about 10 ms and thus would not present any problems to the subject other than the psychological effect of hearing the thump.

Fig.10a shows a typical trace of running into the end stops.

Electronic limits for acceleration and displacement are included. These are adjustable between 0 and 2 g and 0 and 10 in. Any excursion outside the set limits initiates the Emergency Stop procedure.

Various other fault conditions can initiate the Emergency Stop procedure, they are oil level pressure, temperature or filter, air temperature, platform tilt and external safety. If any of these conditions necessitates shutdown, then the signal from the internal oscillator is decayed to zero in approximately one second, a ramp signal is fed to the actuators to return them to the rest position of the table, and the hydraulic pump is shut down. The g level associated with these emergency stops was always found to be less than the original oscillator setting.

Fig.10b shows a typical trace where the original setting was an oscillation of ±1 g vertically at 5 Hz. When the emergency stop button was pressed the oscillation decayed to zero in about 1 second. The two subsequent spikes were a result of the platform grounding on the two vertical actuator trunnions.

Feedback failure, induced by the removal of a wire, produced a transient oscillation of ±4.7 g at the driving frequency which developed into an oscillation of ±2.3 g at 63 Hz. This frequency could also be induced by increasing the feedback until the loop went unstable.
For this test the emergency stop circuit was by-passed, so in real-life the rig would be shut down after about 1 second.

Fig. 10c again shows a typical trace where a 3Hz oscillation at ±1 in the horizontal axis develops as described above.

The control of the vibrator under Emergency Stop conditions is clearly dependent on the mains voltage. As an additional safeguard the system included a mains protection unit, consisting of a set of rechargeable batteries with sufficient power to control the ramping down of the rig and stopping the pump if the mains voltage fails.

The above results were roughly the same for the two axes, and were all done using the maximum payload with maximum offset of the C of G. Various levels and frequencies within the range 0 to 2 g and 1 to 10 Hz were covered.

Apart from the transient peak when the rig was run into the snubbers, which should not affect the subject, the specification was met.

3.4 Other factors

This section will be confined to a repeat of the relevant section in the specification followed by a brief report of the appropriate tests.

4.1 The platform is required to move in the horizontal and vertical planes with amplitudes as follows from mid-position of the stroke.

(a) Horizontal 10 in (total stroke movement 20 in).

(b) Vertical 10 in (total stroke movement 20 in).

The console mounted displacement meters were calibrated by applying a dc voltage to the rig inputs and measuring the resulting displacements with a foot rule. The oscillators were then set to 1 Hz and the amplitude increased to the required ±10 in.

4.2 The horizontal and vertical movements are required to take place either simultaneously or independently, with a different signal to each plane of movement. It is required that all movements will give the following effect on the platform when it is moving in one or both planes either independently,
simultaneously or with any random movement resulting from a combination of horizontal and vertical displacements.

(a) Frequency 1-10 Hz.

(b) Acceleration of sinusoidal wave-form in accordance with that shown in Appendix I to give a peak g pattern over the required frequency range of 1-10 Hz. Similar acceleration rates obtainable with waveform patterns of square, stepped ramp, saw-tooth, random etc., are desirable but are subservient to that of the sinusoidal requirement.

(c) Tenderers should indicate if it is possible to extend the frequency range of platform oscillations from 1 to 10 Hz to 1 to 50 Hz and an indication of performance at the extremities of the extended range would be appreciated.

Following calibration of the accelerometers mounted on the table, cross coupling of acceleration between the two axes were measured, to ensure that any signal put onto one axis was unaffected by the presence of any signal on the other axis. It was found that across the frequency range 1 to 10 Hz and within the required acceleration range, the mean value of the cross coupling term was 3%. It should be noted that the usual figure quoted for cross-axis-sensitivity of accelerometers is of the order of 2%.

The extension of the frequency range required in (c) is covered in 4.3(a).

The above tests were conducted with the maximum payload and maximum offset of the C of G both vertically and horizontally.

4.4 The total back-lash of all platform actuating mechanisms must not give a free oscillatory movement of < 0.02 in at the platform.

It was decided to test the system by demanding a response from the table corresponding to an amplitude of ±0.001 in (1 thou). If the resulting acceleration waveform was of the correct amplitude and relatively distortion free then we could say that the back-lash in the system was at least an order less than the amplitude used. The figures chosen were ±0.16 g at
40 Hz, for both the vertical and horizontal axes. Maximum payload and maximum offset of the C of G were used. The results indicated that the backlash in the rig was insignificant.

Specification met.

4.5 Stability of the whole system must be such that its response remains stable over a three hour period of operation at maximum endurance.

It was decided to run the rig for eight hours (not three), as being more representative of a working day, at ±1 g in both heave and sway at 5 Hz. The phase angle between the axes was arbitrarily chosen at 320°. Readings of distortion on both axes were taken every 30 minutes as a check on performance. The tests produced no snags, the rig performance and distortion figures remaining constant apart from a slight rise in the overall acceleration level on the heave axis after about 6 hours (an increase of about 3%).

Specification met.

4.7 It is anticipated that the C of G of the payload will fall within a 12 in diameter circle at the centre of the platform and will not be more than 24 in above its top surface.

Most of the tests described were conducted with the maximum load (600 lb) and maximum offset of the C of G (24 in above the table and 12 in lateral offset at right angles to the line joining the two vertical jacks).

In order to check the effect of load and C of G position, various spot checks were made, mainly of distortion and frequency response. Change of C of G and payload had little effect on distortion produced (approximately 1% difference).

The frequency response curves were generally unaffected by C of G position but were significantly different when the load was removed. However, it is envisaged that for all tests planned the payload will always be made up to the maximum 600 lb.

Results acceptable.

4.9 The platform must be constructed so that vertical forces applied to its centre are reproduced at its extremities. It should be approximately
6ft long × 4ft wide. The resonant frequency of the platform when unloaded should be not less than 500 Hz.

In order to test the stiffness factor of the table it was decided, first to excite the table vertically and measure the resulting vibration at the extreme edge of the table, and then excite horizontally and again measure the vibration at the edge. The results indicated that for both cases the vibration at the edge of the table was at worst 6% of, or 6% greater than, that at the centre of the table.

By crudely striking the table when the weight was removed, it was found that the apparent resonant frequency of the table was in excess of 200 Hz, i.e. considerably in excess of the required frequency range.

The table stiffness tests were conducted with maximum payload and offset of the C of G.

Other conditions.

Section 3.4 required that the noise generated by the complete facility when operating at full power must not exceed 75 dBA at the platform. The maximum noise levels measured were for the condition ±1 g at 35 Hz, when the noise level was approximately 74 dBA. It should be noted that these readings were taken with the pump enclosure incomplete in that the door had no positive closing mechanism and the inlet fan was uncowled.
CONCLUSIONS

Before the present rig was commissioned, any work on vibration, be it performance or biodynamics, had to put up with a high level of harmonic distortion (greater than 50%) and poor frequency response characteristics. So poor in fact that it was virtually impossible to play tape recordings into a rig and faithfully reproduce real-life accelerations up to say 20 Hz. This Chapter has shown that we now have a rig which has a high degree of built-in safety, low distortion and an adequate frequency response.

Work conducted on the rig, subsequent to the commissioning trials, has borne out all the promise laid out in specification. Flight profiles measured in transport and fighter aircraft - for vertical and horizontal motion - have been reproduced on the table and biodynamic and performance experiments conducted.
APPENDIX I

Specification for a Vibration Machine and Associated Equipment

to be installed at the Royal Aircraft Establishment

Farnborough, Hants.
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1. **FORWARD**

A requirement exists at RAE, Farnborough for a test rig to provide oscillatory movement in the vertical and horizontal planes to a work table or platform. Mounted on the platform will be an instrument panel, an aircrew seat and an aircraft control column. Accommodated on the seat will be a human subject who will be carrying out various operational tasks whilst the work table is moving. The movement will be simulating vibrations as may be experienced by aircrew in the flight control cabin of an aircraft. It is planned to install the test rig in Building R132 which is located in the Main Area of RAE.

2. **GENERAL CONDITIONS AND SCOPE OF THE SPECIFICATION**

2.1 The successful Tenderer, hereafter referred to as the Contractor, will be responsible for the whole of the facility and as such must co-ordinate all sub-contract work and be responsible for its proper execution to ensure satisfactory completion of the facility. The Contractor will be responsible for the suitability of all individual products manufactured by sub-contractors and for ensuring that a complete and reliable facility is handed over to RAE ready for operation. All work is to be carried out to the satisfaction of the Director, RAE or his Representative Officer (RO).

2.2 Adequate safety precautions must be taken by the Contractor at all times during the execution of work and the Contractor will be required to make good at his own expense any loss or damage arising as a result of work being carried out by him or on his behalf.

2.3 In addition to all standard conditions of Government Contracts as set out in the attached schedules and forms this specification and accompanying drawings are intended to cover all work and material necessary and proper for the facility contemplated. In case by inadvertence or otherwise, the specification and drawings omit to require or indicate some work and material necessary for that purpose, the Contractor shall nevertheless be required to provide the same so that the facility may be completed according to the true intent and purpose of the specification.

2.4 The Contractor will be required to commission and prove the facility ready for operation before handing it over to RAE. This includes the preparation, cleaning, flushing out, intermediate and final priming and charging of all systems and a demonstration on site that the whole facility meets the performance requirements as detailed in this specification or that offered in his tender and contractually accepted in lieu thereof.

3. **GENERAL DESCRIPTION AND REQUIREMENT**

3.1 The preferred location and relative positions of the major parts of the facility are as indicated. This must not be regarded as a working drawing and Tenderers may put forward alternative schemes or suggestions for consideration.
3.2 It is desirable that the vibration platform, its supports, guides and actuators are designed as one unit capable of being bolted to a foundation block at floor level. An access platform and safety rails are to be provided at the mid position of the vertical stroke. Means of moving the platform and locating it in the mid vertical and horizontal positions are to be provided.

3.3 The execution of all foundation and associated building work will be the responsibility of RAE. The Contractor must supply full and comprehensive drawings for this work bearing in mind that special precautions must be taken to prevent the transmission of vibration to other nearby laboratories and offices and Tenderers should clearly show the means they propose to adopt to install the machine and physically isolate it from its surrounds.

3.4 Hydraulic power packs and associated equipment which operate at high noise levels should be contained within a noise attenuating enclosure preferably located near the southern wall of R132 Building. The enclosure may be ventilated via the south wall of the building or the roof. Alternative means such as individual enclosure of units may be provided to ensure that noise generated by the complete facility does not materially add to the overall noise level in the building. Noise generated by the complete facility when operating at full power conditions must not exceed 75dB 'A' measured at the platform.

3.5 The Tenderer must include for the supply and installation of all necessary interconnecting pipework, cables and wiring for satisfactory and efficient operation of the machinery and equipment being supplied.

3.6 RAE will bring services up to terminal points located in the sound-proof housing shown. The Contractor must give RAE details of the requirement as early as possible, i.e. total load in kilowatts, power requirement of pump motor, etc, so that arrangements can be made with the Establishment's Service Department to fit into their work programme the laying of water pipe lines and cables into the building. The exact position and form of termination will be agreed with the Contractor before this work is started. Generally water lines will be terminated with a stop valve and cable runs with a switch fuse or isolator.

3.7 RAE will be responsible for connecting the analogue computer, amplifiers and tape recorder supplied by them to the equipments supplied by the Contractor.

3.8 RAE will be responsible for carrying out civil and building work necessary for installation of the facility. A complete set of detailed working drawings in triplicate must be submitted not less than 4 months after the contract has been let. The Contractor should present to RAE an outline of installation proposals not later than 5 weeks after the contract has been let and this must be followed with drawings as above.
3.9 It is desirable that all controls are grouped together on a single console. If however this complicates the design, controls positioned near their respective units will be considered. Clear indication of the Tenderer's proposals on this particular feature is required in order to assess how conveniently the plant can be operated.

3.10 It is intended that control, monitoring and recording of the machines' performance will be done by the equipment which RAE are supplying under Appendices III, IV and V. All equipment which the Tenderer may offer to augment the RAE units being supplied should be located in the area indicated.

3.11 All Tenderer's quotations must show a breakdown of costs under the following two main sections:

(i) Cost of machine, power actuators, etc and installation work.

(ii) Cost of control console and control equipment, if any, in addition to that supplied by RAE under Appendices III, IV and V and that normally supplied with hydraulic power packs necessary for the machine to meet the performance requirements under para. 4.

3.12 Tenderers may put forward proposals to actuate the machine by means other than hydraulic as outlined in this specification but the performance requirements remain the same.

3.13 The Contractor must supply the services of competent staff to work under the direction of a Resident Engineer to supervise the whole installation of the facility and its commissioning. It is required that the Resident Engineer or his appointed deputy will instruct RAE personnel in efficient operation of the facility. An outline programme of work from Confirmation of Contract to completion and hand-over of the facility should be included with the Tender Documents.

3.14 Final "making good" of surface finishes etc. occasioned by installation work is the responsibility of the Contractor and must be carried out to the satisfaction of RAE.

On completion of all site work and before the facility is accepted the Contractor must remove all surplus material from the site and leave it in a clean and tidy state to the satisfaction of the RAE RO.

4. DESIGN REQUIREMENTS AND PERFORMANCE

4.1 The platform is required to move in the horizontal and vertical planes with amplitudes as follows from mid-position of the stroke.

(a) Horizontal 10 inches (total stroke movement 20 inches).

(b) Vertical 10 inches (total stroke movement 20 inches).
4.2 The horizontal and vertical movements are required to take place either simultaneously or independently, with a different signal to each plane of movement. It is required that all movements will give the following effect on the platform when it is moving in one or both planes either independently, simultaneously or with any random movement resulting from a combination of horizontal and vertical displacements.

(a) Frequency 1-10 Hz.

(b) Acceleration of sinusoidal waveform in accordance with that shown in Appendix I to give a peak "g" pattern over the required frequency range of 1-10 Hz. Similar acceleration rates obtainable with waveform patterns of square, stepped, ramp, saw-tooth, random etc are desirable but are subservient to that of the sinusoidal requirement.

(c) Tenderers should indicate if it is possible to extend the frequency of platform oscillations from 1 to 10 Hz to \( \frac{1}{2} \) to 50 Hz and an indication of performance at the extremities of the extended range would be appreciated.

4.3 The response of the complete facility from signal input to platform movement should be as follows:

(a) Frequency 0 to \(-12\) dB over the entire range, i.e. 0/ \( \text{dB} > 50 \) Hz.

(b) "Cut-off" frequency \( > 40 \) Hz, above 40 Hz permissible fall 12 dB per octave.

(c) Phase lag not to exceed 20° at 5 Hz, performance at other frequencies within the range 1 to 10 Hz to be stated.

(d) Distortion of the fundamental acceleration sine wave within the frequency range 1 to 10 Hz as measured at the platform when it is fully loaded must not exceed 15 per cent error and may be defined as

\[
\frac{\sqrt{A_1^2 + A_2^2 + A_3^2 + \ldots + A_n^2}}{A_0} = 0.15
\]

up to the 20th harmonic.

The amplitude of the harmonics shall lie under a curve bounded by 15 per cent at the first harmonic to 2 per cent at the 10th. It would be acceptable if the above limits are obtained with the aid of mechanical filters.

4.4 The total "back-lash" of all platform actuating mechanisms must not give a "free oscillatory" movement \( > 0.02 \) inches at the platform.

4.5 Stability of the whole system must be such that its response remains stable over a three hour period of operation at maximum endurance.
4.6 The total "payload" to be placed on the platform is 600 lb which includes a human subject, aircrew seat and simulated flight control equipment. An allowance of approximately 200 lb should be made for the human subject and his personal equipment.

4.7 It is anticipated that the C of G of the "payload" will fall within a 12 inch diameter circle at the centre of the platform and will not be more than 24 inches above its top surface.

4.8 The machine must be robustly constructed and all moving parts adequately supported in guide rails or slideways to promote long life of actuator equipment and associated ancillaries.

4.9 The platform must be constructed so that vertical forces applied to its centre are reproduced at its extremities. It should be approximately 6 ft long x 4 ft wide. The resonant frequency of the platform when unloaded should be not less than 500 Hz.

4.10 The top surface of the platform must be flat within a tolerance of \( \pm \frac{1}{16} \) inch over its whole surface. It must be so constructed that holes may be drilled in it for bolting down equipment. The Contractor is to submit a detailed drawing of the platform to RAE before manufacture is commenced.

4.11 Means must be provided at the platform and control console to enable either the subject or controlling operator to stop all motion immediately with a maximum 10g retardation in the event of an emergency. Means must also be provided to absorb and contain the kinetic and potential energy at the extremities of movements in the event of the control system failing and "run-away" of the machine taking place.

4.12 Where applicable all components, joints, attachments, screwed fittings etc must be capable of withstanding vibration and rapid load reversal without failure or working loose.

4.13 The complete facility should be designed to give a useful service of not less than 10,000 hours. The Tenderer must indicate those parts of the facility where this condition cannot be met and give an indication of the estimated useful service life to be expected.

4.14 The hydraulic power system must be a complete and self-contained assembly suitable to provide the motive power to the machine to give the required performance as previously detailed. It must be complete with electric motor driven pumps, platform actuators, reservoir and header tanks, filters, coolers, shut-off, isolating and pressure release valves, pipework, accumulators, etc, and include the following features in the assembly design.

(a) A full flow 5 micron filter complete with a means to warn of filter blockage and a filter bypass line complete with valves to isolate the filter if necessary without shutting down the plant.

(b) System filling point containing a 10 micron filter.
(c) A sampling point and equipment to check oil cleanliness.
(d) Catchment and drainage of undesirable leakage i.e. drip trays.
(e) Strategically positioned drain valves, so that the complete system may be drained.
(f) Strategically positioned vent valves so that the whole system may be replenished without trapping or entraining air.
(g) Means to readily check the hydraulic fluid levels.
(h) Adequate support and fixing of all pipework, valves, etc and compensating means of relieving all strain imposed by temperature changes.
(i) Sufficient thermometer pockets to check malfunctioning of parts of the system or individual components which may lead to an increase in oil temperature.
(j) Sufficient capacity to avoid, high rates of circulation, temperature cycling, fluid decomposition and foaming.
(k) Operating pressure of the hydraulic fluid should be as low as possible consistent with the required performance in order to minimise leakage and noise generation.
(l) Provision should be made for manual control of the hydraulic system so that the platform position may be adjusted to facilitate setting up experiments or carrying out maintenance service work. Positive and foolproof locking of the platform in any required position within its maximum displacement must be provided.
(m) The hydraulic fluid which the system is finally filled with must be non-toxic, non-corrosive and non-flammable to a reasonable degree as accepted in commercially marketed products.
(n) Tenders must state the number of particles of a given size in a representative 100 millilitre sample of hydraulic fluid at which precision sections of the hydraulic system will function satisfactorily.

The preferred range and increments of particle size are as follows:

Particle sizes:  
5 - 14 microns
15 - 20 microns
24 - 49 microns
50 - 100 microns
100 and over microns
(o) An indication of hydraulic fluid flow in low pressure sections of the circuits is required, e.g. header tank return, pump suction, cooler inlet or outlet.

(p) Sufficient pressure and temperature indicating gauges must be provided to permit operators to visually check satisfactory operation of all the main and important circuits of the equipment.

(q) Adequate means must be provided to combat pressure surges under all operating conditions.

(r) Sliding surfaces must be protected with flexible gaiters to exclude dust and foreign matter. Where this is not possible surface wiper blades or pads must be attached to all moving components to prevent trapping of foreign matter which may cause defacing of sliding or rolling surfaces.

(s) Lubrication of all moving parts must be adequate for the duties involved and automatically operated immediately the test rig is set in motion. It may take the form of either:

(i) Pressurised recirculating oil system. (This must be complete with pump, tanks, pipework etc and cooler if necessary.)

(ii) Pressurised total loss oil system. (This must incorporate a means of collecting expended oil and directing it to a common sump referred to at 4.14 (o).)

(iii) Grease lubricating system comprising a central dispensing unit, augmented where necessary with Stauffer cups or grease gun nipples if it is not possible to obtain a supply from the central unit. It is essential that cups or nipples are located in easily accessible places.

Lubrication seals must be fitted in all instances where seepage is likely to be a nuisance.

(t) Indicators must be provided to show that cooling water is being supplied to the hydraulic fluid cooler.

(u) Tenderers should forward outline dimensioned sketch plans of their proposed installation indicating the position of all major items of equipment.

4.15 All cooling water pipework must be hot-dip galvanised in accordance with BS.729: Part 1: 1961. Other component surfaces wetted by cooling water must be coated with a corrosion resistant paint or alternatively sprayed with a suitable metallic substance.

4.16 Where it is necessary to thermally insulate pipework, heat exchangers etc the lagging must be fire and vermin proof in accordance with BS.799.
4.17 All exposed surfaces shall be cleaned free from rust and millscale and suitably prepared, e.g. welds dressed and sharp corners removed prior to being painted or enamelled at the Contractor's Works in battleship grey or similar colour.

Pipework shall be painted in bands of the correct BS Colour Code appropriate to services or functions.

Machined faces, shafts and other similar bright surfaces normally left unpainted are to be adequately protected against rust before leaving the Contractor's Works.

5. ELECTRICAL

5.1 The standard available supply is 400V, three phase, 50 Hz and 230V, one phase, 50 Hz both subject to a voltage fluctuation of ±10 per cent. Automatic control equipment shall be suitable for operation under these conditions without adverse effect on controlled parameters.

5.2 The electrical equipment supplied is to be constructed throughout in accordance with the appropriate British Standard. It must also comply with the Regulations for Electrical Equipment of Buildings, as issued by the IEE wherever applicable.

5.3 All earths, main cable and auxiliary wiring connections must be provided with permanent identification and be in accordance with that shown on all relevant drawings supplied with the equipment.

For identification purposes the following colour code shall be used for power conductors.

(i) AC phase conductors: Red, Yellow and White.

(ii) AC neutral conductors: Black.

(iii) AC earth conductors: Green.

Different colours from the above shall be used for control and instrumentation circuits.

Electrical apparatus such as switchgear, distribution boxes etc shall be painted in the approved BS Colour (orange) at the Contractor's Works.

5.4 All necessary switchgear for safe operation and maintenance of the electrical equipment shall be supplied and fixed by the Contractor. In the event of a fault or breakdown the defective unit must be automatically disconnected from its supply. Fuses shall be HRC type throughout. Overload protection where fitted shall be suitable for manual re-setting and should be the thermal type. Fuse ratings and overload settings shall be determined by the Contractor to ensure adequate protection of the equipment and where applicable discrimination between protective devices shall be incorporated into the overall scheme.

5.5 Guards or enclosures must be provided on all equipment to prevent accidental contact with any live or moving part.
5.6 Means must be provided to readily make the equipment safe for servicing and maintenance work.

5.7 All cubicles and panels should be of sheet metal construction, floor mounted with adequate means of access. Ventilation louvres should be provided as necessary to keep within the design operating temperatures of the electrical insulating materials employed.

5.8 In all instances extensive lengths of small bore pipework for the purpose of indicating the condition of fluids such as cooling water or hydraulic media should be avoided. Where it is necessary to indicate remotely from the point of measurement electrical transmitters should be used and in all instances where an instrument is mounted in a cubicle or a control panel containing other electrical equipment or instruments this method must be employed.

6. STANDARD CONDITIONS

6.1 The available electrical power supply on site is as given under Section 5.1.

Surface water drainage is to a nearby gully.

A domestic water supply can be made available for the normal requirements of building work.

RAE will make all temporary connections to site services as required by the Contractor during installation work.

6.2 The Contractor is responsible for ensuring that all equipment supplied under the contract related to this specification meets the mandatory requirements and recommendations of the Factory Acts and other Statutory Bodies. He must also ensure that a reasonable factor of safety is incorporated into the design of the facility to give efficient reliable operation consistent with Section 4.13.

6.3 Suitable attachments such as lugs, eye bolts and shackles must be provided on all equipment requiring the use of lifting tackle.

6.4 Where special tools or other accessories are required for the maintenance of the vibration machine and its associated plant and equipment, one complete set shall be supplied. Each item must be clearly labelled and its designation and function listed.

6.5 Each unit shall have a nameplate of durable material attached in a prominent position. It shall give details such as the manufacturer's name, serial or model number, type, year of manufacture, rating and other important relevant information.

6.6 The approximate weight of all major items of the facility should be stated in the Tender Documents.

6.7 Three copies of all test certificates as required by the Factory Acts and insurance surveyors appropriate to individual items of equipment must be supplied.
6.8 The Contractor must supply three complete "As Installed" sets of drawings of the complete facility at the time the plant is handed over to RAE. All electrical drawings and circuit diagrams must agree with the coding and identification labels of the equipment installed.

Six copies of Operating Instructions and Service Manuals must be supplied, the former before the facility is handed over to RAE, the latter before the expiration of the guarantee period. The Contract will be deemed to be incomplete while these items remain outstanding.

