by the police, ambulance and fire brigade and often confused with the number of bus passengers, injured people and interested onlookers.

Contacts were established so that in the event of an accident involving bus passenger injury or seat deformation, we were to be notified. Official emergency organizations such as the Police Accident INvestigation Squad, Ambulance and the State Emergency Service were contacted, as were bus proprietors and bus body builders, most of whom perform repair work on damaged buses.

The request for notification of a bus accident was primarily confined to Victoria, but requests were made to state authorities in New South Wales and South Australia. The decision to limit the inspection of crashed buses was based on both geographical and financial considerations.

During the course of the investigation we were only able to inspect two bus accidents at first hand. The first, a Toyota Coaster Mini-bus, was involved in an accident on Sunday 3rd August 1980 on the Eildon-Warburton Road, north of Big River Camp at approximately 6.35 pm. The second, a School Bus, was involved in an accident on the morning of 26th April 1981 on Merrymans Creek Road, Cormondale.

Although the information gained from inspecting the buses was interesting, it could not be regarded as statistically useful. Recourse was therefore made to the further study of bus accident case histories.

5.2 A REVIEW OF BUS ACCIDENT REPORTS

5.2.1 Introduction

In order to understand the types of bus accidents and the degree of damage that buses sustain to both their structural body work and internal fittings, it was decided to study indepth the accident reports together with post-crash vehicle inspection reports. The documentation held by the TRB often includes the police reports and listings of those injured and sometimes their position in the vehicle. Statements from witnesses, passengers and the driver and photographs of the scene and dometer reading from the bus involved are also sometimes included. Correspondence between the bus porprietor and the TRB notifying the TRB of the occurrence of any form of incident is held in the files. A study of bus accidents on the TRB files was conducted back 1973 when possible, and were checked with the RoSTA data. A newspaper clipping file maintained in the "Age" Library was another source of basic bus accident data.

Case studies presented by the Traffic Accident Research Unit (TRAU) and the University of Adelaide Road Accident Research Unit (RARU) were also examined along with police traffic accident report forms.

5.2.2 Comments on the Bus Accident Case Studies

As a result of studying bus accident case histories, several points arose concerning the accidents reviewed.

- where accidents involving the bus running off the road and rolling over is concerned, there was both considerable bus body damage, particularly to the section above the lower level of the window accompanied by a high proportion of passenger casualties.
- There was a small proportion of accidents caused by a mechanical failure.
- Even in apparently severe accidents, there was a surprisingly small number of bus passengers seriously injured.
- 4) There have been several bus accidents which involved the penetration of the bus body to the extent that the passenger survival space was infringed. In such accidents the injury rate and severity were high.

- Bus seat anchorage failure has occurred in accidents in the past, causing the passengers and the seat to be unrestrained.
- 6) On the basis of the one accident examined it would appear from the damage observed both in terms of deformation and fracture, that seats in mini-buses may be less crashworthy than conventional omnibuses, although further investigation would be needed to confirm this.
- Accidents which involve the impact of the bus with a truck, generally result in high severity and risk injuries.
- 8) It would appear that the most common "on-path" accident was the frontal collision while negotiating a bend or corner. Such as accident usually involved casualties on the part of the bus passengers.
- 9) A large number of accidents occurred when the bus was empty or very nearly empty. Typically, the driver was on his was back to the depot.
- 10) The indicence of fires onboard buses appeared to be significant, although the resultant injury rate was very low.
- 11) If the bus impacted a car, the risk of serious injury was very much higher in the car than it was in the bus. Although there would appear to be some justification in concluding that the bus passengers are more likely to sustain minor injuries than the car occupants.
- 12) The types of injuries sustained in bus accidents were largely lacerations, bruises to the head, face and extremities in the category of minor injuries, however, with the more severe injuries, head injuries and skull fractures were predominant.

13) Only accidents involving a collision were recorded so any indication of transit bus passenger falls involved in non-collision accidents was not available.

5.3 CONCLUSION

As a result of inspecting two buses that were involved in an accident and by studying accident reports and post-crash inspection reports, it has been possible to grasp an understanding of the conditions which lead to injury causation in the event of a bus collision. In the less severe category of accidents where the bus will typically be involved in an impact with a car and remain on its wheels on the road the injuries sustained by the bus passengers are generally minor. Such accidents often result in bruises caused by contact either with the seat and other internal fittings of the bus or with the floor of the vehicle once the passenger has been dislodged from his seat. The top of the seat back (particularly low back seats) has been shown to be an object within the vehicle which is often contacted in the event of an accident. Evidence of blood is often the characteristic sign of contact with the top of the seat back. Futhermore, the easily damaged regions of the face such as the nose and teeth, are likely to sustain impacts with the often inadequately protected seat back. The mandatory introduction of padded "roll-top" seat backs for low backed seat backs in Victoria by the TRB, must reduce the likelihood of passenger injury due to the distribution of the impact laod, assuming of course, that proper deisgn methods and energy abosrbing materials of sufficient thickness have been used. Rigid bars, which are still common throughout Australia on the tops of seat backs, provide an object which in more severe accidents, has in the past, more likely been the cause of skull and facial boon gracture. The more serious injuries resulting from bus accidents have largely been due to head injuries and of people killed on board a bus, there has been a high incidence of head injuries. Often in these more serious accidents, the passenger survival space is destroyed due to collapse of the roof structure as a result of roll-over or an

impacting vehicle penetration. Lacerations are another common form of unjury sustained by bus passengers in the even of an accident. Internal items such as non-recessed window latches and saloon lights are probably the cause of such injuries. The practise of locating ashtrays on the rear of seat backs has led to passenger injury due to both their sharp or tight radius corners or the tendency for the plastic items to fracture, leaving sharp ragged edges. In the event of more serious collisions where roof collapse occurs, the separation of internal panels needs to be guarded against, as such a situation may be particularly dangerous and highly likely to cause severe laceration. Standee passengers are particularly vulnerable to impacting the seats, floor, the driver's protection barrier and the fare-box. In the event of a bus roll-over, the occurence of either partial or total passenger ejection is likely and such an event is potentially dangerous.

It needs to be noted, in conclusion, that the provision for studying bus accidents injuries in detail is extremely difficult with the existing accident report procedure and it is only when in-depth case studies are performed that sufficient detail is provided to allow a meaningful investigation into injury causation to be undertaken.

CHAPTER 6

FORMULATION OF TESTS FOR ASSESSING THE STRENGTH OF SEATS AND SEAT ANCHORAGES

6.1 INTRODUCTION

Throughout the literature reviewed in this project, there has been a distinct lack of documentation of:-

- The strength, stiffness and crashworthiness of existing Australian made bus seats.
- 2) The strength, stiffness and crashworthiness required for bus seats operating under Australian conditions.

It was therefore concluded that the tests performed during this project should aim to establish certain characteristics of bus seats that are being used in buses on Australian roads. There are a large number of different bus seats available and it was thought impractical to consider testing every different make. Instead a representative sample of seats was sought. The parameters considered important to investigate were:

- 1) The energy absorbing characteristics of the seat.
- 2) The maximum force sustained by a seat during the collapse mode.
- 3) The elastic and plastic stiffnesses.

At the same time, it was clearly necessary to establish the crashworthiness of different types of anchorage systems. During an impact, it is essential that the anchorage stays intact, and prevents the seat from moving.

It was therefore decided that the test be a relatively simple static deflection test, using an hydraulic ram to apply a forwards directed force to the top rail of the back of the seat in order to load the structure in a similar manner to that achieved in a head on collision. That is to simulate the force of one or two occupants seated behind the seat who, on the occurence of a front end impact, would collide with the back structure of the seat in front. It was considered important to retain as much as was practical of the bus wall, floor and subframe in the test jíg so as to be able to test the seat and seat anchorage system in a realistic manner.

There was a need to keep the equipment needed for monitoring and recording the force and deflection of the seat as simple as possible. This was because the tests were being carried out at an existing test jig owned by Ansair. This testing rig, together with several skilled trandesmen was made available to the authors at no cost. Making use of the Ansair facilities meant:

- 1) Saving in both time and money,
- a need for the monitoring and recording equipment to be able to be quickly and easily installed and removed in the event of the jig being needed for other tests,
- 3) a need for the test equipment to be maintenance free and relatively robust as the tests were carried out in a factory environment.

6.2 TEST METHODOLOGY

Several decisions were made regarding the method of testing so as to extract the most useful data possible.

6.2.1 The Method of Load Application

A single hydraulic ram was used in conjunction with a loading bar which was attached via a load cell to the ram. The loading bar fitted over the top of each seat back was pivotted so as to be free to allow for any skewing which may occur in the seat during the loading sequence. All the seats investigated were tested in the untrimmed state to allow for:-

- (1) the ability to observe the type and location of failure,
- (2) a positive means of applying the load to the top of the seat back in such a way as to minimize the possibility of slippage,
- (3) a reduction in the cost of purchasing.

Having the point of application of load at the top rail of each seat means that the bending moment applied about the floor anchorages would be a function of seat geometry in each test,

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the important parameter being of course the distance of the top rail from the floor anchorages or the point of failure of the seat back.

This however, is not critical, as it is a relatively easy task to compute the effective bending moment, once the geometry of the seat is known. It is indeed more meaningful to be able to calculate the bending moment about the point of failure (usually the base of the seat back).

A logical alternative to loading the seats would be to use a pressure form on the back of the seat and a hydraulic ram fixed at a given height above the floor for all seats and adjusted so that the ram moved horizontally all the time. This would simulate chest loading in an accident situation. However, it was considered that this method would not give a true accident simulation since both knee and head regions also contribute to the loading of the seat. When a seat is loaded during the deceleration process involved in a head-on accident, the rates of loading for the three regions, head, chest and knee are different due to the movement of the passenger which is predictable, but complicated. Thus, the only sure method of determining data on seat characteristics in an accident situation is by dynamic testing with fully instrumented manikins and high speed cinematography. Using a pressure form on the back of the seat in a static test neither,

- fully tests the entire seat as any point of the seat above the point of load application would be unstressed and not contributing to the overall energy absorbing characteristics of the seat,
- nor 2) makes the process of determining the bending moment about the point of failure of the seat any easier, as the resultant bending moment would be difficult to ascertain and inaccurate,

nor 3) gives a fair representation of the accident situation.

6.2.2 FLOOR/WALL TEST BED RECONSTRUCTION

In the early stages of the planning of the test program, it was considered important that the testing method should be as realistic as possible. To this end, three different chassis test jigs were constructed, each one being effectively a module from a bus or coach and employing the same sized structural members and the same method of construction. Thus, the wall floor, floor bearings and chassis rails were identical to that which would be found in the buses manufactured by the three companies that constructed the test beds; Ansair, Domino and Denning. Due to industrial disputation all the tests conducted for this report were carried out on one test bed, furthermore four tests had to be dropped from the testing programme.

6.2.3 Seat Anchorages

The remaining link necessary to complete the test bed was the method of attaching the seats to the floor. Again, for reasons of wishing to test the entire system as realistically as possible, the seat manufacturers or coachbuilders were asked to supply the necessary hardware that would normally be used for seat retention. The tests covered not only a variety of seats, but also a sample of retaining methods both for the floor and wall mountings.

6.3 TEST DESCRIPTION

6.3.1 Test Preparation

The following is a list of tasks that were formed before each test

1) The seat was attached in the usual method used for that particular seat. Attachment to the floor was such that maximum ram travel was achieved. Proper anchorage of the seat often meant the removal of existing tapping plates used in previous tests and the modification of the test bed to accept the necessary tapping or backing plates

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for both the floor and wall mounting positions of the new seat. This task sometimes meant removal of the wooden floor to allow access to the test bed frame.

- 2) Positioning of the hydraulic ram across the width of the seat to ensure that the centreline of the ram coincided with the centreline of the seat and that the height of the ram was such that the ram remained substantially horizontal during the entire tests. An error analysis of the small amount of inclination and declination that occurred was carried out and yielded maximum errors of less than 1% on effective horizontal ram force in the course of any one test.
- Check the force and deflection measuring equipment both electrically and mechanically and note the necessary voltage levels.

6.3.2 Test Description

The tests needed two people; both in the preparation and testing stages.

