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TRAFFIC ACCIDENT RESEARCH UNIT

DEPARTMENT OF MOTOR TRANSPORT N.S.W.

DYNAMIC TESTS FOR SEAT BELTS

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The Traffic Accident Research Unit was established within the Department of Motor Transport, New South Wales, in May 1969 to provide a scientific approach to the traffic accident problem.

This paper is one of a number which report the results of research work undertaken by the Unit's team of medical, statistical, engineering and other scientists and is published for the information of all those interested in the prevention of traffic accidents and the amelioration of their effects.

D.R. Coleman

Commissioner.

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DEPARTMENT OF MOTOR TRANSPORT,
NEW SOUTH WALES.

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CONTENTS

PREFACE

A dynamic test for seat-belt assemblies is soon to be a mandatory requirement for new cars sold in Australia.

As an aid to the development of suitable test requirements, a meeting was held at the Department of Motor Transport in November, 1972 in order to bring together those organisations and individuals most closely concerned with them.

This report comprises papers prepared by staff of the Traffic Accident Research Unit for this meeting. As well as describing some of the Unit's own activities in this area, attention is drawn to the work of other research groups and the activities of the various bodies concerned in rulemaking around the world.

The report is now being published for general information and wider circulation, and in the interests of uniformity and reproducibility of test procedures.

1. AN INTRODUCTION TO DYNAMIC TESTS ON SEAT BELTS

Historically the seat belt has been mainly a "hung-on" accessory to a motor vehicle and it is only recently that the necessity to provide seat belts has influenced styling and engineering of many cars. It is not surprising then that the test requirements for seat belts still ignore almost completely the influence of vehicle dynamics on seat belt performance.

Australian Design Rules and Standards currently require that the layout geometry of the anchorages in real cars shall be used on the test rig for testing belts but other vehicle factors are rarely considered. All belts regardless of their application are required to have the same strength of 4000 pounds, this being based on the deceleration at 20g of a 200 pound wearer (or at 25g of an average 160 pound male). In the case of combination belts, such as the lap/sash type commonly installed in Australia, the static method of test applies the 4000 pound proof load in such a manner as to stretch the lap and sash parts equally; thus, if their lengths happen to be similar, the load is distributed equally to lap and sash sections.

This equal division of load between lap and sash is fully justified from work[1] at Holloman Air Force Base on live adult male subjects provided that no load is transmitted from the legs to the vehicle, and provided that the seat is incapable of transmitting large loads from the buttocks to the floor pan. In real cars some load is often transmitted through the legs, the driver's chest often transmits load to the steering wheel (although in new cars this load is now limited by an Australian Design Rule) and large loads are transmitted from the buttocks to the car seat.

The proportion of the seat belt load that could be transmitted to the seat appears from laboratory studies to be very large. However the present Design Rule No. 3 requires merely that the seat-to-floor anchorage shall be capable of withstanding its own inertial force at 20g and the (rearward) headrestraint load. It must be expected under such circumstances that where a seat belt is worn in a severe head-on collision, the seat-to-floor mounting would break away, even where the floor pan deceleration did not exceed 20g. Thus, the full load restraining the wearer would be transmitted to the seat belt and, following failure of the seat anchorage, the belt would also be loaded by the seat. Except where vehicles are designed with seats secured in a more adequate fashion, it is evident that seat belts should be designed to carry at least the full load required to restrain the wearer. (This disregards the additional load from a break-away seat). To be consistent with the 20g seat anchorage requirement, it would seem desirable to require belts to restrain the wearer at 20g floor pan deceleration; this is not exactly the same as specifying 20g deceleration of the wearer. Where a seat can withstand seat belt loads, the seat cushion is depressed progressively as the wearer is thrown forwards with the result that he slides down a slope of about 10 degrees. A car driver may be decelerated in a head-on crash by the steering wheel, as well as by his belt but this is not the case of tightly strapped passengers. Some load is often transmitted through the feet or knee caps but there is not at present a rational basis for apportioning some of the restraining force to feet or knees.

