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**Crashworthiness of Composite Seats
For Civil Aircraft**

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ABSTRACT

A study has been conducted into the design of civil aircraft seats which are forward-facing and use the lap-belt method of restraint. Within these terms of reference, the response of the seat restraint occupant system (SROS) to impact loading has been analysed using physical (dynamic testing) and analytical (computer simulation) modelling techniques.

With the increasing use of fibre-reinforced polymer composites in aircraft for weight efficiency, and the consequent appearance of composite seats, attention must be given to the crash performance of these structures. Composite structures are characterised by brittle failure with low impact energy absorption, in comparison to the collapse of metal structures which may exhibit plastic deformation prior to failure. However, using the developing technology of composite sub-structures with high specific energy absorption capability, seat structures have been modified to incorporate composite load-limiting elements.

The redesign process involved the compatibility of energy absorber loads with occupant dynamics to minimise injury potential, together with the alleviation of forces in the structural load path to reduce damage and preclude failure of the seat, floor track, and other components.

Shortcomings of existing seat designs were assessed, and the dynamics of lap-belted occupants analysed, including secondary head impact with the forward seat structure. The computer model created was validated against the results of dynamic tests, and then used in a parametric study of occupant dynamics. Conclusions and recommendations include guidelines relating to the future design of both metal and composite seats.

SUMMARY

A study has been conducted into the design of civil aircraft seats which are forward-facing and use the lap-belt method of restraint. Within these terms of reference, the response of the seat restraint occupant system (SROS) to impact loading has been analysed using physical (dynamic testing) and analytical (computer simulation) modelling techniques.

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NOTATION

The co-ordinate system used throughout is as follows:

X : Longitudinal Axis (+ve forward)
Y : Lateral Axis (+ve left)
Z : Vertical Axis (+ve upward)

All abbreviated units use standard SI or Imperial terminology

The following acronyms are used:

CAA : Civil Aviation Authority
JAR : Joint Aviation Regulations
FAA : Federal Aviation Administration
FAR : Federal Aviation Regulations
CIC : Cranfield Impact Centre
CIT : Cranfield Institute of Technology
ATD : Anthropomorphic Test Dummy
CVS : Crash Victim Simulation
NASA : National Aeronautics and Space Administration
CAMI : Civil Aeromedical Institute
NTSB : National Transportation Safety Board
SROS : Seat-Restraint-Occupant System

INTRODUCTION

It is a sad to note that in recent years, the subject of civil aircraft accidents has become more prominent in the United Kingdom, following the 'Lockerbie' and 'Kegworth' disasters. The latter was particularly important to those working in the field of occupant protection, since it was a potentially survivable accident without a post-crash fire. This made inspection of the structural damage possible, increasing the depth of the accident investigation and reconstruction.

When a British Midland Boeing 737 crashed across the M1 Motorway just short of the runway at East Midlands Airport, 38 people died at the scene as a result of the impact, with 74 passengers receiving severe injuries, and 9 further passengers needing hospital treatment.

It is because of accidents such as that at Kegworth that it is vital to ensure adequate levels of occupant protection in the passenger cabins of Civil Aircraft. The study of the impact performance of vehicle structures, sub-structures, seats and restraint systems continues to be a specialist field where the depth and breadth of knowledge and expertise is continually increasing. 'Crashworthiness' is becoming an increasingly familiar word to designers and engineers, and is a parameter by which the impact performance of vehicle structures and features can be assessed.

1 CONCEPTS OF CRASHWORTHINESS AND OCCUPANT PROTECTION

The principle problems of providing protection against injury and (in the extreme case) death to the occupants of vehicles involved in impacts are basically threefold.

Firstly a survival space must be maintained for the occupant, which must be free from intrusion by vehicle structure, flying objects and other unrestrained contents or occupants.

Secondly, the occupant must be restrained within this survival space in a suitable manner, usually in a seat with the aid of a restraint system. Clearly this also entails restraining the seat to the structure, hence the load-path must have no discontinuities.

Thirdly, the loads transmitted to the occupant should be attenuated such that the probability and severity of injury is minimised. This load attenuation may also aid the process of maintaining a continuous load path.

In this introductory section, the areas of study underlying these problems are discussed, previous work is reviewed, with particular reference to Civil (Transport Category) Aircraft, and the degree of relevance is assessed. An overview of the aims of this thesis and the structure of the work programme is also included.

2 HUMAN TOLERANCE AND BIOMECHANICS

In order to assess the effects of impact on the human body, it is necessary to have an basic understanding of biomechanics, including voluntary tolerance levels, injury mechanisms and injury criteria for use in the modelling of human response to impact. The particular use of injury criteria is in the assessment of seat and restraint systems. This is a vast subject, and is only covered very briefly in this section. However, further reading is suggested at the end of this section.

Pre-World War II research relating to the protection of occupants was precipitated by heavy pilot losses during World War I, and resulted in progression from simple lap-only belts to the addition of shoulder belts. Evaluation of restraint systems was achieved using volunteer human subjects at this time.

It is perhaps fitting that the protection of pilots from impact injury, by the development of ejection seats (crash avoidance), yielded data relating to human tolerance of decelerative forces, applicable to crash impact itself. The work performed in Germany during World War II and the immediate post-war period by Geertz (1946) and Ruff (1950) provided the foundations upon which study of human impact tolerance could be based. Eiband (1959) collated the data and produced a series of curves relating the level of allowable acceleration to the duration of such acceleration. These 'Eiband Curves' for whole-body tolerance were based on fully restrained occupants, but drew on data from human and animal subjects without attempting to identify the injury mechanisms involved.

In an effort to produce whole-body tolerance data applicable in Transport Category aircraft, Laananen (1983) analysed available data on lap-belted subjects. Whilst recognising that survivable acceleration levels would be significantly higher, he suggested that a 20g triangular-shape pulse associated with 30ft/s velocity change in a frontal (spinward) impact would be the tolerable limit level. Similarly, 12g with a 35ft/s velocity change in a vertical (headward) impact would be a tolerable level, based on human volunteer testing. However, the need for modelling direct head impacts with seat backs was recognised.

Drawing on data obtained by Ryan (1962) and Swearingen et al (1962), a tolerance level of 15g at 30ft/s has also been proposed for frontal impact using lap-only restraint, by Glaister in 1976.

In addition to whole-body data, research on specific injury mechanisms which occur in vehicle impacts has been performed, in particular related to spinal column injuries, head impacts, chest, pelvis and leg injuries.

Spinal injuries relating to the use of ejection seats led to mathematical modelling of the body as a mass-spring system and the eventual development of the DRI (Dynamic Response Index) as a measure of injury potential, Stech (1969).

However, the degree to which this work can be applied to anything except fully restrained occupants under purely vertical deceleration is limited. A criterion more generally applicable to Civil Aircraft was developed as part of the U.S. General Aviation Safety Panel (GASP) effort, described by Chandler (1985 a). This was a recommendation that the compressive load at the base of the lumbar spine be limited to 1500lb.

Another specific area of study became that of head injury. The collation of data from tests on cadaver skulls, animals and volunteer human subjects led to the production of the 'Wayne State Tolerance Curve' by Patrick et al (1965), relating head acceleration levels to time duration. Manipulation of this data by Gadd (1966, 1971) led to development of an algorithm for assessing the injury potential by weighted integration of the head acceleration-time history over the entire impact duration. This is variously known as the Gadd Severity Index (GSI) or the Head Severity Index (HSI). An acceptable value of the index is regarded as 1000 for frontal head impact, or 1500 for non-contact accelerations.

As a refinement to this system, a Head Injury Criterion (HIC) was developed to weight the higher level, lower duration events more than the lower, longer duration pulses, following work by Versace (1971). The maximum acceptable value of HIC is regarded as 1000 under current legislation. HIC was originally intended only to be used for direct forehead impacts, and not for facial impacts or inertial loadings (Mellander, 1986).

Thoracic injury is a further area where specific injury criteria have been sought. Where shoulder belts are used, belt load at the upper mounting has provided the preferred indicator of chest injury, following work by Foret-Bruno et al (1978). Other cases may be covered by limits such as those contained in Federal Motor Vehicle Safety Standards. FMVSS 208 recommends that chest acceleration

levels be limited to 60g for 3ms maximum, having previously used a Chest Severity Index (CSI) calculated (in the same manner as HSI) using the resultant acceleration of the chest. Chest deflection limits, related to specific test dummy types, have also been used.

Leg injury criteria have also been developed recently. Federal Aviation Regulations (Appendix A) use a limit of 2250lb compressive load in each femur. Criteria have also been proposed for lower legs, including compressive and bending loads in the tibia, knee and ankle joint shear loads, and compressive load in the knee. However, particularly sophisticated dummy components and transducers are required, and hence are not widely used.

Internal injuries as a result of direct trauma are normally categorised as compression or deceleration injuries, and are prevalent in the head and upper torso. Medical aspects are discussed in detail by McSwain (1978).

These criteria are really only of use in the modelling of impacts. This is achieved either by the use of a physical model, with anthropomorphic test dummies as human surrogates, or mathematical (computer) occupant kinematic simulation. These methods are discussed in the following section.

For a more detailed insight into the biomechanics of impact, the work of Chandler (1985 b) should be consulted.

3 HUMAN SURROGATES

As discussed above, the application of specific injury criteria is made possible by modelling of occupant dynamic response to impact. This is achieved by dynamic testing of vehicle interiors and restraints, with representative physical human surrogates, called anthropomorphic test dummies (ATDs). It can also be achieved by the use of complex mathematical modelling of the occupant, together with the seat and restraint system, using crash victim simulation programs such as SOM-LA, MADYMO and Calspan-CVS.

3.1 Anthropomorphic Test Dummies (ATDs)

Early postwar dynamic test work concentrated on restraint systems, using human volunteers, cadavers and animal subjects. Anthropomorphic dummies first appeared in the late 1940s, and with increasing sophistication, their use became widespread. Test dummy development and matching of performance to human response under impact

conditions was conducted by Alderson, 1967. A standard was prescribed for automobile testing in 1973 by the U.S. National Highway Traffic Safety Administration (NHTSA), under Title 49, Code of Federal Regulations Part 572. This 170lb, 50th percentile 'Part 572' (or Hybrid II) dummy was also accepted as the standard for the dynamic testing of Civil Aircraft seats and restraints, with a suitable modification to measure loads in the lumbar spine (Appendix A). This dummy is less sophisticated than the 'Hybrid III' which has recently become the norm in the Automotive Industry. The Hybrid III is equipped to measure neck loads and chest deflection, in addition to the instrumentation used in the 'Hybrid II'.

Dummies are frequently used in dynamic tests where vehicle structure or sub-structure performance is also being evaluated. However, the use of a mathematical model allows a test to be simulated (cost-effectively) many times over, allowing the effects of parametric changes to be assessed.

3.2 Crash Victim Simulation (CVS)

A number of these analytical models exist, with varying degrees of complexity. Early programs were based on two-dimensional models. The occupant is generally modelled with body segments treated as lumped parameter masses, in many cases simply a mathematical model equivalent to the physical model (such as an ATD). The equations of motion are usually solved by numerical integration with variable step size. Provision for a variety of restraint system configurations, together with user-specified support cushion properties is normal. An acceleration-time pulse is then specified at the seat pan, and occupant dynamics are predicted. If a Finite Element seat structure is included, the input pulse may be applied at the floor.

Interaction of the occupant, seat and restraint system is modelled, together with secondary impact with surrounding structure, possibly including folding seat-backs, airbags, and other features of the vehicle interior. Kinematic plots of the occupant at user-specified times during the impact sequence would normally be provided, together with body segment accelerations, belt loads and seat cushion loads, each plotted against time. Injury Criteria would be normally be predicted.

For Civil Aircraft crash analyses, the SOM-LA / SOM-TA programs developed under FAA sponsorship are recommended for occupant simulation and seat structure analysis. These are described in more detail in Appendix D. Many other programs are available, developed commercially or during research, with emphasis placed either on aircraft

or automotive environments, although much common ground exists. Examples are MADYMO, Calspan-CVS, PROMETHEUS, and Ultrasystems CVS-LA. The effectiveness of a selection of these programs was examined by Frisch et al, 1976.

Most effective use of the methods available can be made by a combination of dynamic tests and computer simulation. This allows fewer dynamic tests to be performed, and the effect of parametric changes to be studied quickly and economically using the validated computer model.

4 AIRCRAFT STRUCTURAL RESPONSE

The general field of structural crashworthiness is large but well documented. Of particular interest in this context is the response of Transport Category Aircraft, with special reference to the airframe and under-floor sub-structure. A large proportion of data has been yielded as a result of the on-going FAA Crash Dynamics effort (Appendix C).

The specific concern of this thesis is the passenger cabin, and the influence of the structural response of the aircraft on the impact conditions can normally be approximated by an acceleration-time pulse measured at the aircraft floor, with the inclusion, if applicable, of warped floor conditions.

This simplifying assumption is made for the majority of dynamic test and simulation work. Thus the seat-restraint-occupant system (SROS) is analysed as an entity, and also possibly in component form. The exceptions are when full-scale aircraft impact tests are performed, or in certain cases where fuselage sections are tested complete with instrumented dummies, such as those described by Johnson (1988).

The FAA Crash Dynamics work is described in detail in Appendix C. As a result of this work, velocity envelopes and floor pulses have been defined at the limit of airframe integrity. This provides a context in which to view the results presented herein.

5 ENERGY ABSORBING STRUCTURE

The principal objective in incorporating energy-absorbing structure into a vehicle subjected to impact, is to change the acceleration time-history experienced in the passenger compartment. This normally takes the form of controlled structural collapse, ensuring that peak loads and onset rates are limited to acceptable or survivable values.

The characteristic response of the human body to the high magnitude, low-duration acceleration commonly experienced in vehicle impacts is known as dynamic overshoot, where the acceleration levels of individual body segments exceeds the input acceleration levels, and is a consequence of bodily articulation and less than complete restraint.

The effect of increasing the distance over which the occupant is decelerated can, with proper design, produce lower level of acceleration, with increased duration, which reduce (or eliminate) dynamic overshoot and result in lower injury severity and probability.

5.1 Aircraft structures

The behaviour of subfloor structures, components and sections has been the subject of much investigation. Specifically, protection of helicopter occupants in vertical impacts has been a particular concern of the U.S. Army for many years. Expertise developed in research, together with field experience, led to the publication of the Aircraft Crash Survival Design Guide (Desjardins et al 1980), first published in 1967, which remains the definitive aircraft crashworthiness manual. A description of the systems approach used in current helicopter design was given by Carper (1988).

Due to the relatively large numbers of occupants killed or injured as a result of vertical impact of helicopters, it is not uncommon for such aircraft to be fitted with energy absorbing seats, such as that described by Desjardins et al (1979). This work also contributed to developing guidelines for energy absorbing seats presented in Volume 4 of the Aircraft Crash Survival Design Guide.

The principle of using the available distance (including in some instances a well beneath the seat) to decelerate the occupant, and hence limit deceleration to a tolerable level, increases the impact velocity which can be regarded as survivable, and reduces injury levels at lower impact velocities.

Efforts to produce crushable metal sub-floor elements for helicopters have been reported by Frese (1988), whilst the work of Kindervater (1988) and Farley (1987) has focused on the use of composite materials.

Five alternative types of crushable underfloor beams for light aircraft were compared by Cronkite (1981), who concluded that the lightest designs were those which made efficient use of existing structure rather than additional structure which was largely redundant, except under impact loading.

Energy absorption in the fuselage structure of Transport category aircraft has been assessed as part of the FAA Crash Dynamics effort. Section drop tests indicated deformation of the lower fuselage section and floor peak acceleration levels of 13-30g for wide-bodied aircraft tested at up to 9.2m/s, and 6-12g for narrow-bodied sections tested at up to 5.2m/s. These acceleration levels were in fact equivalent levels, calculated using a triangular pulse from known impulse duration and velocity change. Measured peak levels in many cases were higher (Wittlin, 1985). Further applicable data was obtained in the Controlled Impact Demonstration, Appendix C.

Significant energy absorption capability of the fuselage in the forward direction is unlikely. The characteristics of acceleration-time histories are assumed to be broadly similar to those obtained in full-scale tests, as described by Reed et al (1965). The possibility of axial crushing of the fuselage when impacting a solid vertical object, producing longitudinal loading, was simulated using the crash dynamics program KRASH by Wittlin (1987).

It was also noted that the acceleration pulse may vary significantly at different stations along the fuselage. In vertical impacts, higher levels have been recorded in the proximity of bulkheads and other 'hard spots'. In longitudinal impacts, lower levels have been recorded in the aft sections.

5.2 Energy Absorbing Devices

A number of these devices have been developed, for a variety of applications. Data from over one hundred publications was reviewed by Head (1975) for Helicopter applications. Three main categories were identified, depending upon the process used to absorb energy. These mechanisms were material deformation, extrusion and friction.

The most widely used method is that of material deformation, commonly using the axial crushing of metal tubes by plastic hinge formation, known as crumple tubes. Flattening, expanding, inverting and flaring of tubes each also use plastic deformation of ductile metals to absorb energy, and devices can be designed to operate in tension or compression. The crushing of foams and aluminium honeycomb are also commonly used techniques.

Extrusion-type devices include airbags, where energy is absorbed by air venting through an orifice, and liquid spring devices employing pistons with orifices. Friction devices tend to be rudimentary, and are not particularly suitable for use in impact energy absorbers, as performance can be highly variable.

The use of energy absorbers in aircraft seats is discussed in Section 6. The main terms used for measuring structural energy absorption are illustrated in Figure I1. Additionally, section geometry in tubes is usually expressed as 'relative density', or the ratio of the tube volume to a solid cylinder of the same external dimensions.

5.3 Composite Sub-Structures

With the increasing use of composite materials in aircraft structure, research has been conducted into the impact performance of composite structural elements.

The performance of different materials in impact has often been assessed by the axial crushing of thin-walled tubes, since high specific energy absorption can be obtained. The crushing mechanics of metal tubes has been the subject of a vast amount of research and can be predicted analytically (Wierzbicki, 1983).

A summary of the state-of-the-art for the axial crushing of composite tubes was produced by Thornton et al in 1985. This identifies the three main families of composite fibres, namely glass, carbon and aramid (kevlar). Principally, the former two crush by brittle failure processes, whilst the latter fails predominately by fibre buckling, with fracture processes less prevalent. The processes of energy absorption in composite tubes were described in detail by Hull (1986). Work has also been completed on the axial crushing of composite cones by Price (1987).

Composite sub-floor beams and helicopter structure have already been mentioned. The influence of the use of composites in the airframe structure of Transport aircraft is also being assessed as part of the Crash Dynamics program (Wittlin 1988).

6 CIVIL AIRCRAFT SEAT DESIGN

The development of commercial aircraft seat design was described by Cannon (1986), and is summarised here for completeness. It is possible, from studying the history of civil aircraft seat development, to identify four distinct phases in the postwar development of Civil Aircraft seat design.

6.1 Early Postwar Seats

The first decade after the war saw seats designed for the first time to comprehensive static test requirements. Current JAR criteria are 9g forward (6g to 1952), 1.5g rearward, 6g downward, 3g upward and 4g sideward.

6.2 Seat Development for Impact

The second phase, from about 1955 to 1970, began with interest developing in rearward facing seats, following research by the RAF, Stapp (1951) and others. This research was supported by USAF accident experience, highlighting head injuries as a frequent cause of death in forward facing seats, compared to no deaths reported in rearward facing seats. This interest culminated in a research programme at Wayne State University in 1961 which produced an aft-facing, energy absorbing seat which was both supported from the floor and suspended from the ceiling of the aircraft. However, airline interest proved to be insufficient to see this design in service.

During this period, with the introduction of jet transport aircraft, seat manufacturers began producing energy absorbing seats. Both Aerotherm and Weber produced designs with extendable rear legs which limited the loads at the aircraft floor. Hardman also produced load limiting lap-belts to prevent failure under inertial loading, but the merits of this type of system are undermined by the belt slack created when the system is deployed.

6.3 Weight Reduction in Seat Design

The third phase of development, from about 1970 to 1985, saw attention turn away from the load limiting designs and the heavier seat structures discussed. The arrival of wide-bodied jets and supersonic transport resulted in demands for lighter, more weight-efficient structures. The reductions demanded in seat-spacing pitch and weight resulted in the production of seats designed to static strength requirements, with no controlled distortion or energy management under impact loading.

6.4 Seat Design for '16g'

The fourth identifiable phase, from about 1985 onwards, has seen the effect of the Controlled Impact Demonstration, the study of seat performance in the FAA Crash Dynamics Programme, and the consequent introduction of dynamic test regulations (Appendix A) affect the attitude of seat manufacturers.

In particular, the modification of seats to accommodate warped floor conditions and dynamic loads has instigated redesign. The imminent introduction of composite seats into Civil Aircraft, and the need for such seats to provide occupant protection under impact conditions, instigated the study described in this thesis.

7 ACCIDENT DATA REVIEW

The most wide ranging study and analysis of accident data related to seat performance and passenger injury in recent years was contained in the FAA seat performance and cost benefit study (Cannon, 1986).

Initially seventy severe survivable accidents in the period 1970-1983 were identified from NTSB, FAA and CAMI files. Of these, twenty were identified in which seat performance could have been a significant factor in occupant survivability, fifteen of which were sufficiently well documented for analysis purposes.

These were categorised according to the degree of fuselage damage, Category 1 containing (7) accidents where the cabin remains intact, although other sections may have separated, and/or floor distortion took place. Category 2 had (5) accidents where cabin fractures occurred, but with the cabin remaining basically in-line after the impact. Category 3 contained (3) accidents where several sections of cabin were spread over an area.

Category 1 accidents were characterised by downwards and/or lateral seat deformation, with no seats breaking loose. They were typically landing accidents which were assumed to have no significant levels of forward deceleration.

Category 2 accidents involved aircraft striking trees, houses, or terrain, such that high longitudinal accelerations were generated, with limited vertical or lateral components, typified by seats being torn from their mountings and displaced forward.

Category 3 accidents were typified by multiple impacts and post-crash fire. No specific conclusions were drawn from these 3 accidents.

This report concluded that most serious injuries were caused either as a result of flailing of the body due to the combination of close seat spacing and the lap-belt method of restraint, or as a consequence of the seat structure separating from the floor, and hence secondary impacts.

It was also clear that the seats were not strong enough to cause belt-related abdominal injuries, since no instances of such injury were reported.

It was observed that spinal injuries in Category 1 accidents tended to occur in the upper spine, but were more prevalent in the lower spine in Category 2 accidents. However, the spinal injuries in Category 2 must be seen in the context of seats breaking loose.

Since seat damage was noted in many instances where occupant injuries were absent, this may have been caused by secondary impact of overhead luggage bins which had broken loose.

8 OVERVIEW AND AIMS

The USA has undoubtedly led the world in its efforts to understand the problems of crashworthiness as they relate to Transport Category aircraft. The FAA Crash Dynamics programme has led to a deeper understanding of the many aspects involved and discussed above. The work conducted and described in the main body of this thesis was conducted during the period July 1986 to June 1989, and hence many of the publications cited have only become available during that period. The work, conducted against the background of the FAA effort, was treated as a stand-alone programme concentrating solely on the subject of passenger seat crashworthiness. This was necessary due to the lack of any other related work being conducted in the UK at that time.

The terms of reference were to investigate the behaviour of forward facing seats, with lap-belt restraints only, on adult occupants. Furthermore, to investigate the performance of composite seats, and identify ways of introducing (composite) energy-absorbing devices to minimise injury potential and attenuate peak structural loads. This was felt to be desirable in the light of the characteristic brittle failure of composites structures in impact.

In previous work, load-limiting in civil aircraft seats has been used as a process to ensure seat retention in circumstances of high forward deceleration. An aim of this work was to establish, by analysis of occupant dynamics, whether it was possible to use occupant injury criteria to optimise the energy absorption process for lap-belted occupants, whilst still providing the valuable load-limiting effects to the floor-track, floor beams, and airframe.

The tools necessary to complete these tasks would be a dynamic test facility for seat testing, a computer simulation program for analysis of the SROS (seat restraint occupant system) and static test equipment for component testing, together with small-scale manufacturing facilities for composite sub-structure.

Chapter 1 describes the redesign of a small dynamic test facility at Cranfield Institute of Technology. The testing of available metal and composite seats (with a 50th percentile male dummy) was performed in a variety of test configurations, and is described in Chapter 2.

The use of the FAA-developed computer simulation program SOM-LA (Seat Occupant Model - Light Aircraft) in further investigating impact response of occupants is described in Chapter 3. Comparisons of rigid, non-yielding seats with energy absorbing variants were conducted, and the effects of energy absorber load characteristics upon occupant response were examined.

The investigation of the energy absorption potential of composite materials in the form of tubular elements was conducted and described in Chapter 4. This resulted in the development of a novel kevlar energy absorbing device suitable for use in a composite seat, Chapter 5. This was subsequently patented, see Appendix F.

The dynamic testing of composite seats in standard and converted energy absorbing form was conducted in a variety of configurations, and this work is reported and discussed in Chapter 6. Following these tests, a series of parametric studies were performed using the modified FAA simulation program SOM-TA (Seat Occupant Model - Transport Aircraft), to validate a baseline model and to further study the effects of secondary head impact, occupant initial position and other variables, Chapter 7.

The conclusions reached after each stage of the study are summarised at the end of each chapter, and presented in consolidated form in a separate section at the end of the thesis.

CHAPTER 1

DEVELOPMENT OF A DYNAMIC SEAT TEST FACILITY

1.1 INTRODUCTION

A rig for aircraft seat testing has existed in the College of Aeronautics at Cranfield since 1982, its design and construction having been described by Ashoori (1981) and Hughes (1982). Following completion, it was tested with impact velocities of up to 16.4ft/s (5m/s), after which it stood idle until the commencement of this research study.

The existing rig consisted of a large mass (110kg) falling from a tower and attached using cables and pulleys to a trolley on a 5m long horizontal track, imparting accelerations of up to 0.283g to the trolley. A triple seat unit was mounted on the trolley using standard mounting track, and a single piezo-ceramic accelerometer mounted to the trolley frame. Dummies were not used during the tests. Upon reaching the end of the track, the trolley impacted polystyrene blocks, which were intended to absorb the kinetic energy of the trolley by controlled collapse. The resultant deceleration pulse was intended to simulate the input pulse at the aircraft floor.

Various shortcomings existed in the capabilities and performance of the rig, and thus a process of redesign was initiated to uprate the test facility to the specific requirements of this research work, within the limitations imposed by economics, time and the availability of space.

The aspects in need of redesign were as follows: propulsion system; track length, surface and alignment; trolley design; method of deceleration; test dummy; available test configurations; instrumentation; other miscellaneous minor improvements.

Details of the various redesign aspects are given below, and the upgraded facility is shown in Figure 1.1.

1.2 PROPULSION SYSTEM

The existing propulsion system was not capable of providing the level of impact velocity normally associated with aircraft impact pulses (Appendix C), nor with subsequently introduced dynamic test regulations (Appendix A). The attainable velocities were insufficient, so a feasibility study was conducted to assess alternative forms of propulsion. Ashoori (1981) identified the falling mass system as the best available option. However,

the acceleration level available proved to be too low to attain a realistic impact velocity in the relatively short distance available.

A proposed alternative system of propulsion was to use shock absorber 'bungee' elasticated cord, acting as a catapult, and capable of providing acceleration levels in excess of 1g. An added advantage of using this system was to retain the existing winch. This was because the process of creating tension in the cord, that is, straining it by a large percentage of its original (free) length, created a known quantity of strain energy, almost all of which was used to provide the trolley with kinetic energy. In the case of the falling mass system, a large proportion of the tension in the winch rope was due to the potential energy associated with suspending this mass against gravity force. Much of this energy was dissipated uselessly upon impact of the falling mass (into a sand pit). Hence by using the elastic cord method of propulsion, one could increase the energy imparted to the trolley without increasing the rated load for the winch.

The installation of such a system involved a number of modifications. The cords were anchored at the launch end of the track, attached to the new winch mounting (see Section 1.3). The cords were passed over a specially designed and manufactured roller at the far end of the track, and attached to the front of the trolley. This therefore allowed the cords to be unstrained when the trolley was at the impact end of the track. As the trolley was winched back, the cords were put in tension; when the trolley was in its launch position, the cords were strained by approximately 73%. This strain energy imparted the necessary kinetic energy to the trolley when released. The roller was supported in rolling element bearings and was dynamically balanced to rotate safely at up to 1200rpm.

Facility for use of between one and five cords was made, therefore if 32mm diameter cords were used, it was possible to generate up to 45kJ of strain energy. By using different numbers of cords, and by adjusting payload a range of different impact velocities was obtainable. However, the main restriction placed on the maximum impact velocity was the need to limit the rate at which the trolley was accelerated to speed. This was important, as it affected the posture of the dummy on impact.

1.3 TRACK DESIGN

Various aspects of track design were considered to be inadequate in view of the uprating programme. These included the length of track being used, its surface finish and its alignment.

The main tracks were mounted on I-section beams, which it was not possible to extend without relocating to another building. However, by moving the winch mounting further back, it was possible to extend the usable track length by 1.5m. Allowing for the length of the trolley itself and for its deceleration during the impact stage, the usable length of track over which the trolley could be accelerated to impact velocity was increased to 7.35m.

The existing right-hand track was an inverted vee-track, which was retained and extended. The left-hand track was flat and had been skinned with 16swg aluminium sheet, attached with double-sided tape, which was removed and replaced by 1/4" steel plate, retained by pairs of flush-fitting countersunk cap-head screws at 3" intervals. Both tracks, previous to these modifications, had been covered with thin rubber sheet, attached with adhesive. This was peeling in several places and had perished. It had been installed in an attempt to reduce the 'ringing' which was adversely affecting the performance of the accelerometer. The rubber was removed and the trolley wheels were tyred to compensate for this (see below).

The main I-beam members of the track were found to be badly misaligned. These were 'built-in' to a tower at the impact end and each supported on two short lengths of beam spaced 5m apart. At the launch end, tie rods were used between the floor and the top flange of the beam. Any misalignment was rectified by correct shimming of the supports, with three supports on each beam, and by ensuring that the two beams were level throughout their length. Additional tie rods were used halfway along the track, between the bottom flange of the beam and the floor.

1.4 IMPACT SLED DESIGN

The existing trolley (or 'sled') was retained, but in modified form. It was decided that rubber coating of the trolley wheels, to give solid 'tyres', rather than retention of the rubber covering on the track, would produce a more durable result. The plain type (left) wheels were turned down to allow for the change in ride height caused by the use of thicker metal skinning on the left track. Also, the coating process results in raising the ride height of the vee-wheels by more than the plain wheels, because the thickness of the coating, nominally 1/4", was measured perpendicular to the surface being coated. This was also taken into account when turning down the plain wheels. All four wheels were commercially coated with automotive quality rubber.

At the rear of the trolley, two large vertical steel plates had been welded on to prevent excessive forward

pitching on impact. These would dig into the underside of the track (on the upper beam flanges) to restrict the tendency to 'ride up'. However, this metal to metal contact was considered to be undesirable, due to the shock-loading effects on the trolley and the likely adverse effects on instrumentation performance, as well as the influence on the deceleration signal and damage to the underside of the track. It was therefore decided to remove these plates and install small (3" diameter) tyred wheels to run just clear of the underside of the track, thus preventing excessive pitching but without allowing 'jamming'. These wheels were attached directly below each of the four large running wheels, using specially installed mounting blocks with integral axles.

Prior to the redesign process described in this chapter, the sled was limited to testing seats measuring 20.75" wide at the floor mountings (Boeing 737). A second set of holes were drilled to allow seats of width 22.64" (Boeing 747, Lockheed Tristar) also to be tested.

The final part of the basic trolley redesign was the addition of an impact plate to the front, in line with the change in retardation method. The reasons for this change are discussed in Section 1.5.

In order to perform dynamic tests in a variety of configurations, various seat mounting apparatus was also designed and manufactured during the course of the test programme, which is described in Section 1.7.

1.5 PRODUCING THE CRASH PULSE

The polystyrene blocks used previously (to decelerate the trolley in a controlled manner by crushing) had shown a tendency to break loose on impact (Hughes, 1982). With the higher energy levels proposed, a reliable and repeatable deceleration pulse was felt to be necessary, so other solutions were sought. The possibility of using aluminium honeycomb was investigated, but although a potentially good energy absorbing medium, was rejected on the grounds of cost.

During concurrent work on energy absorption, relating to the controlled collapse of metal tubes, the use of 'crumpling' tubes of specific grades of aluminium alloy as a means of decelerating the impact trolley had been investigated and was thought to be suitable. This also represented an economical option, because tubes in certain grades of commercially available aluminium alloy tubes could be used, with impact behaviour dependant on material specification and physical dimensions. Furthermore, such tubes had been used successfully in a similar application at Middlesex Polytechnic.

The behaviour of metal tubes was predicted using the equations developed by Abramowicz (1984), and it was found that a 3" diameter 16swg aluminium alloy tube would be expected to collapse at a constant load of approximately 40.2kN. Taking account of strain-rate effects, this could be expected to generate a trapezoidal pulse with a mean deceleration level of approximately 10g to a trolley with a gross mass of 450kg.

The suitability of trapezoidal and triangular pulses for recreating crash pulses in dynamic tests has been investigated by Wittlin (1985 b), who found equivalence between a trapezoidal pulse with a plateau of 10g and a triangular pulse with a 16g peak and of the same velocity. The same study gave a representative longitudinal pulse envelope from data measured during the full-scale L-1649 test; this 10g trapezoidal pulse, with a velocity change of 27ft/s (8.23m/s) was very similar to the pulse used for the dynamic tests conducted and reported herein. It was also within the velocity envelope suggested in the post-CID studies, Appendix C.

Once this method of retardation had been chosen, a suitable crush-tube holder was manufactured to hold the tube in a horizontal position and support it during the impact. Provision was made for bolting this assembly to the tower at the end of the track. It was important to position the tube holder at the same height as (or slightly higher than) the centre of mass of the trolley and 'payload'. It was also necessary to allow for an impact plate to be mounted to the front of the trolley (at the same height as the centre of mass) to contact the crumpling tube. Provision was made for a number of different mounting heights, as the height of the centre of mass would vary with the payload and different test configurations (see Section 7 of this chapter). If the height at which the retarding force was applied to the trolley was not coincident with the height of the centre of gravity, an undesirable pitching moment might have been generated on impact.

The crumple tubes required 'pre-buckling' by use of a special tool, to ensure that failure was initiated at the free end, and in the correct failure mode.

1.6 ANTHROPOMORPHIC TEST DUMMY

In order to realistically model occupant dynamic behaviour, a 50th percentile male anthropomorphic test dummy was procured. Although not by any means the latest model available, the Sierra 'Stan' dummy used was still relatively sophisticated, with provision for head, chest and thigh instrumentation, although no instrumentation was

supplied with the dummy. Nylon vertebrae and spring steel ribs were included in the specification, with joint movement controlled by frictional dampers.

For the purposes of testing multi-occupant seat structures, the remaining occupants were modelled using ballast (sacks containing small bags of lead shot). This enabled the occupant motion to be seen more clearly than attempting to distinguish between several dummies, and reduced test costs.

1.7 TEST CONFIGURATIONS

In its original form, the sled facility was designed to test seats in a forward facing configuration only. During the redesign process and during subsequent seat testing, a total of three standardised test configurations were developed, and these are described below:

1.7.1 Test Configuration (A)

The first configuration, designated (A), was with the seat mounted level in a forward facing attitude, with no pitch and no yaw. Due to the minimal seat pitches (longitudinal spacings) commonly used by operators, the back of the seat positioned in front of the occupant is usually within the head strike envelope. For this reason, a seat back was attached to the trolley in the relevant position, as shown in Figure 1.2. The standard breakover mechanism was utilised. The effective seat pitch modelled could have been varied by changing the longitudinal position of the seat prior to the test. A floor pan was also incorporated to ensure correct initial position of the feet, and was carpeted to generate frictional forces. This configuration was intended to reproduce the impact conditions associated with the 'Category 2' type of accident described in the Introduction, where longitudinal decelerative forces predominate.

1.7.2 Test Configuration (B)

The second configuration was based on FAA dynamic test regulations, which at that time were in the form of a Notice of Proposed Rulemaking (NPRM), with the seat forward-facing but pitched rearwards 60 degrees (Figure 1.3). This is described by the FAA as a 30 degrees nose-down aircraft attitude with a vertical velocity change, it could also be regarded as an emergency landing with the aircraft horizontal, with the impact pulse (and hence velocity change) being mostly vertical and partially forward (30 degrees from vertical). This is perhaps an easier scenario to envisage since no pitching moment would occur on impact. Floor warpage was not induced for this test configuration, designated configuration (B).

This configuration could be seen as representing what was thought to be the case in 'Category 1' accidents, where the vertical component of deceleration dominated seat damage over any longitudinal component. See Appendix A.

1.7.3 Test Configuration (C)

The third configuration was with the seat nominally level and forward-facing, but yawed 10 degrees to the left (or right). Additionally, a special jig was used to deform the seat structure quasi-statically prior to the test, to simulate warpage in the aircraft floor. This entailed rolling one pair of legs (in or out) by 10 degrees and pitching the other pair by 10 degrees, in such a way that the test was representative of the worse case, bearing in mind any seat asymmetry. The seat was deformed in situ on the trolley and the jig left in place for the dynamic test, Figure 1.4.

This configuration, designated (C), was also based on proposals then current for dynamic testing of seats, which have since become Federal Airworthiness Regulations (Appendix A). The impact pulse used for all the test configurations was similar, and an example is given in Figure 1.5.

1.8 INSTRUMENTATION

The usual methods of data collection in dynamic test facilities involve the use of umbilical cords dragging behind or above the trolley. It was felt that for ease of operation and maximum reliability, a solid state impact recorder should be designed and built by the Microsystems Design group within Cranfield Impact Centre.

The finished product recorded transducer data during an impact test and then this data could be downloaded to disk after completion of the test using a portable service unit (PC) for later analysis. Four channels were utilised, three for a triaxial accelerometer, which could be positioned in either the head or the pelvis of an anthropomorphic dummy, and a trolley accelerometer to measure the (input) acceleration at the floor.

No trailing wires from the trolley were used. Prior to a test, the channels were initialised (awaiting trigger) via a serial link from a portable PC to the impact recorder. This was then disconnected and cables from the accelerometers mounted on the trolley and in the dummy were connected to the recorder, which was carried on-board the trolley in a purpose-built foam-lined steel box. Shortly prior to the point of impact, the recording was triggered via a small photocell, by a light source mounted

to the track. Recording took place for a total of 1200ms in each test, including a pre-trigger period of 20ms.

After the test had been completed, the serial link was reconnected to download the recorded data to the PC for manipulation and graphical display. The Head Severity Index could be calculated, as could the resultant acceleration time history from the triaxial accelerometer mounted in the dummy, using the post-processing software package. The following graphical output was then available:

Sled acceleration, head resultant and component accelerations. An example of output is given in Figure 1.5.

The recorder was programmable for gain and filter frequency and met the requirements of SAE J921b as specified for dynamic seat testing. Due to problems beyond the control of the Author, this device was not available until shortly before the final dynamic seat tests (see Chapter 6). For this reason, many of the tests conducted during this work (including the majority of those described in Chapter 2) had to be performed without the benefit of dummy instrumentation.

1.9 MISCELLANEOUS FEATURES

A black and white chequerboard was erected to the right of the impact area to provide reference points, and to improve visual contrast when studying high speed film recorded during tests (cameras were normally positioned to the left in the impact area during tests). A high speed clock was manufactured to assist analysis of films and to facilitate comparisons with computer simulation results.

1.10 SUMMARY

(1) In order to produce a dynamic test facility suitable for this research work, an existing but redundant rig at CIT was redesigned and upgraded.

(2) By changing from a 'falling weight' means of sled propulsion, using potential energy, to a system employing elastic cord to store strain energy and catapult the impact sled to speed, higher velocities became attainable.

(3) The relatively short length of track placed a limit on the sled velocity at impact. This was because unacceptably high acceleration levels would have been needed, affecting the occupant posture at impact and hence the subsequent kinematics.

- (4) Modifications to the track and sled produced a facility which was longer, smoother running and with trolley pitching during impact having been eliminated.
- (5) A suitable 50th percentile male anthropomorphic dummy was borrowed for the duration of the research, and a triaxial accelerometer fitted in the head cavity. Sacks containing smaller lead shot bags were modified to act as representative ballast in other seat places. Each sack weighed 170lb.
- (6) A solid-state data acquisition system with post-processing software was procured and installed, after considerable delay. This allowed for automatic calculation of Head Severity Index and resultant head acceleration-time history.
- (7) A repeatable deceleration pulse was generated by the use of metal 'crumple' tubes, at levels thought suitable for aircraft seat testing (see Chapter 2). Adjustments in deceleration level between tests could be achieved by removal or addition of sled ballast, or by selecting an alternative type of crumple tube, with a different mean collapse load.
- (8) A high speed camera was installed at the impact end of the track, allowing for a side view of occupant kinematics and seat dynamic deformation. A chequerboard was installed on the opposite side to provide contrast, and a high-speed clock installed for ease of analysis.
- (9) The trolley was modified such that seats of various widths could be tested longitudinally. A seat-back, with breakover facility, was modified to be installed in front of the dummy occupant at a variety of seat pitches. A carpeted floor was also installed to support the dummy's feet and provide frictional resistance.
- (10) The trolley was modified such that seats of various widths could be tested in a configuration which involved a combination of vertical and horizontal deceleration.
- (11) The sled was modified to accept a jig used to quasi-statically deform the seat structure prior to dynamic test. This was to represent warped floor conditions, with one set of legs rolled 10 degrees, and one set pitched 10 degrees. The seat could then be tested in this condition, longitudinally but with 10 degrees of yaw to provide a lateral component of deceleration.

CHAPTER 2

DYNAMIC TESTS ON COMMERCIAL CIVIL AIRCRAFT SEATS

2.1 INTRODUCTION

A series of full scale dynamic tests were conducted on various commercial and prototype seat designs. Three different test configurations were used, involving components of forward, downward and lateral acceleration, simulated floor warpage and impact of the head with the seat-back immediately in front of the occupant.

A full description of the test rig and the configurations available was given in Chapter 1. Five basic types of metal (double or triple seat-place) seats were tested together with an initial series of tests on the composite design used as the basis for developing an energy absorbing seat (see Chapter 6). The results of ten tests on the four types of metal seats and four tests on composite seats are presented in this chapter, with a further test programme (exclusively on composite seats) discussed in Chapter 6.

2.2 AIRCRAFT DYNAMIC SEAT TESTING RIG

Development of the dynamic test facility is described in Chapter 1. An impact trolley runs on guided tracks 9.5m in length, and normally carries a civil aircraft double or (more usually) triple seat and occupant(s). The seat is attached to the trolley using conventional mounting track. A 50th percentile anthropomorphic dummy is normally used in the centre seat, with shot bags weighing 170lb in each of the other two places, Figure 2.1. The sled is accelerated to impact velocity using elasticated 'bungee' cords, and is abruptly decelerated by aluminium or steel crumple tubes, giving an approximately trapezoidal deceleration-time pulse shape, such as that shown in Figure 1.5.

For almost all of the tests described in this chapter, the dummy was non-instrumented, due to the delay in development and production of the solid state recorder, which was only available from Test 15 onwards. Instead, the tests were recorded on high speed film and post-test damage assessments made. Output was available from a sled-mounted accelerometer for most tests and timing lights were used to measure impact velocity. Three test configurations were used, and are described in the following section.

2.3 TEST CONFIGURATIONS

(A) The first configuration available was with the seat mounted level in a forward facing attitude, with no pitch and no yaw. Due to the minimal seat pitches (longitudinal spacings) commonly used by operators, the back of the seat positioned in front of the occupant is usually within the head strike envelope. For this reason, a seat back was attached to the trolley in the relevant position, as shown in Figure 1.2. Where this additional seat-back was used, the tests were conducted at a seat spacing pitch of 31". A carpeted floor pan was also incorporated.

(B) The second configuration was based on Federal Aviation Regulations for dynamic seat testing, with the seat forward facing but pitched rearwards 60 degrees (Figure 1.3). This was described by the FAA as a 30 degrees nose-down aircraft attitude with a vertical velocity change. A seat back was not positioned in front of the occupant, as it was observed from initial trials that with seats positioned in this test configuration, the degree of upper torso rotation was unlikely to cause secondary head impact (at the load and energy levels used for such tests). Furthermore, a large amount of additional structure would have been necessary with an unacceptable weight penalty.

(C) The third configuration was with the seat nominally level and forward facing, but yawed 10 degrees to the left (or right). Additionally, a special jig was used to deform the seat structure quasi-statically prior to the test, to simulate warpage in the aircraft floor. This entailed rolling one pair of legs (in or out) by 10 degrees and pitching the other pair (up or down) by 10 degrees, in such a way that the test produced the most severe structural load case. The seat was deformed in situ on the trolley and the jig was left in place for the dynamic test, Figure 1.4.

This configuration was based on proposals for dynamic test regulations put forward by the FAA, which were eventually adopted as FAR 25.562, Appendix A.

2.4 SEAT TESTING PROGRAMME

A variety of new and used metal triple seats were provided for testing, by airlines and seat manufacturers, at various stages during the project. Composite seats were initially brought for testing by the manufacturer. Such seats were also provided in component form and used in the later composite seat test programme (see Chapter 6). For these reasons, the tests were not conducted in any particular order, nor was it possible to conduct tests on all of the seat types in each of the three test

configurations. As discussed, the level of instrumentation available also varied, with only the later tests having the benefit of a triaxial head accelerometer for the dummy.

2.5 SEAT TYPES

Tests on five types of seat are described and discussed in this Chapter. The experience gained from commercial testing of other types of seat also proved useful, although cannot be reported here for reasons of confidentiality. The year of manufacture used in the descriptions below refers to the samples tested rather than the production run of the design type. At the time that the tests were conducted, each seat type was still being either used in service or being produced, although in some cases only in prototype form.

Type 1: Twin tubular spar steel/aluminium triple, square section steel legs, tubular construction, mass 31kg. Approximate years of manufacture : 1982-4 (Figure 1.4).

Type 2: Twin spar steel/aluminium triple, round section steel legs, tubular construction, mass 42kg. Approximate years of manufacture : 1979-80 (Figure 1.2).

Type 3: Single spar (A-frame) steel/aluminium triple, round section front alloy legs, I-section tapered rear legs, tubular spar and stringers, mass 31.5kg. Approximate years of manufacture 1979-88 (Figure 1.3).

Type 4: Composite prototype triple, glass/kevlar or carbon/kevlar spar/pan assembly with foam core, integral carbon/foam stringers, carbon fibre/foam core 'A' legs, mass 30kg. Year of manufacture 1987 (Figure 2.1).

Type 5: Sheet metal construction First Class double, box construction legs and seat pan, integral foot supports, arm-stow picnic tables. Year of manufacture 1984 (Not illustrated)

2.6 TESTS CONDUCTED

A total of fifteen tests were conducted, of which five were in each of the three test configurations described above. Dummy instrumentation and data recording were not available at the time that this series of tests were performed.

The tests were recorded on high speed film to assess occupant motion and monitor any structural failures. A video tape recording of these films is available through

Cranfield Impact Centre. The manufacturers' names may be supplied on request.

The tests were all conducted using a 50th percentile dummy in the centre seat and lumped masses (170lb each) in the outboard seats. The seat mounting tracks were replaced with new lengths after each test or as required.

2.6.1 Seat Type 1

TEST 01:

Impact velocity: 8.84m/s
Peak deceleration: 17g nominal
Mean deceleration: 13.1g
Test confign. A (longitudinal)

RESULTS: Plastic hinges formed in each of the front legs, at the point where the bracing members to the rear spars were connected. Due to the left hand weight bias, damage to the left leg was the more severe than to the right (Figure 2.2). Cracking of the track occurred at the front leg mounting locations, but not failure. Minor bending (2 degrees approximately) of the main spars (forwards and downwards) and elongation of rivetted leg-to-spar connections was also evident.

Occupant motion was not particularly clear from the high speed film. Hence a white chequerboard was erected in the impact area to improve the contrast, and the test repeated with a replacement seat (Test 02). A high speed clock, running at 50 Hz (1 rev = 20 milliseconds), was also installed for subsequent tests.

TEST 02:

Impact velocity: 8.84m/s
Peak deceleration: 17g nominal
Mean deceleration: 12.7g
Test confign. A (longitudinal)

RESULTS: This seat was a right handed triple, producing damage which was almost a mirror-image of that in Test 01, except that only surface damage to the seat track occurred. Post-test seat condition is shown in Figure 2.3. The impact duration was about 100ms. If the time at impact is taken as being $t=0$, occupant forward motion began at $t=50\text{ms}$, and ended at $t=200\text{ms}$. The occupant subsequently rebounded slowly towards the initial position. Head contact with the seat back positioned in front of the occupant was a glancing blow which seemed to leave head motion largely unaffected. The maximum rotation of the upper torso from its initial position was about 90 degrees.

TEST 05:

Impact velocity: 8.36m/s
Peak deceleration: 24.0g
Mean deceleration: 15.5g
Test confign. C (longitudinal/lateral)

RESULTS: This seat had a left-hand overhang. As in Test 01, a plastic hinge formed at the point on the left leg where the diagonal bracing strut was attached. Following separation of the front tubular leg from its anchorage, both the rear legs failed in tension at the top welded joint (Figure 2.4). Such a high input deceleration occurred as a result of unexpectedly high crumple tube failure load. It was subsequently found that the tube used was of the wrong type for this test.

TEST 11:

Impact velocity: 8.20m/s
Peak deceleration: 12.6g
Mean deceleration: 7.0g
Test confign. B (vertical/longitudinal)

RESULTS: The lightweight twin spar design was tested in the partially vertical configuration, and sustained minor damage at the 'knees' of the front legs (Figure 2.5). The right hand weight bias resulted in more damage to the right hand front leg than the left, with loads being transmitted by direct contact of the occupant's legs with the spars and through the lap-belt mountings.

2.6.2 Seat Type 2

TEST 03:

Impact velocity: 8.85m/s
Peak deceleration: 15g nominal
Mean deceleration: 11.5g
Test confign. A (longitudinal)

RESULTS: This type of seat uses thick walled tubing for the legs, hence the weight of these seats is significantly higher than the other (more recent) metal designs. This results in a strong structure, and no plastic deformation occurred during the test. However, a large amount of elastic deformation was clear from studying the high speed recording. This occurred up to $t=100\text{ms}$, at which time the seat began to spring back to its initial position. The loads transmitted to the occupant via the lap belt at this time were presumably responsible for the increased severity of the 'jackknifing' motion. The pulse duration was similar to the previous test, as was the duration of occupant motion. Maximum rotation of the upper torso was about 100 degrees.

This seat type was also to be tested in configuration 3, the combined longitudinal and lateral loading with floor warpage. However, inducing 10 degrees of roll in the left floor rail resulted in track lip failure, due to the very high stiffness of the seat structure (particularly in torsion). This problem was compounded by the use of all-metal front track fittings, with no provision for movement about a roll axis. Other types used have featured plastic lock-rings, which limit the track lip loads.

2.6.3 Seat Type 3

TEST 04:

Impact velocity: 8.84m/s
 Peak deceleration: 16g nominal
 Mean deceleration: 12.2g
 Test confign. A (longitudinal loading)

RESULTS: No permanent structural post-test damage was evident on inspection of this single spar triple seat, with the exception of the seat pans. These were made from plastic sheet with wire inserts, and were attached to the stringers by long wire hooks. These had stretched in the dummy seat place, in particular toward the front of the pan.

The seat mounting track was undamaged. Evidence of a significant amount of elastic deformation was obtained from the high speed film. A more severe head impact was noted than in previous tests; otherwise occupant motion was similar to Test 03.

TEST 10:

Impact velocity: 8.19m/s
 Peak deceleration: 12.8g
 Mean deceleration: 6.9g
 Test confign. B (vertical/longitudinal)

RESULTS: The main structure underwent only elastic deformation during the impact test, although partial collapse of the pans occurred, and it was clear from the high speed film that during the impact the main spar would have been loaded by the occupants' thighs.

Monospar designs have been identified as a potential cause of thigh fracture in vertical impacts (Horsfall, 1980) and it would be useful to have a means of measuring the forces between the leg and the spar. However, the dummy used for this series of tests was not equipped with instrumentation to monitor femur loads. For the purposes of secondary impact with the rear spar of the seat in

front (see below), tibia or lower-leg instrumentation would also be useful.

Occupant upper torso rotation was limited to about 20 degrees, and head rotation to about 90 degrees, with the lower legs coming up until almost horizontal. This could cause shin injuries where contact with the rear spar of the seat in front occurs.

2.6.4 Seat Type 4

TEST 06:

Impact velocity: 8.35m/s
 Peak deceleration: 12.5g
 Mean deceleration: 8.3g
 Test confign. C (longitudinal/lateral)

RESULTS: This test was the first on the commercial prototype composite seat. Various modifications to the standard design had been made by the manufacturer in the light of the need to accommodate floor warpage induced prior to testing. Specifically, these were:

- (1) 'Flattening tubes' had been incorporated to limit the tensile loads and hence stretching of the bolts connecting the leg assemblies to the main spar/pan, and to allow a greater degree of torsional deformation.
- (2) Short channel-section deformable metal fittings had been incorporated between the front legs and track fittings. This was found to be necessary to accommodate the sideways movement of the fittings induced by the warping process.
- (3) The rear track-fittings incorporated spherical bearings with lateral pins, to reduce roll moments and hence eliminate floor-track failure due to unequal lip loading.

Unlike the metal seat designs, there was no tie-bar between the bottom of the front and rear legs on these prototype seats. The front fitting consisted of a simple round stud with a lock ring. Thus the friction provided by such an arrangement, together with the stiffness of the 'A' leg, was all that was available to stop the front foot sliding forward under dynamic loading. This proved inadequate in the test, and the front left foot moved forward by two inches, then reached the front of the mounting jig and parted company with it, leaving the seat supported from three points only.

Due to the height of the jig, considerable damage then followed, although this was not representative of the

events which would occur in a real crash unless a floor break had occurred directly in front of the seat. The post-impact damage is shown in Figure 2.6. The test yielded useful information for detail redesign of the front track fittings.

TEST 07:

Impact velocity: 8.34m/s
Peak deceleration: 12.5g
Mean deceleration: 8.3g
Test confign. C (longitudinal/lateral)

RESULTS: Due to the mode of seat failure in Test 06, a simple steel packing piece was bolted to the floor track in front of each of the front feet to prevent forward movement. Whilst not representing a feasible production modification, it would have much the same effect as the metal tie normally employed between the front and rear floor attachment points on seats, for ease of installation. This was done to expedite matters.

Otherwise the seat was identical to that in Test 06, incorporating flattening tubes and 'spherical bearing' rear floor-track fittings as previously described. Failure of the pin at the left rear leg anchorer occurred, resulting in breakage of the right leg assembly and consequently the seat broke free. This was traced back to a two-piece pin design which failed at the point where the two parts were connected together. Figure 2.7 illustrates post-test seat condition.

TEST 08:

Impact velocity: 8.35m/s
Peak deceleration: 13.6g
Mean deceleration: 7.1g
Test confign. C (longitudinal/lateral)

RESULTS: This test was again on the prototype composite design, but incorporating two additional modifications. The spherical bearing rear track fittings (Figure 2.8) were attached to the forked leg fitting using one-piece pins, and small plates were added between the flattening tubes and the bolts (at the seat pan) to spread the loading over a greater area.

The post-test effect of the modification to the method of mounting the front track fitting (using a U-channel) is illustrated in Figure 2.9. The seat survived the dynamic test, but with collapse of the front portion of the seat pan (in front of the spar), see Figure 2.10. Detail, including behaviour of the flattening tubes, is shown in Figure 2.11.

This deformation was accompanied by excessive rotation of the head and upper torso. It was felt that a stretching of the lap belt contributed to this, as did the lack of a floor to support the occupant's feet.

TEST 09:

Impact velocity: 8.35m/s
Peak deceleration: 13.0g
Mean deceleration: 7.5g
Test confign. C (longitudinal/lateral)

RESULTS: The seat used in Test 08 was left in position, and the dummy was repositioned to enable a further energy input to be made. No further significant visible or measusurable damage occurred. Occupant motion was much the same as in the previous test. This test was performed to check that the seat was not close to failure after Test 08.

2.6.5 Seat Type 5

TEST 12:

Impact velocity: 8.7m/s
Peak deceleration: 14.1g
Mean deceleration: 8.1g
Test confign. B (vertical/longitudinal)

RESULTS: Occupant motion studied on high speed film illustrated the problems associated with seat cushions which are thick and of relatively low stiffness. In this situation, the occupant builds up a velocity relative to the seat pan, increasing severity by causing a secondary impact of the thighs and buttocks with the seat pan. This is likely to increase compressive spinal loads in addition to pelvis, chest and head acceleration levels. The seat structure was examined following the dynamic test and found to have no visible damage.

TEST 15:

Impact velocity: 8.67m/s
Peak deceleration: 17.2g
Mean deceleration: 12.1g
Test confign. A (longitudinal)

RESULTS: For this test in configuration A, full dummy head instrumentation and data collection facilities were available. No seat back was positioned in front of the dummy, mainly due to the practical problems associated with the size of the trolley but also because of the large seat pitches used with first class seats. These seats are fitted with fixed seat backs but lap-only restraints.

The output from the triaxial accelerometer is given as three components and a resultant head acceleration trace, and the Head Severity Index was 559. The longitudinal component is given in Figure 2.12, and shows the 'spike' associated with secondary head impact, in this case with the occupant's own thighs.

The lateral 'kick' shown in Figure 2.13 was as a result of hitting one leg fractionally earlier than the other, and clearly shows that secondary impact took place at 164ms. In work conducted on at Cranfield Impact Centre with child dummies, this lateral component has been found to be a common occurrence where secondary impact with the lower limbs (and not a seat-back) occurs.

The vertical component of head acceleration shows relatively low levels between 60 and 100ms, then rising to a level of 25g, Figure 2.14. This level is maintained as the head and upper torso rotate forward until impact with the legs occurs at 164ms, when the change in loading reduces the vertical component to zero by 200ms.

Study of the resultant head acceleration (Figure 2.15) shows that the vertical component dominates in the time until shortly before secondary head impact, with the combination of forward and lateral components dominant thereafter.

The seat structure was again undamaged according to visual inspection following the test.

2.7 DISCUSSION

2.7.1 Seat Type 1

During proving trials on the test facility, this type of seat was tested in configuration (A) without the benefit of high speed photography, and floor-track failure occurred, as a result of the track being in poor condition and perhaps fatigued, having been used previously in various trial runs. When the test was repeated with replacement seat and track, Test 01, some plastic deformation was evident, although the energy absorption associated with this collapse did not take place in a particularly controlled manner. In particular, the design of the front leg mounting to the track could be improved from the point of view of crashworthiness. The bracing member from the top of the rear leg could be attached to the bottom of the front leg, in a lower position. In this way, the front leg could be designed to collapse in compression rather than in bending, and hence absorb a greater proportion of the impact energy.

Prior to Test 05 (configuration C) the seat was deformed statically, the required applied loads being relatively low. Much of the deformation was taken up as elongation of rivets, rivet holes and other fastener connections.

The failure of the front leg mounting during the dynamic test was as a result of its poor design. The anchorage has a single foot with an integral alloy block which fitted inside the box section steel leg, secured by a lateral rivet pin. Because the leg was closed section, it could not pitch (on the rivet pin) relative to the track mounting to accommodate the leg deformation.

Thus when the lower portion of the front leg pitches forward, a moment is transmitted to the foot and the track. This has been found to result in premature failure of the seat or mounting track.

The behaviour in Test 11 (configuration B) was satisfactory, with little elastic and minor plastic deformation, although the transference of loads through the thighs via a moulded pan to provide distribution of the loads may have been preferable to direct contact with the twin tubular spars.

Of note was the similarity, in terms of the mode of seat deformation, between the tests in configuration A and configuration B. This would imply that it may be difficult to assess the nature of inertial forces in accidents from examination of the deformed seat structures.

2.7.2 Seat Type 2

This type was much heavier than the other (more recent) designs, and deformed elastically only, in dynamic testing (configuration A). This situation resulted in high loads being transmitted to the occupants, with no significant structural energy absorption.

It was attempted to test this type of seat in configuration C, but it was not possible to induce the initial 10 degrees of roll in the left-hand legs without causing failure of the seat mounting track. Therefore no dynamic test could be conducted in this configuration. This indicated that the seat would be unable to accommodate significant aircraft floor deformation, resulting in floor-track failure.

It was not considered productive to test this type of seat in configuration B since instrumentation was at that time unavailable, and it would be a less severe structural test than configuration A. It was considered that nothing new of significance would have been learned.

2.7.3 Seat Type 3

In Test 04 (configuration A), elastic deformation of this single spar seat occurred, thus transmitting high loads to the occupants. Due to the wider seat leg spacing (747/Tristar fitment, not 737), it was not possible to test this design in configuration C, as further special jigs would have needed to be made.

Test 10 (configuration B) indicated significant vertical movement of the occupant, raising concern over the loads between the thighs and the monospar. However, it was impossible to quantify injury levels without the necessary instrumentation.

2.7.4 Seat Type 4

All these tests were conducted in configuration C, due to the commercial considerations of the seat manufacturer. It was intended to perform further tests in other configurations and these are described in Chapter 6.

Test 06 highlighted the shortcomings of having a single stud front foot with no positive locational attachment and no longitudinal tie-bar between the front and rear track fittings, but otherwise was of little use.

Test 07 was a simple case of inadequate design of the rear mounting pin. Tests 08 and 09 illustrated that even with a design which survives structurally, occupant behaviour can still cause grave concern, hence the desire to build 'crashworthiness' into this design. The timing of these tests was suited to the manufacturer's commercial requirements but was not optimal from the viewpoint of this research, especially as regards the lack of dummy instrumentation. Further, the fact that the supply of seat components for testing was rather limited after this series of tests, resulted in an increased need for qualitative and intuitive analysis to supplement quantitative methods.

2.7.5 Seat Type 5

This First-Class type seat utilised a strong, stiff structure unlikely to show significant deformation in a survivable accident. It was not possible to conduct a test in configuration C on this seat type, as the warping jig was built to take seats of 20.75" width only, due to the constraints of time in operation during its construction, and the very large amount of extra resources necessary to create a universal mounting system. However, experience of testing seats such as Type 2 points to high loads being transmitted to the floor track when the seat structure is very stiff (particularly in torsion).

The test in configuration B illustrated one of the problems associated with thick, relatively soft seat cushions. The occupant can build up velocity relative to the seat pan, and then suffer from the effects of secondary impact of the buttocks and thighs with the pan. This is compounded by the fact that high levels of deformation provided by thick cushions allow slack in the lap-belt, which may also reduce occupant protection levels.

Test 15, conducted in configuration A, illustrated the case of secondary impact with the lower limbs, as is the case when large seat pitches are used and the velocity change is primarily longitudinal. This was also the first full test which included dummy head instrumentation, giving test data to supplement the computer simulation work discussed in Chapters 5 and 7.

In summary, the impact performance of this type of design is that in survivable conditions and where floor distortion is not a significant factor, the lack of load limiting and the soft thick cushions may increase occupant injury levels. In warped floor conditions, the seat performance is indeterminate. However, the lack of load-limiting elements, the types of track fitting used and the high structural stiffness are not design features which readily lend themselves to the requirement of controlled distortion.

2.8 SUMMARY

- (1) Lack of a pitch release on front track fittings can result in failure to accomodate structural deformation and cause premature seat or floor-track failure.
- (2) Lack of a roll release on track fittings can cause asymmetric loading of the floor-track and prying of the lips, resulting in premature failure in warped floor conditions.
- (3) Single stud track fittings, used almost universally on production seats, can release easily due to minor seat deformation, especially when the light gauge tie-strap between the fittings is absent.
- (4) The use of twin stud fittings such as those commonly used at the rear leg location, providing positive location, clamping, and anti-rattle, with spherical bearing to provide pitch and roll releases, would solve the problems associated with front floor-track fittings.
- (5) In certain twin spar designs, bracing from the rear spar to the front leg should be achieved such that the

strut joins the front leg as close to the floor-track fitting as possible, to allow axial collapse rather than bending failure of the leg.

(6) Warped floor conditions are more easily accommodated in seat structures using a large number of components, because much of the distortion is taken in the connections and fasteners.

(7) Unyielding seat structures which nevertheless allow large elastic deformations during dynamic loading may result in increased severity of occupant kinematics.

(8) The use of single tubular spars may result in increased probability of thigh-bone fractures where vertical components of acceleration are present.

(9) The use of thick cushions for comfort purposes may lead to secondary impact injuries in the upper legs and lower torso.

(10) Seat structural deformation (for a given design of seat) may show similar failure modes from both vertical and longitudinal inertial loadings. This could increase the difficulty in making post-accident assessments of inertial forces, based upon seat deformation.

CHAPTER 3

DEVELOPMENT OF AN ENERGY-ABSORBING SEAT DESIGN USING COMPUTER SIMULATION PROGRAM SOM-LA

3.1 INTRODUCTION

In order to reduce the probability and severity of injury in a aircraft crash, controlled collapse of the seat structure at predetermined loads may be preferable to either a stiff, rigid seat capable of transmitting high acceleration levels to the occupant, or to one which fails in an uncontrolled manner, with the increased risk of secondary injuries. This type of controlled collapse has been achieved in certain military crew seat designs, using a 'stroking' seat unit on guide tubes, with some form of energy absorber or load-limiter resisting such motion, so as to make the best use of the available space in reducing severity. In the case of the Black Hawk helicopter seat, a well was provided beneath the seat to increase the vertical stroking distance available.

In the design of a 'crashworthy' passenger seat suitable for transport aircraft, it is necessary to define the various loading cases likely to occur in potentially 'survivable' accidents. The structure can then be designed in such a way that loads imposed on the occupant are minimised, together with those transmitted to the floor, and efficient use made of any space available for controlled collapse.

Complex analytical tools are necessary and hence computer simulation techniques can be used to examine the dynamics of the seat and the occupant, and the energy absorber properties tuned to lessen the severity experienced by the occupant. Use of program SOM-LA to these ends is described in this chapter. This program is described in detail in Appendix D.

Two stages in model development were completed using SOM-LA, prior to the availability of the more sophisticated SOM-TA variant. The first of these entailed a simple Finite Element single seat model with non-linear spring element front legs, used as a 'first approximation' in the redesign process, followed by a second stage using more accurate geometric modelling, made possible by the provision of this information by the manufacturer.

The decision to use compressive collapse of the front legs as the energy absorption process was reached at a relatively early stage in the research programme, for two main reasons.

Firstly, energy absorption in composite materials was a subject which was relatively new, with the majority of previous work having concentrated upon compressive collapse, mainly the axial crushing of tubes. This research provided an opportunity to apply and develop this new technology to an engineering structure, and gauge its effectiveness.

Secondly, axial stroking of the front legs was thought to have significant advantages over the other options. The main alternative approach would be to develop a tensile energy absorbing rear leg, such as that proposed in the post-CID study (Appendix C). This solution produces a seat which strokes forwards but not downwards, offering no controlled load attenuation under vertical inertial loading, no matter how severe the impact. Also, because the rear leg must increase in length, the seat clearly would produce a greater obstruction to evacuation than a design where the front leg reduces in length. The only significant advantage to using tension-operated rear legs is that it is easier to accommodate lateral strengthening of the seat, without compromising the leg-room of the passenger seated immediately behind.

3.2 STAGE (A): DEVELOPMENT OF A SIMPLE FINITE ELEMENT ENERGY-ABSORBING SEAT MODEL

Having initially studied the principles of energy absorption by use of the simple 2-degree-of-freedom energy-absorbing seat option, and modelling of commercially available seat structures using the Finite Element seat option, the first stage in the composite seat redesign process was to develop a representative model of an energy-absorbing civil aircraft seat.

The objective of Stage (A) was therefore to develop a simple Finite Element model with energy absorbing (non-linear spring element) front legs, laterally pin jointed at each end, with the rear seat structure behaving as a mechanism pivoting on lateral pins at the rear anchorers.

The simplicity of such a model allowed for an economic means of studying the seat redesign problem, and was the first stage in the process of model development.

3.3 STAGE (A) SIMULATION PARAMETERS

Simplified, idealised deceleration-time signals were used in the previous studies, based on the FAA dynamic test regulations, (Appendix A). The first of these was a 'horizontal' deceleration pulse, triangular in shape with a 16g peak occurring after 90ms, total duration 170ms, and

representing a change in horizontal velocity of 44ft/sec (13.4m/s). The seat and occupant were orientated to be forward facing but yawed 10 degrees, similar to configuration (C) used in the dynamic testing (Chapters 2 and 6).

The second pulse was a 'vertical' deceleration, triangular in shape with a peak of 14g occurring after 80ms, total duration 155ms with a vertical velocity change of 35ft/sec. The orientation of the seat and occupant is 30 degrees pitched down, similar to configuration (B) used in the aforementioned dynamic tests.

These input pulses (as measured at the aircraft floor) were used as 'yardsticks' for the initial development work performed using computer simulation. It was attempted to run each simulation for 210ms, although the program is configured to stop at the point where head rotation (flexion) exceeds 90 degrees, as the equations of motion for the occupant become undefined at this point. This was the case for all the 'horizontal' simulations due to the type of restraint used (lap belt only). This proved to be a significant drawback in the usefulness of the programme.

The following parameters were fixed for this series of simulations:

- Simulation time period: 210ms.
- Lap belt restraint only, fixed to seat.
- Belt force deflection characteristics.
- Cushion force deflection characteristics.
- Occupant initial position in seat.
- Seat undeformed geometry.
- Seat structure FE model, except for front leg structures.

The following parameters were fixed for all 'horizontal' pulse simulations:

- Seat horizontal, yawed 10 degrees to the right, 44ft/sec change in horizontal velocity, triangular deceleration pulse, peak 16g occurring after 90ms. Total pulse length 170ms (see Figure 3.1).
- Three-dimensional occupant model, based on Part 572 anthropomorphic dummy - needed due to lateral component of deceleration.

The following parameters were fixed for all 'vertical' pulse simulations:

- Seat horizontal, pitched 30 degrees forward, 35ft/sec change in vertical velocity, triangular deceleration pulse, peak 14g occurring after 80ms. Total pulse length

155ms (see Figure 3.2). Three-dimensional occupant model, also based on part 572 dummy - not strictly necessary at this stage, but used due to the limited number of runs performed.

The following parameters were varied for all simulations:

The element type for the front legs (beam or non-linear spring).

The spring characteristics for the front legs, defined by an elastic stiffness, a first plastic yield point, a first plastic modulus, a second plastic yield point, a second plastic modulus, and an ultimate strength, see Section 3.5. Elastic unloading, at any stage, takes place at the same elastic stiffness as that specified for loading.

3.4 STAGE (A) FINITE ELEMENT SEAT MODEL

The seat model was created using a minimum of nodes, elements, materials and cross-sections, and served as a 'first approximation'. The initial 'rigid' model had beam element front legs, and hence was not an energy-absorbing design. This served as a baseline for assessing the influence of changing the properties of the spring element front legs.

The basic seat model is shown in Figure 3.3. The seat back and seat pan cushions were 3" thick. The nodes at the seat pan to seat back joints were also used as the lap belt anchorages. The lap belt had zero initial slack. As the front spring elements were the members of interest, the remaining elements were all modelled as 1" square thin-walled steel tubing, to minimise computational costs. The front legs of the 'rigid' seat model were also of this type.

Lateral pin joints at the four floor attachments were specified, and also at the tops of the front spring elements. Further elements joining the front of the seat pan to the top of the seat back were included. These would ideally have been modelled as spring elements, to simulate the rotational friction dampers situated at the seat back hinges, but modelling of 'breakforward' was considered to be an unnecessary complication at this stage.

Various types of spring element front legs were simulated and three such types, designated EA1, EA2 and EA3, are described in the following section.

3.5 STAGE (A) SIMULATION PROCEDURE

A basic, non-energy absorbing ('rigid') beam element seat model was assembled as described above, and simulations performed with each of the two input deceleration pulses described previously.

The beam element front legs were then replaced by non-linear spring elements. These were defined as shown in Figure 3.4. The 'horizontal' simulations were regarded as the 'worst case' (based on the dynamic test results) and hence were performed first, with the 'vertical' pulse simulations run afterwards as a check. It was decided to begin by using a low value of first plastic modulus to simulate a roughly constant collapse load, and the second plastic modulus to simulate 'bottoming out' or exhaustion of the available stroking distance. The elastic loading/unloading stiffness was kept constant throughout at 6000lb/in. Initially the mean load was set at around 700lb, this was a figure approximated from hand calculations to serve as a starting point. Clearly the load would be sensitive to the geometry employed, and hence some initial experimentation was necessary.

The first problem encountered was the refusal by the program to accept negative or low positive values of first plastic modulus. It was desirable to achieve this in order to create constant collapse regions (or other collapse characteristics) which had been possible using the energy absorbing seat option used in previous simulation work. A value of 30lb/in was accepted, but values of 10lb/in resulted in ill-conditioning of the stiffness matrix.

The first non-linear spring characteristic used is referred to as 'EA1'. The initial collapse load (first plastic yield point) was 600lb, increasing to 700lb at a rate of 30lb/in, followed by a second plastic modulus of 100lb/in.

Type 'EA2' had the same plastic moduli as before, but with the yield points at 550lb and 650lb. Type 'EA3' was the same as 'EA1' except that the second plastic yield point was increased to 800lb, with the region of almost constant plastic collapse doubled from 3.33" to 6.67".

Attempts to increase the initial collapse to 800lb resulted in no stroking, producing an effectively rigid design.

3.6 RESULTS OF STAGE (A) SIMULATION

3.6.1 'Horizontal' simulations

Two types of problem encountered with the program SOM-LA affected the degree of success in obtaining meaningful results from this part of the study. Firstly, the problem of limited allowable head rotations made lap-belted occupant simulations, with severe longitudinal deceleration pulses, prone to abortion at a relatively early time during the simulation run. Secondly, dubious performance of the 'energy absorber' elements produced higher than expected stroking distances and reductions in the apparent load-carrying capability of the elements during simulations.

None of the simulations ran beyond 165ms, due to the head rotation limitation described above, whilst the 'rigid' seat ran to only 150ms. Hence the time at which head impact with the lower limbs would have taken place was never reached. It was therefore evident that the limitation of this set of simulations was in that it indicated the possible reductions in loads achievable prior to head impact.

In this context, it can be seen from Table 3.1 that both Chest Severity Index and Head Severity Index were significantly reduced in the case of the Energy Absorbing designs. It was also indicated that seat belt loads would be reduced by about 12%. Plots of typical occupant kinematics in the horizontal configuration are shown in Figure 3.5. The information relating to the energy absorber 'stroke' was dubious for reasons already mentioned. A full discussion of this particular problem is contained in Appendix E.

3.6.2 'Vertical' simulations

The vertical simulations were performed mainly as a check that no undesirable effects were produced as a result of the changes made from the previous simulation work. The results for each of the Energy Absorber types were very similar, and hence only those from 'EA3' have been included here. The reductions in body segment deceleration levels and severity levels (Table 3.1) were fairly modest, but as explained previously the 'worst case' was the 'horizontal' simulation, when a lap belt only was fitted. Lap belt loads for this were not significant, being below 700lb in all simulations. Typical occupant kinematics for an energy absorbing seat in the vertical configuration are illustrated in Figure 3.6.

3.7 STAGE (A) SUMMARY

- (1) The simple yet representative FE model was created using SOM-LA to serve as the first step in a redesign process to incorporate energy absorbing elements in a composite seat, hence improving its crashworthiness.
- (2) Non-linear compression spring elements were used to model energy absorbing front leg elements, and comparisons made with 'rigid' beam element front legs.
- (3) Input decelerative pulses based on FAA dynamic test regulations were used in these comparative studies.
- (4) As a consequence of pulse severity, the program stopped (due to excessive head rotation) prematurely in the case of the horizontal decelerations (150 to 165ms). This prevented modelling of the head impact, and made it difficult to predict maximum body segment deceleration levels, severity indices and the total collapse distance of the energy absorbing elements.
- (5) Useful reductions in peak head deceleration levels (30%) and severity index (40%) were achieved in the horizontal case, prior to the time at which head impact would have occurred.
- (6) Modest reductions in severity were also achieved for the vertical case, including reductions in the injury criteria and peak acceleration levels.
- (7) Doubt was cast on the performance of the spring elements within the FE program, beyond the first plastic yield point. Negative or low positive values of first plastic modulus seemed particularly to cause problems.

3.8 STAGE (B): DEVELOPMENT OF AN ENHANCED FE MODEL FOR THE INCORPORATION OF ENERGY ABSORBERS IN A COMPOSITE SEAT

Following on from the initial FE work, certain model refinements were necessary. Having received a number of components for the composite seats after considerable delay, it was then possible to base the model on the actual geometry of the parts obtained.

At this stage, the single seat model was altered to give the correct seat pan height and width, seat pan angle, lap belt anchor points and floor attachment points, prior to the acquisition of the program SOM-TA, with its triple seat option, Chapter 7. The objective of these modifications was to provide a more accurate model, and to try to solve some of the problems thrown up in the previous stage of the work.

3.9 STAGE (B) SIMULATION PARAMETERS

Simulation parameters were those used in Stage 1 of the simulation work, but with the following exceptions:

Lap belt anchor positions on seat changed.

Seat pan angle and width changed.

Finite Element seat model changed.

NOTE: Non-linear spring elements were again incorporated.

3.10 STAGE (B) FINITE ELEMENT SEAT MODEL

The seat model was changed to a 14 node, 27 element single seat, to incorporate the seat pan geometry and lap belt mounting points of the composite seat, see Figure 3.7. The seat pan was fully triangulated to the rear floor attachment points and lap belt mountings. The front leg elements were 11.5" long; the upper attachment points and floor attachments were placed equidistant from the rear leg to floor attachments. The front leg elements were modelled as non-linear spring elements as in Stage A of the simulation work, with lateral pin joints at each end. The remaining elements were again modelled as being of 1" square 0.2" gauge 1010 steel.

This model represented an initial refinement of the original model. The seat structure was modelled as simply as possible, because it was intended to represent a basically rigid structure (with each element working within its elastic range), behaving as a structure undergoing rigid-body motion, but with non-linear gross deformation of the front leg elements. In other words, the FE program was being used to assess the performance of the energy absorbing elements given a symmetrical, single seat idealisation with the geometry of the existing composite design. Obviously it was also intended to yield useful information such as support structure reactions loads.

3.11 STAGE (B) SIMULATION PROCEDURE

A similar process of model development was employed as in Stage (A), with optimisation of the non-linear spring characteristics using a forward deceleration pulse with ten degrees of yaw, and a (horizontal) velocity change of 44ft/sec (Figure 3.1). The designs were then cross-checked using the vertical deceleration pulse (thirty degrees nose-down) with a 35ft/sec change in velocity, Figure 3.2. This process was intended to ensure that behaviour was optimised for the more severe case (for occupants with lap belt restraints).

As regards the non-linear springs, the elastic stiffness (loading/unloading slope) was kept at 6000lb/in, and the first plastic modulus was set at 20lb/in to model almost constant-load collapse. The first and second plastic yield points were varied. Also, the second plastic modulus used to model bottoming-out was increased to 1000lb/in, in order to make more efficient use of the available stroking distance.

Three of the energy absorber types which were simulated successfully are discussed in this section, together with the equivalent standard (non energy-absorbing) design as a baseline. These types are designated 'CIC1', 'CIC2' and 'CIC3'. Other attempts were either less successful in reducing severity or utilised an unacceptably large stroking distance.

'CIC1' had a first yield point at 800lb, and a second plastic yield point at 925lb. Each of these was 50lb higher for 'CIC2' and 100lb higher for 'CIC3'.

3.12 RESULTS OF STAGE (B) SIMULATIONS

3.12.1 Horizontal simulations

The results are given in Table 3.2. Tabulated results are not included for the 'Rigid' seat, with beam element front legs instead of non-linear spring elements. In previous simulation work, Stage (A), this simulation only ran to 150ms, and for the 'Rigid' composite seat model the same problems occurred, with abortion due to excessive head rotation.

However, when energy absorbing front legs were included, the reduction in severity, and hence in head rotation, allowed the simulation run times to be longer. As mentioned previously, secondary peaks in the body segment accelerations usually occur when direct head contact takes place (at about 140-200ms for the 16g triangular pulse), which obviously precludes the possibility of meaningfully comparing the better energy absorbing types with the rigid seat.

However, at $t=145\text{ms}$, when the program for the rigid seat aborted, the head severity index was 650, compared with 240 for 'CIC2', which at least gives some indication for comparative purposes.

Unfortunately, problems persisted with the energy absorbing elements, as discussed in Section 3.6.1. The results of this part of the study therefore again indicated large stroking distances, approximately twice the magnitude of what was expected (see Table 3.2). This was investigated and is discussed in Appendix E.

3.12.2 Vertical simulations

The results are given in Table 3.2. The figures for only one energy absorbing design (CIC3) are included for comparison with the rigid design, as their behaviour was basically the same. No plastic collapse took place, but small differences in elastic stiffness marginally affected peak accelerations. Severity indices were identical for the two simulations.

These simulations were performed merely as a cross-check, as it was not intended that the seat necessarily stroked in this test configuration.

It was noted that the energy absorbing seats in Stage (A) in fact stroked during the vertical simulations, whilst those in Stage (B) did not. This was due to relatively small geometric differences, and the higher collapse loads of the elements. It could be concluded that the triggering of the energy absorbers in this configuration might be marginal. In particular, the stroking (or non-stroking) could be influenced by differences between the trigger load and the mean collapse load.

3.13 STAGE (B) SUMMARY

(1) A Finite Element model of a single seat based on the leg geometry of the composite triple seat, with the same restraint system mounting positions, seat pan and seat back positions, was incorporated into occupant simulations for the purposes of conducting parametric studies on the energy absorber load characteristics.

(2) A series of simulations was performed on a non-energy absorbing seat model (as outlined above) and on similar designs incorporating a variety of energy absorbing front leg designs. These simulations indicated that reductions in occupant injury severity parameters were possible using this approach.

(3) The behaviour of the spring elements in the program, intended for modelling the performance of energy absorbing elements, was suspect. The load carrying capability of the elements appeared not to be sustained throughout the impact, resulting in higher stroking distances than expected.

(4) With the elements not following the input stiffness relationship, it was not possible to determine stroking distances or optimum collapse loads for the elements. It would therefore be necessary to compare simulation results with test behaviour in order to quantify these parameters.

(5) An investigation into the spring element behaviour in the simulation program would need to be conducted. This study is described in Appendix E.

(6) The purpose of seat redesign was to reduce the severity in instances where the deceleration vector was primarily longitudinal. A cross-check was performed to ensure that no undesirable behaviour occurred for cases where the deceleration was primarily vertical. No stroking of the energy absorbers occurred in this case (except within the elastic range), although this was case was perhaps marginal.

CHAPTER 4

RESEARCH INTO THE COMPRESSIVE AXIAL COLLAPSE OF FIBRE-REINFORCED PLASTIC TUBES

4.1 INTRODUCTION

A programme of design, manufacture, testing and assessment of composite tubes was performed in order to develop structural elements suitable for incorporating into a composite seat structure.

Tests had previously been conducted on both steel and aluminium alloy tubes to observe the failure mechanisms. Such behaviour has been analysed and empirical equations used to predict collapse loads and modes of failure (Abromowicz, 1984).

Due to the infinite possibilities in combinations of fibre orientation, variations in material specification, and other parameters, predictions of mean collapse loads for composite tubes are at best based upon empirical data (which exhibits high degrees of scatter) and hence of little use. A brief description of previous work in this field is given in Section 5 of the Introduction in this thesis. The terms used are illustrated in Figure I1.

For ease of manufacture, composite tubes for testing were made using resin pre-impregnated fibre mat, with either uniaxial fibres or woven mat. The two main types of commercially available high performance fibre were used, carbon-fibre and aramid fibre (kevlar). The possibilities of glass-fibre and hybrids were also considered; glass-fibre was known to exhibit brittle failure modes similar to those produced in carbon-fibre composites, but at lower specific energy levels. The possibility of using it was therefore deferred. The use of hybrid composites was deferred on the grounds of unnecessary complication at the initial stage.

4.2 AXIAL COLLAPSE OF CARBON FIBRE TUBES

As witnessed by the high specific energy levels obtained and controlled collapse achieved by other researchers, carbon-fibre is a potentially useful material for energy absorbing elements. For a high energy absorption capability, global failure of the tube must be avoided. It would also be undesirable for a tube to fail from both ends. It is possible to achieve a microfracture mode of failure, initiated from one end of the tube by chamfering (or bevelling) that end. The stresses required to initiate failure are greater than those required to

sustain failure, and for many geometries and lay-ups the initial load required to trigger microfracture exceeds the global failure load. Thus bevelling one end, and hence reducing the cross sectional area at that end, will reduce the load required to trigger microfracture.

This work ran concurrently with simulation work connected with the overall redesign of the composite seat. As indications of the collapse load design requirements were obtained from computer simulation work, tooling size (and therefore tube internal diameter) was standardised at 50mm.

4.3 MANUFACTURING METHODS

All carbon-fibre tubes were constructed by rolling pre-impregnated woven cloth (pre-preg) or unidirectional material on cylindrical metal mandrels. Initially, tooling was made from aluminium alloy and incorporated a modest draft angle (less than 10 minutes of arc) to assist in freeing the finished tube from the tooling. When release techniques were improved this became unnecessary, and later tooling consisted of parallel sided steel mandrels coated with PTFE release spray.

Various release films, cloths and 'peel plies' were tried in order to increase ease of manufacture. Batch 1 was manufactured using Hercules Magnamite carbon-epoxy unidirectional pre-preg, whilst all subsequent batches were made using Ciba-Geigy Fibredux 913C (unidirectional and/or woven pre-preg).

All carbon-fibre samples were machined to give squared ends, and then one end was further machined to give a 45 degree chamfer. The tubes were not tested (crushed) for their complete length, as complete destruction would have precluded post-test examination of the samples. All quasi-static compression tests were performed at low cross-head speeds in an Instron 1195 test machine.