6.9 The Contractor shall provide three copies of Parts List for all spares which he recommends should be stocked on site to ensure continued and efficient operation of the plant. Spare parts of proprietary manufacture shall be clearly identifiable.

6.10 The Contractor will be required to guarantee the complete installation against defects in materials, design and workmanship for a period of twelve months from the date of acceptance by the RAE RO.

6.11 Before despatch to RAE all equipment shall be packed to give adequate protection against damage during transit from mal-handling and inclement weather. All packages shall be clearly marked with the applicable RAE Order Number.

6.12 Delivery is the responsibility of the Contractor.

6.13 Tenderers must submit sufficient information, including diagrams, illustrations, drawings or other data to permit a full technical appraisal of their offer to be made.

6.14 Tenderers may submit quotations and proposals incorporating their nearest design for standard equipment to meet the requirements of this specification but Section 6.13 must be met.

Alternative proposals may be put forward for consideration but full details must be given in the Tender so that deviations from the requirement of this specification are made known and clearly defined.

7. TECHNICAL ENQUIRIES

7.1 Tenderers may visit the proposed site to take measurements or carry out a survey to enable drawings and a quotation to be prepared.

7.2 All technical enquiries associated with this specification should be addressed to the Director, Royal Aircraft Establishment, Farnborough, Hants., for the attention of Technical Facilities Department.
RANGE OF PEAK 'g' / FREQUENCY.

PEAK 'g' PATTERN.
'PAYLOAD' CG. POSITIONS

TOP FACE OF WORKING PLATFORM

C.G.

ELEVATION

24"

WORKING PLATFORM POSITIONED CENTRALLY ABOUT VERTICAL MOVEMENT.

CENTRE OF VERTICAL MOVEMENT

12" O.A.

PLAN VIEW

'PAYLOAD' MOVEMENTS CG. POSITIONS

ELEVATION VIEW

10"

CENTRE POSITION OF HORIZONTAL AND VERTICAL MOVEMENT.

EXTREMITY OF VERTICAL MOVEMENT

3-35

APPENDIX II

EXTREMITY OF COMBINED MOVEMENT.
FIG 1: SCHEMATIC DIAGRAM OF VIBRATION RIG.
Fig 2: General arrangement of vibration system with console.
Fig 3: Vertical actuators with Test Load.
Fig 4: Horizontal actuator, showing six hydraulic valves.
Fig 5: Distortion values for vertical excitation at 5 Hz
(3 valves per actuator)
Fig 6: Distortion values for vertical excitation at 5 Hz
(1 valve per actuator)
Fig 7: Distortion values for horizontal excitation at 5 Hz
(1 valve)
The governing equation is,

\[
\frac{V_{\text{out}}}{V_{\text{in}}} = \frac{173 \left( 1 + \frac{j f}{2.12} \right) \left( 1 + \frac{j f}{2.84} \right)}{(1 + j f/0.15)(1 + j f/0.15)} \cdot \frac{1}{(1 + j f/48)}
\]

Fig 3: Horizontal actuator compensation network for acceleration inputs.
Fig. 9: Horizontal actuator-final compensated frequency response.
a) Contact with end stops.

b) Emergency stop.

c) Feedback failure.

Fig 10: Safety tests.
CHAPTER FOUR

A new method for the dynamic calibration of accelerometers

for accelerations up to \( \pm 1 \) g
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INTRODUCTION

The ranges of acceleration amplitudes and frequencies which chiefly affect man's performance and comfort under whole body vibration are normally up to ±1 g and from about 0.20 to 50 Hz. To measure such vibrations accurately, the appropriate instrumentation needs to be calibrated over the range of amplitude and frequencies for which it is being used. Most present day accelerometers, which are small enough to be conveniently attached to the man, have ranges of about ±20 g and a flat frequency response (according to the manufacturer's literature) up to about 80 Hz, certainly in excess of the required frequency range. Calibration facilities available in the laboratory are usually a static orientation of the unit through 360° to give +1 g, 0 g, and -1 g (or any value within this range) for accelerometers which respond to steady acceleration, that is piezo-resistive devices; and a small vibrating table. Vibrating tables are not satisfactory for the low frequency end of the above range, for reasons which will be covered in section 2. Thus there is a gap in the frequency range over which calibration is difficult if not impossible. This gap is not so important for piezo-resistive devices as their critical frequency area is at high frequencies (100 Hz plus) and a reasonable calibration can be performed by a dc rotation and a vibrating table at several frequencies above about 30 Hz, where a lot of the problems of using a vibrator have disappeared — although one is still left with the problem of measuring an extremely small displacement (see section 2.1). Then by assuming the device has a conventional first order response, if the response at dc and 30 Hz have the same magnitude, then the response curve is flat between dc and 30 Hz. Other types of accelerometer present a larger problem in that they have no dc response and their critical frequencies are very low, for example — piezo-electric devices, whose response falls off below about 5 Hz.
However, as far as this work is concerned, the accelerometers used have been piezo-resistive ones (type BLA 2 made by Ether Engineering Co. Ltd). The tests described will therefore use such a device, so that the assumption above - of a flat response between dc and 30 Hz - can be verified.

A method of dynamic calibration has therefore been devised which gives maximum accelerations of ±1 g, with the sinusoidal frequency of application selectable to cover the required range of 0.20 to 50 Hz. This enables calibrations of all types of accelerometers to be undertaken, checking the above assumption for piezo-resistive types and the very low frequency area for piezo-electric types.
PRESENT METHODS OF CALIBRATION

2.1 Dynamic methods

(a) Vibrating table and displacement measurement

This is the most common form of calibration. The accelerometer is mounted on a small, usually electrodynamic, vibrator, and the electrical output developed by the transducer is compared with the displacement of the vibrator, and the acceleration calculated assuming sinusoidal motion. By using this method the complete electronic set-up to be used by the experimenter may be calibrated. However at low frequencies most vibrators generate a large amount of harmonic distortion, and as we are measuring the displacement of the vibrator, the harmonic distortion, when applied to the acceleration output can be very large. For example, if we require \( \pm 0.10 \text{ g at 5 Hz} \), then this involves a displacement of only \( \pm 0.04 \text{ in} \). If the displacement trace shows 10% distortion on the third harmonic (in practice a low distortion) this corresponds to an amplitude of \( \pm 0.004 \text{ in} \), but this is equivalent to an acceleration distortion of 90% or \( \pm 0.09 \text{ g} \). Thus very large errors can be involved in this method of calibration.

Displacement also presents a problem for vibrators at the extreme ends of our frequency range. At 1 Hz an acceleration of \( \pm 1.0 \text{ g} \), needs \( \pm 10 \text{ in} \) and most electrodynamic vibrators have a limit of less than \( \pm 2 \text{ in} \), and at 50 Hz our acceleration implies a displacement of \( \pm 0.004 \text{ in} \). This is not a difficult task with micrometer, travelling microscope, etc., but as we wish to know what acceleration distortion is present, then dynamic measurements of 1% of this (0.00004 in) are required. Clearly if accelerations less than \( \pm 1 \text{ g} \) are required then the measurement of any displacement becomes impracticable.

(b) Comparison with sub-standard

Instead of measuring the displacement amplitude and assuming an acceleration, the transducer output may be compared with the output of a sub-standard accelerometer, also mounted on the vibration table. However the manner in which the sub-standard was originally calibrated was probably using a vibration table and hence the drawbacks outlined in (a) above still hold.
Also comparison with a sub-standard, applying particularly to piezo-electric types of accelerometers, usually involves the return of the accelerometer to the manufacturer who compares its performance with a sub-standard kept specifically for this purpose. The method calibrates the transducer alone and presupposes that any electronic equipment used with the transducer (amplifiers, filters etc.), is working to specification and will not overload the transducer.

2.2 Static methods

(a) Whirling arm

For this method, the transducer is attached to a whirling arm or centrifuge in such a way that the active axis of the accelerometer coincides with the centrifugal force axis of the centrifuge. Thus the transducer is subjected to an acceleration of $\omega^2 r$, approximately $40 f^2 r$, where $r$ is the distance from the axis of rotation and $f$ is the frequency of rotation (and $\omega = 2\pi f = \text{circular frequency}$). The obvious drawback to this method is that the calibration is static and does not check transient response behaviour of the accelerometer.

(b) Tilting table

Another method of static calibration, for acceleration below $1 g$, is the tilting table. The table surface can be rotated through $360^\circ$ about a horizontal axis. The accelerometer is attached to the table, and one can thus plot values of the output of the transducer versus $g \cos \theta$, where $\theta$ is the angle between the horizontal and the table and $g$ is the earth's gravitational field. Since such a table is easily portable this method allows the complete system to be calibrated in situ, but again only statically.
3 NEW METHOD

3.1 General

The method of calibration described in this Chapter basically takes the tilting table of 2.2(b) and rotates it at a constant rotational velocity, thus subjecting the accelerometer to \( g \times \sin \omega t \). If the rotational velocity is not constant then unsteady acceleration terms dependent on the rate of change or rotational velocity \( (\text{d}w/\text{d}t) \) will appear and the accelerometer output will no longer be \( g \times \sin \omega t \). The accelerometer is mounted on the shaft of an electric motor, as near as possible to the axis of rotation, and so the axis of rotation is at right angles to the active axis of the accelerometer (see Fig.1). Lower acceleration amplitudes are obtained if the motor or accelerometer is tilted so that the axis of rotation is not horizontal. The limiting case is clearly when the motor or accelerometer axis is vertical when no acceleration is applied. The frequency of the acceleration is only limited by the capability of the motor and the offset between the motor's axis of rotation and the accelerometers seismic axis (see section 3.2). Ultimately the complete system would be made portable, so calibrations can be performed on site.

3.2 Theory

Consider an accelerometer mounted on the shaft of a motor with its active axis at right angles to the axis of rotation of the motor. Let the accelerometer also be displaced a distance \( r \) from the axis of rotation (see Fig.2a). We now have a situation where by rotating the motor by hand we can perform a static calibration of \( \pm g \). If however we power the motor to run at \( \omega \text{ rad/s} \) the accelerometer is still going to give an output of \( \pm g \), no longer statically but in the form of a sine wave at a frequency \( (\omega/2\pi) \) Hz. The effect of the radial displacement \( r \) (between axis of rotation and seismic axis) is to give a constant 'dc' centrifugal acceleration when the motor is running equal to \( \omega^2 r \).
In practice one would strive to mount the accelerometer so that the axis of rotation coincided with the seismic axis but if this were impossible then an arbitrary limit of, say, $\omega^2 r \leq 10\%$ of the specified range of the accelerometer would be set. Typical accelerometers in use by this department have a range of $\pm 20\ g$, so our limit would be set at $2\ g$; equating this to $\omega^2 r$ we arrive at the limiting equation $f^2 r = 20$. As an example, if $10\ Hz$ were the upper frequency of interest, then an offset of $2\ in$ could be allowed before the limit of $2\ g$ was exceeded. The magnitude of this offset can be calculated by measuring the 'dc bias' of the oscillation at any frequency and compared it with the zero volts datum of the dc calibration. Means may then be provided to position the accelerometer to minimize the $\omega^2 r$ term. For dynamic accelerations of less than $\pm 1\ g$, the motor assembly or accelerometer could be equipped with a gimbal type frame so that the axis might be aligned at an angle $\theta$ to the horizontal (see Fig. 2b). Then, instead of imparting $\pm 1\ g$ to the device, as the motor turns the acceleration range will be $\pm (\cos \theta) 1\ g$, e.g. for $\pm 0.50\ g$ an inclination of $60^\circ$ is required. The $\pm (\sin \theta) 1\ g$ component is assumed to act at right angles to the active axis, i.e. along the seismic axis, of the accelerometer and hence can be ignored (representative cross-coupling figures quoted from manufacturer's literature are about $2\%$). The centrifugal acceleration due to the radial offset $r$, as before, gives a constant dc bias of $\omega^2 r$.

In the first instance it is intended to calibrate miniature piezo-resistive accelerometers, such as the Ether type BLA2, using this method, but large accelerometers could be accommodated by mounting them on an end plate and using a more powerful motor.

3.3 Tests

In order to show that the method works, an arrangement was set up in the laboratory with an accelerometer attached to the shaft of a low speed motor with adhesive tape. No slip rings were available, and the rotation
was taken up by the long cable between the accelerometer and the amplifier, so that only a limited number of cycles was possible. A further test was conducted by tilting the motor from the horizontal to about $45^\circ$ (to theoretically give $\pm 0.70 \text{ g}$) and then to the vertical where we should have zero acceleration.
RESULTS

The results of the tests are shown in Fig. 3 and indicate that in the frequency range dc to 10 Hz (the maximum speed of the motor used) the output of the accelerometer was constant at ±1 g. The results of the second, or, inclination test, at 0.58 Hz are shown in Fig. 4. It can be seen that when the motor is inclined, the output from the accelerometer changes from ±1 g to ±0.70 g and then to zero. Also, there is no change in the zero base line of the oscillation indicating that the ±g sin θ component has little effect on the output.

The spikes and discontinuities present on some of the traces in Figs. 3 and 4 are probably due to transient changes in the rotational speed of the motor rather than any fault in the accelerometer.

Thus this brief experiment has demonstrated that the accelerometers used for tests described in the other Chapters, have a flat frequency response up to 10 Hz (and assumed up to 30 Hz) for amplitudes at and below ±1.0 g.
Fig. 1 Typical piezo-resistive accelerometer.
$\omega = \text{Rotational speed of the motor.}$

Earth's gravitational field (constant in strength and direction).

a) Motor at zero inclined angle.

b) Motor at angle $\theta$ ($0 \leq \theta \leq 90^\circ$)

Fig. 2 Schematic diagram of calibrator.
DC calibration $+1g, 0g, -1g$

$0.18 \text{ Hz } \pm 1g$

$0.32 \text{ Hz } \pm 1g$

$5.8 \text{ Hz } \pm 1g$

$9.0 \text{ Hz } \pm 1g$

$10 \text{ Hz } \pm 1g$

Note: All traces to same scale, so constant acceleration level for all frequencies shown

Fig. 3 Variation in accelerometer output for frequencies from 0 to 10 Hz.
Motor horizontal — output = ±1 g

45° — ±0.70 g (approx.)

90° inclination — output zero

45° — ±0.70 g (approx.)

180° — ±1 g

f = frequency, constant at 0.58 Hz

Fig. 4 Effect of motor inclination on accelerometer output
CHAPTER FIVE

Transmission of vertical vibrator to the shoulder of the seated man
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   2.2 The accelerometers and method of mounting
   2.3 Analysing equipment
   2.4 Calibration
   2.5 Subjects and posture
   2.6 Experimental method

3 RESULTS AND DISCUSSION
   3.1 Effect of posture
   3.2 Frequency response
   3.3 Effect of amplitude of acceleration

4 CONCLUSIONS

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INTRODUCTION

As has been described elsewhere, there are a large number of papers in the literature about the transmission of vibration to various parts of the seated human body (Refs. 1 and 2, for example), but since there are infinite numbers of subject postures and possible measuring points, as well as an almost infinite population of possible subjects, any information, even if only strictly applicable to the particular conditions in which it was obtained, adds something to the sum of knowledge.

This Chapter presents some results obtained in a brief run of tests using five subjects seated in approximately car driving position on a hard seat with a hard back. Accelerations were measured by accelerometers mounted on the seat and held onto the shoulder by means of a harness of wide dressmaker's elastic. Generally, the response would appear to be that of at least a second-order mechanical system with its first natural frequency near 5 Hz, and though there were wide variations between the subjects, results from individuals were consistent. The effect of leg position was also considerable, and there was evidence that the system was non-linear, with effects particularly noticeable near the frequencies where maximum transmission occurred.

The results should not be extrapolated to other conditions, but present a picture of the kind of transmission which may occur in practice, with hard seats.

The vibration of the shoulder may be important in affecting manual tasks, and is also one of the few positions on the body to which it is fairly easy to attach accelerometers since the bony structure of the acromion (the extremity of the shoulder blade) is close to the surface and easily located, and presents a near horizontal surface. The experiments utilized nominally sinusoidal vertical vibration, generally of 0.1g or 0.2g rms acceleration, though some additional tests were carried out at...
single frequencies with accelerations up to 0.4g rms. The range of frequency covered was from 1 to 30 Hz in most cases, but in a few cases tests were continued up to 50 Hz.
2 EXPERIMENTAL EQUIPMENT AND PROCEDURE

2.1 The vibrator

All tests were performed while subjects were seated on the small two-axis vibration rig. This consists of a tubular magnesium alloy seat frame with a plywood backrest and rigid wooden seat, the whole of which can be vibrated simultaneously vertically and laterally by two hydraulic jacks, though in these tests only vertical vibration was used. The rig has been described in detail in Ref. 3, and a photograph is presented here as Fig. 1, and a schematic diagram showing dimensions in Fig. 2a.

2.2 The accelerometers and method of mounting

The miniature accelerometers (6 mm x 8 mm x 14 mm) used were of the strain gauged cantilever type, model number BLA2 made by the Ether Engineering Co. Ltd. The strain gauges formed one half of a bridge network, the output of which was fed into an SEL ac carrier amplifier, which contained the other half of the bridge together with a balancing potentiometer which was used as a calibration facility (see section 3). The demodulated output from the amplifier was then fed into a filter (with roll-off frequency of 100 Hz, thus cutting out the accelerometer resonance at 180 Hz) and then into the analysing equipment (see 2.3).

An accelerometer was rigidly mounted in a duraluminium bar (250 mm x 50 mm x 13 mm), and bolted to the underside of the seat so that the accelerometer and measured vertical ('heave') acceleration (±g_z).

The shoulder accelerometer, to measure heave acceleration, was attached by fitting the subject with a harness consisting of two loops: (see Fig. 2b) first a torso hoop under the armpits and second, a vertical loop folded round the first loop and passing over the top of the shoulder. The accelerometer was attached to the second loop on top of the shoulder and held down by it. The loops were made of 1 inch wide dress-maker's elastic, enabling the harness to be tensioned to ensure that the accelerometer would
not lift under the vibration. The position was adjusted so that it measured vertical acceleration when the subject was seated for the test (see 3.4).

2.3 Analysing equipment

The vibration produced by the rig when powered by a sine wave oscillator was found to contain a high percentage of harmonic distortion. In order to ensure that the amplitude ratios calculated were accurate, the outputs of the accelerometers were fed into a Solartron Digital Transfer Function Analyser which separated the fundamentals from the distorted wave forms. Thus all amplitude ratios included are for the ratio of the fundamentals at the frequency quoted.

Table 2 shows typical distortion figures for the rig for an input acceleration level of 0.2g rms. Corresponding figures for an input level of 0.1g rms would be about 15% higher than those quoted.

2.4 Calibration

Before any testing or calibration started, the accelerometers, amplifiers and filters were allowed to warm up for at least half an hour. The amplifier gains were then selected to give the appropriate amplitude on the analyser and the outputs balanced for zero voltage output for a horizontal attitude. The accelerometers were then inverted to give -1 g, for a check calibration on site.

The settings of the balance potentiometer, for zero output voltage, on the face of the amplifiers for conditions of +1 g, 0 g and -1 g were also carefully noted, and these results were used to check their orientation on the seat and shoulder. The dynamic calibration of the accelerometers used is covered in Chapter 4.

2.5 Subjects and posture

Five subjects were used, four men and one woman. Table 1 gives their basic anthropometric details. These can be summed up as, the men were of average build, aged from 26 to 52 and reasonably fit (able to perform a
vibration experiment), the woman, aged 26 and of small build. All subjects were volunteers from Human Engineering Division.

The subjects were instructed to sit on the seat in an erect posture. It has been found that the best method of attaining a reasonable posture is to tell the subjects to slump completely and then to sit rigidly upright. Having demonstrated the limits of posture each subject can then better appreciate what is required in asking for a normal erect posture. In all the tests the subject rested his hands on his thighs and he was instructed to look straight ahead. No task, mental or manual, was given. Each subject was instructed to sit with his knees flexed and feet resting on a specific part of the foot rest. This resulted in a thigh/shin angle of about 110° with the thigh angle nearly horizontal. A sketch showing the dimensions of the rig is given in Fig.2.

2.6 Experimental method

Each subject was instructed to maintain a normal erect posture with feet on the 'rudder bars' and knees bent. No seat harness was used. When he was in a comfortable position the shoulder accelerometer was adjusted to measure vertically (see section 2.4).

A series of tests were carried out with the subject's knees flexed, with accelerometer levels of 0.1g and 0.2g rms normally over the frequency range 1-30 Hz, but going up to 50 Hz with some subjects. A similar test was also carried out on one subject (No.1) with his legs straight and knees 'locked'. The variation of transmission with frequency (amplitude and phase) for each subject are shown in Fig.3-5, and the average response is shown in Fig.6.

Following each test, and after inspecting the results to locate peak response, a number of measurements with input acceleration amplitudes from 0.05g to 0.4g rms were made at a fixed frequency near the peak transmission. For one subject (No.4) a similar test was also carried out at a frequency well away from the peak transmissibility. The object of these tests was to explore the degree of linearity of the response, and the results are shown as polar plots in Figs.8 and 9.
3 RESULTS AND DISCUSSION

3.1 Effect of posture

The importance of leg position in determining transmission from seat to shoulder is clearly illustrated in Fig.3, where the response for the same subject is shown in exactly the same conditions, except that in one his knees were flexed, and in the other they were locked. Comparing the transmission, it is seen that for frequencies up to about 5 Hz, at least for this man, there is very little difference between the effect of the two attitudes; from about 5 Hz to 10 Hz, transmission is reduced with the knees flexed; and above 10 Hz, transmission is reduced with knees locked. It is obvious that in any practical situation, a man will try to accommodate himself to reduce transmission according to the predominant frequencies of vibration to which he is exposed, and which parts of the body he most wishes to protect from vibration. This particular subject, for instance, might manage to produce a response like that shown in Fig.7 if he were asked to adjust his posture for each frequency with the criterion of producing the minimum shoulder movement. How far a man can in fact maintain the same mechanical characteristics throughout a test in which he is subjected to different kinds of vibration is another question of interest. Such a gross feature as the knees locked may be determined, but it seems very likely that minor adjustments will take place, even if the subject firmly intends that they should not. This may possibly account for the difficulties which are experienced in attempting to determine human mechanical characteristics from vibration experiments.

3.2 Frequency response

3.2.1 At 0.1g rms

The results shown in Fig.3-5 for the individual subjects, indicate that the general form of the response was the same in each case. It can be said that, with the knees flexed and an input of 0.1g rms, each subject
showed a peak transmission with an amplification of about 2 in the region of 5 Hz, a dip in the neighbourhood of 8-10 Hz, a second peak at about three times the frequency of the first, thereafter a rapid drop in transmission, though some subjects (2 and 3) showed signs of another peak in the neighbourhood of 45 Hz. In each case, a maximum phase lag, generally somewhere about 90° occurred near the first amplitude dip, followed by a minimum at about the frequency of the second peak. As will be seen from Fig.6, these characteristics are preserved in the average, at least up to 30 Hz (above this frequency there were insufficient data to average).

There were, however, very wide individual variations, particularly in the amplitude of the second peak, which in some subjects (2 and 3), was only a third as great as the first peak, while in others (4 and 5) it was nearly as large as the first peak.

3.2.2 At 0.2g rms

Looking at Figs.3-5, it will be seen that the general form of the response at the lower level is repeated, but, in general, the peaks and dips in both amplitude and phase are at slightly lower (about 20% down) frequencies, and generally - though not in every case - with rather lower amplitude ratios, so that the general result is somewhat flatter curves. This frequency shift is preserved in the averages in Fig.6, but the peaks show little difference in amplitude. The phase shift appears to be roughly the same as for the lower acceleration.

3.3 Effect of amplitude of acceleration

As noted already in 3.2.2, the effect of doubling the acceleration (from 0.1g to 0.2g rms) was to shift the response curves downwards by about 20% in frequency and generally to flatten out the peaks and dips slightly. This effect indicates non-linearity in the system, since in a linear system the response is independent of amplitude. The observed shift to low frequencies agrees with what has been found by some other experimenters\(^4,5\) though not generally on this particular part of the body. It indicates
reduced spring stiffness with increasing deflection, whereas most experiments have found increased stiffness. The further data obtained by measurements at different acceleration levels at fixed frequencies near the first peak for each subject, and shown in polar plots in Figs. 8a, b and c, and Figs. 9a and c, confirms the idea of decreasing stiffness with amplitude. These curves also demonstrate how difficult it is to obtain consistent results near resonances. On the other hand, Fig. 9b, which gives the results for a similar range of accelerations well away from a peak, shows very little evidence of non-linearity.
CONCLUSIONS

It must be emphasised that the results given in this experiment relate to a particular posture in a particular seat, with inputs to the body not only through the seat but also through the back and through the feet. Since these inputs were not measured, we do not know how much the specific effects were due to them. On the other hand, there is little doubt that the sort of effects described could occur in practice.

Detailed conclusions cannot therefore be drawn, but it is possible to make some general deductions.