While a test was in progress, one person controlled the hydraulic values regulating the pressure fed to the ram, and the other monitored the plotting of the force/deflection characteristics, observed the method of failure of the seat and checked the voltage readings of the output from the force and deflection equipment.

The test commenced with the placement of the loading bar on the top of the seat back and the drawing of a line on the force/ deflection plot indicating the position of the ram at the commencement of the test. Another line was drawn indicating zero load. The corresponding voltage readings were recorded on the lines drawn on the X-Y plotter.

One of the four hydraulic taps were closed and the compression/ tension lever was eased towards the compression setting. The rate of load increase was dictated by a combination of which the four taps were closed and the position of the compression/tension lever. The rate of load increase was kept as slow as possible, consistent with an approximately constant velocity deformation of the seat until yield occurred where upon if the rate of deformation was too great, the pressure applied to the ram was reduced. During the course of the test, photographs were taken showing the seat in its undeformed position, its maximum deformation stage and its final state after the load had been removed and the residual elastic deformation or spring back had been allowed to occur.

In addition, photographs were taken of specific regions of the seat in which had contributed to failure or deformation.

Once the test was completed, the ram was monitored in both its maximum and minimum displacement positions and calibration check marks were drawn on the force/deflection plot along with noted voltage readings. A check on zero load was carried out after each test. Notes were taken on the method of failure, the state of anchorages, sub-frame and seat back, the amount of skewness and any observations noted during the test. Photographs were taken of the state of the floor anchorages.

Depending on the type of failure and the form taken by the force/deflection characteristics of the seat, a decision was made to retest the seat. This involved removing the seat from the test bed and repositioning it closer to the ram, thus effectively increasing the maximum deflection possible. This can be seen in the ensuing force-deflection curves where the seats were unloaded and reloaded in their new position.

In most cases the seat would reload to a point very close to the previous unloading point, e.g. Test No. 1. The subsequent force-deflection curve would then be a continuation of the previous curve. Thus the effect of unloading and reloading had little effect on the resultant characteristic curve for that seat and in particular the calculation for energy absorbed.

In a few cases reloading did not bring the curve back to its previous unloading point, e.g. Test. No. 10. This was probably due to additional structural deformations and changes taking place during the unloading phase. However after some further deformation had taken place the curve did start to follow the probable extropolated characteristic. Under such circumstances the calculated total energy absorbed probably underestimates the correct figure by (not greater than) 10%.

If a seat was subject to an additional test, the method of testing was identical to the initial test. For each seat tested, new securing bolts and additional hardware were used in case of the possibility of test damage. If a test was subject to a second test, the securing bolts were reused from the first test.

6.4 APPARATUS

6.4.1 Test Jig

The test jig consisted of an open base frame, constructed from RSJ and C section steel lengths. The main requirements of the structure were;

- 1) that it be rigid,
- that it possess the facility to attach loading devices. In this case it was a single hydraulic ram in any position to apply a load in any direction,
- that it have the facility to rigidly attach to it the test article.

The last objective was made possible by using test beds, which will be described later. The test jig also included the hydraulic loading system, consisting of several hydraulic rams (during these tests only one was employed) the pump and control panel.

6.4.2 Test Bed

Three test beds were made, each one representing the floor/ wall structure used by three different coachbuilders; Ansair, Denning and Domino The test bed used for the tests was secured rigidly onto the test jig.

6.4.3 Hydraulic Ram

The ram used to load the seat was a Vickers 2" \emptyset 18" stroke hydraulic ram, with the facility of feeding the ram with 13000 kPa pressure. This is equivalent to a 26,688 N load as output from the ram.

6.4.4 Load Monitoring Equipment

This consists of three components;

- 1) a 22,240 N Interface tension/compression load cell,
- 2) a Gedge systems power supply amplifier,

3) and one channel of the X-Y plotter.

The load cell was screwed into the seat end of the ram and excited by a regulated 10V DC power supply. The load emitted a 4 mV/v signal which was fed back to the power supply/amplifier and amplified to give a 10V reading on full load. The amplified signal was then fed to one channel of the X-Y plotter, the scale of which was changeable. No scale changes were found to be necessary and the load scale was set on 50 mV/cm. Thus 100 mm of plot was equivalent to 1115.9 N. In addition a switching device was arranged so that the following voltages could be displayed on a digital volt meter.

- 1) Excitation voltage (nominally 10V).
- 2) Unamplified load signal 0 40 mV.
- 3) Amplified load signal 0 10V.
- 4) Displacement signal.

6.4.5 Displacement Monitoring Equipment

It was considered that for the accuracy needed in these tests, it was not financially justified in buying a linear displacement transducer. Instead, a 10 turn helipot was mounted to the test jig, and on the end of its shaft a small sprocket was fixed, a length of light weight chain then ran over the sprocket. One end of the chain was attached as close as practical to the centreline of the ram (i.e. the centreline of the seat) while the other end was weighted. In this manner, it was arranged so that the chain between the helipot fixed on the jig and effectively the seat, ran parallel to the centreline of the ram. The helipot was powered with 10V DC from the power supply. The output was then fed to the facia panel of the power supply allowing switching to the digital volt meter, but always retaining continuity with the remaining channel of the X-Y plotter. The scale set on the X-Y plotter was 500 mV/cm, thus giving 100 mm of plot corresponding to 288.5 mm of ram displacement.

6.4.6 X-Y Plotter

The X-Y Plotter, a Rikadenki, plotted on to A3 size paper. There was no need to worry about the possibility of going beyond the allowable deflection range set on the Y channel as the maximum travel of the hydraulic ram was known. Consequently, the scale on the plotter was set so that the maximum movement of the ram could be recorded on the A3 size paper. There was however, a possibility of overshooting the limits of movement of the plotting arm in the X direction load channel primarily because, prior to testing, there seemed to be a great variety maximum forces obtained in previous bus seat tests. Peak loads from 2900 N^{33} to 20000 N^{13} were quoted as being achieved during tests of production seats. In one case, a maximum force of 25,500 N^{33} was achieved from an experimental seat. Thus we were undecided as to what scale setting to use. If the setting was too insensitive, then the plot could not be sensibly read. Alternatively, if the scale setting was too sensitive, then the pen would run the limits of movement of the pen We decided to choose a sensitive scale with the intention of changing scales during the test should the need arise.

The load scale of the plotter was set on 50mV/cm, allowing a maximum force of about 4448N (100 lbf) before a change in scale was required. Fortunately, none of the tests carried on in this project required a change of scale.

6.5 LIST AND DESCRIPTION OF THE SEATS AND ANCHORAGES TESTED

6.5.1 Introduction

A total of 12 seats were tested. These included reclining and non-reclining coach seats, charter bus seats and route bus or school bus seats. Selections were made from the following manufacturers:

Saydair: reclining and non reclining coach, charter, mild and stainless steel route bus seats. Pressed Metal Corporation (Sydney): route bus seat. Ansair: reclining and non-reclining coach (with and without semi cantilevered leg). Denning: reclining coach McConnell: Reclining and non reclining coach, charter and route bus seat. Domino: reclining coach

As a result of discussions with bus seat manufacturers and coach builders it was decided that this range of seats was not only a fair representation of those presently being used in the coachbuilding industry in Australia at the present moment, but also represented the majority of seats being fitted. There are many more manufacturers of seats and many more seat designs, yet in general they are all similar to those that have been incorporated into this testing program.

A variety of seat anchorage systems were also tested. Some of these systems are currently not permitted in Victoria under the TRB bus seat anchorage guidelines. The methods of fastening can be catergorized as follows:

6.5.2 Floor Mountings

- A tapping plate running the length of the bus is welded to the floor bearers. Once the plate, which is usally ½" x 2" bar or angle is in place, there are various ways of fastening a seat to it, viz:-
 - (i) The plate is drilled and tapped and a $\frac{5}{16}$ " bolt or equivalent is used with a spring washer.
 - (ii) The plate is again drilled and tapped and a $\frac{5}{16}$ " UNC bolt is used with a spring washer, but a lock nut is used as well. This method is recommended when thinner backing plates are used.
 - (iii) The plate is drilled and a $\frac{5}{16}$ " self tapping screw is used.
- 2) A backing plate usually $\frac{1}{4}$ " $_{\rm X}$ 2" bar or angle running the length of the bus is welded to the floor bearers. This plate is drilled and $\frac{5}{16}$ " UNC bolts are used in conjunction with spring washers and nuts.
- 3) Individual backing plates that are not welded to the buses body or chassis are used for each seat. These plates are drilled and $\frac{5}{16}$ " UNC bolts, spring washers and nuts are used. This method is not widely used.
- Note: Both metric and imperial sizes are used in describing the tests due to the different systems used by each manufacturer.

- 4) Backing plates of size $2\frac{1}{2}$ " x $1\frac{1}{2}$ " and $\frac{1}{6}$ " thick are positioned for each floor mounting bolt, again usually $\frac{5}{16}$ " UNC and spring washers and nuts are used.
- 5) No backing plate is used. Instead a ½" UNC bolt is used and under the wooden floor a "T"-nut is used. This device has a diameter of 19 mm and has three prongs, which when the bolt is done up, are drawn up into the underside of the wooden floor.

By far, the most common method of floor anchorage is that set out in section 1 above, which mostly uses a drilled and tapped hole without a locking nut. There are normally two floor fasteners per seat.

6.5.3 Wall Mountings

 A tapping plate, usually k" bar or angle is welded to the wall structural members and runs the length of the bus. This tapping plate is often integrated into an inner wall skin, which either runs from the floor to the height of the tapping plate or from the floor to the waist rail.

There are two commonly used methods of fastening the seat to the tapping plate.

- a) Drill and tap the tapping plate and use $\frac{5}{16}$ " UNC bolt with a spring washer.
- b) Drill the tapping plate and use self tapping screws.

In the normal loading of a seat fastened by this method, the bolts are in shear.

- 2) A seat rail running the length of the bus and acting as a ledge to which the seat can be fastened. This is often an integral part of an inner wall skin system, again either stretching between floor and seat rail height or between floor and waist rail. In either case, the inner skin and the chair rail are often made from 18 gauge sheet steel. The seat is then clamped to this chair rail by normally two $\frac{5}{16}$ " UNC bolts with spring washers and nuts.
- 3) An extruded aluminium section running the length of the bus, which allows "T" nuts to slide along and be positioned wherever a seat is placed. This extrusion is secured to the wall so that the centreline of the bolts fastening the seats, are either horizontal or vertical. If the bolts centreline is horizontal, the extrusion is secured to the wall by bolts after drilling and tapping into a backing plate or self tapping screws. If the centreline of the bolt fastening the seat is vertical, a seat rail is used and is essentially identical to a seat rail used to fasten seat directly except it now has the extrusion fastened to it and the seats are fastened to the extrusion.

6.6 TEST RESULTS

On completion of each test, notes on the method of failure and particular areas of deformation were taken and filed with force/deflection plots corresponding.

At a later date, the data in the form of the X-Y plot was analysed and force/deflection coordinates were read and fed onto computer storage tape. This allowed rapid calculation of the initial elastic stiffness of the seat and the energy absorbed by the seat at given deformations. In some cases, the raw data required either interpolation and extrapolation in order to achieve a true picture of the seats characteristics. The interpolation was necessary to smooth perturbations in the plot which were sometimes found after the seats were unloaded and then reloaded. In some cases, a secondary loading was deemed unnecessary as either the seat had failed by fracture and could no longer withstand load or had failed by bending and was work softening in such a predictable manner that reliable extrapolation was clearly permissible.

In some of the tests, the linear elastic range was difficult to establish as can be seen and consequently, from the force/deflection curves the point of yield is open to some conjecture. For each test, an elastic stiffness was calculated and the force and deflection at which yield occured was noted. The definition of yield for this exercise is the point at which the plot becomes non-linear.

6.6.1 Result and comments on mode of failures for each seat.

1) Test No. 1

Peak load - 1705 N
Total energy absorbed = 1272 Nm.
Elastic stiffness - 14289 N/m.
Essentially elastic/plastic deformation, with essentially
constant collapse load after initial yield.
Yield occurred at approx. 90 mm def. under a load of 1300N.
Max. def. was 605 mm.