4.3.1 Batch 1

This consisted of four tubes, each made using 6 layers of unidirectional material; tube 1A was made using 6 layers of 90 degree fibres, tubes 1B and 1C having 2 layers of 0 degree and 4 layers of 90 degree fibres. One sample of each type of type was tested in compression. Tube 1A was made on a tapered mandrel of nominal diameter 50mm, the remainder on 64mm diameter steel tubular mandrels.

4.3.2 Batch 2

The second batch was made using both woven cloth and u/d material. Tubes 2A and 2D were constructed using 3 layers of 0/90 cloth with 1 layer of 90 degree u/d on the inside and 1 layer on the outside. Tubes 2B and 2C were similar but with +/-45 cloth replacing 0/90 cloth. Tube 2A was made on a 50mm mandrel, with the remainder on 67mm tooling.

4.3.3 Batch 3

Mandrel size was then standardised at 50mm (nominal), all tooling having a small draft angle to aid release. All tubes in this batch had 3 layers of +/-45 cloth, with a nominal wall of thickness of 1.5mm (3 layers), the main aim at this stage being to experiment with the manufacturing method.

4.3.4 Batch 4

This batch was also constructed using only +/-45 cloth. Using 2 layers to give a nominal wall thickness of 1.0mm, three such identical tubes were made so that tubes of different lengths could be tested for comparison (4A, B and C). Tube 4D was made using 1 layer of +/-45 cloth, and 4E using 1 layer of 0/90. This was to investigate the possible lower limits of relative density which could be used, whilst still maintaining stable collapse.

4.4 RESULTS OF TESTS ON CARBON-FIBRE TUBES

4.4.1 Batch 1

The results of the static tests are given in Table 4.1. As previously mentioned, the first tubes were made mainly to assess the problems involved in manufacturing carbon fibre tubes. Peak initial loads exceeded subsequent mean loads by about 35-45%, failing in a controlled manner by microfracture processes. High specific energy absorption figures were obtained for both configurations (70-80kJ/kg) compared with typical values of 30-35kJ/kg for aluminium alloy tubes. This was due to two main factors:

(i) The ability to sustain controlled collapse at a relatively high load for the cross-sectional area.

(ii) The ability to maintain crush for the complete length of the tube. By comparison, most metal tubes of similar geometry will crush only over a distance equal to approximately 60% of the original length before 'bottoming out'. Post-test appearance of the tubes is shown in Figure 4.1.

4.4.2 Batch 2

At the time the second batch was made, woven pre-preg cloth and compatible u/d material were both available. Comparing the collapse of the 0/90 tubes with the +/-45 lay-up proved to be difficult. The smaller diameter 0/90 tube was tested successfully, but the collapse load of the larger diameter tube exceeded the maximum rated load of the test machine (10,000lb), and hence it was not possible to complete a static test on this tube. Post-test appearance of one of the tubes is shown in Figure 4.2.

4.4.3 Batch 3

Due to a design requirement to reduce the collapse load, the third batch were all +/-45 lay-up, with wall thickness reduced to a nominal 1.5mm. Tube diameter was standardised at 50mm bore size. The collapse loads and specific energy results indicated good repeatability, although the loads would need to be reduced further. By this stage the manufacturing technique was satisfactory. The post test appearance of all tubes made from +/-45 cloth was similar, (see Figure 4.3), although clearly different to the 'unwound' appearance of the tubes with predominately 90 degree fibres.

4.4.4 Batch 4

For tubes 4A, 4B and 4C the nominal wall thickness was reduced still further to 1.1mm, with fibre orientation standardised at +/-45. Collapse loads in the required range were achieved, with stable collapse by microfracture taking place, see Figure 4.4 for the post-test condition. The force-displacement trace shown in Figure 4.5 is typical of the results obtained for the carbon-epoxy tubes, with a high initial peak followed by a ragged but approximately constant crush.

Tube 4D crushed in a reasonably controlled manner, although at a greatly reduced specific energy absorption, due to unstability in the walls, which were too thin. Tube 4E split catastrophically at a load of 1830lb. Thus the design limit in terms of relative density is above 0.05 for +/-45 cloth, and between 0.05 and 0.09 for 0/90 fibre orientation.

4.5 AXIAL COLLAPSE OF KEVLAR TUBES

Following the series of tests on carbon-fibre tubes, a similar programme of design, manufacture, testing and assessment was conducted using aramid fibre (trade name 'kevlar') in order that comparisons could be made and conclusions drawn about the suitability of each material for the particular application under consideration.

Due to the mode of failure reported for kevlar tubes in other research, it was hoped to satisfy the post-crush structural integrity requirement. Because failure by fibre buckling normally occurs rather than microfracture processes, a sufficient number of unbroken fibres may be present at any point during the collapse to support a tensile load if necessary. In general kevlar has been found to have a lower specific energy absorption capacity than CFRP, but the difference in mass between energy absorber units made from each was considered unlikely to be significant in this context, see Chapter 5.

4.6 MANUFACTURING OF KEVLAR TUBES

The tubes were manufactured using the same methods as those described for carbon-fibre tubes (see Section 4.3). Due to the reportedly poor compressive load carrying capability of +/-45 lay-ups, it was decided that the initial batch would be made using 0/90 cloth, and to develop tube design as dictated by the results. All tubes were tested quasi-statically in an Instron compression test machine, with displacement control and variable cross-head speed. The lower plate of the machine was mounted on a spherical joint allowing a limited amount of tilt. All tubes were made on 50mm steel parallel cylindrical mandrels, which were individually labelled 'A' to 'F', hence each batch did not necessarily include all these reference letters. Batch numbers referred to the series of composite tubes rather than kevlar tubes in particular, hence numbering commenced at '5A'.

4.7 RESULTS OF COMPRESSION TESTS ON KEVLAR TUBES

Comprehensive test results are given in Table 4.2. Figures 4.6 to 4.15 show post-test tube condition and Figures 4.16 to 4.34 give force-displacement traces.

4.7.1 Batch 5

Tubes 5A and 5B were made using 4 layers of 0/90 cloth, giving a wall thickness of about 1.1mm, with 5C and 5D having 5 layers of 0/90. Each tube had a finished length of 300mm, allowing two samples to be cut for testing. It was not possible to machine a bevelled edge on the kevlar using conventional methods. Tubes 5C/2, 5D/1 and 5D/2 were tested with no end supports in a compression test machine in a similar manner to the carbon-fibre tubes. Tubes 5C/2 and 5D/1 were each 100mm long, and crushed by progressive folding, see Figures 4.8 and 4.9. The end at which collapse began was not predictable, and the results indicated a high initial peak, followed by a varying collapse load and bottoming-out, Figures 4.19 and 4.20.

Tube 5D/2 was longer at 200mm, failing first at the top, then later at the bottom. Collapse became unstable and the test was stopped, Figure 4.21. It was obvious that without a means of initiating (and maintaining) collapse from one end, it would be necessary to provide some type of end support. Simple flanged aluminium alloy push-on end pieces were made, which supported the tube for the first 12mm at each end. Tubes 5C/1 and 5A/1 were each 100mm long and end fittings employed at both ends. The collapse of 5C/1 was controlled (Figures 4.8 and 4.18), but 5A/1 was unstable, see Figures 4.6 and 4.16.

Tube 5A/2 was initially 200mm long and tested with end fittings at each end. As can be seen from Figure 4.7, failure occurred towards the middle of the tube along the lines where the edge of the shrink-tape had been positioned during the curing process, causing a stress raiser. Tube 5B/1 was shorter at 150mm, but failed in much the same manner, Figure 4.7. Tube 5B/2 was the final tube from this batch, of length 150mm. This was tested with an end fitting at the bottom and unsupported at the top. Failure was initiated at the top and was stable, with a relatively high peak followed by a short portion with a relatively low load level, and a region of constant crush before bottoming-out, Figure 4.17.

4.7.2 Batch 6

Following the experiences with the previous batch, and given that it would be necessary (for the design application envisaged) to have permanently attached fittings at each end, it was decided that some form of trigger mechanism or collapse initiator would be needed. Two ideas had come to mind as a result of the behaviour of tubes in the previous batch.

Firstly, catastrophic failure had occurred in some tubes by forming fold lines coincident with the position of the edge of the shrink-tape which was wound helically around the tube during the manufacturing process. In regions where the tape had overlapped slightly, the pressure created would be approximately doubled and had resulted in a regions of relatively low resin content. This seemed to cause local weakness, and so it was decided to try an initiator in the form of a circumferential band of shrink-tape close to one end of the tube. This would obviously need to be in addition to the tape wound on during manufacture (which would be subsequently removed). It would also be important to take great care in ensuring that the problems mentioned with the overlapping of shrink-wrap tape were eliminated.

Additionally, a second trigger method had also come to mind. Tubes with low slenderness ratios (length to diameter ratio) had collapsed in a more stable fashion,

and hence a design with a section of reduced cross-section could be incorporated. It was hoped that the tube would then effectively behave as two tubes, the 'short' thin-walled section of tube failing first and hence initiating collapse, and that failure, having started, would continue across the change in cross-section and produce controlled failure in the remainder of the tube.

All tubes were made such that two 150mm samples could be cut from each. Tube 6F/2 was built as a 'control' sample, with no trigger, and failed towards the centre in a similar manner to 5B/1. Push-fit end fittings were used.

Tube 6D/1 had a single wrap of shrink-tape, 7mm wide, positioned 24mm from one end. Two push-on end fittings were employed. Controlled failure occurred, with an initial peak of 4550lb, followed by a roughly constant load of 2300lb, Figure 4.26. Failure began at the position of the shrink-tape trigger. The tube is shown (minus fittings) in Figure 4.11.

Tube 6D/2 was nominally identical to 6D/1 but the trigger was a double wrap of shrink-tape. This reduced the initial collapse load to 3900lb, with a similar load-displacement relationship and again a mean of 2300lb, Figure 4.27. The post-test condition of the tube is shown in Figure 4.10 (with the end fittings removed).

Tube 6F/1 again had double shrink-tape trigger, but in this instance only 16mm from one end (the end fitting intrudes 12mm into the tube). The result was a perfect first fold, with a high peak of 4550lb, followed by folding at a mean load of 2700lb, Figure 4.25. The tube is shown in Figure 4.10 (the trigger was at the top of the tube as viewed).

Tube 6B/1 had a 'reduced cross-section' trigger, reducing from 5 layers of 0/90 cloth to 4 layers for a length of 23mm at the top of the tube. Initial peak load was 3550lb, with a mean of 2000lb, Figure 4.24. Collapse was controlled, following successful triggering, but the force-deflection trace had a second, slightly lower peak before the load became approximately constant, see Figure 4.11.

Tubes 6A/1 and 6A/2 were both made with 5 layers of 0/90 cloth reducing to 3 layers for a length of 25mm at one end. Tube 6A/1 used push-on end fittings, with 6A/2 using the same fittings, but bonded using Araldite 2005 two-part adhesive. This enabled a useful comparison to be made, bearing in mind that some form of bonded fittings was envisaged to satisfy the tensile load-carrying requirement of the energy absorber. Testing of tube 6A/1 gave a peak of 2480lb, with a mean of 2000lb thereafter, Figure 4.22. See Figure 4.11 for post-test condition. The bonded tube

produced a peak of 3510lb with a mean load of approximately 2500lb, but with a high degree of variation over the crush length, Figure 4.23. Tubes 6B/1 and 6C/2 were used as tensile samples to develop the bonding technique, see section 4.7.5.

4.7.3 Batch 7

In order to reduce the number of design variables, and to uncover any problems which might occur due to bonding, all three tubes in this batch were of reduced cross-section type using 0/90 cloth, and all samples had bonded end fittings.

Tubes 7C/1 and 7C/2 were both 150mm long, and with 4 layers of 0/90 cloth reducing to 3 layers for a short portion at one end. For tube 7C/1 this portion was 16mm long, resulting in an initial collapse load of 4220lb, which is more than double the mean of approximately 2100lb, Figure 4.29. For sample 7C/2, the length of the reduced thickness portion was increased to 18mm (including the 12mm into which the bonded end fitting intruded), and this reduced the initial peak to 3210lb. Mean load was also reduced slightly to 2000lb, Figure 4.30. Collapse behaviour for both tubes was stable, and the post-test samples are shown in Figure 4.12.

Tube 7A/2 was made using 5 layers, reducing to 3 layers for a length of 32mm. This length had been increased to see if it was possible to remove the initial peak completely, the length having been estimated from extrapolation of the results from previous tests. In fact the peak was some 17% above the subsequent mean, but with a fairly substantial energy absorption loss occurring immediately after the peak, before the load rose to a steady level, Figure 4.28. The tested tube is shown in Figure 4.13. (Tube 7A/1 was used for tensile tests).

For tube 7B, which again was 5 layers reducing to 3, the length of the reduced portion was only 19mm, and the sample tested was 250mm long. Collapse began at the change in cross-section, but one end of the tube then tried to move sideways, and the test was stopped. This could not be avoided as the lower platen of the test machine was mounted on a limited movement ball-joint. The tube is shown in Figure 4.13.

4.7.4 Batch 8

As a final stage of testing, before deciding upon a design and development programme for prototype energy absorbers, the final batch consisted of tubes with a proportion of +/-45 fibres in order to assess the effects on collapse behaviour. Re-usable push-on end fittings were used for ease of testing.

Tube 8A was manufactured using three layers of 0/90 cloth followed by two layers of +/-45 cloth. For sample 8A/1 this reduced to 3 layers of 0/90 only for 29mm at one end, and push-on flanged fittings at both ends. Tube 8A/2 was of the same lay-up but no reduced cross-section, and a fitting at one end only. Sample 8A/1 had a peak of 2900lb, only 16% higher than the mean of about 2500lb, but at the expense of an energy loss following initial failure (Figure 4.31), whilst 8A/2 indicated a mean collapse load of 2400lb, Figure 4.32.

Tube 8B was made with one layer of 0/90 cloth and 4 layers of +/-45 cloth. As for 8A one sample employed a reduced cross-section 29mm in length with push-on fittings at both ends (8B/1), the other having no trigger mechanism and only one end fitting (8B/2). Both collapsed at a mean load of 2100lb, the peak for the triggered tube being 2500lb, and 3750lb for the free-ended tube, see Figures 4.33 and 4.34. The post test condition of the tubes is illustrated in Figures 4.14 and 4.15.

4.7.5 Tensile Tests

Two tensile tests were conducted to assess the effectiveness of the bonded method. Tube 6B/2 failed at 3450lb using Araldite 2005, with both the end fittings and kevlar tube in 'as manufactured' condition and the bonding cured at 80 C for 6 hours. Tube 6C/2 was tested with the fittings and the inner surface of kevlar roughened using 400 grade wet/dry abrasive paper, and failed at 4010lb due to an end fitting pulling out. An improved method of bonding technique, to increase failure load, is described in Chapter 5.

4.8 DISCUSSION

4.8.1 Carbon-Fibre Tubes

Having manufactured, tested and developed the tubes, the suitability of carbon-fibre for use as the energy absorbing elements in the seat redesign must be assessed.

The following properties recommend its use:

- (1) Predictable force-stroke crush relationship.
- (2) Reasonably constant collapse load.
- (3) Lightweight, with high specific energy absorption.

However, the following problems exist:

- (1) Crushing produces debris which would present a problem of containment.
- (2) Some form of guidance, for example, an internal slider, would be necessary.

- (3) When the available stroke has been exhausted, a means of preventing bottoming-out would have to be provided.
- (4) Due to the nature of the collapse, once crushing has been initiated, the tubes cannot sustain a tensile load.

The containment problem could be solved in many ways, for example a rubber or polypropylene gaiter could be used. Guidance would not necessarily be a problem - due to the smooth nature of the tooling the internal tube surface would be suitable, provided that no draft angle was used on the tooling. Bottoming-out could be solved by using foam, or honeycomb, or other crushable element.

The main stumbling block remaining was thus the tensile requirement. For sustaining normal operating loads, a central cable or similar lightweight tension member could be used. With the bevel end requirement and thus the difficulty in attaching an end fitting, such a device would probably be necessary. However, due to the collapse mode, the ability to sustain a tensile load once crushing had begun would not exist without the addition of a further mechanism, perhaps acting on the cable. This would create a major problem since heavy crash landings can consist of positive and negative vertical components of acceleration.

4.8.2 Kevlar Tubes

Initial testing on Batch 5 indicated the need for a trigger mechanism, which was a problem also identified in previous research. Firstly, in the only instance that controlled collapse was achieved with fittings at both ends (5A/1), peak load exceeded mean load by 175%. Secondly, catastrophic (global) failure was occurring in many instances, so it was necessary to find a means of ensuring that collapse was initiated at one particular end.

Two ideas were thus pursued in Batch 6. The shrink-tape trigger appeared to work successfully, with the peak load approximately double the mean load for one wrap, or 70% higher for two wraps (with push-on end fittings). The chosen wrap width was about 7mm, to allow a single fold to form and this seemed to be about the correct width for this to happen.

The 'reduced cross-section' method of initiating failure also seemed to operate well. Tube 6B/1 was the only tube which reduced from 5 layers to 4 at the trigger, giving a peak some 78% higher than the estimated mean load. With 5 layers reducing to 3 layers (still unbonded), 6A/1 triggered at a load only 24% above the subsequent mean collapse load. The effect of bonding the end fittings for a nominally identical sample (6A/2) was to produce a trigger load 40% higher than the mean, which was itself

25% above the level obtained from the unbonded sample. Untriggered sample 6F/2 illustrated the need for a initiation mechanism.

Samples 7C/1 and 7C/2 gave an indication of the effect of changing the length of the reduced cross-section portion for bonded tubes with 5 layers reducing to 4 layers. Deducting the bonded part, an effective length of 4mm gave a peak 100% above the mean, and 6mm gave a peak 60% above the mean.

For tubes reducing from 5 to 3 layers, 7A/2 with an effective trigger of 20mm showed a peak only 17% above the mean compared to 40% for 6A/2 (13mm trigger). However, significant energy losses occur following initiation. Tube 7B was unsuccessful, raising some interesting points. It was in many ways a 'worst case' designed to draw early attention to any problems likely to occur. The effective length of the reduced cross-section was only 4mm (probably too small), and the tube was of considerable length (250mm) with bonded end fittings. It was useful in indicating the possible need to stabilise collapse for tubes of high slenderness ratio.

Batch 8 gave useful information regarding the use of +/-45 fibres. Samples 8A/1 and 8B/1 could justifiably be compared, with mean collapse load reducing by 16% as the proportion of +/-45 fibres was increased from 40% to 80%.

Tensile tests indicated reasonable bonding performance levels, which it was hoped would be improved by optimising cure cycle, surface preparation and end fitting design, see Chapter 5.

It was decided that kevlar was the more suitable material for use in an energy absorbing element in an aircraft seat, for the following reasons:

(1) Kevlar fails by fibre buckling. The 0 degree fibres are still capable of sustaining tensile loads after collapse of the tube. Carbon fibre fails by fracture processes and a damage front moves down the tube. This discontinuity precludes the possibility of sustaining a tensile load after initiation of failure.

(2) Due to the different modes of failure, debris is produced in the crushing of carbon-fibre tubes, presenting a problem of containment to prevent injury. Such a problem does not exist with kevlar.

(3) In a manner similar to metal tubes, kevlar 'bottoms out' gently when the available stroke is exhausted. As carbon fails by disintegration, the tube carries on crushing until no material remains, at which point the load would increase very rapidly.

4.9 SUMMARY:

4.9.1 Carbon-Fibre Tubes

(1) The characteristic load-displacement behaviour of carbon-epoxy tubes (with one bevelled end for triggering collapse) was predictable, with a large region of collapse at constant load.

(2) The load uniformity (the ratio of trigger load to mean collapse load) was in the range 1.35 to 1.8, and was dependant upon the detail design of the bevel trigger.

(3) Specific Energy Absorption of 80kJ/kg was attainable where a large proportion of 0 and 90 fibres were utilised. SEA in the range 52 to 58kJ/kg was typical for tubes with +/-45 fibre orientation.

(4) Specific Energy Absorption was greatly reduced for tubes of low relative density, (0.05) due to elastic buckling in the section wall. This would mean that for mean collapse loads of below 2000lb, it might be necessary to reduce the tube diameter to below 50mm.

(5) The mean collapse load can be decreased for a given geometry by increasing the proportion of +/-45 fibres. This means that range of collapse load which can be achieved for tubes of a given diameter is increased, before problems associated with low relative density are encountered.

(6) The debris produced, due to the microfracture processes involved in the crushing of carbon-epoxy tubes, would be hazardous in a passenger cabin environment and some form of containment would be needed.

(7) For an energy absorber to replace existing seat structure, the following requirements need to be met in addition to providing a stable resisting force over a high proportion of its length: tensile load carrying capability; post-crush integrity; ease of incorporation.

(8) Due to the mode of failure (propagation of a damage front down a tube) these additional requirements would have to be satisfied by 'add-on' features. This might be difficult to justify in terms of weight, cost and complication, and that in effect the weight advantages of using carbon-fibre are offset by the additional energy absorber components which would be necessary.

4.9.2 Kevlar Tubes

- (1) As the slenderness ratio was increased beyond a value of 2.0 for tubes of 0/90 cloth, and relative densities of about 0.1 with 2 free ends, collapse became unstable.
- (2) 50mm diameter tubes with one built-in end and one free end produce stable collapse with samples up to 150mm in length (greater lengths were not tested).
- (3) Where tubes have supports at each end, global failure occurs for slenderness ratios greater than 2 unless triggering methods were used.
- (4) A method of triggering controlled collapse has been created by using a band of shrink-tape close to one end of a tube.
- (5) A second method of triggering collapse, involving a short portion of reduced cross-section at one end of a tube has also been developed.
- (6) Dependant upon the choice of trigger type and relevant dimensions, collapse loads can be 'tuned' to exceed the mean collapse loads by between 17% and 100%.
- (7) Bonding of end fittings to tubes increases both the peak and mean collapse loads.
- (8) For a given tube geometry, collapse loads can be adjusted by varying the relative proportions of 0/90 and +/-45 fibres. Specific Energy Absorption can thus be varied between 25 and 35kJ/kg.
- (9) A means of stabilising collapse of tubes with high slenderness ratios (above 4) would be necessary.
- (10) Initial tensile tests indicated debonding of the end fittings at a load above 4000lb, although many possible methods of improving this are available.
- (11) The post-crush integrity of the tubes, as a result of the fibre-buckling mode of failure, and the lack of debris, were advantages in using kevlar in comparison with carbon-fibre.
- (12) Although having lower Specific Energy Absorption than the equivalent carbon-fibre tube, the marginal weight penalty of kevlar over carbon per triple seat would be less than 40g, an insignificant proportion of the seat weight.

CHAPTER 5

DEVELOPMENT OF A KEVLAR ENERGY ABSORBER UNIT

5.1 INTRODUCTION

Following the work on the axial collapse of composite tubes described in the previous chapter, the relative merits of the materials tested were assessed, and other materials were considered.

The advantages of using carbon-fibre tubes as energy absorbing elements relate to high specific energy absorption, constant collapse load and predictable behaviour. However the outcome of the tests indicated certain problems existing with the use of carbon-fibre in the application under consideration. Firstly, due to the nature of failure (delamination and other microfracture processes) hazardous debris was produced, presenting a problem of containment.

Secondly, and more significantly, the ability to sustain a tensile load before, during and after impact is a requirement for the energy absorber. This could not be met by carbon-fibre alone, due to the nature of failure. The movement of a damage front along a tube precludes this. Hence a secondary mechanism would have to be incorporated to cope with tensile load requirements.

The work on kevlar tubes had yielded interesting results. The major hurdle to be tackled was to develop a method of triggering failure in these tubes. Two different methods were developed, one using a means of locally reducing the resin content in the region where collapse was to be initiated, the other by creating a reduction in the cross-section close to one end. Failure was predominately by fibre buckling, and produced folding in a comparable manner to metal tubes. The use of bonded end fittings to satisfy tensile loading requirements resulting in increased collapse loads but promoted unstable collapse in slender tubes.

In summary, both types of material would require some form of internal slider: carbon, to provide a means of controlling tube position as the damage front moves down the tube; kevlar as a means of stabilising collapse. Carbon has a higher specific energy absorption, but most of the advantage is nullified by restrictions on the minimum acceptable relative density required for stable collapse. Hence the weight saving would be no more than 20 grams per leg, or 3.5 lb on a 120 seat-place aircraft. Furthermore, the weight penalty applicable to carbon-fibre incurred in incorporating a tensile load-carrying

capability would certainly exceed this, in addition to increased complication and cost. Finally, the problems of containment applicable to carbon fibre do not apply to kevlar.

Alternative materials were available, such as glass fibre, which has a performance disadvantage over the materials tested, but behaves in a similar manner to carbon-fibre under axial crushing, although it is of course cheaper. The possibility of using a carbon-kevlar hybrid was rejected as unnecessary. The presence of carbon fibre would increase collapse load, and hence tube wall thickness would have to be reduced to an unacceptable level. The preferred method for increasing the collapse load would be to increase the proportion of 0/90 fibres used, were this to be necessary.

For these reasons, it was decided to develop a kevlar energy absorbing element, and the development programme is described in this chapter.

5.2 DESIGN REQUIREMENTS

Having decided to use kevlar, various design problems remained to be solved. One specific design requirement was for structural members to act as front legs in a 'crashworthy' conversion to an existing composite seat, with 'A-frame' legs.

The maximum possible length of the complete members, including all fittings, was believed to be 12" long, with a stroke of approximately 5". It was indicated from practical trials that further stroking would hinder subsequent evacuation. Allowing for end fittings, the kevlar tubular portion could be up to 10" long.

Energy absorbers would be required to collapse in a stable manner in uniaxial compression, at a predictable initiation load. Any energy losses induced as a result of employing a particular trigger method would have to be minimised. Subsequent crushing was required to take place at a reasonably constant load, followed by a relatively gentle asymptotic rise upon bottoming out.

Spherical 'rose joint' bearings were to be used at each end of the members to ensure that the units would not be subjected to bending moments.

The legs were required to be capable of meeting the normal static strength requirements and working loads. They were also designed to be capable of sustaining tensile loads before, during and after impact. The weight penalty associated with the conversion was of course borne in mind and is discussed later.

The rear legs were to be pin jointed laterally at the anchorages, as in the FE model, and the redesign is covered in Chapter 6.

In order to satisfy the above design requirements, it was also deemed necessary to change the end fitting design, as described below.

5.3 END FITTINGS

Having produced unsatisfactory collapse in long (10") tubes with bonded ends, it was considered necessary to stabilize the collapse by means of an internal slider. The smooth surface produced by the tooling in the manufacture of the tubes lent itself well to this approach. Bearing in mind the design requirements outlined previously, an end fitting was therefore designed. The main problem would be keeping the uncrushed and crushed portions of the tube on a common axis. Thus a support which allowed sliding was needed. Also, the tensile requirements were borne in mind. The prototype end fitting design is shown in Figure 5.24. This was intended for use in conjunction with the usual simple flanged fitting at the other end of the tube.

The end fitting was bonded at its flanged end, to the end of the kevlar tube at which collapse was to be initiated. The folds formed towards the top end of the stem, where the failure was triggered; simultaneously undamaged portions of the tube moved over the slider, towards the crush zone. The length of the stem (60mm) was intended to be sufficient to allow a crushed portion equivalent to 5" of tube to reside between the slider and the end fitting. If loaded in tension, the slider would attempt to pull the damaged region back into shape, thus providing a resisting force. The flange was radiused in such a way that controlled failure occurred, although not an ideal solution from the viewpoint of bonding. The end fitting concept was granted a patent and is described in more detail in Appendix F.

5.4 DEVELOPMENT TEST PROGRAMME

A three stage development test programme was envisaged:

- (1) Static compression tests (energy absorber only).
- (2) Dynamic compression tests (energy absorber only).
- (3) Tests on seats with energy absorber installed.

The results from parts (1) and (2) are presented in this chapter. In addition, a tensile test was conducted and is reported on. Full-scale tests on prototype seats are covered in Chapter 6.

5.5 STATIC COMPRESSION TEST PROCEDURE

Four different types of tube were tested statically, with push-on end fittings to enable the fittings to be reused. One type (Type IV) was rejected as unsuitable and further tubes of the other three types tested statically, with bonded end fittings to assess any change in collapse behaviour or loads. The types of tube differed in trigger method and fibre orientation, all being nominally 250mm in length. The different types of tube are described below.

NOTE: Batches 9, 10 and 11 were manufactured together and hence do not refer to the order in which they were tested. The reference numbers for each tube type are given in Table 5.1. A 24mm thick piece of phenolic foam was placed inside each tube to prevent sudden bottoming out.

5.6 TUBE TYPES

Type I: Five layers of 0/90 cloth, reduced to three layers for a 30mm length at one end.

Type II: Five layers of 0/90 cloth, with a double wrap of shrink tape about 7mm wide, in the portion 23-30mm from one end.

Type III: Three layers of 0/90 cloth and two layers of +/-45, reducing to three layers of 0/90 for a 30mm length at one end.

Type IV: The same as Type III, but with shrink tape wrapped around the tube immediately after the reduction in cross-section.

5.7 TEST RESULTS

Stable, controlled collapse of the tubes took place for all the samples. The test results are given in Table 5.1 and analysed below (by tube type).

Type I: Tube 11b was tested with push-on end fittings and collapsed at a mean load of 2750lb, with a load variation during crush of +/-27%. Energy losses at initial collapse amounted to about 3% above the mean level, Figure 5.20. Tearing of the fibres occurred however, and the outer layer of woven material had started to peel off (Figure 5.2). The tensile strength would probably not have been affected by such behaviour, since in any event only the fibres in the three inner layers ran for the complete length of the tube.

Tube 11a was nominally identical to the above, but with end fittings bonded to the kevlar using Araldite 2005. Mean collapse load was 9% higher than for the push-on ends, at 3000lb, with a $\pm 20\%$ variation. Initial energy losses amounted to 6%, with a peak load of 3400lb, 13% higher than the mean, Figure 5.19. Again peeling of the outer layer of kevlar was evident.

Type II: Tube 10b was tested with push-on fittings and found to have a mean collapse load of 3000lb ± 600 lb (20%), Figure 5.18. Initial energy losses were less than 5%. The initial peak load was 55% higher than the mean as a result of the shrink-tape trigger. Stable collapse with no tearing and regular folds occurred (Figure 5.2).

Tube 10a was of similar construction but with bonded ends. Mean collapse load was 5% higher at 3150lb $\pm 14\%$, with the peak load 52% higher than the mean, Figure 5.17. Again initial energy losses were less than 5% of the total energy absorption capability.

Type III: Tube 9c (push-on ends) collapsed at a mean load of 2700lb, with the load varying during the collapse by $\pm 15\%$, Figure 5.16. The initial peak load exceeded this mean by 15%. The energy loss at the initiation of failure represented about 5% of the total energy absorption capability. Collapse was again stable with regular folds and no evidence of significant fibre breakage (Figure 5.1).

Tube 9a, with bonded end fittings, produced loads marginally higher than those measured for the unbonded tube. Mean collapse load was 2750lb, again $\pm 15\%$, with a peak of 3210lb (17% above the mean). Initial energy losses of about 6% were found to occur, Figure 5.14. Post test condition is shown in Figure 5.2.

Type IV: Tube 9b had push fit ends and the combined reduced cross-section and shrink tape triggering mechanism. The mean collapse load was approximately 2400lb, Figure 5.15. The peak load was only 5% higher than the mean, but at the expense of a high initial energy loss (about 11%). The tested tube is shown in Figure 5.1.

5.8 DISCUSSION OF RESULTS

The specially developed end fitting operated satisfactorily with stable controlled collapse occurring in all the tubes. The tests gave a range of collapse loads for nominally the same length, diameter and wall thickness. This would enable suitable combinations of trigger method and fibre orientation to be used to give the desired collapse load characteristics.

From results of the initial tests conducted on kevlar tubes (Chapter 4), one would expect the highest peak loads to occur in tubes with a shrink-tape trigger, and for higher mean loads to occur in tubes made only from 0/90 cloth compared with those containing a proportion of ± 45 fibres.

As expected, lowest mean and peak loads occurred for the Type IV tubes, with the double (shrink-tape and reduced cross-section) trigger keeping the peak low, and the ± 45 fibres ensuring a relatively low mean. Similarly, the highest peak and mean loads occurred for the Type II tubes, with the shrink-wrap trigger only, and 0/90 fibres only.

The remaining tubes were Type I and Type III, which had the same (cross-section reduction) trigger but with different fibre orientations. For both push-fit and bonded ends, the mean loads were lower for the tubes containing ± 45 fibres, (Type III). For the bonded tubes, the peak load was also lower for the Type III than the Type I. However, for the push-on end fittings, the peak was lower for the Type I than the Type III. This was thought to be due to the fact that during the initial part of the test on tube 11b, damage occurred at the end of the tube due to fibre end-loading, and consequent splaying of the 0 degree fibres. This would have the effect of reducing the initial collapse load. Examination of the force-displacement trace indicated that the collapse load rose as displacement increased, whereas the other tubes had a roughly constant collapse load. For the purposes of analysis it was therefore felt that the results of 11b could justifiably be excluded.

Comparisons made between the behaviour of the bonded tubes were considered more meaningful than those of unbonded units, due to the elimination of these end loading effects. In general, both the peak and mean collapse loads were 2-9% higher (ignoring 11b) for the bonded tubes than for those with push-on ends, presumably due to the greater degree of restraint afforded by the bonding.

The level of the mean load, excluding the initial collapse (about the first 20mm), varied from $\pm 14\%$ to $\pm 20\%$, which should have been acceptable, subject to satisfactory results from dynamic tests.

Immediately following failure, load drop-off tended to occur over a crush distance of about 20mm before the constant collapse region became established. The area under the graph represents the energy absorbed by the tube, therefore the dip represents the shortfall in the energy when compared to an idealised rectangular, constant load relationship. Using numerical methods, this shortfall has been estimated as a percentage of the energy absorbed during the full (135mm) stroke, Table 5.1. For either trigger method, this amounts to 4-6% or 11% in the case of both trigger methods being employed simultaneously (Type IV). Thus the use of the double trigger was considered to involve unacceptably high reductions in energy absorption capability when compared to a single trigger.

5.9 DYNAMIC COMPRESSION TEST PROCEDURE

Dynamic compression tests were conducted using tube types which had already been tested statically, with both push-on end fittings and bonded end fittings. Subsequently, a set of tubes of lower wall thickness were also tested, and the results from these tests are included. The tubes were mounted to the impact face at the end of the seat testing rig, and the bare trolley (mass 212kg) catapulted towards the end. Each sample was tested individually. The impact energy was generally about 1.5kJ. A load cell comprising 3 load washers was mounted at the impact end of the tube and a datalogger with class 60 filter utilised.

5.10 TUBE TYPES FOR DYNAMIC TESTS

Tubes of Types I, II, and III (as defined for the static tests) were tested dynamically, with bonded end fittings. In addition, tubes of Types II and III with push-on ends were tested. Type IV was dropped from the test programme due to the energy losses experienced in the static tests.

Types V, VI and VII were all thinner walled tubes than employed previously, consisting of four layers of woven cloth rather than five, in order to increase the available range of collapse loads. This was considered to be necessary due to increases in the collapse loads as a result of bonding method and/or dynamic behaviour. Furthermore, many multi-occupant seats have a high degree of asymmetry resulting in a weight bias, and hence the

loads may not be shared equally between the two energy absorbers. Under these circumstances, the collapse load requirement for each unit would be different. These additional types were tested dynamically with bonded end fittings only, without recourse to static testing. The complete range of tube types is described below:

Type I: Five layers of 0/90 cloth, reduced to three layers for a 30mm length at one end.

Type II: Five layers of 0.90 cloth, with a double wrap of shrink tape about 7mm wide, in the portion 23-30mm from one end.

Type III: Three layers of 0/90 cloth and two layers of +/-45, reducing to three layers of 0/90 for a 30mm length at one end.

Type IV: The same as Type III, but with shrink tape wrapped around the tube immediately after the reduction in cross-section.

Type V: Three layers of 0/90 cloth and one layer of +/-45 cloth, reducing to three layers of 0/90 for a short portion (30mm for Type Va, 25mm for Type Vb) at one end.

Type VI: Two layers of 0/90 cloth and two layers of +/-45 cloth, reducing to three layers of 0/90 and one layer of +/-45 for a short portion (30mm for Type Va, 25mm for Type Vb) at one end.

Type VII: One layer of 0/90 cloth and three layers of +/-45 cloth, reducing to one layer of 0/90 and two layers of +/-45 for a short portion (30mm) at one end.

5.11 DYNAMIC TEST RESULTS AND DISCUSSION

The results from the series of dynamic compression tests are given in Table 5.2 and discussed in detail below:

Type I: Tube 11c was tested with bonded end fittings at impact speed of 3.71m/s and collapsed in a controlled manner, although tearing of the fibres and inter-laminar separation were evident, producing the effects shown in Figure 5.4. The load cell matrix indicated a mean collapse load of 3230lb, in comparison to the static load of 2900lb (11% increase). The collapse load rose gradually over the duration of the impact, Figure 5.21.

Type II: Tube 10c had reusable push-on end fittings, and collapsed in an acceptable manner, but since this was the first test conducted, the energy input was

insufficient to allow for a high crush distance. The impact velocity was 3.09m/s, but was increased for subsequent tests. However, collapse took place at an approximately constant load (3330lb), Figure 5.21. By comparison, the static failure load was 3000lb (again 11% higher). The post test specimen is shown in Figure 5.3.

Tube 12a again had the shrink-tape trigger, but with bonded end fittings. The shrink-tape trigger failed to operate, and failure was initiated from the wrong end of the tube. The collapse is shown in Figure 5.5. Failure due to the tube being subjected to combined compressive and bending loads was suspected, created by pitching of the trolley on impact. In its given application, this loading (and hence behaviour) should not arise. However, the outcome of this test is worth noting, but the collapse loads are not worthy of analysis due to the failure mode.

Type III: Tube 12c was tested with push-on end fittings and collapsed by 'scrolling' over the end fitting, Figure 5.6. This method of failure is undesirable for the application required, although the collapse load (3920lb) was steady and the energy absorption high. This mode of failure could be investigated for other applications (with no tensile load requirement) but was outside the scope of this project, and has thus not been pursued. Failure in this mode would have been prevented if the end fittings had been bonded on.

Tube 14c was the bonded version of the Type III tube, and was tested at an impact velocity of 3.63m/s. The mean collapse of the load was 3490lb, compared with 2750lb statically some 27% higher. A satisfactory shape for the force-time relation was obtained, Figure 5.21. The post test condition of the energy absorber is shown in Figure 5.8. Separation of the two outer layers of +/-45 cloth was evident at the change cross-section.

Type IV: This tube type was rejected for the reasons previously stated.

Type V: Tube 14a was a Type Va tube and the length of the reduced cross-section part was 30mm; this was tested at 3.67m/s. The mean collapse load recorded was 2805lb, with controlled collapse and tight folds, Figure 5.7. The force-time trace was satisfactory and is shown in Figure 5.22.

Tube 15c was Type Vb (25mm 'reduced cross-section' trigger length) tested at 3.60m/s, with the load fairly constant at 2400lb. Again collapse was controlled, with the slider operating correctly, Figure 5.12. The decrease

in the length of the reduced cross-section part had the effect of minimising the initial energy shortfall.

Type VI: As previously discussed, the proportion of +/-45 fibres was increased in comparison to Type V. As expected, collapse loads fell slightly. Tube 14b was of Type VIa, collapsing at a mean load of 2760lb, and displaying good crush characteristics, see Figures 5.22 and 5.9.

Tube 15b (Type Vb) gave a slightly improved pulse shape, otherwise collapse was as for 14b, with the mean collapse load reduced to 2240lb. The energy absorber is shown in its post-test condition in Figure 5.11.

Type VII: This final type tested had only one layer of 0/90 fibres, with three layers of +/-45. Collapse load was 2260lb, with constant collapse characteristic, but high initial energy loss, Figure 5.22. The post-test condition of tube 15a is shown in Figure 5.10.

5.12 TENSILE TEST

Tube 14a had been tested dynamically in compression, Figure 5.7, the length of the kevlar tube being 145mm after this test. The tube was then subjected to a tensile test, and pulled back to its initial length, 250mm. The load-displacement trace is shown in Figure 5.23. The initial resisting force was only 500N for the first 19mm (0.75"), increasing to a maximum of approximately 11.3kN (2540lb) before the load was removed due to fibre tearing.

The low initial load was as a result of the incomplete crush of the energy absorber, and would not have occurred if the damage zone filled the end fitting between the slider and the flange. The subsequent increase in load is as a result of damaged material being pulled back into shape as it passed over the slider in the 'wrong' direction. The appearance of the tube during and at the end of the test is shown in Figure 5.13. There was no evidence of debonding of the end fittings.

If the mass of a triple seat is taken as 75lb, and that of the occupants as 170lb each, this makes a total of 585lb. Assuming that the total tensile load is reacted by the front legs only (equally), then the mean tensile resisting load of 1.3kN is equivalent to 1g and the ultimate failure load equivalent to 8.7g. This was felt to represent acceptable performance, taking account of the likely application of the energy absorber.

5.13 SUMMARY

- (1) A (basically) rectangular load-displacement relationship can be obtained for the kevlar energy absorber with a single trigger method, but with the penalty of an initial energy shortfall of 3-6% indicated from static tests. The load variation about the mean value during collapse was typically +/-15 to 20%.
- (2) The load at which failure was initiated could be varied according to the method of triggering failure which was used.
- (3) Use of a trigger comprising a double wrap of shrink-tape close to one end of the energy absorber, resulted in initial failure at a load about 55% higher than the mean collapse load.
- (4) Use of a trigger comprising a reduction in cross section 30mm from one end of the tube resulted in initial failure at a load about 15% higher than the mean collapse load.
- (5) Use of a combined shrink-tape and reduced cross-section trigger resulted in an initial peak load comparable to the mean load. However, the subsequent reduction in the load carrying capability resulted in a reduction of over 10% in the energy absorbing ability of the unit.
- (6) The mean collapse load for a tube of given geometry could be varied by changing the fibre orientation. In general, increasing the proportion of 0/90 fibres in relation to the proportion of +/-45 fibres increased the mean collapse load.
- (7) Bonding of the end fittings as opposed to push-on end fittings typically raised the failure loads by about 5%.
- (8) An end fitting has been designed which stabilised the collapse for relatively long tubes ($L/D=5$), with no evidence of undesirable collapse behaviour during static tests.
- (9) Following on from static tests a series of dynamic tests were performed on energy absorber units based on kevlar tubes with alloy end fittings, incorporating an internal slider. Some tests were conducted with push-on end fittings in order that they could be reused, but this practice was stopped following poor dynamic performances.
- (10) Dynamic test results indicated measured loads to be 11-27% higher than in the equivalent static compression tests.

(11) With bonded end fittings, only one tube failed to operate correctly. This was due to failure of the shrink-tape trigger to operate, and the tube failed in bending from the 'wrong' end. With the type of spherical bearings to be used this effect would probably be eliminated, although use of the (alternative) reduced cross-section trigger could be considered preferable, because the initial collapse load can be more readily tuned to the design requirements.

(12) Tests conducted to establish post-crush structural integrity and tensile load-carrying capability were carried out, and indicated an ultimate residual tensile strength in excess of the mean compressive failure load.

CHAPTER 6

DEVELOPMENT AND DYNAMIC TESTING OF COMPOSITE AIRCRAFT SEATS - STANDARD AND ENERGY-ABSORBING TYPES

6.1 INTRODUCTION

Following the design of suitable kevlar energy absorbing units, various modifications to the existing seat structure were necessary in order to incorporate an energy absorbing capability. To assess the dynamic performance of energy absorbing seats in comparison to standard designs, a series of dynamic tests was conducted, supplementing the computer simulation work. Limitations in the amount of available test parts resulted in a relatively small number of tests being conducted, supplemented by a series of post-test parametric studies being conducted using further computer simulation, as described in Chapter 7.

The object of the tests was to assess the structural performance in various configurations and the effect of imposing warped floor conditions on the structure, together with assessing the effects on occupant kinematic behaviour. It was initially intended to test both standard composite seats and modified (energy-absorbing) seats in each of three dynamic test configurations.

6.2 SEAT REDESIGN TO INCORPORATE ENERGY ABSORBERS

The particular composite seats being used for this series of tests were available with two alternative types of leg, known as the C-leg and the A-leg. The geometry of the legs in terms of floor fitting attachment points and seat pan attachment points were identical, and both the front and rear types of floor-track fitting were common to each type of leg. It was essential that the A-leg type was used (in modified form) for the energy-absorbing seat, as the C-leg did not lend itself to modification for such an application. As only a single pair of A-legs were available, the C-legs were used for the standard seats. This hardware shortage was due to the unforeseen closure of the UK composite seat manufacturing facility, by the American parent company.

This use of the C-legs for the standard seats was considered to be satisfactory, as the object of the testing was to directly compare an effectively rigid seat design to an energy-absorbing variant with identical geometry. One problem proved to be that the same set of legs had to be used throughout the test programme for the energy absorbing seats.

In general, the seat lent itself well to the proposed conversion, with the pan having a large surface area over which leg and buttock loads could be spread, and a generous radius at the rear to minimise the likelihood of shin injuries from the seat in front.

6.2.1 Seat Leg Modification

The object of the modification was to allow for incorporation of energy absorbing elements in the seat structure. The use of existing leg structure as the basis for a test prototype was the most expedient route to commence testing. It was realised at an early stage that the reduction in the lateral strength as a result of the modifications would require minor redesign of the rear legs, and the original intention was that such a modified design would be specially manufactured. Due to the subsequent lack of manufacturing capability, this was not possible and it was not therefore possible to test the seat successfully in the lateral direction.

Hence the A-legs were modified to test forward-facing with both horizontal and combined vertical horizontal loading conditions. (At this stage it was intended to have a modified set of legs made to test with a combined forward and lateral loading with 10 degree yaw). As described in Chapter 2, the legs were of carbon-epoxy composite with a foam core. The front portion of each leg was cut off and a steel forked mounting plate bonded as shown in Figure 6.1. This was constructed in such a way that a pin-jointed fitting such as a spherical bearing could be used at the top of the energy absorber.

The legs (and hence the seat structure) were thus intended to pivot about the lateral pin-joints at the rear floor-track fittings as the energy absorbers stroked downwards. The fork was designed in such a way that the distance from the lateral pins at the rear floor mounting to the upper energy absorber pin-joints equalled the distance from the rear floor-track mountings to the pin-joint at the front floor-track mounting (at the bottom of the energy absorbers). Any production design would have tie-bars from the front to the rear track fittings for ease of installation, although manufacture of the parts was not deemed necessary for these tests.

Forked fittings were used at the lower ends of the energy absorbers in order that standard (rear) floor-track fittings could be used with pin-joints. Due to the geometry of the set of legs available, the length of the composite tubes was reduced to 190mm. With the length of the slider stems reduced to 50mm, this allowed 80mm of crush.

The asymmetry of the seats was such that the static loading on the legs was nominally in the ratio 2:1. Hence an energy absorber with a mean crush load of 3000lb was used on the side with maximum overhang, and 1500lb on the other side. The collapse load was chosen as a result of the simulation effort described in Chapter 3, taking account of the change from a single-occupant seat to a triple-occupant seat, and the findings regarding load drop-off discussed in Appendix E. The simulations described were also repeated with pulses similar to those which were to be used in dynamic testing, indicating that approximately half the amount of stroke would be used as would be the case for testing to FAR 25.562 at 16g and 44ft/sec.

All tubes used a reduced cross-section for a length of 25mm at the top of the tube, with the top 12mm being bonded to the end fitting. The stronger tube was constructed using 5 layers of aramid pre-preg reducing to 3 layers at the top, whilst the weaker tube used 3 layers reducing to 2 layers at the top.

When the seats were weighed, they were devoid of armrests, food trays, cushions and trim. The modified seat complete with energy absorbers weighed 13.8kg compared with 13.5kg for the standard seat with C-legs. This weight penalty is negligible given that the composite design can give a considerable weight saving over equivalent metal designs.

It is very difficult to categorise seats which use lap-belt restraint systems in terms of dynamic collapse loads. For example, one could not say that a seat strokes at 16g dynamically, because different occupant kinematics will be associated with different '16g' deceleration pulses, producing different seat loading regimes. For example, the pulse shape and the velocity change applicable will influence the loads applied to the seat and belt anchorages. The seat is loaded primarily by a combination of restraint system and seat pan loads, many permutations of which could result in the seat stroking (for a given direction of inertial loading).

A more general means of expressing the load at which stroking begins is to use static test criteria. It would perhaps be appropriate to quote Chandler (1985) regarding the dangers of designing to such criteria: "For example, consider a seat in which the front legs are designed to limit the load on the occupant by collapsing at a pre-determined level as an energy absorber. If such a seat were made so the front legs collapsed under the action of a body block the loads required to collapse the legs under dynamic conditions may be so much greater that the protective capability of the load limiting concept would be lost".

6.3 COMPOSITE SEAT DYNAMIC TEST PROGRAMME

In order to verify results obtained in computer simulation work and to broaden the database of knowledge built up during the course of this work, a series of dynamic tests on composite seats in standard and modified trim was planned. The results obtained for the tests on standard seats would form a baseline, both for comparison with the energy absorbing version of the seat and for use in parametric studies conducted using computer simulation programs SOM-LA and SOM-TA, as described in Chapter 7.

6.3.1 Test Configurations

Three types of test configuration were used, based on the scenarios associated with survivable aircraft accidents. These are described in detail in Chapter 1 and are again outlined below:

(A) 0 degrees pitch, roll and yaw, horizontal change in velocity in the range 28-33ft/sec, peak deceleration 14-18g. Seat back to be positioned in front of occupant at the equivalent of 30" seat pitch to study secondary head impact.

(B) 30 degrees nose down pitch, zero roll, zero yaw, vertical change in velocity in the range 28-33ft/sec, peak deceleration 12-16g.

(C) 0 degrees pitch and roll, 10 degrees of yaw, but with 'floor warpage'. Floor is warped quasi-statically, to give 10 degrees of roll for one seat track, 10 degrees of pitch (forward) for the other, prior to dynamic testing. Horizontal change in velocity in the range 28-33ft/sec, peak deceleration 12-16g. Seat back positioned in front of occupant to study secondary head impact.

The deceleration levels quoted were based on those sustained for a minimum of 5ms rather than instantaneous peak levels obtained.

6.3.2 Tests Conducted and Results

As described earlier, it was initially proposed to conduct tests on both standard and energy absorbing seat designs in each of three configurations. The test numbering system used is merely chronologically based, and is discontinuous, as the test facility was also used for contract test work during this time period.

Static tests were also conducted on energy absorbers used in the seat tests, where the full available crush distance was not used during the dynamic test, to check for stable collapse over the complete stroking distance. These tests are described briefly in Section 6.7.

6.4 DYNAMIC TESTS CONFIGURATION (A)

6.4.1 'Standard' Composite Seat

The standard seat was tested, in a forward facing configuration with zero pitch or yaw, in Test 24. The seat is shown immediately prior to test in Figure 6.2a and after the test in Figure 6.2b. Impact velocity was 8.3m/s (27.2ft/s), and the deceleration-time pulse measured on the sled is given in Figure 6.16. When the baseline error is removed, the pulse has a nominal peak of about 17g, if levels attained for less than 1ms are discounted. This represented the standard pulse for this configuration, the differences between crush tubes being insignificant, since they were all cut from the same original length of tube.

The longitudinal component and vertical component of head acceleration are given in Figures 6.17 and 6.18 respectively (the lateral component was negligible). Dynamic response of the occupant is relatively slow, with most of the head deceleration prior to impact being in the vertical direction (relative to the head), with a period of sustained 30g deceleration immediately prior to head impact. Secondary head impact (with the seat-back positioned in front of the occupant) took place at 137ms, and is clear from the longitudinal component trace. The peak is 73g in the longitudinal direction. This was accompanied by a reversal in loading in the vertical direction on impact.

Head resultant acceleration-time history (Figure 6.19) shows the lower g, higher duration deceleration phase followed by the higher g, high onset, low duration spike characteristic of secondary head impact. Following this, the seat back broke forward with a constant resisting moment, but as the point of contact moved down the seat back and the angle between the head and the seat-back changed, then the force increased as the final head deceleration took place.

Head Severity Index calculated by the post-processing software was 330. This would therefore be classed as a survivable impact.

Seat structural damage amounted to a broken (left outboard) stringer, with a lateral split in the seat pan at the rear of the spar from the broken stringer towards the inboard side of the left seat place. A view from underneath the seat shows the post-test damage, Figure 6.3. Any effect this would have had on an occupant cannot be quantified, as the test dummy was occupying the centre seat place. However, it is a very undesirable situation when the stringer (carrying the restraint system anchorages) begins to separate from the seat pan, as this represents a load-path discontinuity. Study of high speed

film shows a high degree of relative movement of the ballast in the relevant seat place and this could well have had a detrimental effect on an occupant. In a situation where the velocity change (and therefore energy) is higher, this could have resulted in a more serious problem.

Other minor damage occurred, including elongation of the lap belt mounting rings and downward deflection of the seat pan 'nose', the section in front of the main spar. This was particularly evident in the centre seat place, and reflected slight differences in the loading caused by the dummy and that caused by ballast. Both these aspects of structural deformation were deliberate load-limiting design features.

6.4.2 'Energy-Absorbing' Composite Seat

The modified seat was tested in the forward facing horizontal attitude, Test 17. The impact velocity was 8.6m/s (28.3ft/sec) and the standard 17g nominal trolley deceleration pulse was used. As for all this series of tests, a triaxial accelerometer was fitted in the head of the test dummy.

The longitudinal component of head acceleration (Figure 6.20) was negligible to 140ms, and the head struck the seat-back at 165ms. The energy associated with this secondary impact was very low and produced a 27g peak, compared to 73g when the standard seat was tested. A second peak was again produced shortly before 200ms, indicative of the seat-back rotation having reached the limit of available travel.

The vertical component of head acceleration-time history was similar (in comparison to Test 24) up to 80ms, Figure 6.21. Differences occurred in the period 80-180ms. Beyond 80ms, the level rose progressively to a peak of 25g at 140ms, in comparison to 30g at 120ms and sustained at that level to 140ms. This change in behaviour coincided with operation (stroking) of the energy absorber elements. The much less violent jackknifing of the body produced in this case resulted in a head impact which took place later (by 25ms) and at lower velocity, and hence produced lower severity. The vertical component of the head strike was negligible, presumably because the seat-back had rotated forward more due to the later impact.

The lateral component again showed no significant levels, as would be expected for this test configuration. Head Severity Index was 74, an extremely low level when considering the configuration, restraint type and g levels. Resultant head acceleration is shown in Figure 6.22.

Post-test examination of the seat structure (see Figure 6.1) revealed that the energy absorbers had stroked by an average of 35mm, and the tube collapse appeared stable, with end fittings parallel and on a common axis. As for all tests on the modified seats, the energy absorbers were to be tested statically for the remainder of their travel to confirm stable collapse. These tests were performed successfully and the details given in the relevant section at the end of this chapter.

Visual examination of the seat revealed no structural damage (such as had been found when standard seat legs had been used) except for a slight downwards deflection of the nose section of the seat pan similar to that produced in the previous test.

6.5 DYNAMIC TESTS CONFIGURATION (B)

6.5.1 'Standard' Composite Seat

The second configuration involved testing the seats pitched back by 60 degrees and tested on the sled with a mainly vertical and partially horizontal change in velocity, to simulate a heavy landing. This configuration is described in detail in Chapter 1. Due to the restrictive size of the sled, it was not feasible to include the seat structure in front of the occupant. Whilst head contact was unlikely in this configuration at the energy levels used, shin contact with the rear spar of the seat in front was likely, and hence the refinement of this test configuration would be recommended for future work. The inclusion of a 'floor' to support the occupant's feet would also be a worthwhile modification.

The standard seat was tested in this configuration at an impact velocity of 8.15m/s (Test 27), the crumple tubes used to decelerate the trolley for this group of tests producing peak levels of about 20g, after deduction of the baseline offset, Figure 6.23. For the purposes of later simulation work, requiring a 16-point approximation, this was treated as a 17g plateau level, see Chapter 7.

Analysis of the components of head acceleration indicated a reasonably rapid response to the input deceleration, any delay being as a result of the seat cushion loading-up and of spinal compression. Unfortunately older types of dummy, such as that used, do not have provision for the inclusion of a lumbar load cell to monitor compressive spinal loading. Resultant head acceleration levels (Figure 6.26) similar to those input were attained, with the direction changing from vertical to longitudinal at around 100ms, as the head rotated forward and downwards, Figures 6.24 and 6.25. As mentioned previously, this configuration produced

consistently lower severity levels for lap-belted occupants than was the case for the horizontal impact case. A Head Severity Index of 57 was indicated for this test.

No visible structural damage to the seat structure was caused as a result of this test, Figure 6.4.

6.5.2 'Energy-Absorbing' Composite Seat

Test 26 involved testing a modified seat at 8.15m/s (26.7ft/s), in the combined vertical and horizontal configuration, with nominally the same deceleration pulse as used for the standard seat, Figure 6.23.

Study of high speed film indicated that the energy absorbers began stroking at approximately 35ms. The effect on head acceleration was evident on the trace for the vertical component (Figure 6.28) at about 40ms, when the level reduced to around 10g. This reduced to zero at about 120ms, with no evidence of subsequent longitudinal deceleration levels (Figure 6.27). This might reflect the fact that energy had been taken out of the system as the seat had stroked. The Head Severity Index for this impact was 41, and resultant head acceleration is shown in Figure 6.29.

Following the crushing of the tubes during the impact sequence, the energy absorbers returned slowly to their original length due to the static loads caused by the mounting method. When using a sled to simulate vertical components of deceleration, the direction of the gravity vector is different to that which would normally exist in an impact. As with all the energy absorbers used in dynamic seat tests, subsequent static compression tests were conducted. From the results of these tests it was possible to accurately assess the amount of crush obtained during the dynamic tests, as described in the later section in this chapter. The mean crush obtained during this test was 30mm. Apart from the controlled collapse of the energy absorbers, no structural failure of the seat took place, Figure 6.5.

Pre-test simulation work (Chapter 3) had indicated that under the test conditions described in FAA regulations for a similar test configuration (14g / 35ft/sec), no stroking of the energy absorbers would take place. Factors contributing to the different behaviour were the (higher) loads generated, together with the lack of floor support for the feet, causing an increase in the compressive loads generated in the seat support structure and hence the energy absorbing elements. If a floor had been installed, it would only have been loaded by the centre occupant, since a full complement of three anthropomorphic test dummies was not available. The proportion of the load

transmitted through the seat structure (and energy absorbers) would therefore be artificially high. This aspect is discussed in more detail in the post-test parametric study, Chapter 7.

It is perhaps worth noting that Test 26 was in fact a repeat of Test 25, where similar occupant kinematics and energy absorber stroking levels were observed, but failure of the power supply to the data recorder resulted in a total loss of transducer output.

6.6 DYNAMIC TESTS CONFIGURATION (C)

6.6.1 'Standard' Composite Seat

The third test configuration involved two stages, namely the quasi-static floor warp test and the subsequent combined forward and lateral dynamic test, achieved by mounting the seat on the sled at a yaw angle of 10 degrees.

As explained previously, the first stage was to induce 10 degrees of roll in one pair of legs (in this case the left legs, rolled outwards) and the second stage was to induce 10 degrees of pitch (in this case the right legs were pitched down).

During the static induction of floor deformation particularly high forces were needed to induce the deformation in relation to seats tested in the past. This was partly because those composite seats tested with A-legs (as described in Chapter 2) had glass fibre spars, whereas these seats had carbon fibre spars. The pitching was induced by tightening a large bolt at the front of the jig, and very high torque was needed due to the high stiffness and torsional rigidity of the structure. As the test proceeded, there was audible evidence of significant internal delamination, accompanied by visible damage.

A damaged area appeared on the outer face of the rear flange of the left leg, where the web and flange meet, Figure 6.6. Deformation of the front flange of the same leg was also evident in a central vertical area where the skin thickness was lower, Figure 6.7. The loading of this leg was complex, but included bending induced by the 10 degrees of roll, torsion about a lateral axis (induced by the pitching deformation and consequent spar torsion) produced compression in the front flange and tension in the rear flange. Also torsion about a vertical axis, produced as a result of fixing the rear anchorage, thus caused the right side of the spar to move forward as the front right anchorage was pulled down. These effects are illustrated in Figure 6.6.

Severe spar deformation was also evident, with the warping shown clearly in Figure 6.9. Due to the extent of the audible delamination and visible deformation and damage, a decision on whether to test the seat dynamically was deferred. In the intervening period, surface deformation appeared on the seat pan where the upper skin of the composite had separated and lifted from the foam core, Figure 6.8.

At this stage in the test programme, shortage of hardware, namely seat components, had become a problem. It was thought that as the legs and the pan would be no use for subsequent tests, this test should proceed, even though the seat might not have survived.

It was decided to test the seat at a lower velocity than had been planned to reduce the energy input, as there was clearly a significant amount of energy stored in the seat structure and the trolley due to the quasi-static deformation. There was evidence that significant warping of the sled itself had taken place, with the left rear tyre only in intermittent contact with the track, but the wheel on the underside of the track in contact (this would not of course normally be the case). This was despite the sturdy construction of the sled, and brought into question once more the realism associated with a test that required a certain amount of deformation to be induced without regard to the magnitude of the reaction loads, or indeed the load carrying capability of the floor beams in the relevant aircraft.

The seat was tested at 7.95m/s, although erroneous triggering of the impact recorder resulted in no transducer output. The seat structure remained intact, although the amount of visible damage increased. The nose of the seat pan collapsed and cracked at the left end where the nose and the span meet (Figure 6.10) and at the centre seat location. The value of the dynamic part of this test was limited by the recorder failure, since A-leg versions of the seat had already been tested (with non-instrumented dummies) in this configuration during earlier test work, see Chapter 2.

6.6.2 'Energy Absorbing' Composite Seat

Due to the closure of the manufacturing facility by the seat manufacturer, it became evident during the series of tests that it would not be possible to obtain seat parts beyond those originally obtained. These included only one pair of A-legs, which had been converted to include energy absorbing elements and used in three previous dynamic tests. Whilst there was no visible damage, it was not usual to subject such components to more than one dynamic impact test. It was the original intention to have a second set of (different) legs made for the test which

involved a lateral component of deceleration, configuration (C).

The reason behind this was the reduction in lateral strength resulting from using energy absorbing elements. It was necessary to use spherical bearings at the energy absorber mountings in order not to load the units in bending. However, this meant that the lateral loads must all be reacted by the remaining seat structure, and hence the lateral strength would need to be increased by increasing the width of the legs at the top where they join the spar and pan assembly. As no such legs were available, the results are limited to the quasi-static floor warp test.

The loads needed to induce the 'floor warp' requirement were very much lower than those needed for the C-leg seat, and for the purposes of discussion can be compared to those experienced when testing the standard A-leg seat, as described in Chapter 2.

For those tests, the rear leg-to-track mountings were improved by the addition of cheek plates and the use of special track fittings using spherical bearings. Such fittings were not available for this test, and suitable substitute parts were manufactured, Figure 6.11. Crush tubes on the pan-to-leg connections were again used, both to promote easy comparison and to get a measure of the relative magnitude of the forces generated at those locations.

The presence of the spherical joint at the top of the energy absorber, in comparison to the continuous member present in the standard A-leg, provided a bending moment release and hence increased compliance to floor warping (Figure 6.12), but reduced lateral seat strength (as discussed above).

No seat damage or permanent deformation occurred during the floor warping test on this seat.

6.7 STATIC ENERGY ABSORBERS TESTS

The energy-absorbing seats used in the dynamic tests were dismantled in order that static tests could be performed on the individual energy absorbing units. The purposes of these tests were as follows:

- (i) To ensure that controlled collapse would continue for the complete stroking distance.
- (ii) To ensure that collapse load would be maintained at the levels measured during previous static tests.

(iii) To calculate the crush distance used during dynamic tests in configuration (B), where the energy absorber had 'pulled back' to its original length due to the nature of the test. This was due to the unrealistic seat mounting necessary when sled testing, as explained previously.

The tests were performed successfully, with the post-test condition shown in Figures 6.13 and 6.14. The stroking distance associated with the dynamic test was obtained by monitoring the point at which the load carrying capability rose to the normal level. The figures obtained are included in the results of the dynamic tests and discussed in the preceding sections.

For the purposes of direct comparison, two energy absorbers of the type used in the dynamic tests are shown in unused and fully stroked condition respectively, Figure 6.15.

6.8 SUMMARY

(1) One pair of carbon-fibre A-legs were converted to accept energy absorber elements in place of the front part of each leg. This was the simplest possible modification using existing components. The seat would pivot on the lateral pins at the rear floor-track fittings and stroke forwards and downwards at force levels controlled by the energy absorbers, allowing a maximum stroke of 80mm before bottoming-out progressively.

(2) Seats fitted with these legs were tested dynamically, as were 'standard' seats which used C-legs. Occupant kinematics and injury criteria together with seat structural performance and damage were compared.

(3) A 'standard' type seat was tested in configuration (A), to assess the behaviour in longitudinal decelerations. Structural failure of the outboard stringer would have affected occupant kinematics, but this seat place was balasted rather than occupied by an instrumented dummy. This failure led to lateral splitting of the seat pan, which could have led to complete seat failure in situations of higher velocity. Severity levels measured on the centre occupant included a peak head deceleration of 73g and a Head Severity Index of 330.

(4) A modified 'energy-absorbing' seat was also tested in configuration (A), and stroked a distance of 35mm. No damage to the seat occurred, with peak loads having been attenuated. Peak head acceleration of the dummy was 27g, with a Head Severity Index of 74. Clearly this represented a significant reduction in injury potential, and was indicative of a secondary head impact of relatively low energy.

(5) A 'standard' seat was tested in configuration (B) to assess performance in conditions of combined downwards and forwards inertial impact loading. No seat structural damage occurred, and Head Severity Index was 57.

(6) An energy absorbing seat tested under similar conditions stroked by 30mm and produced an HSI of 41. If it had been possible to use three dummies and a floor structure or other foot supports, the reduction in loads might have resulted in little or no seat stroking at this level of velocity and acceleration.

(7) A 'standard' seat was deformed quasi-statically to simulate conditions of floor warpage. The only seat pan available for this test had a carbon-fibre spar rather than a glass-fibre item, requiring the use of extreme loads to produce the required deformation. A high degree of seat damage resulted.

(8) This seat, whilst still in the deformed state described, was tested in configuration (C). The seat remained intact although further damage occurred including spar breakage. Higher energy levels could have resulted in seat failure. Erroneous triggering of the impact recorder resulted in no instrumentation data being collected.

(9) The closure of the composite seat manufacturing plant created a shortage of seat parts, including A-legs for modification and precluded the possibility of having a set of legs made which had increased lateral strength, needed as a result of incorporating energy absorbers. Thus it was not possible to successfully perform lateral tests, or tests with a significant lateral component of acceleration (configuration (C)).

(10) Despite the above, it was possible to perform floor warp tests on an 'energy absorbing' seat in the form used for the other dynamic tests. Due to the various releases at the floor fittings, and the increased compliance resulting from the conversion, floor warping was induced at lower force levels than for the standard seat (equipped with either C-legs or standard A-legs), see Chapter 2.

(11) All energy absorber units used in the dynamic tests were further compressed in a static test machine to ensure continued controlled collapse at sustained load levels. Such behaviour was confirmed.

(12) Results from the programme of tests were used in validation of a baseline computer seat and occupant model, and the use of the model in a series of parametric studies. This work is described in Chapter 7.

CHAPTER 7

POST TEST-PROGRAMME PARAMETRIC STUDIES USING COMPUTER SIMULATION PROGRAM SOM-TA

7.1 INTRODUCTION

Following on from the test programme on standard and energy-absorbing composite seat designs, a wide-ranging parametric study was performed using computer simulation program SOM-TA.

This took the form of four distinct study areas:

- (1) A refined model of the horizontal test on the rigid seat design using a triple seat in SOM-TA, to include head impact with the forward seat back. This would serve as a baseline model for the other studies described below. This model would also be used to simulate the combined vertical and horizontal deceleration, Test 27.
- (2) A study of the effect of the presence in the head strike envelope of the forward seat back at different seat spacings.
- (3) A study of the effect of seat-back design on the secondary head impact event, in particular the seat-back breakover characteristics and local stiffness properties.
- (4) A study of the effects of adopting a crash position prior to impact, through the effects of occupant initial position on injury severity measurements.

7.2 PARAMETRIC STUDY - DEVELOPMENT OF A BASELINE MODEL AND CORRELATION WITH DYNAMIC TEST DATA

7.2.1 Introduction

The development of a baseline model was useful for a number of reasons. Firstly, the output from the simulation could be compared to test results, and conclusions drawn regarding the degree of correlation between the test and the model. Secondly, information not available from test could be gleaned from the simulation, assuming the latter to have more sophisticated output. Thirdly, further studies could be performed to assess the effect of parametric changes which are relevant to the survivability of the occupant and the crashworthiness of the seat.

The baseline model was developed as a rigid triple seat with a seat back positioned in front for the

forward-facing test, and therefore was effectively a simulation of Test 24. As a check, the pulse was also changed to that used in Test 27, the combined vertical and horizontal deceleration test.

7.2.2 Description of Baseline Model

The object initially was to model Test 24 as accurately as possible, using the standard composite triple seat, with three occupants in a forward facing orientation and an horizontal 17g deceleration vector.

With a lap-belted occupant, a greater degree of limb flailing (and hence dynamic overshoot) typically occurs when the deceleration vector is largely horizontal, and secondary impact with the forward seat back normally occurs. Hence this case provided a more exacting validation of program accuracy. Simply changing the input pulse and removing the forward seat back enabled Test 27 (combined vertical and horizontal loading) to be simulated.

The seat used in the baseline was configured such that it could be of either Finite Element or rigid type. The method used was to create a Finite Element model based on the correct geometry. This was an assymetric triple seat developed from the single seat used in SOM-LA, see Chapter 3. As no significant seat deformation was being modelled, the FE data could then be omitted. It was found that the occupant output was identical (as expected), so where seat structural output was not required, a 'rigid' seat was used to minimise computational costs.

The seat back and pan angles and dimensions were correctly reproduced, together with lap belt anchorage geometry. Force-deflection characteristics for the lap belt and seat cushions were input. Zero lap belt slack and 3" thick cushions were used, as in the tests. A forward seat back was positioned at 30" seat pitch in front of the occupant, with the dimensions and break-forward moment measured directly from that used during testing.

The occupant models were chosen as three dimensional 50th percentile males, with initial position determined by seat and headrest angles. Tabulated and graphical output from the program was based on the centre occupant as standard.

The horizontal input deceleration pulse was based on that obtained during Test 24 (Figure 7.1), using a sixteen point approximation as shown in Figure 7.2, representing a velocity change of 27.22ft/sec.

Similarly, the pulse from Test 27 (Figure 7.3) was approximated for simulation purposes (Figure 7.4) for the combined horizontal and vertical deceleration, with a velocity change of 26.8ft/sec.

7.2.3 Results of Baseline Simulation and Comparison to Test 24

The output from running the simulation included occupant kinematic plots and acceleration-time histories for the head, chest and pelvis. As discussed previously, the most important of these was the head (for the purposes of this test configuration) hence this was where the instrumentation was positioned during these tests.

Comparisons of the various head acceleration components, resultant head acceleration output, severity indices, occupant kinematics and seat break-forward behaviour can be used to assess the degree of correlation between test and simulation.

Study of the head resultant acceleration-time histories (Figures 7.5 and 7.6) revealed a high degree of correlation, with an initial (rounded) peak of about 40ms duration with a mean about 20g and peak level 30-35g. This was followed by a spike of about 5-8ms, with a peak level of 70-75g. A further (lower) peak occurred some 40-50ms later. It was also evident that the various features mentioned occurred about 25ms later in the test than in the simulation. This is explained later in the text.

Study of the components of head acceleration also yielded useful information. Good agreement was obtained in the general shape of the longitudinal component of head acceleration, Figures (7.7 and 7.8). Initially, low (5g) acceleration of the head occurred. Forward speed was then reduced as the effect of floor deceleration was transmitted through the seat and restraint system, and then head impact occurred. This reduced the forward velocity of the head drastically over a short time period, producing a high peak. Further reduction then occurred as the head was brought to rest.

Head Severity Index showed satisfactory agreement, 330 for Test 24 and 334 indicated from the output of the baseline simulation.

The level of lateral (y-component) head acceleration was of course negligible for the forward-facing impact orientation. The vertical (z-component) head acceleration was negative initially, as the whole body moved downwards as the cushion compressed (due to the lap belt method of restraint), see Figures 7.9 and 7.10. It then went slightly positive as the body became upright. Subsequent

tensile loading in the neck and spine decelerated the head until impact. Reversal of the loading occurred upon head impact.

Study of high speed film revealed the possible reason for an apparent time lag of 25ms during the dynamic test when compared to the baseline simulation. Upon impact, the dummy could clearly be seen to slide forward with no change in attitude, with the relative positions of the body segments remaining the same. This was due to either lap belt slack or low initial belt stiffness. Subsequent kinematic behaviour indicated good agreement with simulation. During the period of 25ms when this slack was being taken up, the seat back in front was of course breaking forward under its own inertia. Therefore, when secondary impact occurred, the breakover angle was slightly higher in the test, the difference being about 6 degrees.

Due to the mathematical nature of occupant simulation, it lends itself to providing the fullest possible selection of output parameters, whereas this may entail huge expenditure if comparable instrumentation were to be utilised for dynamic testing. For example, load cells or other force transducers at every contact surface between dummy body segment and seat, floor and restraint system is beyond the level used at even the most sophisticated test facilities.

Hence the simulation of Test 24 can give an indication of parameters for which no instrumentation was present. These include belt loads, seat and floor reaction loads, and chest and pelvis acceleration-time histories.

Restraint system loads and seat cushion forces are best analysed if considered together, Figures 7.11 and 7.12. Lap belt load increased until 35ms, then a dip occurred coincident with increased seat cushion loads. As the lower limbs were thrown forward, cushion forces reduced to 150lb and lap belt forces increased, peaking at about 1250lb, after 100ms. Beyond 110ms, the lower limbs moved back down and the head impacted the seat back. The occupant was loading the seat cushion and the seat-back in front, and lap belt loads reduced to zero for a period of about 20ms. After 145ms, the bias changed again as loads were reacted more through the lap belt than the seat, and the buttocks rose due to the angle of the thigh contact with the front portion of the seat base.

Pelvic acceleration-time history (Figure 7.6) clearly bore much more resemblance to the input (sled) deceleration pulse than that of the head, with a lap belt restraint, one would expect this. The chest deceleration was higher than the pelvis but lower than the head, also shown in Figure 7.6.

7.2.4 Baseline Model with Combined Vertical and Horizontal Deceleration

As further validation of the baseline model, the deceleration pulse was changed to simulate a combined vertical and horizontal loading based on that measured during Test 27.

The resultant head acceleration-time history from the test is shown in Figure 7.13, with the head, chest and pelvis accelerations from the simulation shown in Figure 7.14. Comparison shows a broadly similar response, with a peak of 17g at 38ms recorded in the test, compared with 22g at 42ms in the simulation.

The Head Severity Index was 57 from the test and 58 from simulation, with a Chest Severity Index of 64. Other output from the simulation gave a useful insight into likely levels of other injury measurements. For example, peak vertebral compressive load (Figure 7.15) was only marginally below the maximum permissible load of 1500lb.

Restraint system output indicated that loading of the belt occurred only after the peak acceleration levels of the body segments, Figure 7.16. It was indicated from the simulation that head contact with the forward seat-back would not occur in this configuration (assuming 30° pitch).

7.3 PARAMETRIC STUDY - THE EFFECTS OF SEAT ROW PITCH

7.3.1 Introduction

The presence of a seat-back in the occupant head strike envelope will affect the kinematics of the head and hence affect the acceleration-time history of this and other occupant body segments. Furthermore, such direct contact in the form of a secondary impact warrants investigation and the use of further measurements of severity, such as HIC.

Amongst the parameters which will affect such a secondary impact are the position and the properties of the forward seat-back. The seat row pitch will influence which part of the seat-back is struck, and the angle of incidence with the head. Since the seat-back breakover moment-rotation characteristics is nominally constant, the force applied will also vary with the point and angle of contact. The force-deflection properties of the surface struck will also affect the severity of the secondary impact. Seat belt slack and stiffness may also have an influence.

For a particular size of occupant, the seat row pitch dictates the proximity of the seat-back in front. Having tested at 30" seat pitch, the baseline model was used in a parametric study comparing this to 28" and 32" pitches, and also to the case where no forward seat exists within the occupant head strike envelope.

7.3.2 28" Seat Row Pitch

Study of the results for 28" pitch (Table 7.1) indicated that head impact occurred some 6ms earlier than for 30" pitch, with the seat back breakover at impact 20.9 degrees and a slightly higher head velocity. The angular velocity of the seat back was slightly higher at 2.1rad/sec, equivalent to a tangential speed of about 2ft/sec at a radius of 24", and thus has little effect on reducing the head to seat-back relative velocity.

Study of the resultant acceleration graphs for the body segments (Figure 7.17) shows broadly similar behaviour to the 30" pitch case (Figure 7.6), up to the point of head impact. Any slight differences up to this point were as a result of shin impacts at different times. The 'spike' associated with the head impact was of greater magnitude and duration.

The main differences between the two cases were as follows:

- (1) The impact position was higher on the seat-back for the 28" seat pitch, and hence the force required to cause seat breakover would be lower, compounded by the seat breaking forward under its own inertia.
- (2) The time at which secondary impact occurred was earlier, hence the acceleration levels immediately prior to impact were different, affecting the levels during head impact.
- (3) The angle of incidence between the head and the seat-back surfaces was different, since the occupant would be sitting closer to vertical for shorter seat pitches. Thus the angle of incidence would be lower for the shorter seat pitch, producing a greater head-on component. This assumes the same or similar levels of seat-back breakover on impact, 21 degrees for the purposes of this comparison.

The fact that the head impact is 'superimposed' on the acceleration signal (at different points) makes it difficult to separate the effect of each of these factors. Study of the longitudinal and vertical components of head acceleration shows significant increases in both.

The graphs of severity indices (Figures 7.19 and 7.20) for the 30" and 28" pitch cases show that the portion associated with the initial deceleration (up to 112ms) is about 190, with the subsequent increases associated with the secondary head impact. This was very different for the two cases, and perhaps confirms that the severity index for the impact conditions considered in the baseline model (and therefore in test 24) was dominated by the secondary head impact event.

7.3.3 32" Seat Row Pitch

For this case, secondary impact of the head took place at 146ms, well after the initial deceleration phase. Head velocity was hence considerably lower at impact than for the shorter seat pitches, Table 7.1. Head Severity Index was again higher than was the case for the 30" seat pitch case, and with peak head deceleration of 77g compared with 70g, Figure 7.18.

For this case the contact area on the seat-back was lower than the other cases, and the body had pitched further forward upon impact. The effective force necessary for seat breakover was also higher because of the reduced moment arm, and the seat back had actually stopped breaking forward under its own inertia, compounding this effect and to a certain extent offsetting the benefits of lower head velocity.

7.3.4 Large Seat Row Pitches

Simulations indicated that for seat pitches in excess of 44", no secondary impact of any limb with the forward seat structure is likely with a 50th percentile male occupant. Head impact may be eliminated at row pitches of less than 44" (assuming no lap belt slack) but lower limb impacts would not be.

For simulations with lap-belted occupants involving large forward rotations of the upper body segments, and especially the head, programs SOM-LA and SOM-TA have a drawback, in as much as the program stops when the limit of rotation of a particular joint is reached. This meant that it was not possible to run certain simulations for their entirety, a problem recurrent throughout this programme of study.

Such was the case when attempting to quantify the pitch at which head impact would (just) be avoided.

7.4 PARAMETRIC STUDY - SEAT-BACK DESIGN VARIATIONS

The seat-back breakover moment-rotation characteristics will influence the angle at which head-strike occurs, and to a certain degree the relative velocity of head and seat-back. The stiffness of the struck surface will also affect the severity of the secondary impact. Each of these influences is examined in this section, to assess their relative significance.

7.4.1 Seat Breakover Moment-Rotation Characteristic

For a seat-back of given mass, moment of inertia, and surface force-deflection properties, the moment-rotation characteristics will influence the deflection under inertial forces of the seat-back, and hence the angle and time at which the head strike occurs, as well as the subsequent head acceleration-time history.

The standard breakover moment used in the baseline model and measured prior to dynamic testing was 40ft-lb. For comparative purposes, simulation runs using 35 and 45ft-lb were performed, together with a 'restricted breakover' seat-back similar to those used adjacent to emergency exits. The results are presented in Table 7.2. For these runs, it was assumed that the seat back would not break at its mountings due to the loading imposed by the head impact.

7.4.2 35ft-lb Breakover Moment

Direct comparison of the results from a simulation using a 35ft-lb seat-back breakover moment with the standard 40ft-lb moment shows that the head impact took place much later, 146ms compared to 118ms. Body segment accelerations are shown in Figure 7.21, with those from the baseline simulation in Figure 7.22. The acceleration level prior to impact and the head velocity on impact were both lower, the seat back had rotated by a further 3 degrees and had stopped. The angle of incidence was basically unchanged, but the point of contact on the seat was lower, requiring a higher breakover force due to the reduced lever arm. This offset the reduced breakover moment, and the result was higher severity indices. Due to the complex nature of the occupant dynamics it was not possible to completely separate all variables. For example, hand impact with the food tray (on the seat-back) occurred prior to head impact for the case of the lower breakover moment, but not in the baseline model. Also, the dynamics of the seat back at the time of head impact were absent.

Reducing the seat-back breakover moment would appear from the simulations to create similar impact conditions to those indicated when increasing seat pitch to 32". The

head contact again took place later, with the seat-back stationary after breaking forward under its own inertia and having stopped by the time secondary impact occurred. The head velocity on impact was also similar at 30.2ft/sec. The breakover moment was lower, hence one might have expected secondary head impact to be less severe, with a lower peak 'g'. However, this was not the case, because the head impacted the seat back at a different point, and with the seat back at a different position, as explained previously.

The occupant kinematics were similar for the two cases, 32" pitch with 40ft-lb breakover and the 30" pitch with 35ft-lb breakover, up to the point of secondary head impact. The only differences up to this point were as a result of the slightly different times at which shin and hand impacts took place. The main difference between these two cases would appear to be the angle of incidence between the head and the seat back. The higher seat pitch case results in a higher angle of incidence, less head-on force and more vertical. However, as explained in the seat pitch study, the different geometry produced forces of a higher magnitude due to the much lower contact point.

7.4.3 45ft-lb Breakover Moment

Occupant body segment acceleration levels are shown in Figure 7.23. Increasing the breakover moment produced a case with secondary impact conditions more readily comparable to the baseline model (with a 40ft-lb breakover moment). These included 37ft/sec impact velocity, with impact times only differing by 3.5ms, and impacting a moving seat-back. Coincidentally, hand and head secondary impacts occurred simultaneously at 114.5ms, but any effects of this were indeterminate. The lower angle of breakover at impact would result in a point of contact higher on the seat back, offsetting to an extent the higher breakover moment.

By comparing the severity indices and peak head deceleration levels, one might well conclude that the higher breakover moment had contributed to the increases in severity. However, separating the variables would involve many more simulation runs. The worth of such a detailed study is doubtful when one considers the fact that occupants are likely to be of dimensions spread across the anthropometric range, therefore suggesting optimum levels for such parameters would not be advisable.

However, an overview of the situation can be obtained from the relatively small number of simulation runs used in a parametric study such as this.

7.4.4 Restricted Breakover Moment

Certain seat places on Transport Aircraft have 'fixed' position seat backs, due to proximity to emergency exits. Such a seat-back was simulated for comparative purposes, with a 60ft-lb breakover facility for only 15 degrees of forward rotation. The properties used were those measured from an actual seat at CAMI, and thought to be typical.

At impact, the velocity of the head was comparable with the baseline model, with secondary impact some 6ms earlier. However, the seat back had been thrown forward under its own inertia, had reached the limit of its travel, and was elastically unloading at the time of impact. Elastic movement of the seat back and the force-deflection characteristics of the contact surfaces would influence the severity of the head strike, with no available breakover rotation.

The results indicated a potentially non-survivable secondary head impact, with both HIC and HSI in excess of 1000. The low duration, high deceleration 'spike' produced when an extremely small distance over which to retard the occupant (or in this case head and upper torso) is available, is clearly illustrated in Figure 7.24. This obviously suggests that the load limiting effect of seat-back breakover, in this type of impact, could make the difference between survival and non-survival.

N.B. This simulation was repeated with a breakover moment of 40ft-lb and no significant differences in the results were obtained.

7.4.5 Seat-Back Stiffness

Simulations were performed to assess the effect of the force-deflection characteristics of the seat-back surfaces. These typically consist of a central food tray, normally a plastic coated foam 'sandwich panel', with a cloth-covered head support structure above this, and a cushion support structure in the lower part of the seat back. Various characteristics had been obtained from test data, data from program user manuals and other literature.

Three different values were used, a typical food tray, a firm seat-back and a soft seat-back. Whilst the peak g levels obtained varied by up to 18%, the maximum and minimum observed HIC only differed by only 3% and head severity indices by 2%. This implies that where a seat breakover facility exists, the influence of the seat-back surface force-deflection properties is minimal, within the range studied here.

7.5 PARAMETRIC STUDY - THE EFFECT OF ADOPTING THE 'CRASH POSITION'

7.5.1 Introduction

It is common practice amongst operators to have a policy of advising passengers to adopt a 'braced' or 'crash' position in a potential impact situation. This position varies from one airline to another, but generally speaking all entail forward rotation of the upper torso and head.

For the purposes of dynamic test and computer simulation work, it is the usual practice to position surrogates in normal upright seated positions, as this is assumed to be the worse case. It is certainly reasonable to assume that if the crew is able to impart the relevant instructions to the passengers, and the correct position is adopted, then under certain circumstances, survivability is increased.

Simulation runs were conducted to compare normal and braced initial positions in two types of impact, and also to investigate any differences caused by the proximity of a forward seat-back.

7.5.2 Horizontal Simulations

The problems discussed elsewhere in this text, relating to limits on body segment rotations involved with the use of program SOM-TA, again appeared in these simulation runs. It was therefore not possible to run simulations beyond 75ms, rendering them of no real value.