1. All the subjects when seated with knees flexed exhibited the characteristics of at least a second order system, with peak transmissions in the neighbourhood of 5 and 15 Hz, but the amplitude of the peaks varied widely between subjects.

2. The effect of locking the knees (in one subject only) was to bring the two peaks closer together, so that in effect they merged into one peak.

3. All the subjects showed marked non-linearity of response, producing peaks at rather lower frequencies as the amplitude increases. Changes of response with amplitude were especially marked near the peaks. How far these effects were due to insensible changes of posture and muscle tension and how far to genuine non-linearities in the mechanics of the body has not been explored.

In future work it is intended to attempt to correlate the transmission of vibration to different parts of the body, measuring as many as possible of the inputs and outputs. In the meanwhile, the results presented here can only be taken as an indication of the kind of response which may occur in vehicle seats, and of the dangers of generalization and extrapolation.
Table 1
SUBJECT DATA

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<th>Sex</th>
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<th>Height (m)</th>
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<td>26</td>
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Table 2
DISTORTION VALUES FOR VIBRATION RIG

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<th>4</th>
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<th>8</th>
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<td>% distortion</td>
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<td>30</td>
<td>26</td>
<td>30</td>
<td>41</td>
<td>26</td>
<td>35</td>
<td>53</td>
<td>57</td>
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* The distortion figures are calculated by taking the square root of the sum of the squares of the distortion percentage values up to the fourth harmonic.
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<th>Title, etc.</th>
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<tr>
<td>1</td>
<td>S. Lippert (ED)</td>
<td>Human Vibration Research, Pergamon Press (1963)</td>
</tr>
<tr>
<td>2</td>
<td>G.F. Rowlands</td>
<td>The transmission of harmonically distorted low frequency vibration to the head of the seated man. RAE Technical Report 72080 (1972)</td>
</tr>
<tr>
<td>3</td>
<td>E.J. Lovesey</td>
<td>Some effects of dual-axis heave and sway vibration upon compensatory tracking. RAE Technical Memorandum EP 484</td>
</tr>
</tbody>
</table>
The cushion is 410 mm deep, 450 mm wide and is inclined 6° to the horizontal and is 50 mm thick at the front edge.

**Fig. 2a** Dimensions of rig

**Fig. 2b** Accelerometer harness
Frequency (Hz)

Amplitude ratio

Phase lag

Subject 1 - knees bent

Subject 1 - knees locked
FIG. 4 Vibrational transmission. Subjects 2 and 3.

Subject 2 – knees bent

Subject 3 – knees bent
FIG. 5 Vibration transmission, Subjects 4 and 5.
Fig. 6 Average vibration transmission

Fig. 7 Optimized transmission for subject 1
Fig. 8 Variation of transmission with amplitude near resonance.

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<th>m/s² rms</th>
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Variation of transmission with amplitude, near and away from resonance.
CHAPTER SIX

Seating Dynamics
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<td>Second seat</td>
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INTRODUCTION

Any investigation which attempts to quantify the manner in which vibration, present on the floor or structure of a vehicle or building, is modified to arrive at specific parts of the human body, must include discussion of, and possible measurement on, the vibration transmission characteristics of seats and their associated cushions. The major topic of this thesis is the vibration characteristics of the body itself, defined as frequency response functions between the feet, or seat of the pants, and other parts of the body and clearly another separate thesis could be written on seating dynamics. This Chapter will therefore merely report some ad hoc work done by the author on the frequency responses of aircraft seats. It will be appreciated that the dynamic response of seats and cushions is related to the dynamics of the man sitting on the seat, i.e. the man's impedance loads the seat impedance, (this part is covered in more detail in Appendix A) so that seating dynamics and body dynamics are both necessary factors in defining the acceleration levels present on the head etc., when the subject is seated on a "soft" seat. However the theoretical interaction of such systems will not be fully pursued as of necessity each seating arrangement will give a different response and therefore each seat will have to be tested on its own. This of course is one of the reasons why a hard seat was selected for the main body of the experiments detailed. A lot of the early reports on body response failed to mention the type of seat used for their subjects. If soft cushions were used then the results found may well have been a function of the cushion's response rather than the subjects. Alternatively had a hard seat been used, even with a contoured seat pan, then the discomfort factor, especially when subjected to many minutes of sinusoidal input, may again have influenced the results.

The present investigation was initiated when the Department of Trade and Industry visited the RAE, having heard that the use of an RAE dry-fluid
('bird-seed') cushion could isolate the pilot from any vibration causing eye-ball resonance. Inspection by the author of the soft cushion being used on the pilot's seat indicated that the problem would not be with eye-ball resonance which is at about 20 Hz, but more likely with whole-body/cushion resonance around 5 Hz, which could amplify any vibration present at the floor of the cockpit arising from the first two aircraft fuselage bending modes (about 2.3 and 4.5 Hz). It was therefore decided that, with the agreement and help of aircraft manufacturers and DTI, an investigation should be made into the frequency response of pilots' seats as used in the aircraft. These seats would be tested when fitted with the normal soft cushion and also when fitted with the bird-seed cushion, or a personal survival pack (PSP), with typical subjects seated on them.

The tests were carried out using sinusoidal excitation in the vertical and lateral directions, and also using the semi-random vertical acceleration recorded during take-off on various runways. Runway input tests were necessary in case the response of the man/cushion/seat system should prove to be non-linear, in which case, the results of any sinusoidal tests done at amplitudes around ±0.2 g, could not be correlated with performance of the system in real life where non steady state accelerations greater than ±1.0 g may well be encountered. Also the loading by the main is not passive, and conscious or subconscious posture changes could alter the biomechanics of the man/cushion combination, the degree of adaptation differing between the repetitive and random vibration inputs.

Conclusions are drawn regarding the frequency responses of the various cushion/seat systems and recommendations made on how to minimize the vibration reaching the crew as far as the seat/cushion system is concerned, in the critical frequency region 1 to 6 Hz.

A second seat was tested which had been designed from the outset as being the ideal from an anthropometric point of view and RAE was requested to make an engineering appraisal of the seat and it was decided that a survey of the vibration characteristics of the seat within the frequency
range 1.0-30 Hz would be included. The seat was available for vibration testing for a period of only two weeks. This permitted a general investigation to be carried out but did not allow sufficient time for any detailed modal analysis. Some general response curves are given at the end of the Chapter.

It is necessary when studying the dynamics of any seat to have a man seated in it, since it is at present impossible to simulate all the dynamic characteristics of the human body which may affect the response of the seat. Three different subjects were used for these tests and the seat was vibrated vertically and laterally using various seat configurations. The results show a broadly similar response to other aircrew seats examined although the resonance peaks appear to be less pronounced. The seat is in general considered to have good vibration characteristics by present standards.
**EQUIPMENT**

2.1 **Seats**

2.1.1 **First seat and cushions**

The seat used was a production aircraft pilot's seat, with normal fittings including the soft back-pack and the harness. The seats were mounted on their correct rails on the vibrator. In order that the tolerances for fitting the seat and rails should be as in the aircraft, the aircraft firm's personnel undertook to fit the seat onto its rails, and then RAE personnel fitted the complete system onto the vibrator. A Dexion frame was provided in the rudder-bar position, to support the subjects' feet. Due to the short period of time that it was possible to have the seats at the RAE, it was not possible to provide a control stick so each subject was instructed to keep his hands on his thighs throughout the tests. Before testing began a test pilot visited the RAE and demonstrated the normal seat-back and seat-pan angles and normal leg position, which were then kept constant for the duration of the tests, except for those tests designed to explore the effects of varying these parameters. Fig.2 shows the seat with normal cushion on the vibrator.

The cushions used were:

(i) the normal soft cushion usually used with the seat;
(ii) the personal survival pack (PSP) as fitted to the seat;
(iii) a dry fluid or 'bird-seed' cushion. This is an innovation which is basically a soft cover containing about 8000 \( \times 10 \text{mm} \) diameter hollow plastic balls, on a thin layer of soft foam rubber. The basic idea is that when a subject sits on such a cushion the balls will 'flow' to accommodate his shape and, even more important, if he changes his position on the cushion, the balls will flow again to provide him with a comfortable contour. It must be stressed that the bird-seed cushion used in these experiments had not been designed specifically to suit this seat. More time than was available would have been needed to
determine the optimum packing density of the balls and the thickness and consistency of the foam rubber used.

2.1.2 Second seat

The seat tested is illustrated in Fig. 3. The back comprises shoulder and lumbar pads which can together be raked by increments from vertical $+5^\circ$ to vertical $+35^\circ$. The height of these pads can also be adjusted together relative to the seat pan. The lumbar pad has an in and out movement relative to the back at any angle of rake. An adjustment of the contoured seat pan and cushion height is provided. The thigh-supporting pads are sprung and also adjustable in height. The arm rests are adjustable through an arc of $24^\circ$ which provides height adjustment at their forward ends. They may also be stowed behind the seat back pads when not required.

The lower part of the seat structure is divided into two parts; one unit including the seat pan and height adjustment mechanism, and the base unit incorporating the rail rollers and the fore and aft adjustment lever. This lever has three positions; seat free to travel on the rails, seat located in position and seat rollers clamped to the rails. The clamped position is designed to prevent rattling of the seat on its rails. Interchangeable versions of the base unit are available to suit the rail configurations of different aircraft.

A seat mounting base complete with rails and rudder pedals was supplied by the manufacturers. This comprised a substantial wooden board, stiffened along its underside, to which were attached a pair of seat rails and a pair of aircraft rudder pedals. This complete base was fitted to the vibration rig platform using two lengths of angle iron as clamping bars one fitted just behind the seat and one in front.

2.2 Vibrator (see also Chapter 3)

The rig, manufactured by Servotest Ltd., consists basically of a flat aluminium table ($1.8 \times 1.2$ m), weighing about 250 kg, supported by three
hydraulic actuators. Two jacks support the table in the vertical axis and the third is attached laterally to the side of the table in the same plane as the vertical jacks. The acceleration and frequency ranges possible from the rig, in both axes, separately or simultaneously, are 0 to ±2 g, and 0.5 to 50 Hz, with a working load of around 300 kg.

The rig is equipped with its own oscillators connected as displacement inputs. The frequency applied to the two axes can be the same, (with adjustable phase angle) or different, in the range 0.5 to 50 Hz. The control console has input sockets where external displacement or acceleration inputs can be connected to the rig. The acceleration input socket is connected internally to the displacement socket via two integrators. These integrators have a feedback network which compensates for the frequency response of the rig, allowing acceleration records from various forms of transports, in our case the acceleration history of an aircraft taking off, to be reproduced on the table. For the tests described, the internal oscillators were used to power the rig for the sinusoidal tests and a tape recorder (see section 2.4.1) was connected to the external acceleration input socket for the runway tests.

Two feedback loops are provided to purify the waveform and provide servo-control differential pressure across the piston and displacement of each actuator. The resulting harmonic distortion at the acceleration levels used was about 20%.

2.3 Instrumentation

2.3.1 The transducers, filters, amplifiers and tape recorder

Acceleration was measured at three positions, (i) on the table, (ii) under the cushion, on top of the seat-pan and (iii) on top of the cushion. The transducers used were Ether type BLA2 (semi-conductor bonded resistance strain gauge) accelerometers mounted in light alloy bars (250 mm x 50 mm x 12 mm) in the appropriate directions. One bar was fixed
to the mounting plate, the second to the seat-pan and the third to the top of the cushion, positioned under the ischial tuberosities, so as to measure the acceleration input to the body. This is a simplification, all the accelerations across the face of the cushion ought to be included. This is obviously impossible to do in practice and it is considered that the best compromise is a seat-bar of small area (so that the pressure distribution across the cushion is not significantly altered by it), placed at the region of the likely major vibration input to the skeletal system.

The strain gauges on each accelerometer formed one half of a bridge network, which was connected into an SEL ac carrier amplifier, containing the other half of the bridge. The demodulated output from the amplifier was fed into a low-pass filter, set to give a cut-off frequency of 100 Hz, thus filtering out spurious signals corresponding to the accelerometer's mechanical resonance at about 180 Hz. The filtered signal was fed into a tape recorder or analysing equipment depending on the test. The tape recorder used was a four track FM Tannberg type 100. The fourth track has a voice interrupt facility which was used to record test details.

2.3.2 Pre-test calibration

The accelerometers were calibrated on site in two ways. First they were rotated about their axes to give a dc calibration equivalent to ±1 g (= 9.81 m/s²). Then as an additional check, all three systems were mounted close together so that they all received the same vibration, and their outputs could be compared in amplitude and phase for various frequencies. The frequency response of the transducers were checked by using an accelerometer permanently fitted on the rig as a standard. The low frequency response of the transducers has already been checked and found to be satisfactory (see Chapter 4).

During the tests the dc capability of the accelerometers was utilized to ensure that the seat bar measured in the true vertical sense. The bar
was placed on a known horizontal surface and the output voltage from the amplifiers adjusted to give zero volts. Then when the subject sat on the bar any change in the output voltage implied an angle to the vertical of the bar and the bar could then be moved to give zero volts output. This facility implies that the accelerometers can be used as electronic spirit levels.

2.3.3 Sinusoidal excitation analysis equipment

For the sinusoidal tests, the output from the accelerometers was fed into a Solartron Digital Transfer Function Analyser (DTFA) which extracted the fundamental amplitude from the distorted waveform from the vibrator. The output from each transducer was compared with the oscillator output driving the vibrating table, in amplitude and phase, so that the fundamentals of the outputs of the transducers could be compared with each other.

2.3.4 Random (runway) excitation analysis equipment

For the runway tests, the outputs from the transducers were recorded on the FM tape recorder and subsequently analysed using the Fast Fourier Transform facility of a Hewlett-Packard 5451A Fourier Analyser. This instrument can break down any input that varies with time and show the amplitude of the component frequencies that make up the time function. Then by dividing the appropriate amplitudes for various frequencies the computer can generate a frequency response curve. The equipment is fitted with its own analogue-to-digital converters, so the analogue outputs from the accelerometers can be fed directly into it.

The accelerations at the vibrator floor, during the simulated vibration pattern at take-off were also analysed by recording the total run filtered through octave filters centred at 1, 2, 4, 8 and 16 Hz to show the time history of the acceleration present in the set octave bands.
3. **TEST PROCEDURES**

The subject sat in the seat and was strapped in, care being taken to ensure that the accelerometer bar was under the ischial tuberosities. The output from the accelerometers was checked to make sure that the seat-bar was measuring true vertical acceleration (see section 2.3.2). If it was the subject's first test, the safety and emergency procedures were explained to him. Throughout the tests it was emphasised that the subjects were volunteers and that they had the right to stop the experiment at any stage if they felt apprehension or physical discomfort, and, if they wished, without giving a reason. The object of the particular experiment was then explained to the subject and the test began. The sinusoidal tests lasted about 20 minutes for each condition, and the runway tests lasted about 50 seconds for each take-off. In general no subject did more than two sinusoidal tests at a lower level or one test at a higher level in any one day.

The sinusoidal tests investigated the effects of harness, subject weight, seat height and cushion variety.

In order to validate the results on the first seat of the sinusoidal tests described, acceleration records taken from actual aircraft take-offs were played into the vibrator so that the vertical acceleration of the vibrator table was similar to that measured in the aircraft.

Two subjects were used. Each subject did take-offs from three 'airfields'. Three cushions were used, the normal cushion, the PSP and the RAE bird-seed cushion. Thus each subject was exposed to twelve take-offs. Measurements were again taken at the floor, seat-pan, and top of cushion.

The test procedure for the second seat closely followed that for the first seat, except that only sinusoidal excitation was used.
4 ANALYSIS THEORY

4.1 Sinusoidal excitation

As explained in section 2.3.3, the acceleration output from the vibrator in response to a pure sine wave driving signal is composed of a fundamental signal plus harmonic distortion, the distortion being due mainly to the non-linearity of the hydraulic system. As a result, all the accelerometer signals, floor, seat-pan and cushion, contained unwanted harmonics. These unwanted signals were eliminated by feeding a reference signal from the oscillator driving the rig into a Solartron Digital Transfer Function Analyser, which triggers a crystal oscillator controlling a digital filter. Then, by passing each transducer signal through this filter, it was possible to find the absolute voltage level of the fundamental frequency of the signal and its phase with respect to the reference signal. Thus by comparing any two accelerometer signals, it was possible to find their complex ratio (amplitude ratio and phase angle).

4.2 Random (runway) excitation

4.2.1 Analysis of input signals

Two forms of analysis were performed on the runway signal. First, in order to find the dominant frequencies in the signal and the manner in which their amplitude varied with time during the take-off, the cockpit floor acceleration signals were passed through a set of octave-band filters with mid-band frequencies set at 1, 2, 4, 8 and 16 Hz. With the bandwidth of these filters set for one octave there was no attenuation of the signals across the filters, within the selected octave.

Secondly in order to find their linear spectra the signals were analysed using the Hewlett Packard Fourier Analyser, described in section 2.3.4. Before calculating the linear spectrum, the signals were fed through a low-pass filter to eliminate aliasing errors.

4.2.2 Calculation of seat response

The cushion linear spectrum was produced as described in section 4.2.1. The cushion and input transforms were then divided in the computer and a
print-out of their ratio was produced. The input linear spectrum was also printed out. The frequencies in the input spectrum were found to be very discrete and thus at many points on the frequency scale there was little or no input information present whereby one could produce a reliable ratio. The computer having a dynamic range of greater than 60 dB (1000:1) will quite happily produce what is in reality noise over noise. It was therefore necessary to use the 'Mk.1 eyeballs computer' to assess whether the quotient was meaningful. A rough level of confidence was found to be 10% of maximum value at the predominant frequency, i.e. when the spectrum level fell below 10% of the value at the predominant frequency, the results of the analysis were ignored. It will be seen that a lot of points had in fact to be eliminated.

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5. RESULTS

5.1 First seat

It is not proposed to reproduce all the results that were drawn up for the Report on which this Chapter is based. Rather, typical graphs will be shown and general conclusions taken from the Report.

Fig. 4 shows the range of cushion results for vertical vibration input, covering all subjects for the cushion normally used with the seat. The amplitude ratio shown is across the cushion only. Fig. 5 shows a similar overview for the seat itself (seat-pan/floor), covering all subjects and all conditions. The results obtained from the three cushions used, under vertical excitation, are given in Fig. 6 and cover one subject only. Fig. 7 gives the responses, both for the cushions and the seat, for lateral excitation.

Figs. 8, 9 and 10 cover the response using, as an input, the vibration measured in the cockpit during take-off. Fig. 8 gives the octave band analysis of the vibrator and Fig. 9 its linear spectrum. Fig. 10 then shows the comparison between the results for random (take-off) vibration and sinusoidal vibration.

Table 2 gives a summary of the vertical frequency responses, for random and sinusoidal inputs.

5.2 Second seat

Again it is not proposed to go into any great detail regarding the results, but rather to give a general overview and to reproduce the general conclusions from the appropriate report.

Fig. 11 shows some typical results for vertical excitation of the seat covering the response across the seat and also across the cushion. Phase angle information is also included.

Fig. 12 shows the corresponding results for lateral excitation.
6 DISCUSSION OF RESULTS (FIRST SEAT)

6.1 Acceleration inputs

As mentioned in section 2.2, the accelerations measured on the table had a distortion content of only about 20%, so that for the repetitive tests the subjects could hardly detect that the motion was not a pure sinusoid. This distortion was eliminated during the analysis of all the recordings.

For the random (runway) tests, comparison of the accelerations as measured on the table, with the flight records of the accelerations as measured in the aircraft indicated a good reproduction except at the higher frequencies (> 20 Hz) where the vibration rig tended to amplify the signal. During the initial setting of the rig a test pilot pronounced the ride to be very realistic. Fig. 8 shows the time history of the major frequency bands for the take-off used. The graph shows that apart from near 1 Hz (the aircraft whole body mode) and about 2 to 2.5 Hz (aircraft first bending mode), there is little energy in any frequency regime. This is reflected in Fig. 9 which shows the computed linear spectra. The vertical axis on Fig. 9 corresponds to a linear unit of acceleration, the absolute value of which has little meaning as the computation procedure assumes that the acceleration history is a transient, and the linear spectrum values of a transient are a function of the width of the transient with respect to the computational time window width, as well as the height of the transient.

6.2 Vertical vibration

6.2.1 Transmission from floor to seat-pan

Fig. 5 indicates that the seat itself is basically rigid in the frequency range tested. However, between 2 and 10 Hz there is some slight amplification in the response giving a ratio of 1.05. This figure may appear insignificant, but it will help later to explain another factor (see section 6.2.2(e)). The phase lag between 2 and 10 Hz is reasonably constant at about 5°.
6.2.2 Transmission from seat-pan to cushion top and from floor to cushion top

Inspection of Fig. 5, 6a and 10a and Table 2, show that the normal seat and cushion, has a clearly defined resonance at about 4 Hz giving a peak of about 1.64 and a phase lag of about 30° in the cushion top/floor resonance. The curve also shows a factor of 1 (0° lag) at 1 Hz and a factor of 1.18 (3° lag) at 2.25 Hz. Variations in these responses due to subjects, seat parameters, cushions, etc. are covered in the following sections. In general the responses found show an amplitude ratio of 1 and a zero phase lag at 1 Hz, so particular attention will be centred on the 2.25 Hz critical frequency only.

(a) Effect of subjects

There is little difference between the subjects' results, the only noticeable difference being that one subject, who was appreciably heavier than the other subjects, produced a resonant peak at a slightly lower frequency and with a slightly lower amplitude ratio.

(b) Effect of varying sinusoidal acceleration amplitudes

Increasing the input level of vibration to ±0.40 g from ±0.20 g increases the amplitude ratio at resonance and increased the phase lag especially around the 5.5 Hz for two out of the three subjects. This could imply a degree of non-linearity of the seat or cushion, or subject, or (as is more likely) the subject could be adapting to the higher level, especially under the untypical sinusoidal excitation. This point will be further discussed in section 6.2.3, where the random tests, in which the acceleration levels are even higher, will be considered.

(c) Effect of runway inputs

Fig. 10b shows the cushion top/floor curves for the runway input for subject (1) and the three types of cushion used. The curves show that the normal cushion again gives a resonance at about 4 Hz, but this time with a peak amplitude ratio of 1.55, compared with 1.68 for the sinusoidal input (see Table 2 and Fig. 10). The phase lag at resonance is again about 25°-30°.
(d) **Effect of harness and seat parameters**

Comparison between results for subject (1) indicated that slackening the harness had little effect on the response curves except that the curve for no harness shifted towards the harnessed $\pm 0.40$ g curve, whilst for another subject, varying seat height and tilt parameters made little difference.

(e) **Effect of different cushions, during sinusoidal tests**

The difference in the response of various cushions is covered in Figs. 6a and 6b. Fig. 6b shows that with the RAE cushion or PSP the cushion top/floor results no longer show a marked resonance at 4 Hz. At 4 Hz, the bird-seed cushion has an amplitude ratio of 1.05 with a phase lag of $5^\circ$. If one takes into account the seat-pan/floor response shown in Fig. 5 (and discussed in section 6.2.1) then clearly the RAE cushion has a flat 1:1 response with zero phase lag. Fig. 6b also shows that the PSP amplitude ratio is considerably lower and the phase lag $10^\circ$ less for the frequency range 3 to 5 Hz than that for the normal cushion. However, at 2.25 Hz the difference is between 1.18 and 1.10 compared with 1 for the RAE cushion. Fig. 6b also indicates what might be a possible disadvantage in using the RAE cushion, and to a lesser degree the PSP. At frequencies above about 10 Hz the normal cushion begins to attenuate the input vibration (the PSP also has this tendency although it shows a peak again at about 18 Hz), the RAE cushion however has a basically flat response and shows an amplitude ratio of 1.0 up to 20 Hz and possibly beyond. Clearly if any high frequency vibration is present, the RAE cushion will transmit it without loss, and it is interesting to note that several of the subjects commented that they could feel a 'tingling' at the back of the neck at the higher frequencies used. Above 20 Hz the response appears to fall and it is felt that this is due either to the thin layer of foam rubber used at the bottom of the RAE cushion providing an attenuating factor, or to the chattering or
resonance of the balls used in the cushion, providing a damping term. Clearly this could be an important factor as such cushions are to be used in dynamic situations in the future.

6.2.3 Comparison between sinusoidal and random inputs

As already mentioned in section 6.1, there was very little energy above 3 Hz in the runway spectra, so little importance should be attached to the higher frequency inputs. On the whole, however, it appears that the use of the runway tape produces the same sort of frequency response curves as those for sinusoidal inputs, for the normal cushion. The value of the amplitude ratios at 2.25 Hz for subject (1), from Fig.10 are 1.18 and 1.17. As quoted in section 6.2.2(c), the resonances are both at 4 Hz but the random tape gives an 8% lower amplitude. The phase responses for the normal cushion also compare very favourably showing about 25° lag at resonance and about 60° lag at 10 Hz for both inputs.