The foregoing remarks give strong support to the argument that seat belt dynamic tests should preferably be performed in the cars to which they are to be fitted without any unnatural reinforcement of the seat-to-floor anchorages. In the not too distant future this should be in the interests of both manufacturers and occupants. In the meantime, and in the absence of hard data about the performance of vehicles and seats, it is reasonable for Australia to adopt the present overseas practice of performing dynamic tests on rigid seats of specified geometry; this also has the practical advantage of ensuring long life for the test rig. However, it still leaves open for discussion the angle of inclination of the seat. Most national standards and both international specifications (E.C.E. and I.S.O.)† specify an upward sloping seat at 10 degrees although S.A.E.* specifies a horizontal seat. Upward sloping seats transmit frictional forces to dummies and these may be large if the protrusions under the pelvis (the ischial tuberosities) dig into the seat under load, which does occur with Sierra 1050 dummies. If the objective in testing is to eliminate the influence of friction between seat and dummy then a downward sloping seat would be preferable and a choice of -10 degrees would also satisfy the geometric requirement up to the point of failure of the seat-to-floor anchorage.

The two greatest problems for E.C.E. and I.S.O. in prescribing a dynamic test have been the choice of dummy and the choice of sled deceleration pulse. Most national standardizing bodies originally made fairly arbitrary selections of both dummy and pulse; the need for international agreement, particularly in the European Common Market (E.E.C., not to be confused with E.C.E.) has highlighted the need for a more rational approach. Essentially the problem has been one of selecting one of two controlling parameters, and specifying their values; all other parameters could then be left to individual countries to specify. The I.S.O. in 1968 decided to approach the problem from the strength level required of belts that is, by starting with the concept of decelerating the average man at 25g. To this was added the requirement, derived from live subject work[1], that the sash and lap sections of a lap/sash belt should be equally loaded by a valid dummy. The S.A.E. Committee was so impressed with the

† E.C.E. is the United Nations Organisation's Economic Commission for Europe (Land Transport Commission, Working Party 29). I.S.O. is the International Standards Organisation (Motor Vehicles, Sub-Committee 12).

* S.A.E. is the (U.S.A.) Society of Automotive Engineers.

simplicity of this concept that they adopted it in principle at their next meeting; S.A.E. J117 as it stands today, requires that the sled and dummy shall be adjusted to produce equal loads in lap and sash and, in the tested belts, a minimum belt tension of 6000 pounds. S.A.E. leave open the means of achieving these loads, specifying the equipment only in the broadest terms.

The I.S.O. recommended method of test, as approved by a majority of members, also specifies that the load calibration technique shall be used. The British Standards Institution (B.S.I.), which had been one of the first national bodies to introduce dynamic testing, changed in 1971 from the arbitrary specification of pulse and dummy to the I.S.O. calibration procedure with a 20kN (4496 pound) calibration load but with a nominated design of dummy that reportedly provides equal loading of sash and lap calibrating straps.

The E.C.E., proceeding along a different path, adopted what appears to have been an early French proposal to use a polyurethane tubular programmer for pulse generation and proceeded to lay down rigid limits for the resulting pulse. The polyurethane programmer was later adopted by Holland. (It is understood that temperature control of the polyurethane is necessary for reasonable reproducibility of test results and that programmers have to be rested several days after each use.) The E.C.E. also specified that the sled velocity change must be 50km/h (31 mph), the stopping distance 400 mm, the pulse duration 50 to 70 ms and sled mass 400 kg. The dummy dimensions, mass distribution and degree of articulation were also specified, and its compliance (that is, its energy-absorbing and momentum-changing properties) was specified in some, but not all respects.