However, when the deceleration vector is horizontal, it is generally held to be beneficial to adopt a 'crash' position. If the head is braced against the forward seat-back then the head will be supported during the impact, whilst the breakover capability (if available) should provide a means of load-limiting. Secondary head impact with the seat back would therefore only be likely after the primary impact. In the case of a shorter occupant or a higher seat pitch, the head may only contact the seat back under impact loading, although one would expect lower velocity and severity than for an occupant who was initially upright.

During the course of testing, one attempt was made to test an adult dummy in a braced position, but the process of accelerating the sled to impact velocity caused the dummy to move back to an upright position. Resources were not available to pursue this study further as it was peripheral to the main area of investigation. However, it is generally held that adopting this position helps to offset a number of the shortcomings of the lap-belt method of restraint. Testing in this configuration has been conducted at CAMI and is reported by Chandler (1985 c).

7.5.3 Vertical Simulations

The second inertial loading condition considered was the combined vertical and horizontal deceleration pulse, such as that used in the baseline model validation based on Test 27. This simulation was repeated using a 2-dimensional occupant model in order to obtain spinal loading results. Two further simulations performed, both of these used an occupant in a 'crash' position, in one case with a 30" seat pitch, enabling the head to be braced against the seat-back, and in the other using a seat back positioned such that contact would not occur. For 50th percentile occupants, this would normally be the case where seat pitch exceeds 36", assuming a crash position is adopted and with inertial loading comparable to that used in this set of simulations.

Case 1 was a model of Test 27, but with a 2-dimensional occupant model and represented the 'control' case of an occupant subjected to a combined vertical and horizontal deceleration, with an upright initial position, Figure 7.25. For this situation, head contact with the seat in front did not occur.

Case 2 consisted of identical loading conditions to case 1 but with the occupant inclined forward and with the head braced against a seat-back positioned to give 30" seat-row pitch, Figure 7.26. Resultant pelvis peak acceleration was slightly higher than the peak input, with peak chest acceleration the same as peak input, and head acceleration lower. All acceleration levels were lower than those indicated for Case 1. The seat-back breakover during the impact and the subsequent 20ms amounted to 10 degrees. There was no seat-back rotation beyond 135ms. This clearly illustrated the load-limiting effect of the seat-back breakover.

Case 3 illustrated what may happen when occupant height or seat-row pitch dictate that whilst the crash position can be adopted, it is not possible to brace the head against a seat-back, Figure 7.27. Peak head and pelvis acceleration levels are significantly higher, and large increases indicated in spinal loads and bending moments, as might be expected. This trend could be accentuated during a more severe impact, such as the conditions modelled in Regulatory dynamic tests, such that secondary impact of the head occurs with the lower limbs (although this is conjecture).

Results of this study are given in Table 7.3. Whatever criteria are selected, it would be fair to list the three cases in order of increasing severity as follows: Case 2, Case 1, Case 3.

It would appear therefore, at least for the limited set of circumstances considered here, that the benefit of adopting the crash position during emergency landings involving significant vertical decelerations, may be limited to cases where it is possible for the occupant to brace his or her head against a forward seat back. This would normally be the case for the majority of passengers.

The reason for this is that the upright passenger bears the majority of the vertical loading as compressive forces in the spine, whilst the occupant in a crash position with no direct head support suffers relatively high bending moments in the neck and spine.

7.6 SUMMARY

(1) Simulation of a rigid triple-occupant seat was performed based upon Test 24 (longitudinal impact), with lap-belted occupants. Seat-back breakover and head impact were modelled successfully.

(2) Correlation between the results from the test and those from the simulation was obtained, including the severity of the head impact event.

(3) This 'baseline' model was used with an deceleration pulse based upon Test 27 (combined vertical and horizontal impact) and also correlated well with test results.

(4) The characteristic response of lap-belted occupants to impact pulses was established. For longitudinal impacts, a relatively high duration, low magnitude response is compounded by a short period of high deceleration, as the head strikes the seat-back in front, which breaks forward under inertial loading.

(5) The effects of seat spacing pitch have been studied, and compared with the baseline model (30" pitch). The effects of closer spacing included earlier head impact, increased head-on component of acceleration, and increased head velocity at impact. Impact took place higher up the moving seat-back, which therefore offered a lower resisting force. The net effect of this combination was detrimental, for the example considered.

(6) The effect of increasing seat spacing pitch was to delay head impact considerably, which then occurred at lower velocity with a stationary seat-back. The contact point was lower, producing a higher resisting force, and an angle of incidence changed to produce a more 'glancing' impact.

(7) The effect of the seat-back breakover moment was studied. Low values of this parameter resulted in more rapid rotation of the seat-back, and hence greater rotational displacement at the time of head impact. The head impact occurred later and at lower velocity. However, the lower point of contact and the change in the contact angle produced higher contact forces in the example considered.

(8) The effect of increasing the breakover moment was to produce an earlier impact at higher forces and higher velocity.

(9) The effect of a seat-back with breakover restricted to 15 degrees was to produce potentially fatal injury to the head. The seat-back reached its limit of rotation under inertial loading and the impact severity increased significantly.

(10) Although affecting peak head acceleration levels, seat-back contact stiffness did not significantly affect occupant HIC or HSI.

(11) The design of the seat-back and its position in the head strike envelope were found to be crucial to occupant injury. In certain cases, sufficient kinetic energy was absorbed to produce lower impact severity than would result from head impact with the lower limbs. In other cases, potentially fatal head impacts were generated.

(12) Attempts were made to simulate the effect of adopting a braced 'crash position' prior to impact. These were only successful for simulations of combined vertical and longitudinal decelerations. The results indicated differences in the carrying of spinal loads. Whilst the spine of the upright occupant was loaded primarily in compression, the braced occupant was subjected to higher spinal bending moments, especially where no seat-back was available against which to brace.

CONCLUSIONS

(1) A dynamic seat test facility, for research purposes, has been developed at CIT for the evaluation of civil aircraft seats. A series of tests on both metal and composite seat structures was conducted using an anthropomorphic test dummy.

(2) Many problems in seat performance under impact loading were found to be as a result of the floor-track fitting design. Lack of a pitch release on front floor-track fittings can result in failure to accommodate seat structure deformation and cause premature seat or floor-track failure. Lack of a roll release on floor-track fittings can cause asymmetric loading of the floor-track and prying of the lips, resulting in premature failure in warped floor conditions. Single stud floor-track fittings can release easily following minor seat deformation.

(3) The use of twin stud fittings, such as those commonly used at the rear leg location, providing positive location, clamping, and anti-rattle, with the addition of spherical bearings to provide pitch and roll releases, would solve the problems associated with front floor-track fittings in the majority of applications.

(4) Warped floor conditions can be more easily accommodated in seat structures using a large number of components, because much of the distortion can be taken up in the connections and fasteners. Unyielding seat structures which nevertheless allow large elastic deformations during dynamic loading may result in increased severity of occupant injury.

(5) The use of single tubular spars may result in undesirable loading of the femurs, where vertical components of acceleration are present, and where the seat-pan used is inadequate for distribution of loads.

(6) The use of thick cushions (for comfort) may lead to secondary impact injuries in the upper legs and lower torso, as the occupant is allowed to build up (downwards) velocity relative to the seat structure.

(7) Post-impact structural deformation (for a given design of seat) may show similarities between primarily vertical and primarily longitudinal inertial loadings. This may increase the difficulty in making post-accident estimates of inertial forces, based on the nature of seat deformation.

(8) It was indicated from computer modelling of the seat-restraint-occupant system (SROS) that reduced severity of injury (including body segment accelerations and severity indices) was achievable in longitudinal impacts by use of an energy absorbing seat, even with lap-belt restraint only.

(9) Where composite seats are utilised, brittle structural failure can be avoided by the use of load-limiting elements, the collapse characteristics of which should be optimized to minimise occupant injury severity, having regard to the available and acceptable space for seat displacement, and the severity of impact loading in potentially survivable accidents.

(10) Comparative studies were performed to assess the suitability of carbon-fibre and kevlar reinforced plastics, for use in tubular structures which collapse axially to absorb impact energy. Kevlar was identified as a preferable material for civil aircraft seats.

(11) A suitable compression-type energy absorber, working on the principle of virtual work, was developed using kevlar-based composite materials, which collapse by fibre buckling. This uses a unique internal stabilizing mechanism, which has been patented.

(12) Incorporation of such devices in prototype composite seat structures, to produce controlled collapse under impact loading, was found to reduce seat structural damage and occupant injury severity levels.

(13) Advanced computer modelling techniques were used successfully to recreate dynamic seat tests and to study the response of lap-belted occupants to impact loading. Occupant response to longitudinal deceleration was found to be characterised by (relatively) high duration low level head acceleration, compounded by a high level, short duration period during secondary head impact with the seat-back in front (which breaks forward under its own inertia).

(14) The design of the seat-back and its position in the occupant head-strike envelope would seem to have a significant effect upon occupant injury potential. The effects of parameters such as seat spacing pitch and seat-back breakover moment have been studied. In certain cases the effect was to absorb significant amounts of kinetic energy of the head, and produce lower severity than head impact with the lower limbs. In other cases, potentially fatal decelerations were generated.

(15) Attempts were made to simulate the effect of the braced 'crash position'. The effects, where significant vertical velocity components were present, indicated differences in the generation of spinal loads. The spine of the upright occupant was loaded primarily in compression, whereas the braced occupant suffered high spinal bending moments, if no seat-back was available for the purposes of adopting a braced position.

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JOINT AVIATION REQUIREMENTS

The paragraphs shown below describe the dynamic test conditions to satisfy JAR 25.562. Identical requirements are contained within Federal Aviation Regulations (FAR) Part 25.562.

JAR 25.562 Emergency landing dynamic conditions

(a) The seat and restraint system in the aeroplane must be designed as prescribed in this paragraph to protect each occupant during an emergency landing condition when —

- (1) Proper use is made of seats, safety belts, and shoulder harnesses provided for in the design; and
- (2) The occupant is exposed to loads resulting from the conditions prescribed in this paragraph.

(b) Each seat type design approved for passenger occupancy must successfully complete dynamic tests or be demonstrated by rational analysis based on dynamic tests of a similar type seat, in accordance with each of the following emergency landing conditions. The

Tests must be conducted with an occupant simulated by a 170-pound (77.11 kg) anthropomorphic, test dummy sitting in the normal upright position.

(1) A change in downward vertical velocity, (Δv) of not less than 35 feet per second (10.67 m/s), with the aeroplane's longitudinal axis canted downward 30 degrees with respect to the horizontal plane and with the wings level. Peak floor deceleration must occur in not more than 0.08 seconds after impact and must reach a minimum of 14 g.

(2) A change in forward longitudinal velocity (Δv) of not less than 44 feet per second (13.41 m/s), with the aeroplane's longitudinal axis horizontal and yawed 10 degrees either right or left, whichever would cause the greatest likelihood of the upper torso restraint system (where installed) moving off the occupant's shoulder, and with the wings level. Peak floor deceleration must occur in not more than 0.09 seconds after impact and must reach a minimum of 16 g. Where floor rails or floor fittings are used to attach the seating devices to the test fixture, the rails or fittings must be misaligned with respect to the adjacent set of rails or fittings by at least 10 degrees vertically (i.e. out of parallel) with one rolled 10 degrees.

(c) The following performance measures must not be exceeded during the dynamic tests conducted in accordance with sub-paragraph (b) of this paragraph:

(1) Where upper torso straps are used tension loads in individual straps must not exceed 1750 pounds (793.78 kg). If dual straps are used for restraining the upper torso, the total strap tension loads must not exceed 2000 pounds (907.18 kg).

(2) The maximum compressive load measured between the pelvis and the lumbar column of the anthropomorphic dummy must not exceed 1500 pounds (680.38 kg).

(3) The upper torso restraint straps (where installed) must remain on the occupant's shoulder during the impact.

(4) The lap safety belt must remain on the occupant's pelvis during the impact.

(5) Each occupant must be protected from serious head injury under the conditions prescribed in sub-paragraph (b) of this paragraph. Where head contact with seats or other structure can occur, protection must be provided so that the head impact does not exceed a Head Injury Criterion (HIC) of 1000 units. The level of HIC is defined by the equation —

$$HIC = \left\{ (t_2 - t_1) \left[\frac{1}{(t_2 - t_1)} \int_{t_1}^{t_2} a(t) dt \right]^{2.5} \right\}_{\max}$$

Where —

t_1 is the initial integration time,

t_2 is the final integration time, and

$a(t)$ is the total acceleration vs. time curve for the head strike, and where

(t) is in seconds, and (a) is in units of gravity (g).

(6) Where leg injuries may result from contact with seats or other structure, protection must be provided to prevent axially compressive loads exceeding 2250 pounds (1020.58 kg) in each femur.

(7) The seat must remain attached at all points of attachment, although the structure may have yield.

(8) Seats must not yield under the tests specified in sub-paragraphs (b)(1) and (b)(2) of this paragraph to the extent they would impede rapid evacuation of the aeroplane occupants.]

APPENDIX B

A COMPARATIVE STUDY OF THE EFFECTS OF DIFFERENT ACCELERATION-TIME PULSES ON OCCUPANT KINEMATICS USING COMPUTER SIMULATION

INTRODUCTION

When simulating an aircraft crash scenario, the acceleration-time pulse associated with the impact, measured at the aircraft floor, is used as the input pulse to the seat/restraint/occupant system. In a physical impact simulation, such as a dynamic test, this is achieved by orientating the seat on a sled in such a way that the direction of the deceleration vector is correct, and then applying a suitable deceleration pulse to the sled. The area under this acceleration-time curve represents the overall change in velocity associated with the impact, whilst the distance covered during the impact is the area under the velocity-time curve.

Therefore for a given change in velocity over a given time period, the severity of the impact will be influenced by the shape of the acceleration-time pulse. In reality, the pulse is influenced by the structural crashworthiness of the airframe, the topography and physical properties of the impact medium (sea, runway, hillside), and the aircraft attitude and velocity on impact. It will also vary with the position of the seat in the aircraft cabin, proximity of bulkheads, longitudinal position, and so forth. The velocity change associated with the impact pulse in a survivable accident would normally include much of the aircraft vertical velocity (sink speed) if applicable, but only a portion of the forward velocity. The remainder of the velocity change would be related to deceleration at lower levels from skidding, braking and so on. In other words, not all of the aircraft's kinetic energy is necessarily absorbed during the main impact. This subject is discussed in more detail in Appendix C.

It is generally held to be the case that a rectangular pulse is an 'ideal' shape, in as much as the deceleration is kept down to the lowest possible level for a given duration and velocity. The effect of this and other shaped pulses on occupant kinematics and resultant body segment accelerations is of particular interest, as the pulse to which the occupant is subjected can be influenced by the crashworthy design of the vehicle and the seat structure. Thus if an optimum pulse shape can be defined for a fixed velocity change and duration, then the design can be such that the acceleration levels experienced by the occupant are minimised, and the chances of survival, or avoiding serious injury, can then be maximised.

It is also the case that many different methods of producing impact pulses for dynamic tests are used, including hydraulics, tube crushing, wire bending, and a number of other methods. The shape of the pulse produced will vary, and it is useful to know the influence of this parameter for comparative purposes.

Therefore a parametric investigation into the effects of ten different (hypothetical) acceleration-time histories was conducted using the computer simulation program SOM-LA.

SIMULATION PARAMETERS

The effects of ten different pulses on occupant kinematics were investigated, with a view to comparing several parameters related to the relative severity of each pulse. Head, chest and pelvis resultant accelerations, Head Injury Criterion (HIC), cumulative Head Severity Index (HSI) and Chest Severity Index (CSI) were calculated, together with lap belt and shoulder strap restraint system loads, and the forces interacting between occupant and seat cushions, for a total time period of 250ms.

Each pulse was of 90ms duration and of mean deceleration 15g, giving a reduction in forward velocity of 43.5ft/s (13.24m/s). The occupant simulated throughout was a 50th-percentile (part 572) anthropomorphic dummy to enable experimental validation of the results obtained if required. The seat structure used in the simulation was of rigid construction and included a head-rest, with foam cushions three inches thick, with typical stiffness characteristics.

The restraint system used throughout consisted of a diagonal right shoulder belt with lap belt, such that the shoulder belt and one side of the lap belt were one continuous length of webbing, in automotive style. Although lap belts are the normal method of occupant restraint used in Transport Category aircraft, problems with excessive head rotations causing the program to crash had been encountered with SOM-LA simulations using lap belts. This would make useful comparisons difficult to make, and hence it was decided to use some form of upper torso restraint, such that all simulations could run for 250ms.

PULSES INVESTIGATED

Ten pulse shapes were investigated, and are shown in Table B1. All pulses were of the same duration and velocity change. Pulse 01 was a 'worst case', being triangular and with a peak acceleration of 30g, whilst Pulse 02 was what is commonly regarded as an ideal situation, a rectangular time-acceleration history (peak 15g). It is common in crash simulations to find an initial 'spike' in deceleration, where high 'g' is encountered for a relatively short duration. This type of behaviour is characteristic of structures which collapse at an approximately constant load, but which require a higher load to initiate failure. This situation was simulated in Pulses 03, 04, and 05, with peaks of either 22g or 30g. Pulses 06 and 07 represented accelerations gradually reducing and increasing respectively, with upper and lower limits of 10g and 20g. Pulse 08 was the reversed case of Pulse 04, whilst Pulses 09 and 10 were similar to 06 and 07, but with limits of 12g and 18g.

RESULTS OF SIMULATIONS

The stopping distances associated with the deceleration pulses, maximum values of body segment accelerations and severity indices are given in Table B1, together with the Head Injury Criteria Chest and Head Severity Indices.

DISCUSSION OF RESULTS

As previously stated, each pulse was of duration 90ms, with a reduction in velocity of 43.5ft/s and mean deceleration 15g. The principal effect of using this approach is that whilst the area under the acceleration-time graph remains constant for the different pulse shapes, the area under the velocity-time graph does not.

Because this area represents the distance over which the deceleration process occurs, then the stopping distance normally varies from pulse to pulse, unless the pulse is symmetrical. This situation makes realistic comparison of the severity of different pulse shapes more difficult; thus if a change in pulse shape results in a decrease in the severity from the occupant viewpoint, then the proportion of this reduction attributable to any increase in stopping distance must be identified and accounted for in any comparison. In comparing relative severity it is perhaps most convenient to take Pulse 02, the constant 15g deceleration, as a datum; the stopping distance was calculated as 23.7".

Pulse 01 was triangular with a peak of 30g, and the simulation produced the highest body segment accelerations of all the pulses. The stopping distance was 12% less than that taken as the datum, but HIC increased by 69%, chest severity index at 250ms by 177%, maximum pelvic acceleration by 56%, with other parameters showing similar increases; clearly only a small proportion of each increase is attributable to the shorter stopping distance.

Pulse 03 began with a 30g peak, then continued at 15g until 70ms, then reducing. With large deceleration levels early in the pulse, the stopping distance was reduced, hence Pulse 03 recorded the smallest displacement, approximately 17% less than the datum. HIC increased by 18% above that for the datum pulse, severity indices and maximum seat forces also rose in similar proportions, whilst shoulder belt forces increased by 10% and lap belt forces actually reduced. Maximum head, chest and pelvis accelerations rose only marginally, and perhaps it would be fair to say that a large proportion of any increase in severity compared with the datum pulse was due to the reduction in stopping distance. This was as a result of early occurrence of the 30g peak (its positioning rather than simply its presence).

Pulse 04 was (stepped) rectangular, decreasing from 22g to 13g at 20ms. Again the area under the acceleration-time pulse was biased to the left causing a reduced stopping distance (10% below datum). However, the changes in the various severity parameters from the datum case were extremely small, for example HIC increased by 6%, whilst the cumulative Head Severity Index remained unchanged, and chest severity index reduced by 1%.

Pulse 05 had a spiked peak at 5ms reducing to 15g at 10ms and remaining constant until 80ms, then gradually dropping to zero at 90ms. This case was similar, if less severe than, Pulse 03. In comparison to the rectangular datum pulse, the stopping distance was reduced by 9%, with minimal increases in other parameters; HIC rose by 4% with maximum body segment accelerations and severity indices increased by up to 7%.

Pulse 06 was 'wedged' in shape, starting at 20g and reducing to 10g at 90ms. An 8% reduction in stopping distance contributed to increases in the relevant severity parameters of approximately 13%.

Pulse 07 was simply Pulse 06 reversed, increasing from 10g to 20g. As one would expect, stopping distance was increased by 8%, and this pulse actually resulted in a decrease in severity in comparison to the rectangular pulse, with HIC lower by 8% and significant reductions in the maximum body segment accelerations calculated.

Pulse 08 was the reverse of Pulse 04, stepped rectangular, increasing from 13g to 22g at 70ms. As with Pulse 07, the area under the pulse was biased towards the latter part, resulting in an increase of 11% in stopping distance. Head Injury Criterion was reduced to 235, compared with that for the datum pulse of 332, a reduction of 29%, whilst the severity indices fell by at least 18%.

Pulse 09 was similar to 06, but closer to rectangular in shape. Whilst in the case of 06, the percentage increases in severity parameters were greater than the decrease in stopping distance, the simulation for Pulse 09 showed increases in severity of about 6%, with a reduction in stopping distance of 6%. Pulse 10 was a reversal of 09, rising from 12g at the start to 18g after 90ms, and the 6% increase in stopping distance contributed to a 14% reduction in HIC, with small decreases in other parameters.

SUMMARY

(1) The specification of deceleration envelopes for comparative work, such as dynamic testing, should take account of any asymmetry in the pulse and hence any differences in stopping distance.

(2) Deceleration signals of the same duration and velocity change (and hence mean deceleration level) will have different effects upon occupant kinematics. This can be partly due to asymmetry of the pulse, which affects stopping distance. It can also be due to the characteristic shape, which also affects severity.

APPENDIX C

FAA CRASH DYNAMICS PROGRAMME

INTRODUCTION

As part of an ongoing effort in studying aircraft crash dynamics, much useful data has been collected during the 1980s. This has included parametric studies of aircraft crash dynamics using program KRASH and test validation under the joint FAA/NASA programme of airframe section and full scale aircraft testing, including a 'Controlled Impact Demonstration' (CID).

Using the results obtained from the dynamic test work as a framework for on-going validation and application of KRASH, floor acceleration-time pulses have been obtained. This analytical and experimental data has been integrated and a crash design velocity envelope defined. An overview of the work performed was given by Wittlin, 1988.

This database of information can be applied to the design and dynamic testing of aircraft seats with a view to improving crashworthiness, and as input for occupant simulation programs.

Previous relevant work has been incorporated. This includes full-scale tests of a DC7 (Reed, 1965 a), L1649 Constellation (Reed 1965 b), various section tests and the 'Laurenburg' pre-CID full-scale drop test.

CONTROLLED IMPACT DEMONSTRATION (CID)

This full scale impact test of a narrow-bodied jet transport aircraft was conducted at Edwards Airforce Base in December 1984. Pre-test and post-test KRASH models were constructed, and various parametric studies performed. It was necessary to produce a post-test model because the attitude of the aircraft at impact was not as originally planned. Zero initial roll and pitch angles were envisaged, but actual test conditions were -13 degrees roll (left wing down), -13 degrees yaw (nose left). Other parameters were within test specifications, longitudinal velocity 151knts, sink rate 5.27m/s, glide path 3.5 degrees, 0 degrees pitch.

The relevant output test data took the form of impact acceleration-time histories in the vertical, longitudinal and lateral directions at various points (fuselage stations) along the length of the aircraft. Fuselage stations for the L1649 Constellation full-scale tests are shown together with those for the Controlled Impact

Demonstration in Figure C1. The range of peak accelerations recorded at the nose of the aircraft (approximately the first 10m) were significantly higher than in the main fuselage section, 14g approximately, compared to about 7g for vertical acceleration.

Longitudinal accelerations were generally lower, 6g peaks in the nose compared with about 3g in the main fuselage, with lateral acceleration levels lower still, and caused mainly as a result of unforeseen attitude at impact. KRASH analyses are consistent with these findings, and the results are given with the test results in Figure C2.

Typical vertical acceleration time histories at various fuselage sections are shown in Figure C3a. Close to the front of the aircraft, relatively high g levels of relatively short duration were obtained, with lower g and longer durations being more typical for the rest of the fuselage.

Longitudinal deceleration pulses obtained from testing the L1649 are shown in Figure C3b. During the 6 degree slope impact the fuselage remained intact, but during the 20 degree slope impact two fuselage breaks occurred. For each Fuselage Station, the impact at higher velocity generally resulted in a longer duration and a lower g level.

This information is included with those from other tests, for both vertical and longitudinal deceleration pulses, in Figures C4 and C5, showing lines of constant velocity change during the (triangular) impact pulses. Combining all the information from full-scale tests, section tests, and KRASH analysis work, velocity envelopes for Transport category aircraft were defined, Figure C6. Information from vertical, longitudinal and combined pulses was included in the formulation. The deceleration pulse envelope at the floor is higher than the airframe structural integrity envelope and also accounts for rotational and rebound effects. The floor pulses are those applicable to a survivable accident (defined by the limit of fuselage strength). Dynamic testing of seats beyond the airframe pulse envelope would only be applicable if the design strength of the floor and airframe was not exceeded.

Full details of the CID test with KRASH correlation are given by Wittlin (1986), and the post-CID parametric studies mentioned previously were reported by Wittlin (1987). A concise synopsis of the FAA Crash Dynamics effort has been written by the same author (1988).

CID SEAT PROGRAMME

As part of the Controlled Impact Demonstration, modified seats were to be installed alongside standard seats in order to demonstrate improvements possible in occupant crash protection, and the work was described in detail by Cannon (1985). It was also discussed as part of a cost benefit study on improved Transport seats.

The design goals were fourfold:

- (1) To limit loads in order that the seat structure and floor track did not become overstressed.
- (2) Add any necessary lateral bracing without ergonomic penalties.
- (3) Redesign floor fittings to accommodate floor deformation, to fit existing track.
- (4) Redesign seat structure to accommodate floor warping.

Six slightly different types of modified seat were used, all based on the criteria described above. Pre-CID static and dynamic testing were performed to ensure correct seat behaviour. Results from these tests indicated that the modifications design goals had been achieved.

The modified seats installed on the CID aircraft did not stroke. The main reason for this was that the impact pulse differed from that which was envisaged, as mentioned previously.

POST-CID RESEARCH

Further FAA Crash Dynamics research has included the effect of aircraft size on the floor crash pulse. This indicates that for larger aircraft, greater available underfloor crush distances result in lower deceleration levels, accompanied by higher velocity changes and longer durations. This work has been documented by Caifa (1988).

The effect of water and soil impacts is also currently under investigation, with results again as yet unpublished.

The post-CID transport seat work included a cost benefit study for improved, Cannon (1986). This contained a review of accident data, a description of an improved seat and an attempt to compare increased costs, as a result of increased production and development costs together with increased fuel costs, against the (measurable financial) benefits of the reduction in deaths and injuries.

The improved seat was based on the data generated from the pre-CID seat tests, and had extension type load-limiting rear legs, Figure C7. Reinforcement tubes at the seat spar to leg connection points and lateral strengthening are also shown, with improved floor track fittings.

APPENDIX D

SIMULATION PROGRAMS SOM-LA AND SOM-TA

INTRODUCTION

These programs are crash victim simulation (CVS) models, the former for light aircraft and the latter for transport category aircraft. This is in fact an over-simplification, since SOM-LA is a single occupant model, with the capability to model all commonly used restraint system configurations, and includes a Finite Element seat structure modelling capability. Therefore it is possible to model the dynamic interaction of seat, occupant and restraint under a variety of impact configurations. The program has been used for both rotary and fixed wing aircraft, Laananen (1983).

SOM-TA has an enhanced range of capabilities, modelling up to three occupants on the same seat structure, therefore double and triple seats can be modelled in addition to the single seats modelled by SOM-LA. Secondary impacts can be modelled. Such contact may include head impact with the back of the seat directly in front, useful where lap-only belts and shorter seat spacing pitches are used. Full details are given by Laananen (1986).

The two programs have much in common, and features of particular importance are described in the following sections.

OCCUPANT MODELS

Standard data for 50th percentile anthropomorphic dummy (Part 572) and 50th percentile human are included with the program, and either option can be selected. If a non-standard occupant is required, such as a 95th percentile male dummy or 50th percentile 3-year old child, then it would be necessary to provide suitable input data.

Two different types of occupant model are available. A 3-dimensional model consisting of 12 rigid body segments is shown in Figure D1. All joints are modelled as ball and socket type, except for the upper neck, elbow and knee which are modelled as hinge joints. Resistance of each joint is modelled by a non-linear spring and a viscous or friction torsional damper. The relative contribution of each will depend upon the user's selection of either a human occupant or anthropomorphic dummy. Viscous damping is the primary means of controlling joint motion when

modelling human body motion, with frictional torque dominant in dummy motion. The degree of joint rotation is limited to match dummy or human characteristics.

Body segments have length, mass, centre of mass location and moments of inertia based directly on anthropometric data or on those of specific types of dummy. Compliance characteristics are input for chest and abdomen.

Body contact surfaces are also modelled, for interaction of the occupant with seat cushions, restraint systems and surfaces contacted during secondary impact. These are ellipsoids, cylinders and spheres, Figure D3.

A 2-dimensional occupant model can be used where lateral components of deceleration are absent. The eleven segment model includes beam elements for the spine and the neck, and thus requires axial and flexural stiffnesses and damping coefficients to be input for non-standard occupants. The centres of mass may be offset from the spinal elements, thus producing the moments in the vertebral column which are produced by exposure to vertical components of deceleration common in aviation impacts. The basic 2-dimensional occupant is shown in Figure D2.

SEAT AND RESTRAINT SYSTEMS

Various seat-belt configurations are available, from lap belt only to full restraint with shoulder belts and negative-g strap. Belt anchorages are specified by the user, and belt stiffnesses are modelled using a trilinear force-deflection relationship.

Three basic types of seat model are available within the programs. The simplest model is a RIGID SEAT. This models the seat pan and seat back fixed in position and constrained not to deform or displace, thus allowing the effect of parameters other than those affected by seat structure behaviour to be studied, such as changes to the restraint system.

Also available is an ENERGY-ABSORBING SEAT, with one translational and one rotational degree of freedom. This was included to allow modelling of helicopter type stroking seats, where a vertical (or almost vertical) motion of the seat pan is controlled by energy absorbing elements, designed to limit the occupant deceleration levels to the limits of human tolerance.

The linear force-deflection characteristics of the energy absorbing device are defined by a three-line approximation. One degree of freedom is stroking of the seat pan in a predetermined linear direction, such as 5

degrees from vertical, at predetermined force levels. The second degree of freedom is elastic rotation of the support structure about a lateral axis positioned on the aircraft floor. This was included because dynamic tests indicated that the effects of such deformation could be significant.

NOTE: The rotational degree of freedom can be included in the rigid seat model if required.

The third option available is a FINITE ELEMENT SEAT, integrated with the occupant dynamic behaviour. This is loosely based on the WRECKER II (ITT RESEARCH) program, with capacity for up to 75 nodes and offers beam, plate and spring elements. Furthermore it can model large displacements, non-linear material behaviour and local buckling. Internal releases can be used and include shear, moment, thrust and torque releases. The program's ability to model large displacements is based on a formulation to separate the element displacement field into a rigid body rotation and translation (associated with a local co-ordinate system moving with the element) and small element distortions (relative to the current position of the element co-ordinate system).

Material non-linearities are based on uniaxial elastic-plastic stress-strain law (beam and spring elements), and biaxial elastic-plastic stress-strain law (plate elements). A simple local buckling model for thin walled tubes is also included.

Spring elements are defined by one elastic and two plastic spring stiffnesses, which in theory should enable energy absorbing elements to be modelled. This aspect is discussed in detail in Appendix E.

OUTPUT

This is produced in several forms. All output data is related to the simulated time. For example, a typical run might start at time=0ms, with a deceleration-time history lasting 100ms, and the simulation running to 200ms to allow the full extent of occupant motion to be studied.

Plots of the occupant position at up to 8 different points in time are produced, such as 0ms, 30ms, 60ms, and so forth. Typical plots are shown in Figures 3.5 and 3.6. SOM-LA output files plot a single occupant, while SOM-TA output gives plots with up to three occupants, depending on the type of seat (single, double or triple) which is being modelled. Plots of deformed seat structure are also produced if a Finite Element seat model has been included, an example of which is shown in Figure 3.3.

The remainder of the data is output in tabulated or graphical form. FE output takes the form of nodal displacements, element stresses and forces, and floor reaction loads in tabulated form.

Graphical output includes acceleration-time histories for various body segments (head, chest and pelvis) in component and in resultant form. Velocity and displacement time histories for the body segments are also output. Spinal compressive and tensile loads and bending moments are plotted against time when a 2-dimensional occupant has been selected. External forces on the occupant, such as restraint system forces and forces between the seat and the occupant are plotted against time. Head Severity Index and Chest Severity Index are calculated, and Head Injury Criterion can also be obtained where secondary head impact has occurred.

Integration of the acceleration-time history of the aircraft floor (as input by the user) yields velocity and displacement plots.

ADDITIONAL FEATURES IN SOM-TA

Under FAA sponsorship, program SOM-TA was developed to enable sophisticated modelling of multiple occupant, transport category aircraft seating. This can then be used as a design aid for seat manufacturers aiming to comply with dynamic seat testing requirements introduced recently. It has also been used as a research tool in the project described in this thesis.

This program enhancement aims to give a wider range of abilities, namely modelling of floor warpage, multiple occupants and modelling of secondary impact injury. The inclusion of 'floor warpage' requirement in the dynamic seat regulations has prompted the inclusion of an equivalent process in the simulation program. This allows for translational and rotational displacements to be imposed upon the seat structure (FE seat only), hence inducing pre-stressing of the structure before the dynamic simulation begins.

The multiple occupant option is of course important for the accurate modelling of transport category seat structures. For a given design, parametric studies can be carried out and design changes evaluated without the necessity of performing dynamic tests at every stage. It is also possible to identify the worst loading case for a given seat, considering such factors as asymmetry, lateral loads, combinations of occupied and vacant seats, and so on.

It is also possible, using SOM-TA, to model the impact of occupant body segments with the seat structure positioned immediately in front of the occupant being simulated. With the use of the lap-only belt, this can include head impacts with seat back, food tray or armrests, and subsequent break-forward of the seat back. Force-deflection characteristics can be input for each of these contact surfaces, together with the rotational break-forward characteristics of the seat back.

One particular setback in SOM-TA is the removal of spring elements from the FE part of the program. The inclusion of 'Energy Absorber' elements in any future issue of the program would be particularly useful.

APPENDIX E

INVESTIGATION INTO THE BEHAVIOUR OF SPRING ELEMENTS IN SIMULATION PROGRAM SOM-LA

INTRODUCTION

During the simulation work described in Chapter 3, it became clear that a problem existed with the spring elements being used to simulate energy absorbing legs in the seat structure. This was indicated by low values of front leg floor reaction loads calculated by the program during plastic collapse, and by the large amounts of stroke used in the energy absorbing process. It was decided to investigate the problem by performing a number of simulations with identical models except for the stiffness characteristics of the front spring elements, and then to compare the behaviour.

DESCRIPTION OF SPRING ELEMENTS

Each spring element in SOM-LA is a special compressive load carrying beam, described as a force-displacement relationship by input of three loads (first yield force, second yield force and ultimate force) and three stiffnesses, (one elastic and two plastic); these are referred to as F_1 , F_2 , F_3 and E_1 , E_2 , E_3 respectively, see Figure E1. A zero force/zero displacement origin is assumed. This form of modelling thus allows the load-deflection characteristics of an energy absorbing element to be expressed using a three line approximation.

The first line represents the elastic spring stiffness, and hence defines loading and unloading slopes, to the elastic limit. The second and third lines represent the plastic collapse of the element. Ideally, the second line would be used to model the 'constant load' collapse region and the third line to model the 'bottoming out' region. This would enable optimisation of the energy absorbers (for a given seat geometry) to minimise occupant severity parameters within the available limits of stroke.

The nature of the problem encountered appeared to be related to an inexplicable reduction in the load-carrying capability of the element during plastic collapse, and it was decided that the initial method of investigation would be to vary the first plastic modulus, and observe the behaviour of the spring elements. The force and displacement data would then be obtained at each time-step, and compared to the input data.

SIMULATION PARAMETERS

In order to minimise the computing costs, a simple Finite Element model similar to that described in Stage (A) of Chapter 3 was used for this series of simulations. The first spring simulation (SS 01) used a first plastic modulus of 10lb/in (previous work had indicated that the program would not accept zero or negative values of first plastic modulus, and checking of the Fortran code had confirmed this). The method of investigation was to allow the program to run to 200ms, and then to calculate the force and stroke of the spring elements at 5ms intervals. The force was calculated by vectorial addition of the floor reaction components, as the forces in the elements could not be output directly. The stroke of the spring elements was calculated by monitoring the nodal displacement of the tops of the spring elements and calculating the current length of the elements, then simply deducting that from the initial length. Thus a time-independant graph of load versus deflection could be constructed for each simulation performed.

RESULTS

In the series of simulations, the first plastic modulus was varied, and values of 10, 20, 30, 50, 100 and 200lb/in were used. The results of each simulation are given in the form of force-deflection graphs obtained by the method outlined above, comparing the idealised input to the actual behaviour in the simulation.

The initial simulation (SS 01), with a first plastic modulus of 10lb/in, failed after only 97ms with ill-conditioning of the stiffness matrix. The unloading and reloading of the spring elements at a displacement of 1" is shown in Figure E2. In total, 2.5" of plastic collapse of the elements had taken place, and the (output) load levelled off at approximately 500lb, whilst the input was some 20% higher.

Simulation SS 02 utilised a second plastic modulus of 20lb/in, and the results are shown in Figure E3. Whilst the element had shown a high degree of plastic collapse (4.6"), the load carrying capability of the elements had dropped to under 40% of the required (input) value.

The next simulation (SS 03) had much higher plastic moduli, to check if this problem only applied to low values of this parameter. The results showed an improvement in correlation with the input values of 200lb/in and 500lb/in (first and second plastic moduli respectively) in comparison to previous simulations (Figure E4). However, a problem which seemed to occur was

in transition between first and second plastic moduli, although the energy loss associated with this would be relatively small.

Simulation (SS 04) used a reduced first plastic modulus of 100lb/in. Beyond 2" of collapse, the behaviour of the element during the simulation indicated wide variation about the input value, although the general trend was correct, Figure E5.

A reduction in the first plastic modulus to 50lb/in resulted in an improved correlation between the input spring element data (SS 05) and the simulated impact behaviour. As the stroke increased beyond 3" the degree of correlation reduced, although the mean load was close to the idealised input, Figure E6, but with significant deviations above and below the mean value.

A further reduction in first plastic modulus to 30lb/in (SS 06) resulted in a steady drop in the load-carrying capability of the elements, and an input of +30lb/in resulted in an output of approximately -20lb/in, Figure E7. For this case, comparatively little variation in load about a mean value occurred.

The results indicated that it might be possible to create an element which collapses at a constant load, by setting an artificially high plastic modulus (say 40lb/in) as input. However this does not really address the problem adequately, and would at best be usable only when rigorous checking of the output data was performed.

Furthermore, it would be useful if the initial trigger load could be set to a different value than that defined by the intersection of the elastic stiffness and the first plastic stiffness lines in the simulations. This would require a new type of 'energy absorber' element to replace the spring element. This would then enable accurate prediction of the required trigger load and constant collapse characteristics, taking full account of available stroking distance and the levels of occupant severity parameters.

One solution to the problems with spring elements would be to define a more versatile element, within SOM-LA, with distinct trigger load and collapse loads. Such an 'energy absorber' element would allow crashworthy seats to be developed effectively, using the program as an effective design aid.

SUMMARY

- (1) A problem was found in the use of spring elements in the simulation program SOM-LA. This related to the modelling of the plastic collapse region.
- (2) The program will only run with compressive spring elements if positive non-zero values of plastic moduli are selected.
- (3) Values of 30lb/in (or lower) for plastic spring rate result in a reduction of the load carrying capability of the elements. It is highly probable that the validation procedures did not include spring elements with such values as input.
- (4) A value of 40 to 45lb/in (positive slope) may, under certain circumstances, result in an approximately constant load collapse for these elements. This may provide a temporary solution.
- (5) At values of over 20lb/in, as used in the simulations in Chapter 3, the reduction in load-carrying capability was such that the predicted stroking distance was much greater than that which would be expected.

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Page 128 to 136

APPENDIX F

PATENT DOCUMENTATION

This Appendix contains copies of documentation relating to the Patent granted to the Author of this thesis, relating to the Composite Energy Absorber designed and developed during this study.

INVENTION REPORT No 5860

Security: <u>UNCLASSIFIED</u>	Establishment <u>RAE(F)</u> Section <u>MS6</u>
	Section Head <u>Mr R T Potter</u>
This copy to: _____	Division Head <u>Dr A C Ham</u> (Please initial and return to Pats 1)
	1A ☐ 1B ☐ CR ☐ X ☐
Inventors: <u>MR V M STEPHENS</u>	D/PDPats/ P0656

Details of the invention described below were received by Pats 1 on date 20 October 1988

This report is copied to give you notice of Pats 1B preliminary assessment and intentions.

Division Heads will be consulted before clearance of any DTE Information Sheet and again before any formal option is offered to DTE.

TITLE OF INVENTION:
ENERGY ABSORBING DEVICE FOR AIRCRAFT PASSENGER SEATING

Ownership assessment:

☐ Made in normal duty by MOD staff ☒ Made under contract no <u>2028/154</u>	☒ 100% MOD owned ☐ DTI funded ☐ Not 100% MOD owned (give details overleaf)
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Contractor CRANFIELD INSTITUTE OF TECHNOLOGY

Pats 1 proposed action: (a) Recording or patenting; (b) Exploitation action at least 3 weeks after date below:

☐ Record ☐ Record and await development ☒ File patent application	☐ Disclose to DTE - allow DTE database entry and dissemination of abstract ☐ Disclose to Pats 6 for direct licensing or advice ☐ No disclosure until further developed - Not proven or premature for exploitation
---	--

Keywords: CRASHWORTHY, SEATING, ENERGY-ABSORBING Subject Codes: A43, J69

Descriptive abstract:

The energy absorbing device is designed to sustain a compressive load over a high degree of deformation or axial collapse and comprises a composite tube with a special end fitting.

The thin-walled tube is made of a composite fibrous material which absorbs compressive energy by deformation of the fibres. The material of choice is aramid fibre such as Kevlar, although high performance polyethylene has been proposed as an alternative. So that subsequent tensile loads can be borne, it is important that the fibres should buckle rather than fracture.

A stabilising end fitting helps to control the collapse of the tube. This fitting comprises a flange, a stem portion and a slider which is a close fit inside the tube. The initial collapse of the tube is designed to occur near the flange and then the tube collapses concertina-fashion with the buckled portion constrained between the flange and the slider.

Abstract written by D Watkins *D. Watkins* on date 21 October 1988

Please address any comments to Pats 1B, Main Office, RAE Pyestock

United States Patent [19]

Sadeghi et al.

[11] Patent Number: **5,035,307**[45] Date of Patent: **Jul. 30, 1991**[54] **ENERGY ABSORBING DEVICE**[75] Inventors: **Majid M. Sadeghi, Newport Pagnell;
Vivian M. Stephens, Neaton, both of
Great Britain**[73] Assignee: **The Secretary of State for Defence in
Her Britannic Majesty's Government
of the United Kingdom of Great
Britain and Northern Ireland,
London, England**[21] Appl. No.: **421,593**[22] Filed: **Oct. 16, 1989**[30] **Foreign Application Priority Data**

Feb. 22, 1989 [GB] United Kingdom 8903994

[51] Int. Cl.⁵ **F16F 7/52**[52] U.S. Cl. **188/372; 74/592;
188/371; 280/777; 280/784**[58] Field of Search **188/371, 373, 376, 377;
74/592; 293/133; 297/473; 280/727, 784, 805**[56] **References Cited****U.S. PATENT DOCUMENTS**

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Primary Examiner—George E. A. Halvosa
Attorney, Agent, or Firm—Nixon & Vanderhye

[57] **ABSTRACT**

An energy absorbing device, suitable for use in the leg structure of aircraft passenger seating for the purpose of attenuating crash forces, having first and second attachment fittings 11, 21 with a tube 10 of fiber reinforced plastics material extending between them. Tube 10 has a crush initiation band 17 at which buckling of the tube commences upon compressive overload. Long, tough fibers are used as the reinforcing material in the tube so that, when compression of the device has concluded, there is a residual tensile capability. Successful operation of the device is facilitated by providing attachment fitting 12 with a slider 15 which helps to maintain the collapsing portion of the tube in axial alignment with the uncollapsed portion. The slider also acts as a tube wall straightener to unbuckle the collapsed fibers when tensile load is applied.

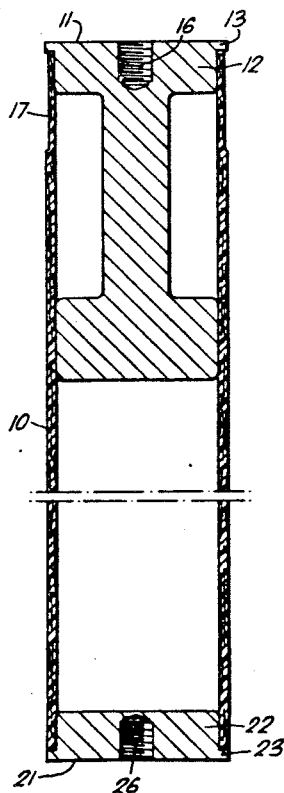
7 Claims, 3 Drawing Sheets

Fig. 1.

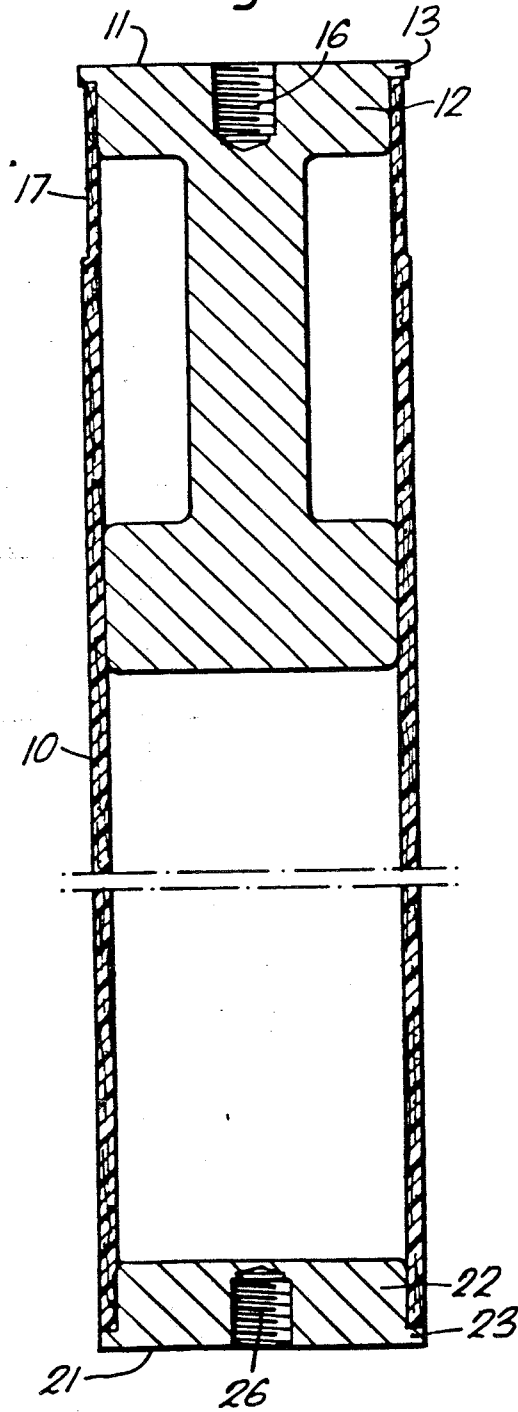
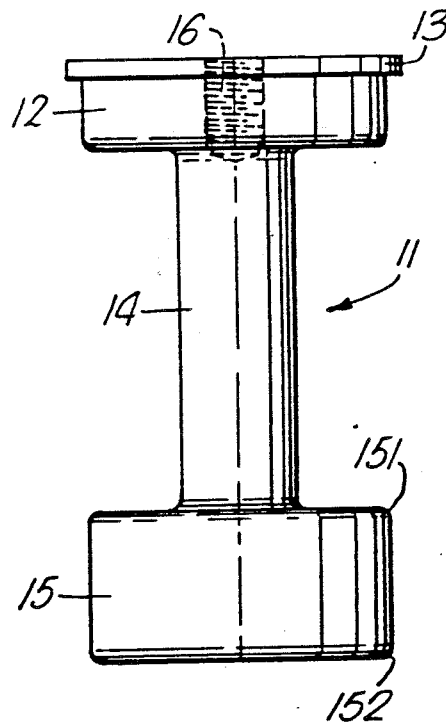


Fig. 2.



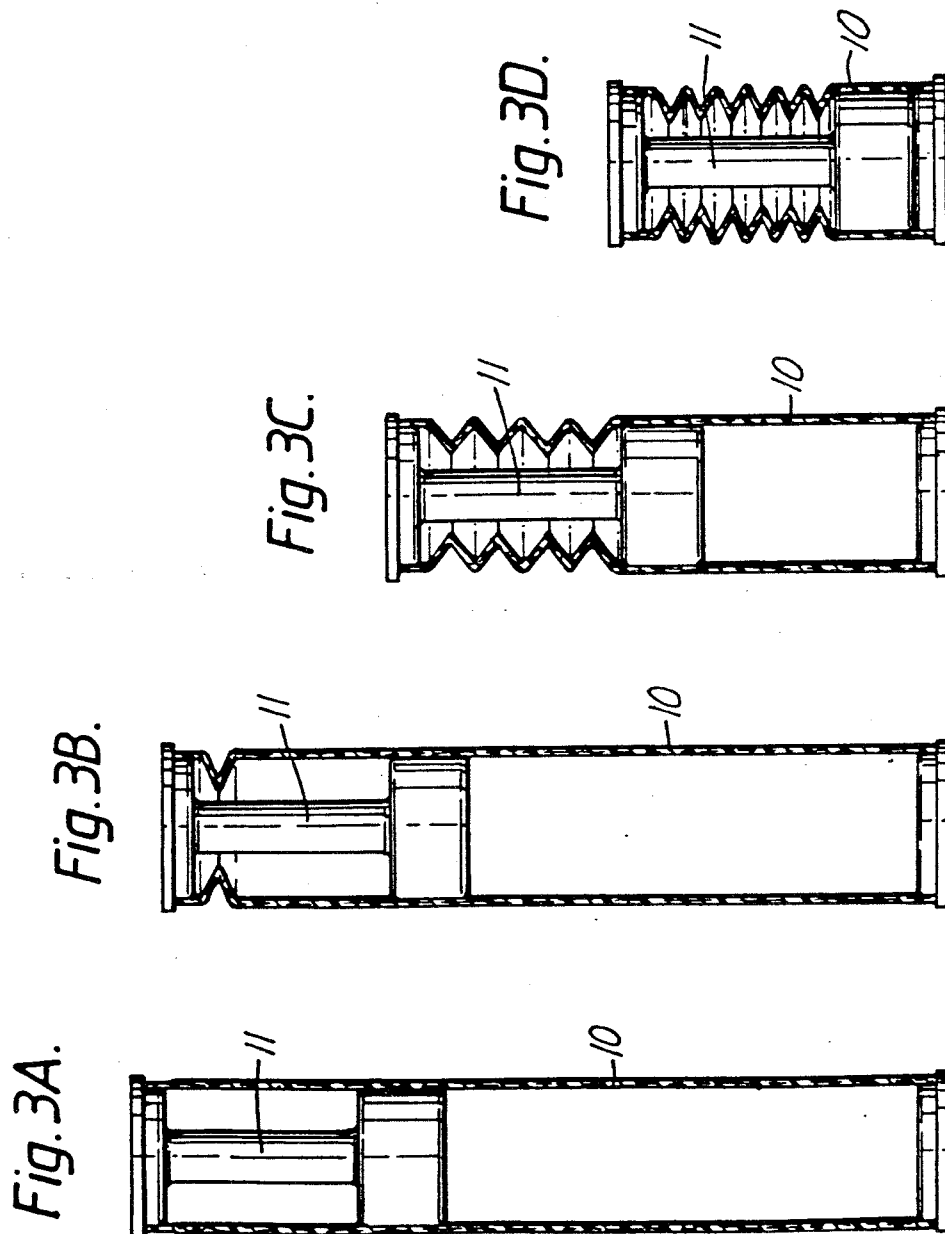


Fig. 4.

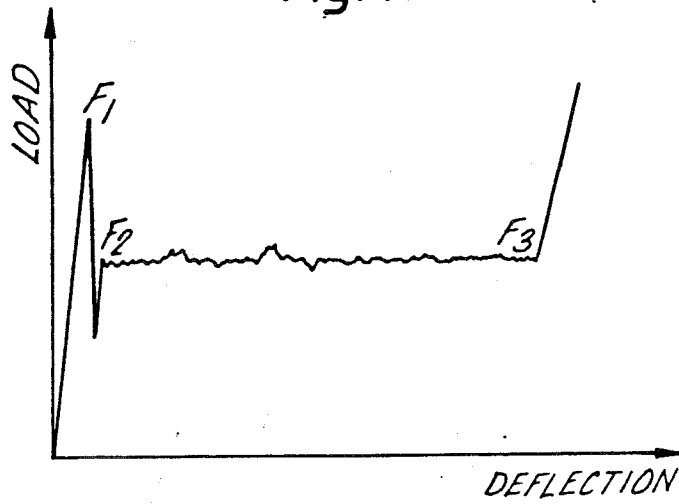
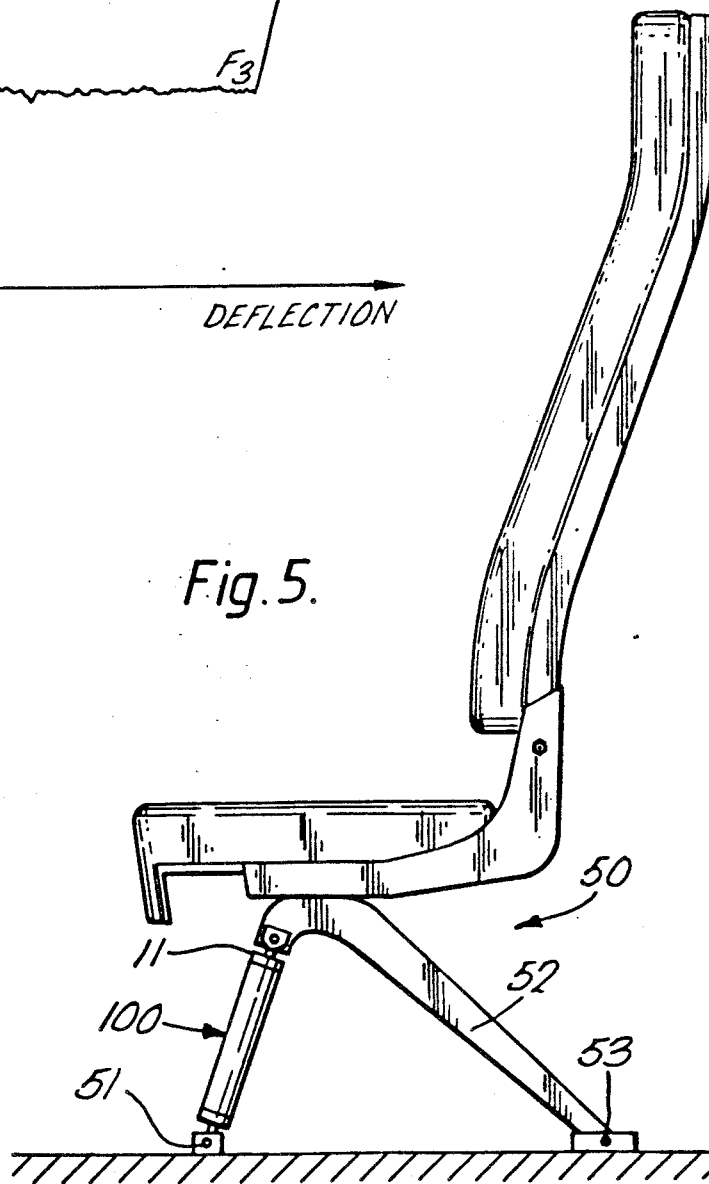


Fig. 5.



5,035,307

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ENERGY ABSORBING DEVICE

This invention relates to an energy absorbing device of a type suitable for use in the leg structure of aircraft passenger seating, where it provides the seat occupants with a degree of relief from the forces experienced in a crash or crash landing situation.

Although the invention is described in the context of aircraft seating (including helicopters) its utility is not limited to this application.

Research has indicated that many aircraft crashes are potentially survivable except that the force of the impact is often sufficient to cause injuries which hinder the passengers' ability to escape from the cabin area. Whilst the injuries in themselves are not necessarily fatal, passengers are incapacitated to a degree which causes them to be overcome by fire or smoke before escape is possible. Those passengers fortunate enough to survive the impact without suffering broken limbs and without falling unconscious due to the sudden deceleration may still find their exit hampered by debris such as unsecured seats which have broken free as a result of the crash forces.

It follows that there is a significant desire to improve the crashworthiness of aircraft passenger seating, but coupled with this there is also a strong commercial desire to reduce the overall weight of aircraft to increase their fuel efficiency. Safer seating which incurs a weight penalty is therefore unlikely to be favourably received.

It is an object of the invention to satisfy both of the above needs by providing an energy absorbing device which is suitable for incorporation into the leg structure of an aircraft passenger seat assembly and which is formed, in large part, from lightweight fibre reinforced plastics material.

The following characteristics are desirable, if not essential, in regard to the primary application of the invention:

(1) The device should form an integral part of the seat assembly leg structure and should be capable of repeatedly withstanding normal loads without deterioration.

(2) It should absorb large compressive forces in order to safeguard the seat occupant.

(3) It should remain intact after controlled collapse to tether the seat in position and thereby maintain escape pathways.

(4) It should resist re-extension in order to attenuate the "rebound" force which follows initial impact.

In order to meet requirements (3) and (4) above, the fibre/resin system must retain some structural integrity after collapse and to this end highly oriented synthetic polymer fibres are preferred.

The use of fibre reinforced plastics materials as energy absorbers is reviewed by Thornton, Harwood and Beardmore in *Composites Science and Technology* 24 (1985) at pages 275-98. This paper discusses both materials and structural aspects of composite energy absorbers and considers how parameters such as fibre and/or resin choice, the geometry of the structure and the fibre arrangements can influence energy absorbing capacity. It also recognises that composite structures having thin wall-sections require a trigger or crush initiation device to ensure that collapse is initiated in a stable mode. However, the article is guarded about the precise form of the crush initiation device best suited to precipitate

2

collapse in those fibre reinforced plastics which crush by local wall buckling rather than by disintegration.

The invention is an energy absorbing device reactive to a sudden compressive overload and subsequent reversal of load comprising:

a first attachment fitting and a second attachment fitting;

a tube of fibre reinforced plastics material extending between the attachment fittings and secured thereto, said tube having a fibre reinforcement of long tough fibres such that it collapses in a local wall buckling mode in response to the overload without snapping of the fibres;

a crush initiation band disposed on or within the tube, at which wall buckling is precipitated;

guide means to maintain the collapsed portion of the tube in axial alignment with the uncollapsed portion when compression takes place, and

a tube wall straightener responsive to re-extension forces which acts so as to press against the tube wall to remove convolutions therein.

The expression "long tough fibres" is used in its conventional sense for fibre composite technology. That is to say, the fibres are of sufficient length to behave indistinguishably from continuous fibres in sustaining tube collapse by local wall buckling and are sufficiently tough to resist brittle fracture when they become bent into a small radius. Very short fibres or whiskers are unlikely to facilitate controlled tube collapse in the desired fashion.

The choice of fibre/resin combination will, to a large extent, determine the collapse characteristics of the device. The most important factor for successful operation of the invention is the mode of collapse of the fibres. As indicated above, the device as a whole is required to collapse by local wall buckling, in which energy is absorbed by crushing of the matrix, failure of the fibre/matrix interface and collapse of the individual fibres. It is clear that a considerable amount of disintegration takes place when the device collapses and this is important for maximizing energy absorption. However, to ensure that some structural integrity is retained after collapse and to provide a residual tensile capability it is essential that a majority of the individual fibres should remain intact in the buckled structure.

Fibres which fracture on compression are unsuitable for exclusive use in this device because they are not capable of resisting tensile loads once fractured. Thus materials such as carbon fibres and glass should be avoided as the sole fibre material. Although they have inherently good tensile properties, the fibres are too brittle to withstand severe buckling without flexural failure.

That is not to say that brittle fibres are totally unsuited to load-limiting applications. In some situations their use is perfectly legitimate because fibre fracture is a recognised energy-absorbing process. It may be possible to absorb the requisite amount of compressive energy, but such a device would not fulfil the objects of the invention especially with regard to its tensile capability.

For this reason the preferred materials are fully aromatic polyamide or "aramid" fibres such as Kevlar (Trade Mark). It is also possible to use fibres of high performance polyethylene which is polyethylene specially formulated for structural applications and having a softening point above 180° C. (see I M WARD: Ultra

high modulus polyolefins; Phil Trans R Soc Lond A 294 (1980) pp 473-82).

On bending, aramid fibres are prone to fibrillation which means that the fibres split longitudinally in the bending zone to form a number of discrete fibrils whose limiting bending radius is much smaller than that of the mother fibre. This is similar in many respects to what happens when a green stick is bent and even if the fibres are twisted they prove very difficult to break. This property of aramid fibres is fully exploited in the device according to the invention. Where higher compressive strengths are required it is possible to use "hybrid" tubes in which a mixture of aramid fibres and some other fibre material is utilized.

The selection of a resin material for the matrix is not so critical. Phenolic and epoxy resin systems are preferred, but the toxicity of combustion products means that epoxy resins are unsuitable for use in confined spaces or where air supply is limited. In applications where elevated temperatures are unlikely to be encountered it may even be possible to use some thermoplastic resins.

As indicated above, it is necessary to include a crush initiation device or trigger near one end of the tube to ensure that initial failure of the tube takes place at a predetermined position and in a controlled manner. This is a circumferentially disposed band which acts symmetrically about the tube and may take the form of a locally reduced wall section or a collar of shrink-wrap tape which is applied to the fibre lay-ups prior to curing and thereby increase the fibre volume fraction to cause a concentration in stress. This collar is preferably removed after curing and before the tube is put into service. It is best to avoid scoring of the tube as an alternative crush initiation mechanism because this is likely to break some of the fibres and may induce an unpredictable mode of collapse.

The collapse characteristics of the device can be varied by altering the geometry of the tube, for example by changing its wall thickness, increasing its diameter or by varying the fibre orientation. In particular, where the tube is formed from a number of pre-impregnated fibre mats, it is possible to influence the collapse characteristics by varying the lay-up sequence: Preferably at least some of the fibres are disposed with a 0/90° orientation relative to the longitudinal axis of the tube so that the 0° fibres are able to sustain tensile loads after collapse of the device. However, it is expected that the tube will often consist of a combination of 0/90° fibres and fibres disposed in some other orthogonal relationship.

Other characteristics of the device according to the invention include:

- (1) Reduced susceptibility to Euler buckling or other catastrophic failure modes, and
- (2) Approximately constant collapse loads over a relatively large axial distance.

The invention will now be described by way of example only with reference to the following drawings, in which:

FIG. 1 shows in longitudinal section an energy absorbing device in accordance with the invention;

FIG. 2 shows in detail the configuration of a preferred sliding end attachment;

FIGS. 3A-3D show a collapse sequence for an embodiment of the device using the preferred sliding end attachment;

FIG. 4 illustrates graphically the variations in load during the collapse sequence;

FIG. 5 shows the device in position on an aircraft passenger seat.

As shown in FIG. 1, an energy absorbing device comprises a fibre reinforced plastics tube 10 extending between a first attachment fitting 11 and a second attachment fitting 21. The tube has a reinforcement of long tough fibres disposed in a compatible resin matrix, with at least some of the fibres disposed with a 0/90° orientation relative to the longitudinal axis of the tube.

The fibres may be aramid fibres, high performance polyethylene fibres or possibly a hybrid of one or more of these with carbon and/or glass fibres. The matrix may be a phenolic, epoxy or any other resin compatible with the selected fibres and which is suitable for the conditions encountered in end use of the device.

The tube is formed with a smooth internal bore by laying up a number of pre-impregnated woven fibre mats onto an internal mandrel and then curing by the application of heat. Reference numeral 17 indicates a crush initiation band which is here represented as a local section of reduced wall thickness where some of the fibre layers have been terminated short of the tube end.

The attachment fittings 11, 21 are each formed from a light metal such as aluminium alloy and comprise a flange 12, 22 receivable inside the tube 10 and adhesively bonded thereto, with lip portions 13, 23. These extend beyond the outer circumference of the tube and serve to transmit compressive forces evenly over its end. The attachment fittings may also be provided with tapped holes 16, 26 to receive fastening means which facilitate attachment to the component to be supported.

Attachment fitting 11 is integrally formed as a combined guide and wall straightener and is shown in more detail in FIG. 2. Additionally it comprises a solid stem portion 14 depending from the flange 12 with a slider 15 at its distal end which is dimensioned to be a sliding fit inside the tube 10. The stem portion 14 has an axial dimension which is sufficient to allow a predetermined length of the tube to collapse and a diameter small enough to accommodate the folds of the collapsed tube. The slider 15 has an axial dimension which is sufficient to resist lateral forces and thereby helps to maintain the tube in axial alignment during collapse and re-extension. Typically, the length of the slider is equal to half its diameter.

The upper and lower faces of the slider 15 are radiussed as shown at 151 and 152, respectively, to assist in easing passage of the slider through the tube. These radiussed corners should neither be too large in case they compromise the slider's resistance to lateral forces, nor too small lest they act to sever the fibres of the tube. This latter consideration is particularly important when the device is reacting under tensile loading and the slider acts as a wall straightener by pressing against the tube wall to remove its convolutions.

Collapse of the energy absorbing device is depicted in FIG. 3 as a sequence of figures numbered 3A to 3D.

FIG. 3A represents the normal condition before compressive overload, where the device is able repeatedly to sustain everyday loads without deterioration.

FIG. 3B shows the condition when compressive overload is encountered and collapse is initiated. Thanks to the crush initiation band, collapse begins in a predetermined position and occurs symmetrically about the tube.

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FIG. 3C shows the tube undergoing collapse in the so-called constant collapse load region. The tube collapses concertina-fashion with the convolutions constrained between the flange and the slider of the sliding attachment fitting. The collapsed portion of the tube is maintained in axial alignment with the uncollapsed portion by virtue of the close fit between the slider and the interior of the tube.

FIG. 3D represents the condition when the available stroke is exhausted.

In order to avoid sudden bottoming-out when the fully-collapsed condition is reached, cushioning means may be provided inside the tube. This may take the form of a piece of phenolic foam which the slider squeezes as it approaches the end of its travel. However, to take full advantage of the weight-saving offered by the device according to the invention it is preferable to avoid additional weight such as cushioning and in any case it is probably best to avoid resilient cushioning which might contribute to the rebound that the invention is designed to attenuate.

At any point during its collapse the device is able to resist a reversal of load and sustain a tensile force by the mechanism of fibre unbuckling as the slider is withdrawn along the longitudinal axis of the tube. The fibres are not pulled straight. Rather, the convolutions are straightened by the pressing action of the slider against the wall of the tube.

Once the device has collapsed and been fully re-extended, any portion of the tube which has been compressed will have very little structural integrity because the fibres in the compressed portion can do little more than serve as a flexible tether. However, it is expected that, if complete collapse followed by full re-extension of the device is sufficient to attenuate the sudden impact load and subsequent rebound load to a tolerable level, any buffeting which follows will be within normal tolerances.

FIG. 4 shows a typical load/displacement curve for an energy absorbing device such as that depicted in FIGS. 1 to 3. The peak collapse load is encountered at

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absorbing leg which incorporates a device 100 according to the invention. The energy absorbing device 100 is anchored to the floor section of the aircraft cabin by its attachment fitting 21 through the agency of a spherical "rose" joint 51. This anchoring arrangement allows angular movement of the device relative to the plane of the floor section to accommodate warpage thereof (in accordance with forthcoming Federal Aviation Authority regulations). The top of the energy absorbing device is secured to the A-frame via attachment fitting 11 in similar fashion. The rear leg 52 is secured to the floor section with a lateral pin joint 53 and thus collapse of the seat assembly as a whole is designed to take place in a forward-tipping motion. This will help to attenuate the forces experienced in a typical crash situation, in which loads are exerted in a generally downward and forward direction.

As an indication of the way in which collapse characteristics can be varied by changing the tube structure, a number of energy absorbing devices were constructed in accordance with the invention and their static performances compared. The tubes were all made from Kevlar (Trade Mark) woven fibre mats disposed in a phenolic resin matrix to a length of 250 mm with a 50 mm internal bore. The lay-up characteristics were as follows:

TYPE A: Five layers of 0/90° prepreg, reducing to three layers for a length of 30 mm at one end;

TYPE B: Five layers of 0/90° prepreg with a double wrap of shrink tape about 7 mm wide in the region 23 to 30 mm from one end;

TYPE C: Three layers of 0/90° prepreg and two layers of $\pm 45^\circ$ prepreg, reducing to three layers of 0/90° prepreg for a length of 30 mm at one end;

TYPE D: As type C with the addition of a shrink wrap tape around the tube immediately after the reduction in wall thickness.

With the exception of Type D tubes, the devices were tested with both push-fit and bonded attachment fittings.

The results are compared in Table 1.

TABLE 1

RESULTS OF STATIC TESTS ON KEVLAR ENERGY ABSORBERS WITH SLIDING END ATTACHMENTS						
TUBE TYPE	END ATTACHMENT	MEAN LOAD (TONNES)	VARIA- TION ABOUT MEAN	INITIAL PEAK LOAD (TONNES)	PEAK LOAD/ MEAN LOAD	ENERGY SHORTFALL
A	PUSH-FIT	1.25	$\pm 27\%$	1.28	1.30	3%
A	BONDED	1.36	$\pm 20\%$	1.54	1.13	6%
B	PUSH-FIT	1.36	$\pm 20\%$	2.11	1.55	5%
B	BONDED	1.43	$\pm 14\%$	2.18	1.52	5%
C	PUSH-FIT	1.22	$\pm 15\%$	1.41	1.15	4%
C	BONDED	1.25	$\pm 15\%$	1.46	1.17	6%
D	PUSH-FIT	1.08	$\pm 17\%$	1.15	1.50	11%

point F1 where collapse is initiated and then the curve dips to a relatively low level, corresponding to a displacement of between about 5 and 10%, before climbing again to the constant collapse load region. This constant collapse load is represented by the substantially horizontal line between the points F2 and F3. At F3 the available stroke of the device is exhausted and further displacement would only be possible under the exertion of much bigger loads.

FIG. 5 illustrates the device incorporated into an aircraft passenger seat, acting as the front legs in a so-called "A frame" construction. The A-frame is a light-weight metal assembly designated by the reference numeral 50 with the front leg substituted by an energy

The reference to "energy shortfall" in Table 1 is meant to indicate the depth of the trough which appears in the load/displacement graph immediately following initial collapse at peak load. It is referred to as a shortfall because where less energy is absorbed the curve deviates from the horizontal plot which would be obtained in the "ideal" case.

From the table it can be seen that tube configuration gives characteristic peak and mean loads for nominally the same length, diameter and wall thickness of tube. This enables the best crush initiation device and/or fibre orientations to be selected for the desired collapse characteristics in a particular application.

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Higher peak loads occur in tubes where the crush initiation device is a band of shrink wrap tape and higher mean loads occur in tubes with a greater proportion of 0/90° oriented fibres. The lowest peak and mean loads are therefore found in the Type D devices where the crush initiation device is a combined shrink wrap and reduced wall thickness mechanism and where two of the fibre layers have their fibres disposed in a $\pm 45^\circ$ orientation relative to the longitudinal axis of the tube. Conversely, the higher peak and mean loads occur in the type B devices where a shrink wrap tape is used on its own for crush initiation purposes and where the fibres are all disposed with a 0/90° orientation.

The devices with bonded end attachments show slightly enhanced values for both peak and mean loads, presumably because the bonding process helps to stabilize the tubes against collapse.

Although the invention has been described with particular reference to its application in improving the crashworthiness of aircraft passenger seating, this should not be construed as a limitation on the scope of the following claims.

We claim:

1. A lightweight energy absorber reactive to sudden compressive overload and subsequent rebound forces, comprising:

- a collapsible composite tube comprising a matrix of plastics material reinforced with long fibers in which the reinforcement comprises tough fibers;
- a trigger zone adjacent one end of said tube which comprises a localized circumferential band of weakening in the tube wall which concentrates stresses relative to the other parts of the tube wall;
- a pair of attachment fittings, each one affixed to a respective end of said tube;
- a slider located within the said tube and being a sliding fit therewithin, and

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a stem substantially coaxial with the tube, said stem being affixed at one of its ends to the respective attachment fitting at the end of said tube adjacent to the trigger zone and affixed at its other end to the slider, the length of the stem being such that the slider is held in spaced-apart relationship from said respective attachment fitting at a position beyond the trigger zone,

whereby longitudinal collapse of the tube commences at the trigger zone through the attachment fittings on application of a compressive load in excess of a predetermined compressive threshold and propagates beyond said trigger zone by a local wall buckling mode on continuance of said excess compressive load, and whereby axial alignment of the tube is maintained during collapse thereof by cooperation between the slider and the uncollapsed portion of the tube.

2. A lightweight energy absorber as claimed in claim 1, wherein the attachment fitting at the end of the tube adjacent to the trigger zone is formed as a unitary fitting with the stem and the slider.

3. A lightweight energy absorber as claimed in claim 1, wherein the trigger zone comprises a circumferential portion of tube having uniformly reduced wall thickness.

4. A lightweight energy absorber as claimed in claim 1, wherein the trigger zone comprises a circumferential portion of tube having a locally increased population of fibers relative to the volume of plastics matrix material.

5. A lightweight energy absorber as claimed in claim 1, wherein the fiber reinforcement comprises fully aromatic polyamide fibers.

6. A lightweight energy absorber as claimed in claim 1, wherein the fiber reinforcement comprises high performance polyethylene fibers.