The question remains from section 6.1 as to whether or not the system under test is behaving in a linear manner. The results from the runway tests, at 2.25 Hz, where the acceleration level on take-off may be as great as ±0.80 g, differ little from the repetitive test results for ±0.20 g, for amplitude ratio and phase lag. It would be of little use comparing the results at resonance as the vibration level at 4 Hz (or at frequencies other than about 1 Hz and 2.25 Hz), during a take-off, is probably less than ±0.10 g. Thus taking the results overall, comparing Fig.10a where the input levels are constant across the frequency band and Fig.10b where the curves represent a 10:1 range of acceleration values across the frequency scale, then the close similarity up to about 8 Hz between these figures, for amplitude and phase, indicates that the responses obtained in the sinusoidal tests may be used to predict the effects of random vibrations in the range occurring during take-off. Above 8 Hz there is little energy in the random inputs, so no comparisons may be drawn. It is probable that the difference apparent between the ±0.20 g and ±0.40 g curves is due to involuntary
tensing of the subject's muscles altering his body characteristics at the higher level.

6.3 **Seat lateral vibration characteristics**

In the lateral mode Fig. 7a shows that the overall cushion/floor response has a resonance at between 12 and 15 Hz with an amplitude ratio of about three and a phase lag of 90°. The RAE and normal cushions resonate at 15 Hz and the PSP at 12 Hz. There is some evidence of a second peak below 1 Hz and the curves begin to climb from about 4 Hz to a factor of about 1.80 at 10 Hz. Fig. 7b shows the seat-pan/floor results and indicates a resonance at about 15 Hz with a ratio of about 2.8 and phase lag of 90°, for all subjects and conditions (except with the PSP which would not accommodate an accelerometer bar on the seat-pan), with very little scatter. Thus the cushion/floor resonance discussed above is mainly due to the seat itself rather than to the cushion characteristics (apart from the slight difference in the values of resonant frequency shown in Fig. 7a). The slight peak at or below 1 Hz is missing from Fig. 7b indicating that this effect is from the cushion.

Summing up, over the range of frequencies one would expect to meet in the lateral mode in the aircraft say 1 to 6 Hz, the seat response is constant at 1:1, with about 10° phase lag.
DISCUSSION OF RESULTS (SECOND SEAT)

7.1 Response of the seat to vertical vibration

It can be seen in Fig. 11 that a vertical resonance in the response at the cushion/man interface, occurs at a frequency of 3.5–4 Hz with a peak amplitude ratio of approximately 1.5. At frequencies above this resonance the amplitude ratio decreases until at 20 Hz it is in the region of 0.5. It could also be seen that neither the difference in build between the subjects nor the use of the armrests had much effect in the frequency or amplitude ratio in this mode. However, the higher level of input showed some change in the shape of the frequency response curve with a slight decrease in resonance frequency and increase in amplitude ratio of the primary resonance and a more noticeable difference at the secondary resonance. These differences could be due to changes in the subject or non-linearities in the seat response.

The vertical resonance of the cushion occurring at a frequency of between 3.5 and 4 Hz, is unfortunate, since this could fall within the frequency range associated with major vertical vibration modes of the fuselage in some large modern civil aircraft. It is also, according to the ISO Guide in the range in which the human body is most sensitive. It should be noted however that this cushion is by no means unusual in this respect, since all soft cushions which are of a similar stiffness, must have a resonance in this region when coupled with the dynamic system of the human body. Nevertheless, if this resonance could be reduced in level and the resonance frequency raised, a considerable improvement could be achieved in the comfort of aircrew and their ability to perform their tasks. This would be of particular importance when an aircraft is vibrating severely, for example, during heavy turbulence and take-offs from rough runways.

The increase in seat pan-to-floor amplitude ratio above about 20 Hz is probably of little importance in most circumstances, since the attenuating
properties of the present soft cushion at this frequency affectively prevents most of the vibration reaching the subject.

It can be concluded that the vertical resonance found in the investigation, is undesirable and is predominantly of the cushion. Some preliminary work carried out at RAE has shown that a dry fluid cushion, variations of which are described by Bolton, have some advantages in comfort under low frequency vibration. Measurements have also been made which show that cushions of this type have an amplitude ratio of unity from 1 Hz up to at least 20 Hz. The use of such a cushion would obviate any amplification of vibration levels in the frequency range to which the human body is most susceptible. However, it is desirable to maintain some of the higher frequency attenuating properties of a soft cushion, especially since there are some signs that there is a resonance in the seat/floor response in the neighbourhood of 30 Hz. It might be possible therefore to develop a composite cushion consisting of a dry fluid filling on top of a thin layer (say 10 mm) of fairly firm foam plastic or rubber.

7.2 Response of the seat to lateral vibration

The response of the seat to lateral vibration is illustrated in Fig.12. There appears to be two modes; one with a resonance frequency of about 1 Hz and an amplitude ratio of approximately 1.2 and the other with a resonance frequency between 14 and 20 Hz and an amplitude ratio of between 1.4 and 1.75. The difference in response is small between subjects used. However, when the armrests are in use the resonance frequency of the second mode is higher at about 18 Hz and its level lower at about 1.55 amplitude ratio. A small difference could be seen in the resonance frequency and amplitude of both modes when the height of the seat is varied.

The first of these two resonance conditions, which occurs at approximately 1 Hz, is believed to be basically a body resonance. The motion of the body at this frequency is reflected at the seat due, in part, to a low level of rigidity in the mechanism which adjusts the seat height. This mechanism,
which has its locking pin on the left hand side of the seat, supports that side of the seat directly but the opposite side is connected to it by rods which pass under the seat-pan. At this low frequency body resonance, these rods appear to flex in torsion so allowing the right hand side of the seat to move up and down, which results in the whole seat swaying in sympathy with the subject. Any reduction in this sway would improve the support and consequently the comfort and capability of aircrew under this type of vibration.

It can also be seen in Fig. 12 that not all of the flexing at this frequency takes place on the seat structure, the cushion also allows a certain amount of additional motion to take place.

The second lateral resonance which occurred between 14 and 20 Hz appeared to be a resonance of the whole seat structure. This resonance should not cause much discomfort especially in a fixed wing aircraft which sustain very low levels of vibration in this frequency range, but may pose a problem in rotary winged aircraft, which have blade passage frequencies in this area.
CONCLUSIONS

8.1 First seat

The vertical and lateral vibration transmission characteristics of the aircraft seat when fitted with its normal soft cushions, as well as the RAE dry fluid cushion and the PSP, have been investigated using four subjects, under sinusoidal and typical runway vertical vibration and sinusoidal lateral vibration.

The results indicate that for vertical vibration, the seat when soft cushions is fitted, amplifies cockpit floor vibration by about 20% in the important 2.25 Hz regime, where due to aircraft characteristics, the main energy input is likely to occur. The seat exhibits a resonance around 4 Hz where a peak amplification of about 1.6 occurs. Both of these amplifications were virtually eliminated when an experimental RAE bird-seed cushion was substituted for the normal soft cushion (and to a lesser degree when a standard PSP was used). These alternative cushions however do not provide the good attenuation of frequencies above 10 Hz afforded by the soft cushion.

For lateral vibration the seat structure gave a resonance at about 15 Hz at the seat-pan and all the cushions transmitted the vibration almost unmodified to the man.

Comparison of the results obtained with sinusoidal and random (or runway) type vertical vibration, indicates that sinusoidal excitation is a valid way of obtaining vibration transmission characteristics under the more representative random input conditions.

8.2 Second seat

Of the limited number of seats tested in vibration by this Department, this one was found by the subjects to be the most comfortable to sit in for the duration of the tests.

The vibration measurements made had a very limited scope, only the basic responses at the seat-pan and cushion top have been investigated. The seat was found to have broadly similar vibration characteristics to

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to other aircrew seats tested, although amplitude ratios appeared a little lower at resonances both in the vertical and lateral axes. The vertical resonances at 3.5-4 Hz and the horizontal resonance at about 1 Hz would almost certainly adversely affect the performance and comfort of the crew if similar vibration frequencies occurred at a high enough level in an aircraft. Any reduction in amplitude ratio in these frequency ranges would increase the comfort of aircrew and, under some conditions, their ability to perform their tasks.
Appendix A

THEORETICAL SUBJECT/CUSHION RELATIONSHIP

Fig. 1 shows a standard two-degree-of-freedom system. We shall assume that the top system is the man (obviously an approximation) and the lower system represents the cushion.

As usual the governing equations are,

\[ m_2 \ddot{z} = K_2(y - z) + C_2(\dot{y} - \dot{z}) \]  \hspace{1cm} (1)

and

\[ m_1 \ddot{y} = C_2(\dot{z} - \dot{y}) + K_2(z - y) - C_1(\dot{z} - \dot{x}) - K_1(y - x). \]  \hspace{1cm} (2)

Let us assume that the cushion has zero mass, i.e. \( m_1 = 0 \), then equation 2 becomes (using the operator \( D = \frac{d}{dt} \))

\[ [(C_1D + K_1) + (C_2D + K_2)] y = (C_2D + K_2)z + (C_1D + K_1)x \]  \hspace{1cm} (3)

Equation (1) gives us the frequency response of the man

\[ \frac{z}{y} = \frac{(C_2D + K_2)}{(m_2D^2 + C_2D + K_2)} = \bar{H} \]  \hspace{1cm} (4)

Also the impedance of the man can be shown to be given by

\[ \frac{\text{input force}}{\text{input velocity}} = \frac{(C_2D + K_2)(z - y)}{\dot{y}} = \frac{m_2D}{\bar{Z}} \]  \hspace{1cm} (5)

It can be shown that this is a perfectly general equation, not necessarily relating to a single order system.

If we now eliminate \( \ddot{z} \) from equations (3) and (4), and include equation (5) then we have that
\[ [y/x] = \text{frequency response of the cushion} \]

\[
= \frac{1}{1 + \frac{D}{C_2D + K_2}} \quad : (6)
\]

The term \((C_2D + K_2)\) can be extended into a general term defining the cushion damping and stiffness — again not necessarily governed by a first order system response.

Thus in general terms we can say

\[
\text{frequency response across the cushion} = \frac{1}{1 + \frac{D(\text{subjective impedance})}{\text{cushion characteristics}}} 
\]

We have thus demonstrated that the response across the cushion is a function of the impedance of the subject sitting on it.
### Table 1
SUBJECT HEIGHTS AND WEIGHTS

<table>
<thead>
<tr>
<th>Subject Number</th>
<th>Height (m)</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.75</td>
<td>61.2</td>
</tr>
<tr>
<td>2</td>
<td>1.72</td>
<td>60.3</td>
</tr>
<tr>
<td>3</td>
<td>1.78</td>
<td>83.0</td>
</tr>
<tr>
<td>4</td>
<td>1.75</td>
<td>68.7</td>
</tr>
<tr>
<td><strong>First seat</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1.80</td>
<td>63.6</td>
</tr>
<tr>
<td>2</td>
<td>1.80</td>
<td>79.4</td>
</tr>
<tr>
<td>3</td>
<td>1.83</td>
<td>76.2</td>
</tr>
<tr>
<td><strong>Second seat</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Input</td>
<td>Sinusoidal</td>
<td>Runway</td>
</tr>
<tr>
<td>--------------</td>
<td>------------</td>
<td>--------</td>
</tr>
<tr>
<td></td>
<td>Cushion</td>
<td>Normal</td>
</tr>
<tr>
<td>Level (±m/s²)</td>
<td>2</td>
<td>4</td>
</tr>
<tr>
<td>Subject</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>f_{max} (Hz)</td>
<td>4.5</td>
<td>4.25</td>
</tr>
<tr>
<td>R_{f_{max}}</td>
<td>1.68</td>
<td>1.58</td>
</tr>
<tr>
<td>R_{2.25}</td>
<td>1.17</td>
<td>1.15</td>
</tr>
</tbody>
</table>

f_{max} is the resonant frequency

R_{f_{max}} is the amplitude ratio at f_{max}

R_{2.25} is the amplitude ratio at 2.25 Hz
Fig 1: Subject/cushion schematic diagram.
Fig 2: First seat, fitted with a normal cushion.
Fig 3: Second seat, showing attachment to vibration rig.
Fig 4. Range of response at cushion top, for first seat, for normal cushion under vertical vibration. (all subjects and conditions)
Fig 5: Range of responses, first seat, at seat pan, vertical vibration (all subjects, all conditions).
Fig 6: First seat, cushion top response, vertical vibration, effect of level and different cushions.
Fig 7: First seat lateral response.

- Normal cushion •
- RAE cushion +
- PSP □

Amplitude ratio vs Frequency (Hz)

- Shows upper and lower limits of results

Frequency (Hz)

a) Cushion top response.
b) Seat pan response (all conditions).
Fig 8: Vertical vibration (with octave band analysis), measured at cockpit floor during take off, using first seat.
Fig 9: Computed linear spectrum at floor of cockpit during take off.

Acceleration in non-dimensional computer units

0 1 2 3 4 5 6 7 Hz
a) Sinusoidal.

b) Random (runway) input.
Fig 11: Second seat, response curves for vertical excitation.
Fig 12: Second seat, response curves for lateral excitation.
CHAPTER SEVEN

THE TRANSMISSION OF VERTICAL VIBRATION TO THE HEAD AND SHOULDER OF THE SEATED MAN SUBJECTED TO A LINEAR SWEPT SINE WAVE ACCELERATION INPUT
INTRODUCTION

So far all the experiments detailed in this Thesis have ignored any possible variation of the body's vibration transmission characteristics due to changes in posture, be it involuntary or controlled. In fact the assumption has been made that either the posture has little effect on the results, or the posture has remained unchanged throughout the duration of the tests. Chapters 2 and 5 have attempted to investigate the effects the acceleration level and limb positions have on the response characteristics. Investigation of the literature has shown that very little research has been done on the effects of these parameters on the required response, possibly because in the past almost all the work on body response has been done using sinusoidal input functions, which require a long period to cover all the necessary permutations of frequencies and postures etc. (see section 3.1.1). So long in fact that it is dubious whether it would ever be possible to maintain the same posture for the duration of the test if so required.

When the new vibration rig (Chapter 3) therefore became available for human response work, with its ability to reproduce transient vibration, together with the Hewlett Packard Fourier Analyser which could analyse such transient inputs and their respective outputs, it was decided to devise an experiment which would aim at quantifying the human frequency response under conditions of varying acceleration level, posture and limb positions. These responses would relate to the head and the shoulders, with respect to the input, as it is assumed that these positions relate closely to the centres which govern man's response to vibration. The use of transient inputs can be justified on several accounts (see sections 3.1.1 and 3.1.2), the most important being that in real life one meets transient rather than sinusoidal vibration, and the use of a short transient (normally less than 30 seconds) enables the subject to maintain any required posture for the required time. Also as will be seen later, one set of tests on one subject covering all the
parameter variations took only about 15 minutes, whereas to cover all the variations under sinusoidal testing may well have taken hours.

The need to perform an experiment, such as detailed in this Chapter, can be summed up as answering the comment of Potemkin¹, "... hitherto (1971) there have been no papers on the investigation of the effect of vibrations on the dynamic reactions of man dependent upon the change in his working posture".
2 EQUIPMENT

2.1 Vibrator

The vibrator used was the new two-axis facility, which is fully described in Chapter 3.

2.2 Seat

The seat used was a rigid wooden structure fitted with arm rests. A sketch of the seat (which is one of a pair fitted on the rig) showing various angles and dimensions is shown in Fig. 1. The question arises as to why a rigid seat was used and what use are the results from an experiment using such seats. Chapter 6 has outlined some of the work that has been done at the RAE on the dynamic properties of cushions and seats. In the past it has been assumed that the response of soft cushions has been that of a simply loaded single spring; however the results given in Chapter 6 plainly indicate that frequency responses across the cushion — when a man is sitting on it, not a dummy — are some form of reflection of the bodies response and hence equally as complicated. As a corollary, one cannot really understand or explain the behaviour of cushions under these circumstances until one fully understands the transfer function of the man. Therefore it was decided to isolate the man from any other possible dynamic systems and test him in isolation. Clearly at a later date one would have to validate the results found here in an environment where soft cushions are used, clearly differences will exist, especially in the area of back cushions, but the research into the subject on a hard seat must come first and provide the basis for future studies.

2.3 Harness

For the purpose of these tests, no seat harness was used. In reality, the subjects retained a lapstrap, so slack, that no restraint was offered, simply as a rig safety precaution. The point can again be made here that harness was not used, as it would give us, like cushions, another dynamic system to complicate the required response of the man.

The accelerometers used, were held in position by head and shoulder harnesses; these are fully described in section 3.
2.4 **Accelerometers and associated amplification/recording equipment**

The instrumentation used was basically similar to that detailed in Chapter 5 except that instead of terminating the signals in our analyser, they were recorded on a Tannberg type 100 magnetic, frequency modulation tape recorder. This tape recorder has a voice track which can be used to provide a check of run number, level, subject, etc.

2.5 **Computation facility**

For analysis purposes, the tape recorder was connected to a Hewlett Packard type 5451A Fourier Analyser. This is an instrument which can break down any input that varies with time and show its component frequencies. It does this digitally, which means it is faster and more accurate than previous analogue equipment used. A main feature of the Hewlett Packard equipment is the provision of a keyboard on which the user can punch the keys for a variety of mathematical functions to be performed on the frequency domain data. These keys can be used to calculate the transfer function between two signals entering the system. The transfer function (or frequency response) is the mathematical description of the system, be it an engine, structure, or the human body, and can be defined as:

\[
\text{transfer function} = \frac{\text{Fourier transform of the output}}{\text{Fourier transform of the input}}
\]

This function is calculated as a complex quantity, i.e. has a modulus and a phase relationship. As the computer has a two channel simultaneous input and its own built-in analogue-to-digital convertors, phase information is preserved.

The actual analysis theory is covered in greater depth in section 4.

2.6 **Subjects**

Six subjects were used, selected to represent the normal range of the population. It could be argued that six is an exceedingly small sample from which to draw conclusions, but it was strongly felt that it would be preferable to thoroughly test a few subjects and establish the variation in their responses under several body conditions rather than to test a lot of subjects under
relatively few test conditions. Seldom in this work will be mentioned average response, as by and large one is not interested in the mean person, but rather in the range of variations, so there is no need for a large sample for statistical significance. Table 1 shows each subject's height, weight, sitting height, age etc. Section 7, discussion of results, includes the variation in responses between the subjects and attempts to correlate these variations with the corresponding variation in the above anthropometric parameters.

Prior to any experimentation, all subjects were asked to fill in a 'Subject File'. It was stressed that the information given would be kept strictly confidential and any reference to a subject would be made by a subject number - never by his name. The proforma used is attached as Appendix A and was drawn up by the author following his service on a committee set up to discuss safety in vibration experiments. This committee eventually reported to the British Standards Institute who subsequently published a Draft for Development covering the safety aspects of human vibration experiments. The acceleration levels used in this experiment did not exceed the current "fatigue decreased proficiency" set out by the International Standards Organization, and hence following the safety guide's recommendations for schedule I type vibration, no medical certification, other than that contained in the proforma, was undertaken.
3 EXPERIMENTAL DESIGN

3.1 Input characteristics

3.1.1 Choice of input function

Of the reports found dealing with the transmission of vibration across the body, almost all of them have used a sinusoidal forcing function as an input. Very early on in our interest in vibration it was realised that although one may start off with a pure sine wave output from our oscillator, by the time the signal reaches the floor of the vibrator, a large amount of harmonic distortion may well be present in our input. Clearly this distortion can provide erroneous answers (see introduction to Chapter 2). The early experiments detailed in this thesis used nominally sinusoidal wave forms as inputs, but the importance of the distortion was appreciated, in fact Chapter 2 gives details of attempts to use this harmonic distortion as an input factor enabling us to measure at frequencies outside the normal range of the vibration facility. For both the experiments in Chapters 2 and 5, the distortion was filtered out or measured by using a harmonic analyser. Vibrations measured in the field clearly demonstrated that rarely does one meet sinusoidal oscillations of constant amplitude and frequency, such as were used as input signals for previous experiments, which set out to measure body impedance or transmissibility. In these experiments subjects were exposed to perhaps many minutes of sinusoidal oscillation whereas in real life rarely can one find records showing more than ten or so oscillations (perhaps at about 4 Hz giving an exposure time of a few seconds only). Thus in the past one has been faced with the problem of testing subjects under unrealistic conditions. Also Dieckmann has shown that under vibration of a sinusoidal type, man exhibits an elliptic motion at the head. Shurmer has argued that this is because "the centre of gravity of the upper torso of a man standing or sitting erect is forward of the spine and thus non-vertical motions will occur even for forces applied vertically. The extent of these 'spurious' rotational movements will depend on the body tone adopted and on the restraint provided in the form of a harness".
One really wonders therefore what past results really mean, as we are effectively saying that under sinusoidal input conditions the accelerations measured on the head are a combination of translational and rotational components and as nobody appears to have accurately measured both quantities, then the validity of these results may be questioned. As will be discussed later in this Chapter, past results show a large degree of scatter, this may in part be due to the rotational components of the measured values.

If one precludes the use of sinusoids due to the foregoing arguments, then one is left with the choice of either using a random signal, or a 'swept sine' signal, as an input. In a swept sine the frequency of application is made a function of time, so that normally the frequency increases (or decreases) linearly with time. Typical values might be sweeping from 1 Hz to 30 Hz in 10 seconds.

A frequency sweep or swept sine wave is defined as,

$$ F(t) = F_0 \sin \phi(t) \quad 0 \leq t \leq T $$

The function is therefore of constant amplitude and time dependent frequency, the instantaneous variation of frequency being given by

$$ \omega(t) = \frac{d\phi(t)}{dt} $$

A linear variation of frequency with time is given by

$$ F(t) = F_0 \sin (at^2 + bt) $$

where, if $ f_1 $ = initial frequency and $ f_2 $ = final frequency, then $ a = \pi(f_2 - f_1) $ and $ b = 2\pi f_1 $.

It can be shown that the spectrum of such a swept sine wave exhibits the following properties

(i) the modulus is reasonably flat or rectangular;
(ii) the mean value of the spectrum is $(\pi/4a)$ and
(iii) the amplitude of the ripple superimposed on the mean spectrum level is proportional to $1/\sqrt{T}$, where $T$ is the duration of the transient.
A swept sine signal however is also completely deterministic, one has the input transient cause and the output transient effect. Thence by division of the frequency transforms of the two quantities, one can use a single sweep function to calculate the required transmission function. Another important factor in favour of this swept sine technique is that it may overcome the difficulty concerning the rotational vibration present at the head in sinusoidal testing. It is a pity that none of the early tests gave a history of the rotational component. It can be argued or surmised that the rotation is a steady state phenomenon and thus needs time to build up - the fact that the form of the rotational motion varies with applied frequency, from an ellipse to a circle and back to an ellipse for increasing frequency - tends to bear out this point. The swept sine is therefore preferable in that these components are not allowed to build up, to the extent that the acceleration readings are affected.

If a random signal is used then the question of statistical reliability and random error arises. It can be shown that \( \text{Consider Bendat, 1972}\) for reasonable random and bias errors then record lengths of the order of 100 to 200 seconds must be considered. Even then the functions generated can never by definition be completely deterministic.

Returning to the point regarding length of experiment, in past experiments, some conducted by the author, subjects have had to undergo tests lasting in excess of 30 minutes. Clearly subjects are unable to maintain the same posture, muscle tone, arm, leg and head positions for such a period, so conditions must have varied across the test. Even using the technique of 'balancing' the experiment for effect of order will not eliminate this effect, it will still result in an increase in scatter at certain points. Using the swept technique, constant conditions only need to be preserved for approximately 30 seconds, then conditions will be altered in a controlled manner and another test begun. Hornick has shown that over a period of 2 minutes, head transmissibility can increase from 25% to 39% at 5 Hz, and from 68% to 93% at 2 Hz, for a subject standing.
with bent legs. Clearly this is an extreme condition but it does indicate that values can alter over a period of time, and again the point can be made that the experiment is unrealistic in that in real life such a subject would not be subjected to such a vibration for such a length of time.

The swept sine technique, although praised in the preceding paragraphs, has several disadvantages. Firstly analysis is difficult, a digital computer complete with Fourier transform software is desirable, if not essential, whereas for sinusoidal work, only an rms meter and filter are necessary. Secondly the problem of non-linear response. This point will be covered in more detail later, but the point will be made here that if the response of the body to vibration inputs is grossly non-linear then the only technique which can be used to give a general solution is that of sinusoidal input. Even this technique is suspect as, by definition of non-linearity, an input sinusoid of frequency \( f_1 \) can generate super-harmonics in the output of a non-linear system, of frequencies \( 2f_1, 3f_1 \) etc., depending on the type of non-linearity (see Chapter 2, Appendix C). As will be seen some of the early work in this thesis, using sine waves, looked at the question of non-linearity and for the final experiment the null hypothesis was made that the body was linear and that the results would have to prove otherwise. In fact if the results of such an experiment did show a high degree of non-linearity, qualified by the conditions of the acceleration levels and frequencies used, then no test will really apply enough to give us a general picture covering the bodies frequencies response. Again if this is so then if one wishes to define the vibration levels at specific parts of the body - the basic requirement of this thesis - then these levels will have to be determined for the particular vibration input as measured on the vehicle in question, and the results will apply to that vibration condition and that one alone.