Failure of this seat resulted from the bending of two components governing the pivoting of the two seat squabs (Photo 6.2). The external vertical seat squab tube of each seat back is crimped at the base of the squab to allow welding of a bush which forms the pivoting axis of the

seat backs. The squab tubing extended downward below the bush and acted as a lever for the spring/piston device used to control seat back angle. One of the members that bent was this lever section of the squab tube. The bending took place at the point where the crimped and non-crimped tube met, just above the pivoting bush. The other component that underwent bending was the bracket used to attach the seat squabs on both external sides of the seat to the seat sub frame (Photo 6.1). The crimped section of squab tubing which acted as a lever for the reclining mechanism is also used as a positive stop for seat back movement. The seat back pivots forward until the crimped tube comes up against a plate welded to the seat sub frame. As a result of the bending of both the anchorage bracket for the seat squabs and the crimped tube at the base of the seat backs, the crimped tube slipped past the reclining stopping plate, (Photo 6.2). Thus the load was no longer taken by the stopping plate and was transferred through the crimped tube to the spring/piston reclining device situated underneath the seat cushions near the external edges of both cushions.



The ultimate failure of the seats took place when shafts from both spring/piston reclining devices fractured suddenly (Photo 6.3). The shaft, which runs through these pistons had a threaded end which is screwed into a shackle mounted at the end of the crimped seat squab tube. The fractures occured at the end of each of these shafts where the threaded section commences (Photo 6.4). Once the fractures had occurred both seat backs were free to pivot forward unrestrained. Since both shafts fractured simultaneously, there was no skewing of the seat.

There was no apparent anchorage distortion.



Photograph 6.1 Seat prior to Test No. 1



Photograph 6.2 Seat back pivot. Note the bending of the plate on the left and the crimped tubing (part of the seat back) on the right which combined to render the positive stop (bottom centre) ineffective



Photograph 6.3. Ultimate failure. Note the bending inwards of the main side plate of the seat cushion in the region of the seat back pivoting bolt. The two adjustable reclining mechanisms can be seen hanging from the front of the seat cushion frame, failure of the shaft resulted in the springs falling to the floor.

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Photograph 6.4. The point of failure can be seen at the threaded end of the piston shaft. Note, also the bending of the side plate of the cushion framing in the region of the seat back pivot.

II)Test No. 2

Peak Load - 1905 N Total Energy Absorbed 1562 N Elastic Stiffness 21269 N/m Essentially elastic/plastic deformation, followed by work softening. Yield occurred at approx. 60 mm deflection under a load of 1300 N. Max. def. was 765 mm.

The mode of failure of this seat was by bending of the four vertical seat back tubes (Photo 6.7). The point at which bending occurred was at the upper end of the stiffening insert tube (Photo 6.8), which is added to strengthen the bend in the lower segment of these tubes. Anchorages remained sound, although the rear seat leg lifted and bent the floor attachment plate (Photo 6.9). There was no noticeable skewness.



Fig. 6.2



Photograph 6.5 Seat prior to Test No. 2. Note the non-continuous floor mounting plates at the ends of the legs (compare to photo 6.1).



Photograph 6.6 Test No. 2 in progress Note the already obviou lifting of the rear flo mounting plate.

Photograph 6.7.

Maximum deflection reach in Test No. 2. Note the localized bending, i.e. pivot at the base of the seat back.



Photograph 6.8. The localized bending can clearly be seen just above the upper edge of the inserted strengthening tube.



Photograph 6.9. The lifting of the rear seat leg attachment plate is obvious, although there was no sign of imminent failure. III) Test No. 3

Peak Load - 1851 N

Total Energy Absorbed 1506 Nm

Elastic Stiffness 18020 N/m

Essentially elastic/plastic deformation followed by work softening.

Yield occurred at approx. 90 mm deformation under a load of 1700 N.

Max. def. 757 mm.



Fig. 6.3.

Failure of this seat was due to localized bending of the four vertical seat back frame tube members at the base of the seat backs, immediately above the end of the stiffening tube inserts as in Test No. 2 (Photo 6.11). There was no noticeable damage to either the seat sub-frame or seat anchorages. Nor was there any apparent skewness.



Photograph 6.10. Prior to Test No. 3. Note that the seat legs are positioned between the two seating positions and thus the aisle side seat is cantilevered. Also note the thick (5 mm) continuous floor attachment plate welded between the seat legs.



Photograph 6.11.

Identical failure mode and position to that exhibited in Test No. 2 (photos 6.7 and 6.8), i.e. bending of the four seat back tubes just above the upper edge of the inserted strengthening tube. Note that there is no distortion of the floor anchorage plate in contrast to photo 6.9. This seat was the same as the seat tested in Test No. 2, except for the repositioning of the legs and was used to confirm Test No. 2 results and to investigate the possible effects of altering the seat sub frame.

IV) Test No. 4.

Peak Load - 1750 N

Total Energy Absorbed 1581 Nm

Elastic Stiffness 14591 N/m

Essentially elastic/plastic deformation yield occurred at approx. 80 mm deformation under a load of 1200 N. Max. def. 715 mm.



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Photograph 6.12. Prior to test No.4. Note that the seat legs are mounted to the sub framing between the two seating positions, the aisle side seat is cantilevered.

Failure of this seat was initially due to localized deformation at the base of the seat back where the frame tube had been crimped to receive the bush for the pivot pin of the reclining seat back (photo 6.12). The crimped tube member deflected inwards, causing the seat squab frame to miss its stop. Once the seat squab stop was rendered useless, the entire force was taken in the shaft of the reclining cylinder fractured at the threaded end of the cylinder shaft (photo 6.13). There was no noticeable damage to either the seat sub frame or the anchorages, nor was there any apparent skewness of the seat during testing, although upon unloading the inboard seat back was 25 mm aft of the outboard seat back.

This seat is identical to the one tested in Test No.1, except for the modification to the seat legs, which cantilever for the aisle side seating position. This test was used to confirm the results of Test No. 1 and to investigate the effect of sub frame changes (refer photo's 6.24 and 6.4 for modes of failure).



Photograph 6.13. Ultima failure of the seat resulted from fracture of the shafts in the two reclining devices, seen dangling from the front of the seat cushion framing.

V) Test No. 5 Peak Load - 2938 N Total Energy Absorbed 1183 Nm Elastic Stiffness 21198 N/m Yield occurred at approx. 70 mm deformation under a load of 1500 N. Max. def. 400 mm.





Photograph 6.14. Prior to Test No. 5. Note the absence of the seat back board and the pocketing channels used to retain it. Most route bus seats use plywood boards, usually 'thick, for the basis of both seat back and cushion. It was originally thought that the absence of these boards from the test seats would have very little effect on the test results, as it was considered that the backing boards would not alter the strength of the seat frames significantly. With this particular seat, the method of "pocketing" the seat back board was such that the timber was not held rigidly but instead, was slotted in between two stainless steel channels. Thus the board contributed no strength to the frame as it was not rigidly attached down the sides of seat back.

The seat failed by buckling at a point on the two seat back tubes between the two attachment points of the board (Photo 6.15). It is considered that if the board had been in place, it would not have influenced either the position or type of failure, not would it have altered the load at which buckling occurred. However, as the tubes buckled and the distance between the two back board locating channels decreased, the timber



Photograph 6.15. Note the height of the points of buckling and that they fall between the retaining channels for the seat back board.

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It is questionable whether there would have been sufficient freeplay between the board and the locating channels to allow the buckling of the tubes to continue to the point where the seat no longer was useful as a passenger retainer in the event of an accident. It is conceivable that the board may fracture, with the consequence of leaving a splintered timber edge and a possible source of injury. Alternatively, since the upper channel was held in place by two pop rivets, it is possible that the top section of the seat back could become dislodged. In this event the injury inflicting consequences were two fold. Firstly, the seat back tubes could probably be left protruding and unprotected and secondly, the upper section of the seat may have acted as potentially damaging projectile.

There was no apparent reason why the inboard seat back tube buckled earlier than the outboard one, nor was there any explanation as to why buckling occurred at different positions on the two seat back tubes. On the inboard tube, the height above the floor of the buckle was 470 mm, while on the outboard one, it was 545 mm. The angular deflection of the inboard tube, at the point of failure was larger than for the outboard one. The culmination of the different location of buckling, different angular deflection and a slight twisting of the seat sub frame, mainly due to asymmetrical seat anchorages resulting in the outboard side of the seat being stiffened by the wall mountings, resulted in a substantial degree of skewness on completion of the test. Once the load had been relaxed and 75 mm of elastic deformation had been recovered, there was a skewness of 8.5° with the aisle side of the seat back loading the wall side (Photo 6.16).

There was no anchorage failure although the wall mounting bracket, which is part of the seat and the means of attaching the seat to the wall had been distorted (Photo 6.17). The deformation was such that the front of the seat had moved down, while the rear had moved up.

There was also a slight amount of distortion in the floor anchorage plate, which is the means of attaching the seat legs to the floor (Photo 6.18). The seat legs had angled forward slightly as the plate had distorted, giving rise to the skewness of the sub frame. The lifting of the plate was more noticeable behind the front leg.



Photograph 6.16. The obvious skewing of the seat back can be seen. In the event of an accident, it is probabl that such a situation would not assist the ta of passenger retention.



Photograph 6.17. Post test inspection shows the distortion of the wall attachment bracket although there was no sign of failure.



Photograph 6.18. Slight deformation of the floor attachment plate caused by the angling of the seat legs about their bases.


Photograph 6.19. The maximum deflection achieved in Test No. (

IV) Test No. 6.

Peak Load - 2235 N

Total Energy Absorbed 1465 Nm

Elastic Stiffness 13849 N/m

Essentially elastic deformation then after yield the structure exhibited work hardening until final failure.

Yield occurred at approx. 80 mm def. under a load of 1100 N.

Max. def. 652 mm.



Fig. 6.6.



Photograph 6.20. Prior to Test No. 6.

As the seat was loaded the initial type of failure and approximate location of failure were identical to test No. 5. That is, buckling of the seat back tubing, except on this occasion it took place somewhat higher, viz: 690 mm above the floor and the point of failure was at the same height for both tubes. This seat was fitted with the wooden backing boards for both cushion and squab. The method of attachment of the seat back board was slightly different. Instead of having two channel sections that the board sat in, the lower edge of the wood was fastened to a piece of flat bar running across the seat while the upper edge was held by a channel. On further loading of the seat, there was a secondary mode of failure. This occurred at a deflection of 540 mm and resulted in a dramatic drop of load from the peak load of 2235 N to 1780 N in a distance of 10 mm. The condition of the seat back tubes, which had already buckled, bent at the base of the seat. The secondary failure occurred in this section of tube on the inboard side of the seat. This tube underwent considerable bending before a crack developed on the underside of the tube, which because of the bending was subject to tensile forces. The crack propagated quickly as indicated by the rate at which the load dropped off. The position of the fracture along the frame member was approximately in the middle of the section of the tube (Photo 6.21).

Interestingly, the wall anchorage bracket stiffened the equivalent tube member on the outboard side of the seat. Thus there was only slight distortion of this member and no sign of fracture. However, there was quite considerable distortion of the wall anchorage bracket itself. The distortion being of the same kind as for test No. 5, a twisting of the bracket so that the front of the seat had been lowered while the rear had been raised (refer Photo 6.17).

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Photograph 6.21. Maximum deflection reached in test No. 6. Note the two modes of failure. 1) Buckling of the seat back tubes, approx. 680 mm above the floor and above the upper edge of the stiffening insert tube. 2) Fracture of the cushion frame tubing on the aisle side between the legs (bottom centre). The skewing of the seat back is noticeable.

This assymptical secondary failure had introduced a slight degree of skewness so that the inboard side of the seat was leading the outboard side. There was no apparent anchorage distortion. The wooden seat cushion backing board had lifted away from the frame at the rear where it had been secured by one self tapping screw (one of three holding the cushion to the frame).

VII) Test No. 7.

Peak Load - 1531 N Total energy absorbed 733 Nm Elastic stiffness 21269 N/m Essentially elastic deformation followed by work softening. Yield occurred at approx. 40 mm def. under a load of 850 N. Max. def. 468 mm.



Fig. 6.7.