7. A lightweight energy absorber as claimed in claim 1, wherein the tube is circular in cross-section.

* * * * *

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ENERGY ABSORBER DESIGN				RESULTS OF HORIZONTAL SIMULATION			
SPRING REF.NO	E(1),F(1)	E(2),F(2)	E(3),F(3)	PEAK DECELERATIONS		SEVERITY	SPRING STROKE USED
EA1	6000 lb/in 600 lb	30 lb/in 700 lb	100 lb/in 3000 lb	PELVIS 20g	CHEST 26g HEAD 43g	HEAD 380 CHEST 100	7.36 inches
EA2	6000 lb/in 550 lb	30 lb/in 650 lb	100 lb/in 3000 lb	PELVIS 19g	CHEST 27g HEAD 44g	HEAD 380 CHEST 105	7.52 inches
EA3	6000 lb/in 600 lb	30 lb/in 800 lb	100 lb/in 3000 lb	PELVIS 20g	CHEST 25g HEAD 42g	HEAD 380 CHEST 100	8.03 inches
'RIGID'	- -	- -	- -	PELVIS 20g	CHEST 33g HEAD 60g	HEAD 620 CHEST 170	-
ENERGY ABSORBER DESIGN				RESULTS OF VERTICAL SIMULATION			
SPRING REF.NO	E(1),F(1)	E(2),F(2)	E(3),F(3)	PEAK DECELERATIONS		SEVERITY	SPRING STROKE USED
EA3	6000 lb/in 600 lb	30 lb/in 800 lb	100 lb/in 3000 lb	PELVIS 18g	CHEST 17g HEAD 17g	HEAD 63 CHEST 58	4.58 inches
'RIGID'	- -	- -	- -	PELVIS 22g	CHEST 20g HEAD 19g	HEAD 80 CHEST 65	-

TABLE 3.1 : Stage (A) Simulation Results

ENERGY ABSORBER DESIGN			RESULTS OF HORIZONTAL SIMULATION			
DESIGN REF.NO	E(1),F(1)	E(2),F(2)	E(3),F(3)	PEAK DECELERATIONS	SEVERITY	SPRING STROKE USED
CIC1	6000 lb/in 800 lb	20 lb/in 925 lb	1000 lb/in 7000 lb	PELVIS 35g CHEST 40g HEAD 89g	HEAD 895 CHEST 185	7.82 inches
CIC2	6000 lb/in 850 lb	20 lb/in 975 lb	1000 lb/in 7000 lb	PELVIS 32g CHEST 38g HEAD 85g	HEAD 865 CHEST 180	7.69 inches
CIC3	6000 lb/in 900 lb	20 lb/in 1025 lb	1000 lb/in 7000 lb	PELVIS 29g CHEST 38g HEAD 60g	HEAD 775 CHEST 195	8.98 inches
ENERGY ABSORBER DESIGN			RESULTS OF VERTICAL SIMULATION			
DESIGN REF.NO	E(1),F(1)	E(2),F(2)	E(3),F(3)	PEAK DECELERATIONS	SEVERITY	SPRING STROKE USED
CIC3	6000 lb/in 900 lb	20 lb/in 1025 lb	1000 lb/in 7000 lb	PELVIS 24g CHEST 22g HEAD 21g	HEAD 78 CHEST 68	—
'RIGID'	- -	- -	- -	PELVIS 22g CHEST 20g HEAD 19g	HEAD 78 CHEST 68	—

TABLE 3.2 : Stage (B) Simulation Results

SPECIMEN	LAY-UP	DIMENSIONS l x D x t (mm)	MASS (g)	RELATIVE DENSITY	PEAK/MEAN COLLAPSE LOAD (lb)	COLLAPSE LOAD (kN)	S.E.A. (kJ/kg)
1A	90 ₂ 0 ₁ 90 ₂	101.0x53.3x1.9	41.1	0.14	9950/7300	44.26/32.47	79.8
1B	90 ₆	101.0x66.3x1.4	39.2	0.08	8570/6100	38.12/27.13	69.9
2A	90(0/90) ₃ 90	101.0x52.6x1.7	36.6	0.13	8510/5100	37.85/22.69	62.6
2B	90(+45) ₃ 90	101.0x67.0x1.6	46.3	0.09	9050/5900	40.26/26.24	57.3
2C	90(+45) ₃ 90	101.0x67.0x1.7	48.4	0.10	9500/6200	42.26/27.58	57.6
2D	90(0/90) ₃ 90	101.0x67.0x1.6	44.8	0.09	-	-	-
3A	(+45) ₃	100.0x52.2x1.4	30.4	0.10	5850/3900	26.02/17.35	57.1
3B	(+45) ₃	100.0x51.8x1.5	30.6	0.11	6080/3950	27.05/17.57	57.4
3C	(+45) ₃	100.6x52.5x1.5	31.7	0.11	6300/4100	28.02/18.24	57.9
4A/1	(+45) ₂	100.1x51.2x1.1	20.6	0.08	3780/2450	16.81/10.90	53.0
4A/2	(+45) ₂	149.2x51.3x1.1	30.8	0.08	4120/2550	18.33/11.34	55.0
4B	(+45) ₂	250.0x50.6x1.1	51.0	0.09	4180/2400	18.59/10.68	52.3
4C/1	(+45) ₂	100.0x51.7x1.1	20.9	0.08	4700/2600	20.91/11.57	55.3
4C/2	(+45) ₂	149.8x51.5x1.1	31.2	0.08	4340/2600	19.31/11.57	55.5
4D	(+45) ₁	97.0x51.4x0.6	10.7	0.05	1350/450	6.01/2.00	18.1

TABLE 4.1 : STATIC TEST RESULTS FOR CFRP TUBES

REF.	END FITTINGS	TRIGGER METHOD	LAY-UP	DIMENSIONS LxDxt (mm)	MASS g	RELATIVE DENSITY	PEAK (lb)	MEAN (lb)	PEAK/MEAN OVERSHOOT
5A/1	PUSH/PUSH	NONE	0/90 ₄	101x52.1x1.1	24.0	0.08	4130	1500	175%
5A/2	PUSH/PUSH	NONE	0/90 ₄	194x52.1x1.1	46.2	0.08	3800	-	-
5B/1	PUSH/PUSH	NONE	0/90 ₄	150x52.3x1.2	37.0	0.09	3630	-	-
5B/2	PUSH/PUSH	NONE	0/90 ₄	150x52.3x1.2	37.1	0.09	3490	1700	105%
5C/1	PUSH/PUSH	NONE	0/90 ₅	100x52.7x1.3	30.4	0.10	5000	1600	213%
5C/2	NONE/NONE	NONE	0/90 ₅	100x52.7x1.3	30.5	0.10	4530	2550	78%
5D/1	NONE/NONE	NONE	0/90 ₅	101x53.7x1.3	28.9	0.10	4120	2400	72%
5D/2	NONE/NONE	NONE	0/90 ₅	201x53.7x1.3	58.4	0.10	4080	2300	77%
6A/1	PUSH/PUSH	5-3 Red.XS	0/90 ₅	150x52.5x1.3	38.9	0.10	2480	2000	24%
6A/2	BOND/BOND	5-3 Red.XS	0/90 ₅	150x52.05x1.3	39.0	0.10	3510	2500	40%
6B/1	PUSH/PUSH	5-4 Red.XS	0/90 ₅	150x52.6x1.4	41.9	0.10	3550	2000	78%
6D/1	PUSH/PUSH	S/TAPE(1)	0/90 ₅	150x53.6x1.3	43.1	0.10	4550	2300	98%
6D/2	PUSH/PUSH	S/TAPE(2)	0/90 ₅	151x53.6x1.3	42.5	0.10	3900	2300	70%
6F/1	PUSH/PUSH	S/TAPE(2)	0/90 ₅	150x52.5x1.4	43.3	0.10	4550	2700	69%
6F/2	PUSH/PUSH	NONE	0/90 ₅	150x52.5x1.4	43.8	0.10	4600	-	-
7A/2	BOND/BOND	5-3 Red.XS	0/90 ₅	159x52.7x1.3	41.5	0.10	3400	2900	17%
7B	BOND/BOND	5-3 Red.XS	0/90 ₅	250x52.5x1.3	68.9	0.10	4400	-	-
7C/1	BOND/BOND	4-3 Red.XS	0/90 ₄	142x51.9x1.0	28.8	0.08	4220	2100	100%
7C/2	BOND/BOND	4-3 Red.XS	0/90 ₄	144x51.9x1.0	29.6	0.08	3210	2000	60%
8A/1	PUSH/PUSH	5-3 Red.XS	0/90 ₃ + -45 ₂	150x52.7x1.4	40.3	0.10	2900	2500	16%
8A/2	PUSH/NONE	NONE	0/90 ₃ + -45 ₂	150x52.7x1.4	44.3	0.10	4100	2400	71%
8B/1	PUSH/PUSH	5-3 Red.XS	0/90 ₁ + -45 ₄	150x52.6x1.4	38.8	0.10	2500	2100	19%
8B/2	PUSH/NONE	NONE	0/90 ₁ + -45 ₄	150x52.6x1.4	43.0	0.10	3750	2100	79%

TABLE 4.2 : COMPRESSION TEST RESULTS ON KEVLAR TUBES

TUBE TYPE	END FITTINGS	REF NO.	MEAN LOAD(lb)	VARIATION ABOUT MEAN(%)	INITIAL PEAK LOAD(lb)	PEAK/MEAN OVERSHOOT	ENERGY SHORTFALL
I	push-on	11b	2750	+27%	2830	3%	3%
I	bonded	11a	3000	+20%	3400	13%	6%
II	push-on	10b	3000	+20%	4650	55%	5%
II	bonded	10a	3150	+14%	4800	52%	5%
III	push-on	9c	2700	+15%	3100	15%	4%
III	bonded	9a	2750	+15%	3210	17%	6%
IV	push-on	9b	2400	+17%	2530	5%	11%

TABLE 5.1 : RESULTS OF STATIC TESTS ON PROTOTYPE KEVLAR ENERGY ABSORBERS

D Y N A M I C T E S T S				S T A T I C T E S T S		
TUBE TYPE	END FITTINGS	REF NO.	IMPACT SPEED (m/s)	LOAD WASHER MEAN LOAD (lb)	REF NO.	LOAD CELL MEAN LOAD (lb)
I	bonded	11c	3.71	3230	11a	3000
II	push-on	10c	3.10	3330	10b	3000
II	bonded	12a	3.71	2800*	10a	3150
III	push-on	12c	3.10	3600*	9c	2700
III	bonded	14c	3.63	3490	9a	2750
Va	bonded	14a	3.67	2805	-	-
Vb	bonded	15c	3.60	2400	-	-
VIa	bonded	14b	3.59	2760	-	-
VIb	bonded	15b	3.62	2240	-	-
VII	bonded	15a	3.60	2260	-	-

* failure mode differs from static test - see text

TABLE 5.2 : RESULTS OF DYNAMIC TESTS ON PROTOTYPE KEVLAR ENERGY ABSORBERS

Seat Pitch	28"	30"	32"
Head Impact Velocity	37.2ft/s (112ms)	36.8ft/s (118ms)	30.6ft/s (146ms)
Seat Back Angle at Impact	20.9 deg	21.2 deg	21.4 deg
Seat Back Ang. Velocity at Impact	2.1 rad/sec	1.0 rad/sec	0 rad/sec
Head Resultant Acceleration	102g	70g	77g
HIC	388	219	283
HSI	643	334	475
Lap Belt Load (peak)	12301b	13001b	13501b

TABLE 7.1 : Seat Pitch Study Results

Seat Breakover	35ft lb	45ft lb	Restricted
Head Impact Velocity	30.2ft/s (146ms)	37.2ft/s (115ms)	37.4ft/s (112ms)
Seat Back Angle at Impact	24.2 deg	18.6 deg	16 deg (max)
Seat Back Ang. Vel. at Impact	0.0	0.8 rad/sec	-1.6 rad/sec
Head Resultant Acceleration	94g	92g	>160g
HIC	282*	319	1038
HSI	559	518	1490

* Based on incomplete results

TABLE 7.2 : Seat Breakover Study Results

	CASE 1: Upright Occupant Seat @ 31" Pitch (no contact)	CASE 2: Braced Occupant Seat @ 31" Pitch	CASE 3: Braced Occupant No Seat Back
Head Peak Accel.	22g @ 42ms	15g @ 16ms	38g @ 70ms
Chest Peak Accel.	24g @ 42ms	17g @ 43ms	26g @ 77ms
Pelvis Peak Accel	24g @ 42ms	20g @ 30ms	19g @ 68ms
Peak Lap Belt Load	660lb	590lb	655lb
HSI	58	24	134
CSI	64	41	77

TABLE 7.3 : Occupant Initial Position
Results

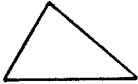
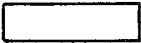
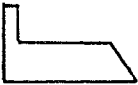
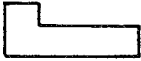
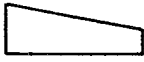
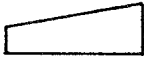
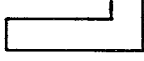
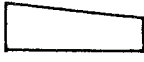
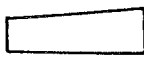
PULSE	SHAPE	STOPPING DISTANCE (in)	MAXIMUM PELVIS (g)	ACCELERATIONS CHEST (g)	HEAD (g)	HIC	CSI	HSI
01		20.9	61	56	75	562	795	957
02		23.7	39	38	63	332	287	520
03		19.6	42	42	67	391	344	585
04		21.3	39	41	64	352	283	520
05		21.5	39	39	64	344	306	533
06		21.8	42	43	67	376	323	564
07		25.6	37	36	56	270	299	433
08		26.2	34	34	55	235	235	424
09		22.3	41	41	65	352	307	537
10		25.2	38	37	58	284	289	460

TABLE B1 : Pulse Simulation Results

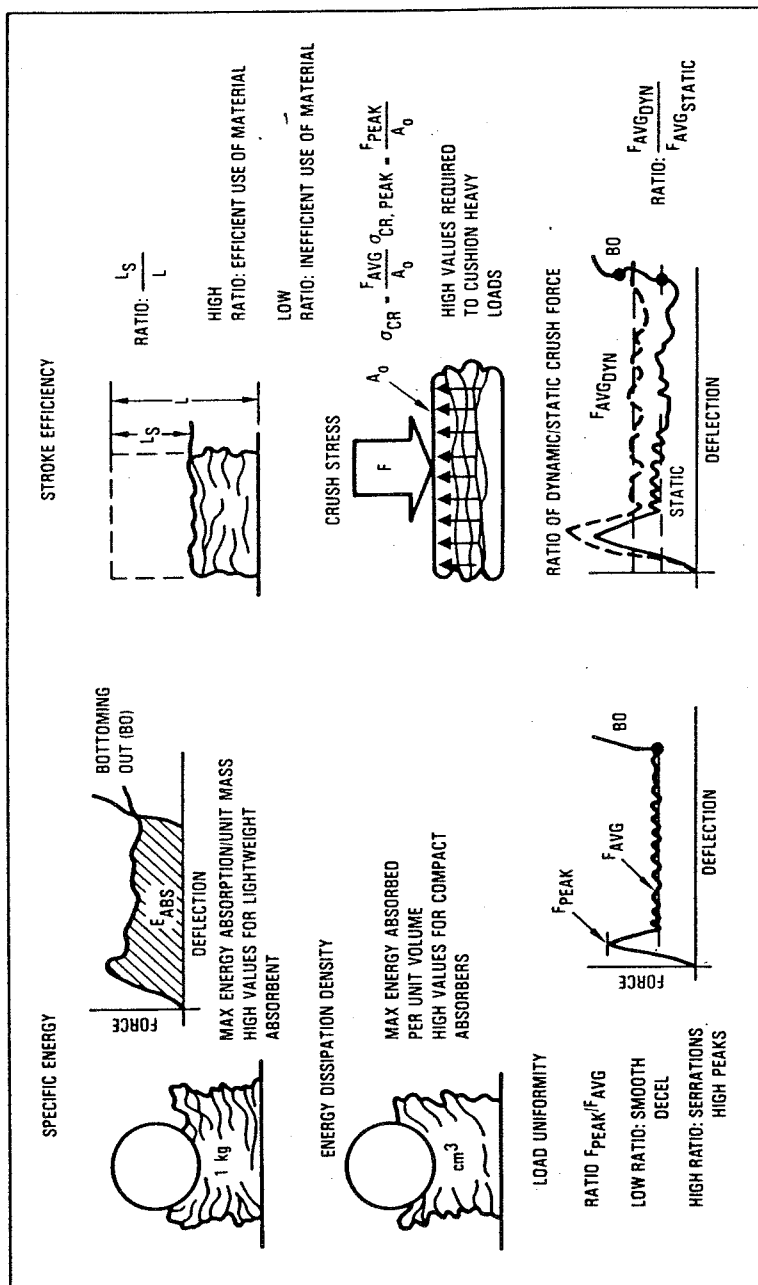


FIGURE I1 : Energy Absorption Terms

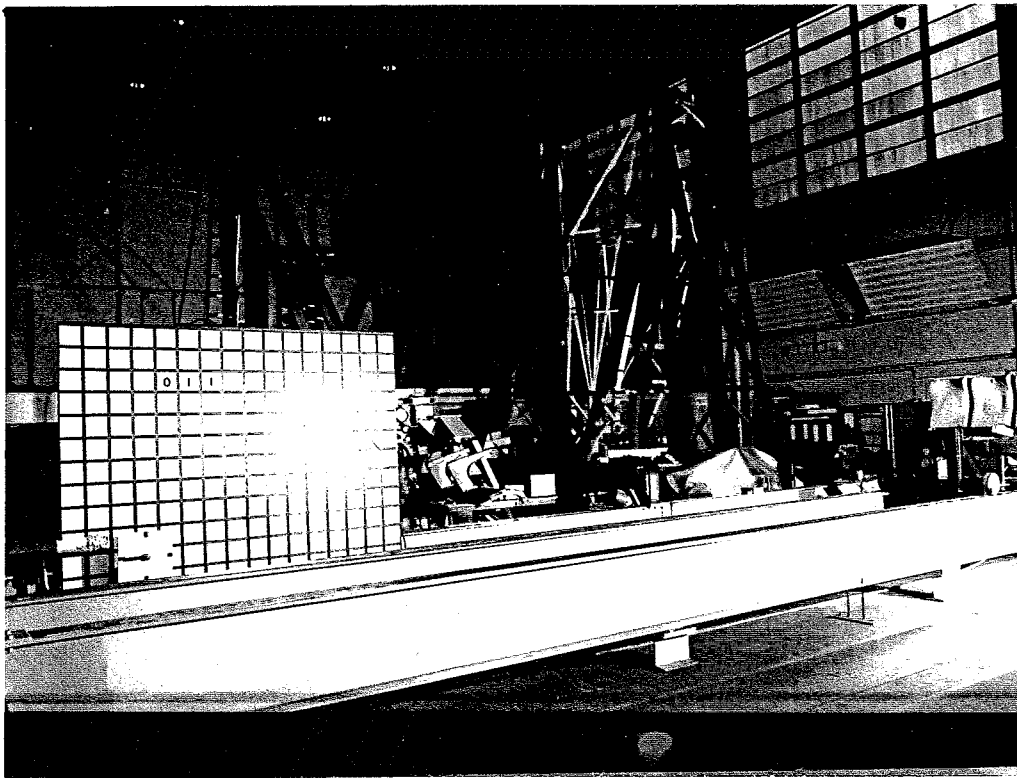


FIGURE 1.1a : CIC Seat Test Facility
View from Impact End

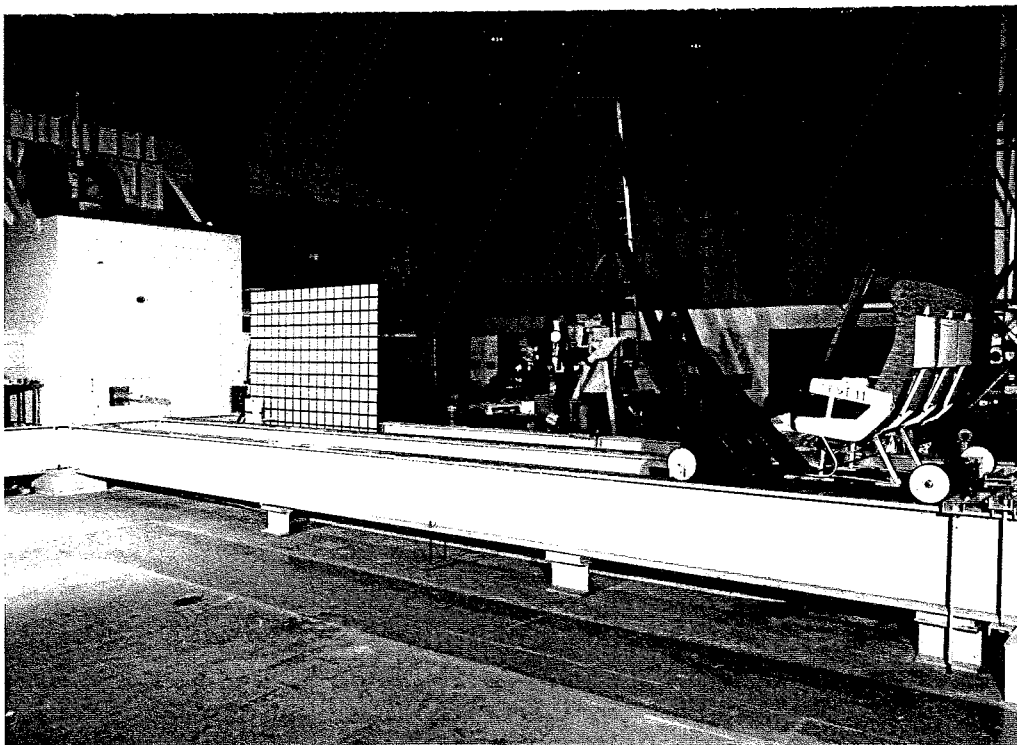


FIGURE 1.1b : CIC Seat Test Facility
View from Launch End

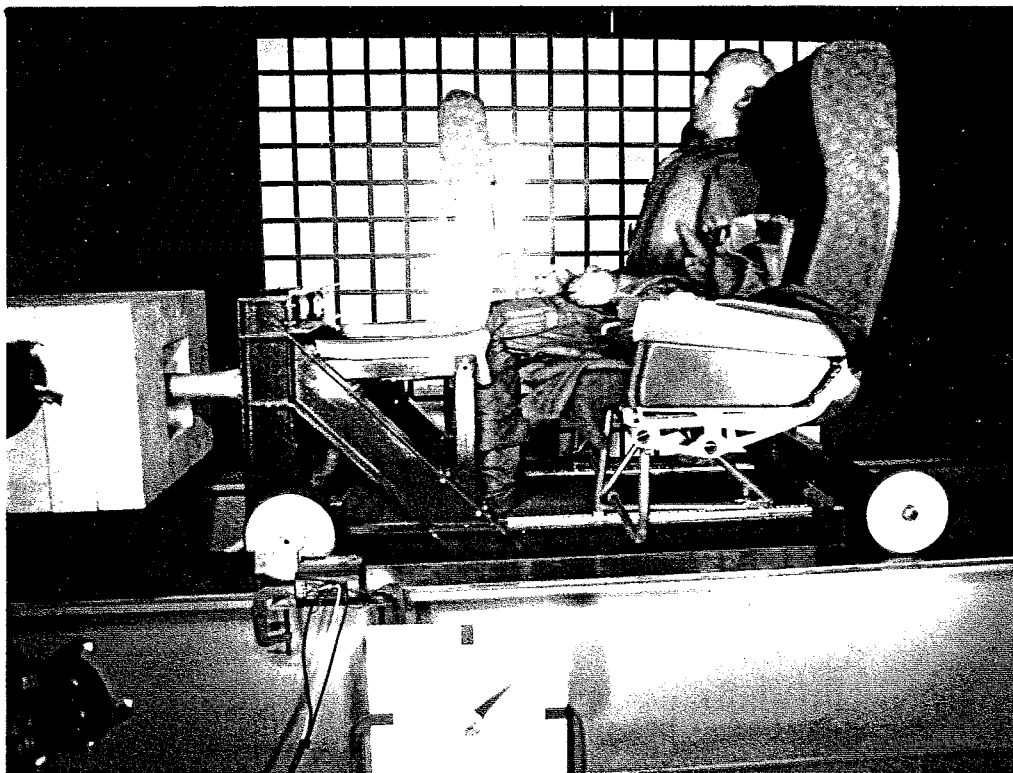


FIGURE 1.2 : Test Configuration (A)
Seat Type 2

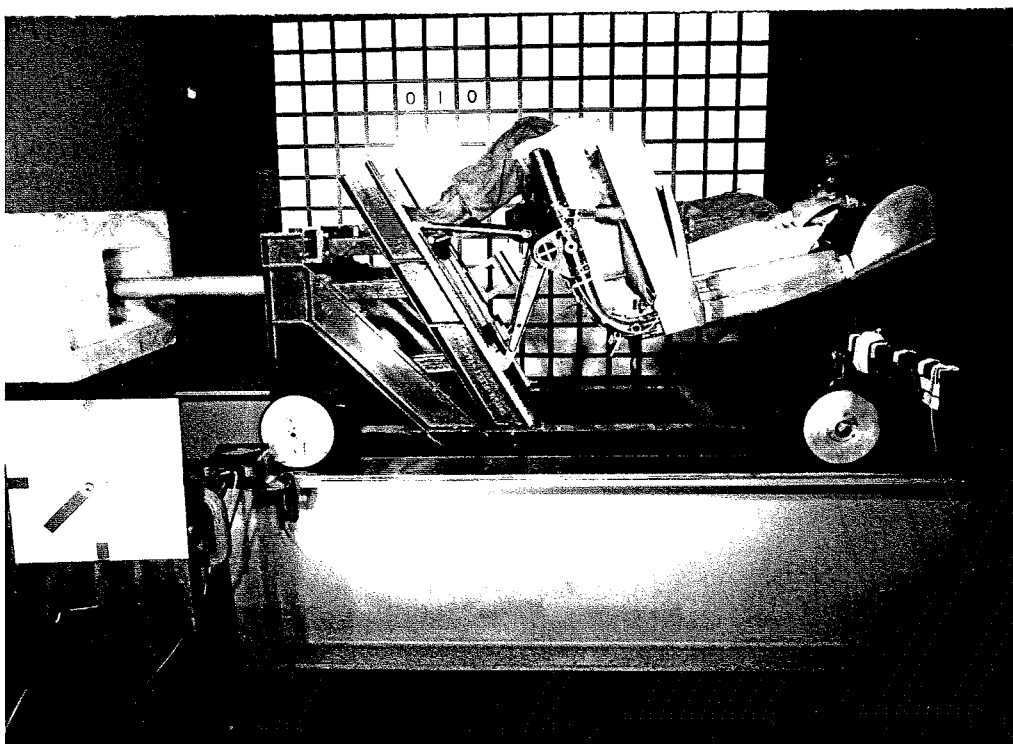


FIGURE 1.3 : Test Configuration (B)
Seat Type 3

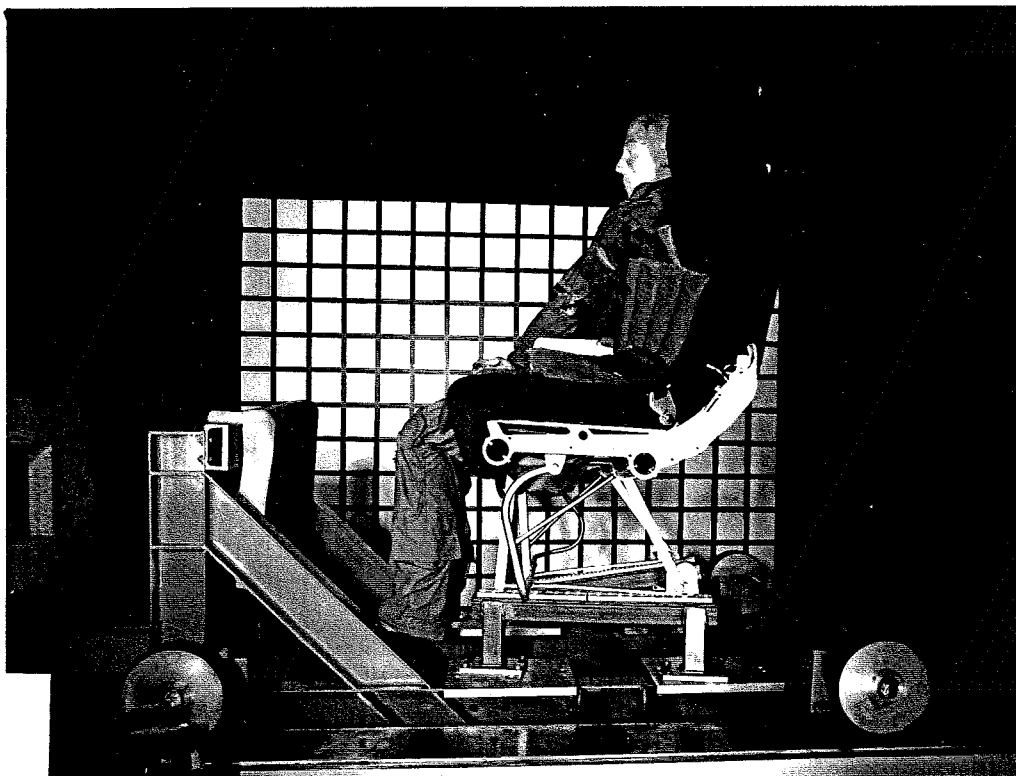


FIGURE 1.4a : Test Configuration (C)
Seat Type 1

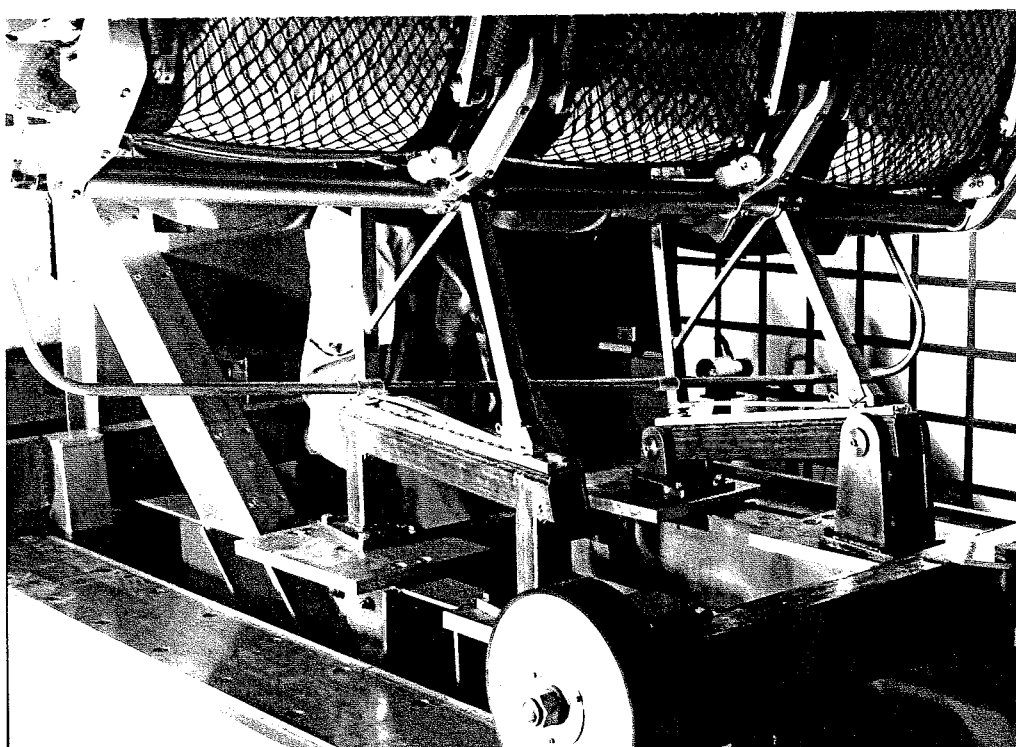


FIGURE 1.4b : Test Configuration (C)
Seat Type 1 (detail)

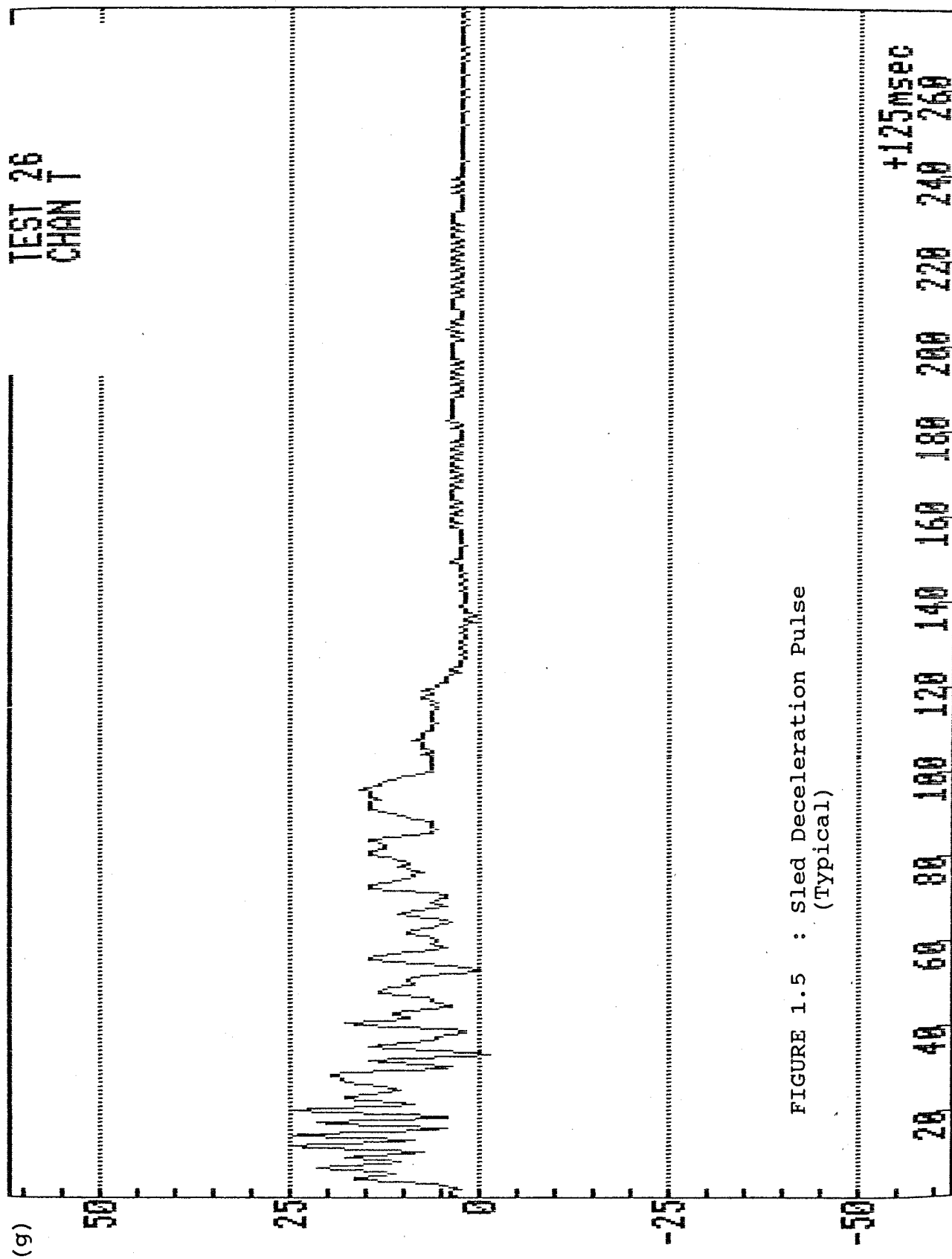


FIGURE 1.5 : Sled Deceleration Pulse
(Typical)



FIGURE 2.1 : Test Configuration (C)
Seat Type 4

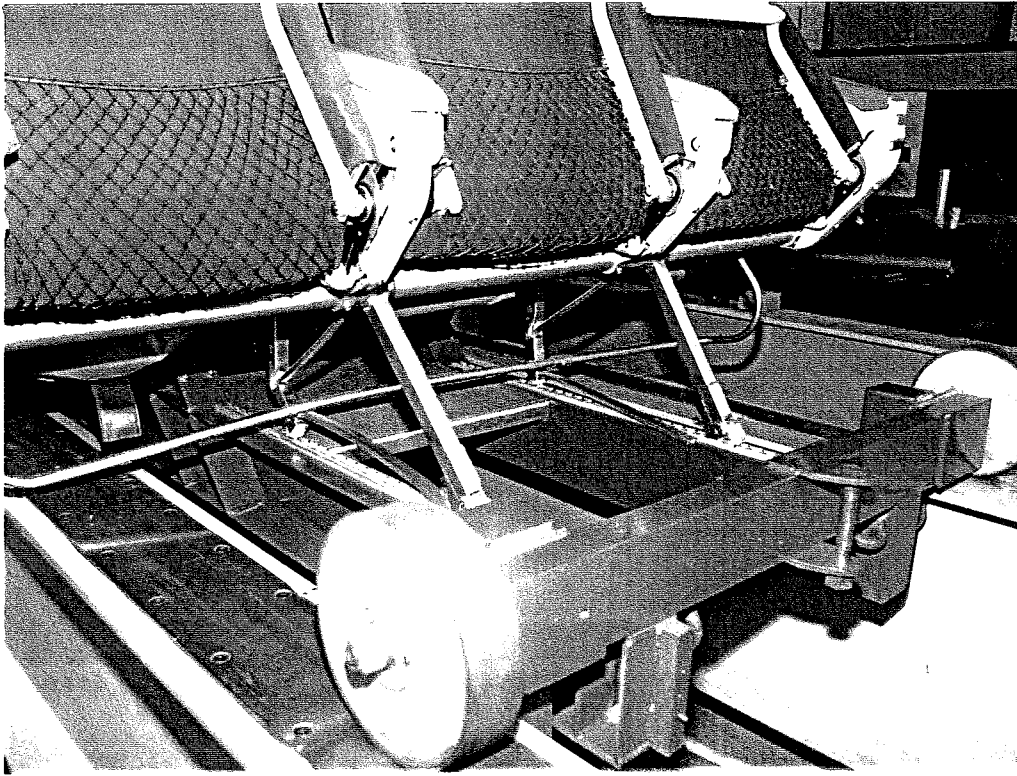


FIGURE 2.2 : Test 01 Post-test

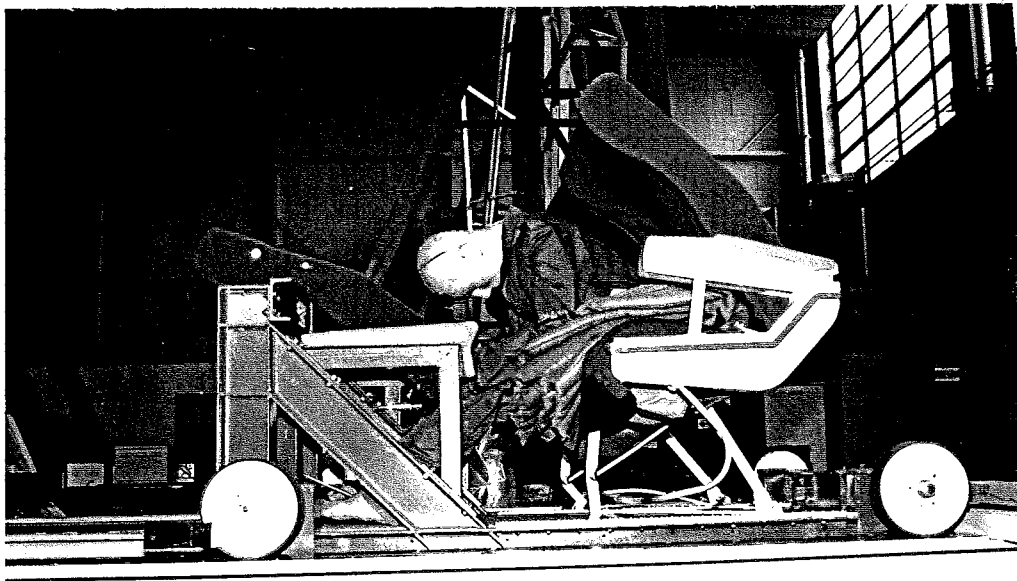


FIGURE 2.3 : Test 02 Post-test

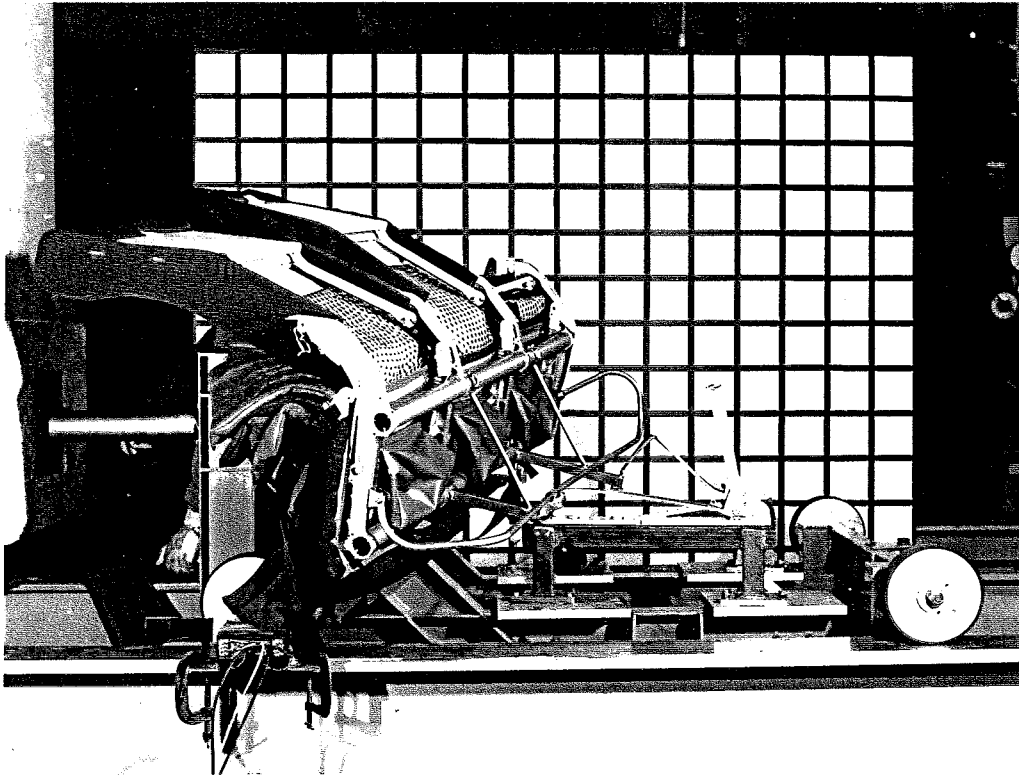


FIGURE 2.4a : Test 05 Post-test

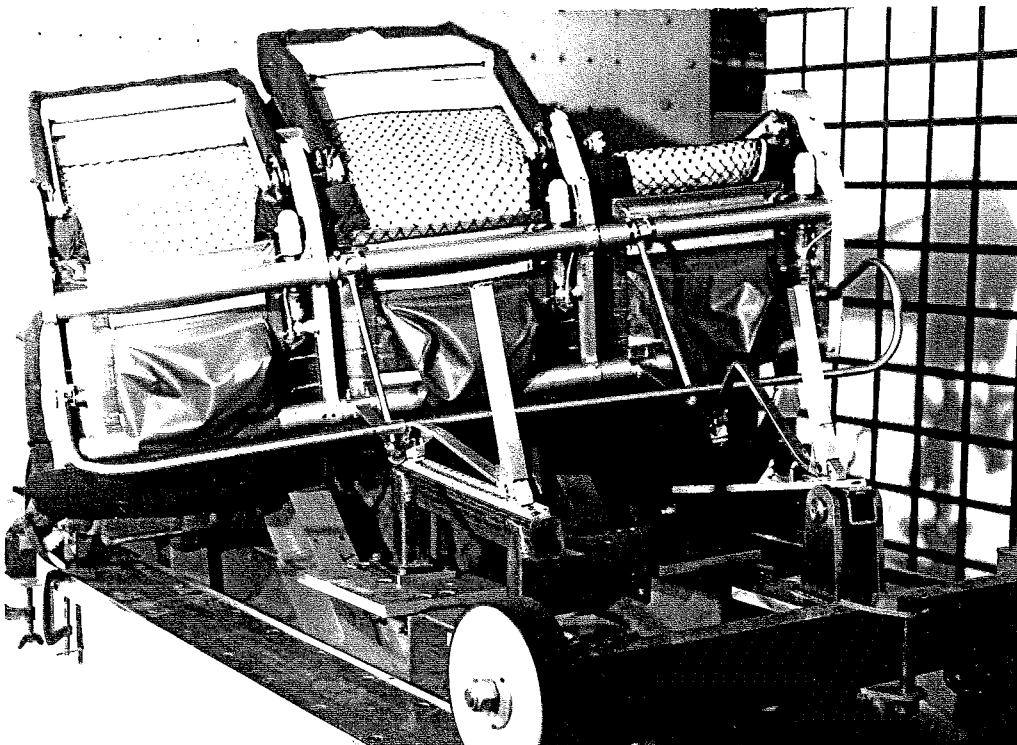


FIGURE 2.4b : Test 05 Post-test

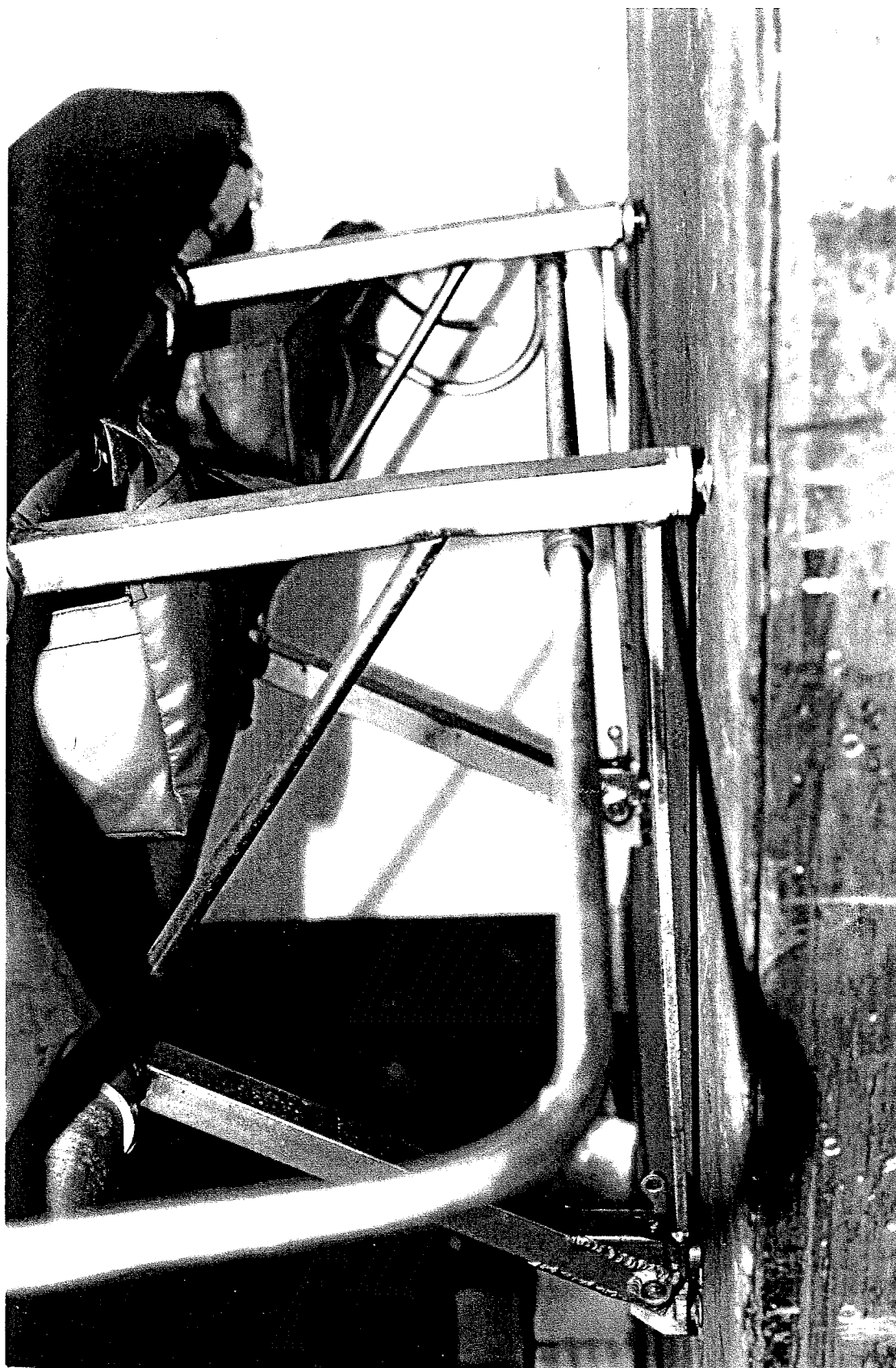


FIGURE 2.5 : Test 11 Post-test

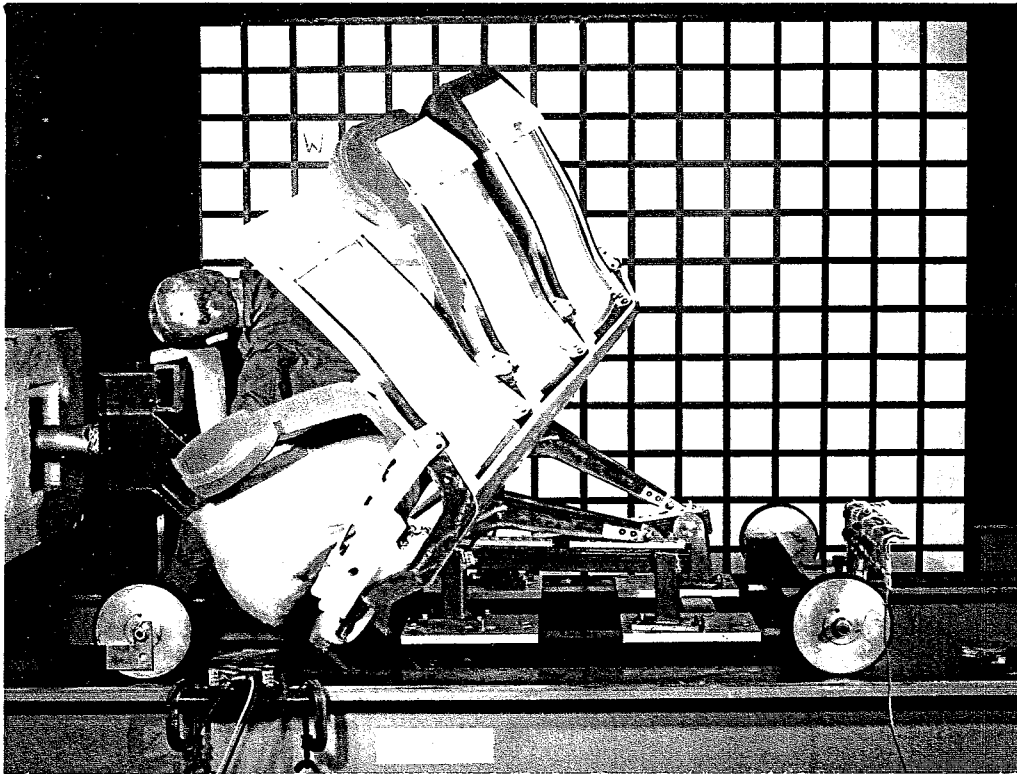


FIGURE 2.6a : Test 06 Post-test

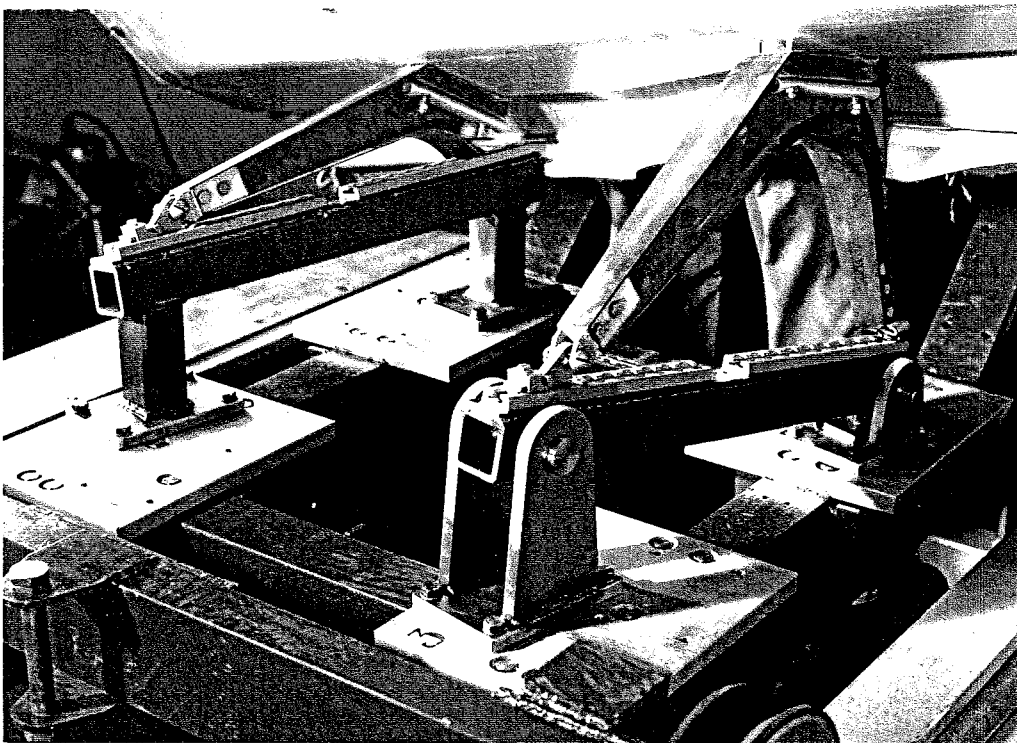


FIGURE 2.6b : Test 06 Post-test

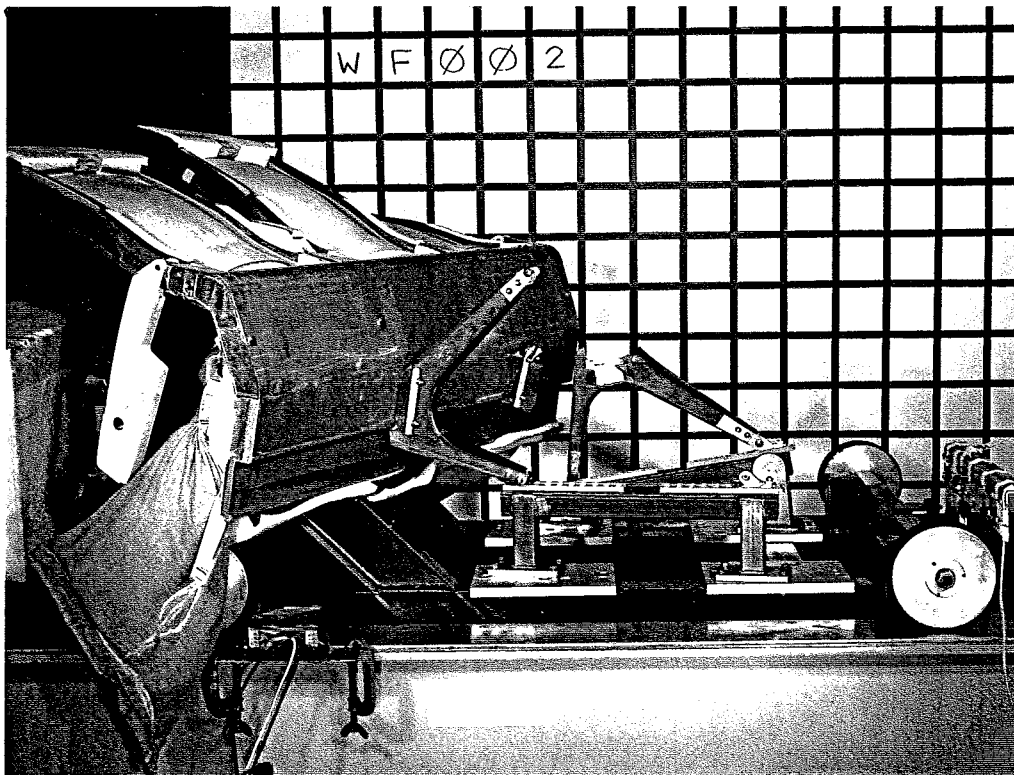


FIGURE 2.7a : Test 07 Post-test

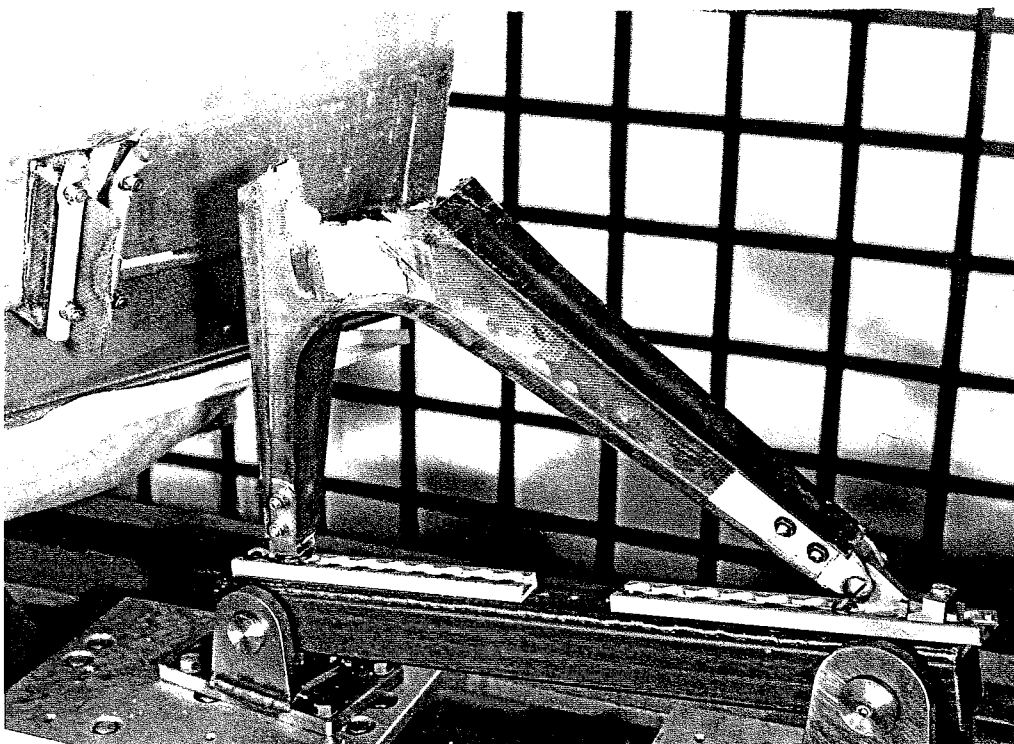


FIGURE 2.7b : Test 07 Post-test

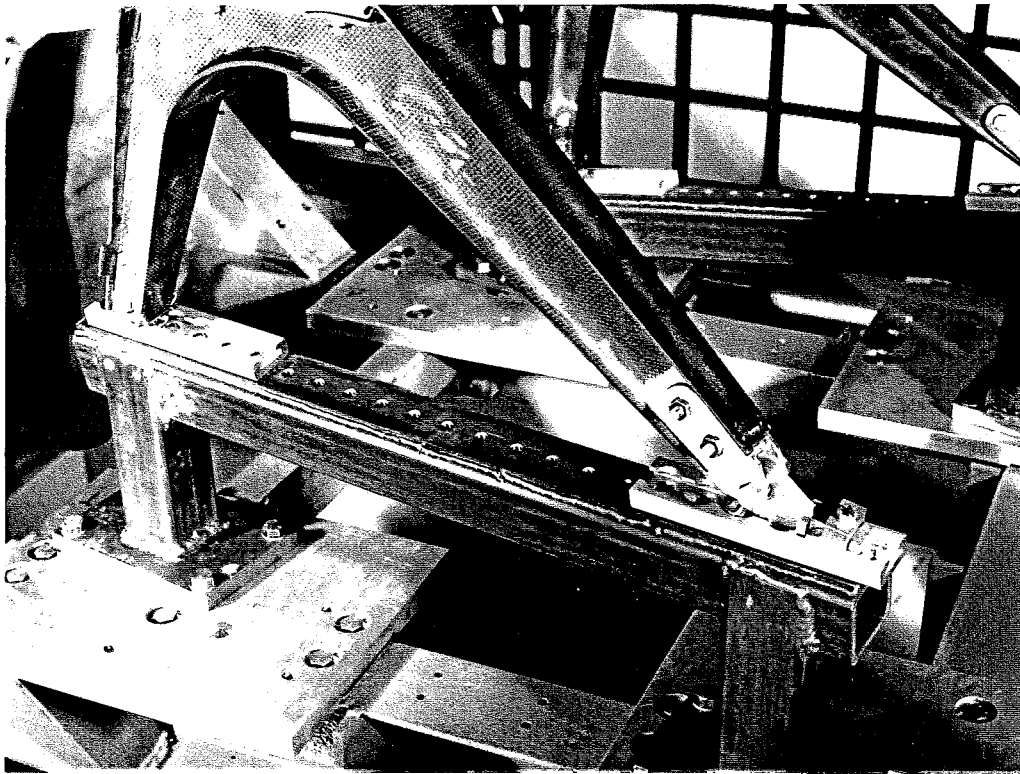


FIGURE 2.8 : Test 08 Post-test

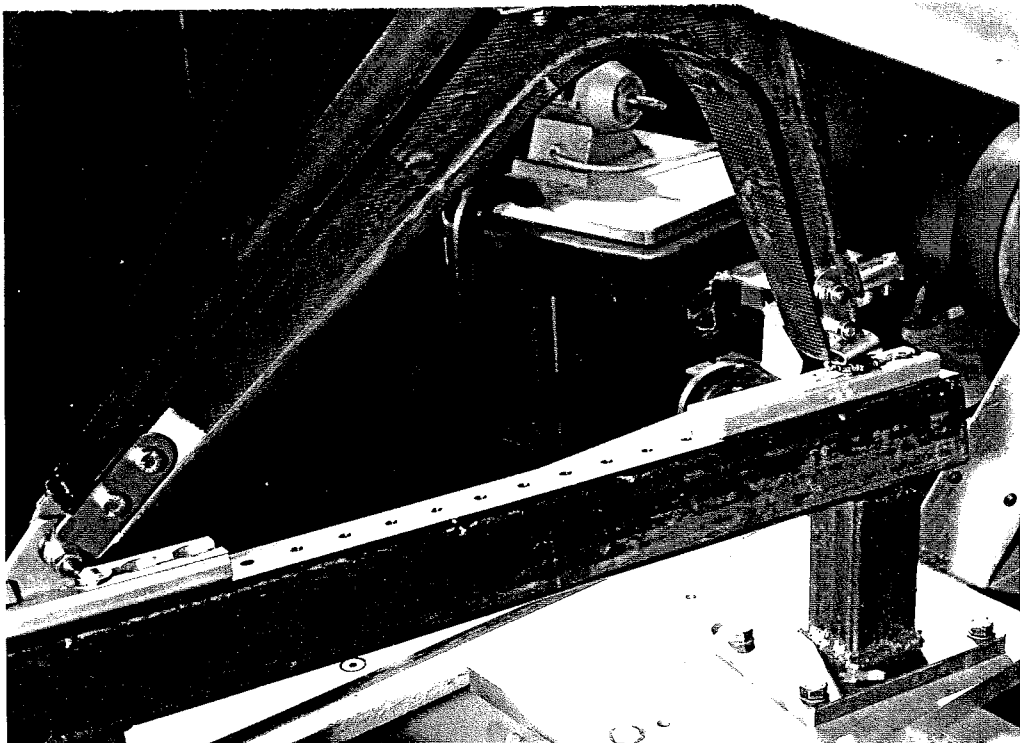


FIGURE 2.9 : Test 08 Post-test

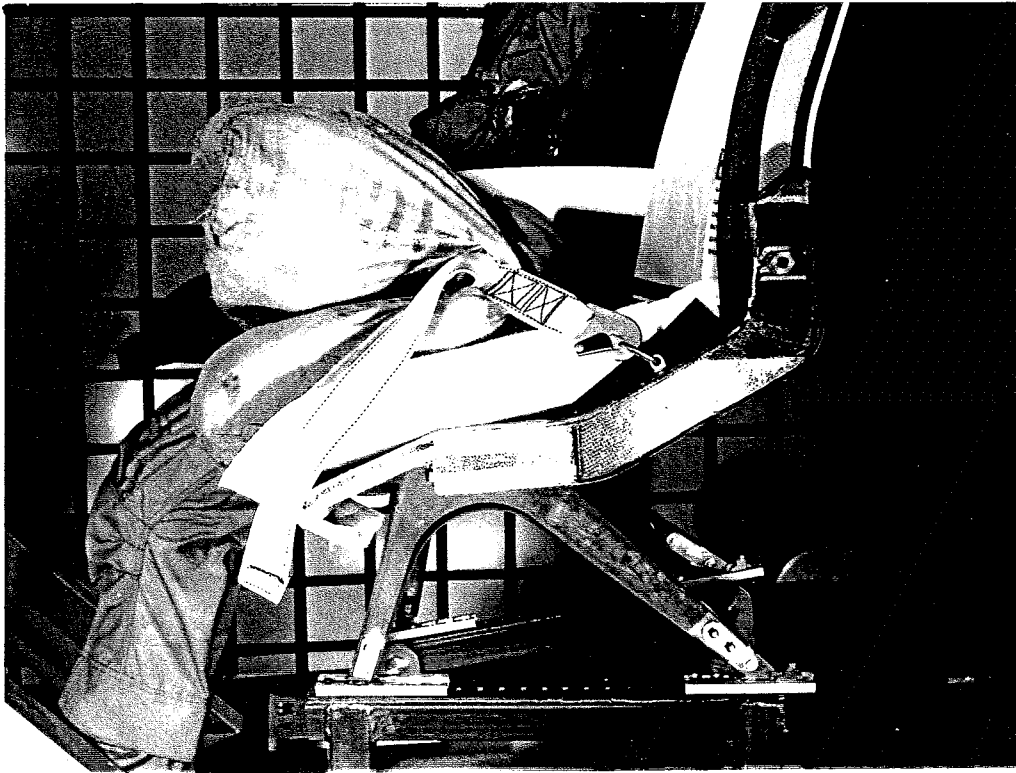


FIGURE 2.10 : Test 08 Post-test

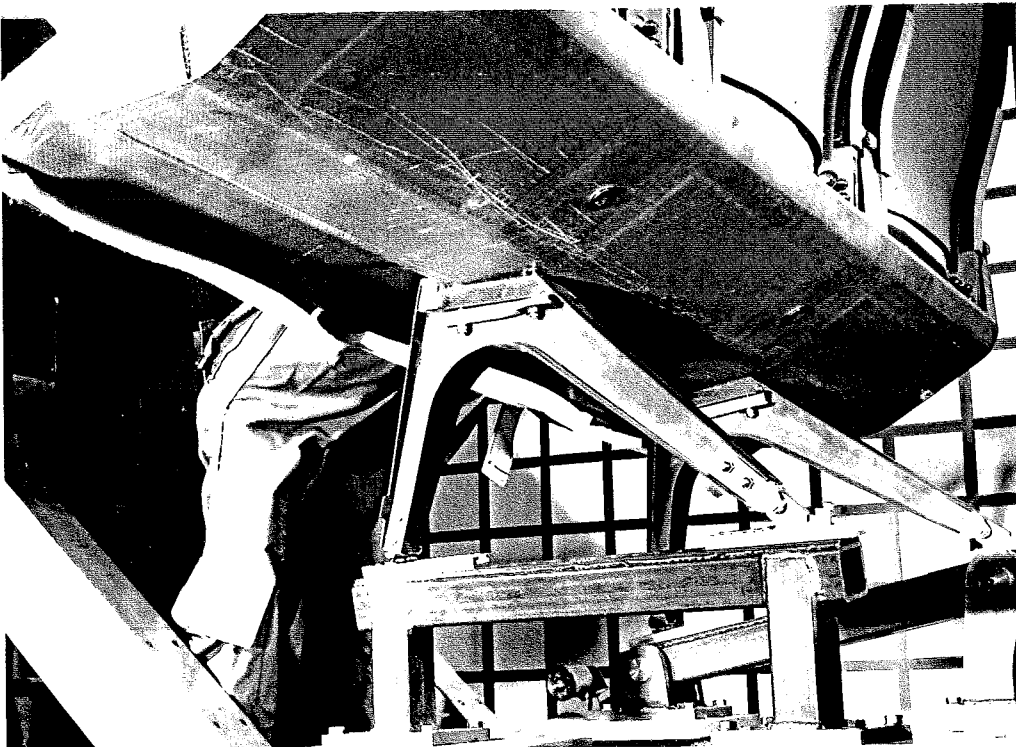
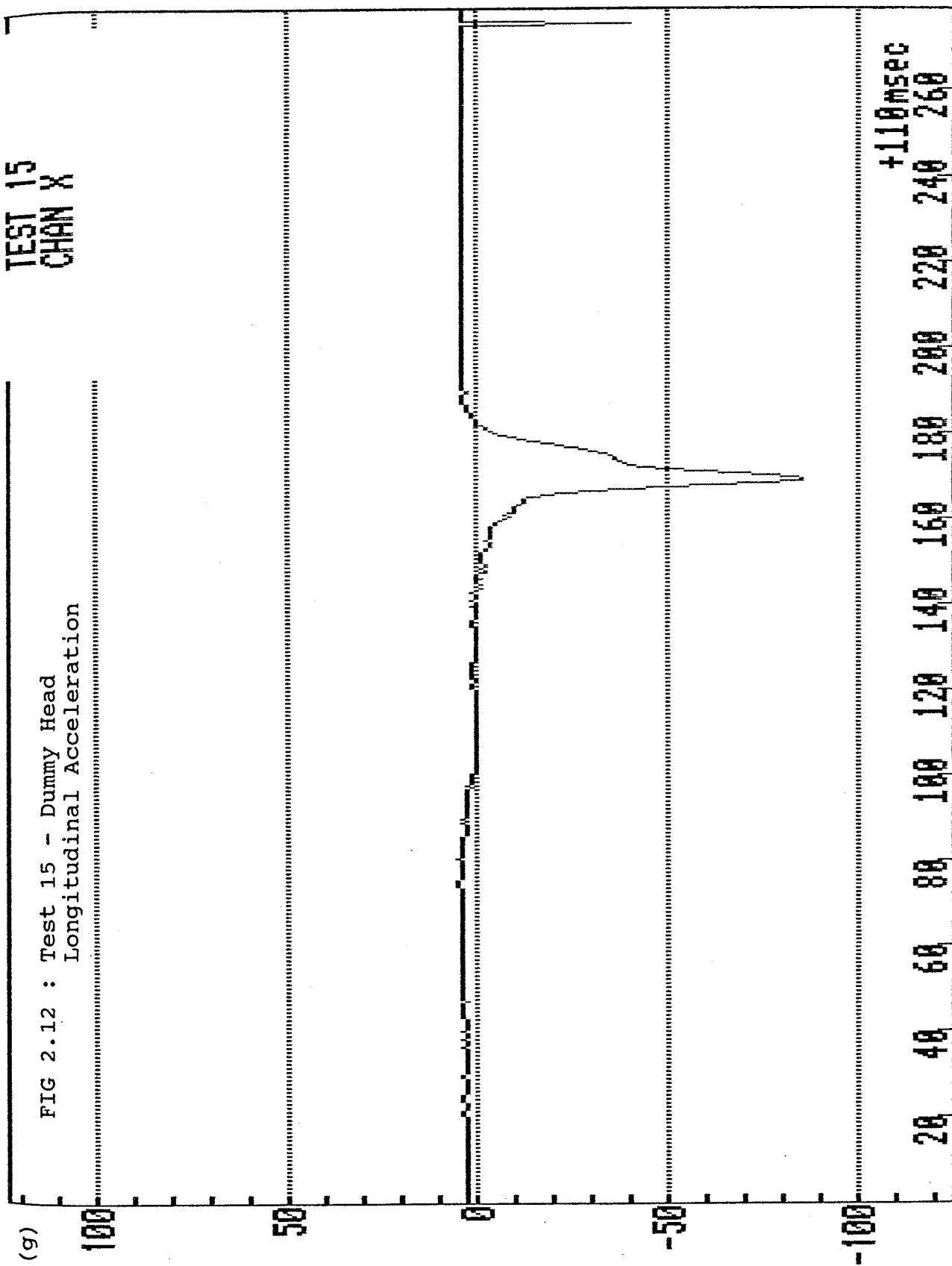
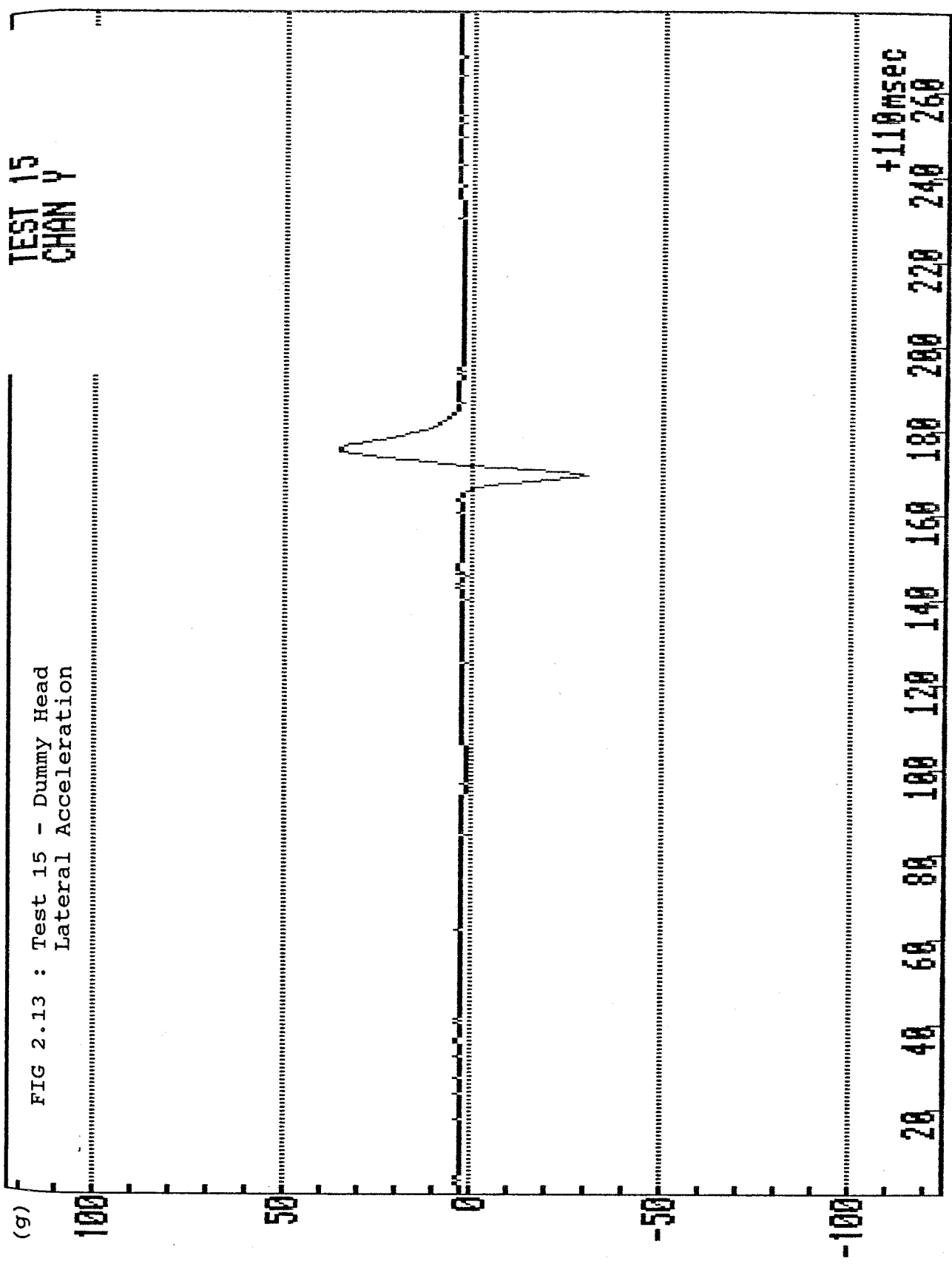


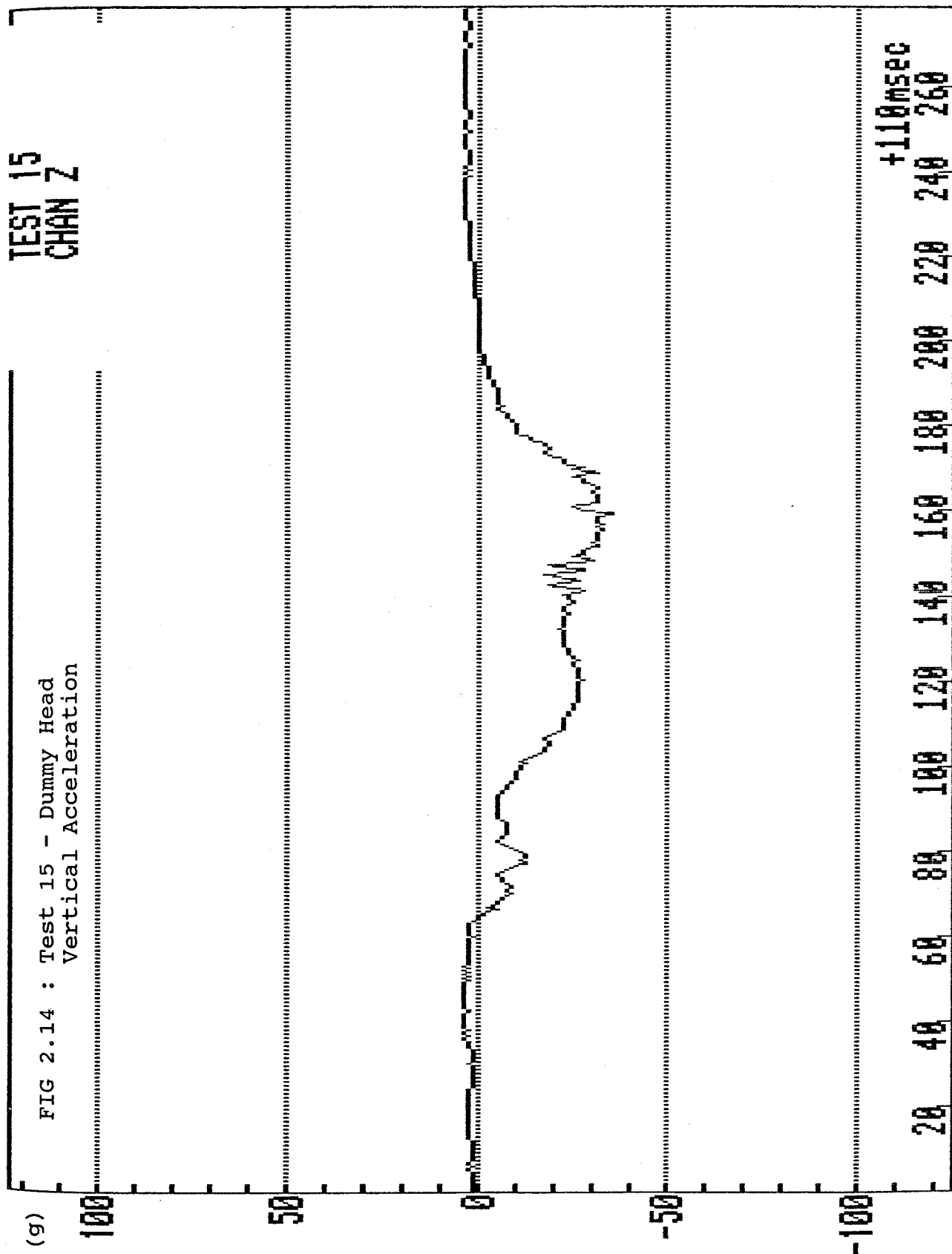
FIGURE 2.11 : Test 08 Post-test

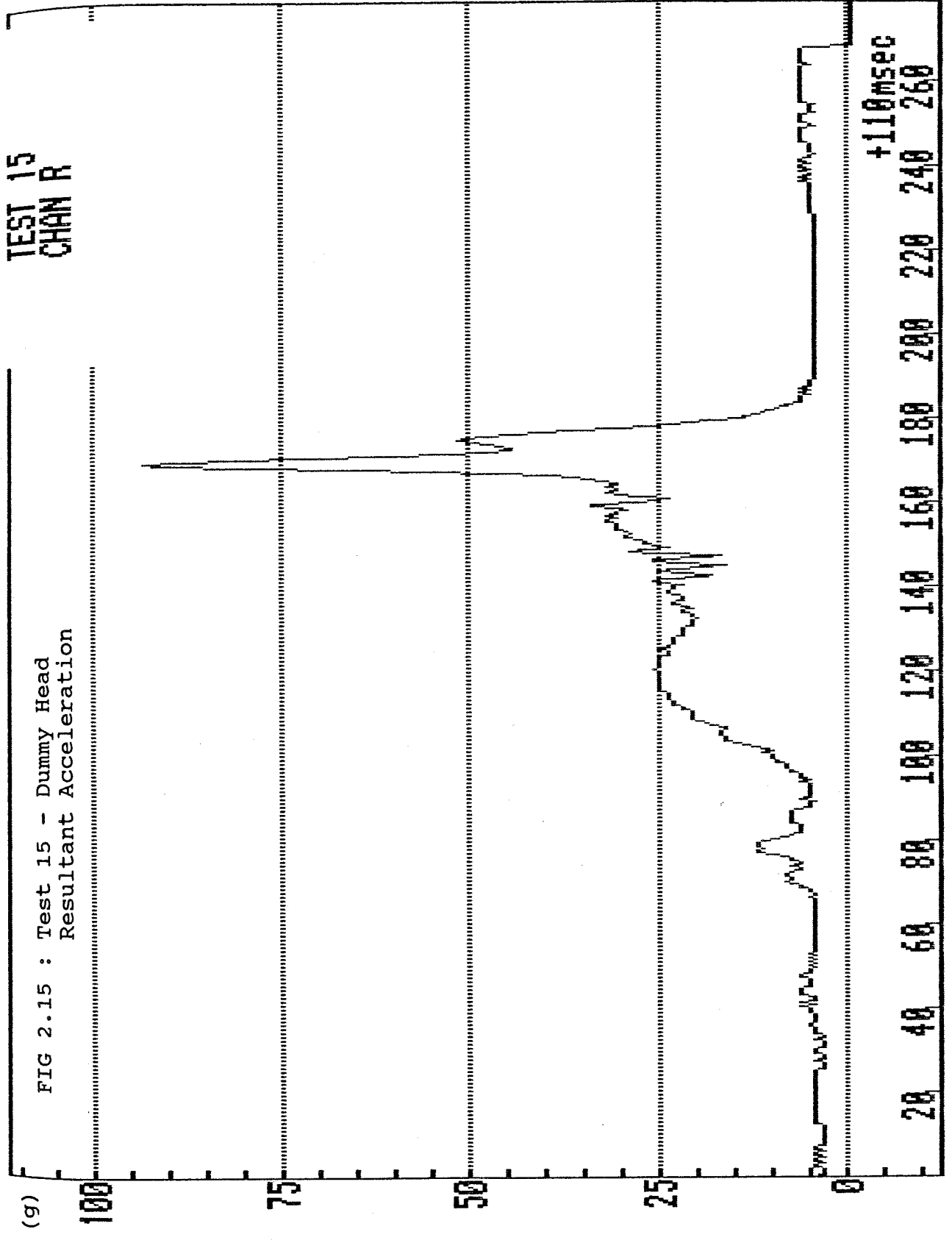
TEST 15
CHAN X

FIG 2.12 : Test 15 - Dummy Head
Longitudinal Acceleration









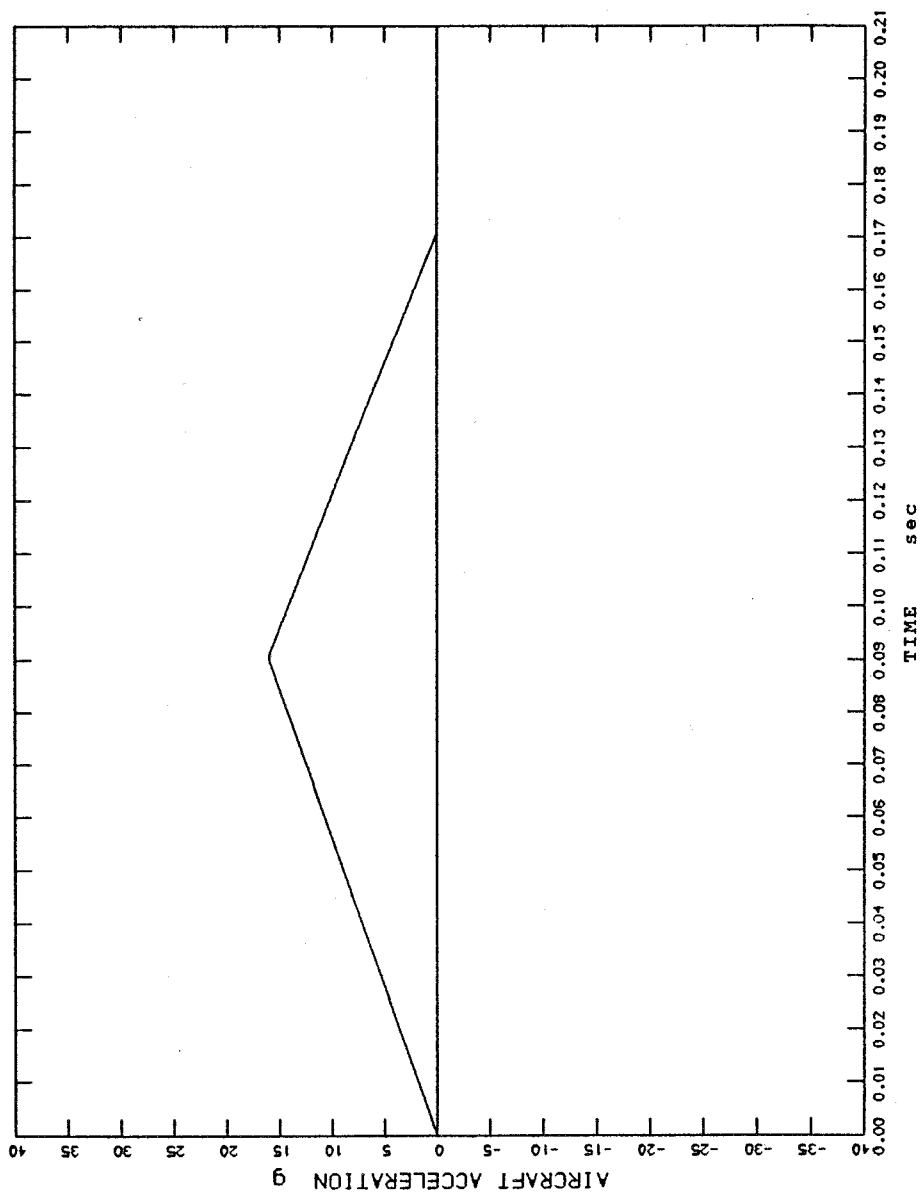


FIGURE 3.1 : Aircraft Acceleration
Horizontal Simulations

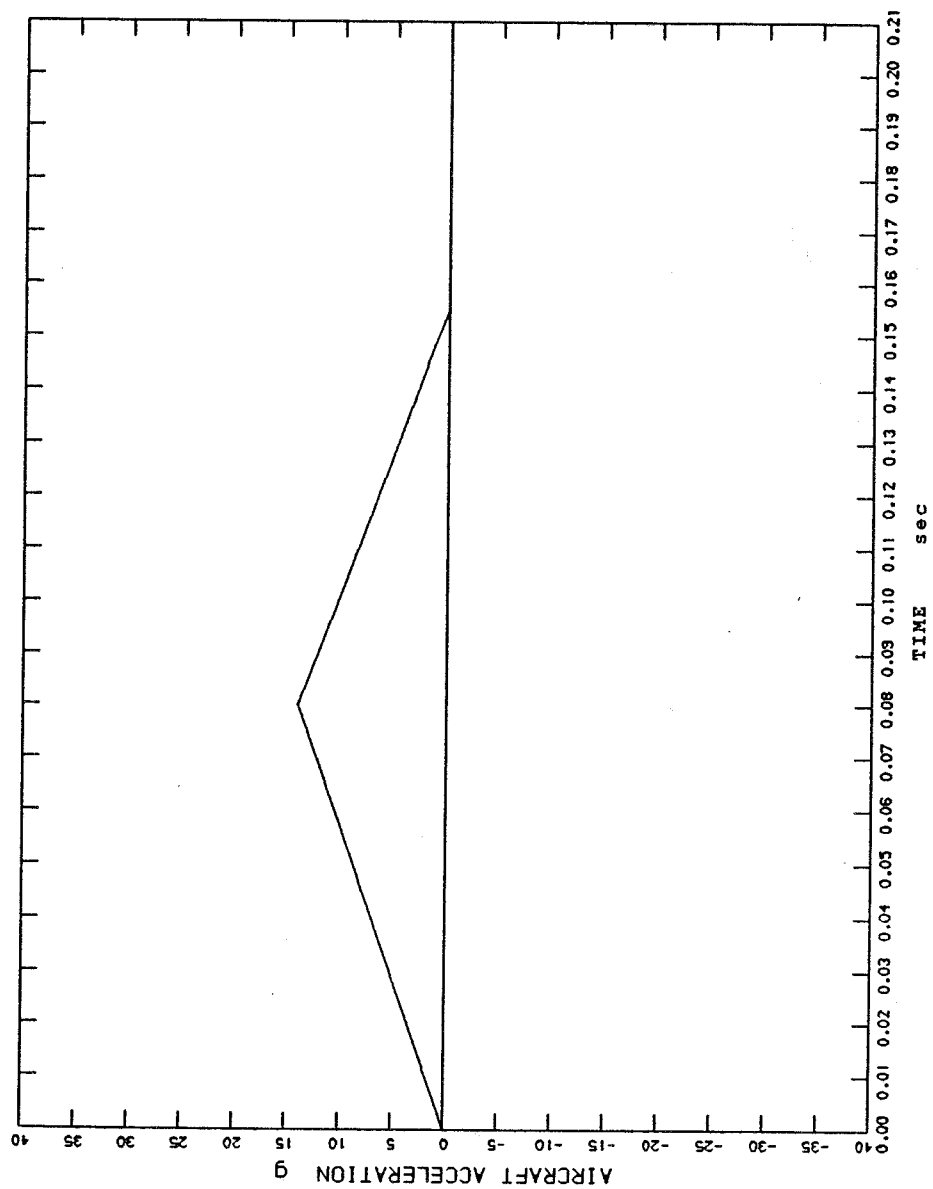
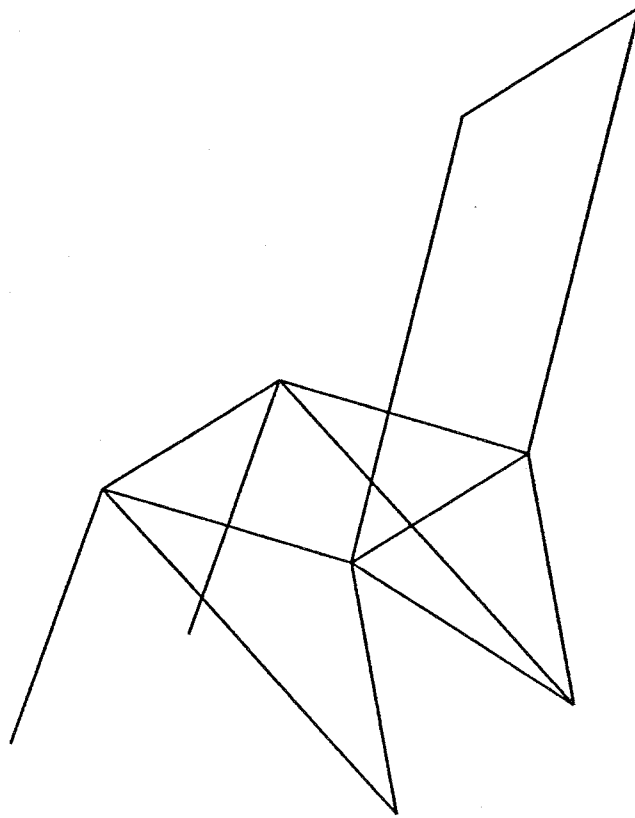


FIGURE 3.2 : Aircraft Acceleration
Vertical Simulations



PROGRAM SOM-LA

FIG 3.3 : Finite Element Seat Model
Stage (A) Simulations

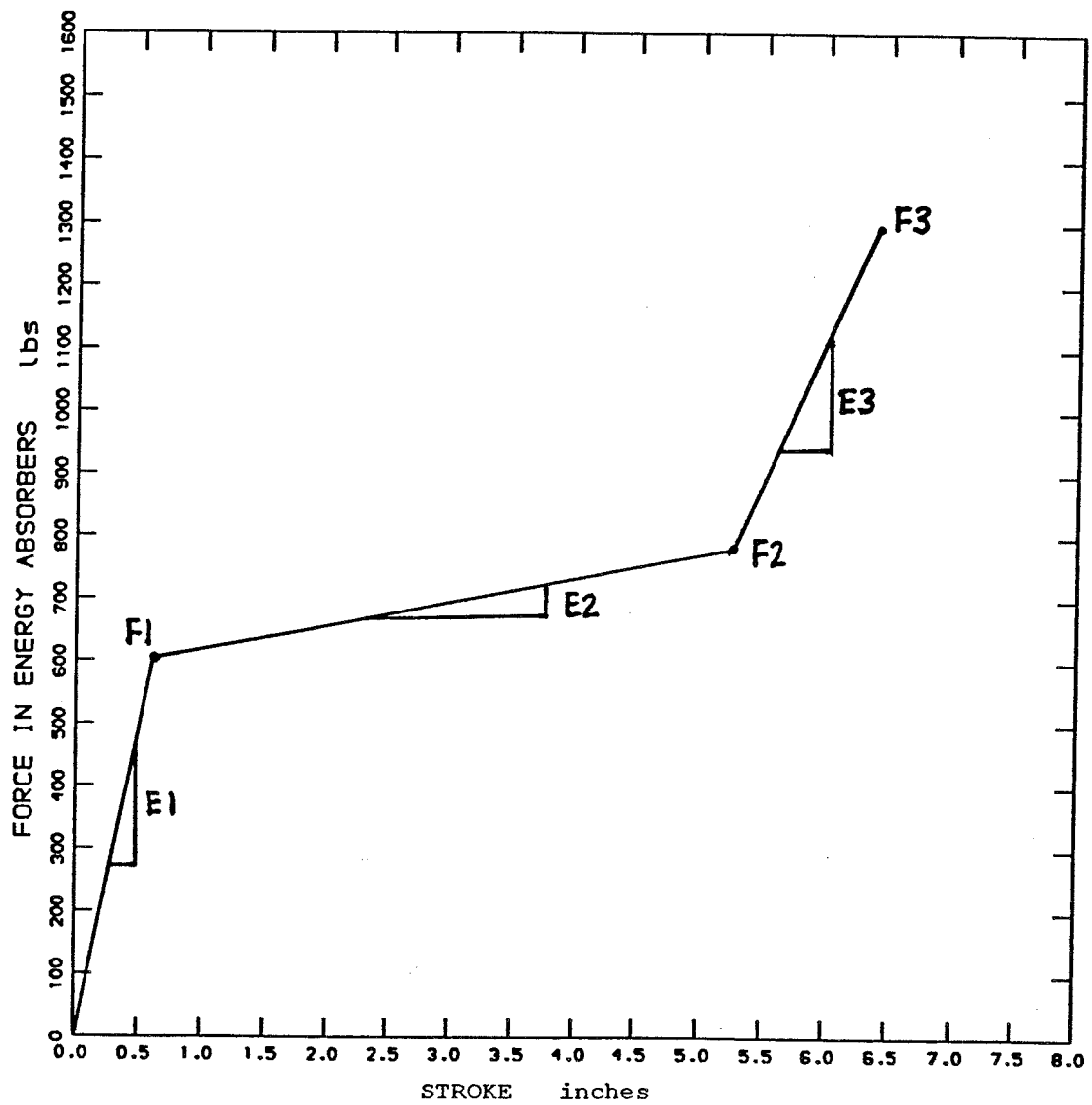
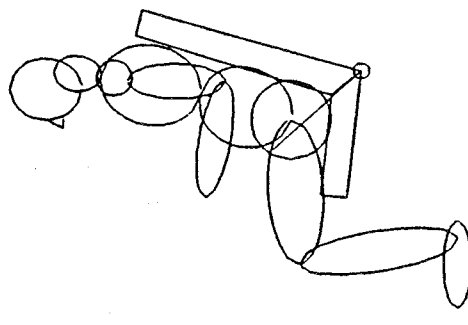
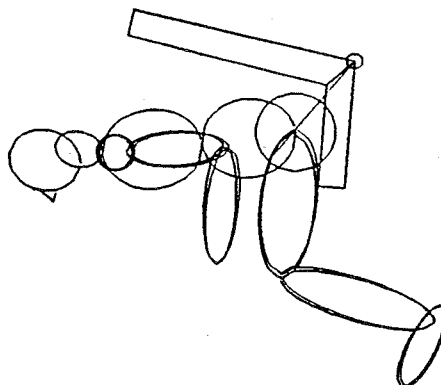


FIG 3.4 : Spring Element Parameters
Force Versus Stroke

PROGRAM SOM-LA
FE / EA SPRING 07 SIMULATION FAA 16G
TIME = 0.0000 SEC.



PROGRAM SOM-LA
FE / EA SPRING 07 SIMULATION FAA 16G
TIME = 0.0900 SEC.



PROGRAM SOM-LA
FE / EA SPRING 07 SIMULATION FAA 16G
TIME = 0.1500 SEC.