The third disadvantage of the swept technique is that by definition it is a transient and it can be shown that initial levels can be generated, which are in excess of the input amplitude. This factor can easily be recognised with a square wave where the initial square front provokes an overshoot in the output.
response. This disadvantage is basically psychological in that the subject becomes aware of a frequency (in reality his resonant frequency) on top of the impressed frequency. Mathematically we are still measuring cause and effect, so the calculation of the frequency response will not be affected.

Summing up, it is felt that in view of the feeling that a lot of the previous reports (including some reported here of the author) suffer from

(i) long duration of experimentation imposing unnatural constraints on the subjects in terms of maintaining posture etc., and
(ii) lack of distortion information - and a lack of the knowledge that this distortion can severely affect ones readings,

then the use of a swept technique is recommended and is used for the final experiment.

A word of explanation would appear to be necessary here to justify the use in this thesis of the two inputs. The early experiments used sinusoidal techniques, and the final one, the swept technique. As will be seen later it is felt that the greatest source of error in past reporting has been the lack of measurement and appreciation of the effects of harmonic distortion in the input and output traces. All the experiments reported in this thesis, measure and appreciate this factor, in fact in one experiment, it is used to advantage. The question of linearity remains and although the early experiments had some conflicting views on linearity, the stand was taken that if the system proved highly non-linear then whatever input was used would result in dubious findings.

The problem of linearity and the need to overcome it are summed up by Potemkin, "the problem of determining the dynamic reactions of the operator under the actions of random (transient) vibrations is of the greatest interest, since it is in fact random and not harmonic oscillations which usually accompany the operator of each machine, however the majority of the papers relate to harmonic excitation".

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3.1.2 Duration of transient input

Theoretically the time taken for the chosen swept sine function should not affect the calculation of the frequency response. In the extreme, of course, the input reduces to an impulse and the output then becomes the impulse response. However the input has to be applied via a high inertia system – the vibrator – and this needs an input signal of finite length otherwise the signal level on the table is very small. Also although it has been argued that swept sine waves do not exist in the real world (nor for that matter do sinusoidal or purely random signals), some reference to what happens in reality must be preserved, and it is quite common for 'bursts' of vibration to be encountered lasting about 20 seconds, e.g. a car traversing a rough patch of road, slowing down at the same time to give a variation in frequency, an aircraft in turbulence, etc. Past researchers who had used swept sine techniques for testing moderately damped systems (the man?) had reached an empirical law which stated that for a reasonable output response signal sweep rates of approximately 1 Hz/s (for the low frequency end of the spectrum) were to be recommended. Thus swept times of 10 s for the 1-10 Hz and 25 s for the 2-30Hz bands were selected.

3.1.3 Frequency ranges

As briefly mentioned in section 3.1.2, two ranges of frequency were used, (a) 1-10 Hz in 10 s and (b) 2-30 Hz in 25 s. The results were then superimposed to give an overall frequency range of 1-25 Hz. This range was decided upon as it covers all the bodies major resonance points, especially in the range 10-30 Hz which has been neglected by past researchers. Two frequency scales were used after it was found in an unreported pilot experiment that the vibrator could not give a flat response over the range 1-30 Hz in one go. The implications and possible errors involved in such superimposition is further discussed in section 5.

3.1.4 Acceleration levels

In order to check for any non-linear effects, the tests were conducted at three input acceleration levels, i.e. this level being the constant (for 25 s or 10 s) amplitude of the frequency varying function. The levels chosen were
those associated with decrement in performance, rather than with comfort or perception, and were approximately $\pm 4.0 \text{m} \cdot \text{s}^{-2}$, $\pm 2.8 \text{m} \cdot \text{s}^{-2}$ and $\pm 2.0 \text{m} \cdot \text{s}^{-2}$. Typical input traces are illustrated later. The range of these levels are very typical of those measured by the author in vehicles under turbulent or rough road conditions.

3.1.5 **Repeatability**

Initially it was intended to repeat every subject's runs three times. However this proved impracticable so two runs per condition were analysed. Also an initial unreported pilot experiment indicated that each subject was able to repeat a 'condition' within narrow limits. As will be seen later the repeatability per subject was excellent, justifying the reduction in runs from three to two.

3.1.6 **Vibration axis**

Again initially the grand plan required the measurement of the human frequency response for all three linear axes and for the standing and sitting man. When the pilot experiment results were analysed it became clear that important original results were being found for the seated man subjected to vertical vibration (which is the main circumstance encountered in real life), therefore it was decided to limit the scope of the thesis for the sake of more detailed analysis of this condition.

3.1.7 **Linear versus rotation**

The rig used for this experiment could only reproduce linear vibration. However it is well known that under certain conditions the body may generate rotational vibration components at certain parts even though the input vibration is linear. This can of course introduce errors as the instrumentation used is basically of a linear type. This point is further discussed in a later section.

3.2 **Output measurements**

As discussed in the introduction, if one is interested in trying to define the effects of vibration on a subject, then one must be able to define the vibration present at that point (or points) at which the subject is affected by the vibration. It is generally agreed that vibration effects are mainly centred
on the visual loop and the manual loop (the third important loop is the psychological one which is not considered for this experiment). It therefore follows that one should wish to measure hand/arm vibration and eye vibration and thus be able to determine frequency response curves between those points and the input - the seat/man interface. However the measurement of both hand and eye vibration poses many problems - and from a practical point of view is impossible. It has been found to be virtually impossible to measure the acceleration levels at the eye especially in the frequency region where the eyes are thought to be most sensitive (area 20 Hz), due to the very small amplitudes involved, even at moderate acceleration levels ($\pm 5.0 \text{ m s}^{-2}$ at about 20 Hz - a level which would solicit adverse comment from a pilot or driver - is equivalent to an amplitude of only $\pm 10$ thou in). Most previous researchers in the field have compromised by measuring head vibration with a harness, and this method was followed for this experiment. It can be argued that the vibration levels on the head are as important as at the eyes, as the skull will effectively transmit the vibration to the brain and to the loose tissues on the head. Thus measurement on the hard part of the head provides a suitable compromise.

The measurement of hand vibration is equally difficult because of the possible rotational components involved in any hand/arm movement and also because of the extreme difficulty in attaching a harness and accelerometers to the hand. It was eventually decided to construct a harness to hold an accelerometer(s) against the acromion, i.e. the top of the shoulder, and to use this as a definition of the input to the hand/arm systems. These harnesses need some degree of elasticity and any mechanical system which uses elastic harnesses to hold down a transducer can clearly give erroneous results. The curves derived may simply be the frequency response curves of the harnesses used rather than of the system under test. It was found to be comparatively easy to validate the head harness, but the shoulder harness had to be assumed rigid in that the harness was tightened to give a high stiffness and hence a high natural frequency - higher than the maximum frequency tested, thus implying a flat characteristic curve over the frequency range used.
The input parameter was seat acceleration (or foot or back acceleration, as the seat used was rigid), measured by a seat bar situated under the ischial tuberosities.

Thus our three output measures are head vibration, shoulder vibration and seat/man interface vibration.

3.2.1 Head vibration

The vibration levels on the head were measured using a specially designed head harness (Fig.2). The main structure is a plastic insert from a standard safety helmet which has an adjustment system whereby each subject could adjust the fit until it was tightly held on his head. A strip of Velcro 'adhesive' was sewn and glued around the outside of the frame of the insert. Thus we have a reasonably rigid mounting frame on which we can affix the accelerometer itself. In order to provide some elasticity for the accelerometer mounting itself, a cruciform of 1in wide dressmaker's elastic was constructed, at each end of which was sewn and glued a small strip of Velcro (this Velcro strip being the 'negative' part of that on the insert). A small cube of aluminium, grooved to accept our accelerometer, was then attached to the centre of the cruciform. Thus by spreading the cruciform across the top of the head and pressing the Velcro parts together, a reasonably rigid harness had been constructed. The position of the accelerometer itself on the top of the head could be varied by adjusting the point at which the Velcro on the insert met the Velcro on the harness. Within the limits of comfort, the harness was tensioned as tight as possible. A short series of runs where the output of the head harness accelerometer was compared with a rigidity held (via a dental bite) accelerometer in the mouth — in order to check the response of the head harness accelerometer — is discussed in a later section.

3.2.2 Shoulder accelerometer

The harness used to measure shoulder acceleration was the same as that used in the experiment detailed in Chapter 5. Two loops of dressmaker's elastic were used to hold the accelerometer, first a torso loop under the armpits and second
a vertical loop folded around the first loop and passing over the top of the shoulder. The accelerometer was attached to the second loop and held down by it. The tension of the elastic was adjusted to ensure that the accelerometer would not lift under vibration conditions.

3.2.3 Seat/man interface accelerometer

The seat acceleration was measured being a standard RAE seat bar (designed by the author) attached to the seat by a strip of Velcro, under the subjects ischial tuberosities. This bar can take three (for the three linear axes) accelerometers and has been used extensively by the RAE for field measurements.

3.3 Body parameters

Having decided on the use of a swept sine wave as our input forcing function, of duration 10 or 25 s, then one can begin to select the variables, the effect of which on the frequency response of the body, one may wish to investigate. The duration of the input function is clearly extremely important in that various conditions may be imposed on the subject which can easily be held or maintained for this short time period, but if imposed for hundreds of seconds (for a random type input) or maybe an hour (sine wave functions), then the subject could not maintain the parameters constant.

Previous work has indicated that the main body parameters, apart from the obvious anthropometric and seating variables, which might affect the transmission of vibration across the body are variations in posture and arm and leg positions. Clearly there must be other parameters which might be considered such as clothing, mental task, time of day etc., but posture and limb positions were selected as the major intra subject variables, all the others being assumed second order.

3.3.1 Posture

There are three main divisions of posture, slumped, normal and erect. The problem is not to define these variables but how to quantify the variables, i.e. how slumped is slumped and is subject 2's idea of slumped the same as for subject 6. Any variation in posture must be associated with an increase or decrease in muscle
activity and as a corollary muscle activity dictates the posture. However this theory deals with the practical aspects of vibration measurements and it was felt that any attempt to enter the field of electromyography would be unwise. Especially as one would have to presuppose that (a) one could find the appropriate muscle or muscles - and instrumentate them - that dictate or control posture and (b) the extremely low voltages involved would be measurable in the dynamic vibration environment, also these low voltages may well be completely swamped by the pseudo muscular activity generated by the relative motions of the body under vibration.

It was therefore decided to allow each subject in acting as his own control, to dictate his needs and to use a method of 'constrained limits'. During the initial briefing of each subject, they were invited to "sit on the vibrator in a normal posture with their back against the back rest, not slumped, not erect, just sitting normally as if about to perform a complex task". Immediately following this the subject was told to assume a slumped posture, to allow himself to relax in the seat, without letting his head fall forward and without letting his back come off the back rest. Then he was asked to sit at attention. Thus the subject had demonstrated the three required postures, and he was told he would be required to assume these postures in later experiments, postures which he had decided on as being his interpretation of slumped, normal and erect. As detailed in the introduction, this experiment was not aimed at defining the average man, in fact in the field of human engineering there is no such thing, we are all anthropometric freaks. Thus as long as each subject maintains his concept of these posture conditions, then it does not matter if subject 2's idea of slumped differs from subject 6's. In other words the intra subject variability is more important than the inter subject one. After all, in an aircraft, the pilot is not really concerned as to whether he is within 10% or 60% of the average, in an emergency situation he becomes the average - in a population of one!

Following analysis of the pilot experiment it was appreciated that a chance remark by one of the subjects had indicated the need for another posture variable
to be included. This subject reported that during the vibration sweeps, especially at the high frequency end of the spectrum, he was able to reduce the vibration he felt in his head region (described as a tingling of the scalp and neck) by raising himself forward off the back rest. In engineering terms clearly he had eliminated one of the inputs to the body — the vibration shear input at the back — thereby apparently reducing the vibration felt on the head. In the final experiment it was stressed to all the subjects, that for one of the conditions they should raise themselves forward off the back rest — but without altering their posture from what had been decided as normal. This may sound a contradiction in terms in that it is impossible to lift off the back without altering muscle tone and hence posture. The assumption was finally made that:

\[ \text{[Muscle activity for erect posture - Muscle activity for normal posture]} \]

\[ \Rightarrow \text{[Muscle activity for back off - Muscle activity for normal posture]}, \]

i.e. the muscles used to lift off the back are a small proportion of the muscles used to maintain a normal posture.

Thus four postures were tested, normal, erect, slumped and back off.

3.3.2 Arm position

The variations in arm position used for this experiment were arms folded, arms in lap (left hand on left thigh etc.), and arms out, resting lightly on the controls of an oscilloscope which was built into the seat structure on the vibrator. Again it can be argued that these variables should be rigidly controlled. Again the answer is one of impracticabilities in terms of measuring muscle activity and also that each subject acts as his own control with little final inter subject averaging. Thus our selection of arm variables covered the situations of a pilot or driver, viewing dial controls etc. (arms in lap and folded), and of a subject actually manipulating such controls in front of him (hands on scope).

3.3.3 Leg position

An earlier experiment, detailed in Chapter 5, had indicated that leg position could influence the level of vibration reaching the shoulder to a
significant degree. Thus leg position was included here as a major variable, and the variations chosen were, legs normal (again decided upon by the subject, legs back (so that the shin/femur angle was approximately a right angle) and legs right forward (subject leg length dictating the final position).

3.3.4 Overall plan

Thus we have arrived at the final choice of variables 4 \times \text{posture}, 3 \times \text{arm position} and 3 \times \text{leg position} – a possible test pattern containing 36 runs. If all these possible combinations were tested then (a) the analysis time and effort involved would be enormous, when one includes the number of subjects, acceleration levels, ratios measured etc., and (b) each subject would have to spend a long time being vibrated and hence it is doubtful whether his normal posture at the beginning of such a run would correspond to normal posture at the end. It was therefore decided to limit the runs to eight – the variation of each parameter when the other two are held normal. The following table illustrates the eight runs which made up one test.

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Posture</th>
<th>Arms</th>
<th>Legs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Normal</td>
<td>Lap</td>
<td>Middle</td>
</tr>
<tr>
<td>2</td>
<td>Erect</td>
<td>Lap</td>
<td>Middle</td>
</tr>
<tr>
<td>3</td>
<td>Slumped</td>
<td>Lap</td>
<td>Middle</td>
</tr>
<tr>
<td>4</td>
<td>Normal</td>
<td>Folded</td>
<td>Middle</td>
</tr>
<tr>
<td>5</td>
<td>Normal</td>
<td>Lap</td>
<td>Forward</td>
</tr>
<tr>
<td>6</td>
<td>Normal</td>
<td>Lap</td>
<td>Back</td>
</tr>
<tr>
<td>7</td>
<td>Normal</td>
<td>On oscilloscope</td>
<td>Middle</td>
</tr>
<tr>
<td>8</td>
<td>Back off</td>
<td>Lap</td>
<td>Middle</td>
</tr>
</tbody>
</table>

Normal condition is defined as, normal posture, arms in lap and legs in middle. Thus runs 1, 2, 3 and 8 give variations in posture, runs 1, 4 and 7 give variations in arm position, and runs 1, 5 and 6 give variations due to leg positions.

3.4 Output parameters

The output parameters required from this experiment are simply head/seat (h/st) and shoulder/seat (sh/st) frequency response curves, giving amplitude
modulus and phase, for all the conditions tested. The calculation of frequency response curves has been preferred to the more usual impedance calculation, for several reasons (reference introduction to Chapter 1). Firstly one is interested in the levels of vibration present at important parts of the body with respect to a specified input level, rather than the dynamic load that the man imposes on the system - the impedance. As discussed in the introduction to Chapter 1, if one has a double spring mass system where the second mass is appreciably smaller than the first, then although this second mass may represent a critical part of the system, the impedance calculations will barely demonstrate the presence of this second mass whereas frequency response curves to this mass will.

This argument assumes that the body behaves as a simple series analogue, whereby impedance and frequency response are mathematically interrelated in that the impedance equation also defines the motion of the centre of mass. However it seems possible that the body responds as a combination of series and parallel loops, in which case the above interrelationship can be lost. This point is discussed in much greater detail in section 7, discussion of results.
The input chosen for this experiment was the linear swept sine wave. This can be defined by the equation

\[ F(t) = F_0 \sin \phi(t) \]  

where \( 0 \leq t \leq T \), \( T \) being the duration of the transient input.

This function is therefore of constant amplitude and time dependent frequency, the instantaneous variation of frequency with time being given by

\[ \omega(t) = \frac{d\phi(t)}{dt} \]  

The use of the swept sine wave in frequency response measurement (particularly in the analysis of building structure responses) has been extensively examined and the linear sweep has been shown to be most suitable as a transient forcing function because the modulus spectrum is rectangular. Thus by selecting our required upper and lower frequency limits for the above function, we can produce a completely deterministic function within these limits — there is no question of signal probability levels giving spurious levels due to random data, the input can be used as a one-shot deterministic function between the selected limits of frequency.

A linear variation of frequency with time is given by

\[ F(t) = F_0 \sin(\omega_1 t + b) \]  

where \( \omega(t) = 2at + b \) and \( d\omega(t)/dt = 2a \) with \( a = (\omega_2 - \omega_1)/2T \) and \( b = \omega_1 \), \( \omega_1 \) and \( \omega_2 \) being the initial and final frequencies.

To examine the spectral characteristics of this function, it is necessary to evaluate the Fourier transform defined as,

\[ F(i\omega) = \int_{-\infty}^{\infty} F(t) e^{-i\omega t} dt \]
It is not proposed to demonstrate here the evaluation of this integral with its complicated Fresnel integral components, as standard mathematical papers can be quoted where the evaluation is performed.

If \( \omega \) is not close to either \( \omega_1 \) or \( \omega_2 \), then via a series of approximations, it can be shown that the mean modulus spectrum level is

\[
|F(\omega)| = F_0 \sqrt{\frac{\pi}{4a}}
\]  

(5)

i.e. independent of frequency.

It can also be shown using the above approximation that the spectrum levels at the frequency limits \( \omega_1 \) and \( \omega_2 \) can be given by

\[
|F(\omega_1)| = |F(\omega_2)| = \frac{F_0}{2} \sqrt{\frac{\pi}{4a}}
\]

(6)

i.e. half the above value.

Thus we have mathematically defined our input function and shown that Fourier transform analysis of such an input yields a rectangular spectrum shape - the function is also deterministic so that single analysis is sufficient to completely define the input.

If one now applies such an input to the system under test, the human body, then the output from the body will again be a swept sine but this time the amplitude will not be constant as the frequency varies. The manner in which it differs will of course be the frequency response of the system under test.

Expressing this in mathematical terms, the frequency response function \( H(i\omega) \) of a linear system may be derived from the response \( y(t) \) to a transient input excitation \( x(t) \) according to

\[
H(f) = \frac{S_y(f)}{S_x(f)}
\]

(7)

where \( S_y(f) \) and \( S_x(f) \) are the Fourier transforms of \( y(t) \) and \( x(t) \) respectively, i.e. if \( x(t) = 0 \) and \( y(t) = 0 \) for \( t < 0 \) then
\[ S_y(f) = \int_0^\infty y(t)e^{-i\omega t} dt \]  

and

\[ S_x(f) = \int_0^\infty x(t)e^{-i\omega t} dt \]  

Thus by calculating the Fourier transforms of our input and output swept functions and dividing these complex quantities, we can generate the required frequency response in terms of modulus amplitude and phase angle.

In order to calculate the above transforms a Hewlett Packard type 5451A Fourier Transform Analyser was used. This analyser utilizes a digital computer to calculate the transforms of the time varying signals.

In order to implement the Fourier transform digitally, one must convert the continuous input signal into a series of discrete data samples. This is accomplished by sampling the input (or output), \( x(t) \) or \( y(t) \), at certain intervals of time. We will assume the samples are spaced uniformly in time, separated by an interval \( \Delta t \). In order to perform the above integrals, the samples must be separated by an infinitesimal amount of time (i.e. \( \Delta t + dt \)). Due to physical constraints on the analogue-to-digital converter, this is not possible. As a result we must calculate

\[ \tilde{S}_x(f) = \Delta t \sum_{n=0}^{n=\infty} x(n\Delta t)e^{-i\omega n\Delta t} \]  

where \( x(n\Delta t) \) are the measured values of the input function.

Equation (10) states that, even though we are dealing with a sampled version of \( x(t) \), we can still calculate a valid Fourier transform. However the Fourier transform, as calculated by equation (10), no longer contains accurate magnitude and phase information at all the frequencies contained in \( S_x(f) \). Rather \( \tilde{S}_x(f) \) accurately describes the spectrum of \( x(t) \) up to some maximum frequency,
$F_{\text{max}}$, which is dependent upon the sampling function $\Delta t$. Shannons sampling theorem states that it requires slightly more than two samples per period to uniquely define a sinusoid. In sampling a time function, this implies that we must sample slightly more than twice per period of the highest frequency we wish to resolve. Translating Shannons theorem into an equation

$$F_{\text{max}} < \frac{1}{2\Delta t} \quad (11)$$

for convenience equation (11) will be written as

$$F_{\text{max}} = \frac{1}{2\Delta t} \quad (12)$$

In order to ensure that aliasing errors do not occur, i.e. frequencies higher than $F_{\text{max}}$ folding back to appear as frequencies lower than $F_{\text{max}}$, the user must make sure that the $F_{\text{max}}$ he sets is higher than the highest frequency in the data. If this is not possible then anti-aliasing filters must be used before the data enters the analogue-to-digital converter.

In order to calculate $\hat{S}_x(f)$, we must take an infinite number of samples of the input waveform (cf. equation (10)). As each sample must be separated by a finite time interval ($\Delta t$), this would take an infinite time. Let us assume that the input signal is sampled from some zero time reference to time $T$ seconds, then we have

$$\frac{T}{\Delta t} = N \quad (13)$$

where $N$ is the number of samples and $T$ is the time window.

As we no longer have an infinite number of time points, we cannot expect to calculate magnitude and phase values at an infinite number of frequencies between zero and $F_{\text{max}}$. Thus we have a discrete finite transform (DFT) given by,

$$\tilde{S}_x(m\Delta f) = \Delta t \sum_{n=0}^{n=N-1} x(n\Delta t)e^{-j2\pi mfn\Delta t} \quad (14)$$
Only periodic functions have such a 'discrete' frequency spectra, and thus equation (14) assumes that the function observed between zero and \( T \) seconds repeats itself with period \( T \) for all time. It is apparent that the DFT, is actually a sampled Fourier series.

Note: There are \( N \) prints in the time series and for our purposes the series always represents a real valued function. However to fully describe a frequency in the spectrum, two values must be calculated - the real and imaginary parts. As a result \( N \) prints in the time domain allow us to define \( N/2 \) complex quantities in the frequency domain.

Then clearly

\[
F_{\text{max}}/(N/2) = \Delta f - \text{our frequency resolution.}
\]  

(15)

If we substitute equations (12) and (13) in equation (15), then we arrive at the equation

\[
\Delta f = 1/T
\]  

(16)

which in terms of basic Fourier series analysis is evident.

Thus we have established the method whereby the components of equation (7) may be calculated using the Hewlett Packard equipment; and hence by simply pressing one key on the keyboard, the calculation and print out of the required frequency response may be performed.

The computer settings for the various analyses performed are as follows:

1-10 Hz in 10 s

Anti-aliasing filters set at 10 Hz for both channels

Computer block size = 256

\( T = 10 \) s

\( F_{\text{max}} = 12.8 \) Hz.

Therefore \( \Delta t = 0.039 \) s

and \( \Delta f = 0.1 \) Hz

Even though the input data contained no information above 10 Hz (this being the upper frequency set for the swept sine wave), the period \( T \) of 10 s
meant that a $F_{\text{max}}$ of 12.8 Hz had to be selected in order to completely fill the selected block size.

2-25 Hz in 25 s

In order to put the experimental recordings into the computer it was found necessary to 'ditch' the frequencies between 25 and 30 Hz. Thus the analysis - which needed a $T$ of 20 s - was taken from 2-25 Hz.

Anti-aliasing filters set at 25 Hz for both channels

Computer block size = 1024

$T = 20$ s

$F_{\text{max}} = 25$ Hz

$\Delta t = 0.020$ s

$\Delta f = 0.05$ Hz.