So far as is known the French test method as published by E.C.E. has not been endorsed by any country. Holland is using the specified programmer and sled but with a very simple dummy that clearly does not comply with the E.C.E. specifications but which has been shown by the Dutch authorities to comply with the S.A.E. lap/sash load distribution requirements originated at the I.S.O. Meeting in 1968. It appears from the laboratory data made available to the author[2] by Bastiaanse, that the E.C.E. pulse with the Dutch dummy produces total belt tensions about 20% higher than permitted by B.S.I. but which are acceptable to S.A.E.

The question of belt strength is an important one. It is known that wearers have survived crashes in which the belts have been broken by overload caused by separation of cars into two pieces. It might be concluded - since the limit of human tolerance would appear from this evidence to be higher than the loads generated in current belts - that the belt strengths should be increased in order to raise them to the human limit. On the other hand it might be argued - since when belts break by overload of the webbing, the webbing stretches many inches at a constant or falling load, so dissipating much of the wearer's kinetic energy - that any increase in strength is likely to reduce the proportion of energy absorbed by the belt, and increase sharply the loads on the wearer even in crashes that were previously survivable with weaker belts.

The latter view is supported by an isolated case of rib cage collapse in a N.S.W. serious collision and by Bastiaanse and Aldman in Europe who argue for the introduction of load limiting devices in sash straps. Suitable load limiters (energy absorbers) have now been developed by Sarrailhe[3].

One item in the E.C.E. specification deserves special consideration because it is absent from the other specifications that have been discussed above, namely the requirement for a particular sled velocity, specified to be 50 km/h in a stopping distance of 400 mm. Terminal velocity is not specified.

During the last three months an attempt has been made in Australia to obtain agreement as to a rational dynamic test based preferably on the requirements of European Governments. The E.C.E. specification was used as a basis because it purports to be the agreed test method in Europe but the present position is that the Swedish and the new British requirements do not remotely resemble it in spite of the fact that both countries are represented on the expert group advising E.C.E. on dynamic tests; moreover Holland, which has performed valuable research for E.C.E. has departed from the E.C.E. dummy specification and the French national laboratories are reported to be now using a calibation procedure for their sled, following difficulties experienced with reproducibility.

Dynamic tests have been performed by the authors using the E.C.E. method and the various anthropometric research dummies available in Australia namely the Sierra 850, Sierra 1050, Alderson F.50.AU and Ogle-MIRA dummies. All these dummies produced high loads with the E.C.E. pulse in spite of the fact that Armstrong[1] found that with his pulse the Sierra 850 dummy produced loads only 25% greater than those produced by human live subjects.

Even if the problem of defining a suitable dummy has now been resolved by Bastiaanse[2] in the Dutch national laboratories in Delft, the problem of specifying a suitable sled pulse remains. Because of the proposed introduction in 1973 of the 30 mph barrier test for evaluation of steering column intrusion, it is natural that Australians should be expecting other safety requirements to be upgraded to the 30 mph level. If seat belts were being evaluated by means of the barrier test, then the problem of defining a suitable sled pulse would be merely one for the vehicle manufacturer. As it is, there is no such intention in the immediate future and in any case, the fact must be faced that real cars range widely in the floor-pan deceleration pulses they produce in barrier crashes. Until recently it has been commonly accepted that a half-sine (deceleration-time) pulse of 20g amplitude and 100 to 150 ms duration was a reasonable representation of medium and large cars, and this is reflected in the B.S.I. test provisions. The current trend however is to design front structures with optimum crush performance and this ranges from triangular and square to trapezoidal pulses according to the designer's notion of what is safest. A complication for the safety scientist in this context arises from the fact that car deceleration levels may be higher in minor crashes in which the crush structure behaves elastically, than in severe crashes which produce the designed plastic collapse. Moreover the introduction into American cars of bumper bar shock absorbers (for protection of the vehicle) may have an adverse effect on the occupants.

Under these circumstances the selection of any particular pulse would at present be rather arbitrary and it is at least questionable whether any selection is justifiable. The alternative of specifying the seat belt loads as in British and American specifications remains for consideration.