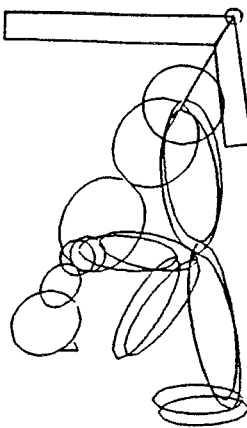
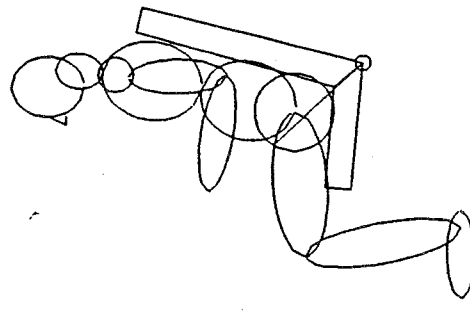
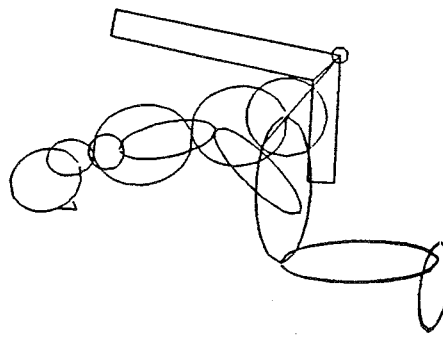


FIG 3.5 : Occupant Kinematic Plots
Horizontal Simulation
'EA2' Seat

PROGRAM SOM-LA
FE / EA SPRING 08V SIMULATION FAA 16G
TIME = 0.0000 SEC.



PROGRAM SOM-LA
FE / EA SPRING 08V SIMULATION FAA 16G
TIME = 0.1200 SEC.



PROGRAM SOM-LA
FE / EA SPRING 08V SIMULATION FAA 16G
TIME = 0.2100 SEC.

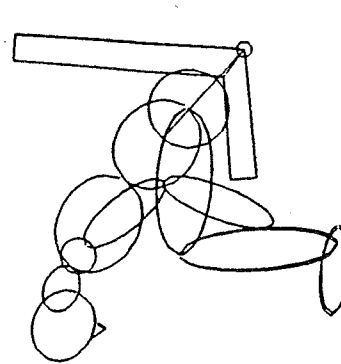
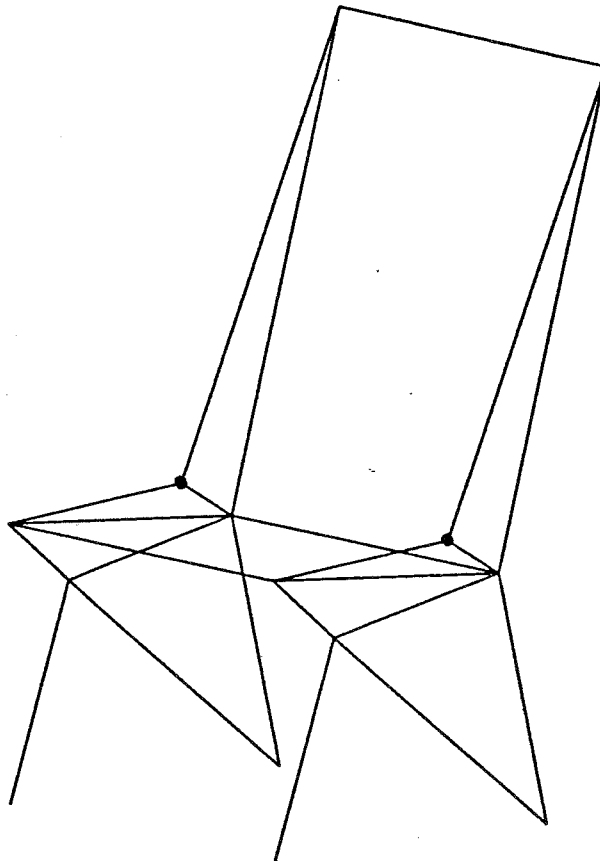


FIG 3.6 : Occupant Kinematic Plots
Vertical Simulation
'EA3' Seat



PROGRAM SOM-LA

FIG 3.7 : Finite Element Seat Model
Stage (B) Simulations



FIGURE 4.1 : Carbon-fibre tubes
Batch 1 - Post-test



FIGURE 4.2 : Carbon-fibre tube
Batch 2 - Post-test



FIGURE 4.3 : Carbon-fibre tubes
Batch 3 - Post-test

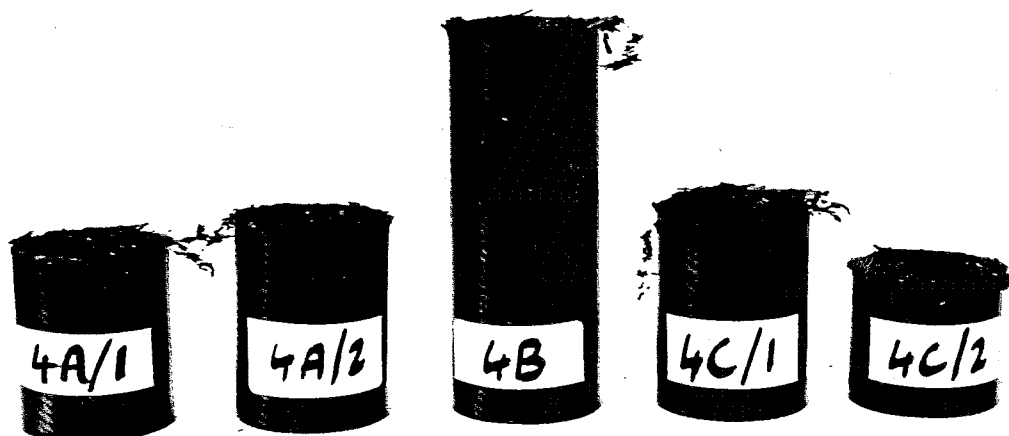


FIGURE 4.4 : Carbon-fibre tubes
Batch 4 - Post-test

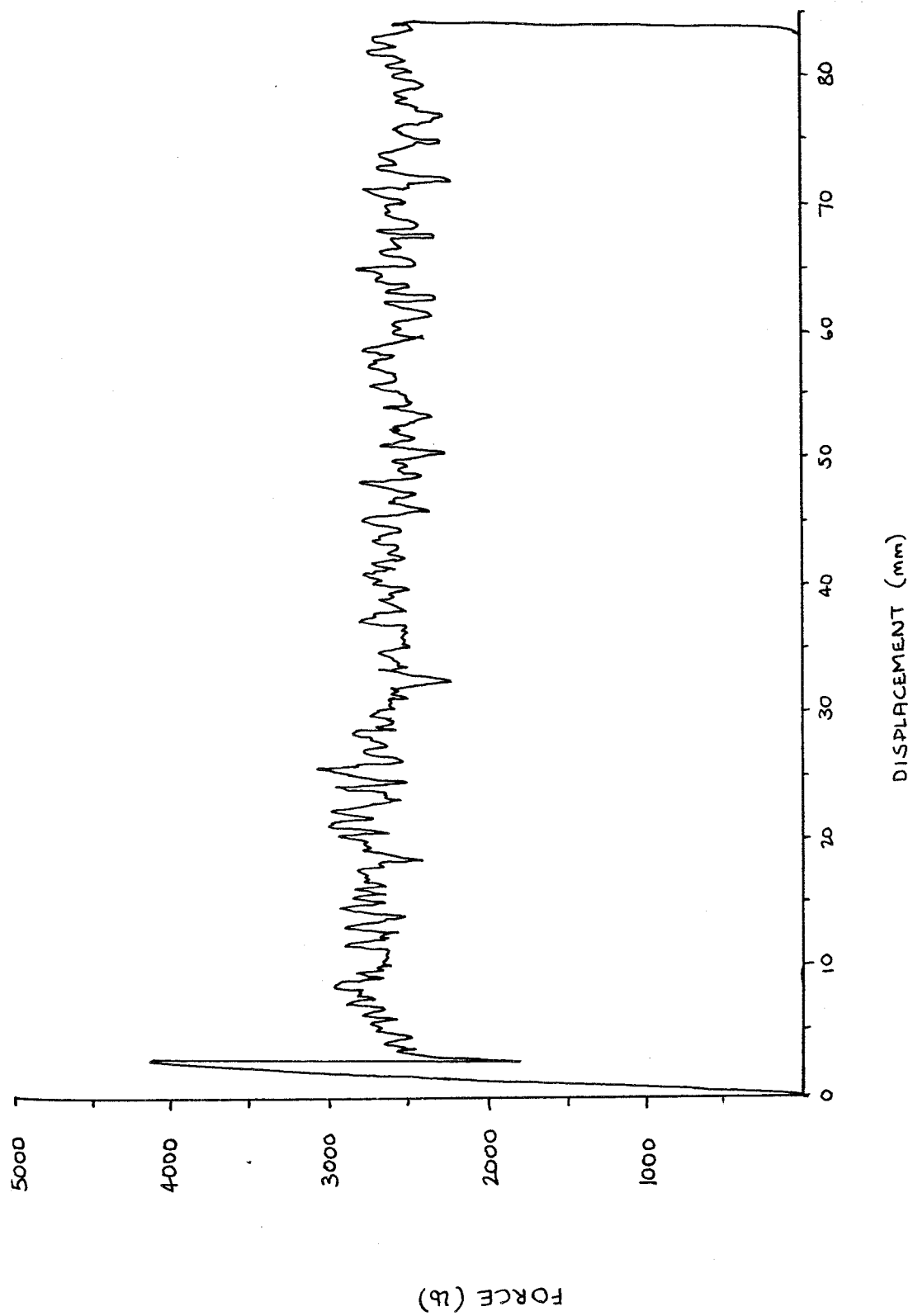


FIGURE 4.5 : Static Compression Test
Results - Tube 4A/2

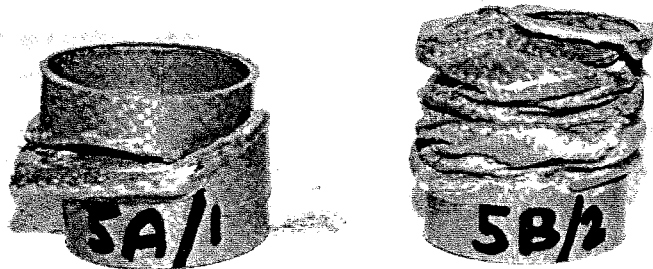


FIGURE 4.6 : Kevlar tubes
Batch 5 - Post-test

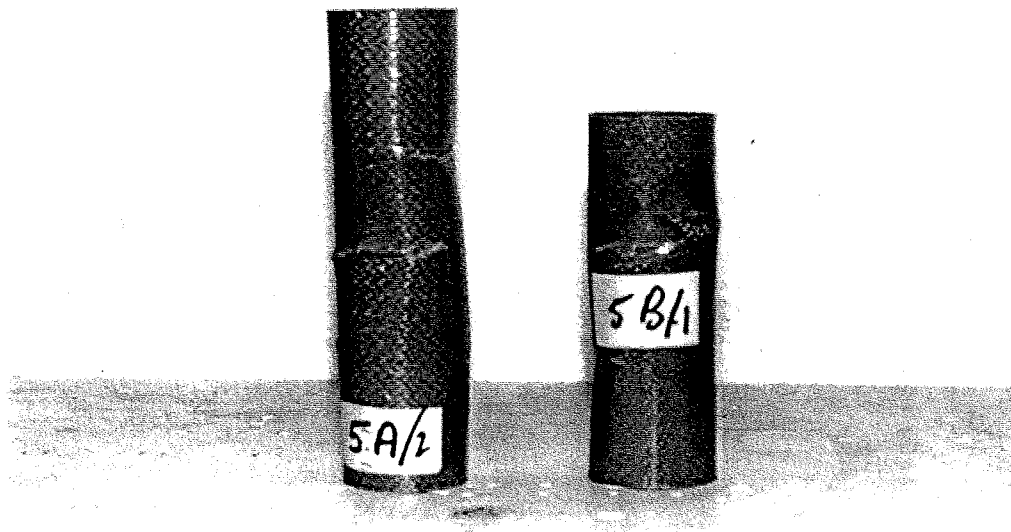


FIGURE 4.7 : Kevlar tubes
Batch 5 - Post-test

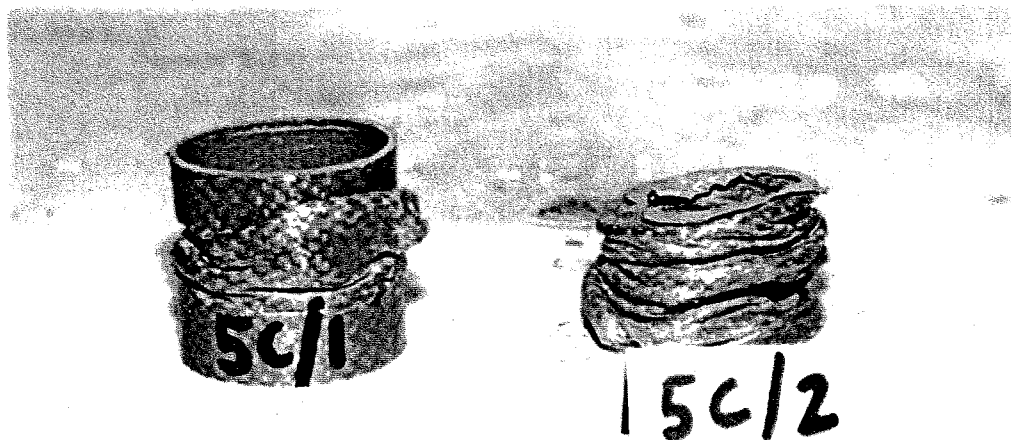


FIGURE 4.8 : Kevlar tubes
Batch 5 - Post-test

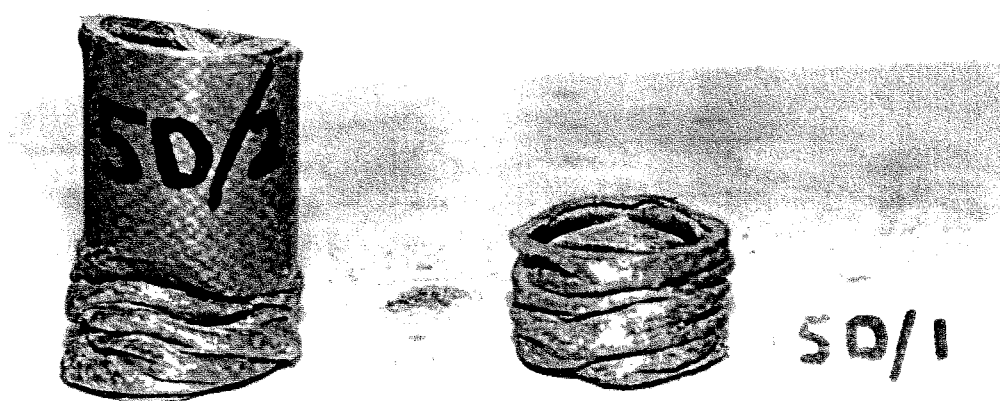


FIGURE 4.9 : Kevlar tubes
Batch 5 - Post-test



FIGURE 4.10 : Kevlar tubes
Batch 6 - Post-test



FIGURE 4.11 : Kevlar tubes
Batch 6 - Post-test

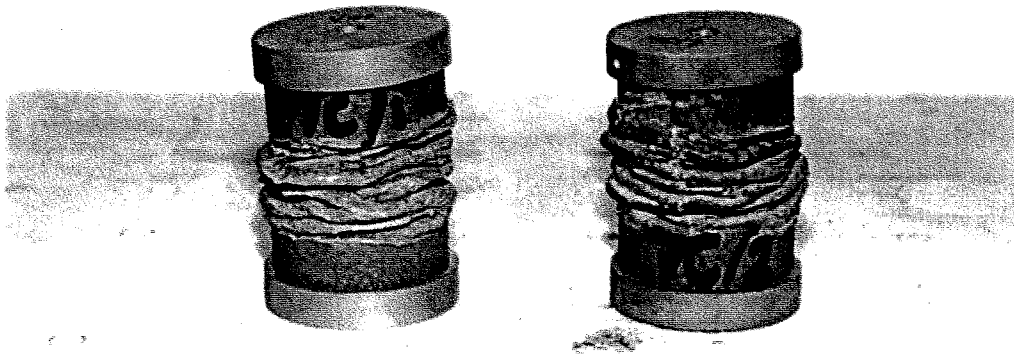


FIGURE 4.12 : Kevlar tubes
Batch 7 - Post-test

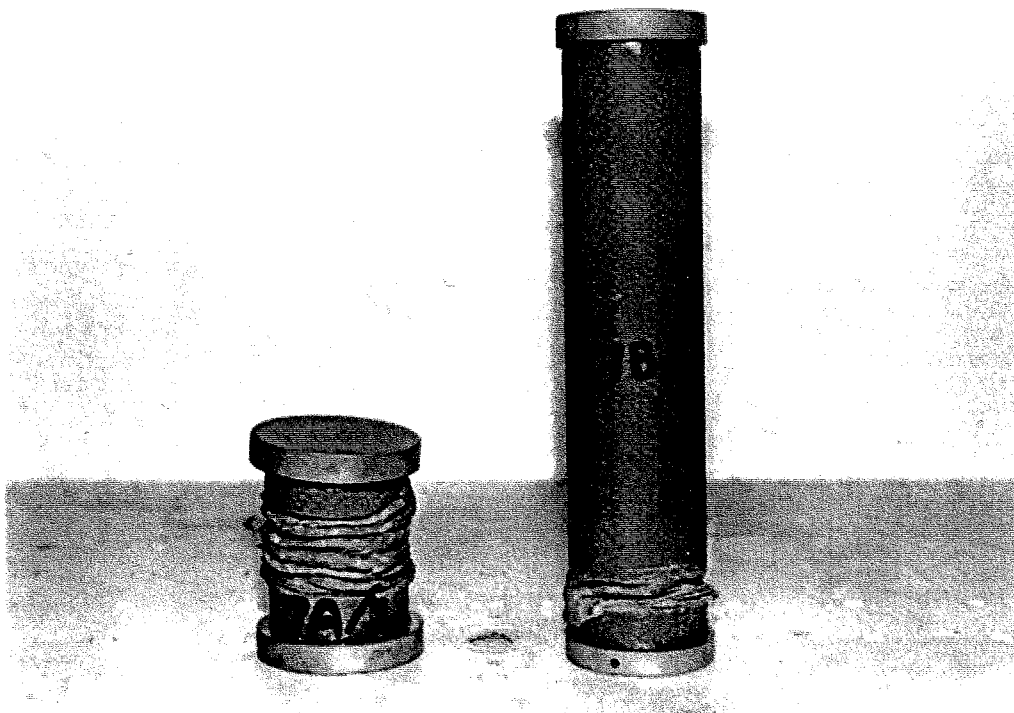


FIGURE 4.13 : Kevlar tubes
Batch 7 - Post-test

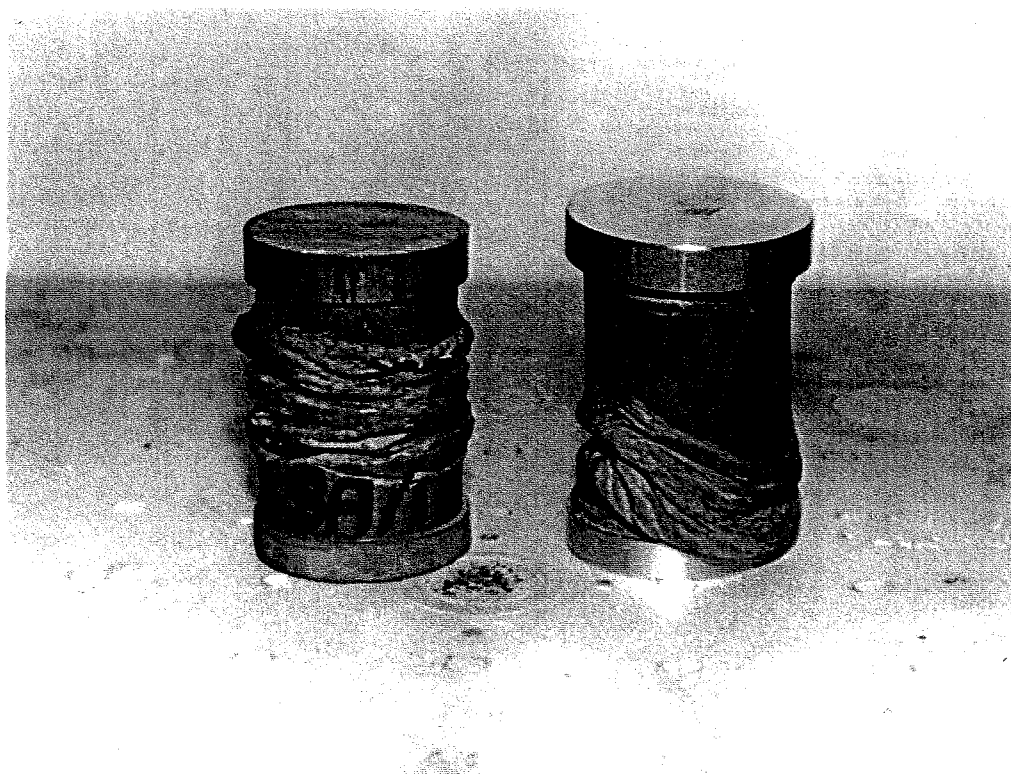


FIGURE 4.14 : Kevlar tubes
Batch 8 - Post-test



FIGURE 4.15 : Kevlar tubes
Batch 8 - Post-test

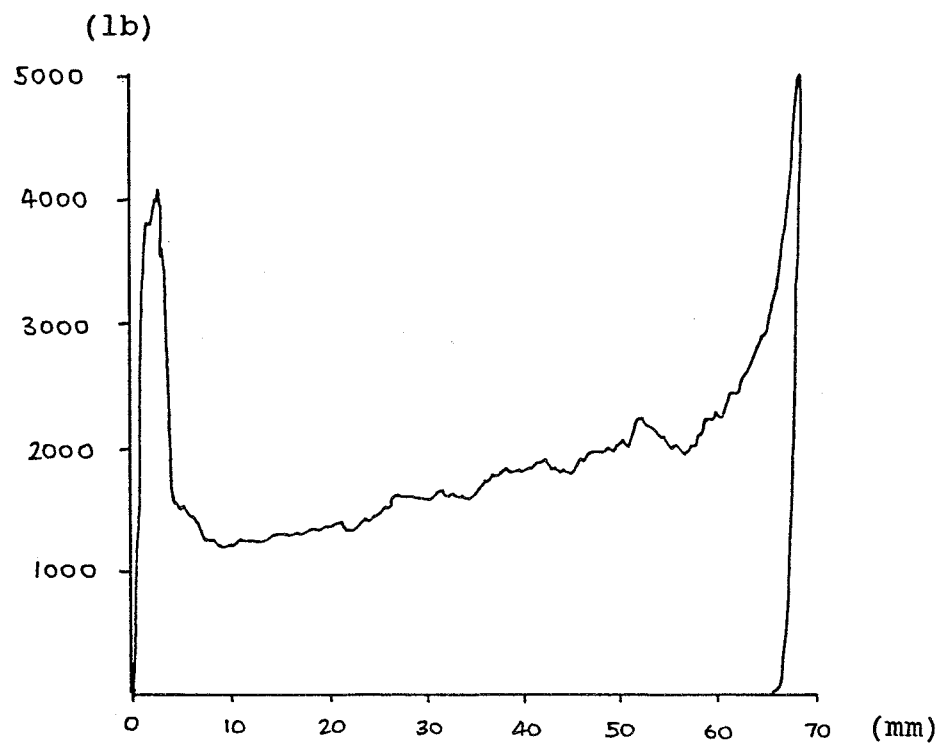


FIGURE 4.16 : STATIC COMPRESION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 5A/1

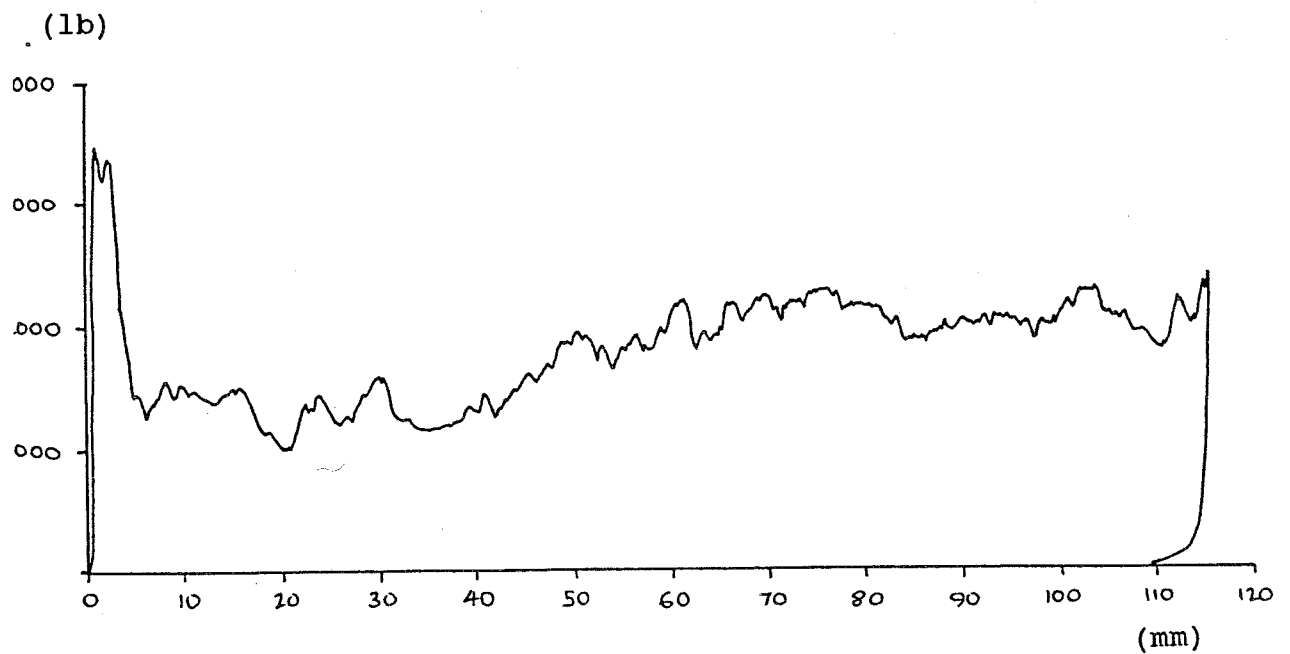


FIGURE 4.17 : STATIC COMPRESION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 5B/2

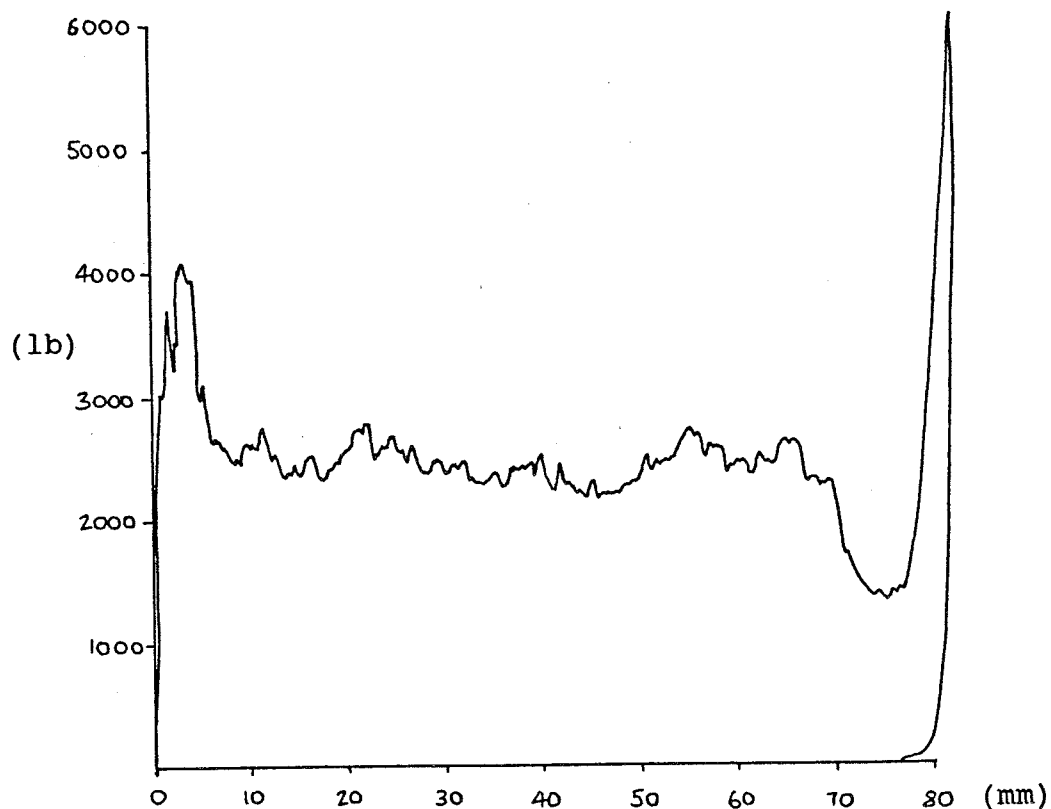


FIGURE 4.20 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 5D/1

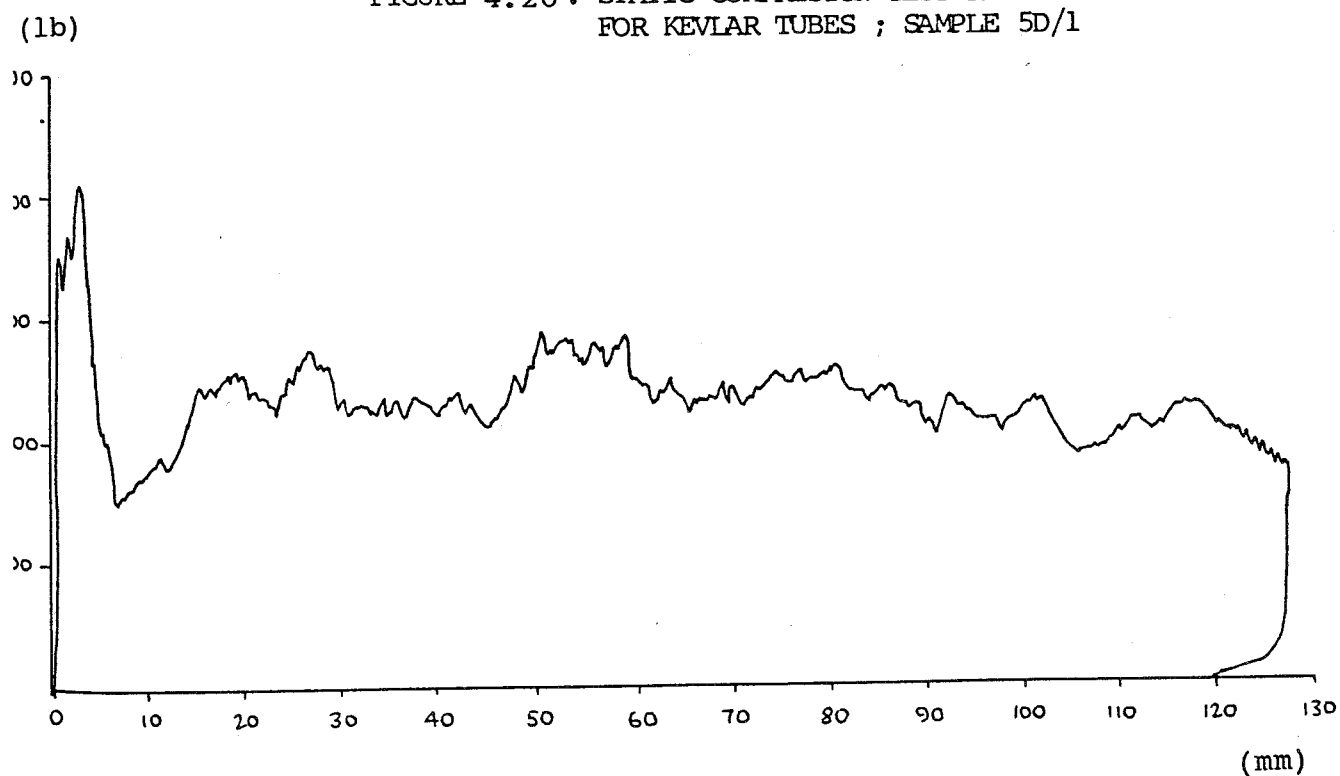


FIGURE 4.21 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 5D/2

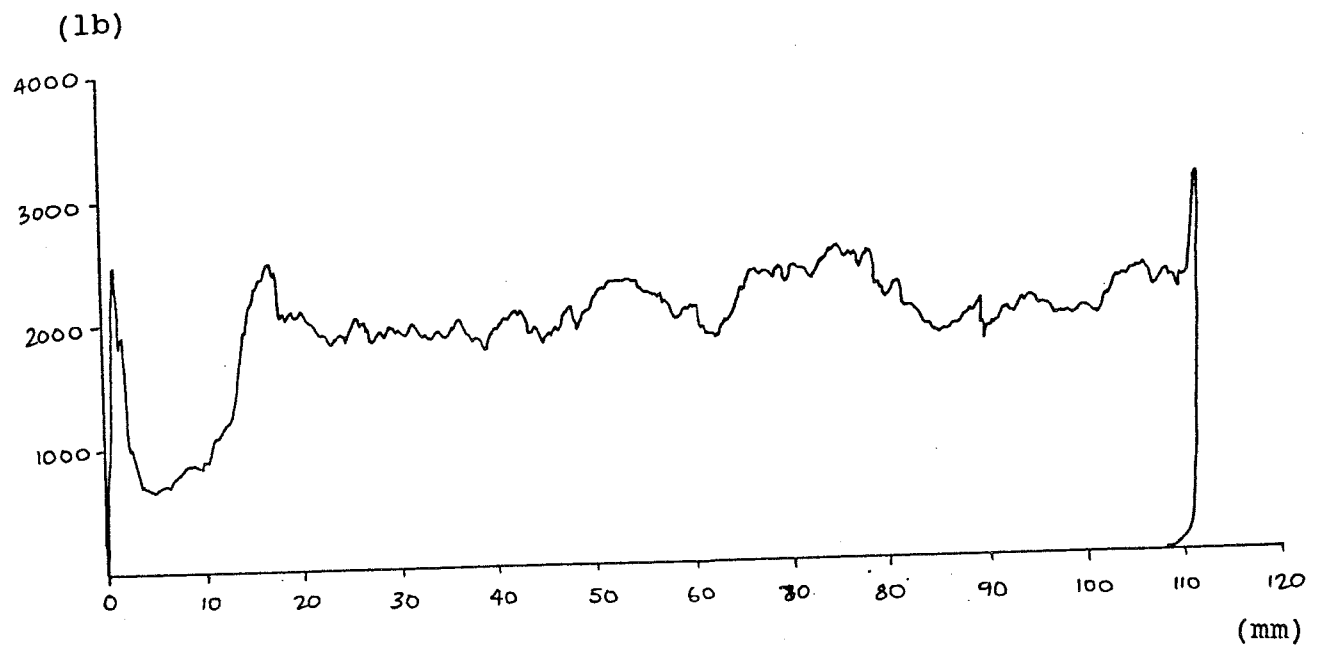


FIGURE 4.22 : STATIC COMPRESION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6A/1

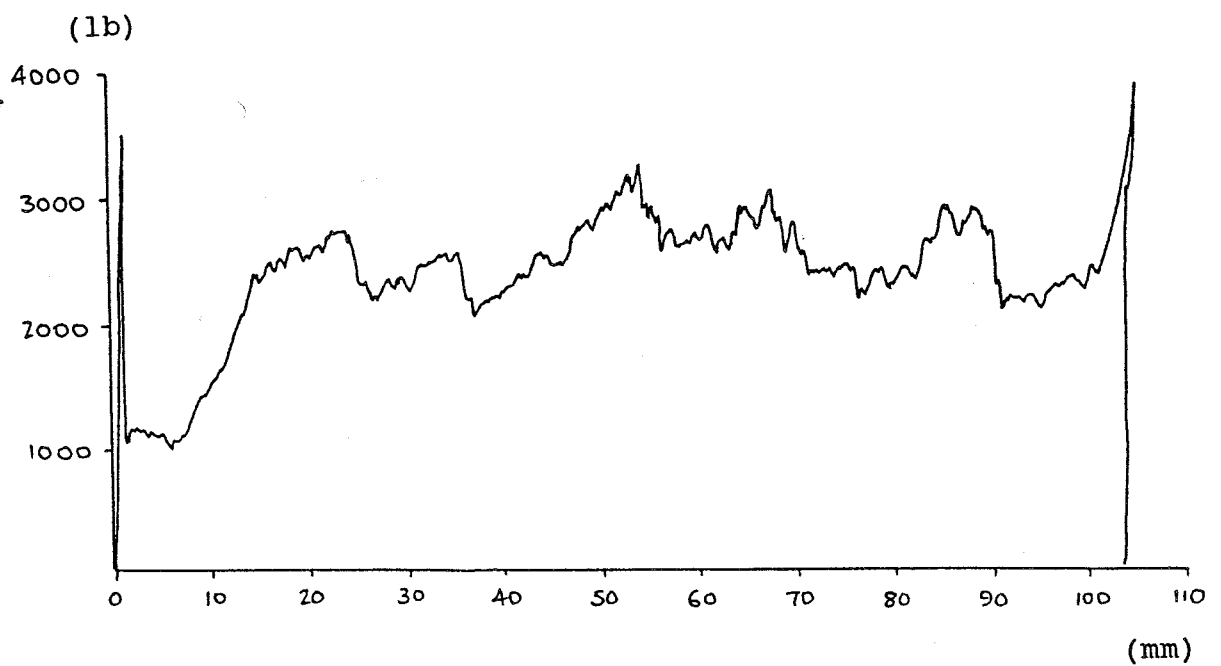


FIGURE 4.23 : STATIC COMPRESION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6A/2