A small keyboard programme was written which inputed the data, performed the Fourier analysis, eliminated the d-c components (corresponding to 0 Hz), converted the frequency domain data into polar co-ordinates, and printed out the results on an X-Y plotter.
METHOD OF TEST

First of all the vibrator was switched on and the oil and cooling water temperatures allowed to rise to their normal working values. The electronic systems were also switched on and allowed to warm up. The accelerometers were then calibrated by rotating them through $360^\circ$ around their operating axis to give $\pm 1$ g. By comparing the voltages generated and comparing them with standard values it was possible to ensure that the complete measuring system was functioning satisfactorily. The signal corresponding to $\pm 1$ g was recorded on the FM magnetic tape recorder and served as a calibrator signal for the subsequent analysis. A low frequency sinusoidal signal was then fed into both axes on the rig to ensure full and free movement in both axes. Sometime during the check the emergency buttons, including the one mounted on the vibrator table itself, were checked for correct operation.

The subject (who had previously completed subject pro-forma and thus been cleared for the experiment, see section 2.6) was then brought into the vibrator room and shown the overall system. The various safety precautions built into the rig were explained to him and the actions of the emergency buttons demonstrated. It was stressed that if an emergency did occur then under no circumstances should the subject attempt to jump off the platform. The complete system was "man-tested" and the safety circuits adequate to ensure that during shut-down procedures, no unsafe vibrators would be transmitted to the subject. If he attempted to leave the platform however, then it would be possible for him to trip or catch his foot between the stationary structure and the rig moving downwards under control of the shut down circuits. The subject then sat in the seat on the vibrator and was instructed to fasten his seat lap harness, loosely. It was explained that the purpose of the harness was simply to dissuade him from leaving the vibrator should a fault occur and also to secure him if the table tilted as a result of any such fault. The tests were to be reported as for 'no-harness', so therefore
the harness should not provide any restraint to the vibration. The experimenter then switched on the generator and demonstrated the highest amplitude swept sine input to the subject effectively saying this is all you are going to feel! Then following the procedure laid down in 3.3.1, variations in posture were demonstrated, first under no vibration and then under vibration conditions. It is interesting to note that most of the subjects commented involuntarily on the lack of vibration at the higher frequencies at the head, when the back-off position was assumed. Following the posture changes, the arm and leg variations were explained and demonstrated (3.3.2 and 3.3.3). The accelerometer harnesses were then attached to the subject following the procedures set out in 3.2.1 and 3.2.2. It was explained that during the vibration runs the subject would be required to keep his head and shoulders in the same vertical plane throughout the tests. Any deviation from the vertical would be sensed by the accelerometers as a dc change in the signal and displayed on the output amplifiers, so that a check could be maintained by the experimenter.

Due to the presence of the head harness, the subject was unable to wear an intercom headset. In order to provide communication between the subject and the 'driver', at the control console outside the vibration room, a second experimenter equipped with an intercom set, sat in the vibration room and relayed instructions. This situation also satisfied a recommendation of the safety guide, that a second observer be present during human runs. This second experimenter also instructed the subject on the required posture, arm and leg positions for the particular run and ensured that these conditions were met, as well as visually monitoring the inclination of the accelerometers in the harness.

Recorded tests then began, the subject being instructed by the observer to proceed through the sequence of eight runs (making one test), set out in 3.3.4, at the highest vibration level. If the sweep duration was 25 seconds,
then a test took about 4 minutes, if 10 seconds about 2 minutes. Then followed a brief rest and the sequence repeated at the same vibration level. The input level was then reduced to the medium and then lowest level and the procedure repeated. In all, for one subject the complete set of runs took about 25 minutes. At a conservative estimate - assuming that it would be possible to maintain the required positions for the length of time involved - in order to cover all the variables under sinusoidal testing a time factor of almost 10 (implying 25 hours of testing) would have been involved.

Each subject received exactly the same test conditions in exactly the same sequence, i.e. no attempt was made to balance the design of the experiment to eliminate any order effects. Results of our unreported pilot trials had indicated that the difference in the response curves from any two conditions, was unaffected by the order of these two conditions in the sequence of eight (3.3.4). Clearly the maximum possible time between any two conditions was only about 3 minutes, assuming that any balancing of the design would be done for a constant input level, so that even without the results of a pilot experiment one could safely assume no order effects. The basic objective of balanced experiments after all is that, during a set of tests perhaps of a performance type which can be intuitively assumed to be time dependent in nature, then in order to compare performance under various test conditions, the time factor should be eliminated. For the experiments reported here when the only time effect involved can be fatigue, then it is extremely doubtful whether 3 minutes can make any difference in a subject's fatigue level and hence alter his biodynamic response.
DISCUSSION OF RESULTS

6.1 Presentation of basic results

If a complete overview of the results is taken, then the total number of graphs required to cover all the results (ignoring for the moment the fact that all the runs were duplicated) can be computed from the matrix,

<table>
<thead>
<tr>
<th>Subjects</th>
<th>Conditions</th>
<th>Levels</th>
<th>Ratios</th>
<th>Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>8</td>
<td>3</td>
<td>2</td>
<td>2</td>
</tr>
</tbody>
</table>

Thus, including the duplication of the runs, this could yield a grand total of \(1152\) separate curves! Assuming that the results are grouped into posture variables, arm position variables and leg position variables (instead of the maximum eight possible conditions), and that amplitude and phase can be presented on the same sheet, then the figure reduces to only \(216\). Clearly it is not possible to present all the results and equally as discussed earlier, it is undesirable to calculate the response of an average man and try and define a standard. Therefore a compromise had to be reached.

Inspection of the results in general indicated that the primary features of the response curves were a peak in the amplitude ratio plot at about 4 Hz, a dip around 8 Hz and a second peak around 13 Hz (which was higher or lower than the first peak depending on whether the response applied to head/seat or shoulder/seat). The phase lag curves showed an increase from 0 to \(\pi\) as the frequency increased at the highest input level, whilst at the lower levels, in general, the increase was from 0 to \(3\pi\). Clearly these are only generalised comments, variation in subject, condition, level etc. producing slight variations. However the pattern described above was consistent for all the subjects and all conditions (ignoring for the moment the particular condition, number 8, where the subject leaned slightly forward to take his back off the seat).
On the assumption therefore that this pattern details the important results, Tables were constructed giving the values, for every subject, every condition and every input level, of the maximum and minimum values on the frequency response curves, together with the frequency and phase lag at which these values occur. These values are given in Tables 2 to 5. The results have been further refined to give an average subjective response for every condition and level - Table 6. Comparison of the values in Tables 2 to 5 and Table 6 gives an indication of intra subject variability as far as the maximum and minimum points are concerned.

Results are then presented for a typical subject (the 'typicality' can be judged by inspection of the results, for the maximum and minimum points, for other subjects, as given in the Tables) covering all the conditions and levels tested. These results are presented in detail, as most of the discussion regarding the variation in response for differing parameters will centre around these curves. Comparisons will be made between the trends observed for this one particular subject and the trends for the other subjects (whose complete response curves are not given). Figs.3 to 11 show the head/seat results for amplitude and phase lag for variations in posture, arm and leg positions for the three levels used. Figs.12 to 20 show similar curves for the shoulder/seat results. Comparisons will also be drawn between the average results and the individual results.

It should be remembered that the curves presented were the result of analysis covering two input times (10 seconds and 25 seconds for the input swept function), and two identical runs for each condition. The variation in response found due to these repeats will be discussed later in this section.

6.2 Head/seat results

Figs.3 to 11 cover the head/seat results for subject number 2. Figs.3, 4 and 5 are for the high input level (posture, arm and leg variations respectively); Figs.6, 7 and 8 for the medium level; Figs.9, 10 and 11 for the low level. In general the curves show clearly defined peaks, one at about
3.5 to 4 Hz, with amplitude ratios about 1.5, and the second around 13 Hz with a peak about 1.9. Between the peaks, at about 8 Hz, there is a dip with amplitude ratio about 1.4. This generalization is maintained for the three input levels and the body conditions tested. The phase results for the high input show a fall from 0 to $\pi$ with an upward trend around 8 Hz. At the other input levels the phase lag increases to an overall value of 0 to nearly 3$\pi$.

6.2.1 Posture variables

Figs. 3, 6 and 9, cover the effects of posture for the three levels selected. The most striking feature of the curves is that the line for condition 8 is consistently much lower than for the other postures for frequencies greater than about 7 Hz. Below this frequency this condition gives marginally higher amplitude ratios. As far as the other three posture variables are concerned the slumped posture gives a lower line than normal or erect, especially at the lower amplitudes. However the difference between normal and slumped postures is much less than the difference between normal and back off (condition 8). Clearly the posture variable covering slumped condition could have meant that the subject leaned slightly forward so that he approached the condition of back completely off. Inspection of the results at the first, lower frequency, peak also backs this theory, in that the highest value is for condition 8 and the next highest is for condition 3 which is the slumped posture. The results for normal and erect postures are seen to be virtually identical. As far as the actual values of the curves at resonance are concerned Tables 2 and 3 give the results, Figures 21 to 26 give every subject's results for conditions 1 and 8 only. It can be seen that for the subject chosen, and for normal posture the maximum amplitude, for the first peak, varies slightly for the three input levels from about 1.18 to 1.20 - the frequency at which this amplitude occurs increasing from 3.1 to 3.7 (20% increase). For condition 8 both the maximum amplitude and the frequency at which it occurs increase, 1.42 to 1.75 and 3.7 to 4 Hz with
increasing level. The implications of these increases in terms of non-linearity are discussed in a later section of this Chapter. The dip in the curves all occur around 7 Hz and have an amplitude ratio of about 0.4. For the second peak at the high level, there is a slight inconsistency in our argument in that the slumped results give a higher peak than the normal or erect (2.24 compared with 2.0). However the results at the other levels show the slumped posture to be below the normal and erect. The results for other subjects show either that the normal, erect and slumped postures produce very similar results, or that the trend described here of a lower curve for slumped is maintained. It is obvious that some of the subjects leaned slightly forward in the slumped position, whilst others were able to slump without leaving the back. [It should be remembered that each subject was allowed to select his own interpretation of slumped, and that each subject showed, on his results, that he was consistent in his selection.] The value of the second peak for condition 8 is seen to be only 0.82, i.e. a third of the value for the other posture conditions. At frequencies higher than 13 Hz, the back off posture tends to a value 0.40 whilst the other postures fall at about 12dB/octave. At 20 Hz, the normal and erect values are about 1.10 and the slumped 0.75. These figures are similar for the medium and low input levels. Thus even at frequencies as high as 20 Hz, we have a factor of 3 between the normal posture and the back off condition. Inspection of Tables 2 and 3, which gives the results at the peaks and dip for all the subjects and all the conditions, and Table 6 which gives the averaged data, shows that subject 2 corresponds very nearly to the averaged results. The results for the normal conditions at the first peak and high level show an average of 1.21 with a spread of 1.05 to 1.45 for a spread in frequency of 2.70 to 4.30. Fig.21 gives plots of all the subjects results for condition 1 at the first peak. The variation in the results between the subjects is evident and is probably due to anthropomorphic variations. The problem of trying to explain
intersubject variability will be discussed in a later section. The important point to make at this point in the discussion is that all the subjects exhibited the same trends so that conclusions may be drawn via one subject's results or by the average response.

The results for phase lag show an interesting phenomena which is replicated by all the subjects. At the highest input level, the curves can be approximated to a gradual increase in phase lag from 0 to around π (at about 30 Hz). In the middle of this curve there is a hump as the lag decreases and then falls off again. This hump occurs at about 8 Hz, just above the frequency of the dip in the amplitude ratio plots. Inspection of the phase plots for the lowest input level reveals that an original hump has become extended so that the curves fall to π at 8 Hz and then proceed to fall a further 2π, making a total lag of 3π at 30 Hz. The plots themselves can only go between ±π but clearly the curves should be extended continuously from +π phase lag. The results for the medium level are a mixture of both types of phase response. Other subjects were much more consistent in that high level results went from 0 to π and all other levels gave a 0 to 3π response. The section on non-linearity explains this changeover as a possible result of the change in damping ratio as the level decreases. The results for condition 8 are again different to the results for the other postures. The phase lag is always of the 0 to π type even at the lowest level. Note that again the slumped posture results lie between the normal and erect variants and the back off.

The results of the posture variation experiments have thus revealed a very important factor. If we wish to minimize head vibration at low frequency, and any input level, then it is marginally better to sit erect with back firmly pressed into the seat as opposed to leaning away from the back rest. If on the other hand we wish to minimize high frequency vibration then the subject should be allowed to lean forward and effectively allow the body to attenuate the vibration naturally. The reduction in
vibration at the head, as opposed to sitting erect in the seat can be as much as one-third. It would also appear that there is an advantage in relative phase angle when the subject sits forward. It is interesting to note that the difference in vibration environments mentioned above theoretically, does occur in practice. Low frequency vibration (< 6 Hz) is common in normal fixed wing aircraft, whilst higher frequency (15-25 Hz) vibration is most common in helicopters.

6.2.2 Arm variables

Figs. 4, 7 and 10 cover the effects of varying arm positions on the response curves. Note that condition 1 is included again as our normal reference response. Clearly the effect of varying arm position is very small. At the high and medium input levels conditions for arms folded and arms on oscilloscope appear to give a slightly lower value at the second peak (13 Hz), whilst at the first peak arms folded gives a slightly higher (6%) peak at a slightly higher frequency. The small differences are much smaller than the intersubject differences for the same conditions. Again the slight trend discussed, is evident for the other subjects.

The phase angle plots, apart from the low level input, arms touching the oscilloscope, when the phase reverts to the 0 to π type, are identical. Results for other subjects again show a more consistent effect in that most of the results for high input give a 0 to π response, and most of the results for the medium and low inputs give a 0 to 3π response.

Thus variation in arm position has little significant effect on the vibration transmitted to the head.

6.2.3 Leg variables

Figs. 5, 8 and 11 cover the response curves for the leg positions tested. Again each set of curves contains condition 1 as a reference curve. Comparing the three curves for each level, again there is little difference, so that the remarks made in 6.2.1 regarding variation in condition 1 for the three levels and the actual values involved, again apply. The results at the high level show an increase in the first peak amplitude for the
condition when the legs were in the forward position. The amplitude
increases from 1.22 to 1.56 and the frequency rises from 3.0 Hz to 3.6 Hz.
When the legs are brought right back, the amplitude again rises, but only to
1.27 and the peak frequency falls to 2.75 Hz. At the other levels the same
trend is apparent except that condition 6 - legs right back has a higher
maximum value.

At other frequencies the leg variable curves are 10% lower than the
normal (high input), coincidental except at the actual second peak frequency
(medium level), and for the lowest level 40% lower. Inspection of other
subjects results indicates that the increase in amplitude at the first peak,
when the legs are outstretched, is maintained but the above figure of 40%
is too high to take as an average value for the low level input.

Thus, in general, the effect of varying leg position is to marginally
increase the maximum amplitude at the first resonance point especially when
the legs are outstretched, and to provide slightly more alternation (com­
pared with the normal condition) at frequencies greater than about 15 Hz.

6.2.4 Conclusions

Summing up, the results of the head/seat responses would be that at
low frequencies, a subject should sit well back into the seat with legs in
a normal position (arms having little effect). At high frequencies, sub­
jects can benefit greatly by leaning forward so that their backs are not in
contact with the back of the seat, and may slightly benefit from having
their legs right back or right forward and will find no difference if their
arm position is varied.

The reduction in high frequency vibration when the posture is back
off can be explained in terms of the body being allowed to naturally atten­
uate the vibration. If the back is firmly pressed into the seat back, then
clearly another input, equal in magnitude and phase to the input at the seat
is being provided along the length of the spine - preventing the body from
attenuating vibration by a relative displacement across the spine.
The variations in the responses due to the leg positions were minor and much smaller than any intersubject variability.

Table 6 gives the results for an average subject for all the conditions and levels. It is interesting to note that the individual results in general all gave the same trends as discussed for subject 2. Thus the trends shown in the averaged results can be taken to apply to all subjects. In order to find the absolute values for the peaks and troughs, i.e. to see the spread due to subject variation Tables 2 and 3 should be inspected. As a generalization, taking the averaged results, normal posture, medium level, the curves show two predominant peaks one at 3.5 Hz with an amplitude of 1.25 and the second at 13.3 Hz with an amplitude of 1.83. In between, the curves form a dip at 8 Hz and amplitude 0.30.

The first peak is probably caused by resonance of the internal organs within the rib cage and abdomen and possibly (as will be seen later, the shoulder/seat results show a similar effect) by the shoulder girdle resonating. The second peak is probably spinal in nature. These conclusions regarding the reason for the resonant peaks has been postulated by many authors of past reports.

6.3 Shoulder/seat results

Figs.12 to 20 give the shoulder/seat results for subject 1. Figs.12, 13 and 14 are for the high level (posture, arm and leg variations respectively), Figs.15, 16 and 17 are for the medium level and Figs.18, 19 and 20 are for the low level. Tables 4 and 5 give the values of the maximum (and the minimum) amplitudes together with the frequencies at which they occur for all the subjects all the levels and all the conditions. Table 6 then gives the averaged values for all levels and conditions of the peaks and frequencies. Comparison of subject 1's results with Tables 4 and 5 enables a check to be made of the 'typicality' of his data, and comparison with the average results in Table 6 shows the validity of extrapolating discussed trends of subject 1's results to the average.
Again, as a general comment, the curves show two predominant peaks in the frequency range tested, with a sharp dip between the peaks. The frequencies at which these peaks and dips occur are very similar to the values found for the head/seat tests, the dip in the plots being a slightly higher frequency, 9 Hz as compared with 8 Hz. The frequencies for both the peaks are also slightly higher than the previous results. This time the amplitude of the first peak is considerably higher than for the head results – the Tables of averages showing an increase of about 60%, 1.4 to about 2.2. On the other hand the second peak for postures other than condition 8 is considerably reduced (by between 50 and 75%) from around 1.7 to 0.70. The value of the amplitude ratio at the dip is about the same – approximately 0.40. Again each subject shows very similar results, any variations being a result of a difference in absolute values of the peaks and frequencies rather than a difference in type of responses, i.e. each subject showed the same trend. The phase angle plots are again very similar to the head/seat results. At the high level, the curves fall to around π, rise again (at a frequency near that for the dip in the amplitude ratio plots) and then fall again to level off, at frequencies greater than 10 Hz, to a figure just under π.

6.3.1 Posture variables

Figs. 12, 15 and 18 give the results covering the variations in posture for the three input acceleration levels. It is immediately apparent that condition 8 again gives a completely different curve than the other three conditions. Also there is again a tendency for condition 3 to be the nearest of the remaining postures to condition 8. Inspection of the results for the other subjects bears out this trend. For the particular subject shown, for frequencies higher than about 9 Hz (i.e. the frequency at which the dip occurs) the line for condition 8 is consistently lower than for the other postures, the difference being greatest at the second peak (0.70 compared with 0.37) and least at the highest frequency tested where all postures curves converge. Inspection of the results for other subjects shows a
similar reduction of about 40% at the second peak converging to the same value at high frequency. The average results show a similar reduction. For frequencies lower than 9 Hz, and the high input level, condition 3 (slumped) gives the highest reading - 2.65 as opposed to 2.39 for the back off condition. For the other levels condition 8 has the maximum amplitude (about 2.30), as was the case for the head/seat results. Conditions 1 and 2, normal and erect gave very similar curves for all the levels throughout the frequency range. Inspection of the results for the other subjects indicated that in general condition 8 gave the maximum value at the first peak for all levels followed by condition 3 and then conditions 1 and 2 together (i.e. subject 1 did not follow the overall trend for this particular point). The average table clearly shows this effect. Figs. 27 and 28 and the Tables of results indicates that as the input level decreases, there is a tendency for the amplitude at the first peak to increase (7%) and the frequency at which it occurs to increase (8%). These differences are well within the inter subject variability. Fig. 27 shows the results for the first peak for all subjects and levels for the normal posture, and the maximum amplitude varies between about 1.75 and 2.10 (20%) and the frequency varies between about 3.40 and 5.30 Hz (55%). The questions of inter subject variability and non-linearity will be covered in later sections. This section is primarily concerned with establishing trends due to variations in posture. The phase angle plots resulting from the posture changes present a similar picture to those for the previous head/seat results. At all levels, for this particular subject, all the phase plots lie very closely together. At the high level, the curves fall from 0 to 150° at about 8 Hz, then climb back up to a lag of 70° at 9 Hz, then fall steeply to 160° at 15 Hz and are flat from then on. The plot for condition 8 is slightly different having a similar shape, but respective values of 165°, 80° and 140°. At the other levels, for this particular subject we no longer have a radical difference between condition 8 and the other postures as we had for the head. The
shoulder results are almost identical for all the postures and the two levels of input, medium and low. The curves fall to $-\pi$ at 9 Hz, then fall another $\pi + 140^\circ$ at 15 Hz and again are then flat for higher frequencies. Inspection of other subjects results show that some condition 8 results do not show such a large phase lag but have the 0 to $\pi$ type response even at the lower input levels. The fact that the head results show such a change whereas the shoulder results do not is discussed in the section non-linearity and explained by saying that the head is mildly non-linear whereas the shoulder is much less so thus having less change in damping, to produce a different phase lag plot.

6.3.2 Arm variables

Figs.13, 16 and 19 show the plots of the results for the three arm variables. Condition 1 (normal) is again included as our standard so that cross references may be made between the results for posture, arm and leg changes. Again all arm variables produce a similar plot with two peaks and a dip. Both arms folded (4) and arms on the oscilloscope (7) have higher amplitude ratios than the normal at frequencies higher than 9 Hz, although the frequency at which the second peak occurs is the same for each condition. This is generally true for all the subjects, the tables of averages giving the best picture perhaps where it can be seen that at the second peak the frequencies are virtually identical for all the conditions but arms folded gives a peak 50% higher than the normal (0.72-1.10) and arms touching the oscilloscope gives a peak 65% higher (0.72-1.20). These figures are taken from the averaged data, as the results presented as typical (subject 1) do not really reflect the other subjects results too well. The difference in amplitude at the second peak reduces as the frequency rises until at about 25 Hz the arm variation plots converge on the normal line. Below the second peak frequency the trend is for the plots for conditions 4 and 7 to have a higher value at the dip frequency (approximately 50%) compared with the reference normal condition line. It is difficult to extract a trend from
the results of the first peak as one subject's results appear to contradict another's. This statement must be taken in the context that we are trying to establish a trend in the results for variations which are well inside the variations due to different subjects, i.e. we are saying that, at the first peak, the inter condition variability is much less than the inter subject variability. Again we will take refuge in the averaged results. These show that the normal condition has a peak value consistently (for every level) higher than the other arm positions (1.89-1.65, 1.98-1.75 and 2.05-1.91 for the three levels). This trend appears in some subjects results but is contradicted in others. Tables 4 and 5 give inter subject variations covering the range 1.60 to 2.12. The frequencies at which our first peaks occur are similar for the normal and arms on oscilloscope case but for arms folded are 10% lower. As far as the phase lag is concerned, for the medium and lower levels the arm variations are identical to our normal curve. For the high level the curves for conditions 4 and 7 showed a marked difference around the first peak where they only fall to about 100° whereas on normal curve falls to 150°, apart from this point the curves are very similar.

In order to establish any trends for varying arm position, we have been forced to contradict a basic premise of this Thesis in using the average results. It was found that this was the only way of establishing a trend, which ultimately proved to give a variation in amplitude much less than the variation in the normal condition between subjects.

Thus in conclusion, the variation on arm positions used made very little difference to the amount of vibration being transmitted to the shoulders. There was however an indication that at high frequency arms in lap provided least transmissibility whilst at lower frequencies it is preferable to either fold the arms or touch the oscilloscope.

6.3.3 Leg variables

Figs. 14, 17 and 20 give the results for varying leg position (1 normal, 5 outstretched, 6 right back) for the three input levels. Of all the
results - except the effect of back off in the posture variables - the curves show the most significant effect. Except for frequencies around the first peak, the lines for the three leg variables can be taken as being the same. This applies for all three input levels. However when the legs are stretched out then the amplitude at the first peak frequency is 50% greater than for the normal condition (approximately 2.05-3.10). When the legs are brought right back (condition 6) then the peak is fractionally higher (< 10%) than the normal. This subjective trend is borne out by the figures in the average results table. Subject 1's results indicate that there is a slight tendency for the peak values to increase (5%) as the level decreases, for all the leg positions, this trend is also apparent for the frequency of the first peak. All leg variables give, per level, the same value of frequency for the first peak.

The phase lag curves differ little from the normal plots.

6.3.4 Conclusions

Again summing up, the results for the shoulder/seat tests indicate that the response is very similar to the head/seat results. This time however the first peak has a much larger amplitude, while on the other hand the second peak is considerably reduced. Again if one wishes to minimize vibration levels at the shoulder, then for high frequency the subject should be allowed to lean forward off the seat back whilst at low frequency he should be firmly against the back. As far as arms are concerned, the effect of varying their position was minimal although there were indications that for high frequency the arms should be in the lap and for low frequency they should be either folded or touching an external device like an oscilloscope. The leg variations had no effect over the frequency range except at the first peak, here stretching the legs out greatly increased transmission to the shoulders, and drawing the legs right back slightly increased transmission.