Photograph 6.22. Prior to Test No. 7. Failure of the seat was due to buckling of the four seat back tubes at the base of the seat squab (Photo 6.23). An inspection was carried out after the test to ascertain whether a stiffening insert tube had been used in the tube members at the base of the seat back. The tubes were sectioned and it was discovered that the insert tubes were missing.

There was no anchorage failure, although there was a slight twisting of the wall tapping plate. There was no apparent skewness of the seat.

This seat was not repositioned and retested because it was apparent from the form of the force/deflection plot that the seat had failed and any further energy absorbing characteristics were minimal and could be deduced from extrapolation of the force/deflection curve.



Photograph 6.23. The extremely localized buckling at the centre of bent tubing. No strengthening insert tube was found in this region of the frame. VIII) Test No. 8.

Peak Load - 1117 N
Total energy absorbed 714 Nm
Elastic stiffness 16025 N/m
Essentially elastic deformation followed by work softening.
Yield occurred at approx. 40 mm. def. under a load of 640 N.
Max. def. 660 mm.



A seat back board was fitted to this seat in the same manner as it would in production. It seemed possible that this board could alter the strength of the seat, largely due to the method in which it is fastened to the seat frame.



Photograph 6.24. Prior to Test No. 8. Note both the stiffening plate and bracket at the base of the seat back. Note also the somewhat thin, 2 mm non-continuous floor attachment plates at the base of each leg.

As it turned out, it was unlikely that the wooden back affected the test results at all.

The seat failed under a very low load (1117N) in a buckling mode at the base of the seat back (Photo 6.25). In an attempt to stiffen this section of the seat, the manufacturer had included a bracing bar on the inboard side of the seat and a bracing plate on the outboard side. These stiffening members buckled simultaneously with the seat tubing.

There was no apparent anchorage distortion or skewness of the seat observable after the test.

This seat was inspected to ascertain if a stiffening insert tube had been used. No such tube was present.

IX) Test No. 9. Peak Load - 2057 N Total energy absorbed 488 Nm Elastic stiffness 17586 N/m Yield occurred at approx. 90 mm def. under a load of 1600 N. Max. def. 459 mm.



Upon loading this seat, the aisle side seat back deformed noticeably more than the wall side seat squab (Photo 6.27). The seat squabs on this seat were pinned at their base in the same manner as a reclining seat, however they were fixed by a pin which fits into a bush welded to the seat squab tube next to the arm rests. This pin is welded to a plate which is bolted to the arm rest on both the wall and aisle sides of the seat (Photo 6.28 and 6.29).



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Photograph 6.25. Maximum deflection achieved in Test No. 8. Note the localized buckling of the seat frame tubing and the buckling of the stiffening bracket. No stiffening insert tube was used in the region of buckling of the frame.



Photograph 6.26. Prior to Test No. 9. Note the raised pedestal leg designed for a ramped bus floor.



Photograph 6.27. Noticeable skewness of the two seat squabs which eventually led to the loading bar slipping off the seat back.

Photograph 6.28.

The location of the seat back restraining pins and mating bushes can be seen just below the upper edge the cast arm rest. The box of the seat cushion frame member which locates the two central seat back tube is detectable.



Photograph 6.29. The position of the pins and bushes that restrain the seat back from pivoting are visible in the far left and right edges of the photograph. Note the twisting of the seat on the left.

Part of the weld retaining this bush on the aisle side seat squab fractured, allowing the seat back to deform forward. As a result of the high degree of skewness caused by the assymetrical failure, the loading bar slipped off the seat. The loading ram was retracted and a close examination of the seat undertaken. The loading bar was then replaced and the seat reloaded. The window side seat squab deformed forward and in so doing, reduced the degree of skewness. The seat squab on the window side failed at the same place and in the same mode as the aisle side seat. In both cases, the fracture of the weld and the steel tubing to which the hushes are welded were located to the rear of the bush where the tube member was in tension (Photo 6.30). The bar running across the seat at the rear of the seat cushion had welded to it the pivot mounts for the seat squabs. The bar as a result

of loading the seat bowed forwards and in so doing, caused the seat squabs to pivot and displace away from the centreline of the seat.

There was no deformation of the seat anchorages. The maximum skewness measured during the test was 11° and was in such a direction that it caused the aisle side seat squab to lead the wall side squab.



Photograph 6.30. The localized failure of the seat back tubes at their point of restraint from pivoting positioned at the rear top corner of the arm rest can be seen.

X) Test No. 10 Peak Load - 3418 N Total energy absorbed 2110 Nm Elastic stiffness 15828 N/m After yield had occurred work hardening took place until failure. Yield occurred at approx. 90 mm def. under a load of 1500 N Max. def. 784 mm.



Near the base of the seat squabs, but above their probable pivot points, was located a square crosssectional tube which ran across each seat squab. The spring and wire suspension used in the seat back was attached to this tube as was a lever which was connected to one end of the piston/spring reclining device. The other end of the infinitely adjustable cylinder was bolted to the front cross member of the seat cushion. As the seat was loaded, the square cross-sectional tube at the base of the seat squabs acted as a torsion bar and underwent twisting. At a displacement of 400 mm the weld holding the lever to the tube undergoing torsional displacement, fractured and the load dropped off.

There was no apparent anchorage failure, although there was a slight amount of distortion of the floor anchorage plate (Photo 6.31). This stainless steel plate had lifted away from the floor just in front of the rear anchorage bolt.

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Photograph 6.31. Maximum displacement reac in Test No. 10. Note th lifting of the rear of th pedestal leg.

Photograph 6.32. Prior to Test No. 11. Not somewhat narrow pedestal leg, attachment plates for arm rests and the contoure seat backs. XI) Test No. 11. Peak Load - 2302 N Total energy absorbed 1685 Nm Elastic stiffness 21499 N/m The linear elastic range was small after which work hardening took place until failure occurred.

Yield occurred at approx. 40 mm def. under a load of 800 N. Max. def. 630 mm.



Failure occurred at the base of the seat squabs in the four vertical seat back tubes. The mode of failure was buckling about 170 mm above the bottom of the seat backs. This coincided with the upper edge of the stiffening insert tube placed inside the lower portion of the seat back.

There was no noticeable skewness of the seat on completion of the test.

The anchorage system used for the legs of this seat used two $2\frac{1}{2}$ " x $1\frac{1}{2}$ " x $\frac{1}{8}$ " thick backing plates; one for each anchorage bolt. These plates were noticeably deformed on removal and inspection after completion of the test. Since this system did not rely upon a plate which was welded to the bus body, it was not surprising to observe a substantial lifting of the wooden floor around the rear anchorage bolt (Photo 6.33). The floor on the test bed was secured to the floor bearer in the same way in which it is done in most buses and the spacing of the self tapping screws was consistent with that found in most buses. No failure of the floor or any of its fasteners took place.

XII)Test No. 12

Peak Load - 1454 N Total energy absorbed 201 Nm Elastic stiffness 11431 N/m Yield occurred at approx. 90 mm def. under a load of 1030 N. Max. def. 183 mm.

Failure of this seat occurred in the device used to control the angle of the reclining seat squabs. A sudden fracture in the shaft of this device resulted in the squabs being unrestrained and free to collapse forward @hoto 6.35). The shaft in which the fracture took place was hollow with an activating rod through it @hotos 6.36 and 6.37). The location of the failure coincided with the first pitch of the shafts threaded end, which is normally screwed into a shackle located at the continuation of the seat squab tubing which in turn, acted as a lever since it extended below the pivot point of the squabs. It was the inboard or aisle side squab that collapsed, however, there seems no logical reason why this one should have failed earlier than the wall side squab.





Photograph 6.33.

Maximum deflection reached in Test No. 11. Note the localized bending of the seat back tubes just above the upper edge on the stiffening insert tubes. The forward angle of the pedestal leg is noticeable in the photograph (compare to photo 6.32) although the subsequent lifting of the floor is not.





Photograph 6.34 Prior to Test No. 12. Note the position of the reclining adjustme mechanism under the se cushion.

Photograph 6.35. Failure of the reclini mechanism of the aisle side seat resulted in the completion of the test. Note the lack o plastic deformation of the non-failed seat ba



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Photograph 6.36. The hollow shaft of the reclining adjustment piston shows the location of failure.



Photograph 6.37. The activating rod which fits into the hollow shaft of the reclining adjustment mechanism is shown protruding from the threaded end of the fractured shaft. There was no apparent skewness of the seat on completion of the test. Although there was no failure of the floor anchorages there was a noticeable lifting of the floor while the seat was loaded and after the test the backing plates were removed and found to be bent.

SUMMARY	RESULT	SHEET

Seat No.	Energy Absor. 355.6 mm (Nm)	Total Energy Absor. (Nm)	Energy Absor. at def. 600(Nm)	Seat Type	Max Force (N)	Max Def. (mm)	Elastic Stiff. (N/m)	Yie Def. (mm)	eld Lc ()
l	591	1272	1270	Reclin'g coach	1705	605	14289	90	13
ц	650	1506	1314	11	1750	715	21269*	80	12
10	1015	2110	1653	11	3418	784	15828	90	15
12	201‡	201	201‡	tt.	1454	183	11431*	90	10
2	833	1905 1562	1357	Fixed back coach	1905	765	21269	60	13
3	836	1506	1320	TT	1851	75 7	18620*	90	17
9	465	488	48 8 †	11	2057	459	17586	90	16
11	854	1685	1684	11	2302	630	21499	40	80
7	614	(861)** 733	826**	Charter	1531	(774)₩ 468	21269	40	85
5	1098	(1343)** 1183	1343†	Route	2938	(510)** 400	21198	70	15
6	704	1465	1414	17	2235	652	13849	80	11
8	467	714	690	13	1117	660	16026	40	64

- Non linear section of plot ignored in the early stages of the test.
- ** Plot was extrapolated to achieve maximum deflection.
- Energy absorbtion is the same for the three deflections due to seat failure at 183 mm.
- Energy absorbed at seat deflection of 600 mm to equal the maximum energy absorbed due to failure of the seat before reaching a deflection of 600 mm.

6.7 COMMENTS ON RESULTS

6.7.1 Classification by Seat Type

I) Long distance reclining coach seats. These seats are the most expensive and sophisticated seats of all those that were tested. There were four such seats tested (No. 1, 4, 10 and 12) and they covered a wide range of characteristics. The maximum force (3418N) and energy absorption (2110Nm) were obtained in test No.10, yet test No.12 displayed the worst energy absorption (201Nm) and the second lowest peak load (1454N).

All the tests with the exception of No.10 <u>failed by sudden</u> <u>fracture of the locking device used to control the reclining</u> <u>mechanism of the seat backs</u>. The probable reason for the failure of the seat in test No.12 was because the rod in the adjusting piston was hollow, to allow the releasing rod to run through it and threaded on the outer surface. Thus the tensile strength of the piston rod would clearly be greatly reduced due to the significant reduction in the cross sectional area of material and associated stress concentration effects.

These infinitely adjustable piston/spring devices do not have a positive locking mechanism. Instead, they rely upon either friction or the seal of a gas piston. As all reclining coach seats are nominally tested in their upright position, which normally means at the end of the travel of the seat ram, these tests did not set out to investigate the locking capabilities used in these devices. Indeed in a static test such as this, a component acting as a dash-pot develops a force proportional to velocity and will unnecessarily complicate the results and diminish their validity if included as part of the structural system. It is conceivable in the event of an accident, with the seat reclined that the piston mechanism

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could provide a favourable energy absorbing profile. The passenger would have a greater chance of being retained by the reclined seat due to the closer proximity of the seat back and the inclined angle of the squab which would tend to prevent ramping over the seat in front. Careful consideration would need to be taken in the construction and trimming of the top of any reclining seat squab, as it is potentially more dangerous due to its increased stiffness and the reduction of contact area initially presented to the impacting rear passenger. The increased stiffness is caused by the effective reduction in the moment arm of the seat back about the pivoting point, due to its inclination. The peak head and throat loading are potentially higher due to the reduction in the contact area of a result of this latter situation.