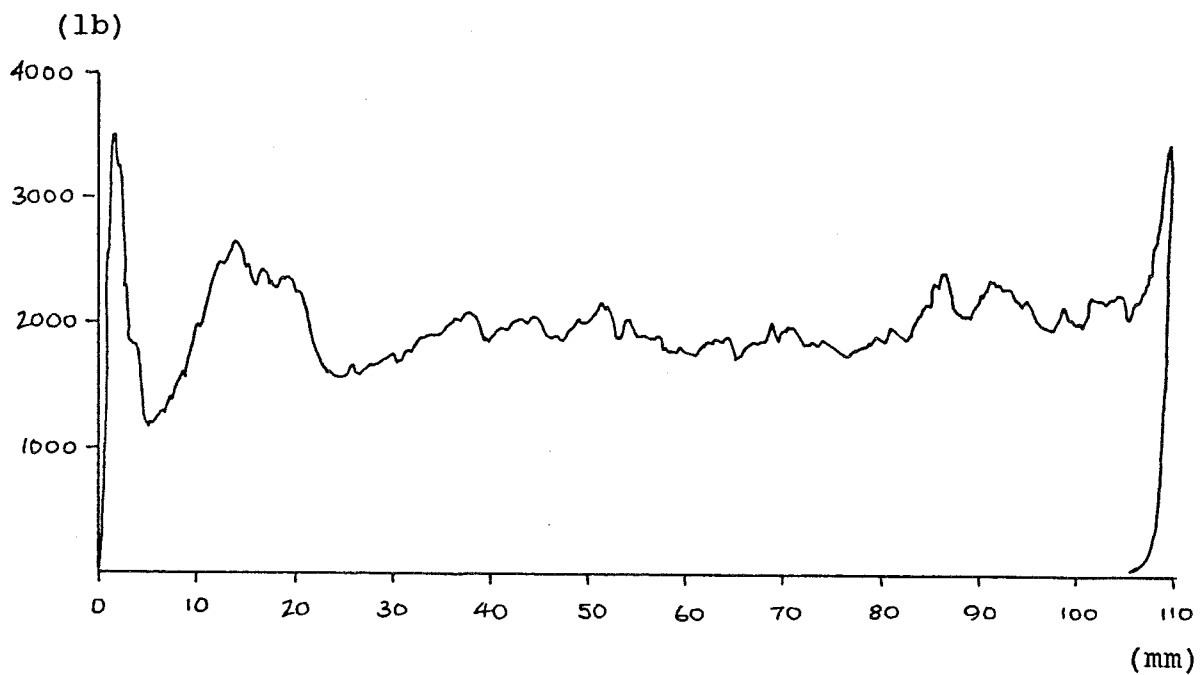


FIGURE 4.24 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6B/1

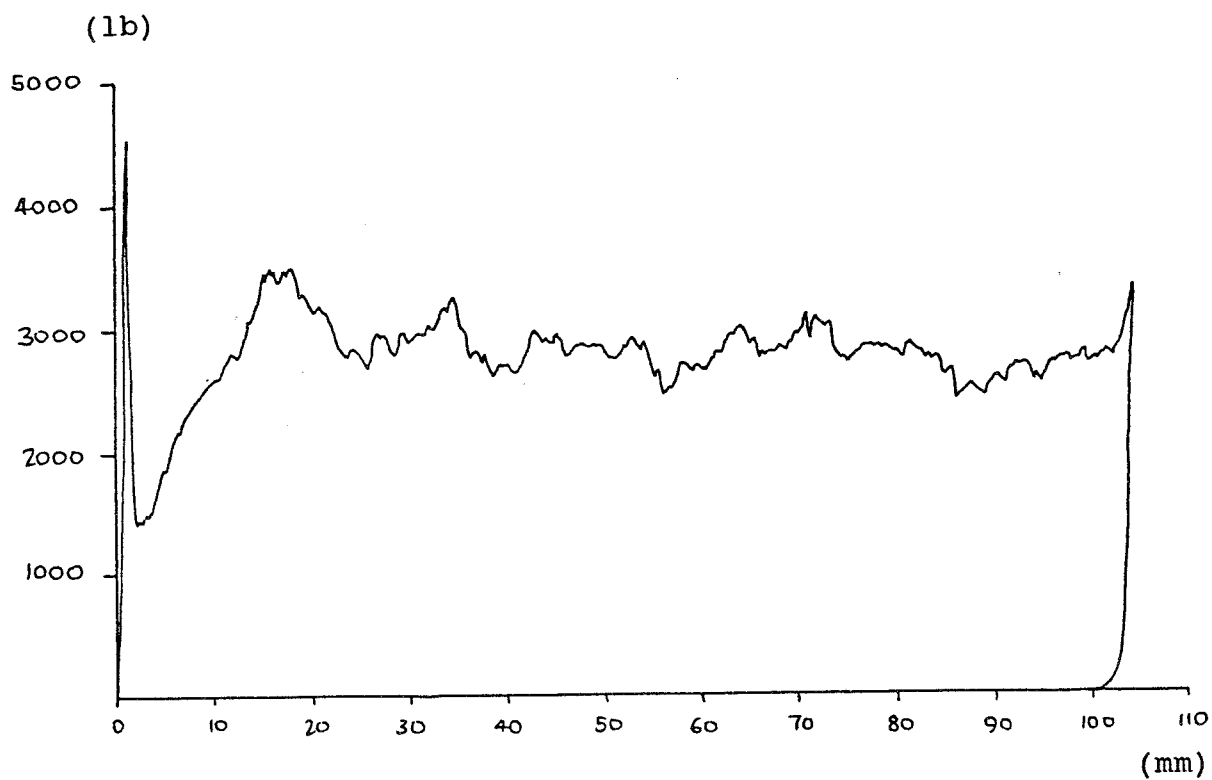


FIGURE 4.25 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6F/1

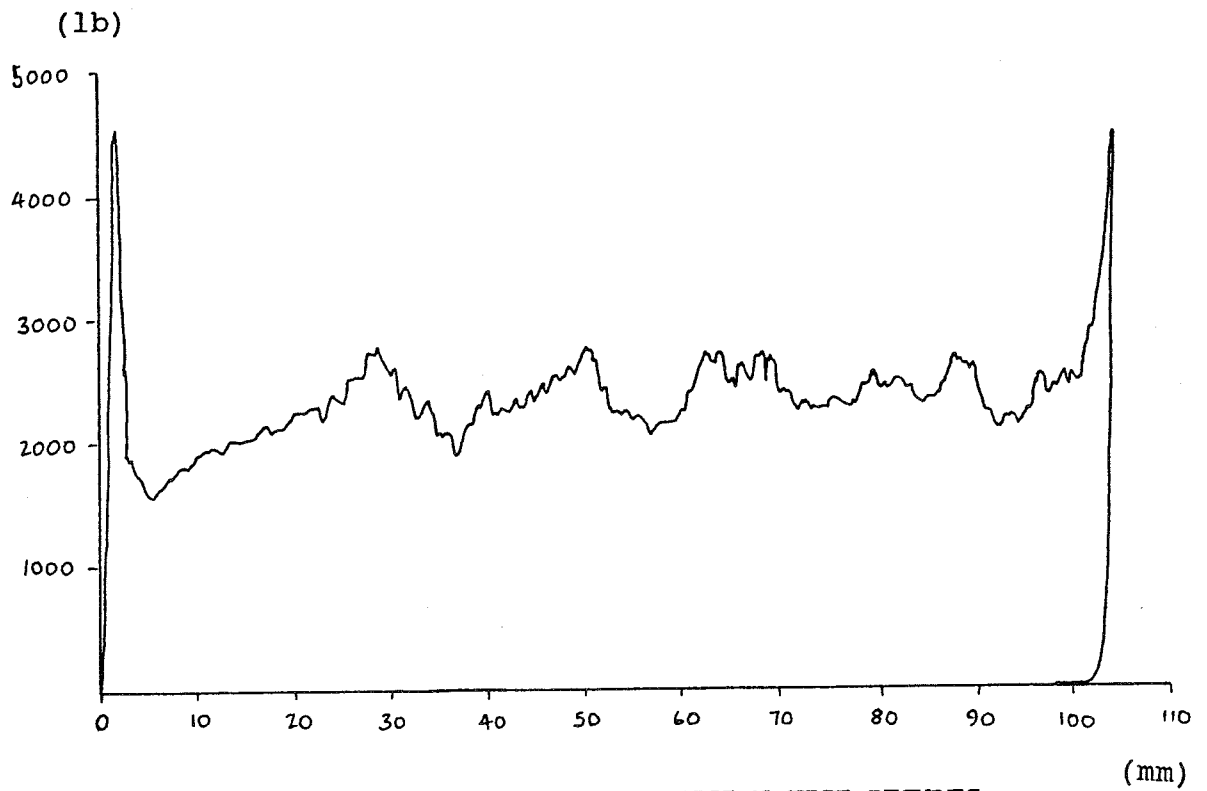


FIGURE 4.26 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6D/1

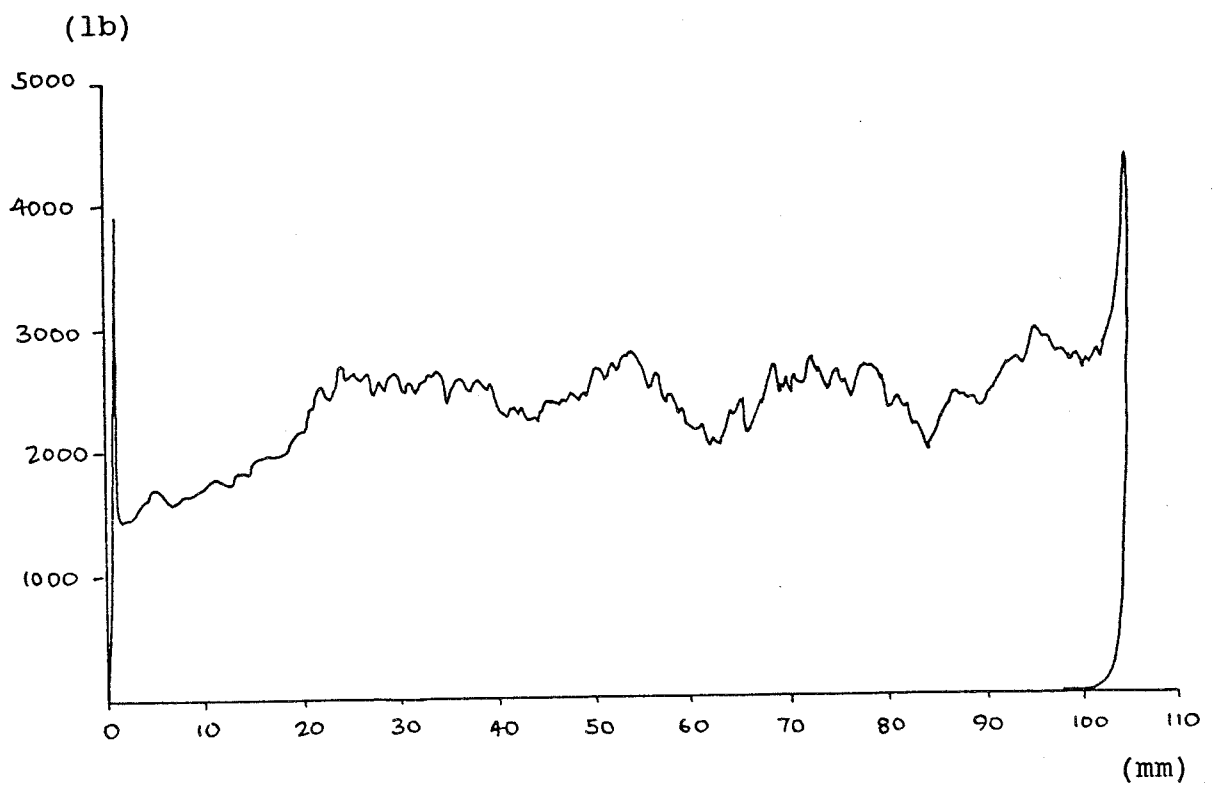


FIGURE 4.27 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 6D/2

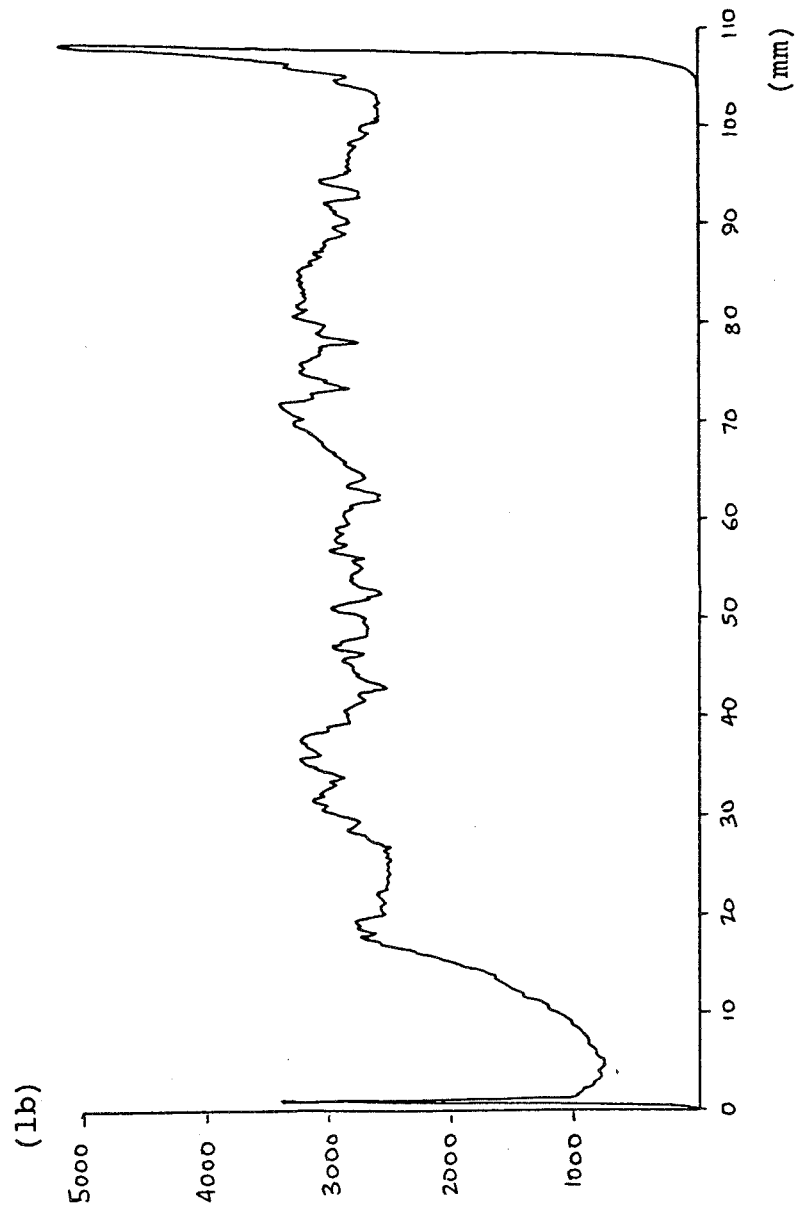


FIGURE 4.28 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 7A/2

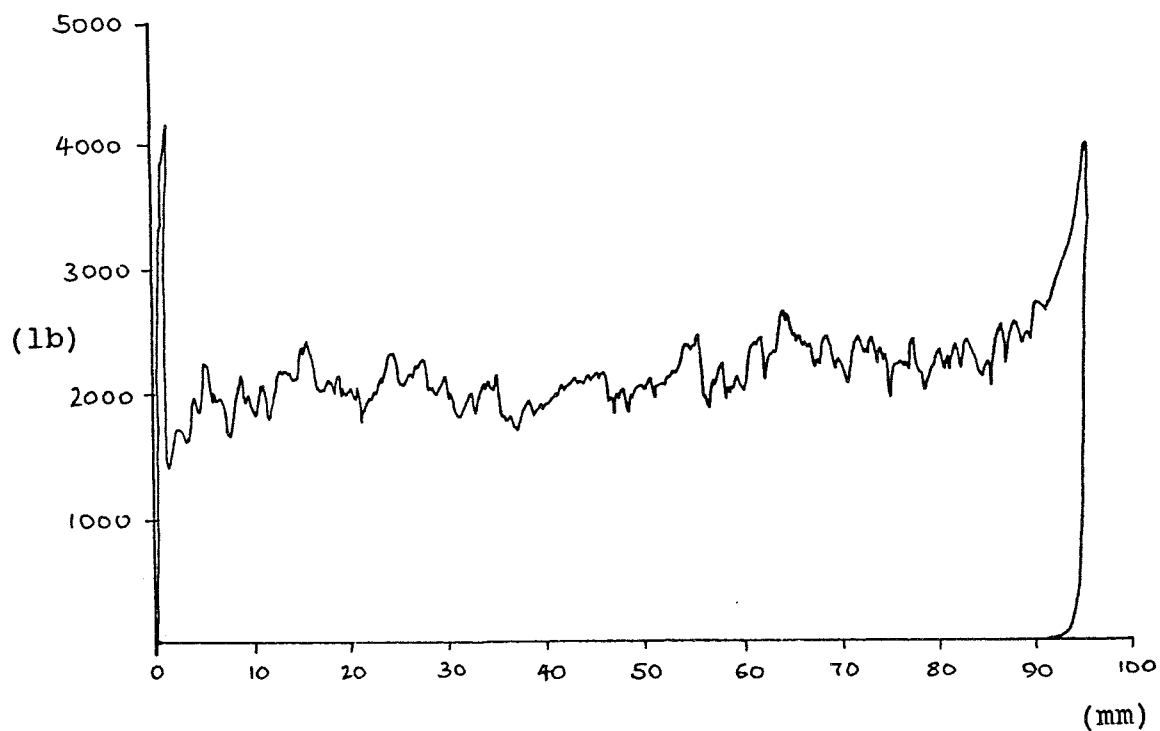


FIGURE 4.29 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 7C/1

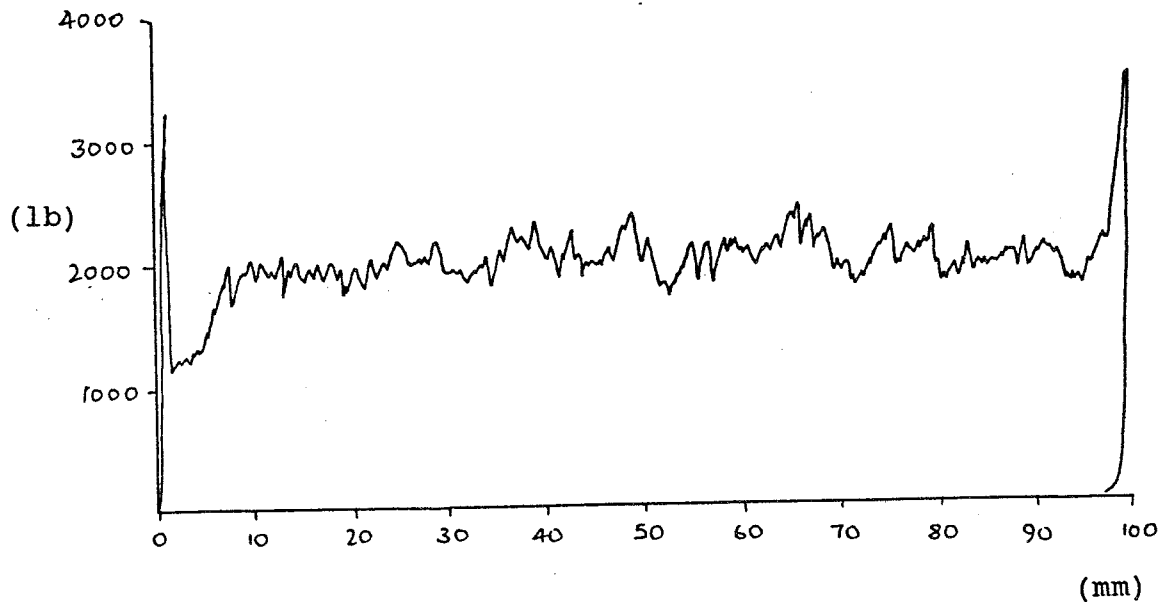


FIGURE 4.30 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 7C/2

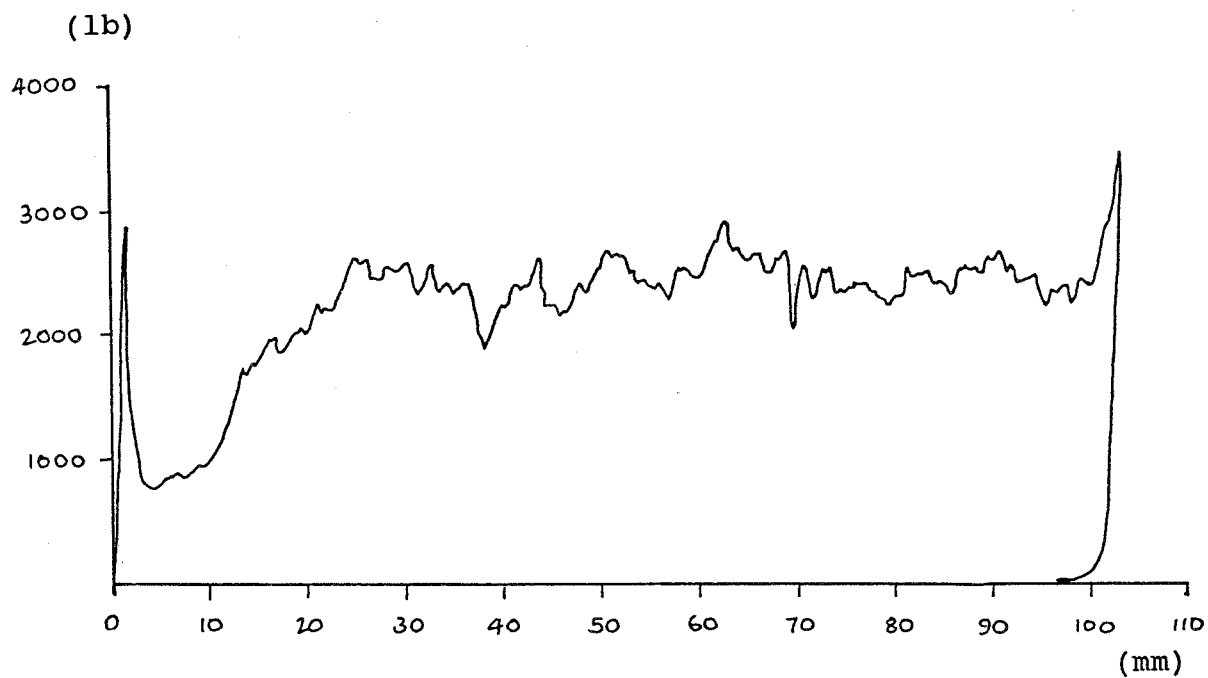


FIGURE 4.31 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 8A/1

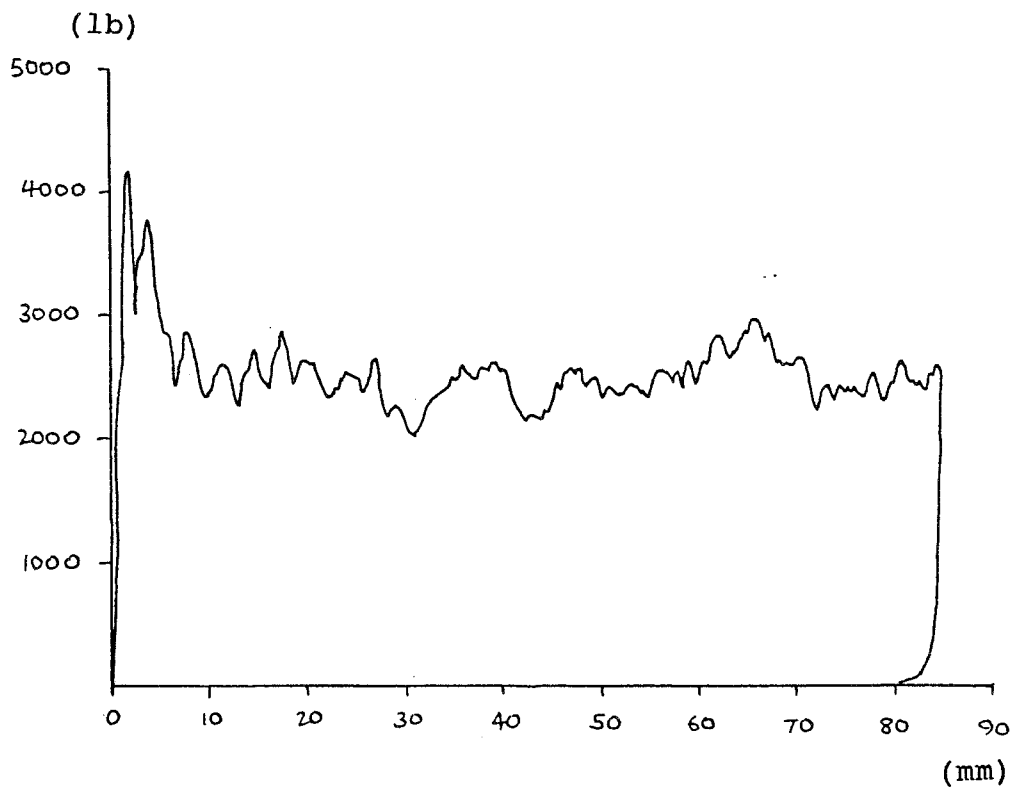


FIGURE 4.32. : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 8A/2

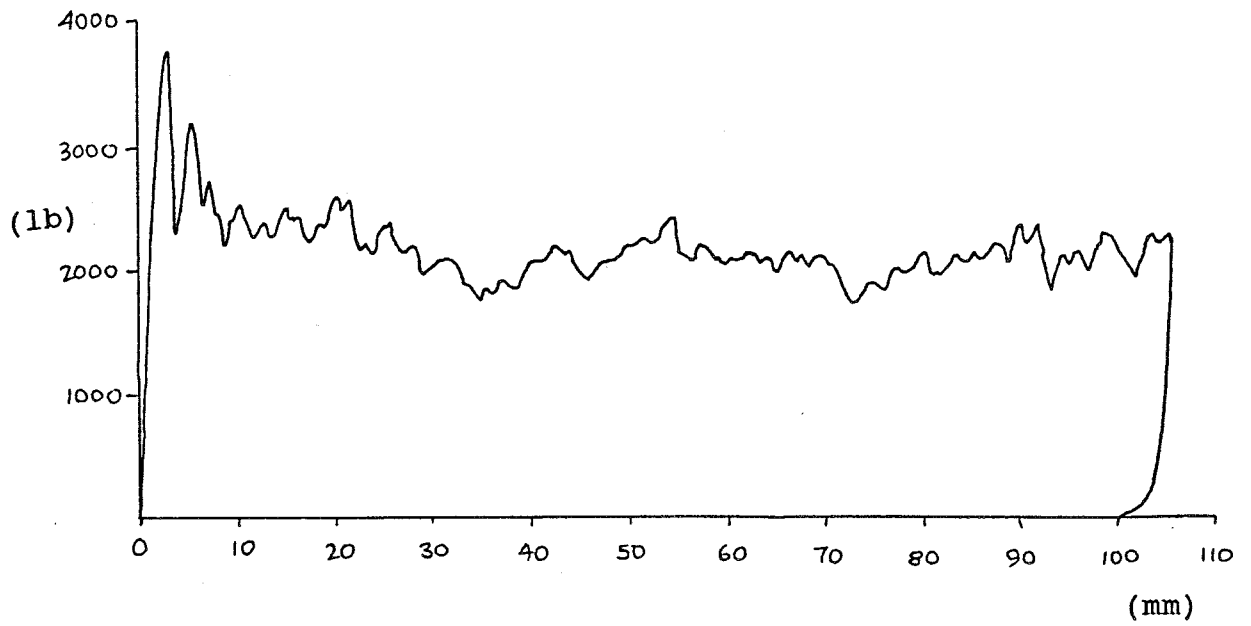


FIGURE 4.33 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 8B/2

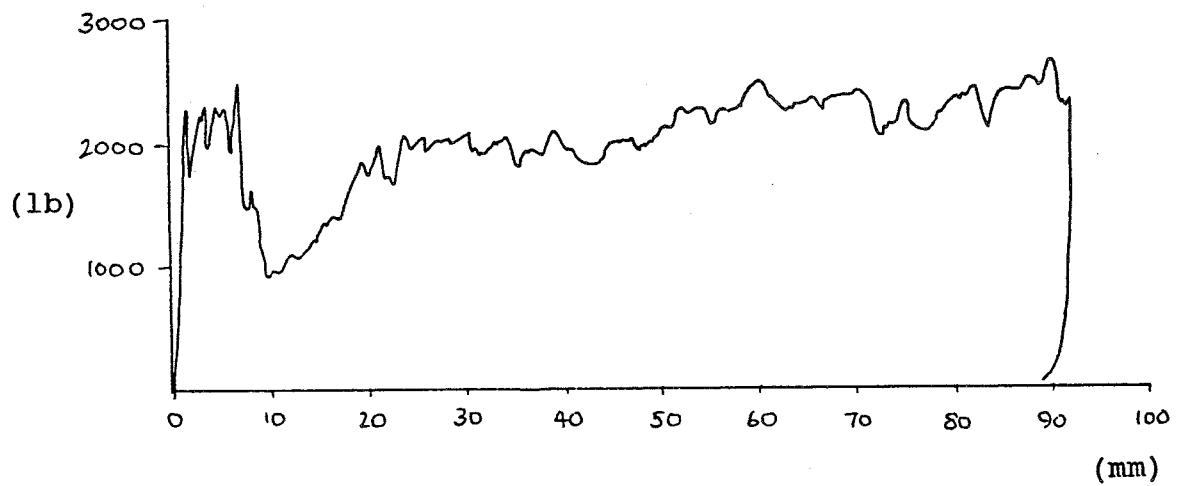


FIGURE 4.34 : STATIC COMPRESSION TEST RESULTS
FOR KEVLAR TUBES ; SAMPLE 8B/1

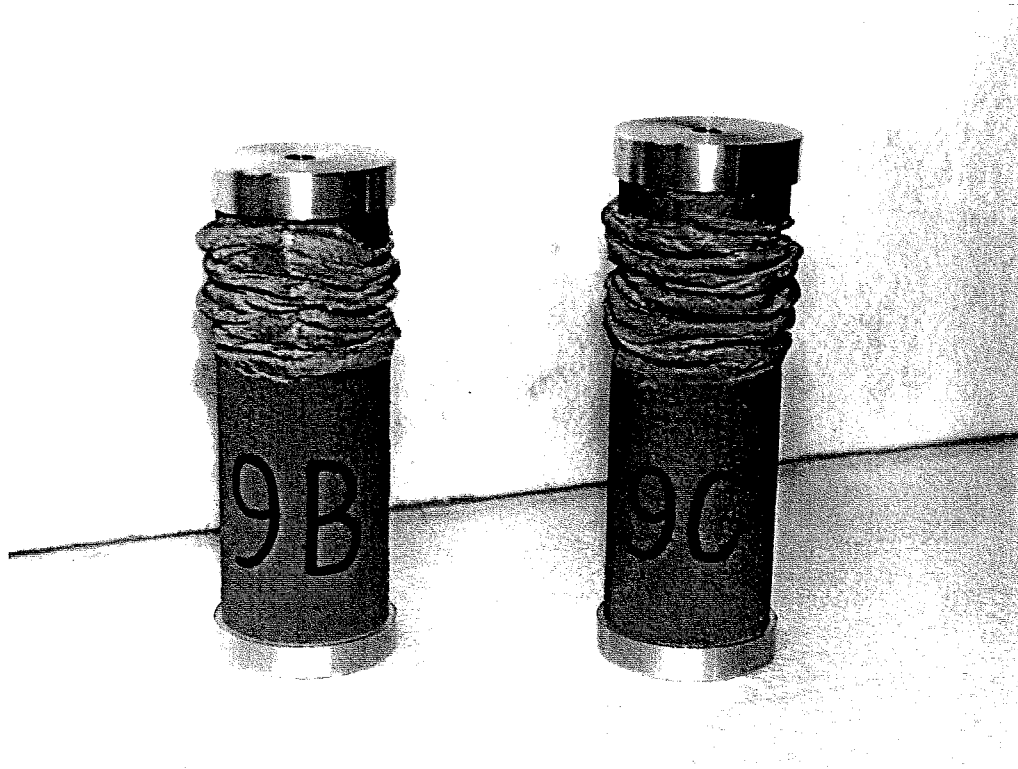


FIGURE 5.1 : Energy Absorbers
After Static Testing

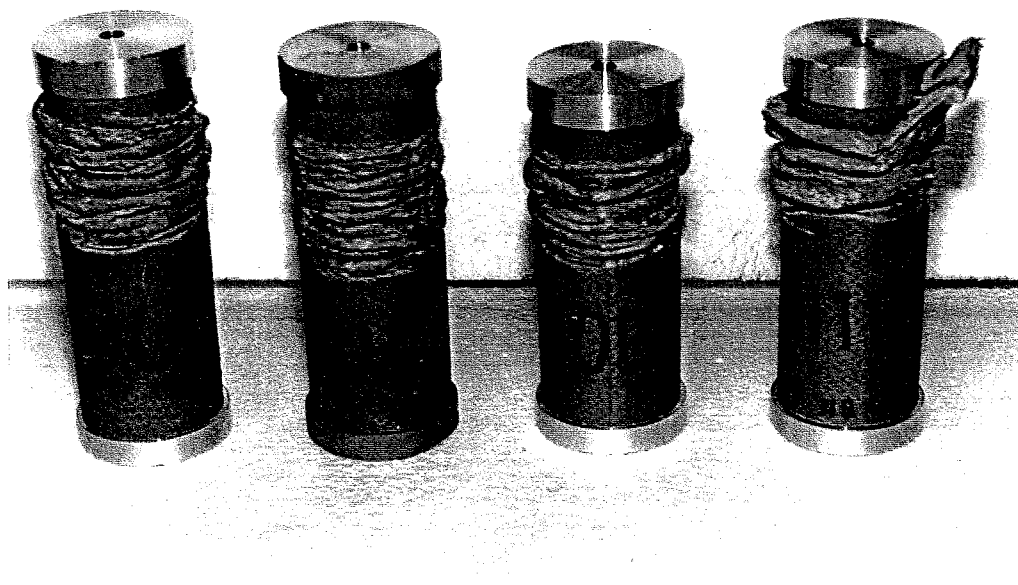


FIGURE 5.2 : Energy Absorbers
After Static Testing

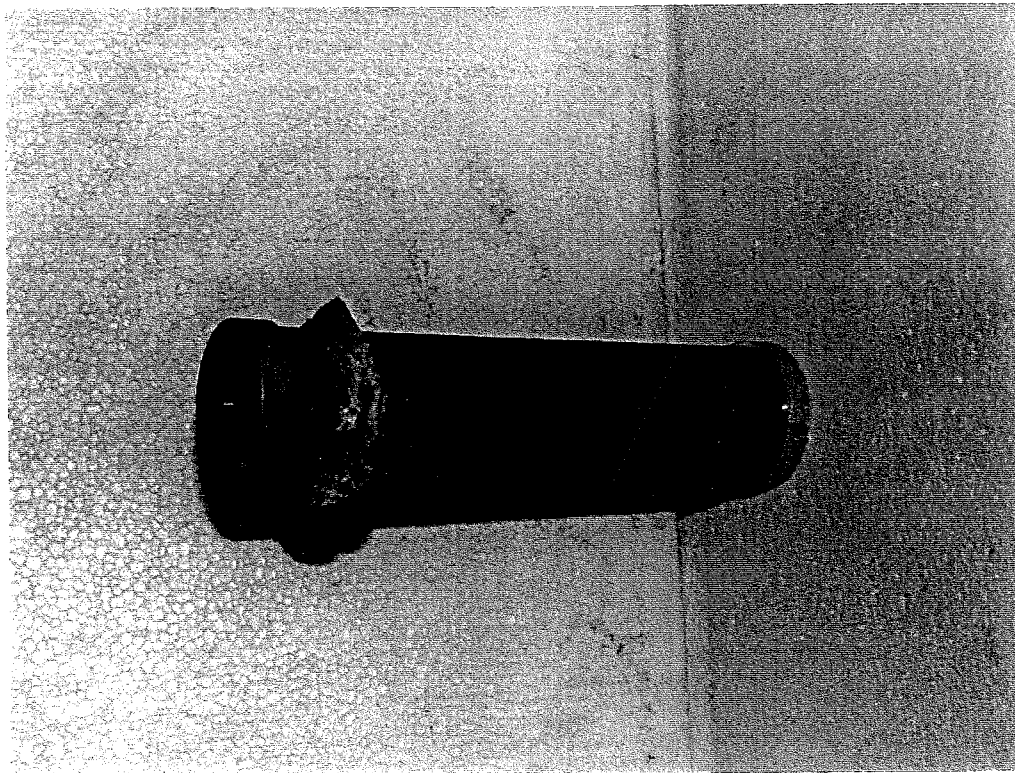


FIGURE 5.3 : Energy Absorber 10C
After Dynamic Testing

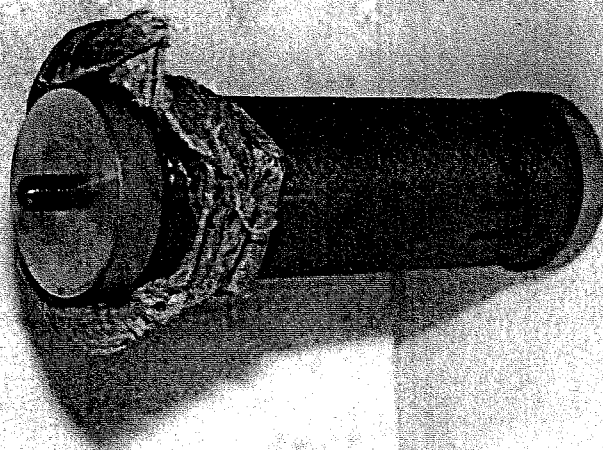


FIGURE 5.4 : Energy Absorber 11C
After Dynamic Testing



FIGURE 5.5 : Energy Absorber 12A
After Dynamic Testing

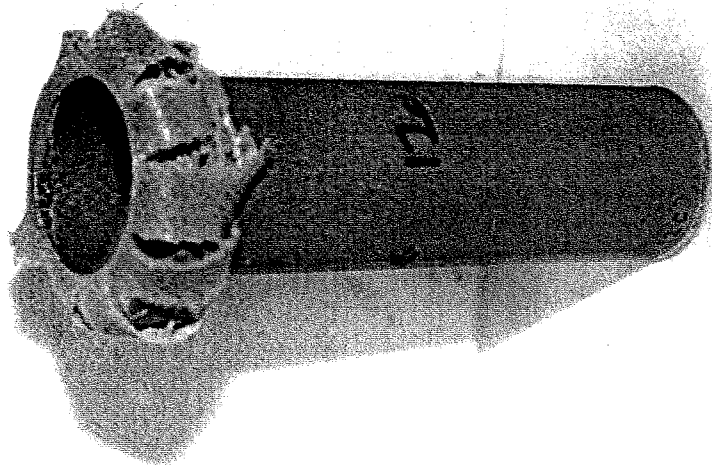


FIGURE 5.6 : Energy Absorber 12C
After Dynamic Testing

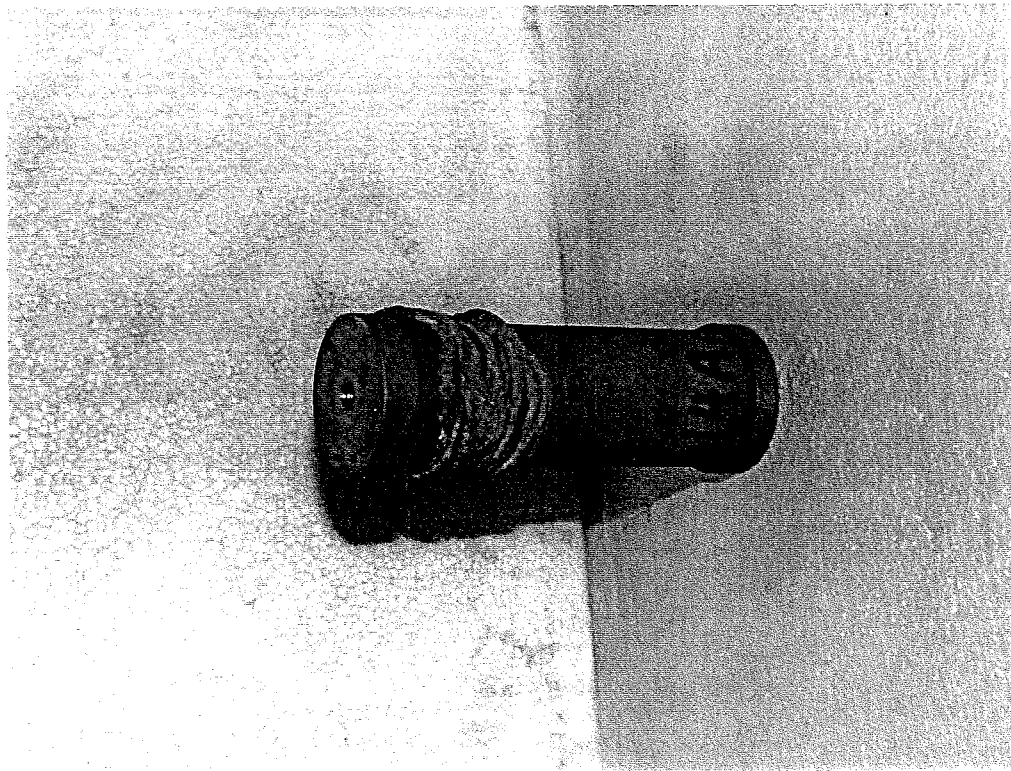


FIGURE 5.7 : Energy Absorber 14A
After Dynamic Testing



FIGURE 5.8 : Energy Absorber 14C
After Dynamic Testing



FIGURE 5.9 : Energy Absorber 14B
After Dynamic Testing



FIGURE 5.10 : Energy Absorber 15A
After Dynamic Testing

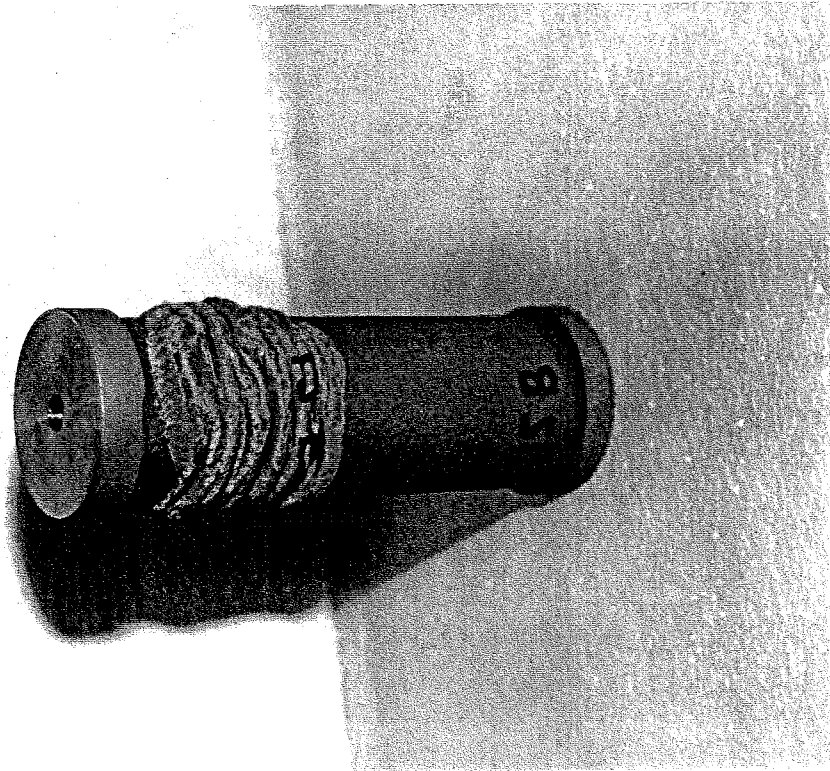


FIGURE 5.11 : Energy Absorber 15B
After Dynamic Testing

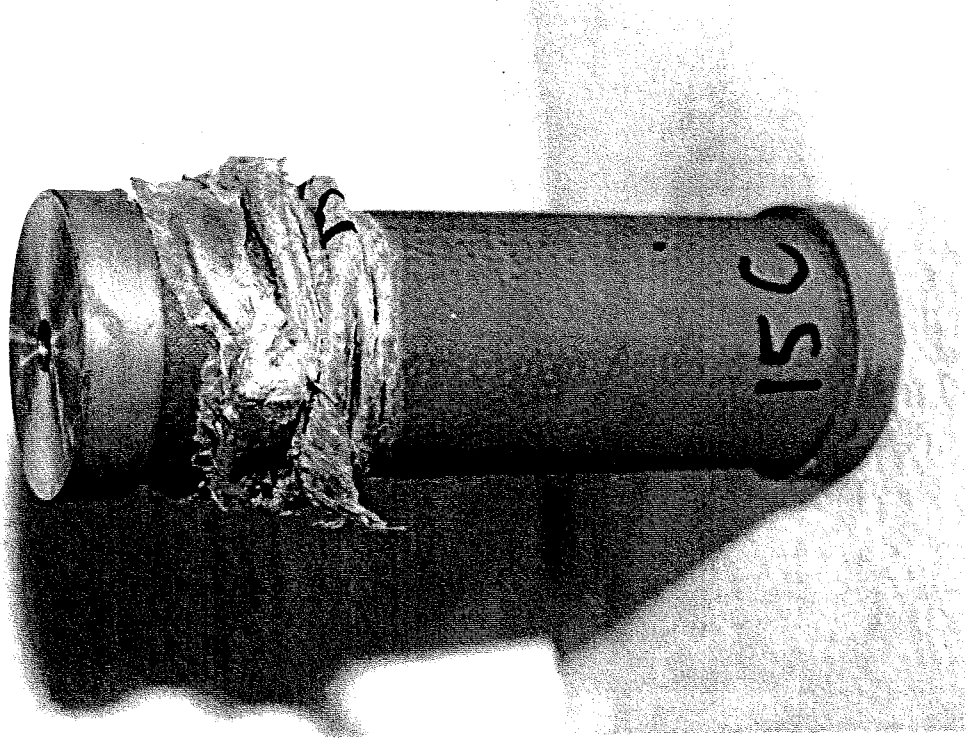


FIGURE 5.12 : Energy Absorber 15C
After Dynamic Testing

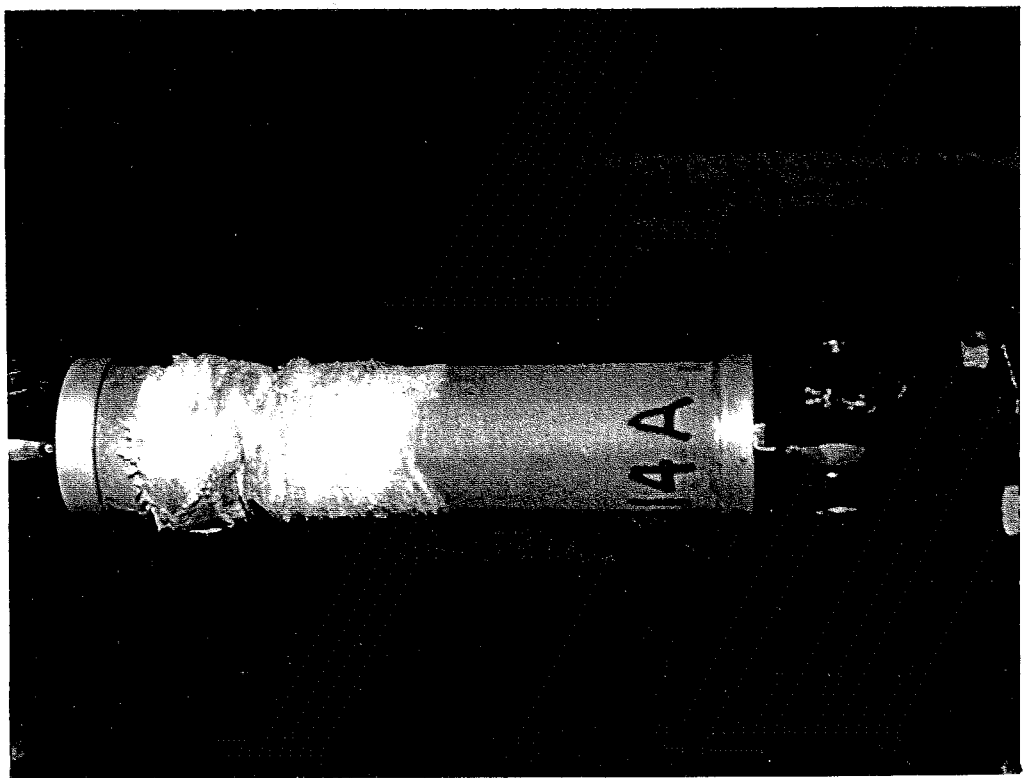


FIGURE 5.13a: Energy Absorber 14A
During Tension Test

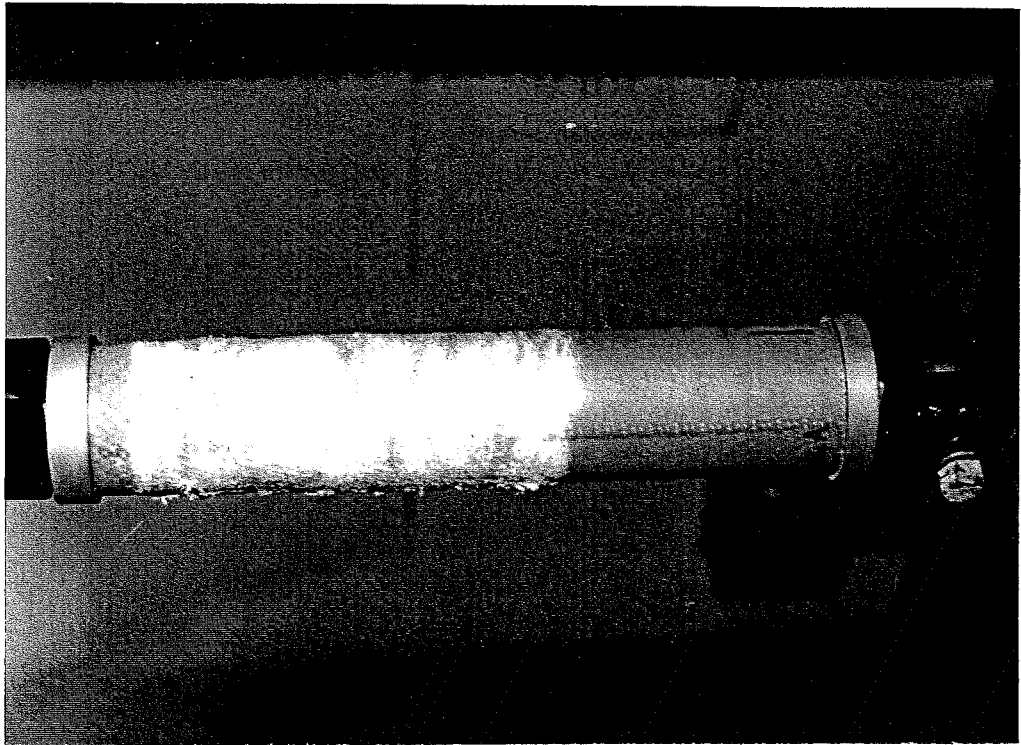


FIGURE 5.13b: Energy Absorber 14A
After Tension Test

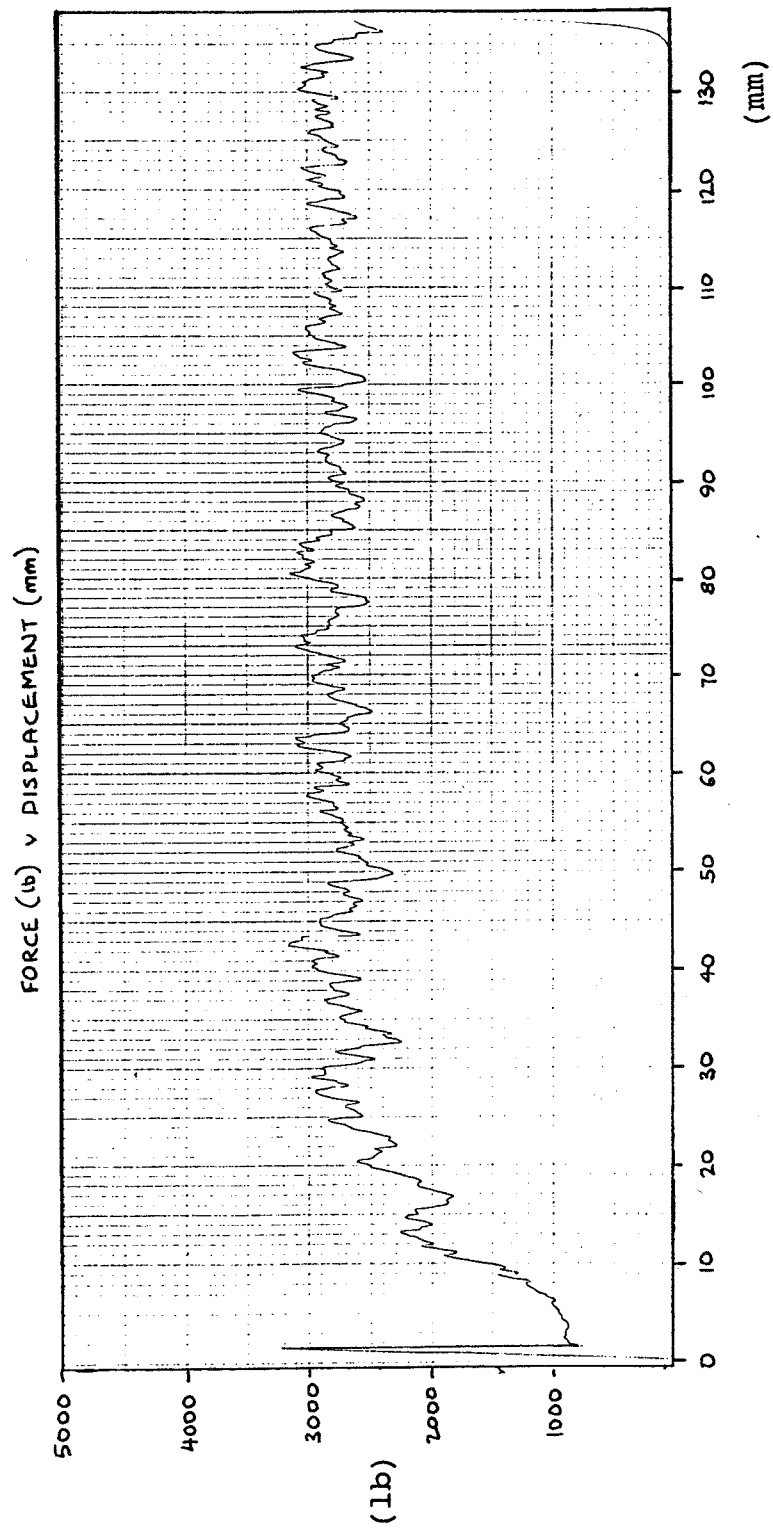


FIGURE 5.14 : Static Compression Test
Energy Absorber 9A

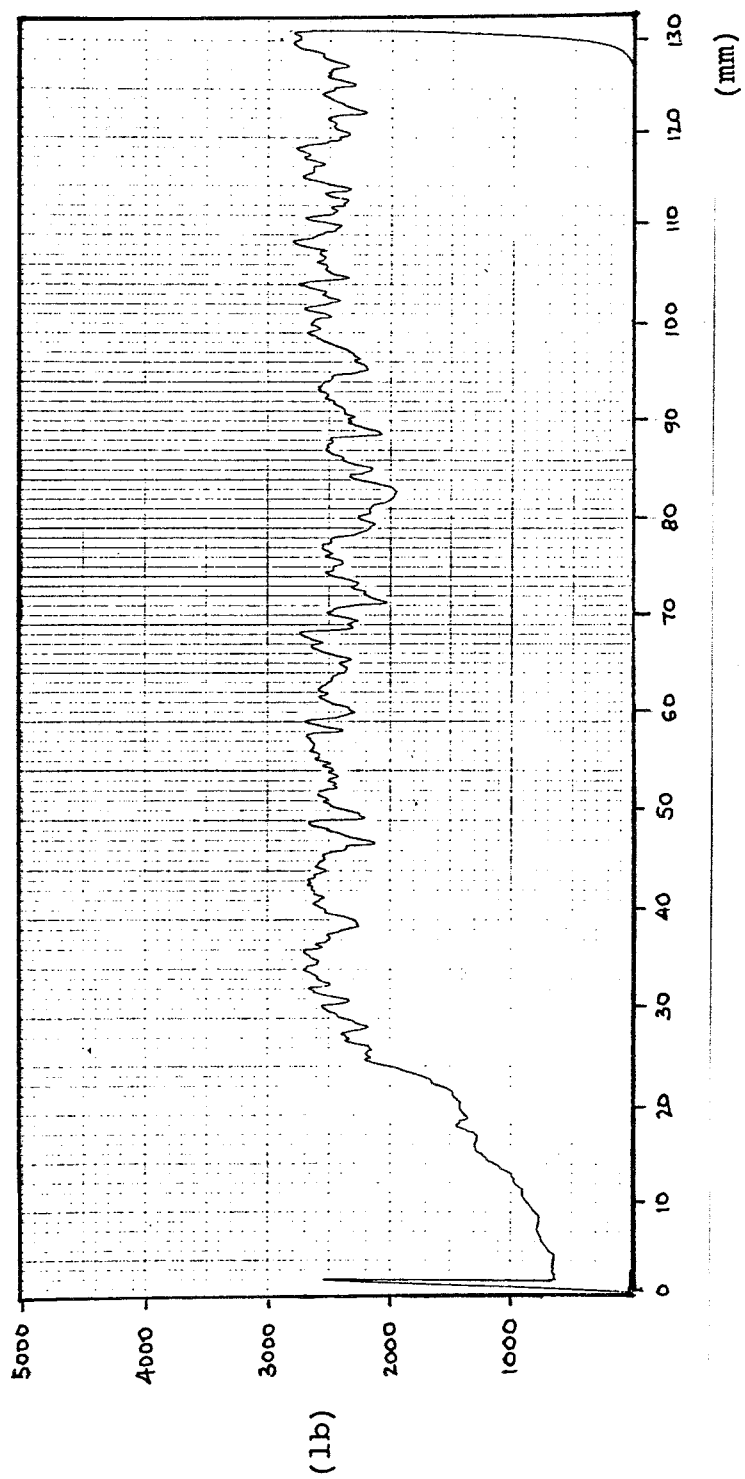


FIGURE 5.15 : Static Compression Test
Energy Absorber 9B

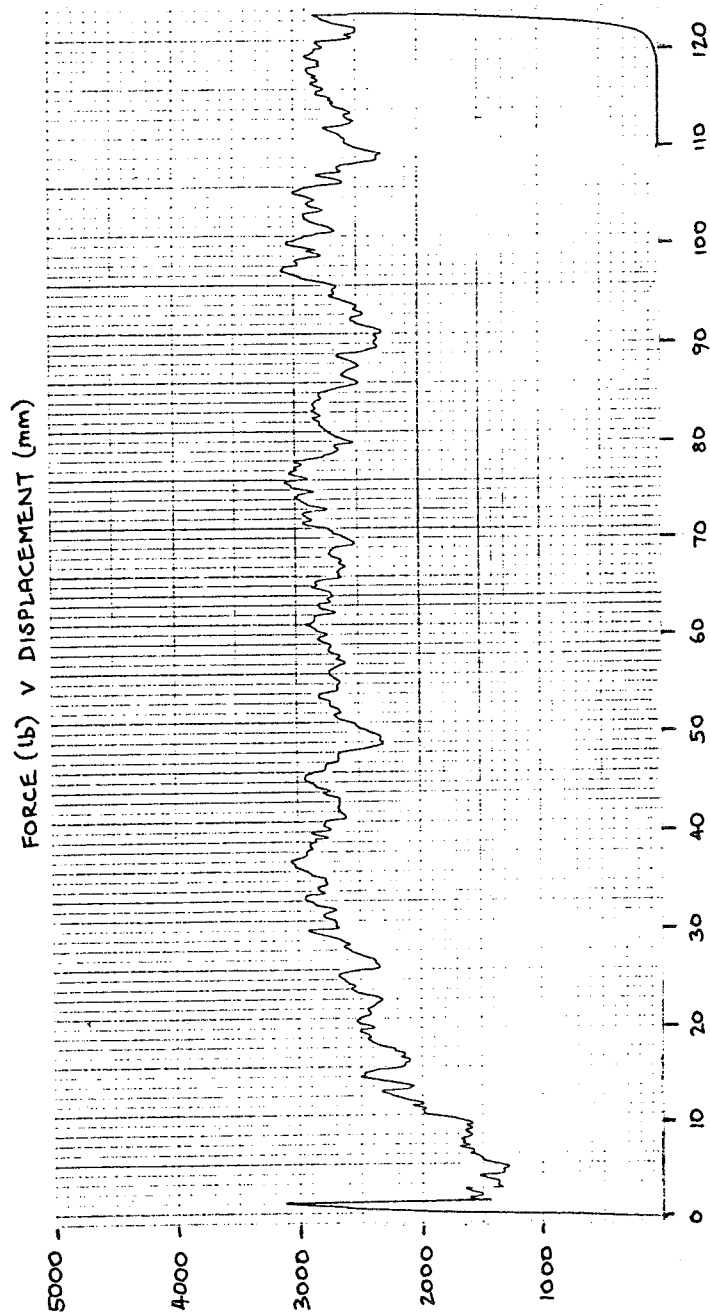


FIGURE 5.16 : Static Compression Test
Energy Absorber 9C

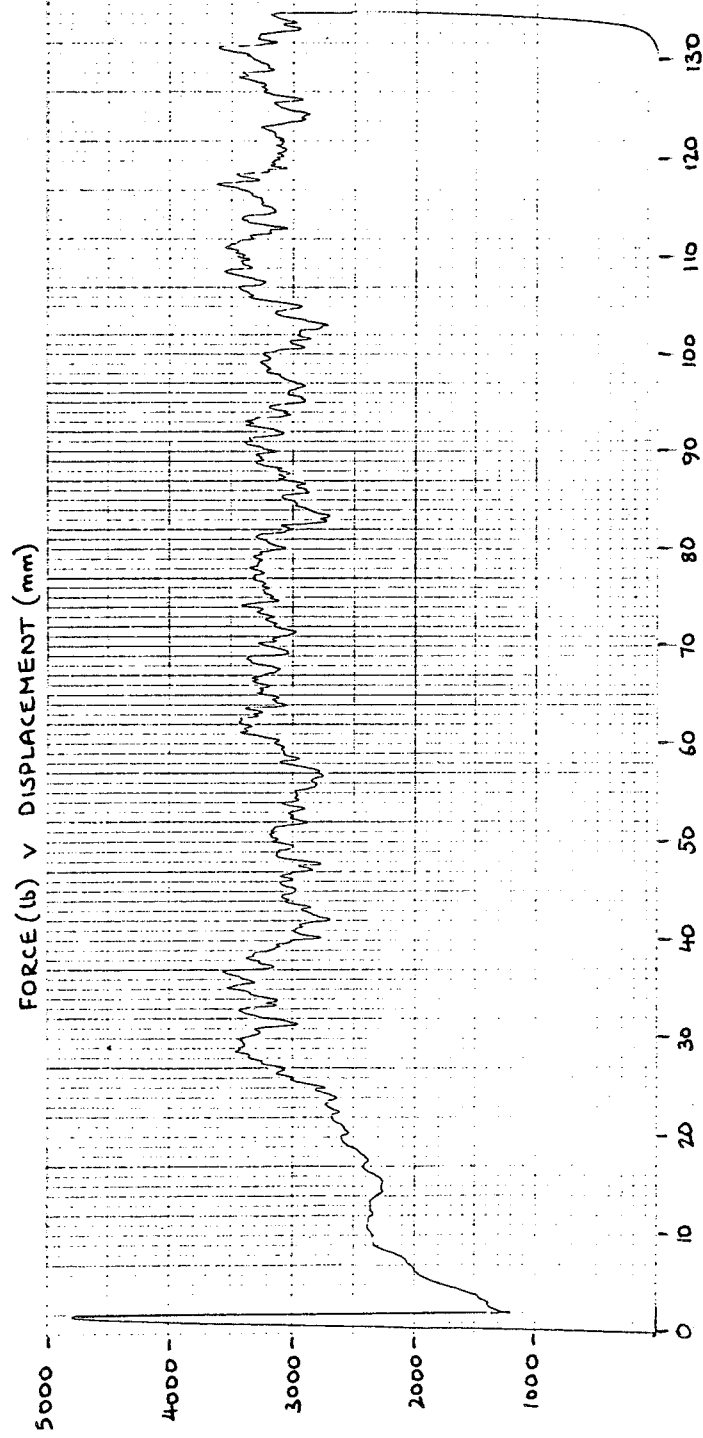


FIGURE 5.17 : Static Compression Test
Energy Absorber 10A

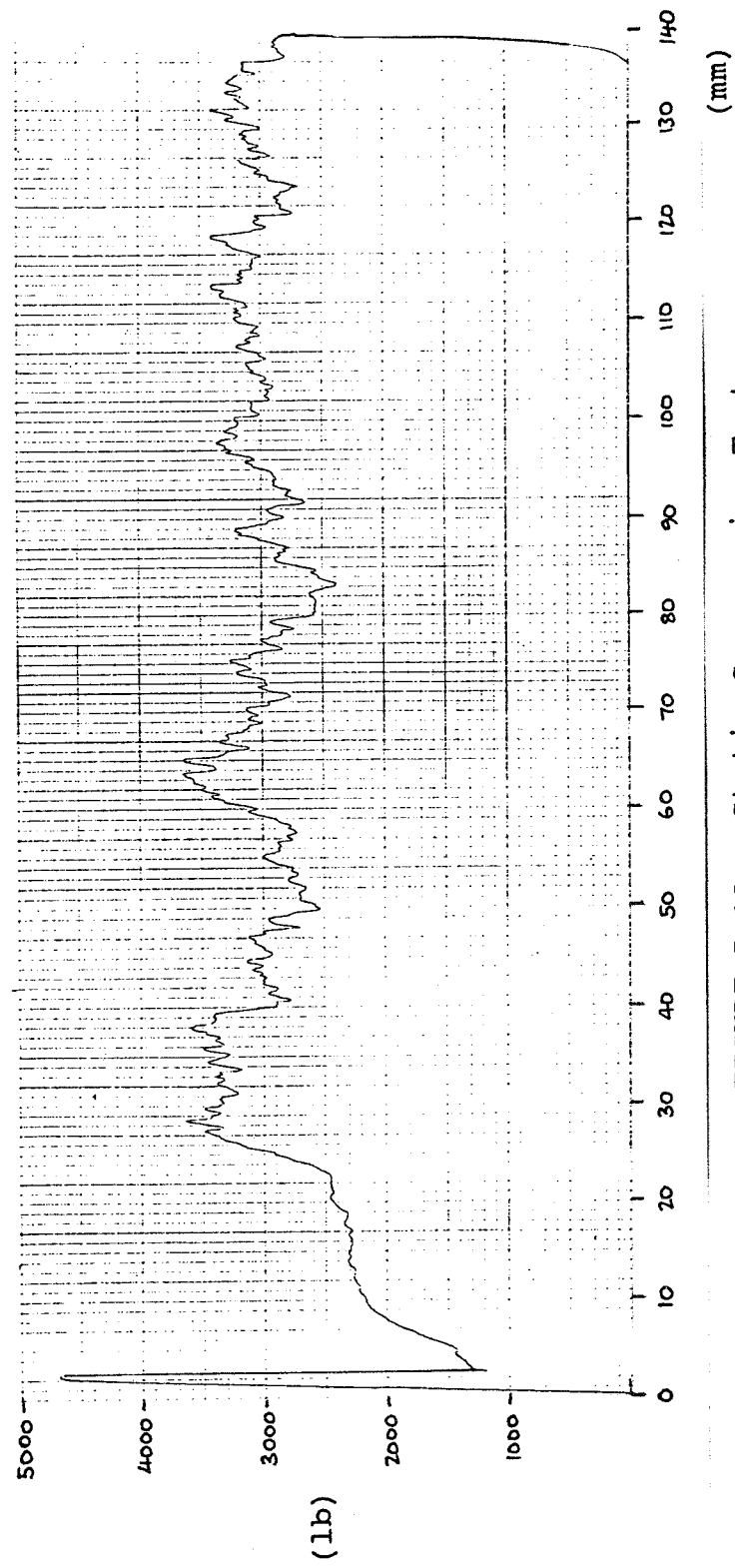


FIGURE 5.18 : Static Compression Test
Energy Absorber 10B

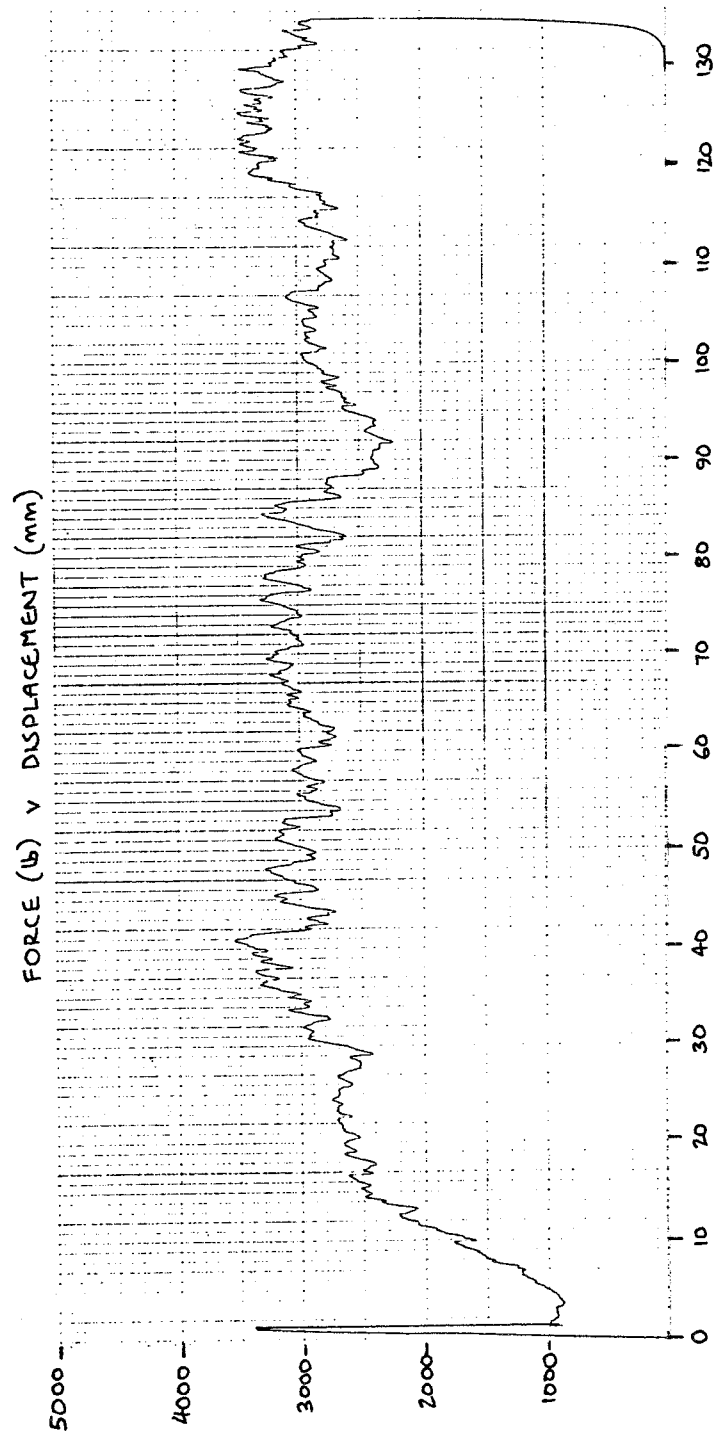


FIGURE 5.19 : Static Compression Test
Energy Absorber 11A

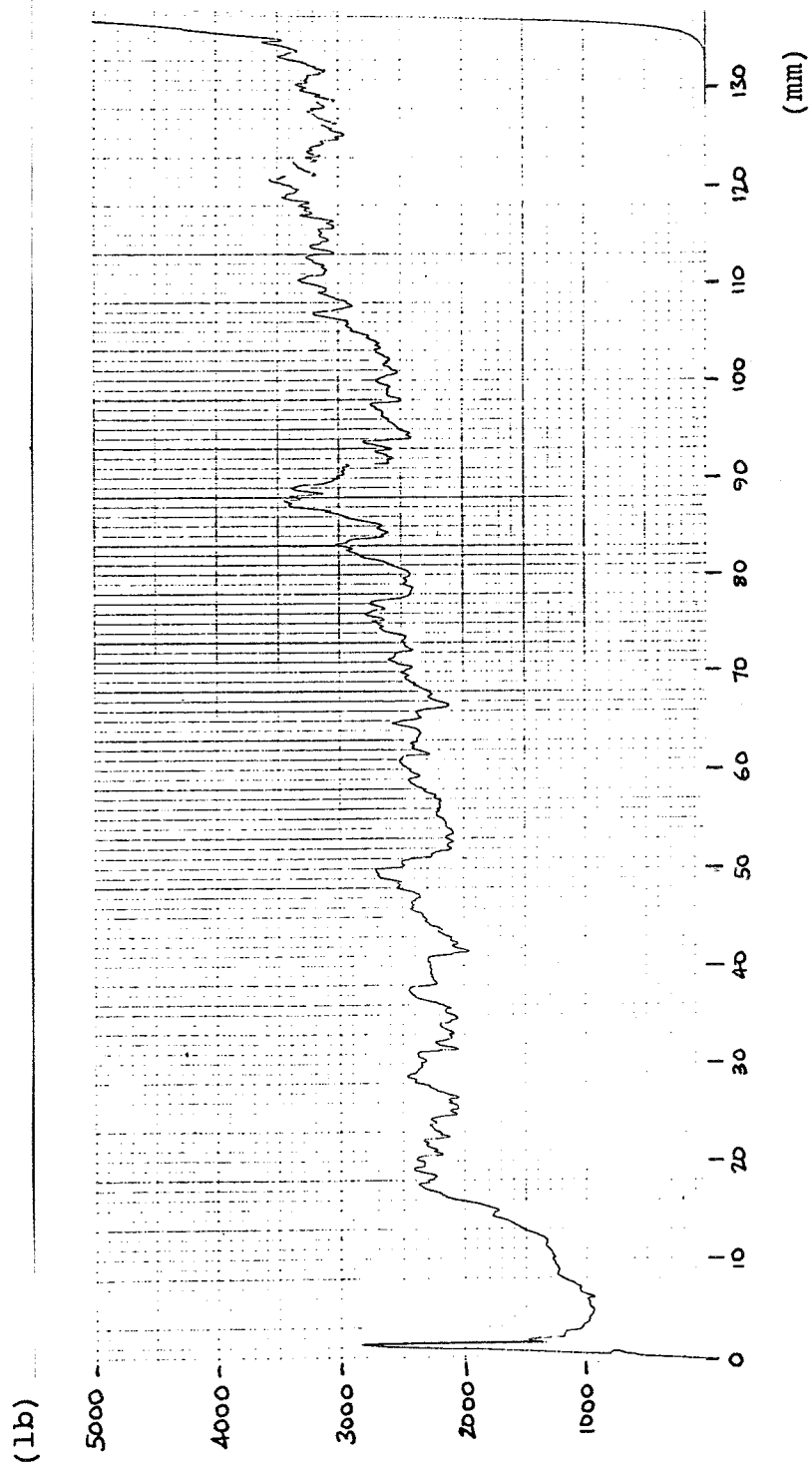
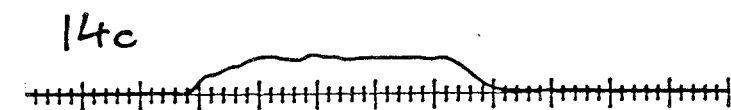
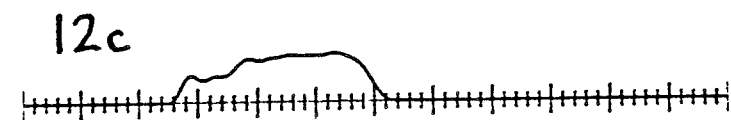
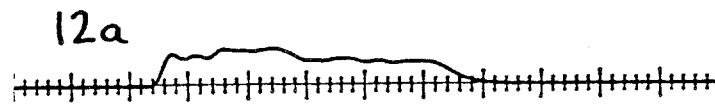
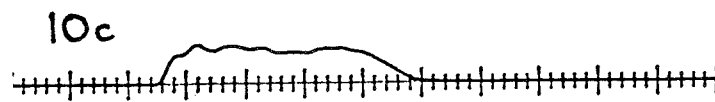
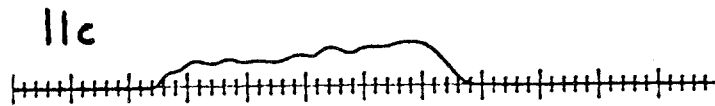
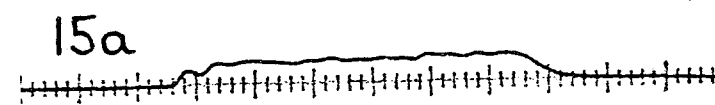
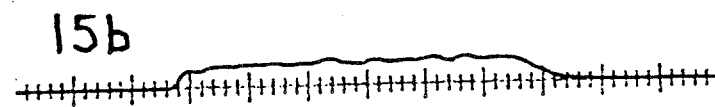
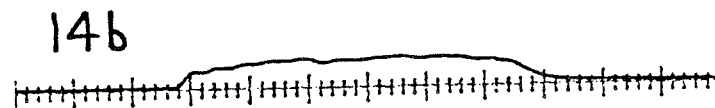
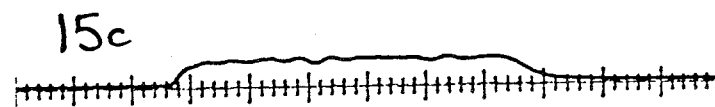
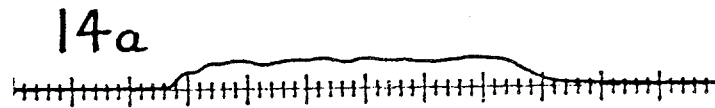


FIGURE 5.20 : Static Compression Test
Energy Absorber 11B



X SCALE=2.5MS/DIV Y SCALE -68.18 to 68.18 KN

FIGURE 5.21 : Dynamic Crush Tests
Kevlar Energy Absorbers



X SCALE=2.5MS/DIV Y SCALE -68.17 to 68.17 KN

FIGURE 5.22 : Dynamic Crush Tests
Kevlar Energy Absorbers

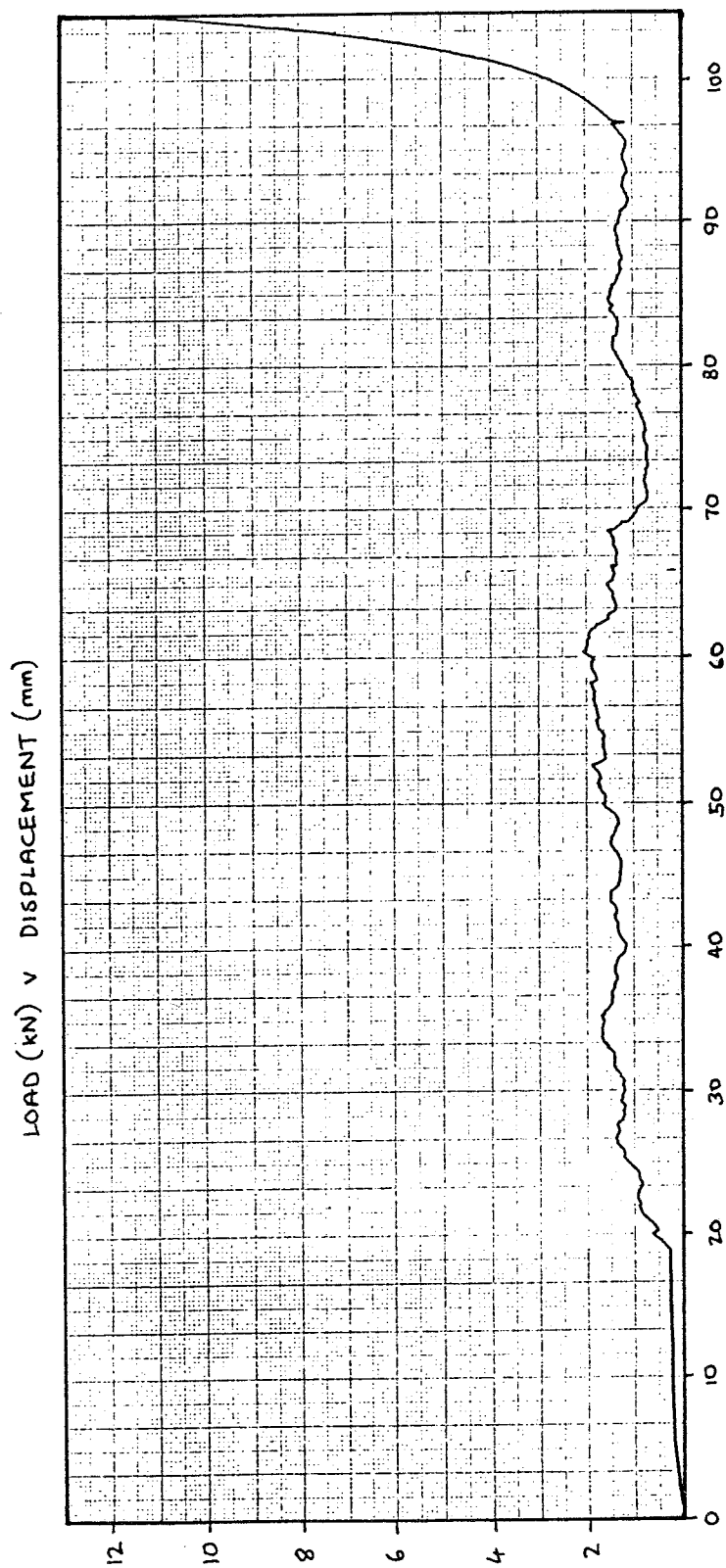


FIG 5.23 : TENSILE TEST ON ENERGY ABSORBER PREVIOUSLY
TESTED DYNAMICALLY IN COMPRESSION; REF 14A

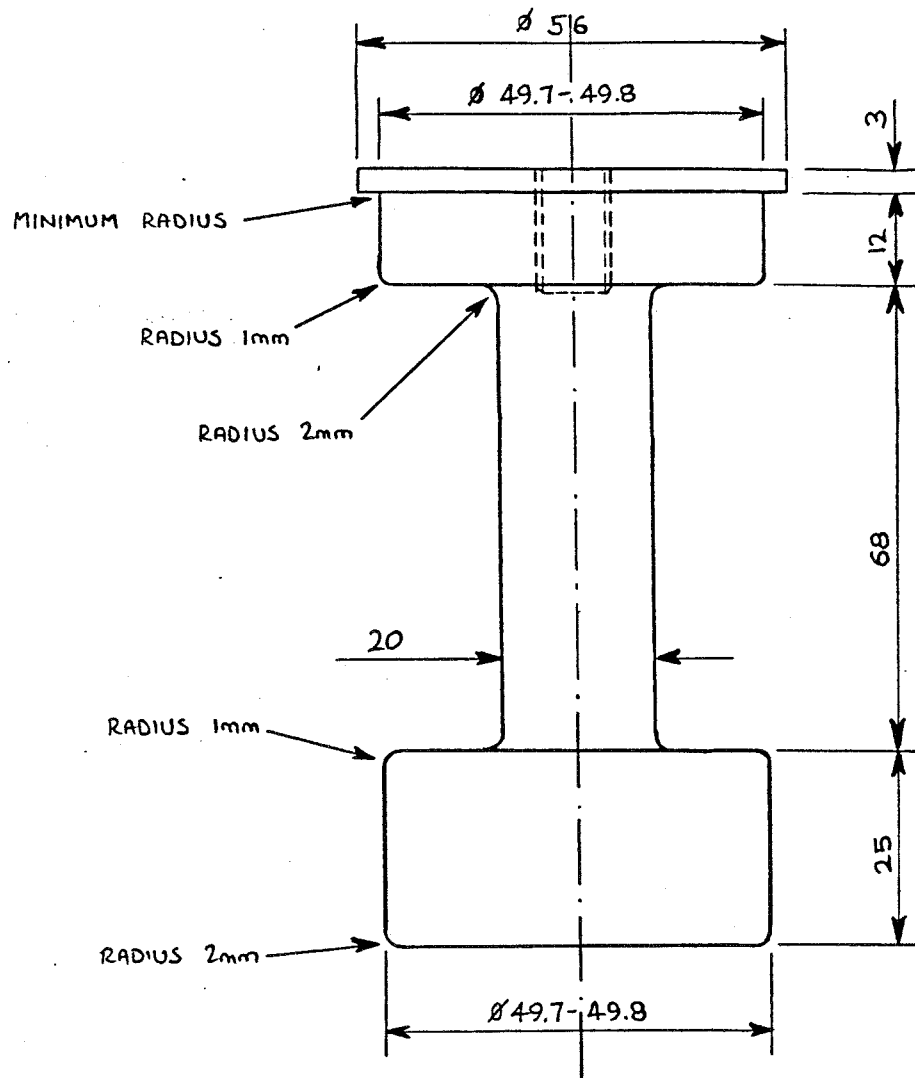


FIGURE 5.24 : Energy Absorber Sliding End Fitting

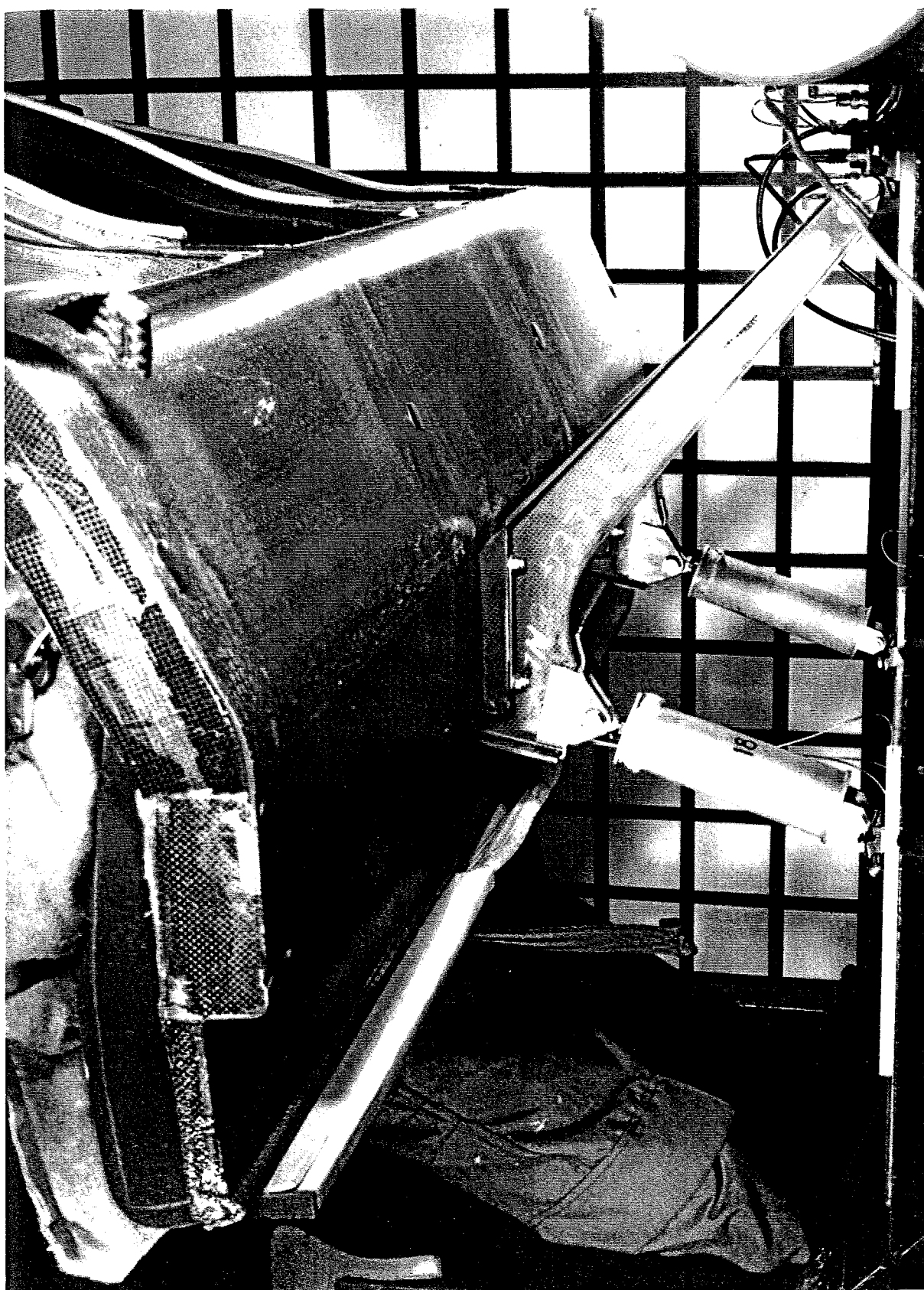


FIGURE 6.1 : Test 17 (Post-test)

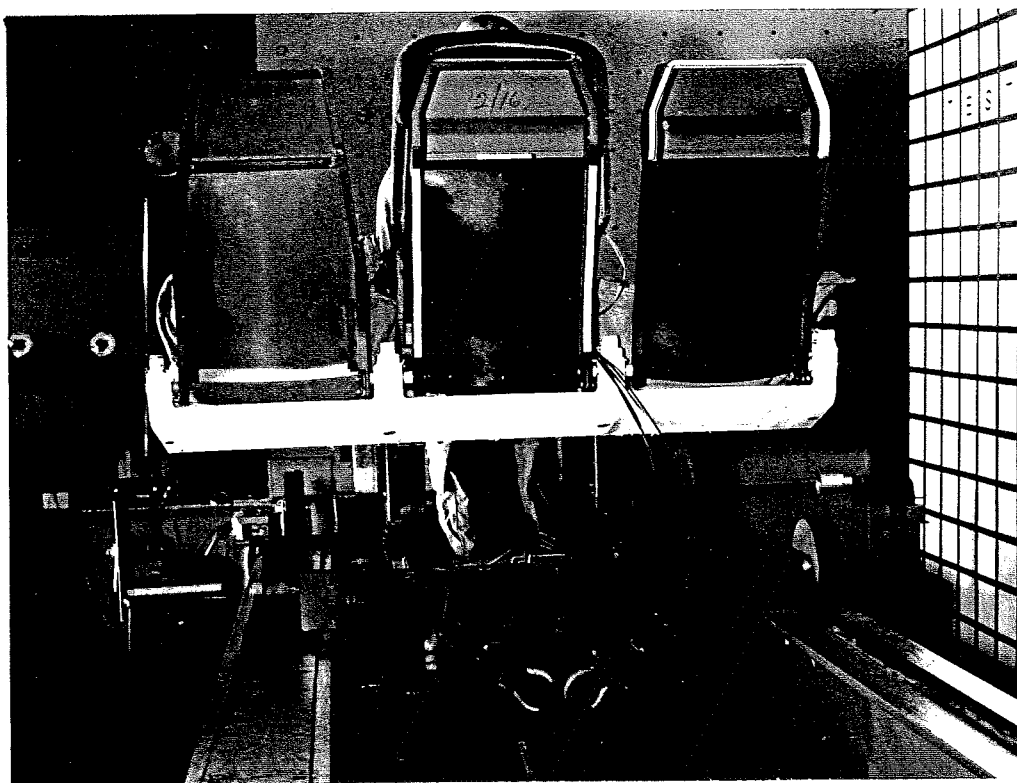


FIGURE 6.2a : Test 24 (Pre-test)



FIGURE 6.2b : Test 24 (Post-test)

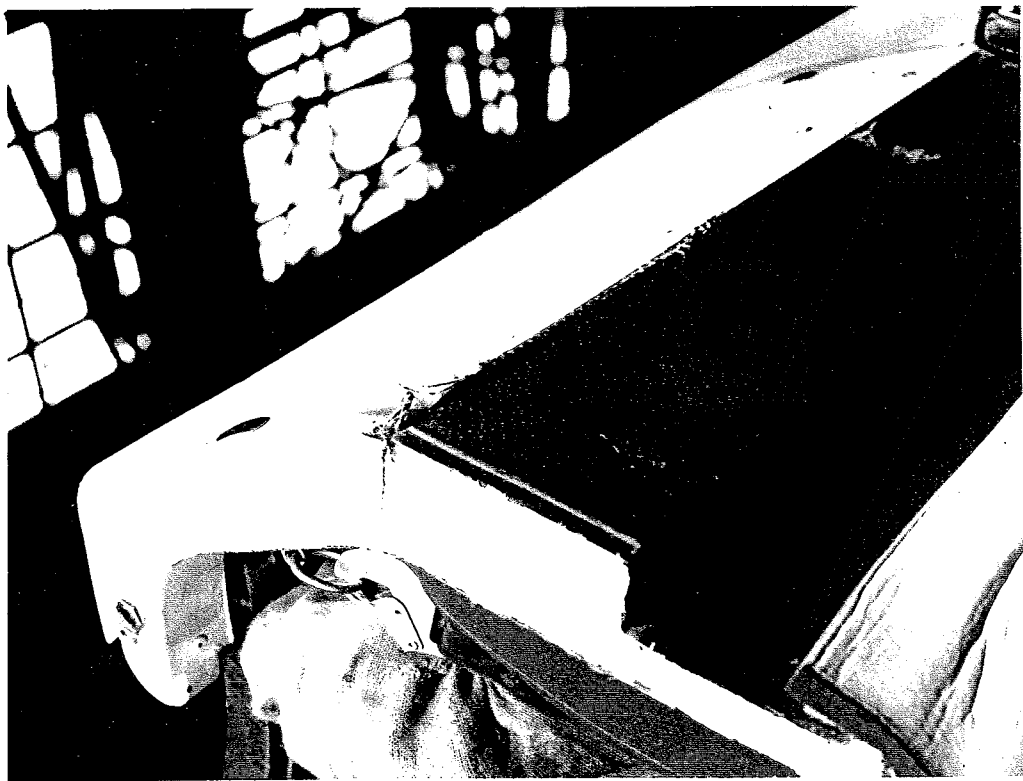


FIGURE 6.3a : Test 24 (Post-test)

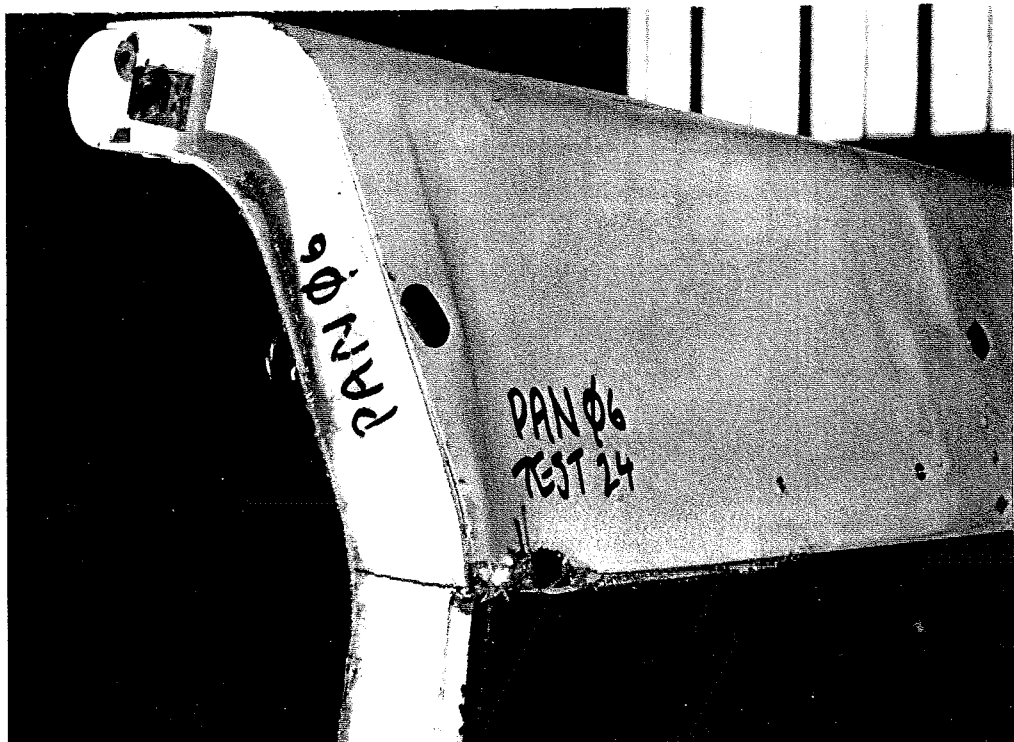


FIGURE 6.3b : Test 24 (Post-test)

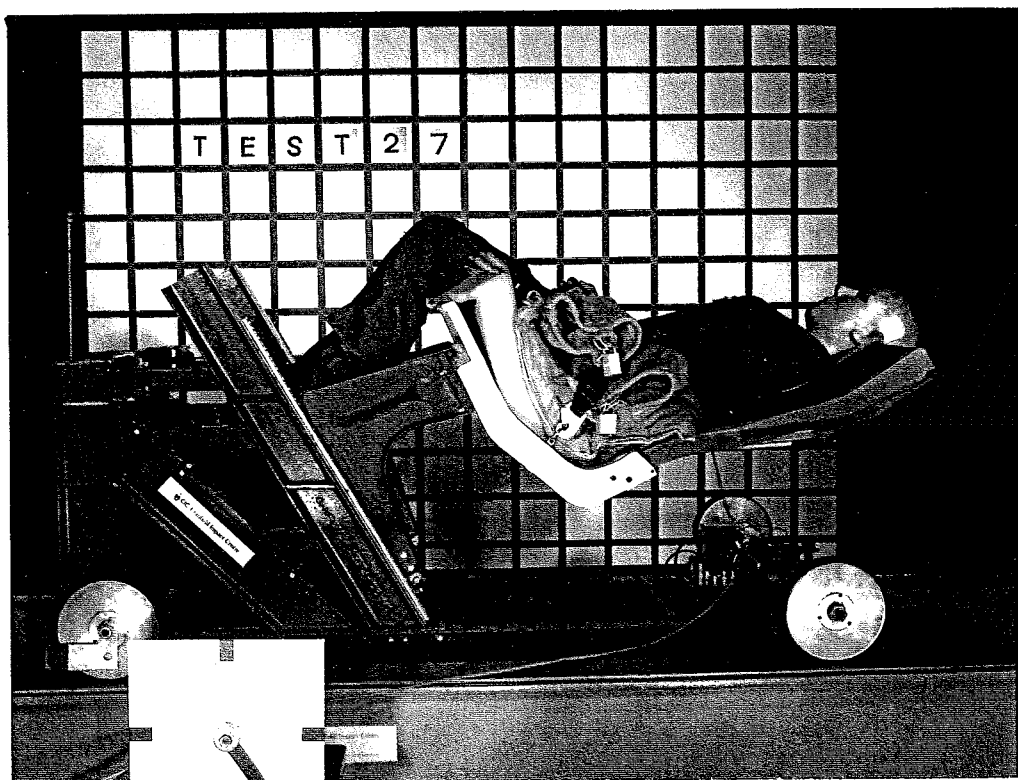


FIGURE 6.4 : Test 27 (Post-test)

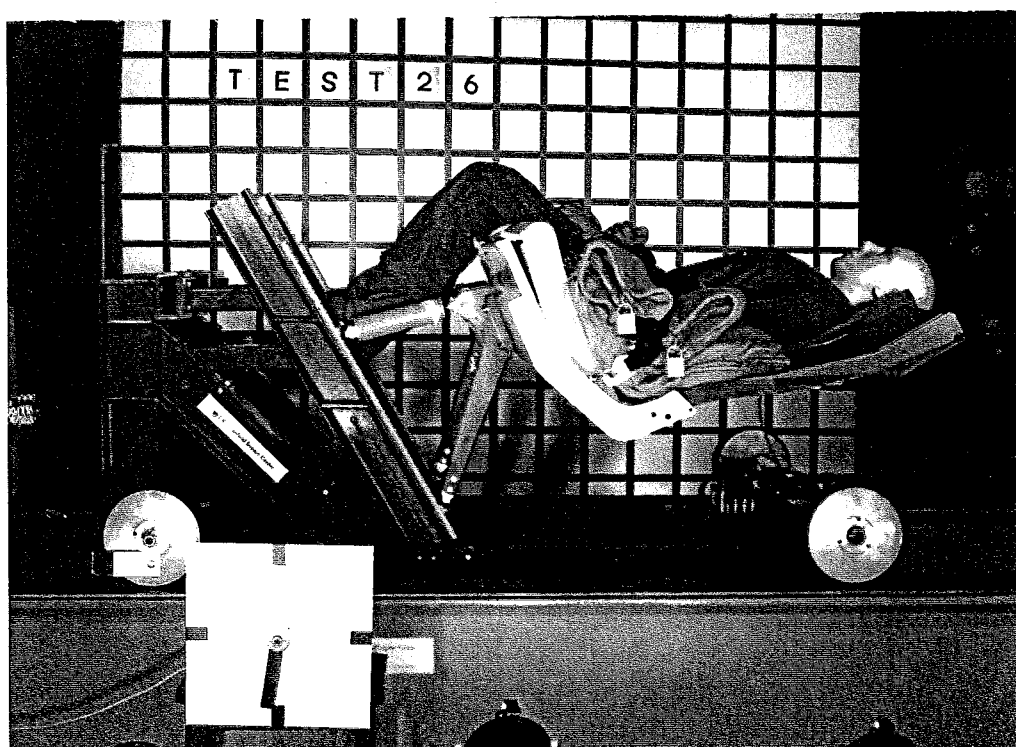


FIGURE 6.5 : Test 26 (Post-test)

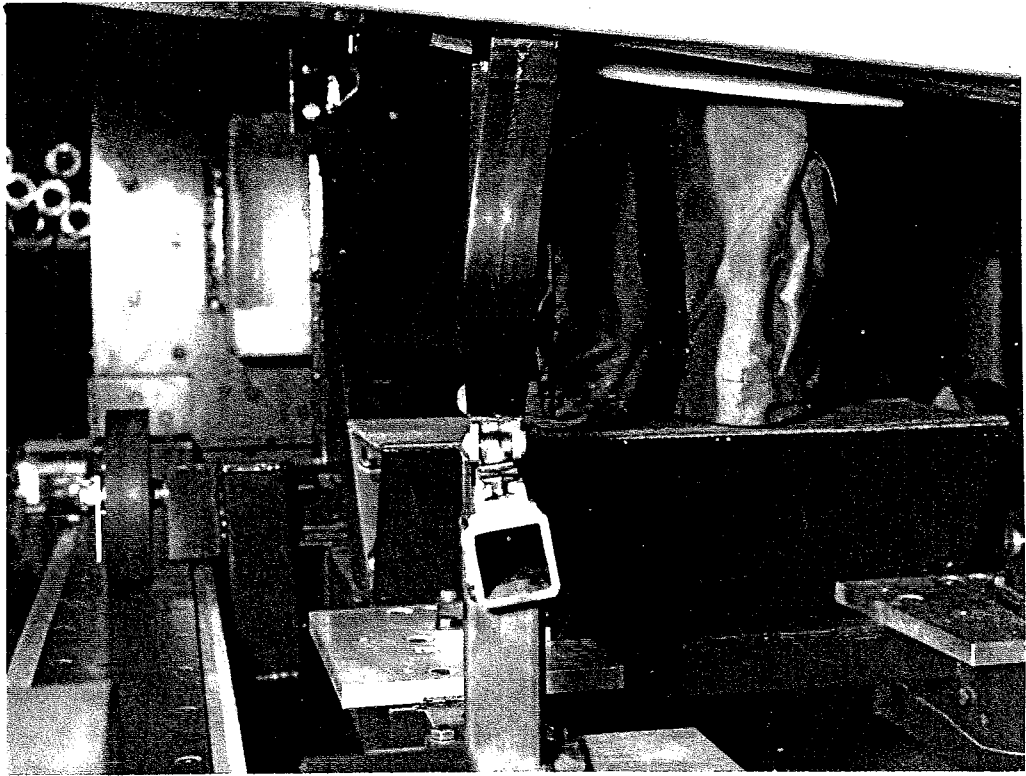


FIGURE 6.6 : Standard Seat
during 'Floor Warping'

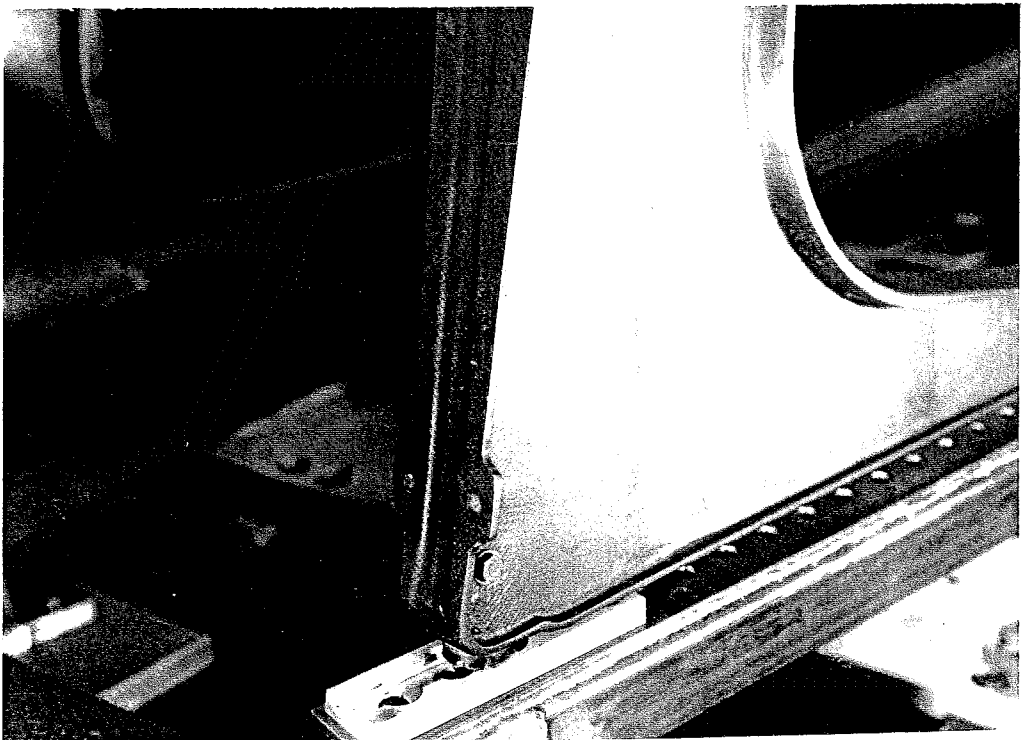


FIGURE 6.7 : Standard Seat
during 'Floor Warping'



FIGURE 6.8 : Standard Seat
during 'Floor Warping'

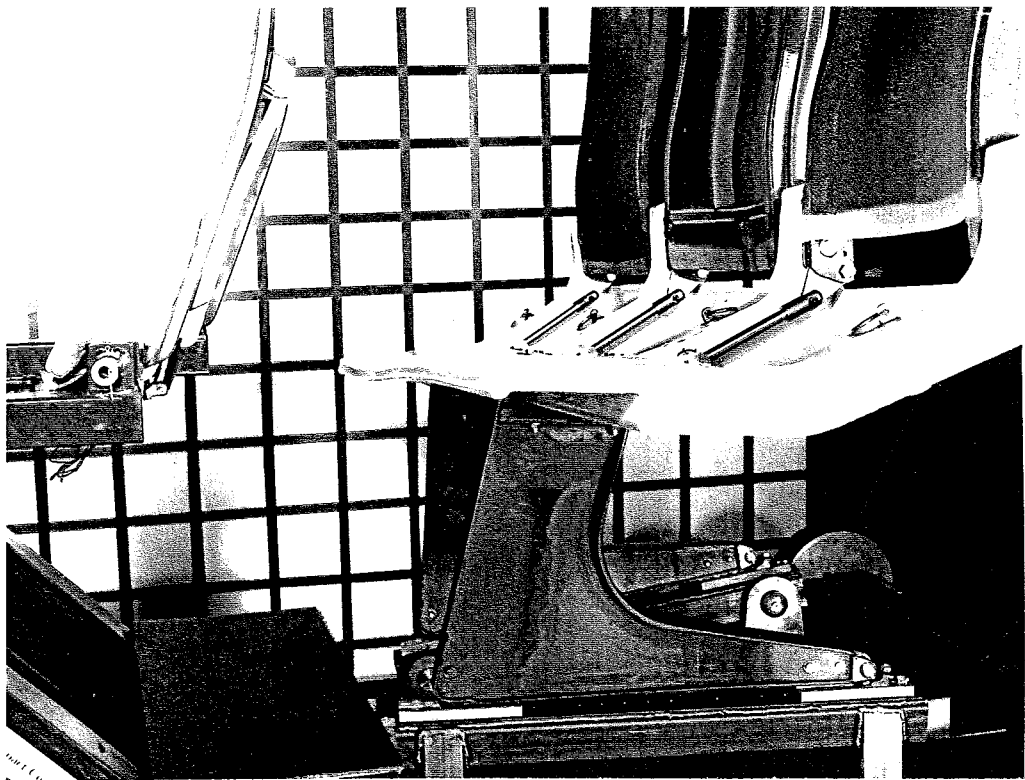


FIGURE 6.9 : Standard Seat
during 'Floor Warping'

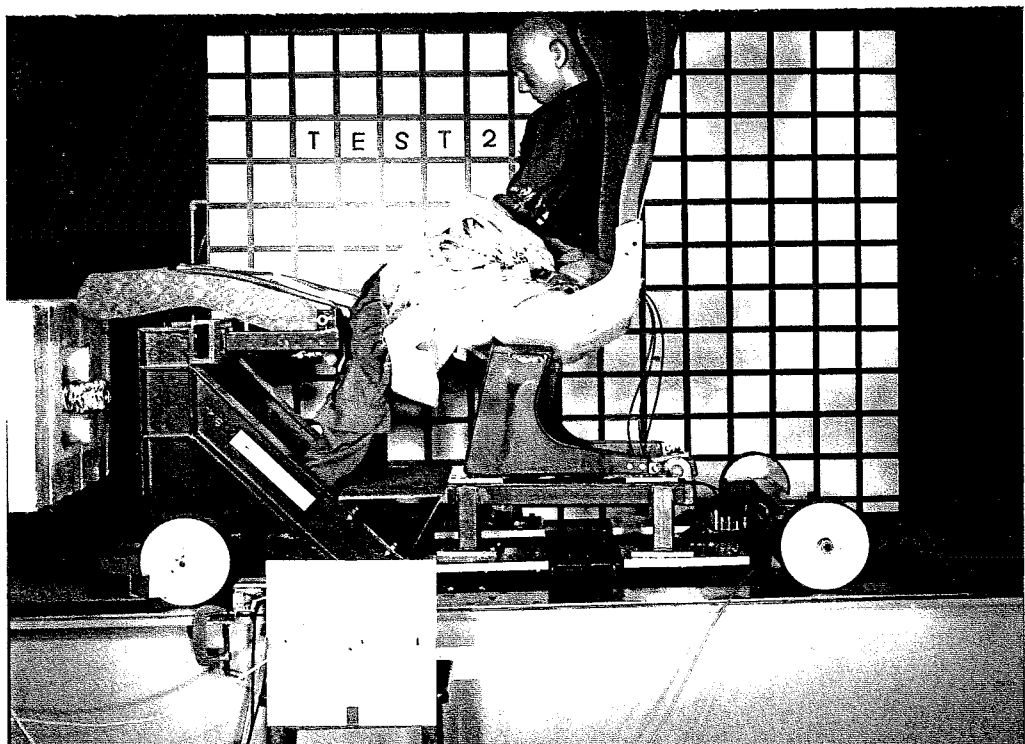


FIGURE 6.10 : Test 29 (Post-Test)

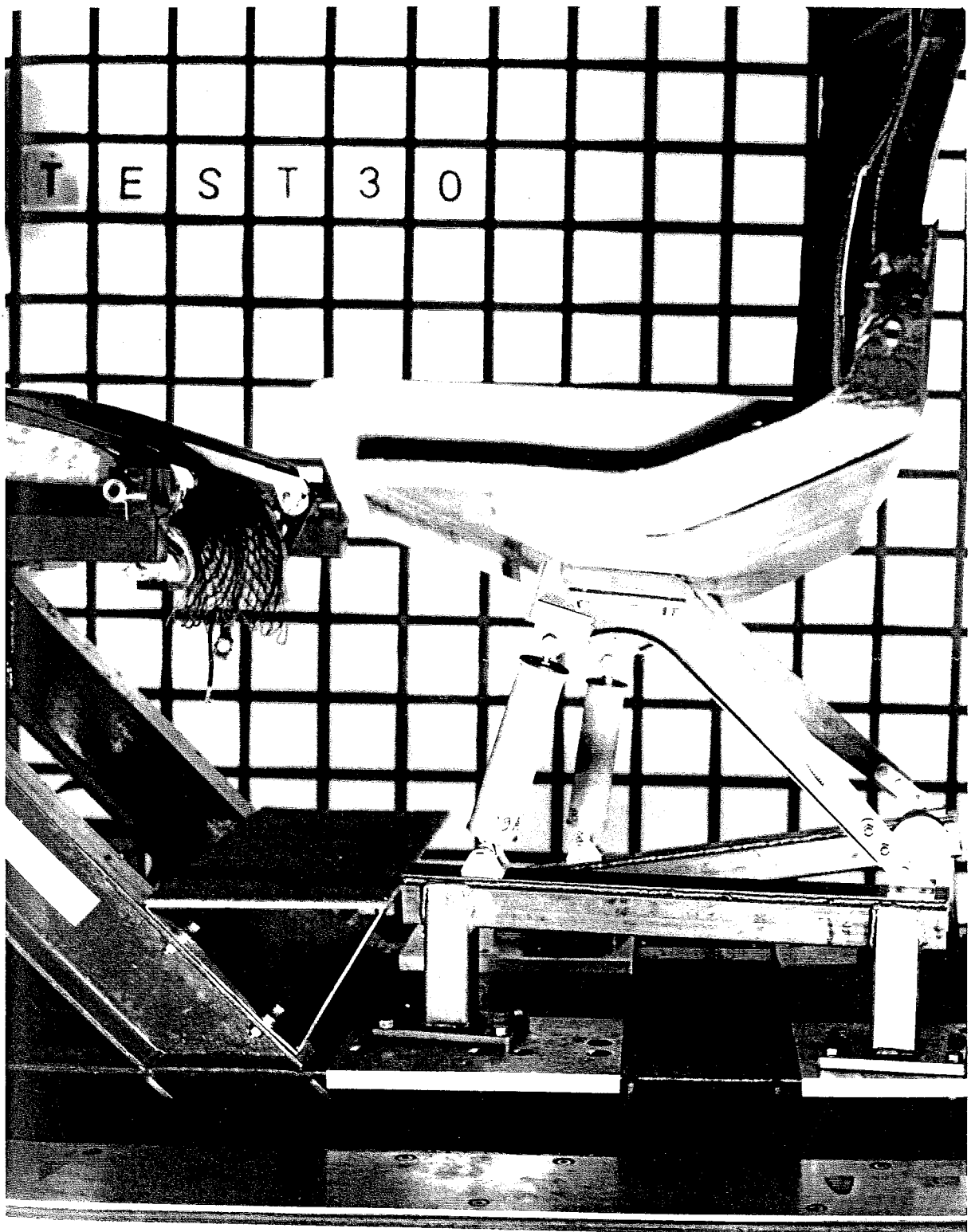


FIGURE 6.11 : Energy Absorbing Seat
during 'Floor Warping'

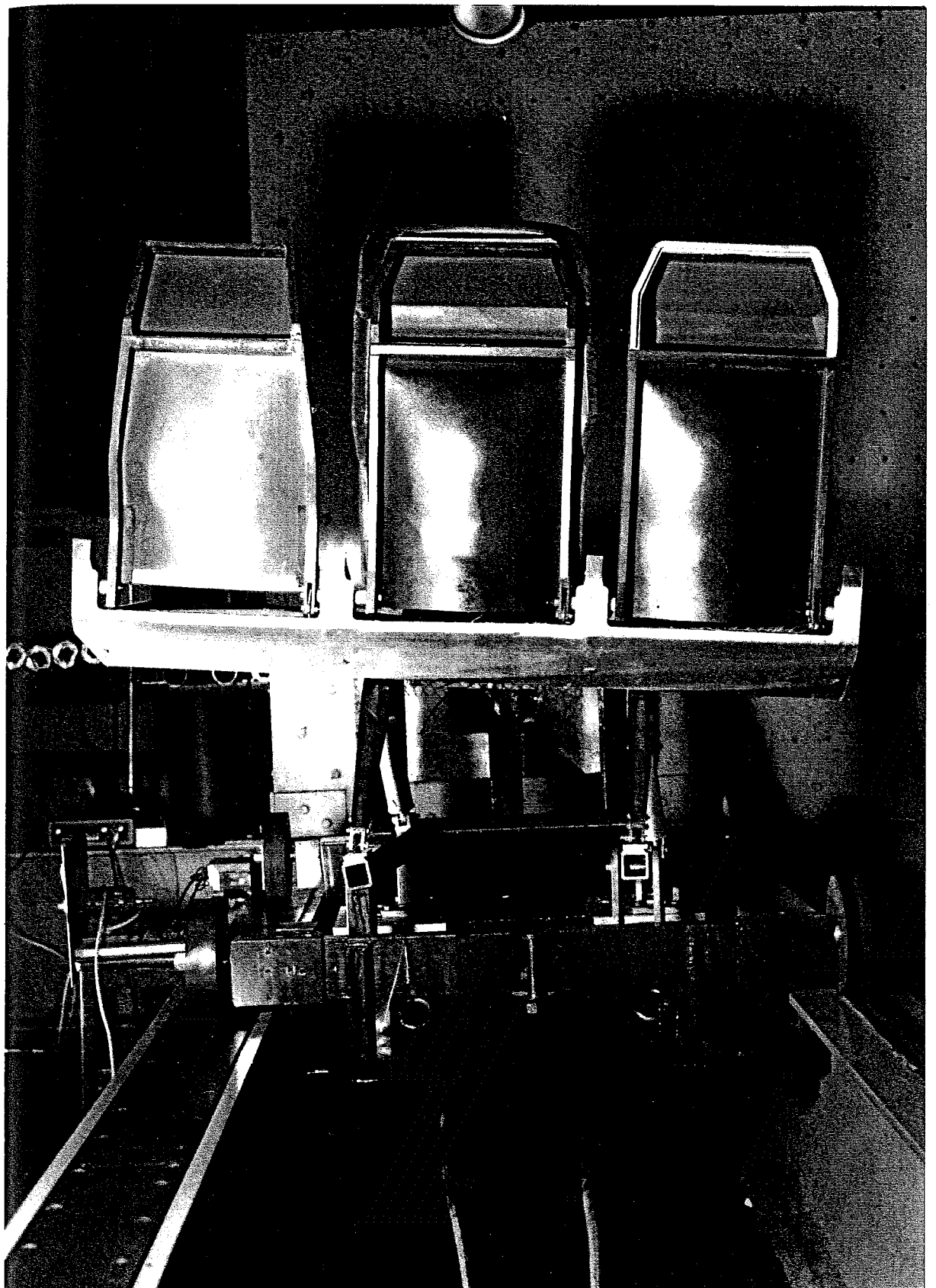


FIGURE 6.12 : Energy Absorbing Seat
during 'Floor Warping'

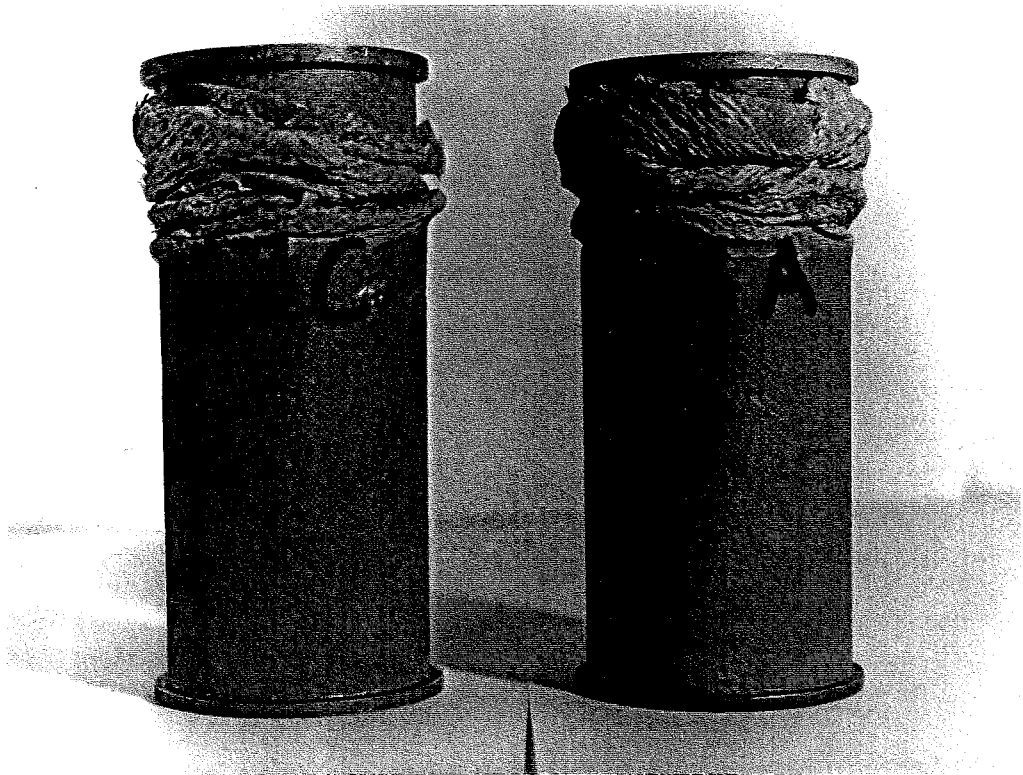


FIGURE 6.13 : Energy Absorbers after
Dynamic and Static Tests



FIGURE 6.14 : Energy Absorbers after
Dynamic and Static Tests



FIGURE 6.15 : Energy Absorbers
Intact and Post-crush

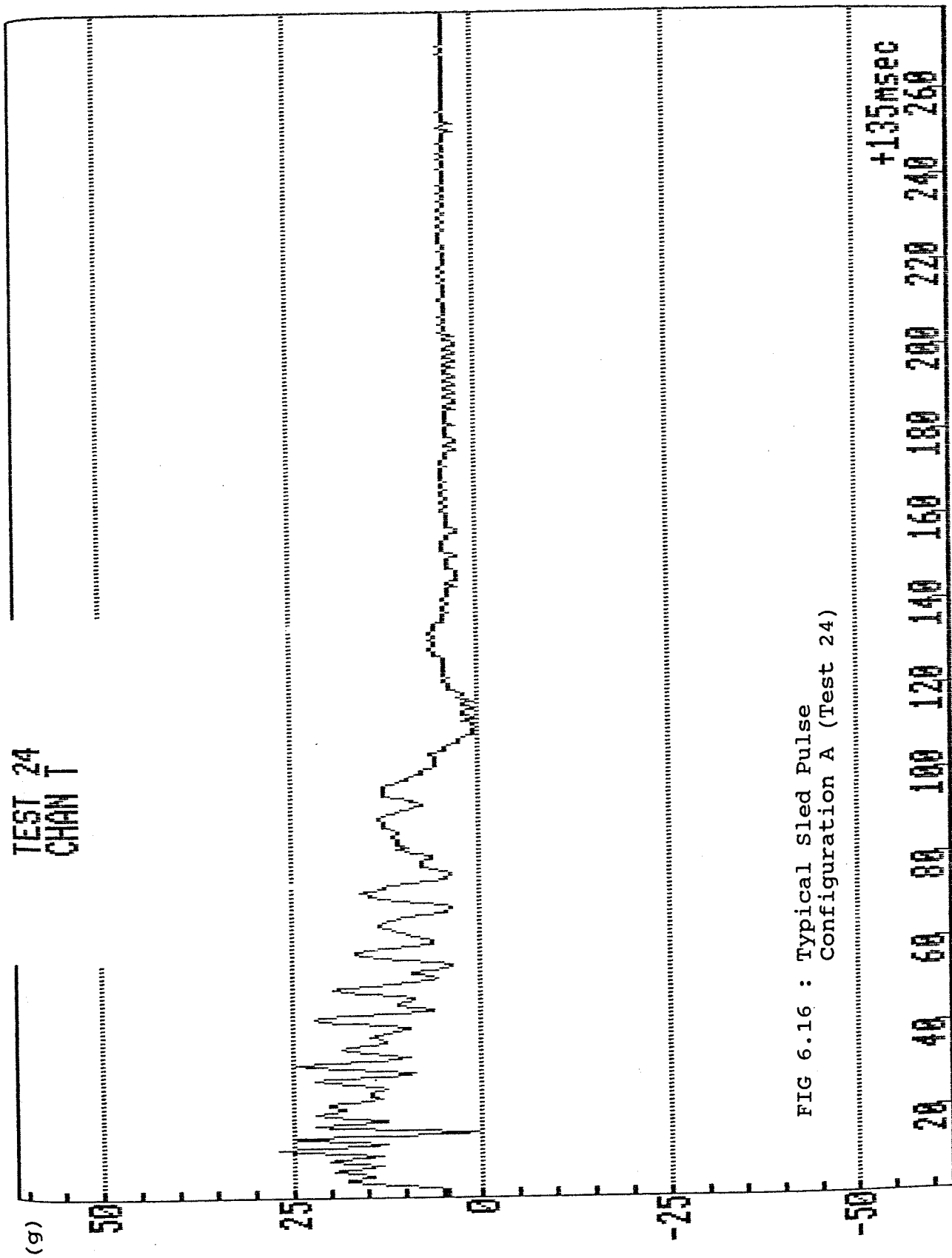
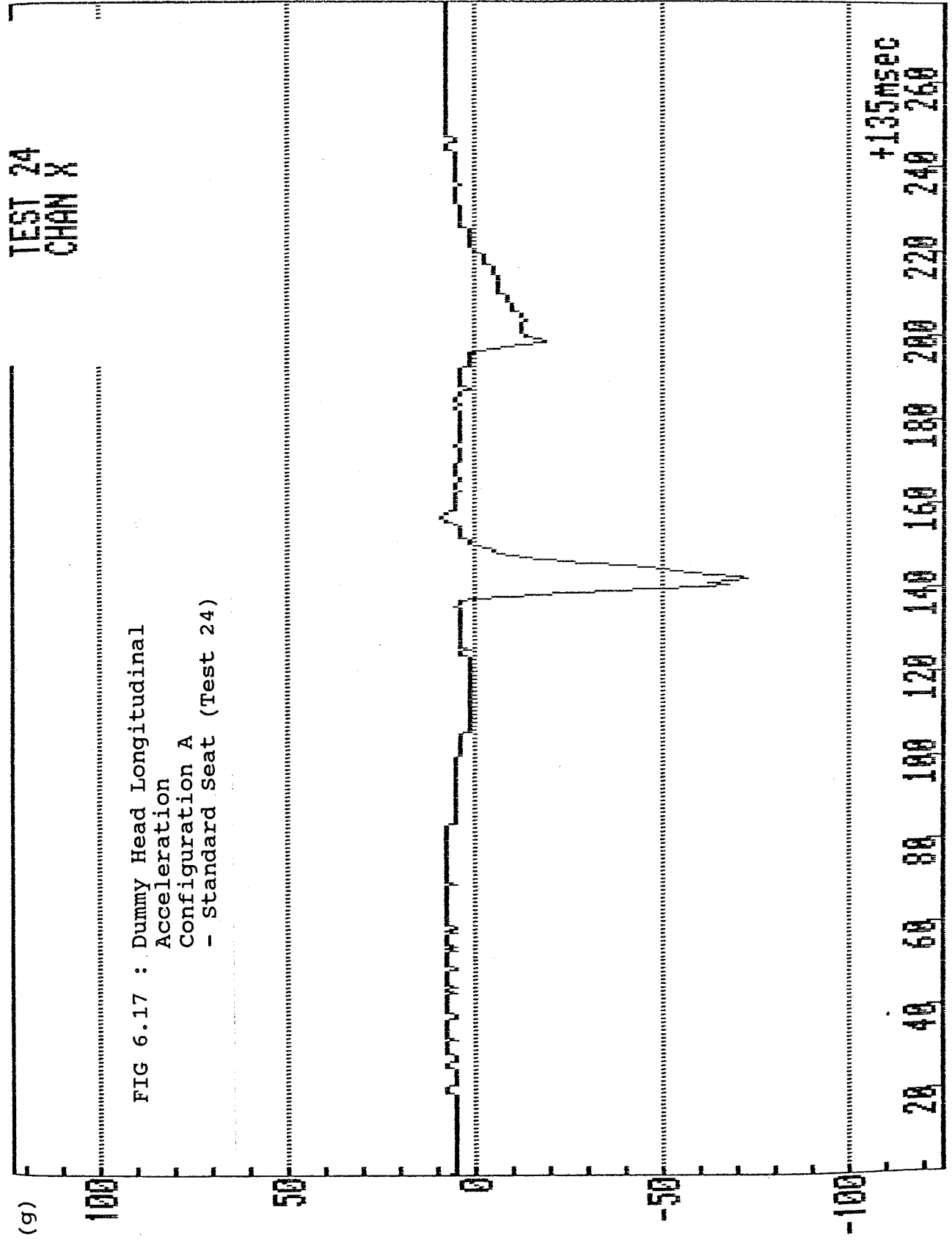
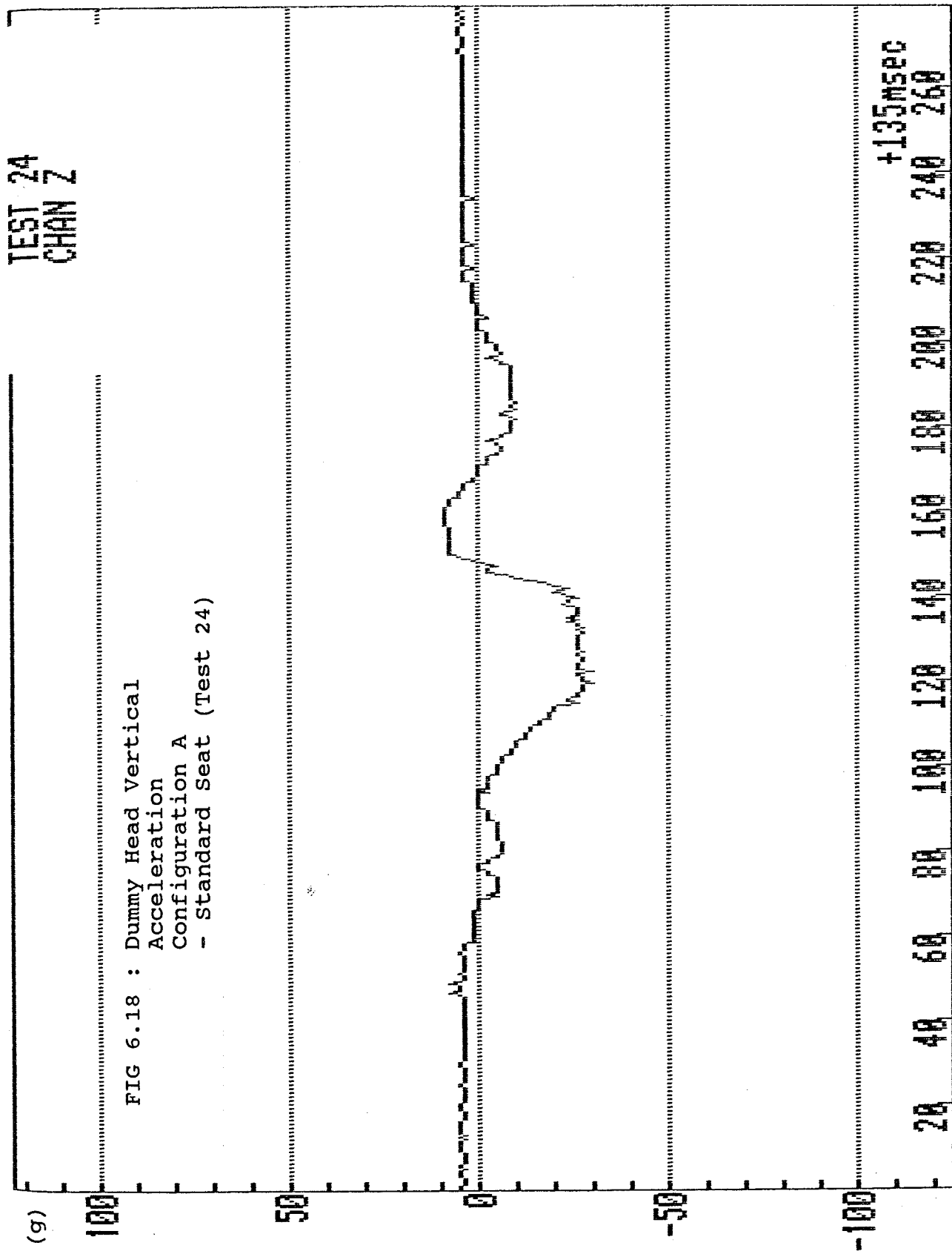
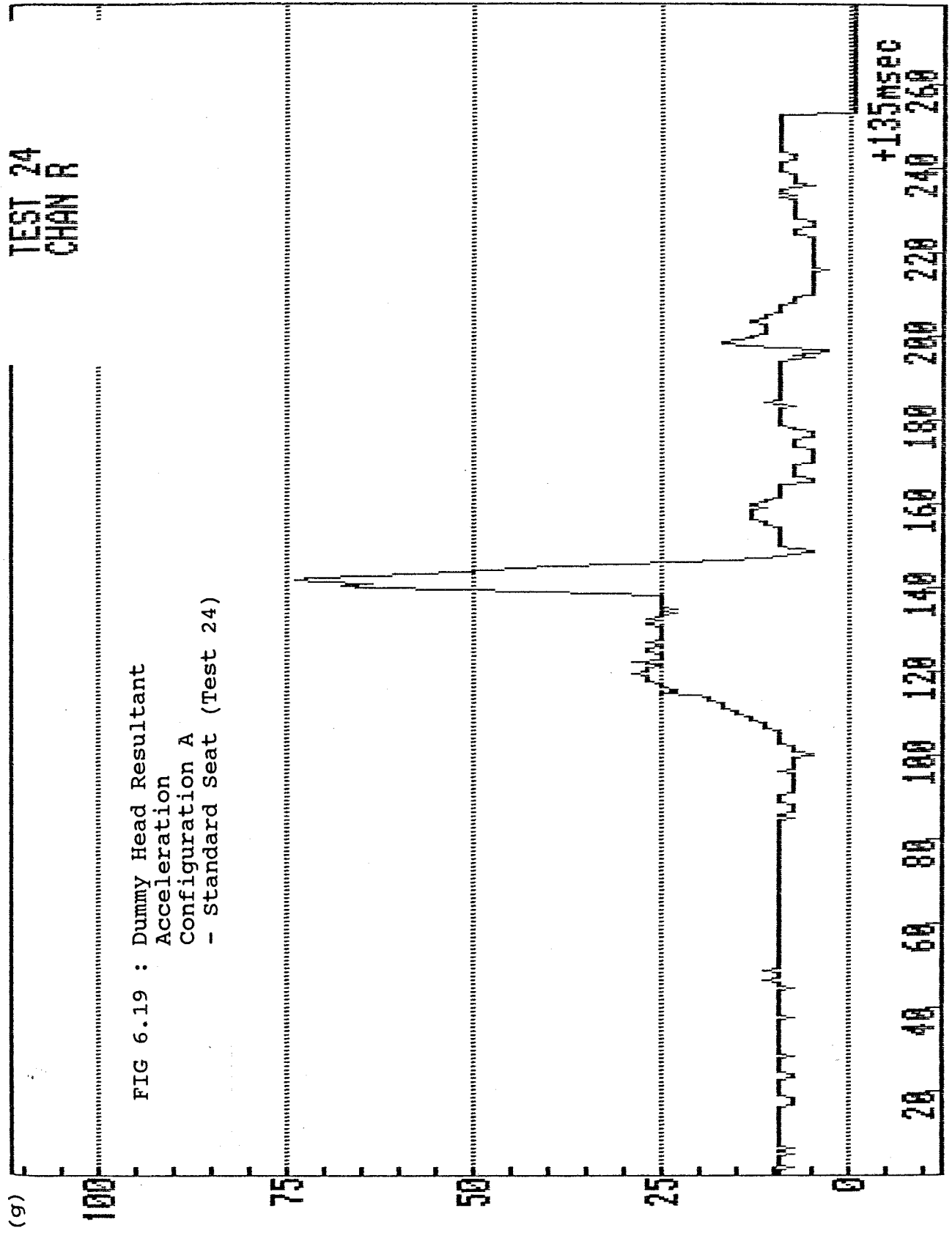