In order to explain these differences, especially those for posture, it should be remembered that very similar effects were found for the head/seat
results, indicating that the mechanism producing these differences is "further down" the chain. As explained in 6.2.4 it is postulated that the difference is produced by allowing or not allowing the spine to naturally attenuate the relative displacement imposed across it. As suggested in the recommendations for future work, these phenomena will be more easily explained when the tests aimed at measuring the response across the spine — by mounting an accelerometer on or near the 7th cervical vertebra — are completed. As far as the differences due to the leg positions are concerned then it is assumed that when the legs are stretched out, they provide a different transmission path to the head or in particular the shoulders, than when the legs are in the normal position.
6.4 Order of systems involved in response curves

One of the easiest, yet crudest and in some cases most inaccurate, way of deciding the kind of system which gives a measured response curve, is to count the number of peaks in the response curve. A single spring-mass-damper system of the type discussed in Appendix A of Chapter 1 and Appendix B of Chapter 2 has one resonance, a two-mass system (Appendix C, Chapter 2) has at least two resonances, etc. Clearly, however, there are limitations to this argument. One can imagine a two-mass system where the two sub-systems have very similar individual responses (although their components governing stiffness and damping may be markedly different) and if the coupling term is small then when the systems are combined then the resulting response curve will be single peaked. Again if the second system is very heavily damped compared with the first system, and possibly has a higher natural frequency - then again only one peak will show. Inspection of the phase response can also aid in defining the number of orders in a multiple system. Fig.14 of Chapter 2 gives the theoretical phase response of a single system and one can logically argue that if, in a multiple system, the sub-systems are sufficiently separated as far as their natural frequencies are concerned, and damping is high, then each part contributes $\pi/2$ (for $\alpha > 5$, say). If however the natural frequencies are close together (their ratio being about 2) and the systems are lightly damped, then each sub-system contributes $\pi$.

For values of $\alpha$ greater than about 2, then the response curve falls off at 6 dB/octave. Thus each part of a multiple system will contribute 6 dB/octave to the final fall off, at a frequency higher than the highest natural frequency. If measured results indicate that the final fall is 30 dB/octave, then five systems are involved in the overall picture. The figure of 6 dB/octave is independent of damping, stiffness, resonant frequency, etc., it is basically an extension of equation (6) in Chapter 1 when $\alpha$ becomes very large. At high frequencies $R$ is proportional to $1/\alpha$ giving a slope of -6 dB/octave for a logarithmic plot.
All the above arguments have assumed a series connection of sub-systems with little coupling, i.e. the factor $\frac{M_2}{M_1}$ defined in equation (C-3) in Appendix C of Chapter 2 is small and negligible. As mentioned in section 2 of Chapter 1 and illustrated in Fig. 8 of that Chapter, if the body does not behave as a 'simple' series network, but rather a combination of series and parallel loops, then the problems of trying to define the characteristics of the system are greatly increased. In a simple series loop the values of the maximum amplitudes and the frequencies at which they occur are directly related to the physical characteristics of the loop and the governing equation for (output/input) has been calculated. If now we superimpose an unknown system as a parallel loop — the governing equation defining its output to the input not being measurable — then only by means of trial and error or intuition (both exceedingly time consuming processes) can one begin to define the sub-systems involved. Any peaks calculated or resonant frequencies found may be reflections of the superimposed parallel system's response rather than the series (and measured) network. Either way, the problems of trying to model the responses are great and the possibility must be faced that if inspection of the response curves (which after all are the basic results which were required from the experiment, rather than an academic breakdown of the system involved) indicates that, using the foregoing logical arguments, (a) the system is not a simple series arrangement, or (b) the system is composed of multiple (>3) sub-systems), then one may just have to accept the response curves as applying to that situation only and abandon any attempts to model the response. The relevant section covering future work will recommend a mathematical exercise to model a typical curve, whilst the designer or engineer would have his basic information ready for use.

As far as the present tests are concerned, for the purposes of arguing about the order of the systems involved, let us assume a standard
response curve. This curve has a peak of 4 Hz (of amplitude 1.40) a trough at 8 Hz (of amplitude 0.40) and a second peak at 13 Hz (of amplitude 1.3). This is simply a theoretical curve covering head and shoulder response for all subjects and all conditions - except condition 8 where the subject leaned forward off the back rest and very probably drastically altered the order of the systems involved.

Theoretically, a response curve which has a resonant frequency at 4 Hz crosses the unity gain axis at a frequency of $\sqrt{2} \times 4$ Hz, or 5.6 Hz. This is true for a single system or for any number of superimposed similar systems. The gradient of the line after this ($f_{\text{max}} \sqrt{2}$) point really determines the number of systems. As mentioned earlier the final fall off will equal the number of sub-systems multiplied by 6 dB/octave. Let us assume that our curves represent a single sub-system which has a resonance at 4 Hz. Our cross over point is therefore 5.6 Hz and the curve falls off at 6 dB/octave. Therefore, by the time we reach the frequency of our second maximum (about 13 Hz) we have increased by about an octave (5.6 Hz $\rightarrow$ 13 Hz) so that we have fallen by about 6 dB. If we assume that we have a simple series connected, two-mass system then the true value of the second peak should be approximately $2 \times 1.3$ or 2.6. Thus we have postulated that our response curves represent a double series system in the following resonant frequencies and maximum amplitudes 4 Hz/1.40 and 13 Hz/2.6. The higher frequency system, when treated in isolation will have an amplitude ratio of roughly 1.11 at 4 Hz, i.e. at the first peak. Thus, in argument our first peak, in isolation, has a true amplitude of 1.27, which corresponds to a damping factor of 0.72 - fairly high damping.

Continuing the argument but relating to the dip in the response curves (which occurs at 8 Hz) then our first peak will only have fallen (assuming 6 dB/octave) by about 30% and our second system will have an amplitude of about 1.50 at the same frequency. Therefore, again simply
series connecting, we arrive at a figure greater than unity at a frequency where experimentally we have a dip. In fact this series connection means that our amplitude ratio curves cannot fall below one between 1 Hz and about 15 Hz. Therefore this cannot be the model to explain our results. The only way that a dip can be generated at around 8 Hz is for our first peak to be the result of more than one sub-system, with similar characteristics. Then the fall off would be $(6\text{dB/octave} \times \text{the number of systems involved})$, but the cross over point of 5.6 Hz would be the same. Thus one can calculate that for a slope of 12 dB/octave, the amplitude ratio following our first peak is 0.55 at 8 Hz, and for 18 dB/octave the figure is 0.40. However, these extreme slopes mean that our second curve now has peak amplitude ratios when treated in isolation of about 5.2 and 10.4 respectively i.e. very lightly damped systems which in turn implies, for the second system again, amplitude ratios of around 1.55 (for 12 and 18 dB/octave) at 8 Hz. When we now series combine our circuits we have at 8 Hz either 0.55 times 1.55 (0.85) or 0.40 times 1.55 (0.62). It also implies that our first peak treated in isolation has a true value of only about 1.20 (0.80 damping ratio)

These figures are certainly less than unity but are a long way from the required figure of 0.40. Intuitively, this logical approach of several superimposed low frequency resonances followed by a lightly damped higher frequency one has advantages and disadvantages. It is well known, and has often been demonstrated by the author to visitors to his vibration facility, that the low frequency resonance of man (around 4 Hz) is brought about by, amongst other things, the resonance of major body organs contained within the rib cage and abdomen. If vibration of a sufficiently high amplitude is used then by slowly varying the frequency between about 3.5 Hz and 6 Hz, the subject can easily identify the oscillations of organs within his own body - such as stomach, lungs, etc. Thus it is extremely likely that our first peak is caused by several similar sub-systems all resonating at about the same frequency and producing one overall peak in the total response
curve, and with a fall off greater than 12 dB/octave. All these sub-systems will have a different connecting point to the major structure of the body, so simple series addition cannot be used. As far as the amplitude ratio curves are involved, in order for our argument to hold across the whole frequency range tested, then we must follow our first peak with a very lightly damped second system (h about 0.05, $R_{\text{max}}$ about 10), with a resonance at about 13 Hz. It is doubtful whether such a lightly damped resonance exists anywhere in the body. Past literature has suggested that this second peak is due to the spine, which intuitively would be regarded as moderately damped.

Inspection of the phase response (taking the response at the medium or low input levels, the change in the shape of the phase response between the high and the other levels will be discussed in the section dealing with non-linearities) reveals that the phase lag has increased to just under $\pi$ by the time we reach the dip in the response curve. Overall the phase lag is around $3\pi$ at the highest frequency tested. Again assuming that our first systems (to give a peak around 4 Hz) are moderately damped, then each sub-system will give $\pi/2$ and it appears that we have two sub-systems. The final phase lag is $3\pi$ so that our second peak needs to contribute $2\pi$. If we again assume a very lightly damped second peak, then $2\pi$ can be obtained by the use of two series circuits. Referring to Fig.14 of Chapter 2, we see that even this is a transient condition and that far enough away from the resonant frequency, the phase curves climb again to a figure between $\pi/2$ and $5\pi/8$.

Summing up it appears that from an amplitude ratio point of view, we have about three systems contributing to the first peak followed by a very lightly damped circuit to give the second peak. The arguments used have assumed simple series connection but intuitively this is unlikely. From a phase lag point of view we have two sub-systems for the first peak,
both heavily damped followed by a further two lightly damped systems for
the higher peak.

Almost all the foregoing argument has assumed simple series connec-
tion and that our frequency range chosen has covered all the major body
resonances. In fact, one of the major points used has been the final
dB/octave fall off. Inspection of the results in general at 25 Hz reveals
that the final fall off is approximately 12 dB/octave - two sub-systems,
suggesting that a complete picture has not been found, as more than two
sub-systems must be involved.

Earlier on, in this section, the point was made that if it was
found that the overall response was not due to simple series addition or
that the system is composed of multiple (>3) sub-systems, then one may
have to accept the results as they are and abandon any modelling techniques.
It is felt that this point has now been reached. Clearly the system is
composed of parallel and series additions and many sub-systems are involved.
Therefore the results are presented for what they are, a record of the
frequency response of the body under numerous input conditions. The
section dealing with recommended future work will be found to contain a
comment regarding the need to perform a computer analysis (both analogue
and digital) of results such as those included in this Thesis, in order to
try to determine the order of the systems involved. Only then will a true
model of the response of the body be found.
6.5 Non-linearity

In the preceding section, dealing with the number of sub-systems involved, the general conclusion was given that the complexity of the situation precluded any real attempt to define the system under test. Any discussions regarding non-linearity can be made (a) on the results as a whole by simply ignoring the order of the system and seeing how the curves vary with varying input level or (b) logical arguments can be made via assumptions regarding the response of simple single systems and fitting the conclusions to the experimental data. As with the preceding section both arguments will be made and conclusions drawn in the light of the findings of both this section and the preceding one.

Basically any non-linearity will show itself by a variation in the response curves for the three levels tested (bearing in mind that there is a factor of 2 to 1 for the input levels used). Clearly it is impossible to plot all the curves for all subjects and conditions so that comparisons between the responses at the three levels can be made. Therefore in order to discuss linearity, an assumption has been made that any effects of non-linearity will be to affect the damping ratio and stiffness of the body (composed of an as yet unspecified number of systems), and therefore to affect the values of any resonances or maxima in the response curves together with the values of the frequencies at which these maxima occur. These changes in $R_{\text{max}}$ and $a_{\text{max}}$ will also be reflected in changes in phase angle plots.

The values for the above maxima for all subjects and conditions are given in Tables 2-5. Inspection of these tables indicates a general trend in the results which is an increase in the maximum amplitudes and the frequencies at which these maxima occur as the input level decreases. A following table, number 6, averages the results and presents the data for the eight conditions and three levels. The trend found for the inter subject results is also shown in the averaged results with a possible exception of condition 3 (slumped posture) for the head/seat results. The trend is apparent for both the head/seat and shoulder/seat results and for both
the peaks. It should be remembered throughout this discussion regarding non-linearity, that changes in values of only about 10% in peak amplitude and resonant frequency for a 50% reduction in input acceleration amplitude are involved. In order to try and understand the problem of quantifying the degree of non-linearity, some of the conditions, contained in the Table, have been plotted as graphs of frequency for maximum amplitude versus the maximum amplitude. The points of these curves have been annotated as being for the high, medium and low input conditions, so that the discussed trend of $R_{\text{max}}$ and $f_{\text{max}}$ both increasing for decreasing input level can be more easily seen. The relevant figures for the averaged results of Table 6 have also been plotted. These results are given in Figs.33 and 34. Also given, Fig.35, is a theoretical curve, assuming a single system, of the variation in $\alpha_{\text{max}}$ versus $R_{\text{max}}$ for various values of damping ratio. By comparing this theoretical curve with the experimental results, an idea of the degree of non-linearity can be made. The results for all the conditions have not been plotted in the above manner because of the large number of figures involved (14 graphs being needed to cover just two conditions). The conditions chosen were regarded as the most important being the standard (normal) position and the condition which, as discussed earlier, produced the most significant result. References to Tables 2-5, provide a check that the other conditions behave in a similar manner.

**Empirical approach**

As mentioned earlier in order to put the whole problem of non-linearity into perspective it must be realised that we are dealing with approximately 10% increases in maximum amplitudes and resonant frequencies for a 50% reduction in input amplitude.

The head/seat results for each subject under condition 1, for the first peak given in Fig.21, indicates that in general the subjects show the trend discussed above. The average results give a very good curve giving a linear
relationship between $f_{\text{max}}$ and $R_{\text{max}}$ for decreasing input levels, a 14% increase in $f_{\text{max}}$ relating to an 8% increase in $R_{\text{max}}$. This figure of "inter level" variability is however much less than the "inter subject" variability for any one level. Inspection of the table for similar results for different conditions indicates that apart from conditions 3 and 8, this trend continues. Both these "rogue" conditions show the increase in $f_{\text{max}}$ but indicate a constant $R_{\text{max}}$ (within approximately 4%). It should be borne in mind that conditions 3 and 8 are interrelated in that condition 3 is slumped posture, where the back may be just starting to leave the back rest, and condition 8 is back completely off, but maintaining an erect posture. The results for the second peak show similar trends (even to the exclusion of conditions 3 and 8 from the general trend) except that the increase in $f_{\text{max}}$ has increased to 25% and the increase in $R_{\text{max}}$ decreased to about 5%. Thus summing up, as far as response to the head is concerned, for the input levels used (in both absolute and relative terms), excluding conditions where the back is off the back rest, the body behaves in a mildly non-linear manner, giving increases in the frequencies at which resonance occurs of about 20% and in the amplitudes at these resonances of about 6%, as the input level decreases. These variations must be related to the inter subject variability, at any one level, of roughly 25% in both $f_{\text{max}}$ and $R_{\text{max}}$. The exclusion of conditions 3 and 8 is of interest as they provide the academic type of system, whereas the other conditions where the subject sits back provide multiple inputs to the body. The actual results for condition 8 plotted on Figs. 22 and 28 show that the first peak results for all the subjects do not follow any particular pattern, but the second peak results show an increase in $f_{\text{max}}$ of about 30% for a very slight decrease in $R_{\text{max}}$. The shoulder/seat results in general do not show to the same extent the trend that the head response shows. The graphs and tables show a much smaller increase in $f_{\text{max}}$ (about 5%) and about 10% increase in $R_{\text{max}}$ for the decreasing levels for the first peak. It is considered that because
(a) these figures are within experimental error variations and (b) they are much smaller than the inter subject variability, even though there is a trend as the level decreases, the body can be regarded as being linear. The second peak has a similar trend as for the head responses, a fairly large increase in *f* max (30%) accompanying a small (5%) increase in *R* max for decreasing levels.

Thus from an empirical point of view, the body can be regarded as mildly non-linear for head response, and essentially linear for shoulder response. The inter level variability for an average subject is always much less than the inter subject variability for one level.

**Theoretical approach**

The basic experimental factor would appear to be that as the input level decreases then the amplitude at resonance and the resonant frequency both increase. Inspection of equations (15) and (16) of Appendix A Chapter 1 indicates that to explain this, then as the input level decreases then the damping ratio must decrease (i.e. lighter damping) and/or the undamped resonant frequency must increase. As the form, or model, of the body's response is not known, then any logical argument will initially be performed via a single system concept, as given in the Appendix. *R* max is simply a function of damping ratio, and hence if the value of *R* max increases then the damping ratio must decrease. Similarly *α* max is a function of damping ratio, but *α* is also a function of *ω* 0, or √(*k/m*), so that *f* max (instead of *α* max) is now a function of √*k* and *h*. Thus inspection of equation (16) reveals that for *f* max to increase then either *h* decreases (which is linked to the increase in *R* max) or *k* increases. In physical terms we are saying that as the input level increases then the system under test becomes more heavily damped (because *R* max decreases and possibly *f* max decreases) and also the stiffness reduces (because *f* max decreases). Intuitively one would expect most physical systems to have hardening characteristics, that is as the extension or relative velocity induced across the system increases then the damping...
and stiffness increase - the ultimate being a member becoming almost infinitely stiff followed by fracture.

Equation (18) of Appendix A chapter I is given as

\[ (\alpha_{\text{max}})^4 = \left[ 1 - \frac{1}{R_{\text{max}}^2} \right] \]  \hspace{1cm} (1)

Let us differentiate this equation and express it as percentage change in \( \alpha_{\text{max}} \) against percentage change in \( R_{\text{max}} \). We then have,

\[ \frac{d(\alpha)}{\alpha} = \frac{1}{2(R^2 - 1)} \]  \hspace{1cm} (2)

to simplify matters we have ignored the suffices given in (1), so in reality \( \alpha = \alpha_{\text{max}} \) and \( R = R_{\text{max}} \).

This equation states that, independent of damping ratio, a change in \( R_{\text{max}} \) is accompanied by a change in \( \alpha_{\text{max}} \), so that our original trend of \( R_{\text{max}} \) and \( \alpha_{\text{max}} \) increasing may be due to the same factor - a decrease in damping, and variation in the stiffness factor may not be necessary to explain the experimental data. As discussed earlier intuition suggests that physical systems have hardening characteristics, so that the earlier suggestion of increasing stiffness with reducing level was suspect.

In the preceding section figures of 14% increase in \( f_{\text{max}} \) and 8% increase in \( R_{\text{max}} \) were stated. If we substitute these values into equation (2), then, solving for \( R \) we find

\[ R_{\text{max}} = 1.14 \]  \hspace{1cm} (3)

Thus we are saying that, assuming a first order system response, then if the maximum amplitude is 1.14, then the percentage changes in \( f_{\text{max}} \) and \( R_{\text{max}} \) can be explained purely on the basis of changing damping (with constant stiffness). The actual values found were between 1.20 and 1.60. Fig.36 show: a plot of equation (2) above for various values of \( R_{\text{max}} \). The preceding section on order of systems involved put forward the theory that the true value of the first peak (when the effect of the second peak was removed) was reduced from 1.40 to 1.27. Examination of Fig.36 reveals that for this sort of figure for \( R_{\text{max}} \) for a multiple system situation, then the experimental
figures for changes in $f_{\text{max}}$ and $R_{\text{max}}$ can be explained as being due simply to a change in damping on the first system.

Summing up the body, for head/seat response, has been found to be mildly non-linear due to a change in damping on the first peak. The shoulder response is regarded as essentially linear. Overall it must be borne in mind that the inter level variability for the average subject is much less than the inter subject variability for any one level.
6.6 Rotational components

Section 3.1.1 of this chapter briefly discussed the problems of rotational vibration, in the context of an experiment which sets out to define the frequency response of the body under linear acceleration input conditions. Reference is made to a paper by Diekmann⁶ (one of the very few papers which attempts to measure head rotation, let alone mention it) which gives plots of the rotational head measurements of standing and sitting subjects under fore-and-aft vibration. These plots show that the head motion plots form a closed figure (indicating that motion is present in at least two linear axes, plus an unknown number of rotational modes), the shape of this loop altering significantly as the frequency is increased. As the shape of the plots is frequency dependent, it can be argued that this rotation is a steady state phenomenon and therefore needs a finite time to develop. The input used for the tests reported in this Chapter—a swept sine wave—is transient in nature and therefore the steady state condition for any one frequency cannot be established.

The frequencies covered by Diekmann's plots of head movement were 1 to 5 Hz, and it is assumed that they were either calculated by summating the outputs of linear accelerometers at the point of measurement or plotted from displacement measures. Neither of these measures can prove rotation of the head, as if measurements are restricted to one point on the head only then the resultant plots can always be the result of more than one translational motion at that point. Rotational acceleration can only be measured by using linear transducers at two positions or by the use of special rotational accelerometers.

Any attempt to define rotational motion (especially at medium or high frequencies—greater than 5 Hz say), by means of displacement methods such as transducers or optical methods is virtually doomed to failure because of the very small displacements involved. As an example let us suppose we have a motion equivalent to ±0.50 g at 7 Hz, and that we are looking for a 20%
rotational effect. The linear displacement necessary to attain this acceleration level is only ±0.100 in, and 20% of this is only ±0.02 in. Considering the image will be vibrating at an equivalent 7 Hz, the problems of measuring and focusing any optical device are large. The other approach is to use rotational accelerometers, these have the problems of being difficult to mount on the head etc. as they are considerably bigger and heavier than the linear accelerometers described in this Chapter. Also once one accepts the need to measure rotational acceleration in a vibration field which also contains linear acceleration, then measurements for all the six possible degrees of freedom are required. Even then the positions of the relevant centres of rotation are required - the assumption being made that they are fixed points. The equations resulting from the use of six transducers pose a severe mathematical problem as the output of any of the linear devices must be composed of the linear motion in the major axis of that particular device plus the resolved components of all the other motions. Thus each output contains six motion unknowns and three radius of rotation unknowns.

As far as this present experiment was concerned it was felt that the transient nature of the input motion would prevent any severe rotational modes from being established and, perhaps more important, there was a need to know the form of the frequency response of vibration to the head and shoulders, in order to be able to draw up limits for use in predicting effects of vertical vibration in terms of performance etc. Any attempt to instrument the head and shoulder in six degrees of freedom would of necessity take a long time whereas an experiment, controlled as far as possible, to minimize the possibility of rotational components could be fairly easily undertaken.

It is significant that the results of our tests showed that the basic form of the response curves was identical for all the levels tested. It is a possibility that head rotation increases markedly as input level increases, as head rotation is probably an involuntary motion due to inertia forces on the mass of the head being applied outside the line of the centre of gravity.
whereas what we are investigating is a biodynamic spring mass effect. The fact that our responses are so similar therefore indicates that head rotation was not a major factor. Visual inspection of the subjects during testing did not indicate any noticeable head rotations, although as said earlier, at high frequencies (>5 Hz) the displacements involved are extremely small.

Therefore, as far as rotation is concerned, the results of this experiment at worst must be regarded as the vertical translational components of the angular accelerations of the head and shoulders, and at best an accurate picture of vertical translational motion at these positions.
6.7 Duration of transient input and repetition of tests

Section 3.1.2 of this chapter explains that two durations of the transient input were used for two frequency ranges, i.e. 1-10 Hz in 10 seconds and 2-25 Hz in 25 seconds, thus keeping to our empirical law of 1 Hz/s for the signal sweep rate. It is important to make sure that when the final response curves are drawn, then in the frequency area where these two ranges coincide, the curves overlap to form one smooth plot, i.e. the frequency responses at the end of one range overlie the frequency responses at the beginning of the other.

The results in general indicated that the total response curves could be drawn by firstly following the 1-10 Hz curve and then smoothly changing across to the higher frequency range, i.e. there were no sharp discontinuities due to a difference in response due to a change in the duration of the input transient. Due to the characteristics of the hydraulic vibrator used - even though the input electrical function had constant amplitude - the amplitude of the acceleration waveform on the table at the low frequency end of the 2-25 Hz range was not constant. Therefore the ranges analysed were 1-10 Hz and 8-25 Hz. Fig.37 shows a typical set of results - from which the final curves were taken. It can be seen that the two frequency ranges merge in together for amplitude and phase, around 6-8 Hz.

It was also explained in section 3 that each run for each subject, for every level and condition, was repeated. Fig.38 is a plot of the two run results and is typical of the repeatability of the analysis. It is considered that the care taken in controlling the posture and limb positions, as well as the short duration of the vibration input contributed greatly to the repeatability of the results. It is however noticeable - as a generalization - in most of the experiments in this Thesis, that intraindividual variability is an order better than interpersonal variability.
6.8 Harness validation

Sections 3.2.1 and 3.2.2 gave details of the harnesses used to measure the accelerations at the head and shoulders. The validation of any such harnesses under vibration is extremely difficult if not impossible. Results from tests may simply be a function of the response of the elastic harness used rather than the response of the object under test, i.e. the head and shoulders. It is difficult to apply a scientific test to such a system and normally the testing of such harnesses comes down to a few *ad hoc* tests which have proved in the past to provide satisfactory measurements.