In the event of an accident with the seats in their upright position, the possibility exists of the seat backs collapsing forward and offering no resistance to the forward motion of the passengers. Should this occur, it would be an extremely dangerous situation likely to result in the bulk of the passengers being flung to the front of the coach and possibly exiting through the front windscreen. Even though the tests carried out were static tests and loading was applied to the top of each, the results are not very satisfactory. In the case of test No.10, which achieved a peak load of 1454 N, this would approximately equate to an equivalent 1.24 G (12.12 m/s^2) deceleration, if two 60 kg passengers were sitting in the seat behind and were projected into the seat back upon collision. Admittedly, the load application in these tests is at the upper extremity of the seat back which is probably not where the seat would actually be loaded in the event of an accident. However, the centre

of force applied to a seat back in the event of an accident will be near the top of the seat squab, particularly once the knees have hit the seat back tending to cause the passenger to move upwards and impact the chest area on the top of the seat. It is difficult to predict the effect of dynamic loading such as would be the case in an accident where the elastic/plastic characteristics could alter the performance of the seat insofar as the actual deceleration levels of impacting passengers is concerned.

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As mentioned, the seat tested in test No. 12, displayed a peak load of 1454 N which was the lowest peak force in this category of tests. If we consider test No.10, which exhibited the highest peak load (3418 N), this amounts to an approximate deceleration of 2.9 G (28.48 m/s^2). In most of the American and European studies into bus safety, a mean deceleration of 5 G and a peak deceleration of 10 G is considered appropriate for the deceleration characteristics of the bus in the event of a head on accident. Indeed, Adams et al4 considers a constant deceleration of 12 G from 30 mph to be a representative reconstruction of a head on bus accident. The anchorages used in the test of the four reclining coach seats remained intact, although the lifting of the floor during test No. 12 reinforced our opinion of the uncertainty of non-secured backing plates. Indeed, the fact that the floor backing plates bent and the floor noticeably lifted under such a small peak load of 1454 N, leads us to speculate as to the performance of $2\frac{1}{2}$ " x $1\frac{1}{2}$ " x $\frac{1}{6}$ " backing plates for individual anchorage bolts when subjected to higher loads. We consider that if the seat had been stronger, then failure of the floor may have eventually taken place. In such an event, the failure of the floor anchorages could conceivably result in the disengagement of the seat itself or perhaps the pivoting of the seat about its wall anchorage in forward motion, in either

case, the passengers would most certainly be unrestrained and projected towards the front of the coach.

This situation has been substantiated by a number of bus accident studies and test crash investigations carried out in the U.S. One study by Stansifer and Romberg² quoted 3% of the 1935 bus occupants involved in a sample of accidents studied were ejected from the bus. Of all the passengers that were ejected, 38% went through the front windscreen. Furthermore, an investigation by Runge, Siegel and Nahum¹ found that the ejection of a passenger will always result in a greater injury risk with the predominant area of serious injury being inflicted to the head.

The general use of some form of suspension for both the squab and cushion section in this type of seat we consider to be preferable in a crash situation. The benefit in having a soft-centred, energy absorbing cushion which when impacted by the knees will deform is significant, especially when the alternative, a rigid piece of $\frac{1}{2}$ " thick plywood is considered.

II) Non reclining coach seats

These seats are sometimes very similar to the reclining coach seats and major components such as the seat squabs, cushion and legs are common to both. The squabs in these seats are retained on their pivots, however, instead of a lever arm and piston attached to the lower part of the squab, a simple pin attached to the arm rest and slotted into the seat squab tubing is used to locate and hold the seat back. This pin is welded to a plate which is bolted to the die-cast arm rest.

The other style of non-reclining coach seat is one that is more closely related to a charter seat. The seat squabs and cushion sections of the chair are not components that are bolted together. Instead, similar to a charter seat, the side tubes of the cushion section bend at the rear of the cushion to form the frame tubing for the squabs. Unlike most charter seats, these seats have individually contoured seat squabs and both cushion and squab have either rubber or spring suspensions.

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A major consideration affecting the crashworthiness of these seats is the fact that they have four seat squab tubes which are either rigidly connected to the seat sub-frame or are the extensions of the sub frame itself. This it would seem, should be stronger than for example, a route bus seat which has two seat back tubes, one up either side of the chair. Since the tubes used in both route and non-reclining coach seats are generally of the same size diameter and wall thickness, the strength of the seat backs should be a function of the length and wall thickness and diameter of the stiffning insert tube, if one is used. The location of this insert tube is at the base of the seat back, where the main tubing frame is bent to form both seat cushion and squab. This is consistent with the results obtained from the three seats of this style that were tested (No's. 2, 3, 4, 11). The maximum loads exerted on the seats in test No's. 2, 3 and 11 respectively were 1905 N, 1851 N and 2303 N. These forces are higher than the maximum forces reached by three of the four reclining coach seats tested, the one exception, being test No. 10, where a squab frame tube acted as a torsion bar.

The seat tested in test No. 9, was a non-reclining coach seat, which used reclining coach cushion and squabs as described earlier in this section. There had been a modification made to the seat squab tubes to allow a locating pin to be slotted into the seat backs. This modification required the drilling of a hole through the rectangular cross-sectional seat squab tubing to allow the fitting of a bush which was welded into place. While this seat reached a moderate maximum load of 2057 N, the energy absorbed was the second lowest of all the seats tested, 488 Nm. The rectangular tubing (32mm x 12.7mm x 1.6mm) increased the strength of the seat squabs over those using round tubing $(1" \emptyset)$ due to the effective increase in second moment of inertia. However, it would seem that the weakening effect of inserting the seat squab locating bush, resulted in failure and quick reduction in the effective As a consequence, there was strength of the members. a rapid fall-off of load and hence a small amount of energy absorbed (488 Nm) compared to the three other fixed back coach seats (1562, 1506 and 1685 Nm for test No's. 2, 3 and 11). This argument was reinforced when the energy absorbtion fixtures for the reclining coach seats were compared. The effect of the bush can be seen if we compare results for test No's. 9 and 10 (fixed back and reclining coach seat respectively). The seat squabs used in these two tests were identical. The energy absorption figures are such that the weakened squabs on the fixed back seat managed to absorb only 488 Nm, while the reclining seat reached the figure of 2110 Nm. Furthermore, the reclining seat surpassed the maximum force exerted on the fixed back seat by 1361 N (reclining 3418 N, fixed back 2057 N). The weakening effect of the holding pin bush into the seat squab of the fixed back seat apparently was substantial.

The anchorages used in the tests of the three nonreclining coach seats, remained intact although the rear leg of the seat in test No.2 lifted away from the floor due to the bending of the floor anchorage plate, welded to the bottoms of each of the legs. The other two non-reclining coach seats showed no sign of floor plate deformation due to the different design of the legs and the anchorage plates welded to them. Test No. 9 used a pedestal leg with a 3 mm thick stainless steel plate, while test No. 3 used a single 5 mm thick plate connecting the bases of the two legs, while the configuration that deformed, used two plates, one on the end of each leg. It would seem that the single floor anchorage plate is superior to the design incorporating a separate tab on each leg.

III) Charter bus seats

This type of seat possessed high back seats similar to the coach seats and had either a single seat back or two individual seat squabs. Unlike coach seats which use a suspended seat squab and cushion and route bus seats which use wooden backing boards, the manufacturers of charter bus seats often combine the two systems. For example, the seat loaded in test No. 7 employed a wooden seat cushion board and a rubber suspension system for the seat back.

One seat of this variety was tested (test No. 7). As a result of the obvious failure of the seat in test No. 7 and the subsequent work softening which took place, we did not retest this seat to investigate its characteristics at greater deflections. It was considered that extrapolation of the force/deflection plot would represent its further behaviour reasonably accurately. Thus the predicted maximum deflection of this seat at total collapse (i.e. zero force) would be 774 mm and the additional energy absor**p**tion of the seat for this further displacement would be 128 Nm. The extrapolated energy absorbtion would therefore be 861 Nm at a seat top deflection of 774 mm. This seat

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managed a peak load of 1531 N, which was the third lowest of all the seats tested and its calculated energy absorbtion (861 Nm) was also the third lowest. The location of the failure and the lack of plastic deformation anywhere else in the seat frame indicated that there was possibly no stiffening insert tubes in the bases of the four seat squab tubes. After testing, the seat was sawn apart to inspect the internal construction of the lower seat squabs. Indeed, it was found that there were no insert tubes, which accounted for the relatively poor performance of this seat, both in terms of maximum force obtained and the amount of energy absorbed. The redeeming design feature of this seat was that it had four frame tubes which determine the performance of the seat back. Even so 1531 N maximum load was the third lowest result, while 861 Nm was the fourth lowest figure obtained for energy absorption.

IV) Route bus seats

This category of seat is (to the best of our knowledge) the cheapest and the most structurally simple of all the seats used in Australia. They employ a simple frame, with a low back and hard backed cushion and seat back. Although the back of the seat is low because the cushion sits above the level of the cushion frame, the distance from the top to the base of the seat back is about the same as the corresponding distance on the high back coach seats. Thus, at a given load, the effective bending moment, about the base of the seat back is approximately the same as the other seats.

Three route bus seats were tested (test No's 5, 6 and 8). The seat tested in test No. 5 was a stainless steel seat using 1" \emptyset 1.2 mm wall thickness tubing. An interesting combination of stainless and mild steel 1" \emptyset tubing was used for seat No. 8, while test No. 6 examined the properties of a mild steel frame again

using 1" Ø 1.6 mm wall thickness tubing. It was interesting to note that the seat combining the non-corrosive properties of stainless steel and low cost of mild steel used two different wall thicknesses, (1.2 mm for stainless and 1.6 mm for mild steel). There were two welds in the seat frame where the stainless and mild steel tubings were welded together.

The maximum force exerted on the stainless steel seat in test No. 5 reached 2938 N, the second highest figure obtained. However, once failure due to buckling, had occurred, the load dropped off, so that the seat had totally collapsed at a deflection of 510 mm and had absorbed 1343 Nm. This energy absorption figure was low when compared to the figure of 2110 Nm obtained by seat No. 10 with a maximum peak load of 3418 N. It was interesting to note that this seat, together with the all-mild steel route bus seat failed in a similar manner which was only exhibited by these seats. The buckling failure, instead of occurring either at the base of the seat back or at the top of the insert tube, occurred some distance up the seat back (approximately $\frac{1}{3}$ to $\frac{1}{3}$ way up the the seat back). The other route bus seat tested, the stainless steel/mild steel hybrid achieved the lowest peak load (1117 N) of all the seats tested and the second lowest energy absorption (714 Nm). Unlike the seat (test No. 12) which exhibited the lowest energy absorption figure, this seat did not fail suddenly by fracture, but rather buckled at the base of the seat back. Once the buckling failure had occurred, the load dropped off quickly. The location of the buckle and the relatively poor performance of this seat suggested that there was no insert tube to stiffen the seat backs. After the test, the seat tubing was cut out to investigate the internal

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construction of the lower seat back. No insert was found. The manufacturers of this seat had welded small stiffening webs on either side of the seat at the base of the seat back in an attempt to reinforce the bend in the tubes. As shown by the test results, these braces had little effect and once simultaneous buckling of the web and tube had occurred, were of little advantage. This seat is used in conjunction with what may be considered to be a possibly unsatisfactory floor anchorage device. However, due to the very low failure load of this seat, the opportunity to investigate the holding capabilities of the fastener was not forthcoming.

6.7.2 Anchorages

None of the anchorage systems tested failed. That is to say in the course of investigating the force/deflection characteristics of the twelve seats tested, the various combinations of tapping and backing plates, chair rails and fasteners held the seats in place. On occasions, the immediate component of the seat to which the anchorage fastener was attached underwent deformation. The components that bent were floor and wall anchorage plates. In the case of the wall mounting, the type of bracket that was most noticeably subject to deformation was the shaped plate used in conjunction with tapping plates. Obviously, the wider the plate, the greater the distance between the wall the the seat and the greater the deformation possible under given loading conditions.

With regard to floor anchorages, the plate through which the floor fastener is placed was often bent following a test. However, at no stage was there any sign of cracks or other forms of failure which would develop into the subsequent disengagement of the seat. The deformation of the floor plate assumed one of two variations:

- Both seat legs angled forward and the floor plate(s) consequently bent or;
- The entire seat attempted to pivot forward on the front floor mounting, resulting in a lifting of the aft of the seat perhaps the compressing of the front floor mounting into the wooden floor.