FIG 6.16 : Typical Sled Pulse
Configuration A (Test 24)







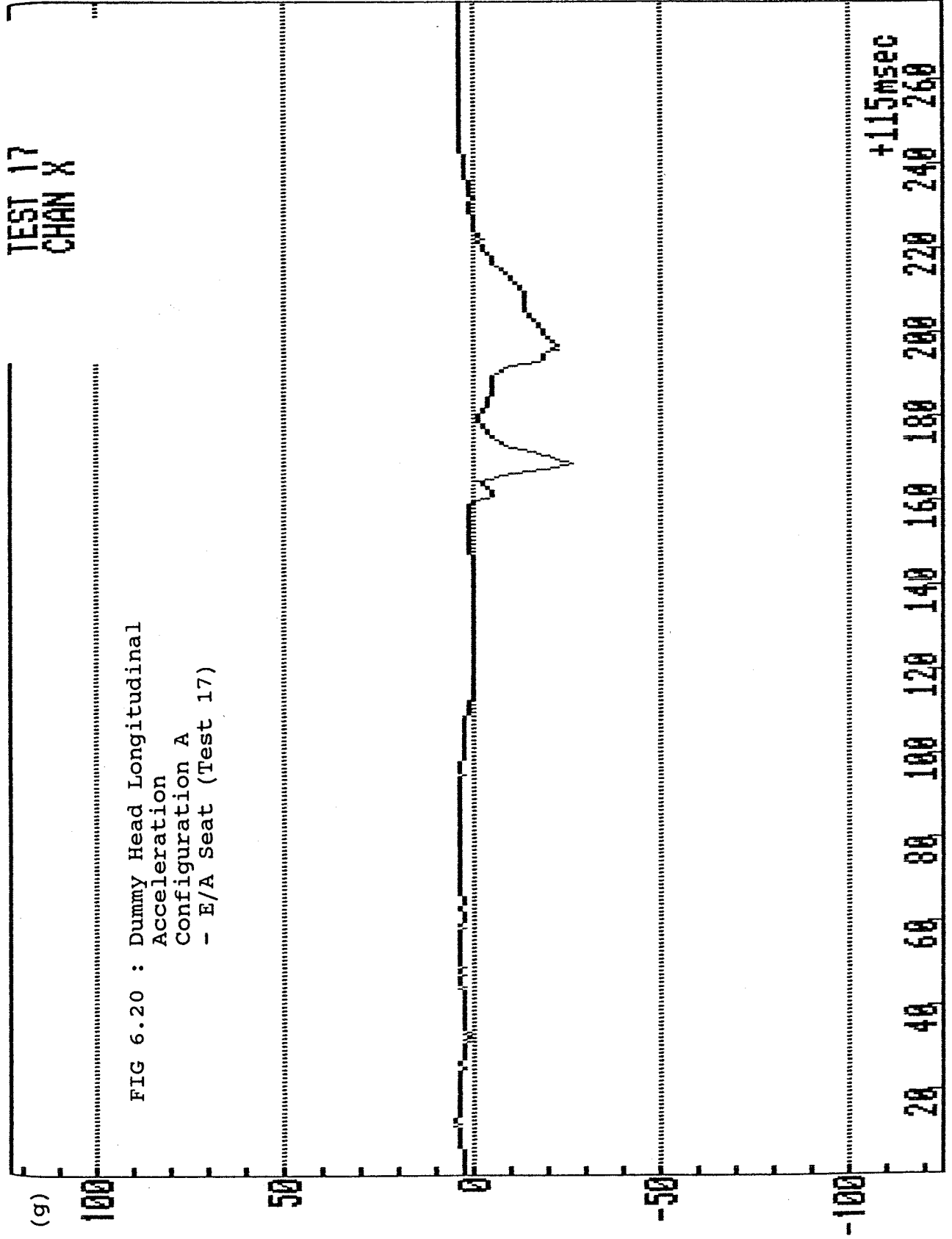
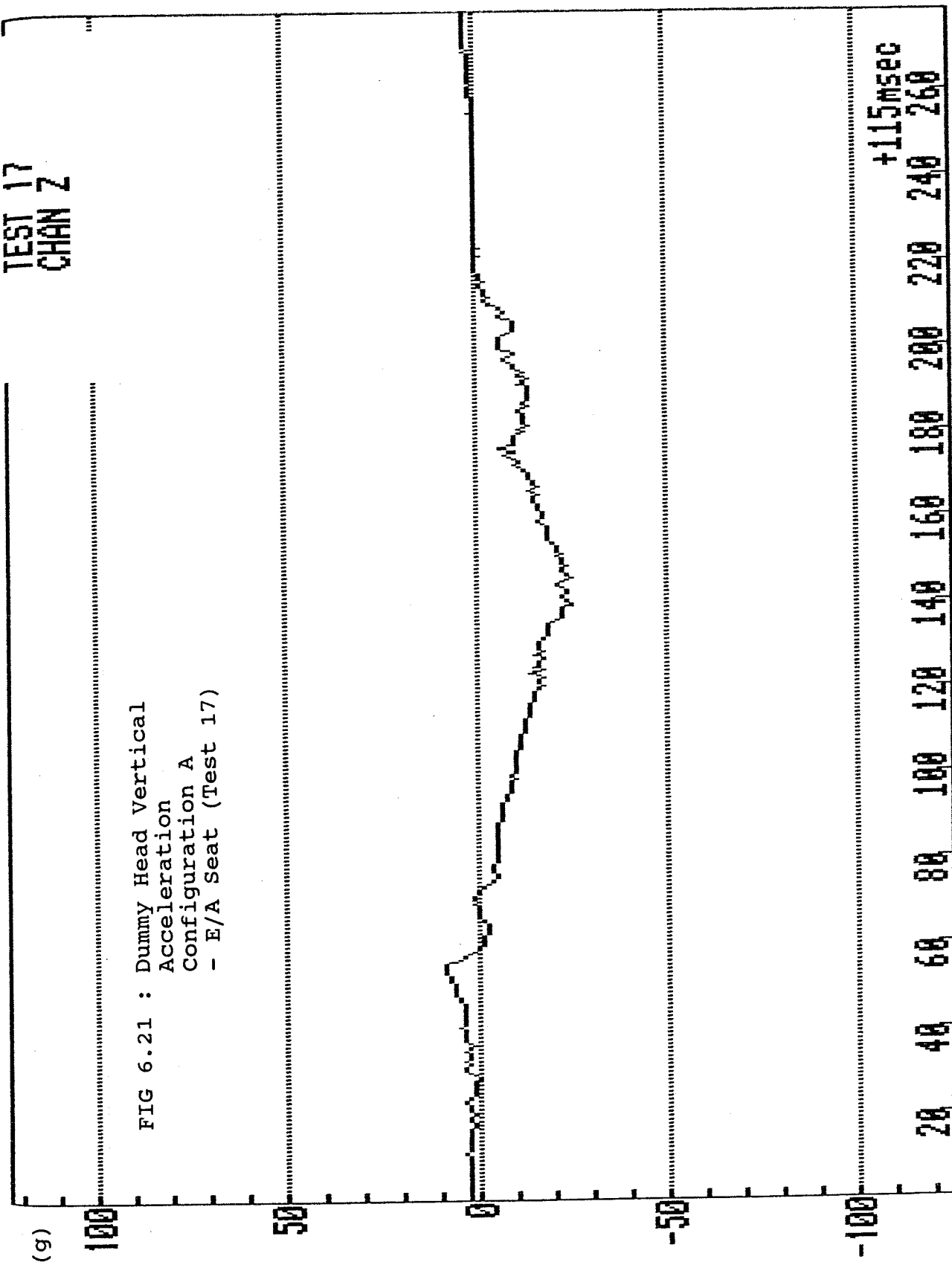


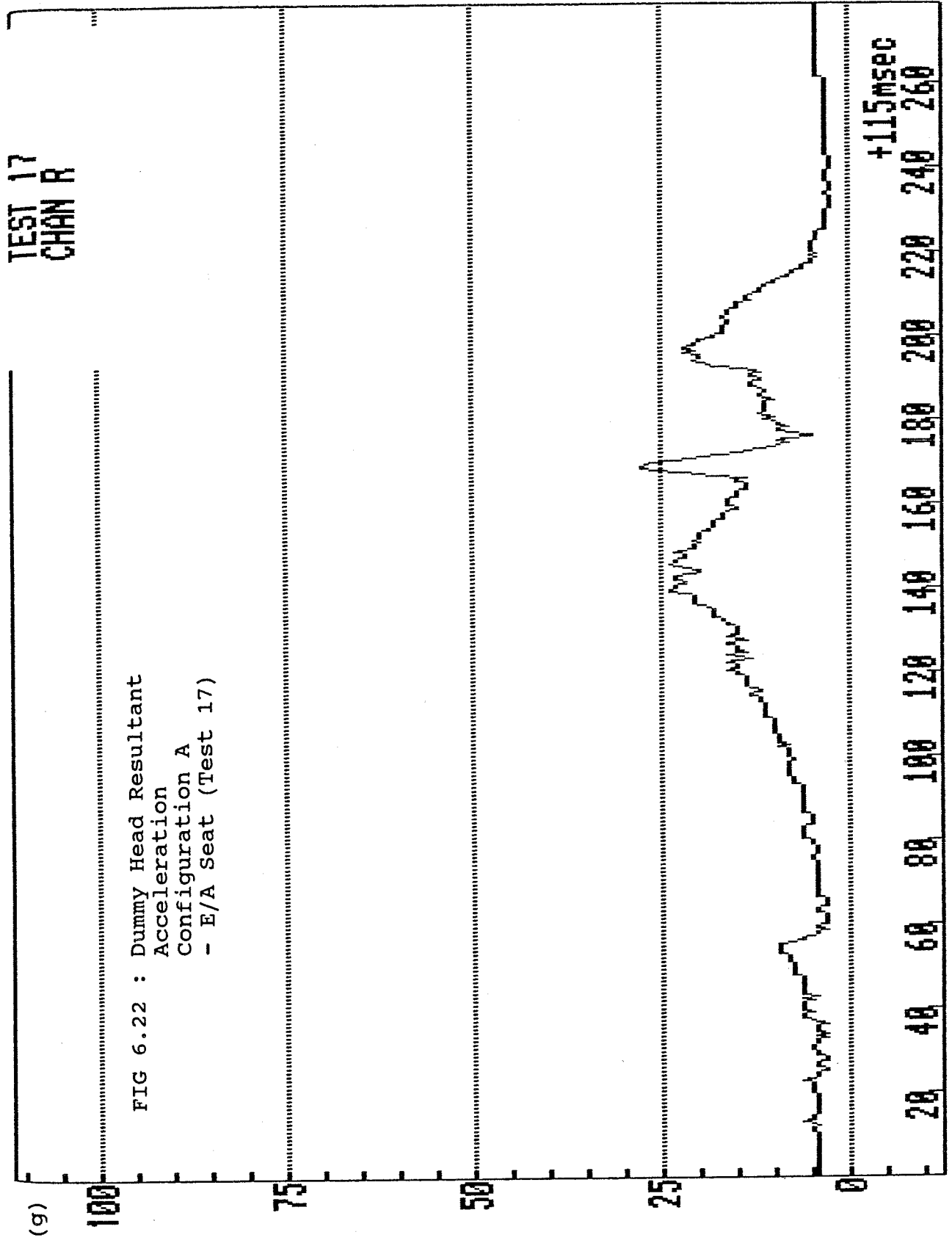
FIG 6.20 : Dummy Head Longitudinal
Acceleration
Configuration A
- E/A Seat (Test 17)

TEST 17
CHAN Z

FIG 6.21 : Dummy Head Vertical
Acceleration
Configuration A
- E/A Seat (Test 17)



+115msec



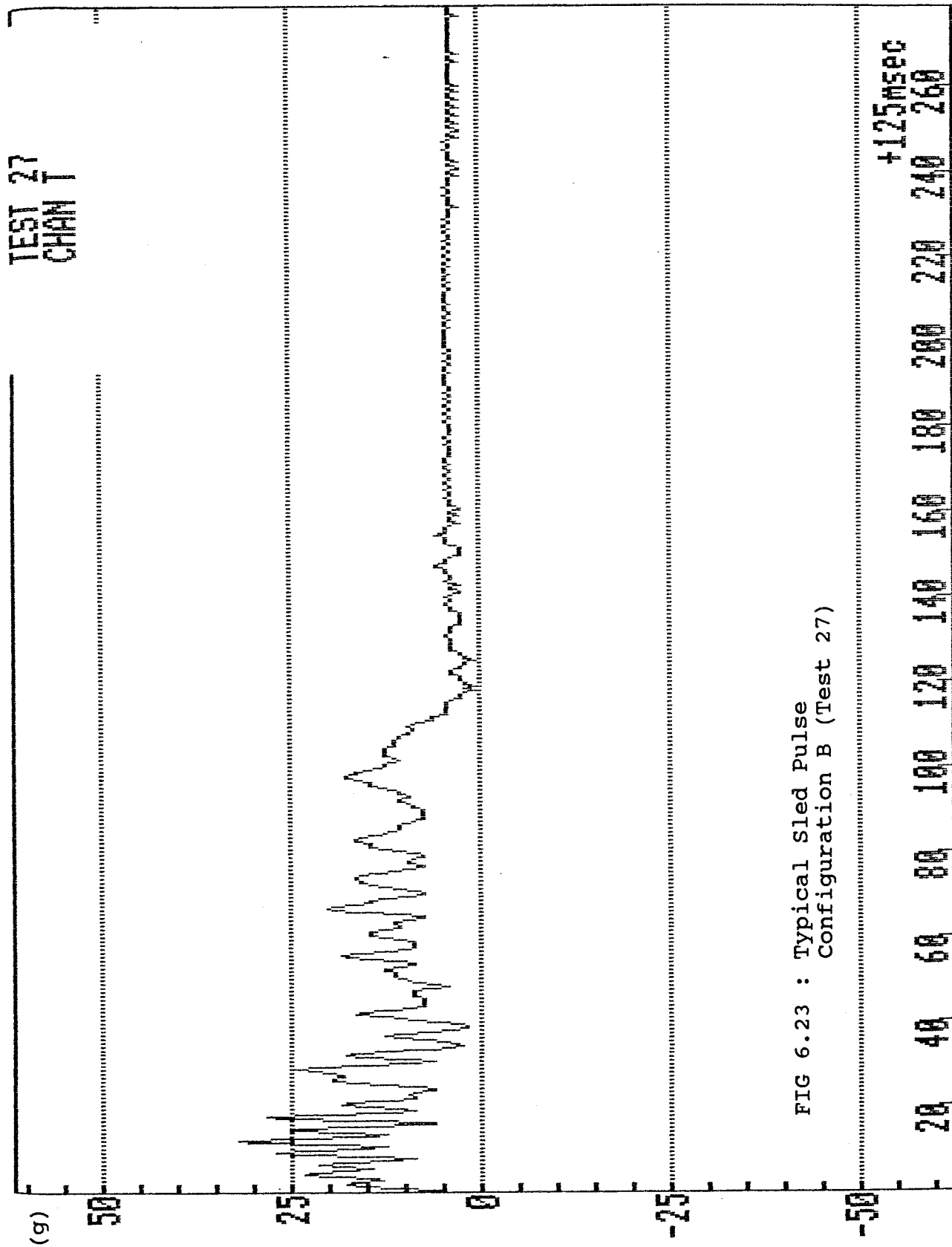
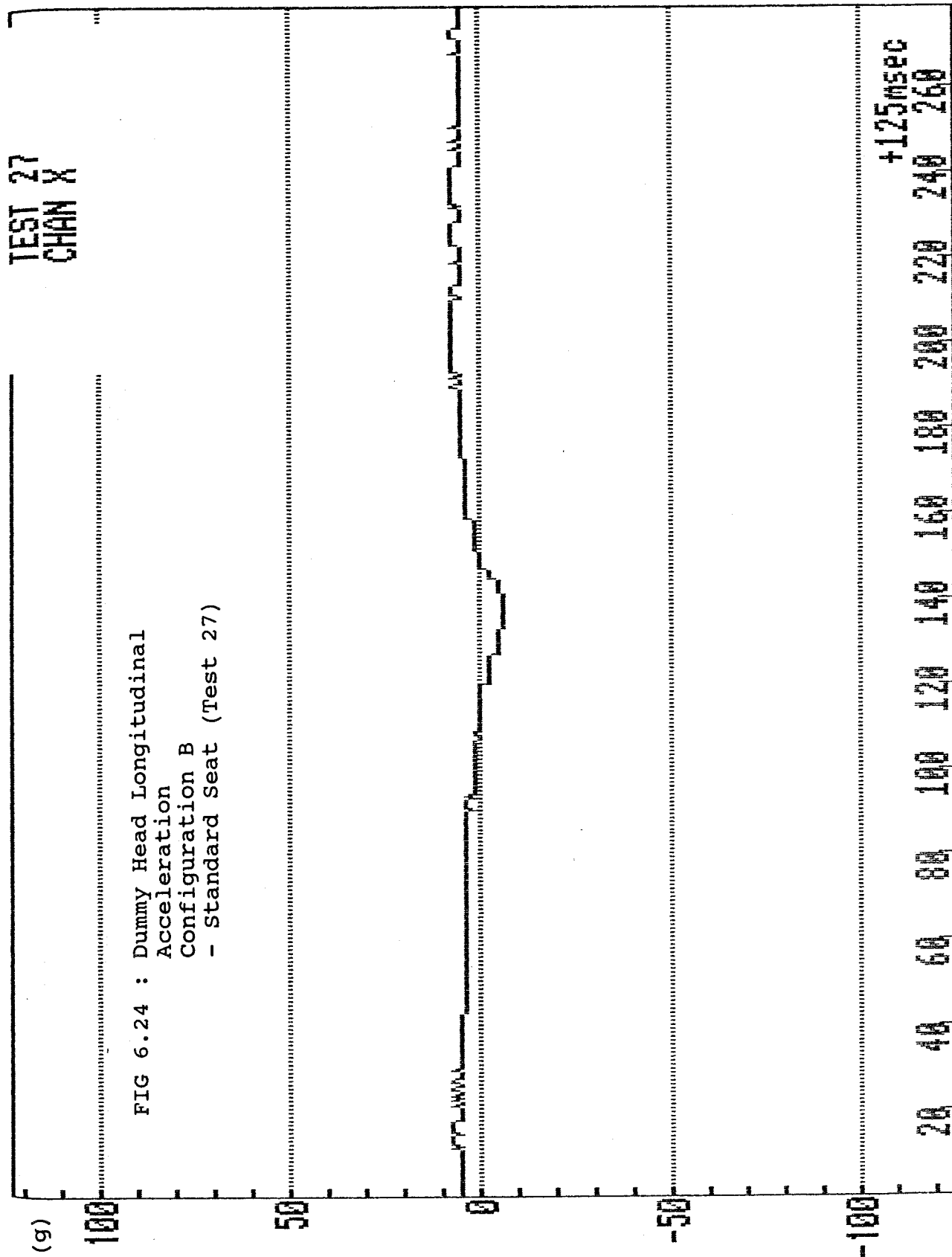
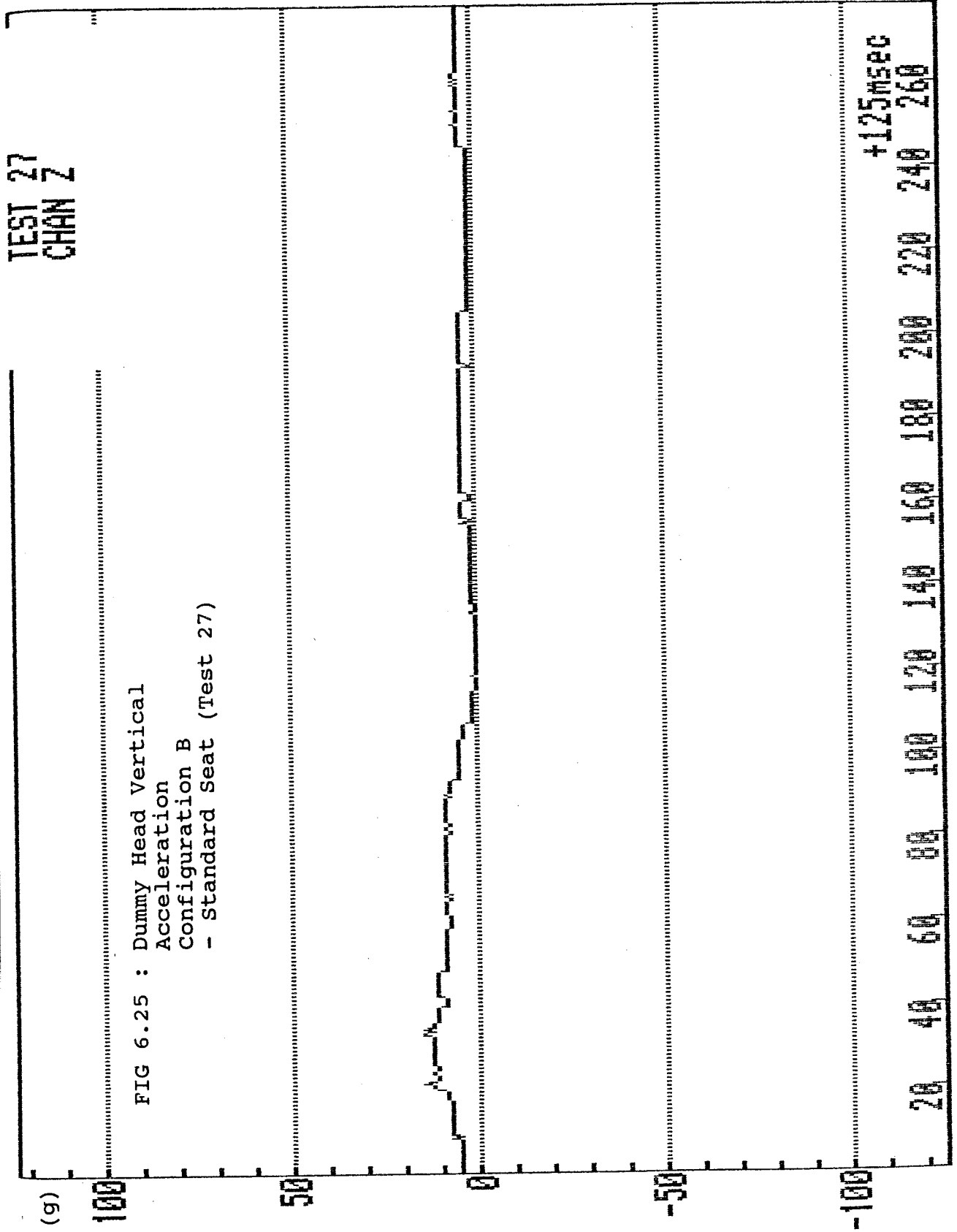


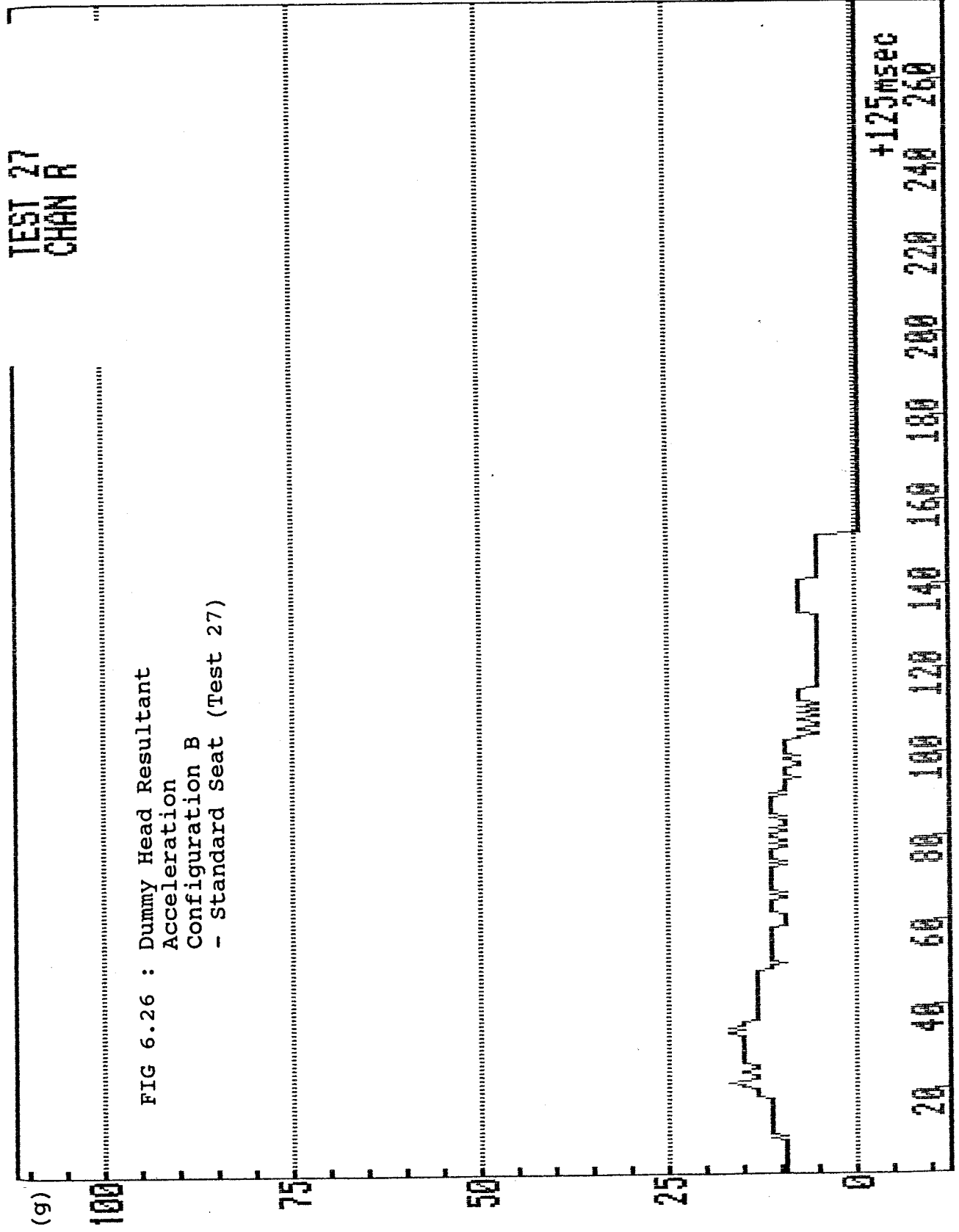
FIG 6.23 : Typical Sled Pulse
Configuration B (Test 27)



TEST 27
CHAN Z

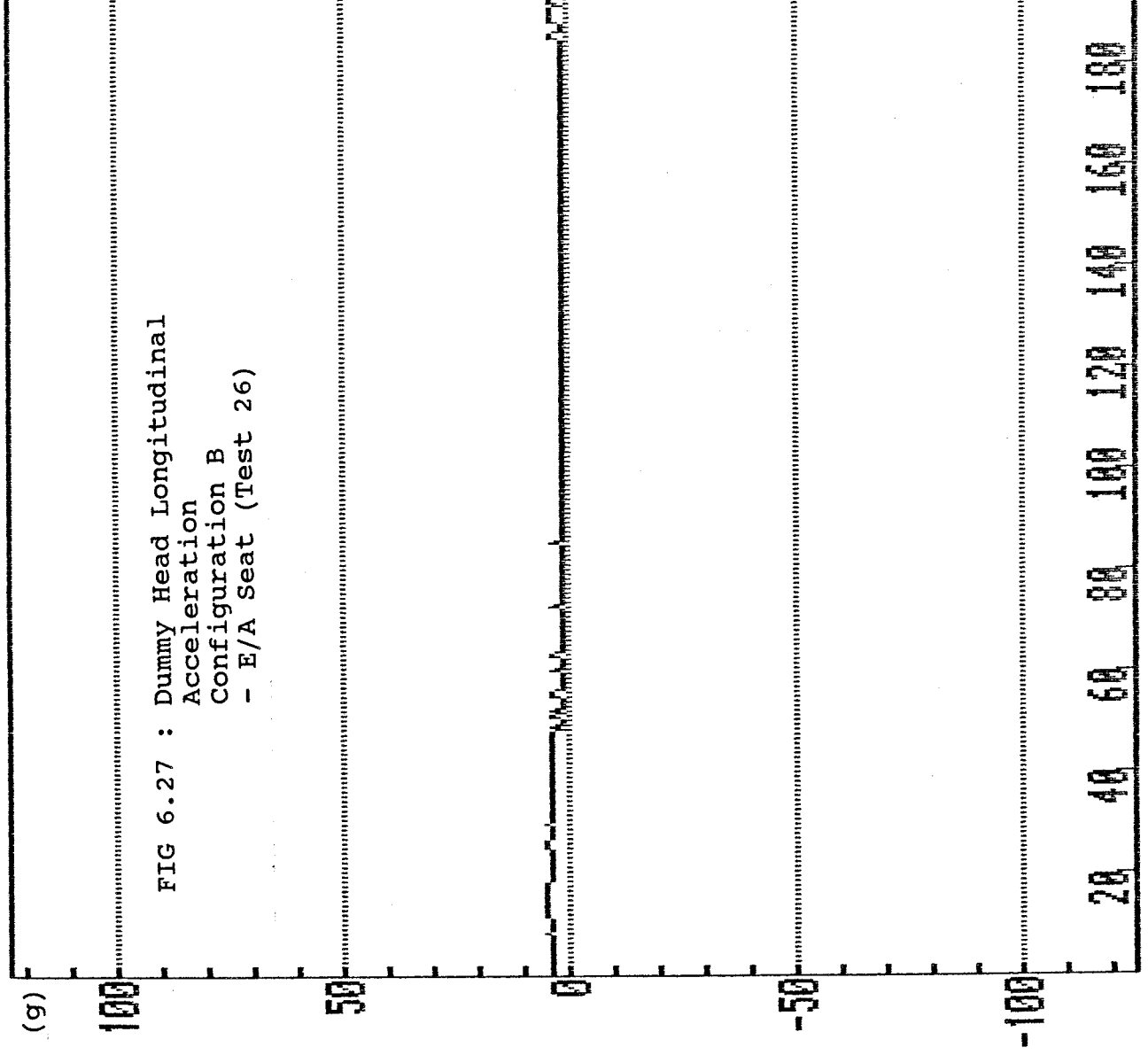
FIG 6.25 : Dummy Head Vertical
Acceleration
Configuration B
- Standard Seat (Test 27)





TEST 26
CHAN X

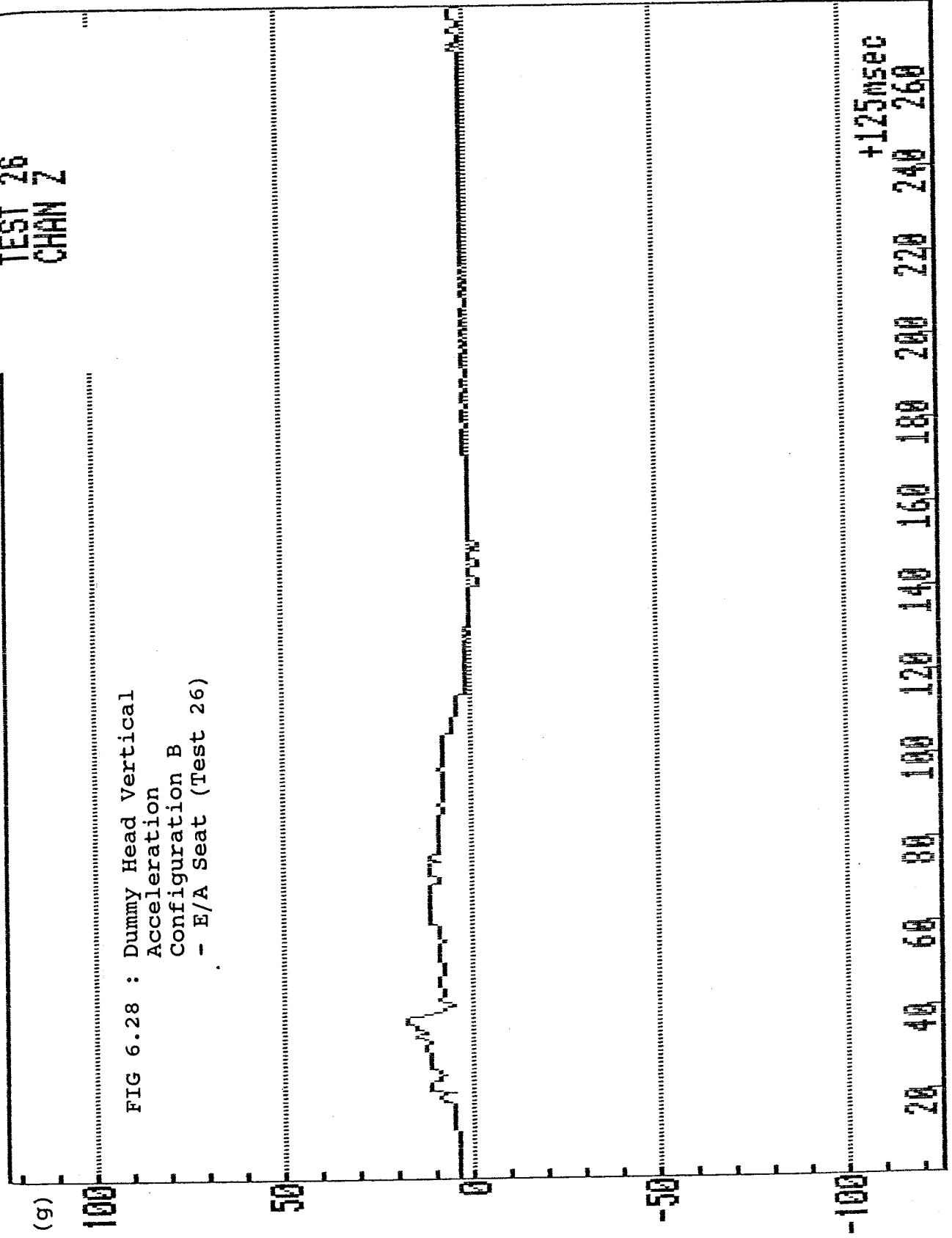
FIG 6.27 : Dummy Head Longitudinal
Acceleration
Configuration B
- E/A Seat (Test 26)



+125msec

TEST 26
CHAN Z

FIG 6.28 : Dummy Head Vertical
Acceleration
Configuration B
- E/A Seat (Test 26)



TEST 26
CHAN R

FIG 6.29 : Dummy Head Resultant
Acceleration
Configuration B
- E/A Seat (Test 26)

(g)

100

75

50

25

0

20

40

60

80

100

120

140

160

180

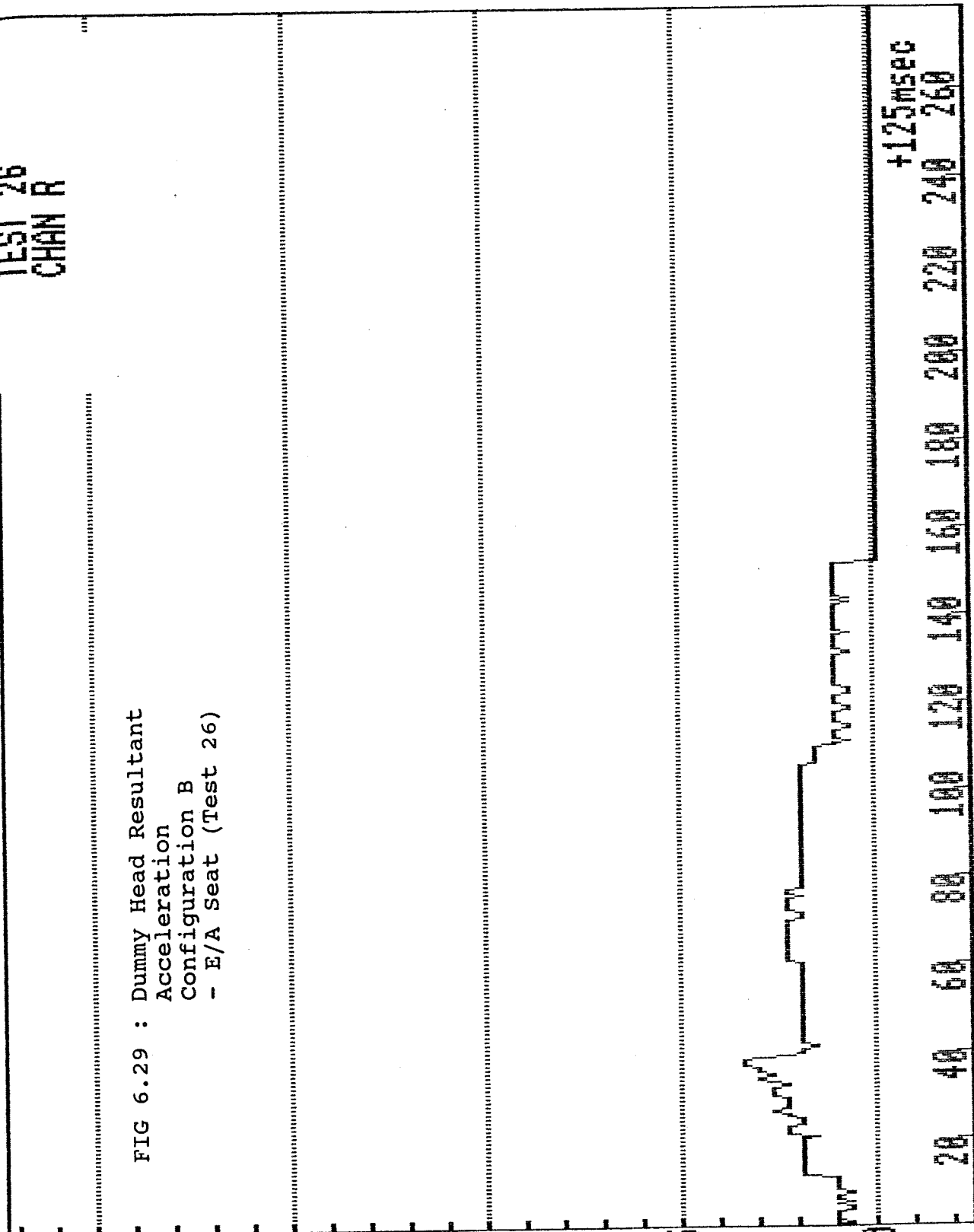
200

220

240

260

+125msec



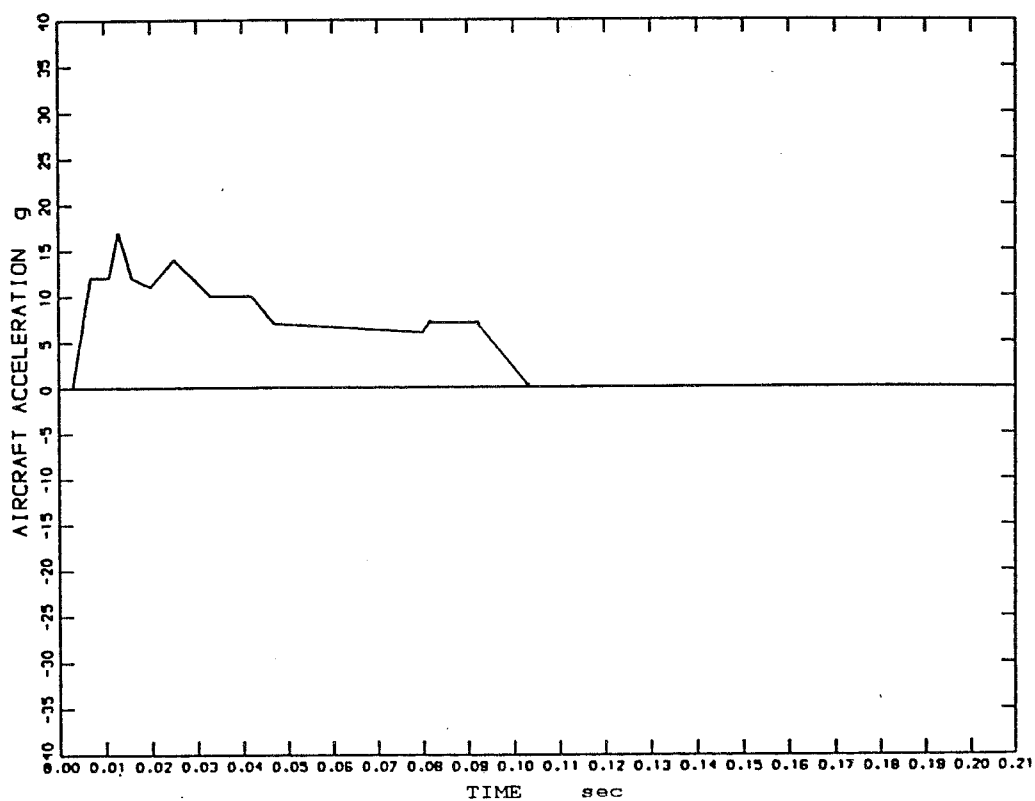
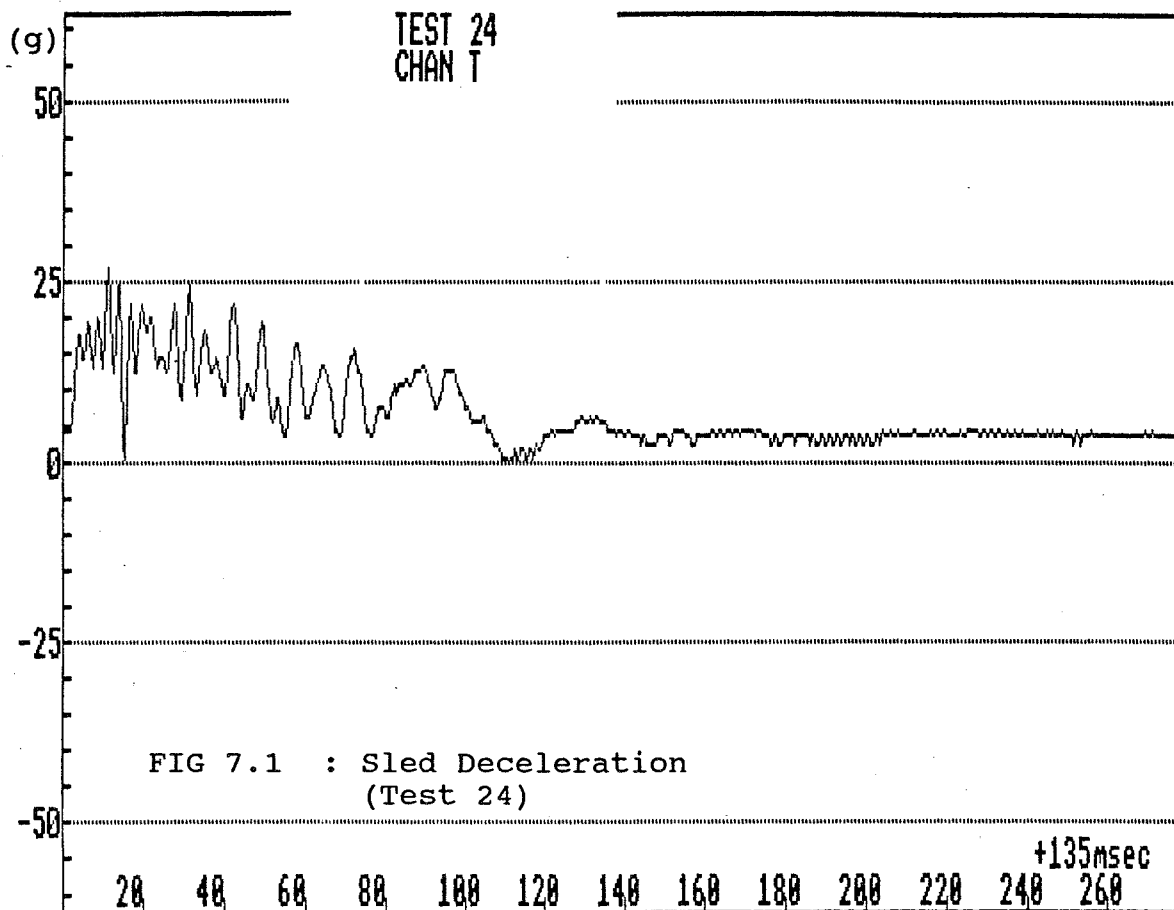
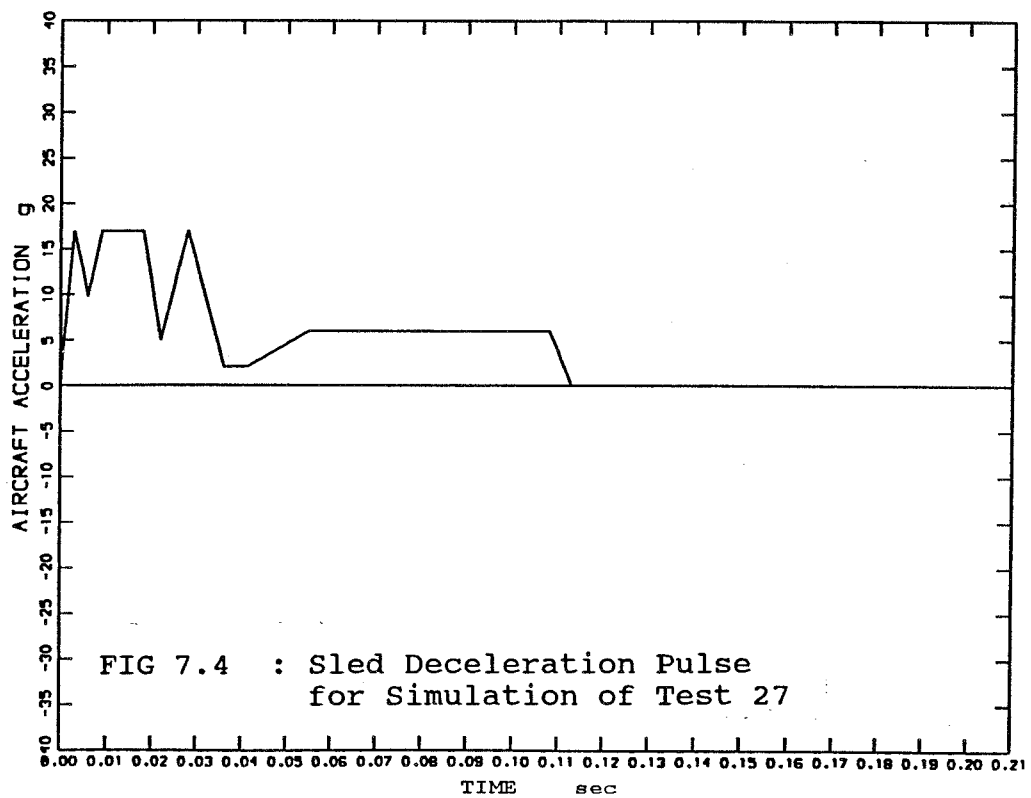
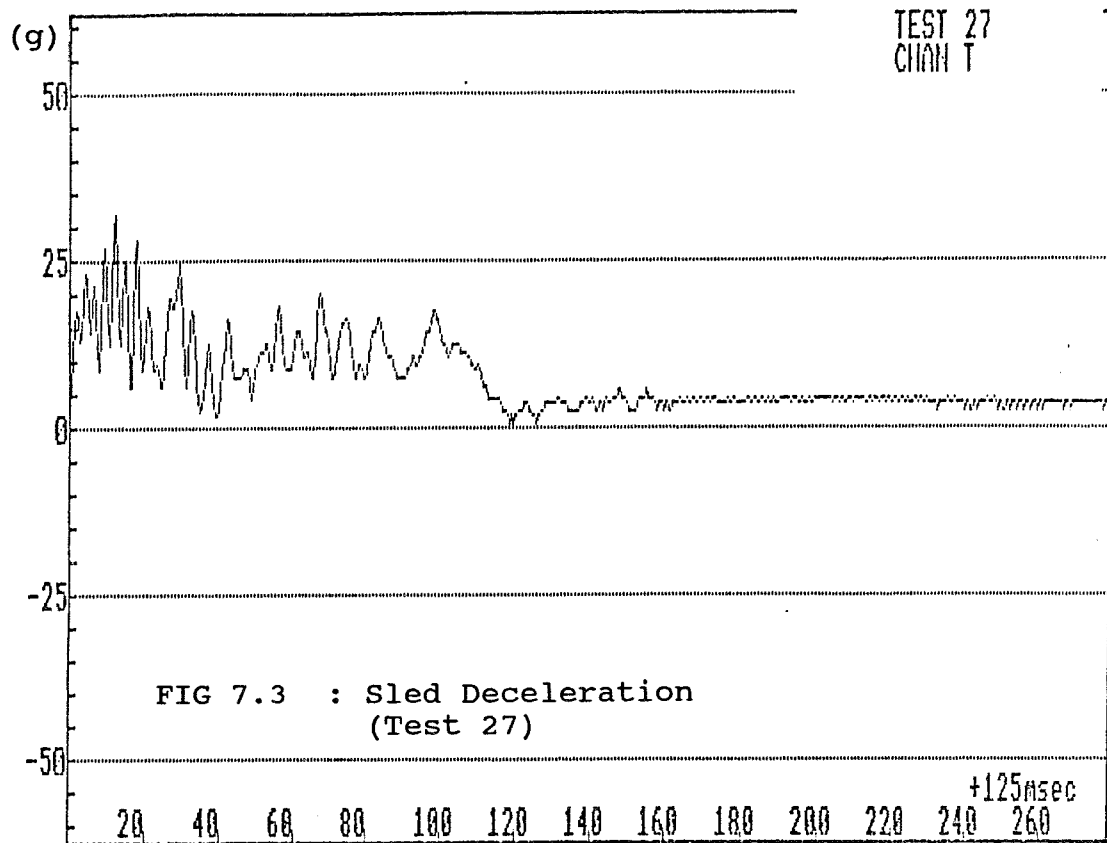


FIG 7.2 : Sled Deceleration Pulse
for Simulation of Test 24



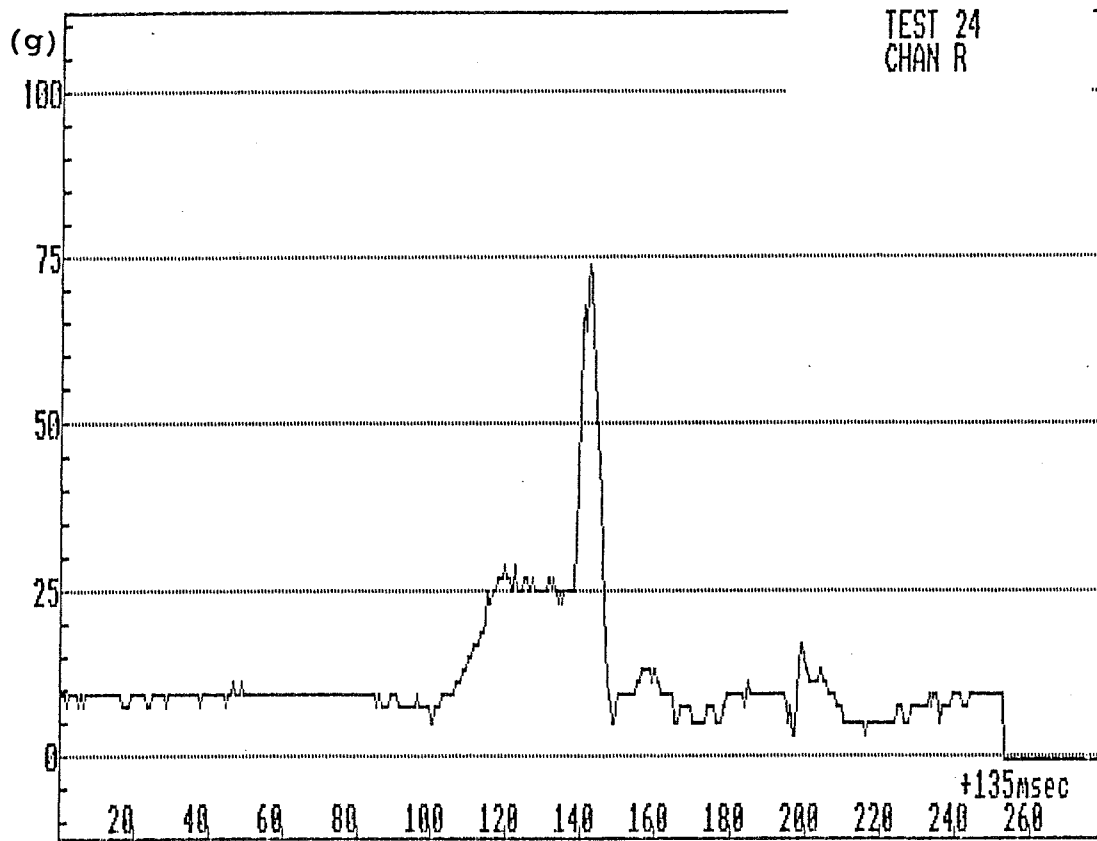


FIG 7.5 : Dummy Head Resultant
Acceleration (Test 24)

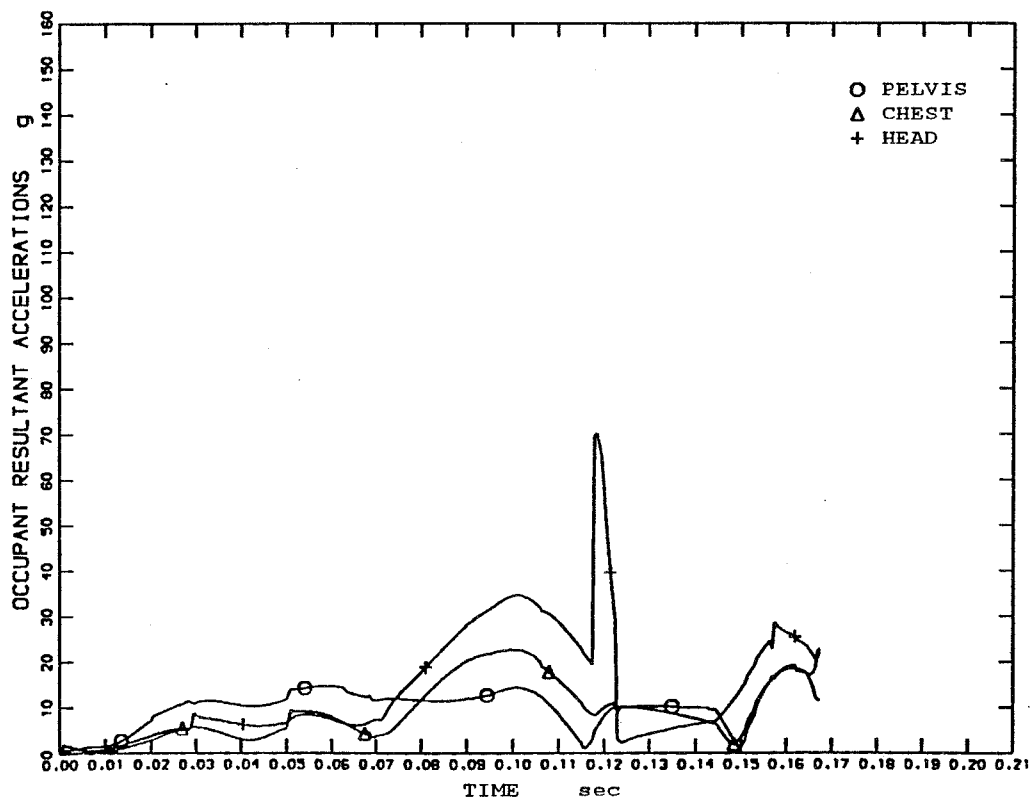


FIG 7.6 : Occupant Body Segment
Accelerations
Baseline model of Test 24

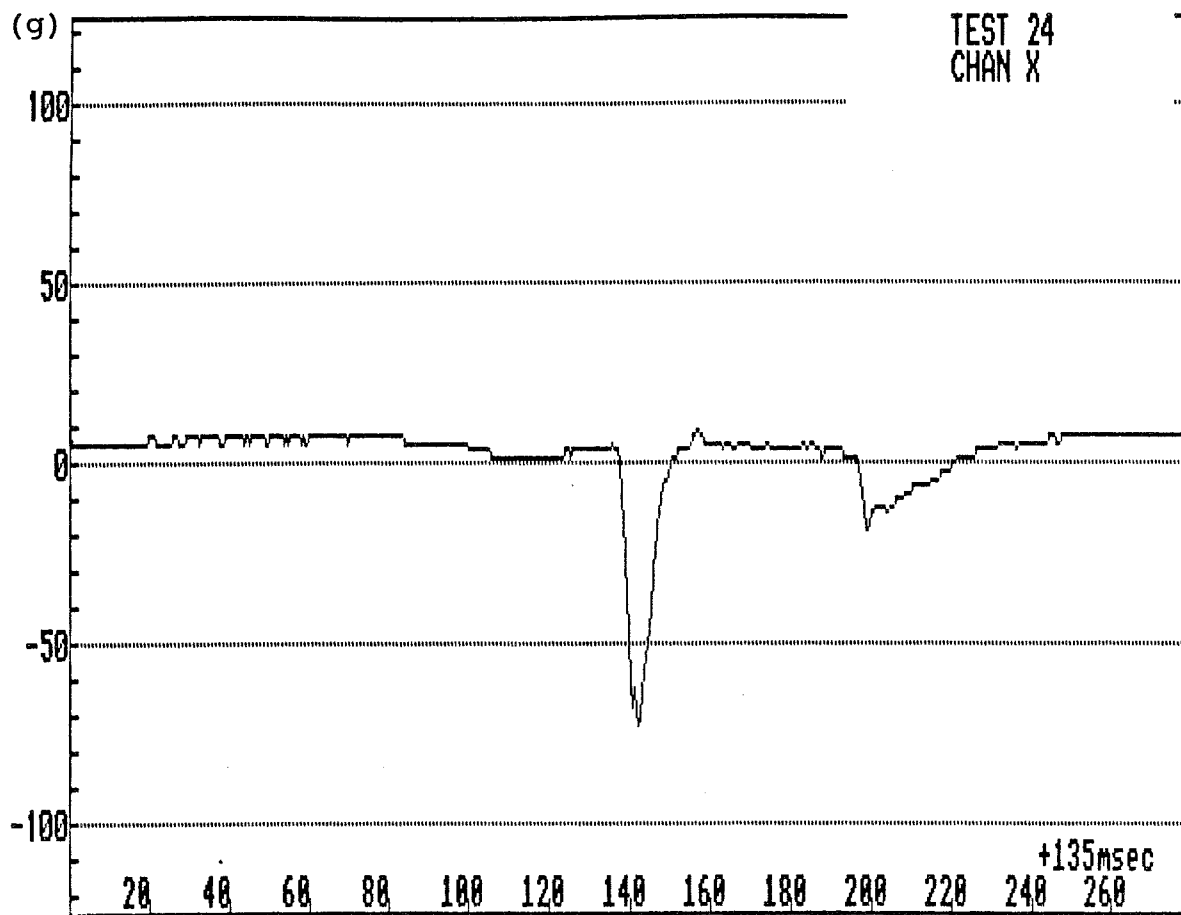


FIG 7.7 : Dummy Head Longitudinal
Acceleration (Test 24)

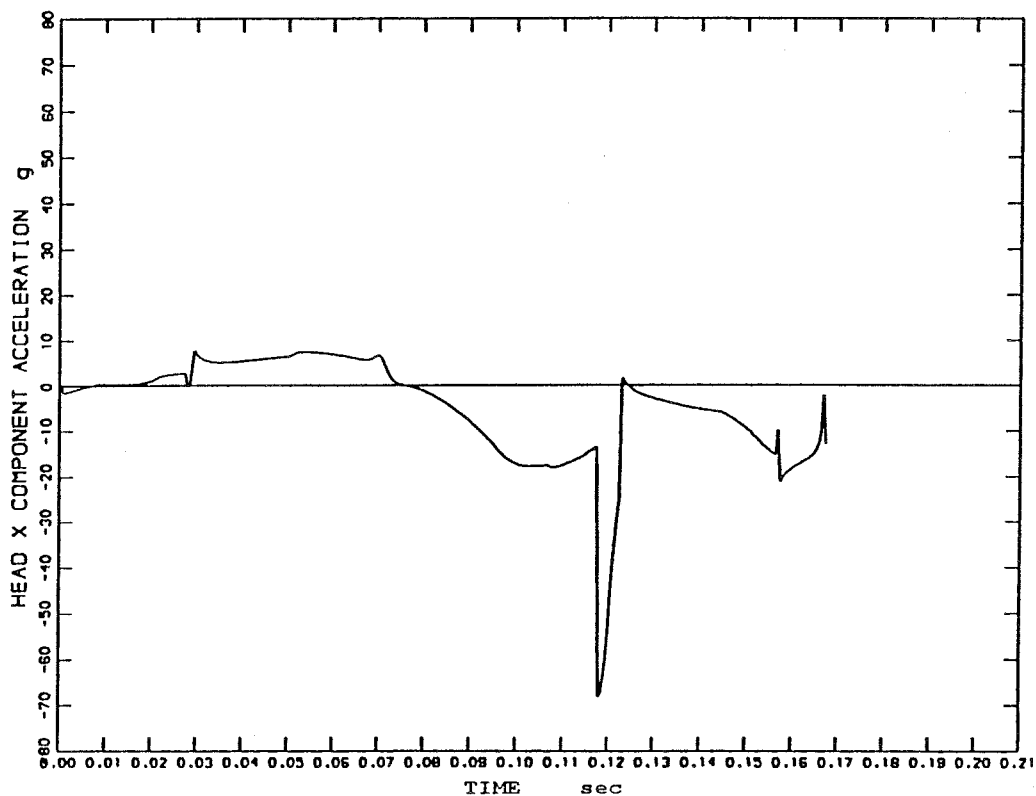
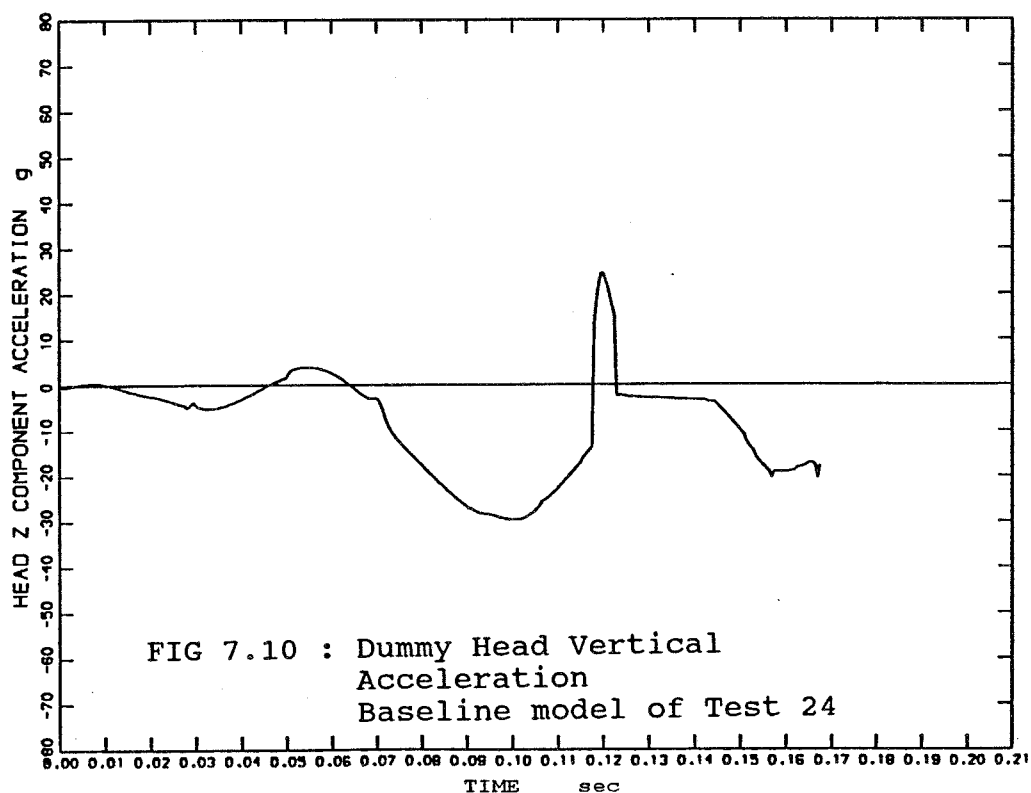
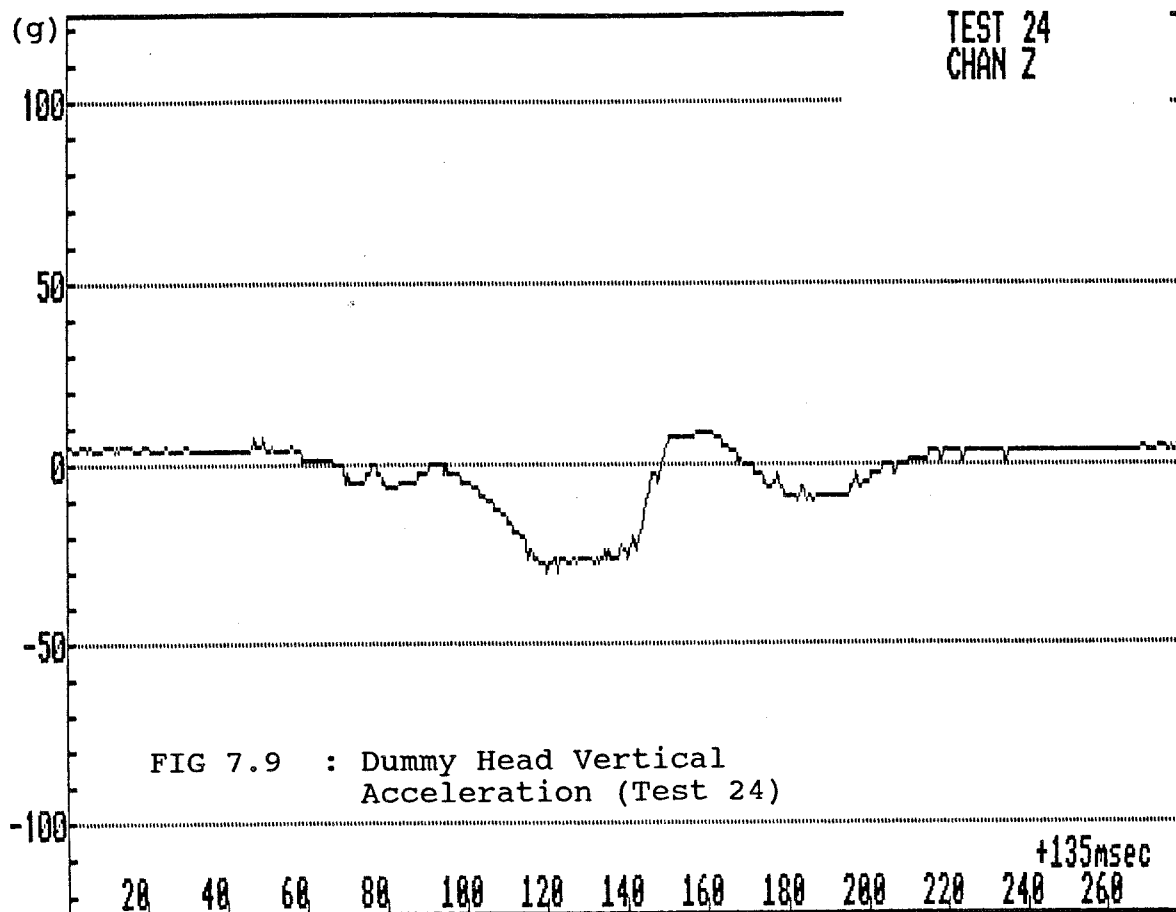


FIG 7.8 : Dummy Head Longitudinal
Acceleration
Baseline model of Test 24



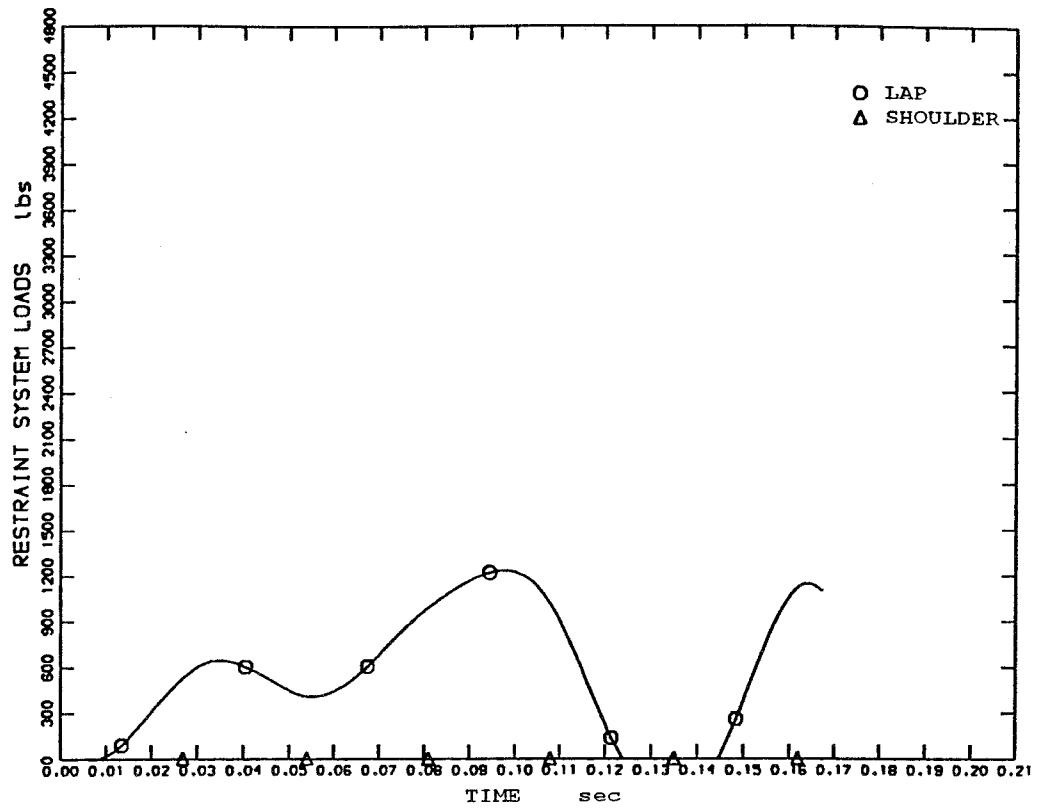


FIG 7.11 : Restraint System Forces
Baseline model of Test 24

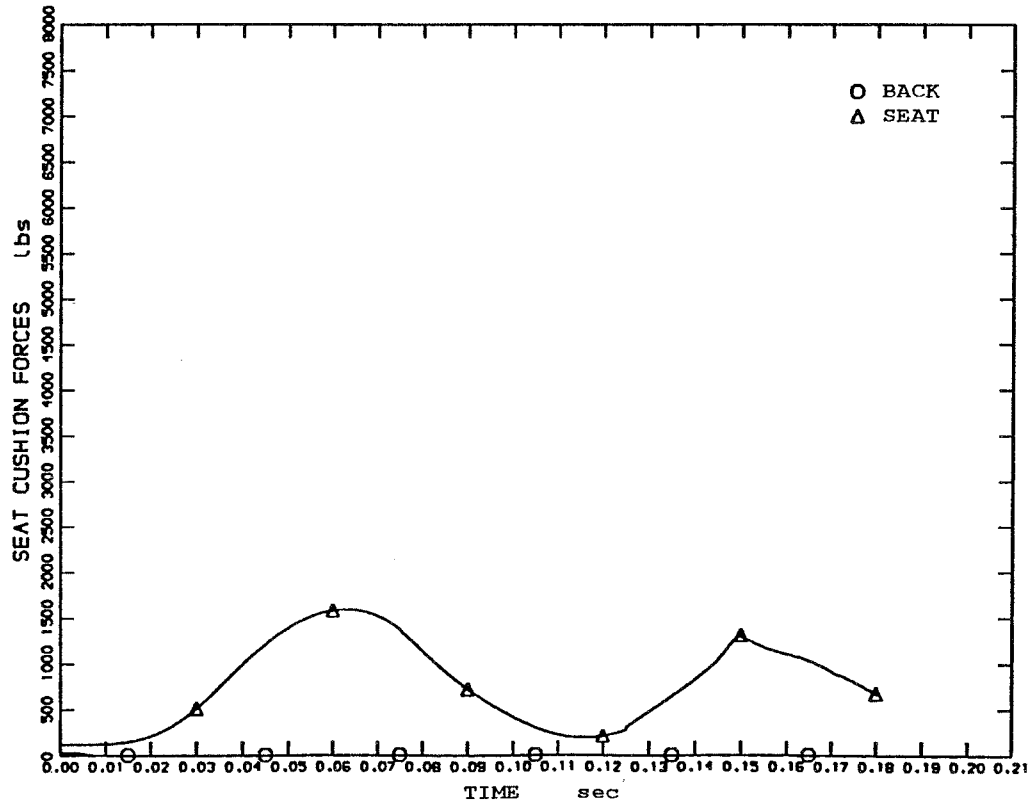


FIG 7.12 : Forces applied to seat
by occupant
Baseline model of Test 24

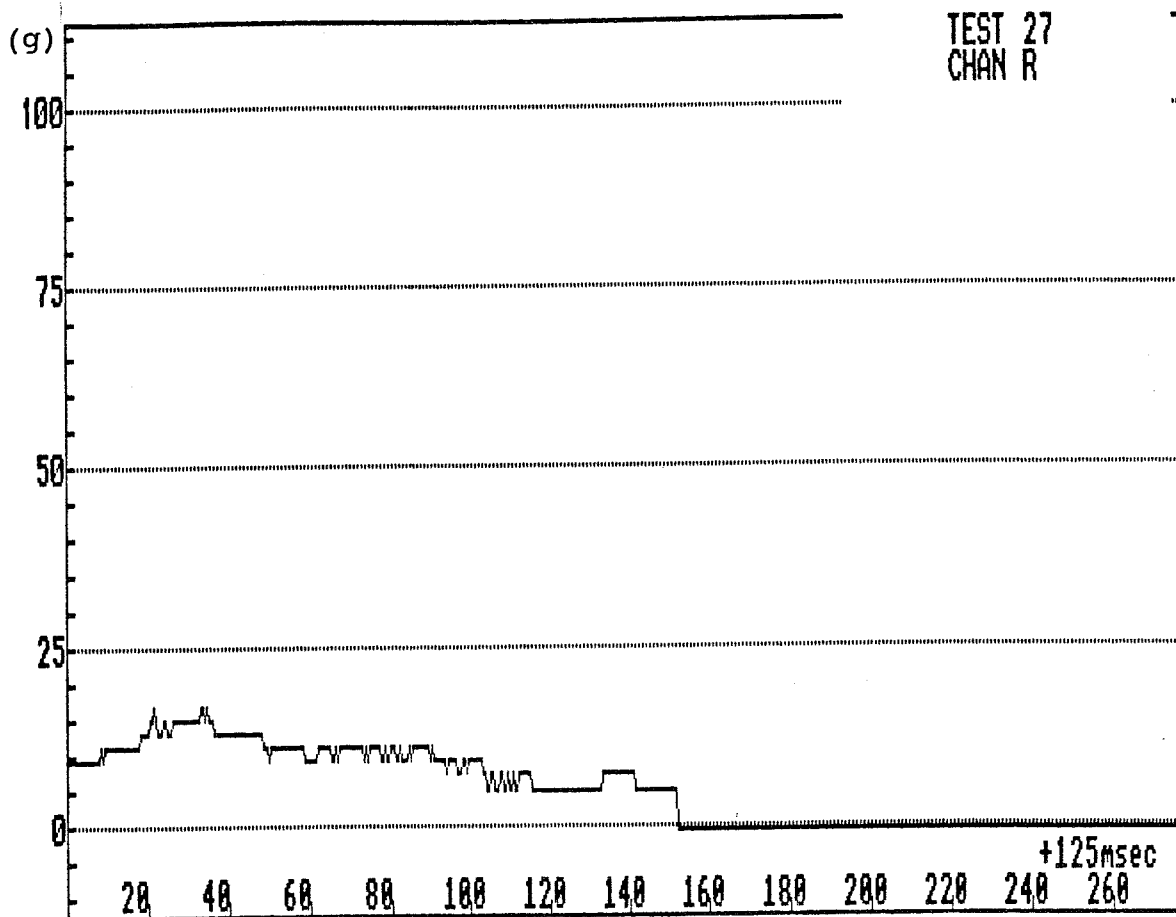


FIG 7.13 : Resultant Dummy Head
Acceleration (Test 27)