First of all the harnesses must be tested as mounted on the subject. It is no use mounting the harness on a rigid block and checking it as the attachment would be completely unrepresentative. Secondly the testing must be performed at the tension which will be used in practice. This really introduces our main method - the harness is tightened until the subject says that it is as tight as he would wish bearing in mind that he will be wearing it for a few minutes only. The author has often been amazed at the discomfort subjects will tolerate in order to help experimenters. When the harness is thus tightened it is possible to check its response - the tendency for it to rise under vibration - by slightly lifting the accelerometer, and allowing it to fall back onto the head or shoulder. Normally the harness is assessed this way on a subjective basis, however for the experiment detailed here, the output of the accelerometer was displayed whilst the accelerometer was being lifted and released. Theoretically one is putting a step function into the system and the output should show a decaying sinusoid. The curve forming the envelope of this sinusoid gives an estimation of the damping ratio and the frequency of the sinusoid gives an estimation of the natural frequency of the harness. The results of this test, which were taken from an oscilloscope display only, showed a very highly damped response - so highly damped that it was difficult to estimate the frequency of the resulting sinusoid. It was
only possible to estimate it is around 40 Hz, i.e. well above the frequency range of the tests reported.

Thus, on a subjective basis, relying on the experimenter's past experience, and on an objective basis where the response was found to give a response curve giving unity gain over the frequency of the reported tests, the harnesses were assumed to be satisfactory.

It is also worthy of note that the results indicated that all the subjects gave very similarly shaped response curves. It could be argued that if the harness was significantly contributing to the results, and as the fit was different for each subject, then each subject would give a different form of response curve. The fact that they did not again bears out the assumption that within the accuracy limits of the experiment, the accelerometers were providing an accurate record of the motion of the head and shoulders.
6.9 *Inter subject variability*

In this section we wish to discuss the inter subject variability of the response curves. A lot of the points of discussion have already been covered in sections 6.1, 6.2 and 6.3 which deal with the presentation of the basic results and the effect of the body parameters. However, in general, these sections were mainly interested in any trends due to variations in posture and limb positions, although both average and individual subject results were considered. In order to discuss inter subject variations, particular attention will be paid to the peak (and trough) amplitudes and the frequencies at which they occur — as these are regarded as the major factors of the response curves. Tables 2-5 give the peak and frequency results for all subjects and all conditions, whilst Table 6 gives averaged results. The average results were generated by averaging the absolute value of the peak amplitudes and the frequencies at which they occur, rather than generating an average curve. After all if we have a subject who has a peak at 3 Hz value 2.0 and another a peak at 4 Hz and a value of 3.0, if we average the curves themselves then we will end up with a curve which has two peaks, one at 3 Hz and one at 4 Hz, with peak amplitudes of about 2.5 and 3.5 respectively. This final curve is completely unrepresentative of either subject's results. For this paper we have averaged the absolute values, so that our final figures are a peak at 3.5 Hz with an amplitude of 2.5. It is felt that this presents a more representative method. Following this argument, it is not proposed to give an average response curve. It is possible of course to plot on one graph all subjects' responses for, shall we say, normal posture, head/seat results, at the high input level. The result however would be confusing and untidy and little information could be gathered regarding the behaviour of an average subject. It is better to give the average results at the peaks with the variations taken from the relevant Tables.
In order to discuss the spread of results, the peaks and the frequencies at which they occur for two conditions, normal posture and back off posture, have been plotted and are given in Figs. 21-32. The equivalent results for an average response are given in Figs. 33 and 34.

Inspection of these curves reveals that there is, for certain conditions, a noticeable trend in the plots as the input level decreases. This has been fully discussed in the section on non-linearity and will not be considered here. The fairest way of assessing the inter subject variability is to calculate each subject's averaged (as regards level) peak and frequency and to attempt to correlate these factors against body weight, age, height etc. We shall confine ourselves to considering conditions 1 and 8 only, these representing the normal and the condition which, following earlier sections, gives us the greatest variation from the normal. Table 7 therefore presents these averages for the first and second peaks in the response. General inspection of this Table shows well the difference in responses at the peak for conditions 1 and 8. For both head and shoulders, almost all the results indicate an increase in amplitude and frequency for the first peak when the subject leans forward slightly, whilst for the second peak the dramatic attenuation of the vibration for both the head and the shoulders under condition 8 is very evident. The frequency at which the second peak occurs is only slightly different between the two conditions. Out of all the results shown only one subject (number 1) for one parameter (first peak response at head - difference between conditions 1 and 8) shows a different trend - and even here the trend is very small, a reduction in level of 3.6%, all other subjects showing increases in level of between 10 and 15%. Thus as far as inter subject variability is concerned, with respect to trends, the variation is extremely consistent. This point has also been covered in section 6.1, 6.2 and 6.3. As far as absolute values, per condition, are concerned, then the Table must be examined in conjunction with Table 1 which details subjective data. The requirement is to try and correlate the results with the subject data, so that
statements can be made as to whether the variation in results is due to a variation in weight, height or age or due to normal experimental, subjective scatter. In order to get the numbers involved into perspective Table 7 also contains a column of calculated means (which can be cross referenced to the relevant averages of Table 6). Inspection of these mean values with respect to the individual subject results will give the variation about the mean. The final column in the table gives the maximum percentage deviation of the individual results about the respective means. The percentages are given for the maximum amplitudes and for the frequency at which the maximum occur, for the first and second peak. Values are given for above and below the mean, e.g. for condition 8, head results the mean maximum amplitude and the frequency at which it occurs - for the first peak are 1.25 and 3.59 Hz, the inter subject variability gives variations of +15% and -14% for 1.25 and +12% and -14% for 3.59 Hz. Inspection of the Table generally reveals that the inter subject variability (except for condition 8 second peak) is about ±15%. In terms of any experiment involving objective measures of subjective responses this is an amazingly low figure! The exception quoted above concerns the amplitude ratio for condition 8, back off at the second peak - the variation in frequency for this condition is again extremely low, as are the results for the same condition at the first peak. No obvious explanation is apparent for this ±40% deviation, but again putting the figures into perspective, this deviation is not excessive. One possibility is that each subject had a different interpretation of 'back-off', and thus not all subjects had the same percentage of their back length in contact with the back. This is borne out to some degree in that the frequency variations between the peaks for conditions 1 and 8 are very close and the only real difference is in the amplitude ratios.

Thus having inspected the basic results, the averaged results and the percentage deviations we need to explain a swing in the results of ±15% in

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terms of body parameters. Table 1 gives each subject's height, weight and age. Logically of all these parameters one would expect weight to play the major part in giving any inter subject variability - probably in terms of resonant frequency rather than peak amplitude. However in order to explore all the possibilities (of the conditions given in Table 7), a computer programme was set up to calculate the correlation coefficients between age, weight and height and all the results given in Table 7. The results are given in Table B. The Figures are grouped in pairs, per condition, the first number represents the correlation against the amplitude ratio and the second against frequency for maximum amplitude. Thus the correlation coefficient of subjective weight against the frequency of the second peak at the shoulder and condition 1 is found to be (-0.69). Theoretically the correlation coefficient must lie in the range ±1.0. Large numbers (> ±0.90) representing a good correlation, i.e. a cause and effect; low numbers implying the variables are independent. Inspection of Table 8, line by line, shows that for the height variable there are no significantly high correlation coefficients - considering the experimental spread in height is only about ±4%, this is hardly surprising. As far as weight is concerned, there appears to be a slight tendency for run number 1, to have 'highish' correlation, -0.86 (the negative number merely implies a negative slope so that as the weight increase then the amplitude decreases). It was postulated that there was likely to be a correlation between weight and resonant frequency (given by the even numbers in the table). Unfortunately the results do not bear this out. The age results yield a very interesting result in that run number 10 shows a high correlation coefficient, 0.90. This is for the first resonant frequency at the shoulders when seated normally. The coefficient is positive showing that as one gets older then the frequency increases. The effect disappears when the back is lifted off the seat, and is totally absent at the head.
Thus the only significant correlations found are (a) age, 0.90 (approximately 2% level) against the first resonant frequency at the shoulders when seated normally (Fig.39) (b) weight, -0.86 (approximately 3% level) against the first peak amplitude at the head when seated normally (Fig.40) and -0.78 (approximately 7% level) against the same peak when leaning forward off the back rest (Fig.40). Fig.40 shows a good deviation in the age factor producing a good deviation in the frequency parameter, whereas Fig.39 shows that the deviation of our weight factor, whilst producing a high correlation against the amplitude parameters shown, is very small. In fact if we ignore the weight of subject 3, in Table 1, then our average weight is 15% with a deviation of only +6%/-9%. If we therefore wish to prove any correlation between weight and frequency, a greater range of weight must be used - such as the deviations shown for the age parameter.

Thus our inter subject variability, apart from a few isolated correlated factors must be assumed to be largely due to experimental scatter. Even so the figure of ±15% is small compared with the scatter in past research.
6.10 Subjective comments

In terms of the duration of the experiments, most of the subjects were surprised that such a short test could give the required information. All the subjects agreed that the method of producing the required conditions in terms of posture and limb/positions was good and that as the duration of each condition was so short they had no difficulty in maintaining these conditions. None of the subjects said that they could detect any rotational motion of the head during the vibration phase of the tests.

As far as the vibration conditions were concerned, reactions to the levels were extremely mild ranging from definitely aware of the high level to just barely aware of the lowest level. Clearly, again, the fact that the tests lasted such a short period affected these comments. The only adverse comment made - by several of the subjects - concerned the head harness, after about fifteen minutes this proved to be uncomfortable. However it was explained to the subjects that in order to measure the body's response at the head - as opposed to the response of the head harness relative to the head - then the harness needed to fit very tightly. Following this explanation, the subjects were quite happy to put up with the discomfort for the sake of the tests.

The only other subjective comment concerned the condition when the subjects leaned forward to take the backs just off the back-rest. Several of the subjects commented involuntarily that under this condition the amount of high frequency vibration - which they described as tingling, buzzing and "making them want to scratch their faces" - was reduced appreciably. This subjective comment bears out the objective results which show a marked reduction in the vibration measured at the head between 10 and 25 Hz for condition number 8.
6.11 Comparison with previous results

As explained in Chapter 1, there have been very few past reports which have dealt specifically with the effects of posture and limb position variables on the resonant frequencies and peaks produced by the body, so comparison with previous results is difficult. Also few reports have used frequencies as high as reported in this Chapter and few have mentioned or measured phase lags. Thus we are breaking new ground.

As far as other papers are concerned the first major body resonance is normally reported to be in the range 4-6 Hz with amplification factors of around 2. The results of this paper have shown frequencies of around 4 Hz for the head response and around 4.5 Hz for the shoulder with respective amplitude ratios of around 1.30 and 2.10. Thus we are in general agreement with past researches. When the results offered for frequencies greater than 10 Hz are considered, then no comparisons are readily available. However the indications of most papers is that around 9 Hz the curves are falling, giving little indication of any higher resonances. As far as input acceleration level are concerned Guignard has indicated that shoulder response is linear up to ±0.50 g, and postulates a trend that increasing the level reduces the amplitude ratio for constant peak frequency. Our results bear this out but also include a reduction in the peak frequency.

As far as other Chapters in this Thesis are concerned, the results of the preliminary experiments given in Chapter 2 (for the no harness condition and up to about 6 Hz) agree fairly well with the present results. The results given in this Chapter are to be preferred in that the levels used in Chapter 2, to give results above 3 Hz were those generated as harmonic overtones of fundamentals. As far as shoulder results are concerned the results of Chapter show a two-peak response - as found in this Chapter. Again the results compare favourably with the peaks occuring at around 5 and 15 Hz (Chapter 2) and 4.40 and 13 Hz (Chapter 7). The amplitude levels at the peaks were also comparable. One of the conclusions from Chapter 5 was that variations in the responses were
possibly due to changes in posture and positions. The basic difference between the two Chapters, is that Chapter 5 used sinusoidal inputs whilst Chapter 7 uses transient swept sine wave inputs - with resulting less scatter.

Thus in general, as far as comparisons are able to be made, the results set out in this Chapter agree well with past work. In addition, information is presented in a form, and covering many variables, not available in past literature.
FUTURE WORK AND RECOMMENDATIONS

7.1 "Back-off" response

As far as direct practical application is concerned, then the major finding of this Chapter is that if a subject leans forward, so as not to be in contact with the back rest, then for certain frequency ranges the amount of vibration reaching his head and shoulders compared with a standard input, is reduced as compared with the normal "back-on" case. Clearly if one has a vehicle which exhibits vibration in such a frequency range then something can be done to make the occupants more comfortable - or to perform their tasks more efficiently. The first recommendation would therefore be to validate the responses found in this Chapter for a 'real-life' seat, i.e. one with cushions and flexible members, and then to design a seat back, with some form of sliding mechanism, so that the subject is effectively isolated from the back. The author has performed some unreported tests which appear to validate the back effect in an aircraft ejection seat, and work has been delegated to a subordinate to design a seat back which if used in the correct environment could reduce vibration at the head and shoulders by a factor of at least two.

7.2 Extension of experimental factors

The present investigation has been restricted to the vertical axis only and to acceleration levels between ±2.0 m\(^{-2}\) and ±4.0 m\(^{-2}\). In the real world these conditions cover most vehicles, but sometimes vibration in other axes and at higher and lower levels can occur. In the literature there are many papers dealing with vertical vibration, and few cover the other axes. Therefore it is desirable that this work be extended to cover similar responses but for the other axes and at other input levels.

7.3 Model responses and linearity

Section 6 of this Chapter went into great detail regarding the type and order of system which would produce the experimental results. The conclusions were that the results were too complicated to model for the moment. The same remarks can be applied to the question of linearity. In the end
empirical approaches were used to explain the results. Future work could therefore be done on the results, whereby mathematical models could be set up on computers (both analogue and digital) covering both parallel and series systems, in order to accurately model the response of the body.

7.4 Spinal response

In order to be able to measure and quantify the response of the body at the head and shoulders in terms of modelling and linearity it would be exceedingly useful if measurements could be made at a position within the system. It is proposed to repeat the experiments detailed in this Chapter measuring as an output quantity the acceleration at the top of the spine, on the 7th cervical vertebra. This is the point on the spine which juts out and provides a convenient mounting platform for an accelerometer. Also this vertebra is low enough to be unaffected by any head rotation. The results of such an experiment would also be of interest and use to people concerned with injuries induced by aircraft ejection seats and with setting acceleration or relative displacement limits for such systems.

7.5 Rotational components

One of the fundamental assumptions made in this Chapter is that under the particular type of acceleration input used, the head exhibits linear motion only. It is well known that under steady state sinusoidal input conditions the head can have a rotational component, but the assumption has been made that for the transient input used, no rotation was present (the enormous problems of trying to measure the rotational as well as the linear components - and the ability to separate them, has been discussed in section 6). In order to prove the assumption, it is recommended that an experiment be set up to measure the rotational accelerations induced at the head by a linear acceleration input of the type used in this Chapter.
APPENDIX A: SUBJECT PRO-FORMA.

SUBJECT FILE - CONFIDENTIAL

EXPERIMENT CODE NO. ____________________________

SUBJECT CODE NO. ____________________________

FORM OF DECLARATION - TO BE FILLED IN PRIOR TO EACH EXPERIMENT.

The following questions regarding your health, past and present, and other personal data are intended to help us obtain useful anthropometric/physiological/medical information, and their effects on the results of any test you may perform on the vibration rig. The medical nature of some of the questions does not imply that any experiments you participate in are dangerous (99% of the vibration levels used are less than the levels one encounters in everyday life from walking, riding, public or private transport etc. - you will be told in advance if the other 1% is to be used). People who have had recent transfusions, intestinal operations, a history of back troubles or other medical treatments could be adversely affected by any vibration. The intention of this questionnaire is to find people in normal health who can help us in our experiments.

Finally we are looking for volunteers, nobody is press-ganging you to take part. If you do not wish to volunteer, say so - no reason will be asked. And if you want to withdraw at any time again no-one will ask why.

Any published information will specifically exclude your name.

PART 1

1) Surname: ____________________________ (Mr/Mrs/Miss)
2) Christian names: ____________________________
3) Date of birth: ____________________________
4) Place of birth: ____________________________
5) Address: ____________________________
   ______________________________________
   ______________________________________
6) Height: __________
7) Weight: __________ 7-73
PART 2

As a protection for you (and for the experimenters), against any possible effects of vibration, we have (with medical assistance) compiled a list of conditions which should as a rule render persons unsuitable to take part in experiments involving whole-body vibration, unless lower levels than those occurring in everyday life are used, e.g., perception levels etc. All the questions in this part are intended simply to ensure that any subject we use is in "normal good" health. The answers will be treated as strictly confidential and any written report will contain your code number - never your name.

1) Do you belong to any of the following groups?

Children under 4 years of age.
People with a history of ear or eye surgery.
People with a history of coughing up, vomiting or passing blood.
People with a history of ulcers.
People with a history of Haemorrhoids (Piles)
People who suffer from intermittent pain, blanching or numbness of the fingers.
People who have had a back injury or bad strain, back pain or ejection experience.
Pregnancy at any stage.

YES/NO

If YES please indicate which group.

2) Have you ever been seriously injured or suffered a serious illness eg. a long stay in hospital?
If YES please explain.

3) Are you at present receiving any sort of medical treatment?

If YES please say what type of treatment.

4) Do you suffer from any sort of disability or defect affecting your daily life, work or travelling?

If YES please give particulars.

5) Would you be willing for the Senior Medical Officer at the RAE to be asked for his opinion as to your fitness to take part as a subject in vibration experiments?

6) Have you during the previous month had a vaccination or inoculation or given blood for transfusion?

If YES please give details.

7) Have you any objections to having accelerometers attached to you?

The method of attachment will normally be via an elasticated harness.

8) Do you normally wear contact lenses or spectacles?

9) Any other remarks or information.
To be completed by all applicants and countersigned by the experimenter(s)

1) I ______________________________ hereby volunteer to be an experimental subject in a vibration experiment at RAE Farnborough, Human Engineering Division, Engineering Physics Department.

2) My replies to all the questions above are correct to the best of my knowledge and belief.

3) I understand that the information about myself which I have given and may give in the course of the experiment will be treated as strictly confidential by the experimenter(s).

4) Satisfactory explanations of the nature of the vibration to be used, and how I may stop the experiment, have been given to me.

5) While agreeing to attend for the purpose of the experiment I fully understand that I may withdraw from taking part in the experiment, and that I am under no obligation to give any reason for my withdrawal, or to attend for further experiments.

6) While in the vibration laboratory, I undertake to obey the regulations in force governing its use and safety, subject only to my right to withdraw.

Signature of applicant: ______________________________

Signature of experimenter: ______________________________

Date: ________________
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<td>5.30</td>
<td>20</td>
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**Table 2:** Basic input data for load response (C1, 2, and 3)
|       | 1     | 2     | 3     | 4     | 5     | 6     | 7     | 8     | 9     | 10    | 11    | 12    | 13    | 14    | 15    | 16    | 17    |
|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|
| Peak 1|       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| Dip   |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| R     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| f     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
|       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| Peak 2|       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| Dip   |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| R     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| f     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
|       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| Peak 3|       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| Dip   |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| R     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |
| f     |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |

**Table 1:** Data from field tests for load resistance (SI units and 5G).
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Table 4: A printed data for shoalier response (set 2, 2nd and 3rd)
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### Average Subject sh/st

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The figures given represent values of $R_{\text{max}}/f_{\text{max}}$ for the two major peaks. The % deviations give the maximum variation of the subject data about the mean (for the lowest and highest values).
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Table 8: Inter subject variability, correlation coefficients
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<td>Representation by models of the biomechanical system man-operator under the action of random vibrations. RAE Library Translation 1651.</td>
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Fig 1: Schematic diagram of seat.
Fig 2: Head harness mounted on dummy head.
Fig 3: Frequency response curves of the head, high level, posture variables. S2
Fig 4: Frequency response curves of the head, high level, arm variables. S2
Fig 5: Frequency response curves of the head, high level, leg variables. S2
Fig 6: Frequency response curves of the head, medium level, posture variables, S2
Fig 7: Frequency response curves of the head, medium level, arm variables. S2
Fig 8: Frequency response curves of the head, medium level, leg variables, S2
Fig 9: Frequency response curves of the head, low level, posture variables, S2
Fig 10: Frequency response curves of the head, low level, arm variables. S2
Fig 11: Frequency response curves of the head, low level, leg variables. S2
Fig 12: Frequency response curves of the shoulder, high level, posture variables, S1
Fig 13: Frequency response curves of the shoulder, high level, arm variables. S1
Fig 14: Frequency response curves of the shoulder, high level, leg variables. S1
Fig. 15.1 Frequency response curves of the shoulder, medium level, posture variables. St
Fig. 17: Frequency response curves of the shoulder, medium level, leg variables.
Fig 18: Frequency response curves of the shoulder, low level, posture variables. S1
Fig 19: Frequency response curves of the shoulder, low level arm variables, S1
Fig. 21: Plots of peak against peak for head response (normal condition) for all subjects and levels, 1st peak.
Fig 22: Plots of 1st peak frequency ($f_{max}$) against $R_{max}$ for head response (back off condition) for all subjects and levels. 1st peak frequency ($f_{max}$) and 1st peak amplitude ($R_{max}$) are plotted on a graph.
Fig 23: Plots of $f_{max}$ against $R_{max}$ for head response (normal condition) for S's 1, 2 and 3, all levels, 2nd peak.
Fig 24: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for head response (normal condition) for subjects 4, 5 and 6, all levels. 2nd peak.
Fig 25: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for head response (back off condition) for S's 1, 2 and 3, all levels. 2nd peak.
Fig 26: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for head response (back off condition) for S's 4, 5 and 6, all levels, 2nd peak.
FIG 27: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for all subjects and levels. 1st peak.

1st peak amplitude ($R_{\text{max}}$).
Fig 28: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for shoulder response (back off) condition, for all subjects and levels. 1st peak frequency ($f_{\text{max}}$)
Fig 29: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for shoulder response (normal condition) for S's 1, 2 and 3, all levels, 2nd peak.
Fig 30: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for shoulder response (normal condition) for S's 4, 5 and 6, all levels, 2nd peak.
Fig 31: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for shoulder response (back off condition) for S's 1, 2 and 3, all levels, 2nd peak

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Fig 32: Plots of $f_{\text{max}}$ against $R_{\text{max}}$ for shoulder response (back off condition) for S's 4, 5 and 6, all levels, 2nd peak.

7-117
Fig 36: Frequency for maximum amplitude versus maximum amplitude for various damping ratios, assuming a single mass model.
Fig 37: Effect of two input durations and frequency ranges (same vibration level), same subject and condition (H = 25 secs, 3 - 25 Hz, L = 10 secs, 1 - 10 Hz).

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Fig 39: Age versus average first peak frequency at shoulder (normal condition), per subject.
Fig 40: Weight versus average first peak amplitude at head, per subject.
CHAPTER EIGHT

Conclusions
Inspection of past literature reveals that there is a large gap in the knowledge, regarding the frequency response characteristics of man under conditions of varying posture, limb position and acceleration level in the vertical and other axes. These results are required so that limits can be drawn up to relate permissible vibration levels in vehicles, aircraft, cars, hovercraft etc., to decrement in performance, comfort etc. In order to be able to draw up such limits, information was therefore required on the human body's frequency response between the input (easily measured on the floor of a vehicle) and those parts of the body which are assumed to be most vibration sensitive. This information is required under normal conditions of seating - using a seat with seat base and back rest. In many cases a seating system is interposed between the input vibration and the subject, and clearly any research into the effect of vibration on man, must take into account any possible frequency response of such interposed cushions and seats. Finally in order to be able to find the information detailed above, it would be necessary to construct a new type of vibrator with an improved frequency response and capability.

This Thesis therefore sets out to provide response curves of the human body to various inputs. In particular it gives variations in the characteristics due to changes in posture, harness and limb positions. Although slight indications of non-linearity in the response of the body were found, in general the human body (for the range of vibrations used) can be taken to be linear. In general the degree of inter subject variability has been found to be small, this has been attributed to the strict control of experimental variables, and, in particular for the final experiment using a transient input, to the short duration of the acceleration input. It has been shown that the vibration transmitted to the head, or shoulders, at high (between 10 and 25 Hz) frequencies can be significantly reduced by altering the position of the body so that the subject is no longer is contact with the back rest. Whilst it is impractical to do this in practice, from an engineering point of view, it would be possible to design a mechanism which provides the same effect.
As well as providing frequency response characteristics of the human body, this Thesis contains a section which gives the response characteristics of seating systems, including soft and other forms of cushion. Again information is given in relation to reducing the vibration transmitted to the body at various frequency ranges.

In order to perform all the required vibration tests, a new vibration facility was commissioned and details of performance etc. are included. Also a brief Chapter details the necessary calibration techniques developed enabling an 'on-line' calibration to be performed in the amplitude and frequency range being measured.

Thus, as described in the Abstract, frequency response curves have been given for the response of the body to the head and shoulders. This basic knowledge will therefore enable designers of vehicles which exhibit vibration, to ensure that there is no conflict between the vibration produced in the vehicle and the response of the human body, in terms of comfort or degradation in performance.