The authors consider deformation of the seat and anchorage brackets (attached to the chair) in the event of an accident, not detrimental to the safety of the passenger, as long as the deformation is plastic and is designed into the structure and does not lead to catastrophic failure. The problem exists that if the seat components plastically deform under inpact loadings, then it is necessary to know how much further the component could deform before failure. In this series of tests, it was established that indeed it is the seat and not the anchorages that fail first.

This raises an important point. Due to the very light peak loads reached in some of the tests, the anchorages were not called upon to perform and thus in effect, were not tested. This is born out by test No. 8 where it was expected that the floor anchorage may fail. The anchorage used no tapping or backing plate and relied upon a female fastener with a contact surface area with the floor equivalent to that of a one cent piece. However, the seat collapsed at such a low load that the anchorage or the surrounding floor structure did not appear damaged in any way.

The most successful floor anchorage, in terms of lack of deformation incorporated the newly introduced pedestal leg. Again, however, it should be questioned what the performance of such a design would be if the seat backs were able to

withstand greater loads. The justification for concern over the pedestal leg is that it effectively increases the chances of failure of the fastener, due to the significant reduction in the distance between the two fasteners. This has the effect of reducing the moment arm and thus increasing the loads on the individual

fasteners.

6.7.3 The Implications of Peak Loads

Investigating the maximum load which can be withstood by the seats before failure is useful for comparisons from test to test. If this load is too high then possibly in an accident situation, the passenger may be subject to loads that could be beyond the normal impact tolerance of the human body. Thus serious injury or death could occur. Of course, the accident loading situation is an extremely complex situation whereby the movement of various parts of the body on impact with the seat determine how subsequent parts of the body will move and thereby determine to a degree the loads imposed on those various parts of the body. The injury severity sustained in an accident cannot, however, be directly related to peak loads. Rather, the injury severity is dependent upon:

- 1) the direction of deceleration,
- 2) the magnitude of deceleration,
- 3) the duration of deceleration,
- 4) the rate of onset of deceleration,
- 5) the type of deceleration (linear or angular).

If the complicated nature of the motion of the passenger relative to the seat in front of him is disregarded, then the factors influencing the correlation between force applied to the passenger and deceleration of the bus are the mass of the occupant being decelerated and the force/deflection characteristics of the seat in the direction of impact.

INJURY INDEX (vs) DECELERATION



UPPER LIMIT

LOWER LIMIT Figure 6.13 presented by Severy et al^{35} shows the relation between peak head deceleration and an injury index. Injury indices above 1000 are regarded as severe to fatal. Furthermore, it is worth noting that Sarraile et al^{3+} states that, with regard to the head, a deceleration of more than 80 g without adequate distribution of the impact force, will probably be fatal. With a suitably distributed load, peak decelerations in excess of 100 G may be tolerable. Sarraile et al^{3+} also consider that blows to the face, involving decelerations in excess of 30 G's will probably cause unconciousness. This appears to be consistent with Severy et $al^{1}s^{35}$ graph.

In the summary report by Adams et al^4 the following restraints were imposed as design criteria concerning injury in a frontal bus accident.

- Head: Resultant deceleration not to exceed a Head Injury Criterion (HIC) of 1000* at the centre of gravity of the head.
- 2) Thorax: Resultant deceleration not to exceed 60 G's (upper) except for intervals whose cumulative duration is not more than 3 milliseconds.
- 3) Femur: Maximum axial force not to exceed 1700 lbs (7562 N) for 50th percentile male adult 1000 lbs (4448 N) for 5th percentile female adult 600 lbs (2669 N) for 50th percentile 6 year old child.

If we consider two passengers each having a mass of 60 kg then the peak deceleration of these people when seated in the seat used in test No. 10 (maximum peak load of all the seats tested 3418 N), would be approximately 28.48 ms^{-2} (2.9 G) if their point of contact was to top of the seat back. As can be seen by the above graph, this amounts to very low risk of injury due to the impact. Compare

* The HIC is as an injury scale so defined that a score of 1000 is rated as severe to fatal HIC = $(t_1 - t_2) \left[\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} a(t) dt \right]^{2.5}$

where, a(t) = resultant acceleration magnitude of the centre of gravity of head (G) and t_1 and t_2 are the two points in time during the impact for which HIC is maximum measured in seconds. this to dynamic sled tests carried out by Adams et al⁴ where peak head decelerations of 75 G's were obtained with a resultant HIC score of 300*. Another test after a seat back padding modification had been made achieved a peak head deceleration of 65 G's, which incurred a HIC score of 250. It should be noted that the HIC scores operate on a formula which incorporates such factors as the distribution of load, rate of acceleration onset (jerk) and duration together with the type of injury sustained. Thus it can be seen that the risk of major head and upper thorax injuries in the seat tested in this project can probably be considered as low.

Whether the femur loads would be acceptable in the seats tested in this project is unknown. However, it is worth noting that no major deformation occurred to any of the seats tested below the lower seat back area which is in the vicinity of the knee impact region. This means that in this region of the seats, the frame is strong. Furthermore, due to the lack of padding and the location of rigid seat frame members preventing the knees from penetrating the seat back, there is a possibility of knee/femur injury.

The question that needs to be answered in conjunction with the failure load analysis above is; will the seat restrain the passenger and prevent him from ramping over the seat once the seat has been impacted?

Although in the event of an accident, the possible deceleration levels of the seats tested in this project would be low, the lack of adequate padding could significantly boost the injury severity.

6.7.4 Energy Absorption

The failure of six of the seats caused by sudden fracture or buckling, resulted in the collapse of the seat with the effect that before full travel of the loading ram had been reached, the seat was no longer resisting forward movement. In such an event, once the load had fallen to zero, the energy
absorbtion potential of these seats had also dropped to zero.

Some of the seats tested and particularly the low back route bus seats had angled forward so far that in the event of an accident, the ability of the seat back to prevent the passengers from ramping over the top of the seat would have had to be very low. For this reason, it was decided to investigate the energy absorbed by the seats after the seat tops had deflected forward by 600 mm. In the absence of any dynamic test evidence it was considered that beyond this deformation, it would be questionable whether the seats possess any passenger retention value due to excessive inclination of the seat back and the lack of knee penetration area.

As it can be seen by Fig.6.14, the drop in energy at a displacement of 600 mm for the seats that had not collapsed is relatively minor. Thus even if we take the maximum total energy absorbed (e.g. in test No. 10 - 2110 Nm), the calculated initial speed of the bus on collision, such that the kinetic energy of two 60 kg passengers is totally dissipated in the seat is about 20 km/h.

If we regard 366 mm (14") to be the maximum deflection of the seat back allowable in order to achieve an effective passenger restraint⁷, then the speed of the bus would be only 15 km/h for the seat which displayed the most energy at 366 mm deflection, (this maximum deflection is the design criteria used by Lewis¹⁵ in the development of a safety bus seat). If we study the seat which achieved the lowest absorption (test No.12), it would be found that for the seat to have totally collapsed yet retain two 60 kg passengers, the initial velocity of the bus in impact would have been only 7km/h. Of course, there are many assumptions made in this elementary method of correlating static test results to a dynamic crash situation. Further full scale dynamic testing would have been preferred by the authors. Indeed previous proposals had been prepared to this effect but failed to reach approval.





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Fig. 6.14.

The most difficult factor pertaining to the performance of bus seats to retain a passenger in an accident situation is the relative movements of parts of the body and the way in which they interact to influence the points of contact with the seat. The points of contact of various parts of the body (predominantly knees, chest, neck and head) with the back of the seat and the consequent loads applied to the seat would influence the capabilities of the seat to retain the passenger. If a major proportion of the load was taken high on the seat back, which is more likely to be the case with the low backed route bus seats, the seat's capability of absorbing the passenger's energy would be less than if the load was taken predominantly lower on the seat back. A point that needs to be remembered is that as the proportion of overall load distribution is lowered on the seat back, the greater the risk of injury as a result of increased effective stiffness of the lower seat back region in conjunction with a seat frame design which is not conducive to knee penetration.

Fig. 6.14 shows the correlation between energy absorbtion and peak load. Three plots are displayed, each one representing a different energy absorbtion criteria eased on the maximum permissible seat back deflection. Energy absorbtion figures for seat back deflections of 366 mm, 600 mm and the maximum deflection are plotted.

6.7.5 The Implications of Static vs Dynamic Testing.

The static load tests carried out during this project give the stiffness characteristics of the seat structure. The results of these tests are useful for the comparison of peak loads, energy absorbtion, elastic stiffness and the general force/ deflection curve shape from seat to seat. Although these results give an indication of the performance of the seat in an accident situation which loads the seat in a dynamic mode, they do not give any indication of the injury potential the seat has upon the passenger. It is possible however, using judgement and experience based on previous studies that have combined static and instrumented dummy dynamic tests to speculate the movement and likely points of contact of the passenger. To ascertain the possibility, type and severity of injury would involve either specific static tests using knee, torso and head forms or preferably instrumented dummy, dynamic tests. Assumptions concerning the movement of the passenger have to be made in the static tests using body forms. Furthermore, these tests require further assumptions concerning load duration, rate of load increase and the distribution of the load in order to predict possible injuries and their severity. The injury causation is dependent upon the movement of the passenger which is governed by:

- 1) Initial velocity of the bus on impact.
- The deceleration profile of the bus (peak and duration of deceleration).
- 3) The orientation of the seat (forward, side or rearward).
- 4) The observations of the passenger and his ability to foresee a collision situation.
- 5) The distance through which a passenger has to move before striking an object which will restrict his motion. According to an American investigation cited by Adams⁴ et al "an overwhelming cause of injury in school bus collisions were the seats". Furthermore, passenger seats contribute to over 90% on the injuries of minor and moderate accidents and 90% of all accidents were of a minor or moderate severity.
- 6) Penetration of the survival space.

The subsequent motion of the passenger upon an impact with the seat is dependent on:

- 1) The mass of the passenger
- 2) The varying stiffness of the seat with progressive deformation.
- 3) The height of the seat back.
- 4) The geometry of the seat.
- 5) The phasing of body component movements. The human body in a real crash situation is more like a combination of small masses strung together rather than a rigid lump mass. Thus there is a tendency for a "whipping" of such parts as the head during a collision. As a result the motion of the body depends upon the bodily components.

The factors which influence the type and severity of injury are:

- 1) The peak deceleration.
- 2) The direction of deceleration.
- 3) The deceleration duration.
- 4) The level of jerk or the onset of deceleration.
- 5) The distribution of the load.
- 6) The body motion phase control ("whipping").
- 6.8 SUMMARY OF STATIC TESTS CARRIED OUT ON BUS SEATS OTHER THAN THOSE PERFORMED DURING THIS PROJECT.

Stiffness curves were obtained from previous tests carried out by International Harvester and Chrysler on three Australian built bus seats. The stiffness curves for these tests are presented in Figures 6.15, 6.16 and 6.17. The loading of the seats was achieved in a similar manner to the method used in the tests conducted during this project. Although the tests were performed in 1974, the design of the seats was essentially the same as the fixed back coach seats that were tested in this investigation.







Another static test was performed by International Harvester on an experimental seat designed by $Hoffmann^{33}$ and the results of that test are tabled along with the other three test results. The force/deflection curves for the Hoffmann seat is shown in Figure 6.18 along with Table 6.1 showing the summary of the results.

The force/deflection curve for an American seat designed by AMF is presented in Figure 6.19 and was tabled in a report by Adams .

A total of five anchorage tests were also carried out and the results of those tests are displayed in Table 6.1.



Fig. 6.18.



Seat	Max.force	Energy absorbtion at 14" def.	Elastic Sti	ffness		
A	2300 N	825 Nm	59055 Nm	Summary of the tes results carried or by International Harvester and Chry		
В	2904	1146	3502			
С	3861	1733	85808			
D	25500	4032	899830 **			
deflection of 16 cm where the load diminished to zero.						
No 5	2038	1099	21198	Summary of maximum		
No.8	111 7	467	16026	obtained during ou		
No.12	1454	201	11431	test program.		
<u> </u>		•				
AMF	16902	4405*Nm	525354 Nm			

TABLE 6.1 Results of Tests on Other Seats

* The static deflection test carried out on this seat terminated at a deflection of 203 mm. Due to lack of positive failure and the associated difficulty in the consequent extrapolation of the compliance curve, the energy absorbtion is calculated at a maximum deflection of 203 mm. (Note all other energy figures in this table are calculated at a maximum deflection of 355.6 mm (14").