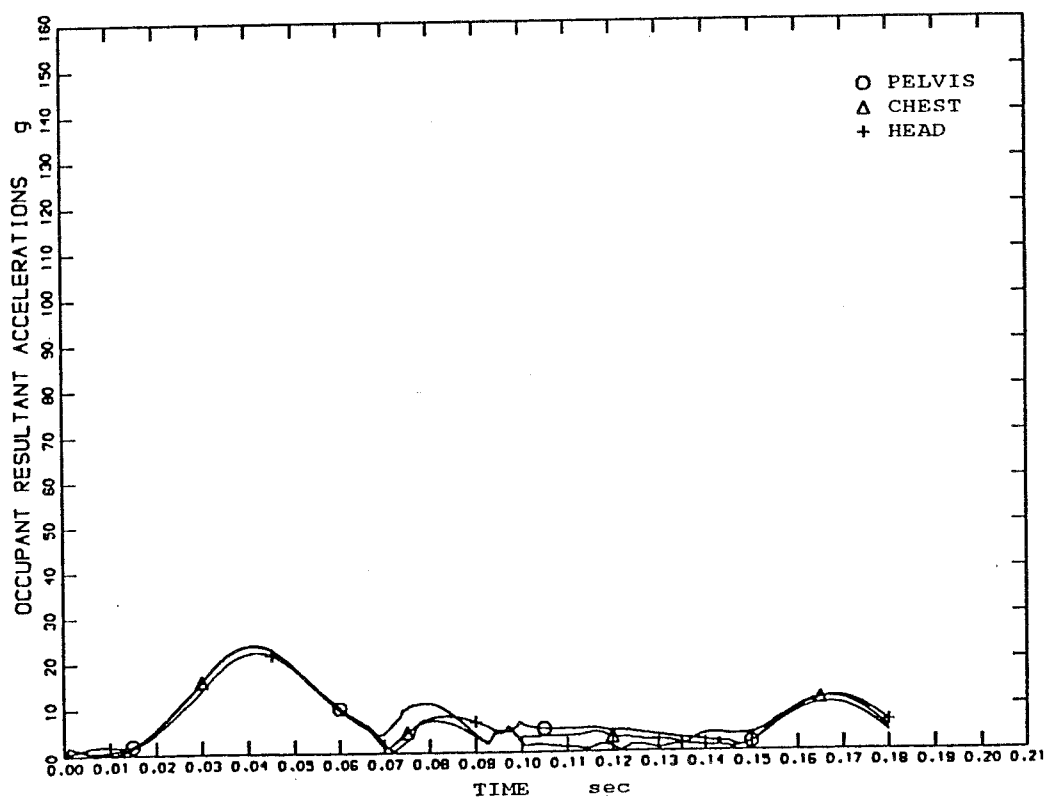


FIG 7.14 : Occupant Body Segment
Accelerations
Baseline model of Test 27

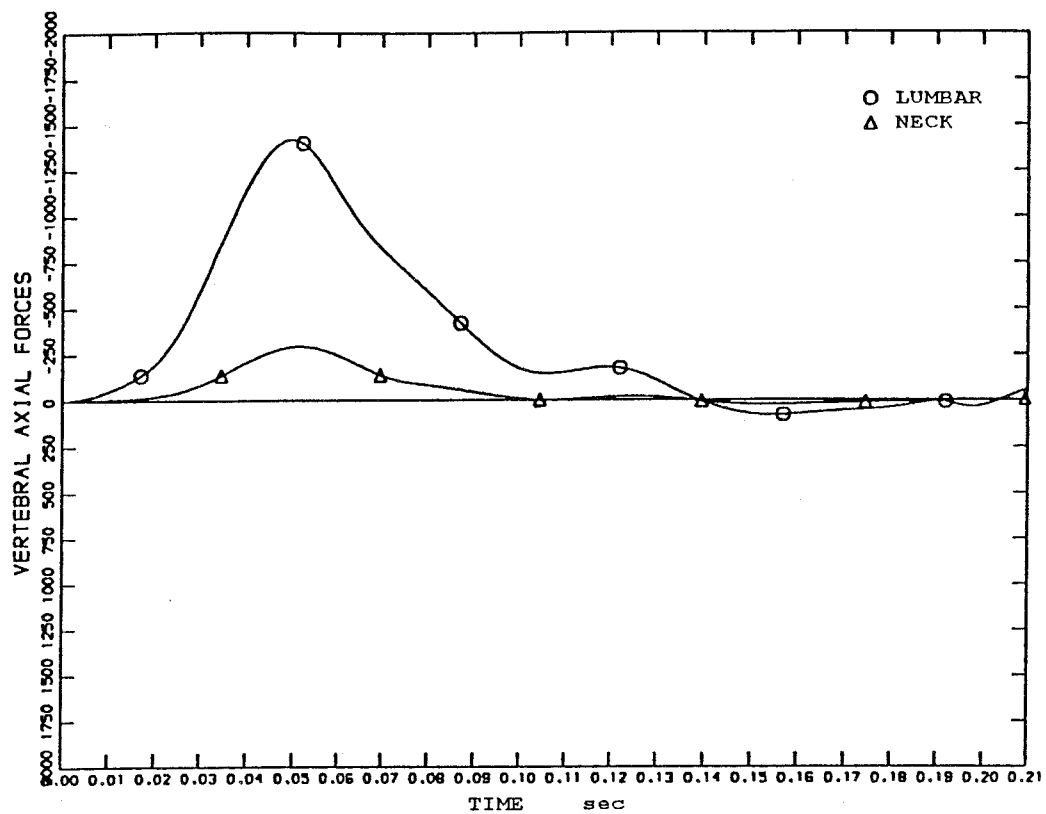


FIG 7.15 : Vertebral Loads
Baseline model of Test 27

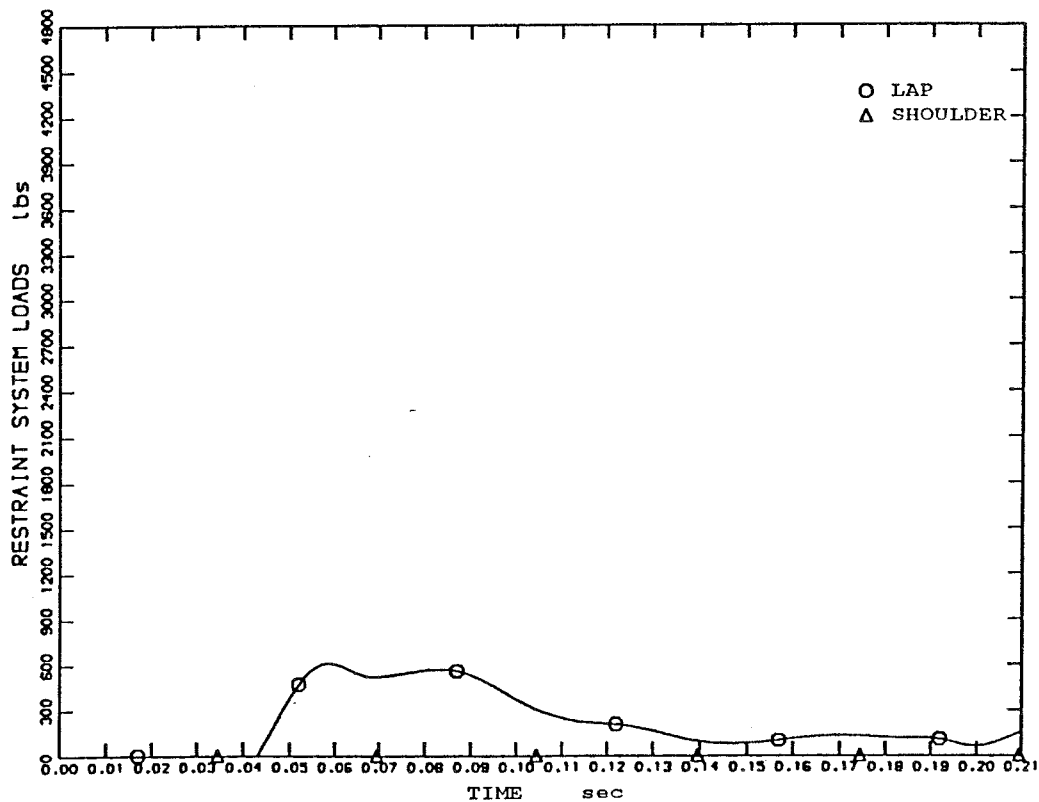


FIG 7.16 : Restraint System Forces
Baseline model of Test 27

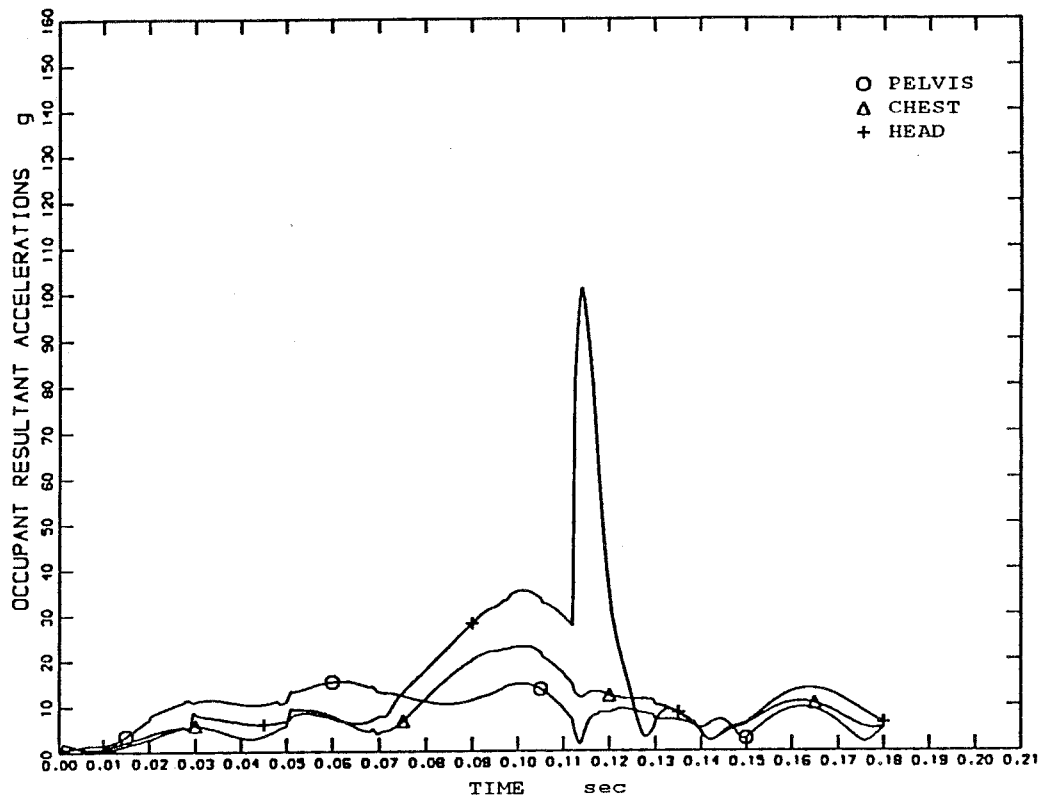


FIG 7.17 : Occupant Body Segment
Accelerations
28" Seat Pitch

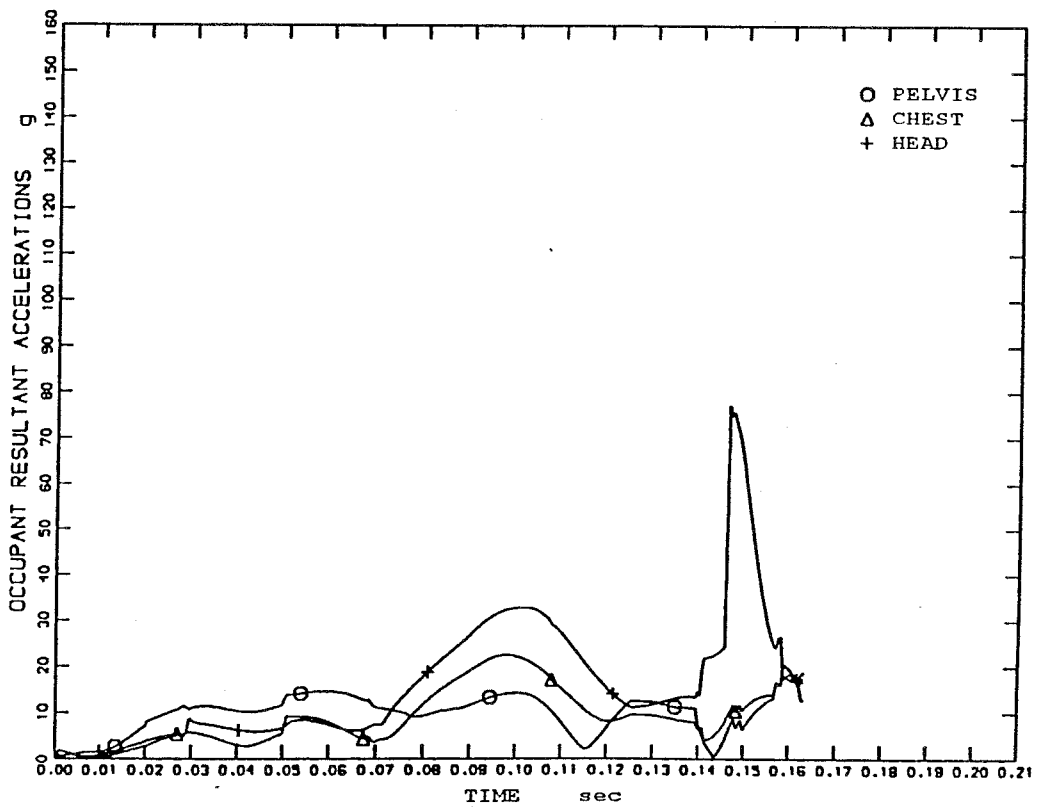


FIG 7.18 : Occupant Body Segment
Accelerations
32" Seat Pitch

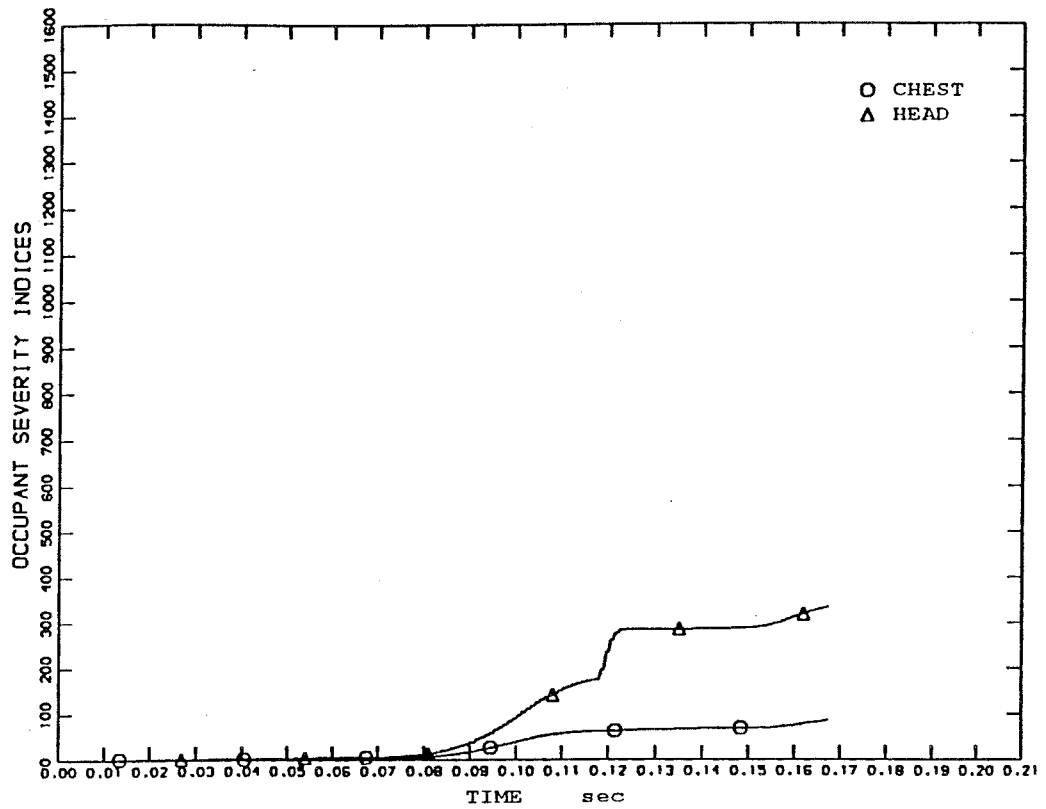


FIG 7.19 : Severity Indices
Baseline model
(30" Seat Pitch)

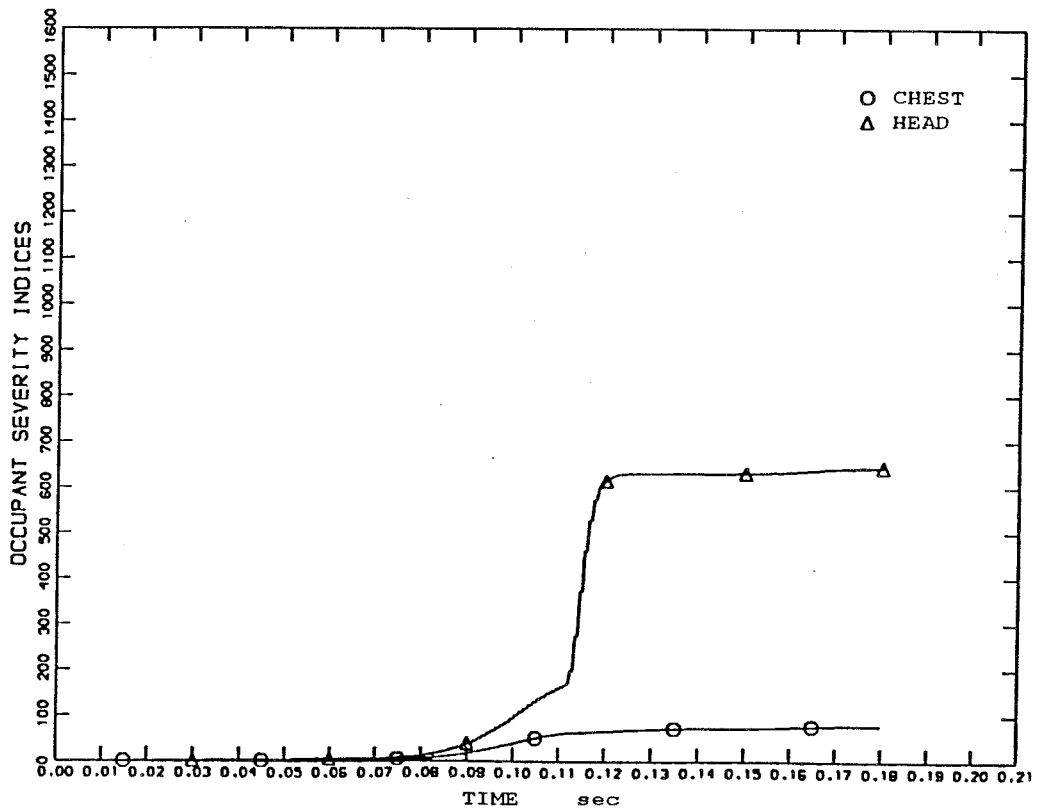


FIG 7.20 : Severity Indices
28" Seat Pitch

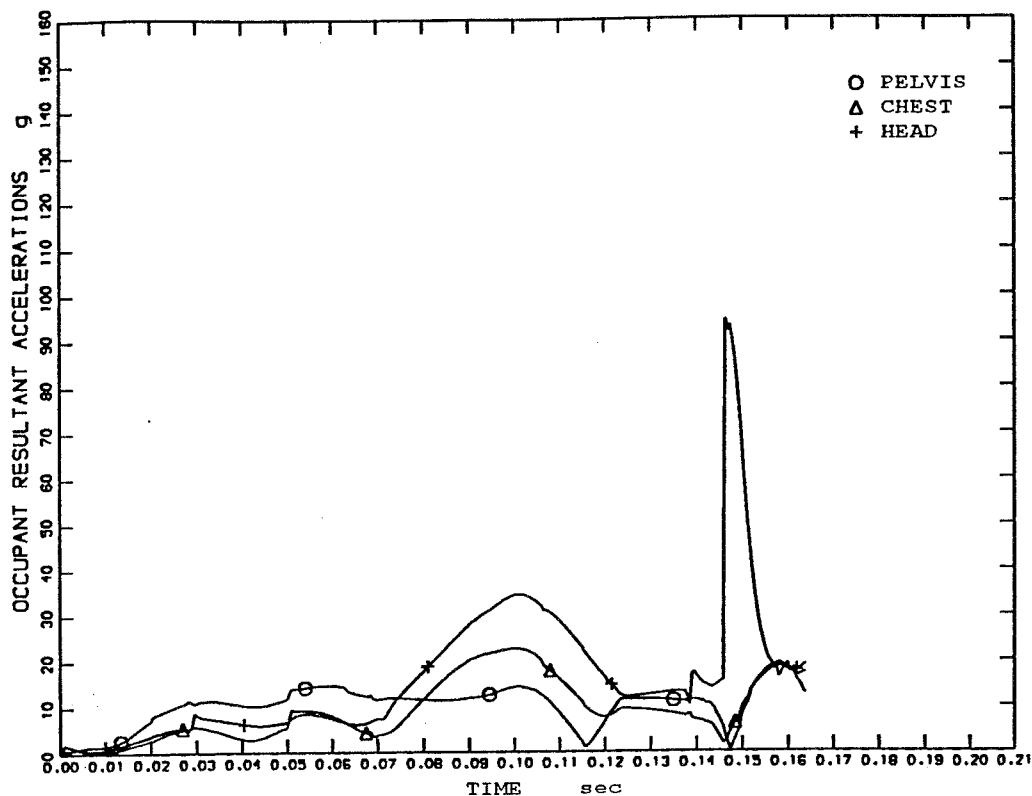


FIG 7.21 : Occupant Body Segment
Accelerations
35ft-lb Breakover

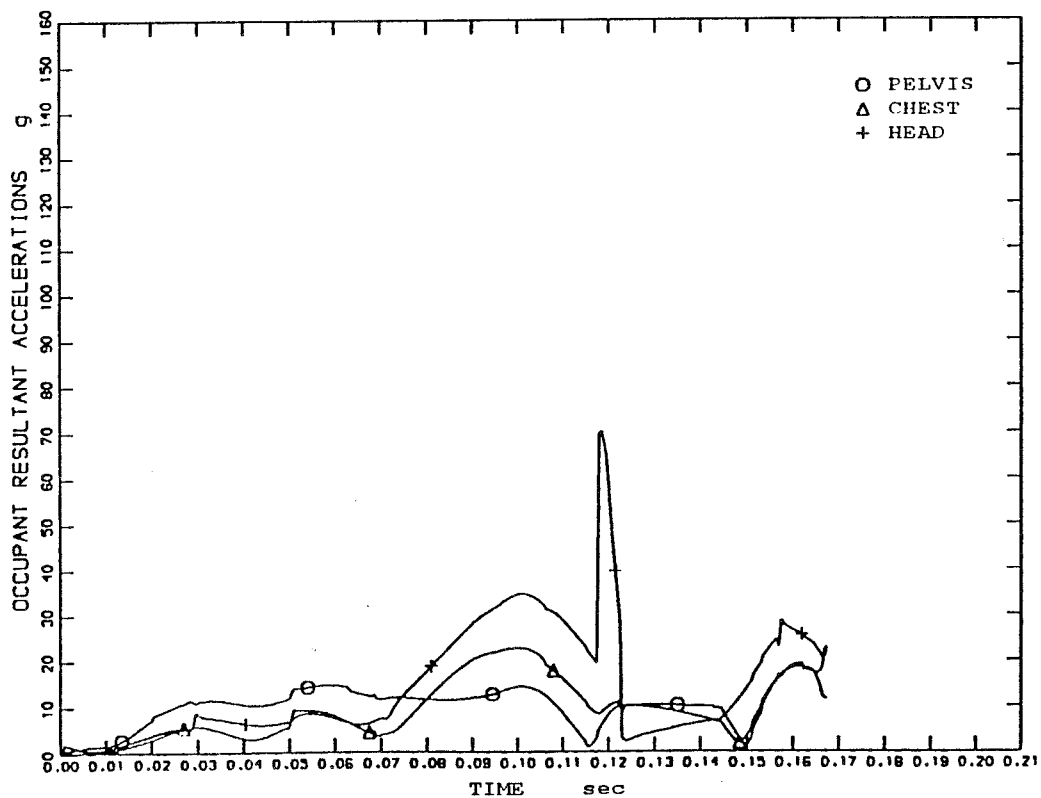


FIG 7.22 : Occupant Body Segment
Accelerations 40ft-lb
Breakover (Baseline)

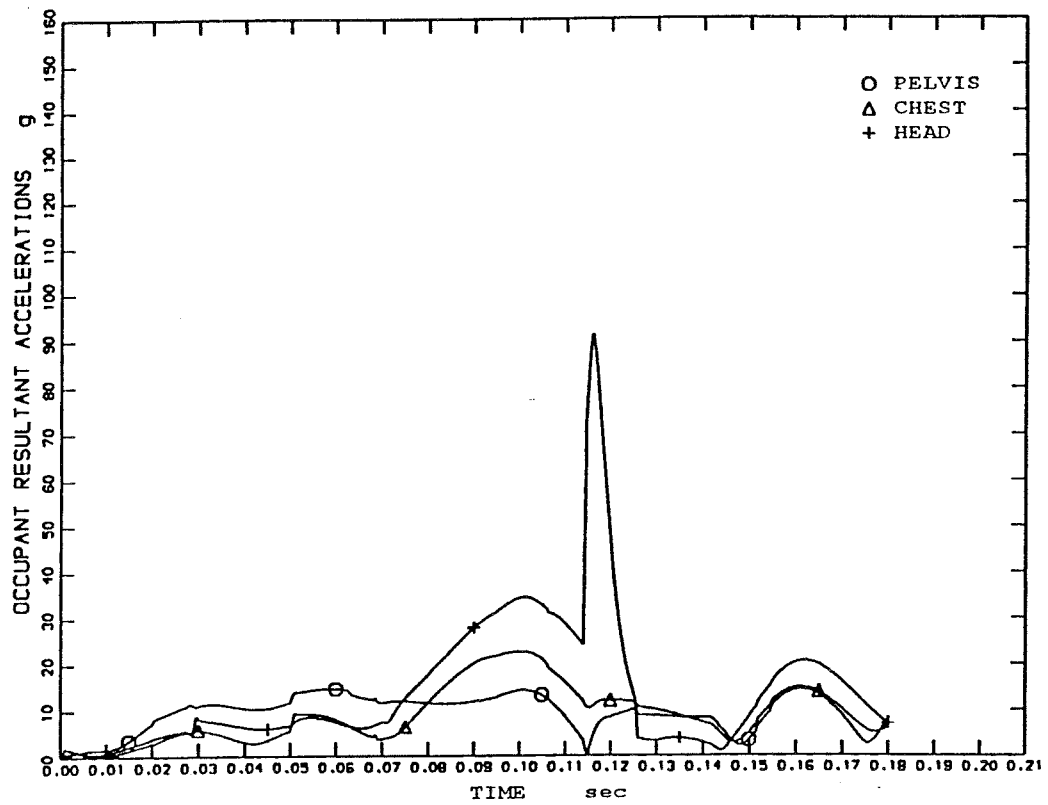


FIG 7.23 : Occupant Body Segment Accelerations
45ft-lb Breakover

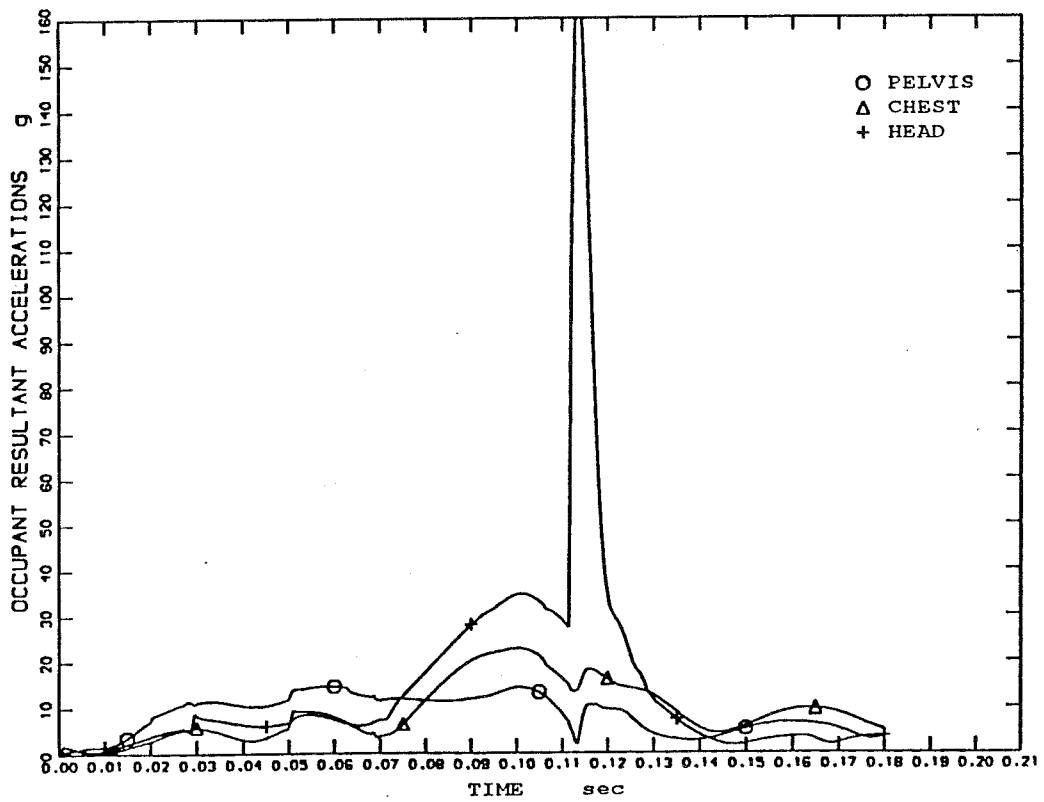


FIG 7.24 : Occupant Body Segment Accelerations
Restricted Breakover

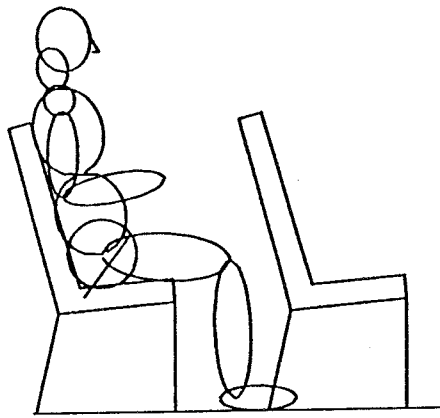


FIG 7.25 : Occupant Initial Position
Case 1

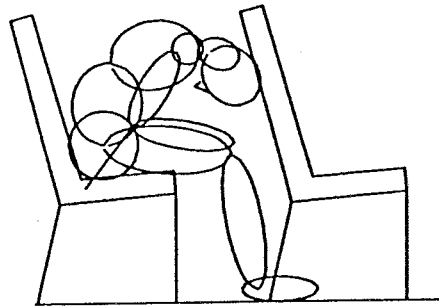


FIG 7.26 : Occupant Initial Position
Case 2

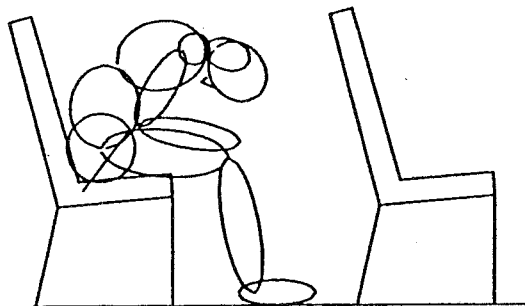


FIG 7.27 : Occupant Initial Position
Case 3

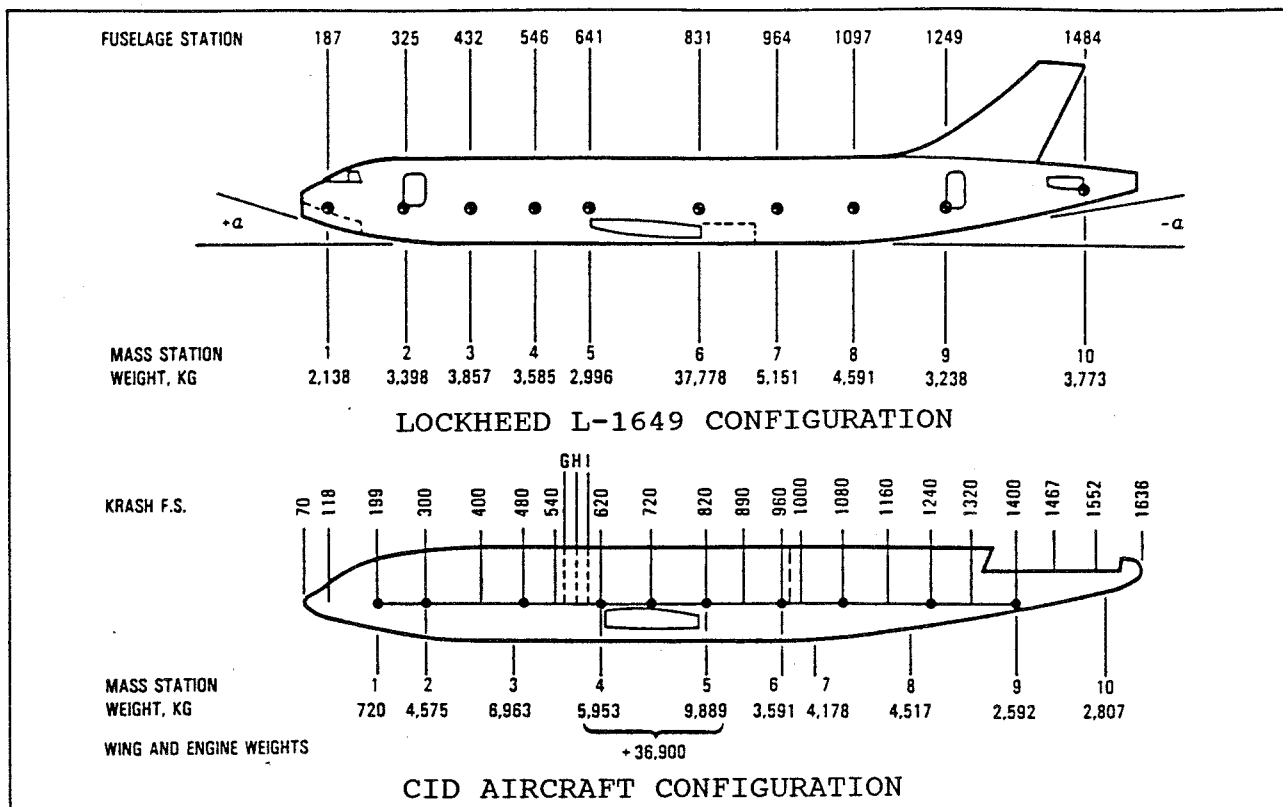


FIGURE C1 : Comparison of Airplane Configurations

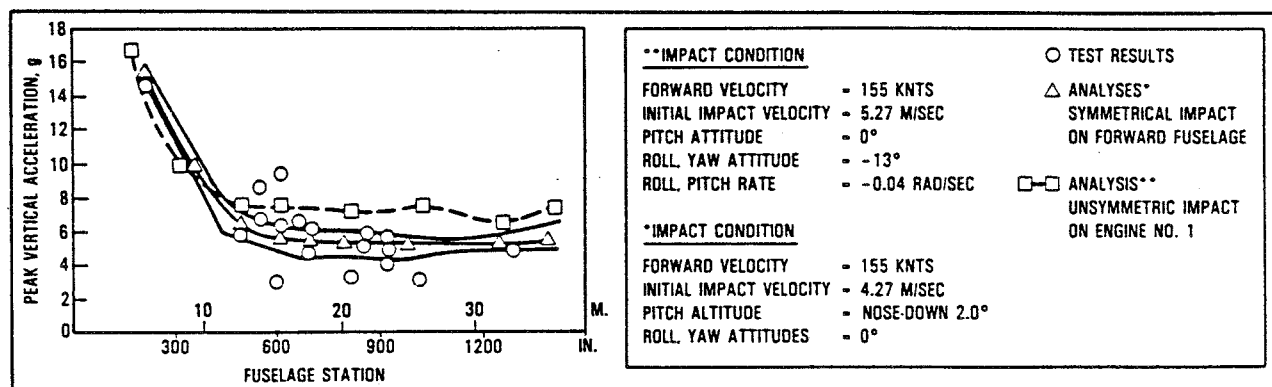


FIGURE C2a : Peak Fuselage Accelerations Distribution

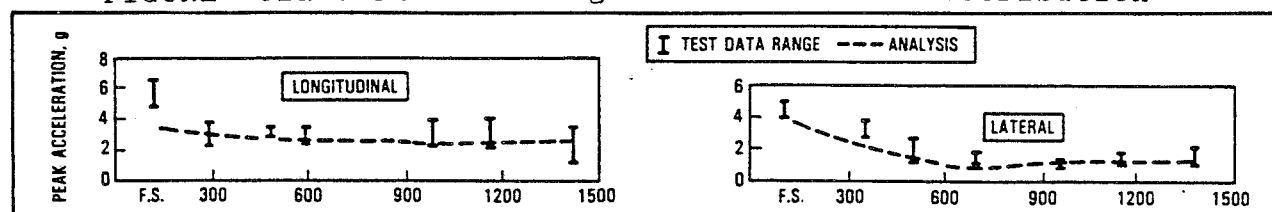


FIGURE C2b : Comparison of Test versus Analysis Results Longitudinal and Lateral Accelerations

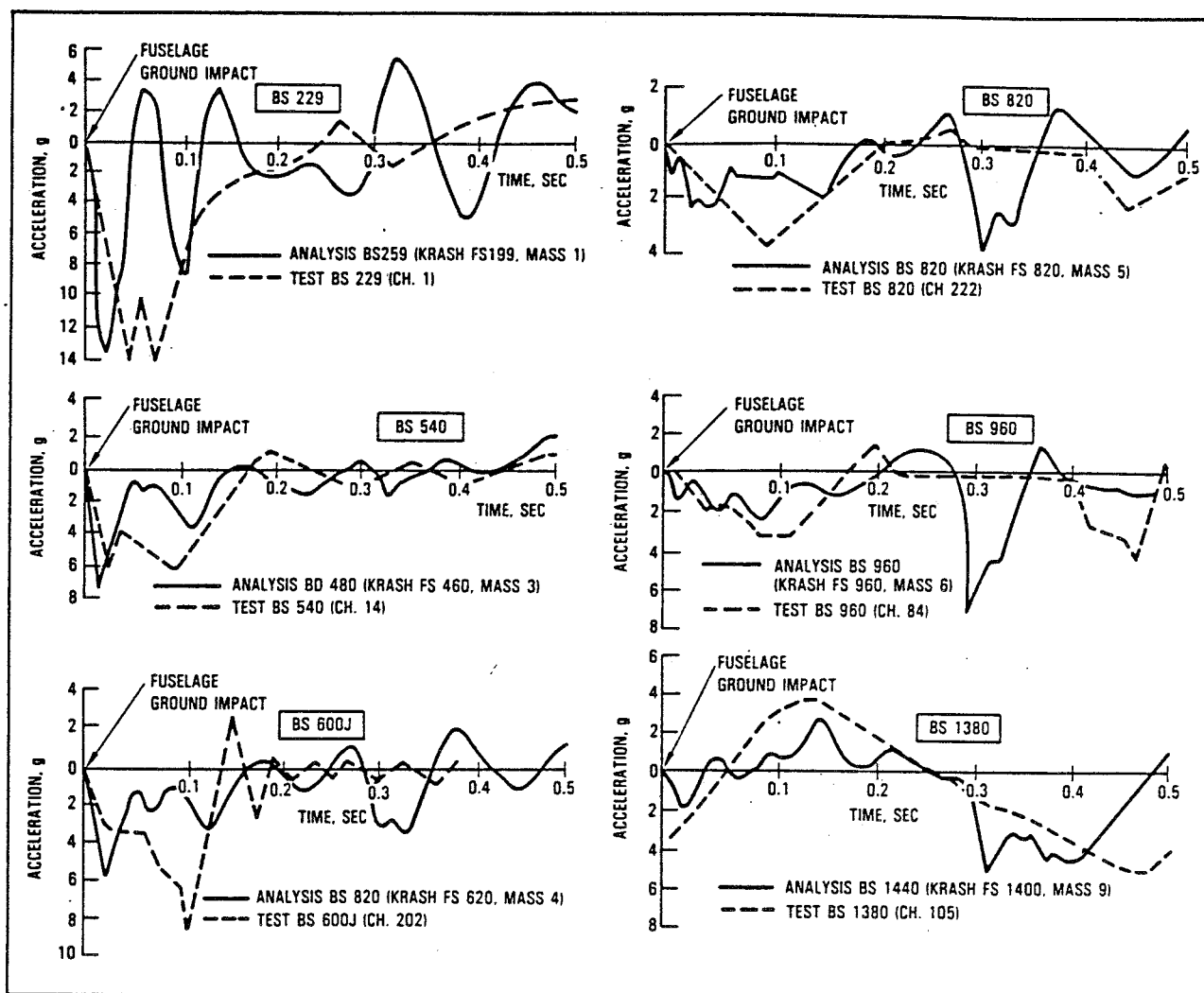


FIG C3a : Vertical Acceleration
Response
KRASH analysis and CID data

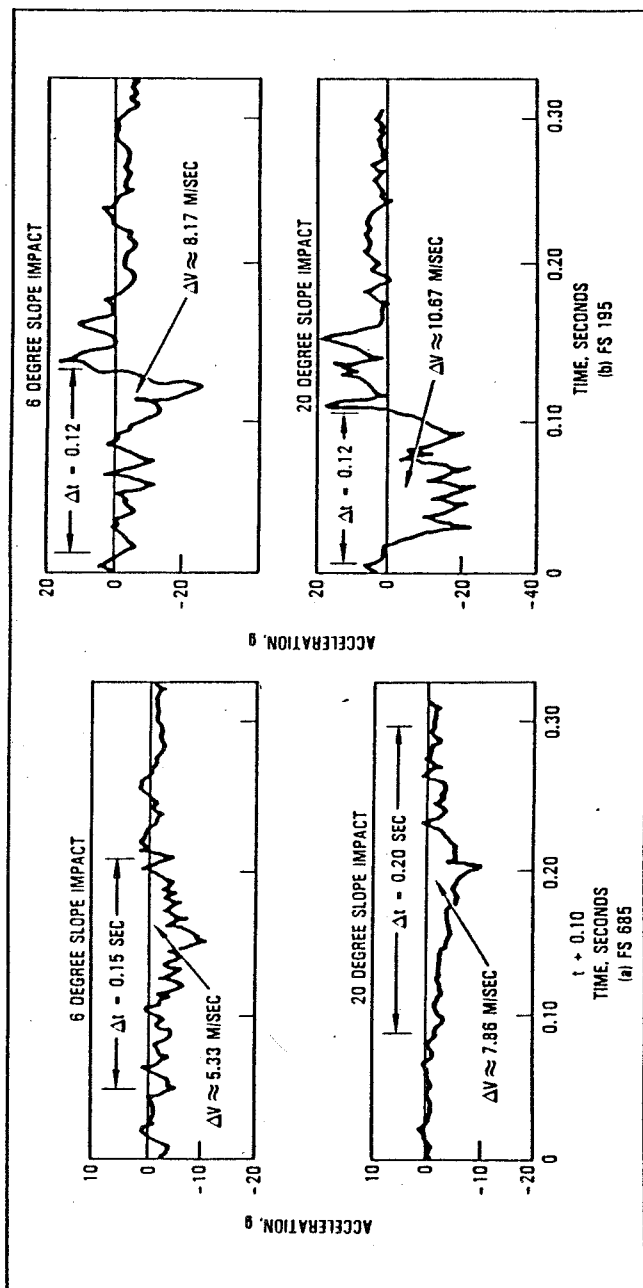


FIG C3b : Lockheed L-1649
Constellation Impact
Measured Longitudinal
Pulses

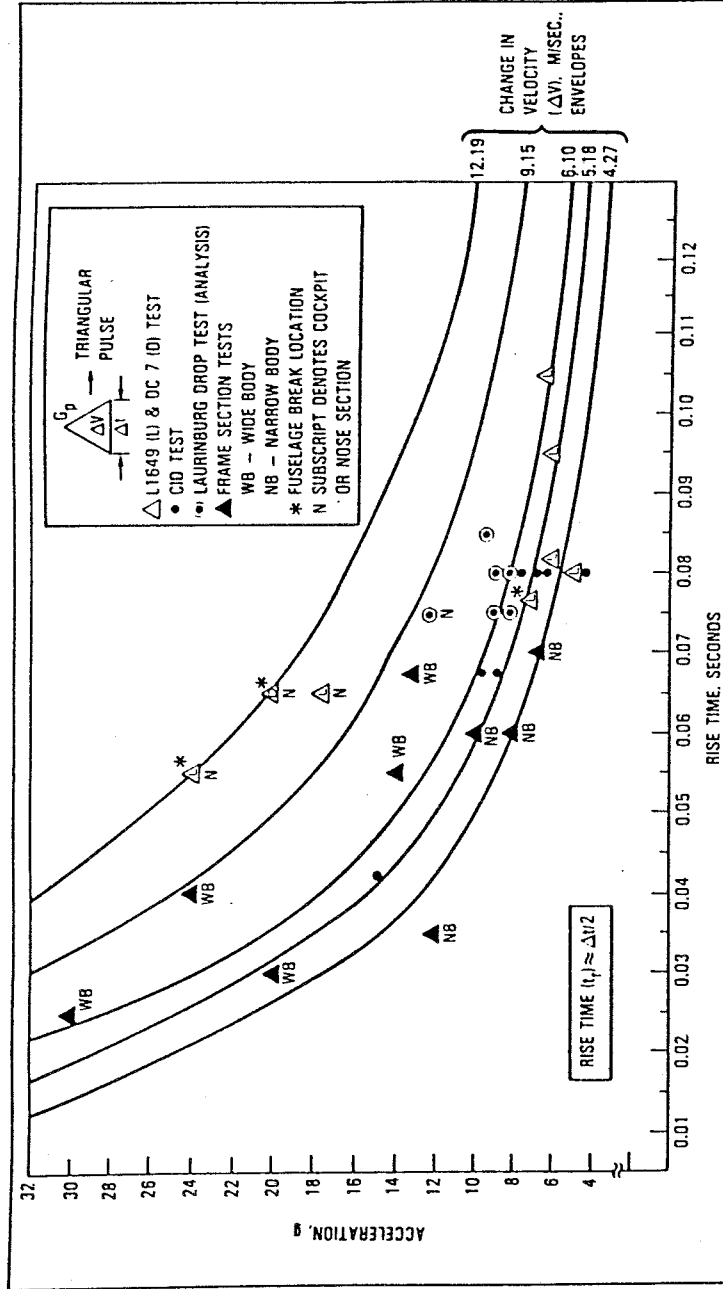


FIG C4 : Summary of Measured Vertical Pulses

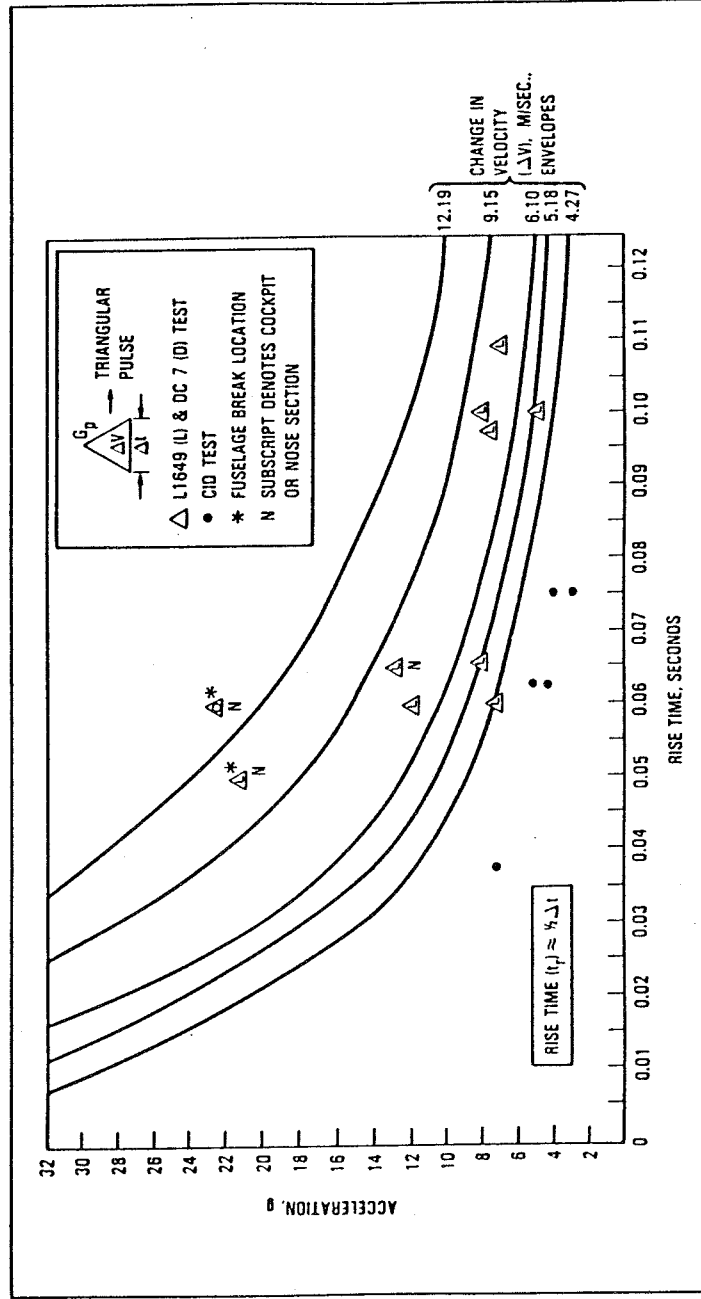


FIG C5 : Summary of Measured Longitudinal Pulses

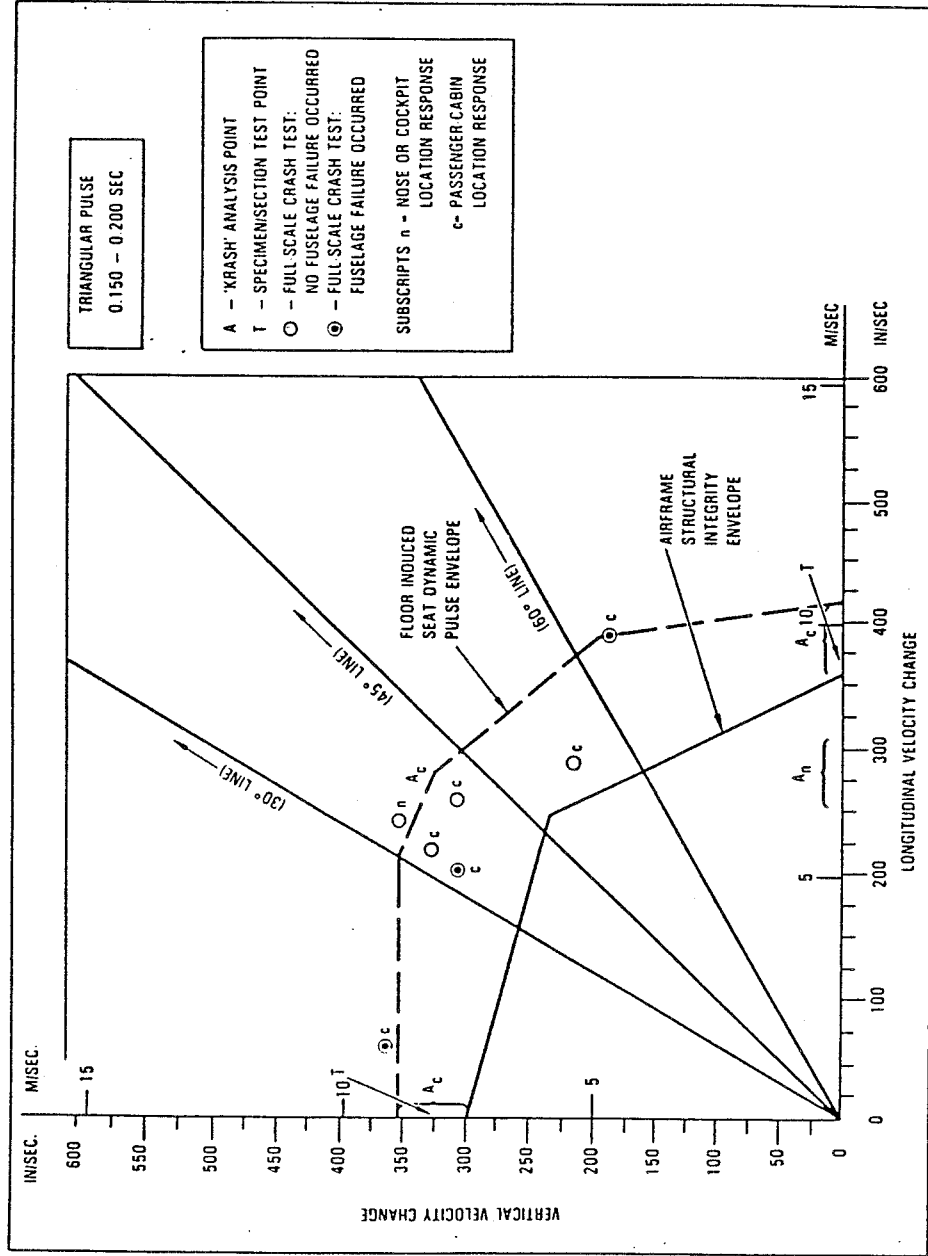
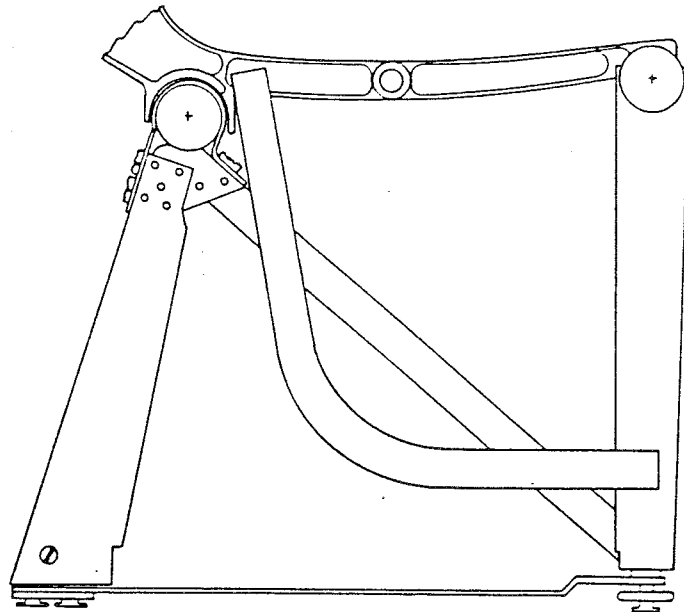
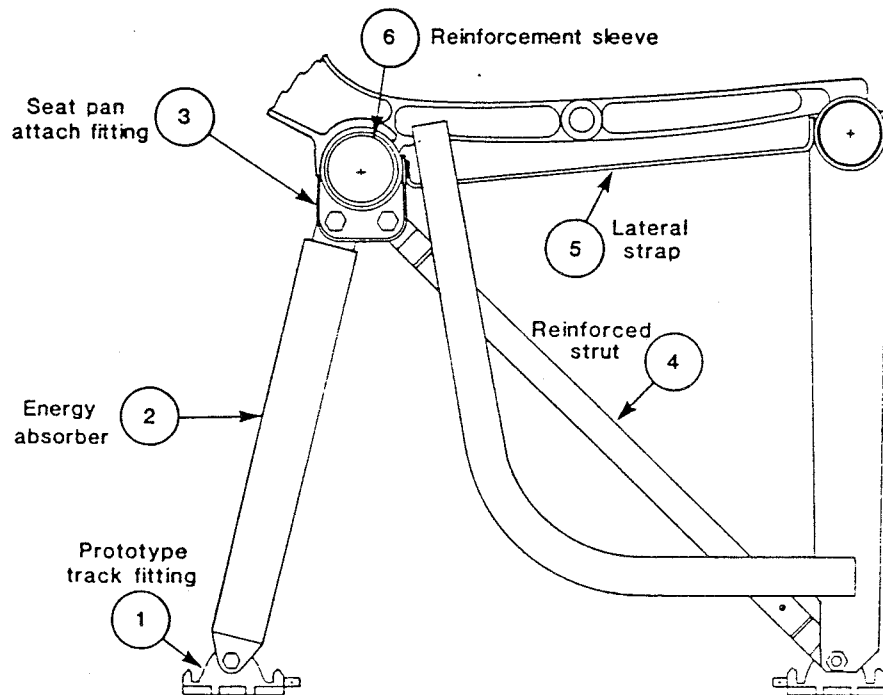


FIGURE C6 : Velocity Envelopes for Transport Category Aircraft



STANDARD SEAT



CONCEPTUAL SEAT

Figure C7. Illustration of standard and conceptual seats.

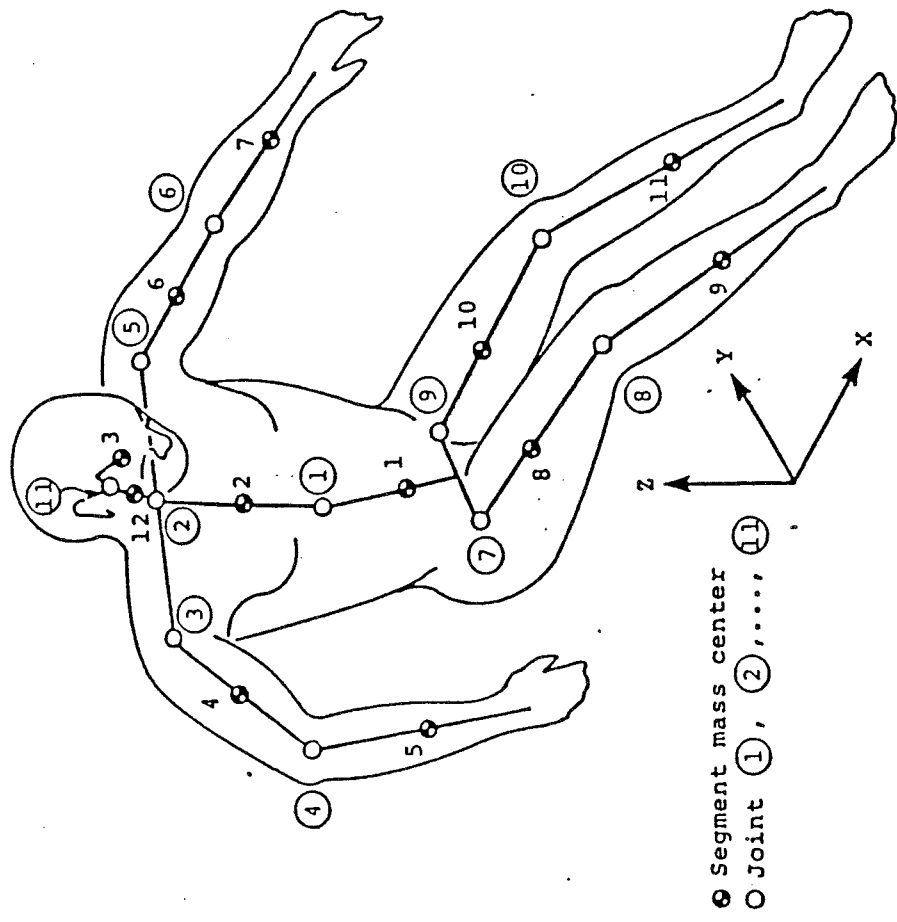


Figure D1 Twelve-segment (three-dimensional) occupant model.

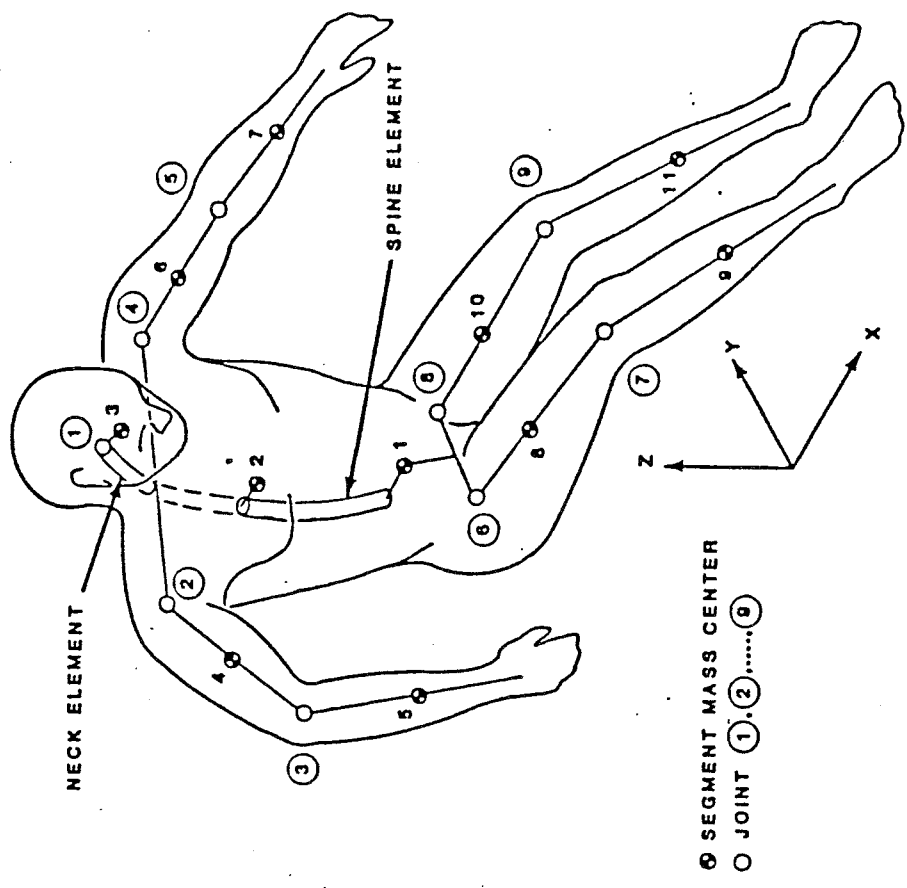
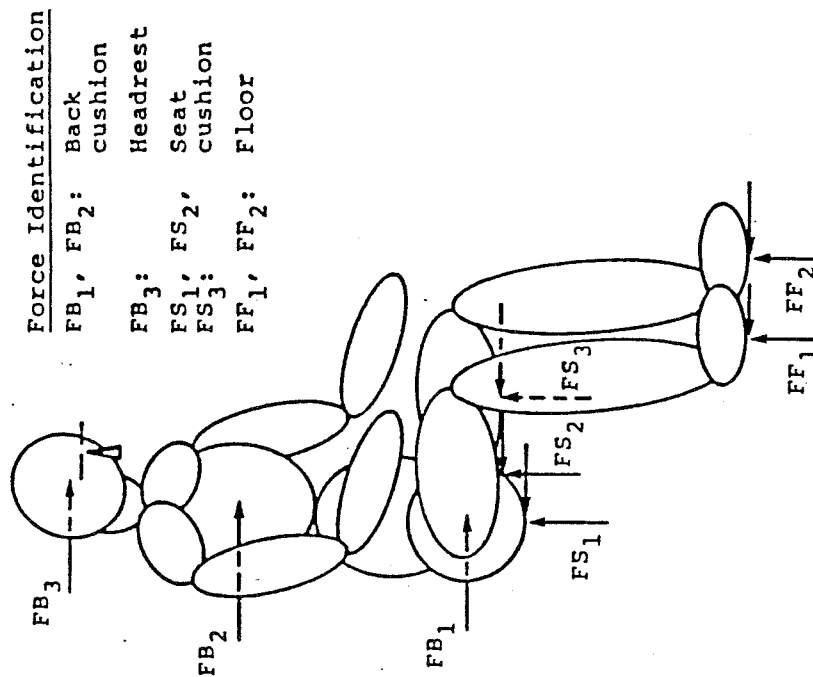
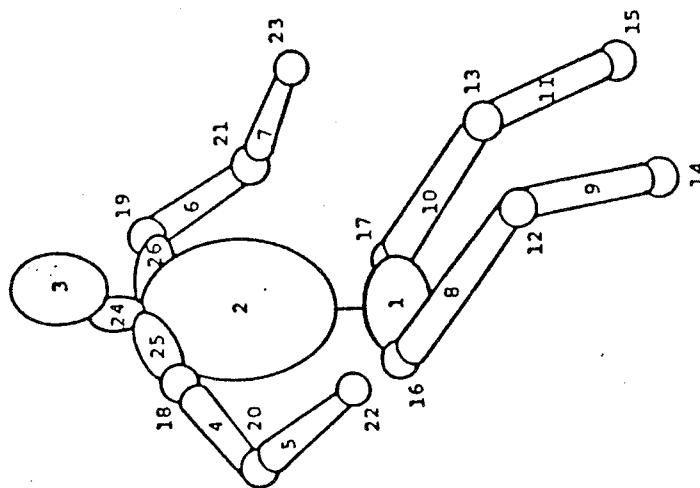


Figure D2 Eleven-segment (two-dimensional) occupant model.

Surface Identification

1. Lower Torso
2. Upper Torso
3. Head
4. Right Upper Arm
5. Right Forearm
6. Left Upper Arm
7. Left Forearm
8. Right Thigh
9. Right Lower Leg
10. Left Thigh
11. Left Lower Leg
12. Right Knee
13. Left Knee
14. Right Foot
15. Left Foot
16. Right Hip
17. Left Hip
18. Right Shoulder Joint
19. Left Shoulder Joint
20. Right Elbow
21. Left Elbow
22. Right Hand
23. Left Hand
24. Neck
25. Right Shoulder
26. Left Shoulder



Force Identification

- FB₁, FB₂: Back cushion
 FB₃: Headrest
 FS₁, FS₂: Seat cushion
 FS₃:
 FF₁, FF₂: Floor

FIG D3 : SOM-LA Occupant Contact Surfaces

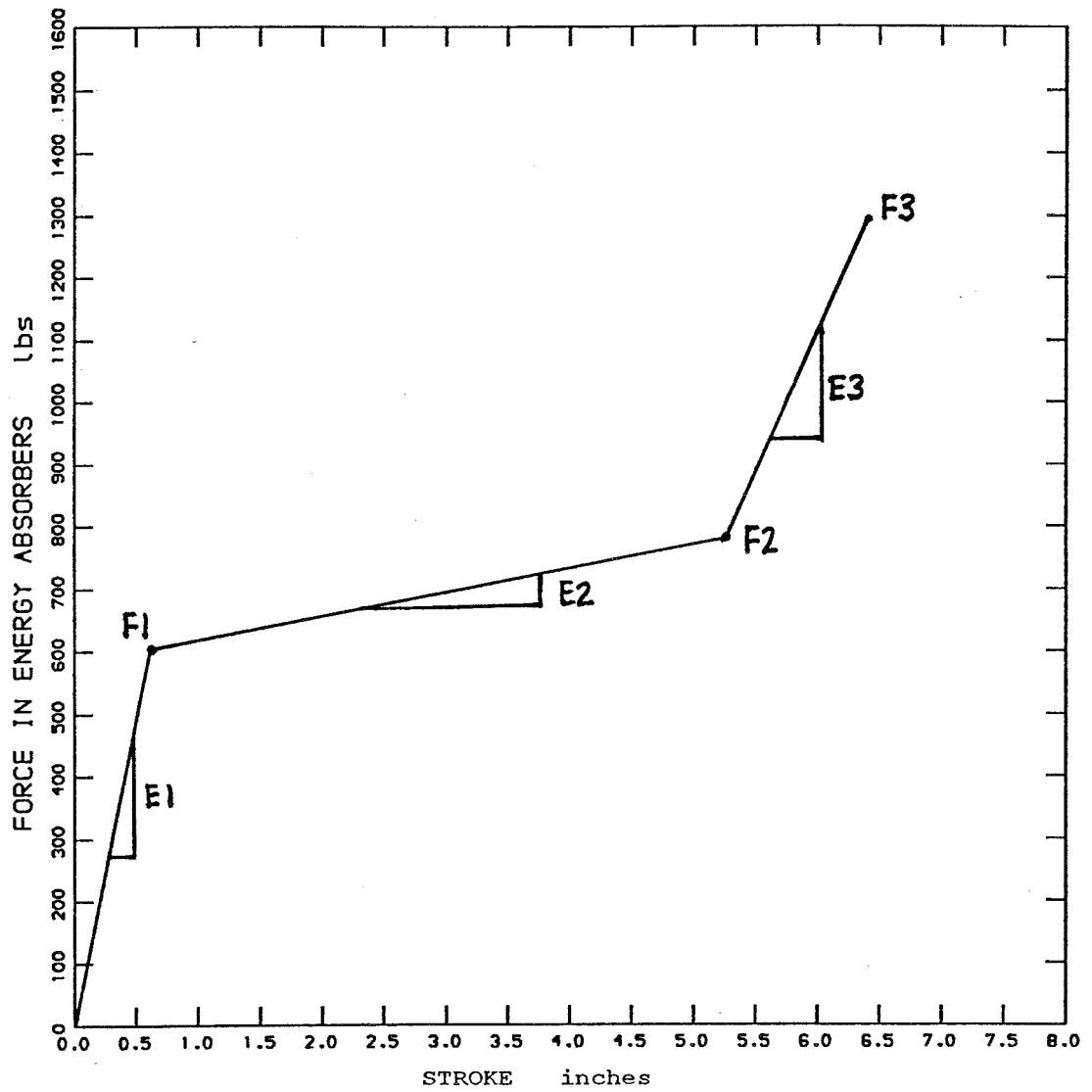


FIG E1 : Spring Element Stiffness
(Notional)

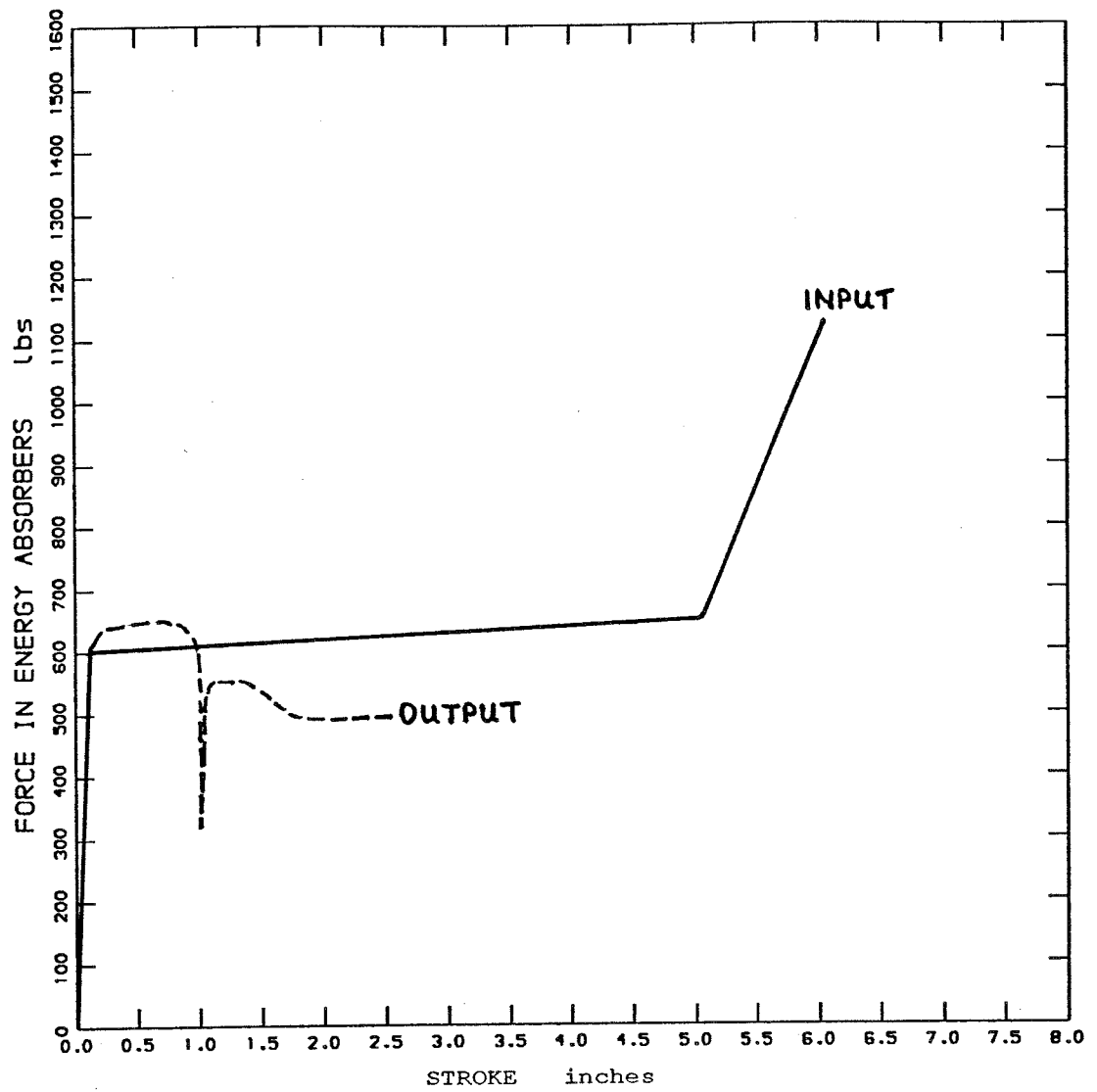


FIG E2 : Spring Element Stiffness
Type SS01

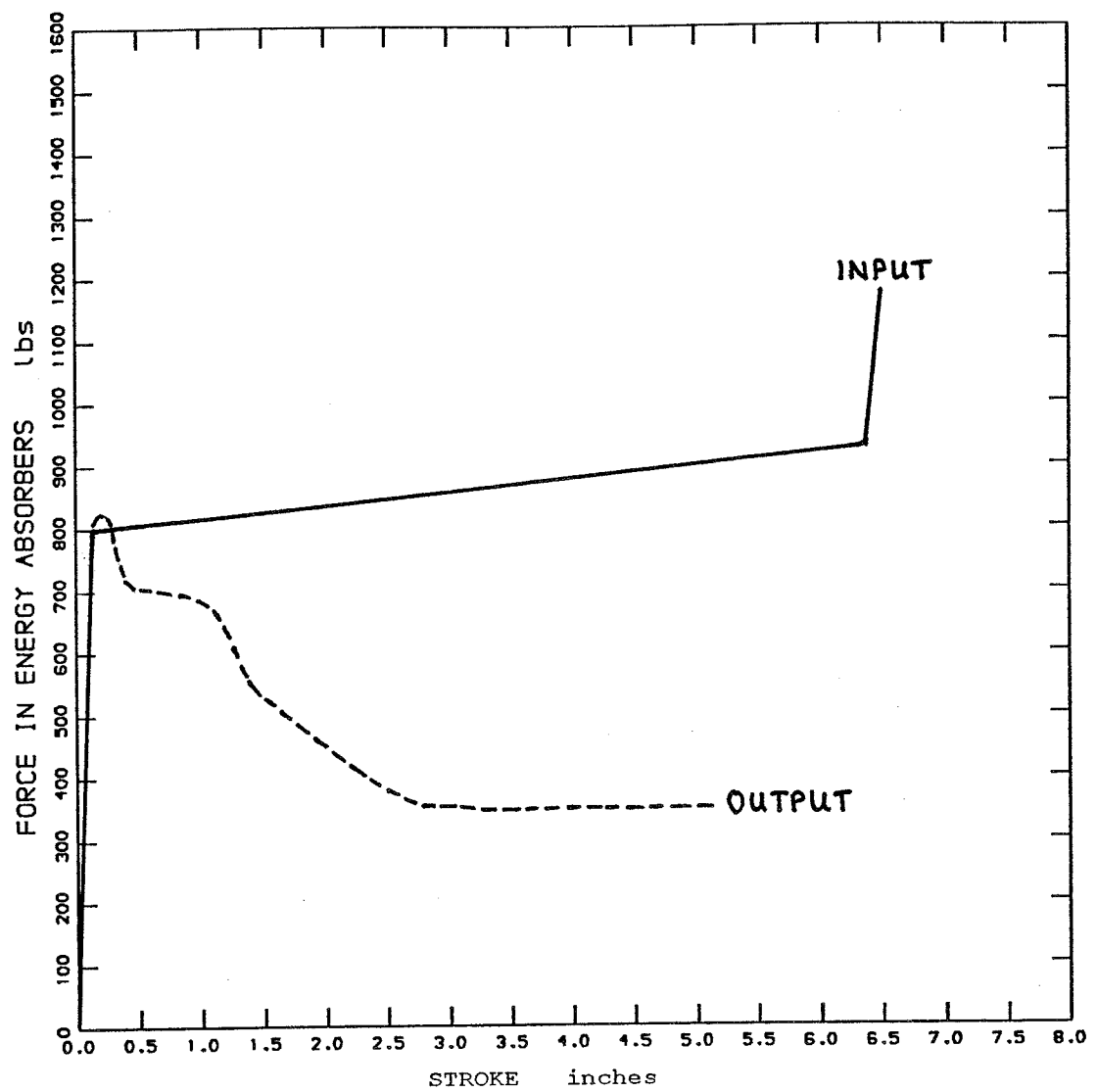


FIG E3 : Spring Element Stiffness
Type SS02

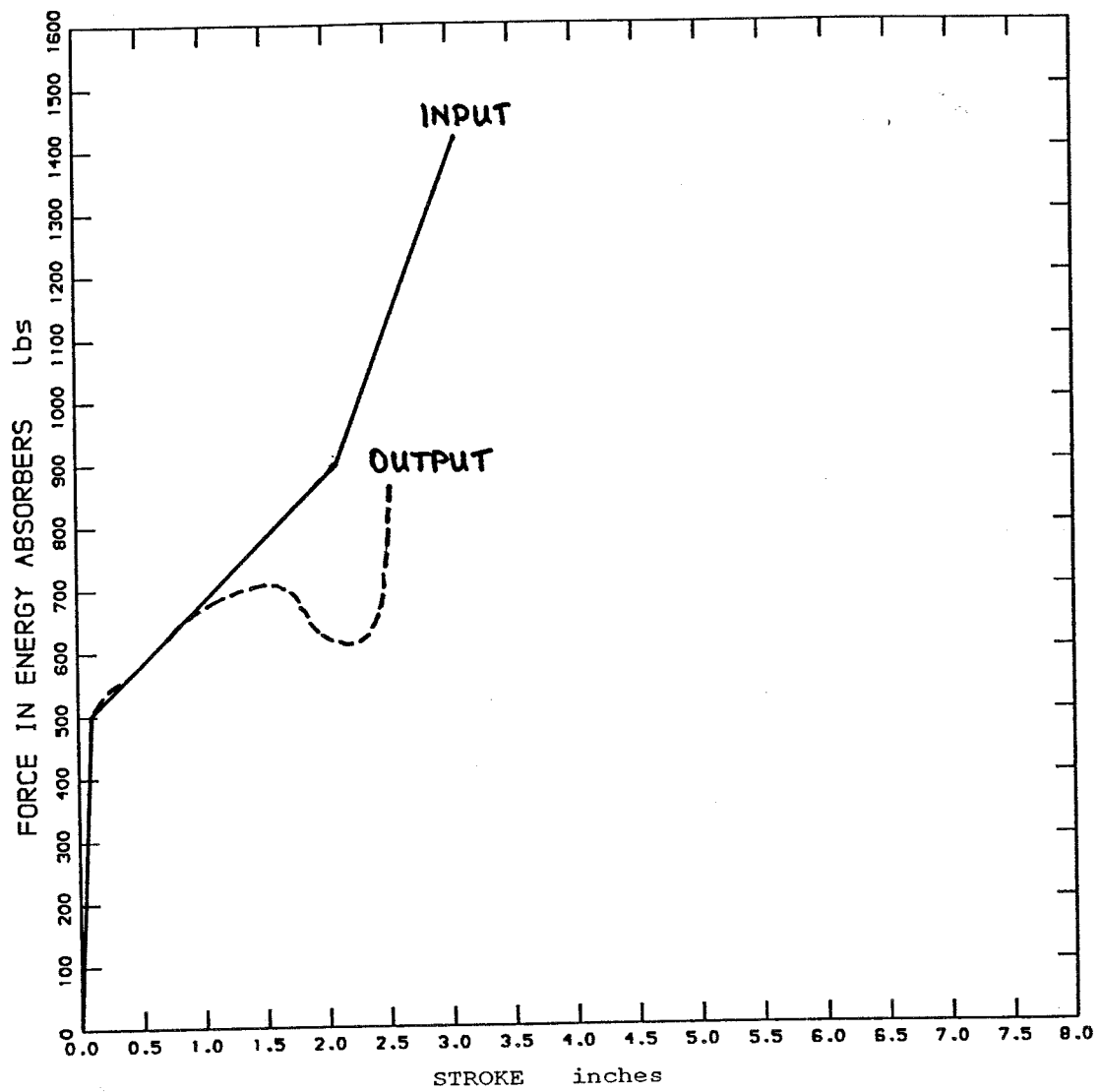


FIG E4 : Spring Element Stiffness
Type SS03

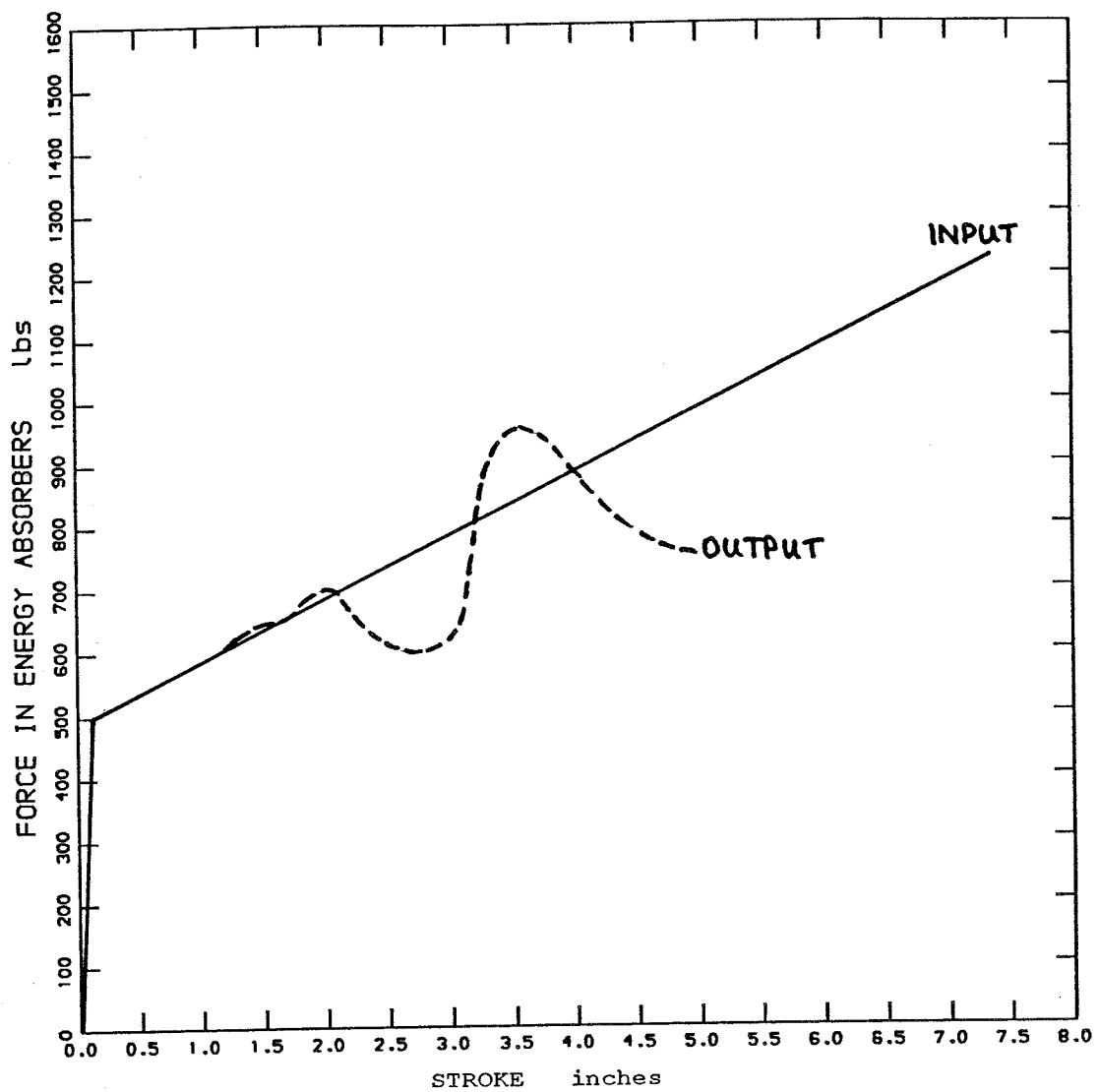


FIG E5 : Spring Element Stiffness
Type SS04

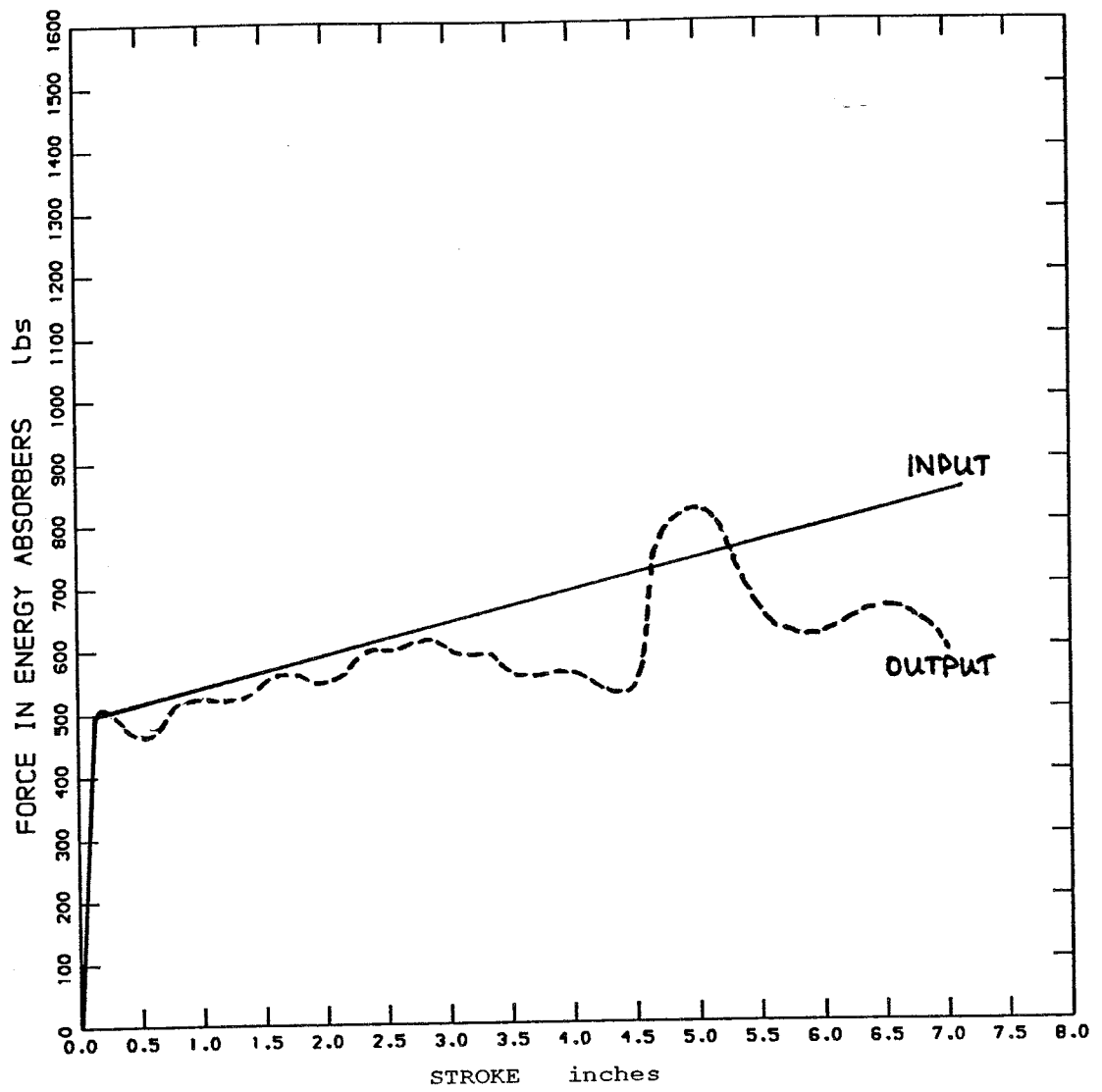


FIG E6 : Spring Element Stiffness
Type SS05

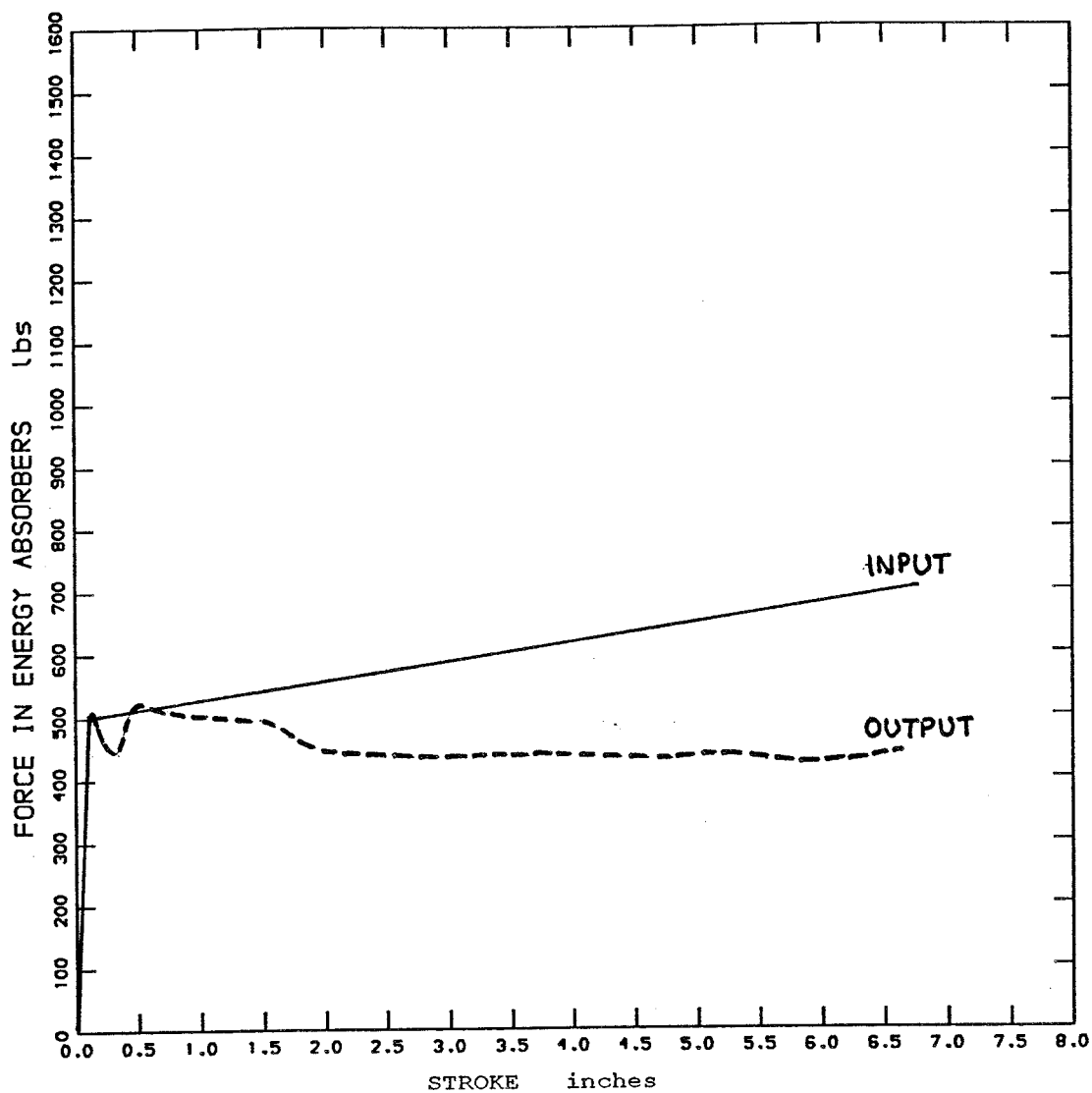


FIG E7 : Spring Element Stiffness
Type SS06