These results are consistent with the figures obtained from the current program. The difference in loading procedure affects the comparison of the results slightly as application of the load in the tests performed by both International Harvester and Chrysler was somewhat below the top of the seat back. Thus we would pred that this arrangement ought to increase the recorded peak loading compared to the results obtained by the current test procedure. Apart from the experimental seat of Hoffmann³³ which displayed a maximum load of magnitude much higher than any of the other seats tested, the peak force and energy absorbtion figures of the other seats are indeed very similar. The experimental seat achieved its peak load at a displacement of 76 mm. Hence, the seat was very much stiffer than any of the other seats, in fact, it was an order of magnitude higher.

If however these results are now compared to a similar static deflection test carried out on an American safety bus seat manufactured by AMF, we find that the AMF seat has a maximum force and elastic stiffness which is much greater than any of the other seats with the exception of the Hoffmann experimental seat. The AMF seat did absorb slightly more energy than the experimental seat. Further tests were carried out on a dynamic sled to determine its crashworthiness on the AMF seat and its ability to retain passengers, by absorbing their kinetic energy. The loads exerted on the instrumented dummy had to comply to the limits set down in the Notice of Proposed Rule Making (NPRM). Suffice to say at this stage, that the seat was not strong enough to withstand the NPRM seat load deflection requirements. However, the injury producing loads and decelerations were measured on the dummy during the dynamic test, under a 12 G square wave deceleration of duration of 210 milliseconds from 30 mph. An ECE paper³⁶ comments on the "strength of seats and their anchorages" and states that the majority of current production seats failed under decelerations of 6 G.

For the purpose of comparison, Figure 6.20 shows the characteristics of the strongest and weakest seats tested in this project along with the behaviour of the AMF and Hoffmann seats.



As an extension of the static tests carried out by International Harvester and Chrysler, the deformed seats were reworked to approximately their original geometry and braced. The bracing extending from the seat back top to the seat cushion frame front, achieved a stiffer structure which could then be loaded in order to test the seat anchorages.

Table 6.2 shows the load at which anchorage failure occurred and the mode of failure.

If we again attempt to predict how the dynamic crash situation relates to these static test results by assuming that the point of load application is the same in both cases, then a peak load of 5800 N (continuously applied) will decelerate two 60 kg passengers at about 5 G. Head-on bus accident decelerations have been shown to involve an average deceleration of 5 G's with peaks of 10-12 G's. Indeed in a study conducted by Wojcik et al⁵ measured peak deceleration during a head-on collision between a truck and a bus both travelling at 30 mph of 21 G.

Anchorage Type	Load of Failure		Mode of Failure
3 grade 5 ³ 8" bolts(floor) 2 grade 5 ⁷ 16" bolts(wall) (Experimental seat)	25500 N (without any sign of anchorage failure)		No failure
2 floor bolts	4060 N		The wall anchorage
2 wall bolts			bolts pulled through
2 will bolts2 large load distributing washers under the floor			the sheetmetal wall member.
2 floor bolts	5800 N	I)	The wall anchorage
2 wall bolts			bolts pulled through
2 large load distributing washers under the floor			the sheetmetal inner skin.
		п)	The rear leg anchorage plate through which the bolt is placed tipped away from the leg.
2 floor bolts	5800 N		The rear wall anchorage
2 wall bolts			bolt pulling through
2 large load distributing washers under the floor			the walls innerskin.
2 floor bolts	4226 N		Forward seat leg
2 wall bolts			pushed through the
2 large load distributing washers under the floor.			floor.

TABLE 6.2 Results of Anchorage Tests

6.9 CONCLUSION

The authors consider that the sample of seats tested in this program were a fair representation of seats being manufactured and fitted to Australian buses and coaches. Furthermore, we conclude after inspection of the failed seats and analysis of the test data that:-

- 1) All the seats failed at relatively low loads (below 2500 N)
- 2) None of the seats appear to comply with any of the studied European and American seat strength standards*.
- 3) Anchorage failure did not occur in any of the tests.
- 4) Three out of the four reclining coach seats tested failed by fracture of the reclining clamping device, which resulted in the sudden total collapse of the seat squabs with minimum energy absorbtion.
- 5) Excluding the reclining seats, the predominant mode of failure was by buckling of the seat frame tubing in the lower region of the seat back.
- 6) The two seats which failed by buckling at very low loads (test No.7; 1531 N and test No.8; 1117 N) had no stiffening insert tube in the lower seat back region.
- 7) Generally seat skewness was not common with the load applied centrally.
- 8) There was very little wall structure deformation and floor movement was only noticed when the anchorage system employed did not include a tapping or backing plate secured to the bus body. In this case, considerable lifting of the wooden floor was evident in the vicinity of the rear floor anchorage bolt.
- 9) Floor anchorage backing plates with dimension $2\frac{1}{3}$ " x $1\frac{1}{3}$ " were found to bend during testing.
- 10) There was no noticeable damage to any of the fasteners used in any of the tests.

11) The deformation characteristics and the energy absorbtion figures appear to be inadequate to restrain passengers when a head-on collision involving an initial speed of 60km/h is studied using various simplifying assumptions concerning the loading of the seat necessary to approximately correlate static test results to a dynamic situation.

CHAPTER 5

EXAMINATION OF BUSES INVOLVED IN BUS ACCIDENTS

5.1 INTRODUCTION

The purpose of inspecting buses involved in an accident was to seek practical field data on the performance of bus seats as a result of passenger impact. Furthermore, it was intended to inspect the buses, with the aim of establishing points of bodily contact and the cause and reason for injury. Thus it was therefore important to inspect buses which had either suffered some form of seat deformation or damage or had been involved in an injury to a bus passenger. Extending this further, it seemed possible for injuries to occur, without the bus having a collision. For example, injuries sometimes occur due to the rapid manoeuvre of a bus, such as when braking, accelerating or swerving. This type of bus injury is particularly prevelant in route/transit buses, where the incidence of stopping/ starting and cornering, together with passenger movement into, out-of and within the bus is high. The types of injuries caused by non-collision incidents are typically low in severity and are largely as a result of tripping. Normal bus inspections after an accident are aimed at establishing the cause of the accident with little or no attention being given to the cause of the injuries. This has been noticed even at the very few accidents attended. Perhaps the inspectors do note that a particular object on the bus is potentially injury inflicting, however there is no established channel of communication open for reporting such an observation.

It was not considered important to inspect the bus at the site of the accident, especially as accident scenes were usually busy with the necessary functions being performed

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 CONCLUSIONS

One of the principal outcomes of this investigation has been the highlighting of the role of the bus seat during an accident situation. It clearly emerges that in the event of an accident, the seat should effectively retain the passenger throughout the impact and allow deceleration to be achieved with a minimum amount of damage to the body.

In order to achieve this, not only must the seat remain firmly attached to the chassis of the bus, but its deformation characteristics should be such that a maximum amount of energy is absorbed at prescribed maximum peak loads.

Past accident studies have indicated that this has not always been achieved with the result that passenger injury due to lack of retention or from the impact of unrestrained seats with otherwise uninjured occupants has occurred.

In an effort to maximise passenger protection, the design of bus seats and their anchorages has generated a considerable amount of work both on the part of designers, legislators and researchers.

Among these groups it is generally agreed that two major objectives need to be achieved in seat performance;

 in the event of a passenger impacting the seat in front, the seat should be capable of local deformation in the knee-chest area to enable "pocketting" of the passenger, so absorbing some of his initial kinetic energy together with controlled deformation of the seat back (without fracture) to absorb the remaining kinetic energy

and prevent the passenger ramping over the top of the seat.

2) through careful design and placement of structural members and the use of adequate energy absorbing padding, the seat should be capable of distributing local impact forces to the head, thorax, chest and knee areas in such a way as to prevent serious injury.

The testing program conducted during this project on a representative sample of bus seats currently being fitted to Australian buses, revealed that all of the seats either collapsed plastically or fractured at relatively low loads (less than 3500 N applied horizontally and forwards to the top rail of the seat back) and absorbed correspondingly low levels of energy in the process (less than 2200 Nm).

None of these seats would have satisfied all the requirements on crashworthiness, force-deflection profiles and energy absorbed at given deflections of the major European and American Bus Seat Standards currently being developed and used.

A lack of adequate energy absorbing padding in the region of knee and head/chest impact was commonly observed. It should be noted however that while in Victoria it is now mandatory for low back seats to incorporate a padded roll-top section over the exposed bar at the top of the seat back, this is not necessarily the case in other States of Australia.

In the course of the testing program, it was found that none of the anchorages failed. In some cases, wall mounting brackets and individual floor mounting plates were bent following a test but at no stage was there any sign of cracks or other forms of failure which could have developed into subsequent disengagement of the seat.

Nevertheless, it should be noted that because of the relatively low collapse loads of the seat backs tested, the anchorages were probably never stressed to their full capacity.

If seat back strengths were to be increased, then a further investigation of seat anchorages would be desireable.

Of all the floor mounting techniques tested, it is concluded on the basis of good engineering practice, that continuous floor mounting plates (rails) welded to the bus chassis along its length, would provide the best anchorage basis for bus seats.

Of particular importance was the failure mechanism of the long distance redlining coach seats. Four such seats were tested and in all but one the failure mechanism was a sudden fracture of locking device used to control the reclining mechanism of the seat backs. In one case it seemed quite clear that the reason for the failure was because in the vicinity of the break the rod in the adjusting piston was hollow (to allow the releasing rod to pass through it) and further, was threaded on its outer surface. Thus, it is believed that the tensile strength of the piston rod had bees significantly reduced due to the reduction in cross-sectional area of material and associated stress concentration effects at the thre roots.

The restraint of passengers during a collision is essential if the number and severity of injuries sustained is to be kept to a minimum. On the evidence in the literature, the fitting of lap type seat belts into buses appears to be neither cost-effective no efficient in reducing accident trauma. It would be difficult to fit lap-sash belts because of the absence of a suitable above shou mounting point for the sash. However, through careful design of seats with adequate energy absorbing padding for knee and heat/che impact regions, a structural design which allows penetration of th knees into the seat back with controlled overall seat back deformat together with correct seat back height, seat spacing and layout; t. occupant could probably be effectively restrained during an accide

Nevertheless, even if a seat is fitted with the above energy absorbing padding, the collapse of such a seat would render these protective devices almost useless, insofar as the occupant could be free to be projected out of his seat. Further to these conclusions, the desirability of improving the secondary safety aspects of buses became evident. To achieve this the entire passenger compartment has to be investigated. If the window pillars and roof structure are insufficient to withstand the loads created in the event of a roll-over accident, then the fitting of properly designed safety seats may not affect the injury rate or severity to any great extent. The same is true for any situation which results in the destruction and/or invasion of the passengers survival space.

Thus not only do bus seats need to be properly designed, padded and securely anchored but the bus structure needs to be capable of remaining intact and resistant to penetration. In addition to this the internal fittings and layout of the interior of the bus can affect the injuries sustained. For example, stanchions, ashtrays, fare-boxes and window latches have all been known to inflict injuries upon bus occupants during an accident.

Both the transit bus accident statistics of the MMTB bus fleet and overseas studies of transit bus accidents indicate that there is an injury causation problem unique to transit bus operation. There is a very high incidence of injuries caused by falls in the bus due to non-collision situations. It would appear that these injuries, which are typically low in severity, could be reduced by careful design of passenger assists, seat backs, floor ramping angles and step size and rise. This high incidence of passenger injuries has been found in overseas studies to be partially due to the lack of driver education concerning smooth driving. As a consequence, levels of acceleration and jerk combined with poorly designed internal layouts and the general nature of transit bus operation results in this problem. High injury risk areas have been established in the vehicle and are substantially localized to the areas of the entrance/exit steps and the front platform area near the driver. Rigid objects especially with sharp protrusions or edges, are obviously extremely dangerous and in many cases the redesign of the fare box is necessary due to its prominence as an object causing a high incidence of severe infuries.

Upon studying Australian bus accident case studies, Victorian bus accident statistics and overseas studies into bus accidents and passenger protection, the conclusion has been drawn that the risk of injury to the passengers in a bus which rolls over is high. The lack of passenger retention which results in uncontrolled body movement and passengers impacting internal bus fittings (largely seats) and other passengers in one of the three most common means of passenger injury in roll-over accidents. Another common injury and one which probably results in the most severe injuries, results from wither partial or full passenger ejection through either windows or doors, or through openings in the passenger compartment caused by the impact of the collision. The remaining mode of injury in the roll-over accident involves the collapse of the side wall/roof structure and consequently a deterioration of the passengers' survival space. Legislative bodies in Europe and America are considering draft regulations concerning the strength of the upper bus body so as to withstand bus rollover. It has been shown in oversear research projects, that adequate strength of the window pillars and of the roof structure in the area of the cant rail is obtainable through careful design.

From studying Victorian bus accident statistics and indepth accident case studies, it is evident that buses are a safe method of transporting people, when compared to the injuries sustained in car accidents. The predominant reason for bus travel displaying such a safety record is the inherent inertia of buses and the fact that in a bus collision, the most commonly impacted object is a car which has far less inertia than a bus and consequently, is subject to correspondingly higher deceleration levels. Thus the injury record of bus accidents appears to be dependent upon the physical nature of the vehicle rather than the designed crashworthiness.

7.2 RECOMMENDATIONS

With respect to the conclusions drawn in this project, it is recommended that the following investigations should be commenced in order to study various problem areas associated with bus travel.

- Further static force/deflection tests to be carried out on the existing test jig. These tests would use reinforced seats and would investigate the ultimate strength of the seat anchorages.
- Dynamic bus seat tests designed to investigate passenger retention and the loads sustained by a bus passenger in a head-on collision.
- 3) Bus crashworthiness investigation. This work would attempt to establish the strength and resistance to deformation of the bus body. There are two viable methods of such a study:
 - (i) Finite element computer programs,
 - (ii) Full scale testing of a section of bus body.
- 4) Development work on the requirements for National Legislative guidelines on the strength of bus seats. their crashworthiness, the strength of anchorages and the energy absorbing characteristic of bus seats, along the lines of the European and American Standards, but taking into account prevailing Australian conditions.
- 5) Further to this, consideration should be given to the introduction of National requirements for the fitting of properly designed "roll-top" energy absorbing padding for low-backed bus seats.

REFERENCES.

- Siegel A.W., Runge D.E. and Nahum A.M.
 "Bus Collision Causation and Injury Patterns."
- 2. Stansifer R.L. and Romberg R.A. "An analysis of accidents involving buses and an assessment of the need for safety belt requirements in such vehicles."
- Johnson, M.A.
 "The National PSV Accident Survey A Vehicle Design Oriented Study." Motor Industry Research Association. Inst. of Mech. Eng. 1977.
- 4. Adams L., Khadilkar A., Pauls L. and Rup W.
 "Development of a unitized school bus."
 Vol. 1, 2 and 3.
 US DOT NHSTA August 1976.
- Wojcik C.K., Sanoles L.R. and ITTE Staff.
 "School bus seat restraint and seat anchorage system."
 US DOT August 1972.
- Severy D.M., Bink H.M., Baird J.D.
 "School bus passenger protection." Inst. of Transportation and Traffic Engineering. Department of Engineering, University of California. L.A. January 1976.
- Adams L.
 "School bus passenger seat testing."
 NHTSA. US DOT AMF System Laboratory. July 1975.
- 8. Leyland National bus barrier test (film).
- Hartley, J.
 "A Bus for America GM's RTS" Automotive Engineering Vol 1. No. 6. August/September 1976.

10. Shanley J.W.

"Safety in Mass Transport : A case study of bus accidents in Washington D.C."

Urban Mass Transportation Administration October 1974.

11. Mateyka J.A.

"Safety Considerations In Design of New Transit bus seats." Booz-Allen Applied Research. 18th STAPP Car Crash Conference.

12. Purswell J.L.

"A study of post crash bus evacuation problems." 22nd Annual Meeting of the human factors Society. October 1978.

- 13. Ursell C.R."A study Relating to seat belts for the use in buses."South West Research Inst. San Antonio Tex. January 1977.
- 14. Keeman D., Miles J.C., Sadeghi M.M., Tidbury G.H."Investigation of PSV roll over safety."I Mech Eng. Symposium. March 1977.
- 15. Lewis F.J., Trunkey D.D., Steele M.R. "Autopsy of a Disaster : The Martinez bus Accident." The Journal of Trauma Vol 20 No. 10. October 1980.
- "Bus Interior Design for Improved Safety." Booz-Allen Applied Research. April 1976.
- 17. Brooks B.M., Edwards H.M., Fraser C.R., Lewis J.A."Passenger Problems on Moving buses."An Analysis by the Leyland Human Factors Group.August 1978.
- "New Bus Lifeguard."
 Traffic Safety. Vol. 23 No. 4, April 1973.
- 19. Keeman D. "Bending Collapse of Rectangular Section tube in Relation to the bus roll over problem." Published 1980.

- 20. Northrop G.M., Ball J.T., Bancroft D.R.E., Reidy J.C. "Evaluation of nine federal Motor Vehicle Safety Standards." US DOT NHSTA March 1978.
- 21. Report of the group of rapporteurs on safety provisions on motor coaches and buses (GRSA) 22nd session."Strength of the Superstructure of public service vehicles."Economic Commission for Europe.14 December 1979.
- 22. Sittler E.L. "Structural verification of the bus passenger seat." US DOT NHTSA June 1973.
- "Prototype bus by Neoplan." Automotive Engineering Vol. 1 No. 6. August/September 1976.

24. "Transbus Public Testing and Evaluation Program." Booz-Allen Applied Research. January 1976.

25. Tovey M.

"Dual Mode seat for Improved efficiency in Bus Operation." Transport Ergonomics and Design Education Ergonomics Vol. 22 No. 2.

- 26. McHugh J., and Tidbury G.H. "The design of Touring Coach Bodywork for safety." Institute of Mech. Eng. conference. Vehicle Safety Legislation Cranfield Inst. of Tech. July 1973.
- 27. Snyder R.G. "Bioengineering of Impact survival in Business Aircraft." SAE National Business Aircraft Meeting. March 1969.
- 28. Brooks B.M. "An investigation into aspects of bus design and passenger requirements." Ergonomics Vol. 22 No. 2, 1979.
- 29. Lewis J.A. "The seated bus passenger - A review."
- 30. "Truck and bus accidents." Vol. 1. Traffic Accident Research Unit (TARU) NSW.

- 31. Traffic Regulation Board's "Post Accident Vehicle examination reports."
- 32. "Special Study inadequate structural assembly of schoolbus bodies." National Transportation safety board. July 1970.
- 33. Hoffmann E.R. "Low Cost Protection for bus passengers." Department of Transport. January 1975.
- 34. Sarraile S.R., Wood C.F. "Heat impact protection on an airline seat." Department of Supply. June 1967.
- 35. Sevey, D.M., Bink H.M., Baird J.D. "Passenger Protection from front end Impacts."
- Report of the Group of Rapporteurs on safety provisions on motor coaches and buses (GRSA), 24th Session.
 Economic Commission for Europe.
 11 - 14 November 1980, Hungary.

Additional literature relavant to the investigation.

- Boulay P.F.
 "The school bus safety improvement program."
 Proceedings of 19th conference of AAAM.
- * Barltrop L.
 "A review of initial investigations into the construction of safe bus bodies."
 National Institute for Transport and Road Research CSIR.
 South Africa, March 1980.
- Cowley J.E. and associates.
 "Heavy Commercial Vehicle speed and Operational Safety."
 ROSTA, December 1978.
- * Appleby M.R.
 "School bus back pads the California experience."
 Proceedings of 19th conference of the AAAM.
- * Barltrop L.
 "Passenger Seats in Public Service Vehicles."
 National Institute for Transport and Road Research CSIR.
 South Africa. May 1980.
- * Vulcan A.P.
 "Vehicle design to Reduce Occupant Injury."
 National Road Safety Symposium.
 14 16 March 1972.
- "Traffic and Safety."
 National Council Publication.
 Vol. 23 No. 4 April 1973.
- Campbell K.J.
 "Preliminary findings on the fleet accident evaluation of federal safety standard 121."
 The HSRI Research Review Vol. 8. No. 2. October 1977.
- Bell, J.L.
 "Seat Belts and anchoraged ~ ADR."
 DOT.
- * Heavy Vehicle Safety."
 Report from the House of Representatives standing committee on Road Safety, April 1977.

- * Joubert, P.N.
 "Review of Truck and bus design in relation to Road Safety."
 DOT February 1973.
- * Miles J.C., Wardill G.A.
 "The analysis and design of a five engine safety cab using finite element methods."
 Institute of Mech. Eng. 1977.
- * Keeman D., Miles J.
 "Application of finite element method to the door instrusion and roof crush analysis of a passenger car."
 3rd International conference of Vehicle Structural Mechanics, October 1979.
- * Tidbury G.H., Keeman D.
 "Investigation into the behaviour of hinges produced by bending collapse of vehicle structural components."
- * Economic Commission for Europe (ECE).
 Group of Rapporteurs on Safety provisions on motor coaches and buses (GRSA).
 Draft. Regulation "Constructional Requirements for PSV's."
- Miles J.C.
 "The determination of collapse loads and energy absorbing properties of thin walled beam structures using matrix methods of analysis." Institute Journal of Mech. Sci. Vol. 18. 1976.
- Keeman D.
 "Bending collapse of rectangular sectioned tubes in relation to the Bus Roll Over Problem."
 1980.
- Sedeghi M.M., Tidbury G.H.
 "Bus Roll Over Accident Simulation."
 3rd International Conference on Vehicle Structural Mechanics, October 1979.

- "Aircraft seats and berths specifications."
 National Aircraft Standards Committee.
- * ADR No. 3 and 3A. "Seat anchorages for Motor Vehicles."
- * Standard No. 207.
 "Seating Systems."
 Federal Motor Vehicle Safety Standard. (FMVSS). US DOT Effective. January 1972.
- "Motor Vehicle seating systems SAE J879b."
 SAE recommended practice.
- * Economic Commission for Europe (ECE).
 Draft regulagion.
 "Uniform provision concerning the approval of public transport vehicles with regard to the strength of seats and their anchorages."
 23rd session.
- "Omibus 'star rating' charter classification." Transport Regulation Board. August 1980.
- * Motor Carrier Safety." Excerpts from the title 13 California Administrative Code.
- "Regulations governing Bus regulations in California."
 California Highway Patrol.
- * Economic Commission for Europe (ECE). Agreement concerning the adoption of uniform conditions of approval and reciprocal recognition of approval for motor vehicle equipment and parts. Addendum 16: Regulations No. 17 to be annexed to the agreement. Revision 1. Uniform provisions concerning the approval of vehicles with regard to the strength of the seats and of their anchorages.
- * "U.S. Passenger seating and crash protection Notice of proposed Rule Making (NPRM). Docket 73.3: Notice 1.
- "School bus passenger seating and crash protection."
 (NHTSA). Docket No. 73.3 Notice 4.
 Federal Register Vol. 40 No. 196, October 1975.

- School bus passenger crash protection" (NHTSA). Docket No. 73.3
 Notice 2.
 Federal Register Vol. 39. No. 147, July 1974.
- "School bus passenger seating and crash protection."
 MVSS 222. Federal Register Vol. 41 No. 19. January 1976.
- "Comments of General Motors Corporation with respect to Notice of Proposed Rule Making - Bus seat and Crash Protection Docket No. 73.3 Notice No. 1."
 May 1973.
- McLean A.J., Brewar N.D. Sandow B.L.
 "Adelaide in-depth accident study 1975 1976."
 Parts 2, 3, 4 and 5.
 DOT4. ARRB.
- Davis M.M.
 "A study of some school bus crashes."
 Proceedings of 21st conference of the AAAM.
- * Turnbow J.W., Haely J.L.
 "A review of crashworthy seat design principles." Air transport and space meeting.
 New York April 1964.
- Saunders D., McCullough K.
 "Problems in Passenger Seat Design."
 Flight International 21 January 1965.
- * Franchini E.
 "The crash survival space."
 Chief Structural dept. Auto Avio Research Laboratory FIAT.