In his technical specification for the Dutch TNO 10 dummy, Bastiaanse quotes[2] the results of a series of calibration tests of the dummy using two types of webbing and both lap and lap/sash seat belts. The present discussion will be confined to lap/sash belts. Figure 1 illustrates the sled pulses in four of the runs, with the E.C.E. pulse limits overlaid. Although errors have possibly been introduced by the authors in reproduction, these can be shown by comparison with the peak values reported in the text, to be small for deceleration values. Consequently, it must be concluded that in those particular runs the TNO sled pulse did not always lie within the cross-hatched area designated by E.C.E.; moreover, the pulse shape is somewhat variable.

It is not clear whether Bastiaanse tried to meet the E.C.E. pulse limits but he did state that one of his objectives was to meet the S.A.E. loading requirement. This requires a total belt tension during testing of at least 6000 pounds (26.7 kN). His data for the four runs illustrated in Figure 1 are reproduced in Table 1 and show belt tensions totalling 29.6, 30.6, 30.6 and 27.4 kN respectively; these results comply with S.A.E. J117 in exceeding 26.7 kN moreover, the lap-to-sash load distributions comply with the S.A.E. 50:50% requirement (permitted tolerance ±10%).

In attempting to produce the E.C.E. pulse shape, the Traffic Accident Research Unit had a choice of two programmers, one giving a square pulse, the other a half-sine. The square pulse was the first choice but it was soon realised that the E.C.E. envelope (shown in Figure 2) by cutting off the top left corner, effectively eliminated square pulses; attention was therefore turned to the Unit's sine wave programmer. Figure 3 illustrates seven successive pulses overlaid from the same data origin. The velocities ranged from 50.6 to 52.4 km/h; these values were obtained from the approach and departure slopes of the sled displacement-time curves, and are somewhat higher than the indications of other instruments. Accurate measurement of velocity is receiving considerable attention from the authors in an endeavour to remove doubts. Velocity change is in any case not easily defined for rebound sleds which characteristically have high velocities both before and after impact, and imprecisely defined instants of entry and exit.

As a possible alternative to the E.C.E. specification, the new B.S.I. provisions in BS AU160:1971 may be considered. These together with other specification requirements, are set out in Appendix 1. It may be seen that the pulse limits consist of duration 80 to 110 ms, 30g maximum and jerk 2000g/s maximum. The calibration procedure of I.S.O. is specified for determining sled velocity using the B.S.I. dummy, the calibration load being 20 $\pm$ 1kN (compared with 26.7 kN minimum test load in S.A.E. J117). The tolerance of $\pm$ 1kN appears to the authors to be somewhat small, although the Unit has insufficient evidence so far to comment with authority on the variability of its own sled. Figure 4 illustrates the Unit's pulses similar to B.S.I.

To conclude, adoption of the E.C.E. pulse would bring belts up to the level of the S.A.E. test but would appear to necessitate about a 20% increase in the strength of Australian seat belts and would present some difficulties in specifying equipment that gave results equivalent to the E.C.E. specified test rig. The B.S.I. standard necessitates no strength increase and the specified calibration procedure (with the addition of the lap/sash load equality requirement) allows straightforward comparison between sleds which is an important matter in the case of dispute and essential in a Standard test method. Detailed questions of dummy design and pulse effects are left for other chapters but alternative test methods, based on E.C.E. and B.S.I. requirements are given in Appendix 2.

REFERENCES

1. Armstrong, R. W. and Waters, H. P., Testing programs and research on restraint systems, S.A.E. Report 690247.
2. Bastiaanse, J. C., Description of the adult dummy TNO 10, T.N.O. Delft, 28th June 1972.
3. Sarrailhe, S. R. , Dynamic tests of a yielding seat belt system, Department of Supply, Australia. Aeronautical Research Laboratories Structures and Materials Report 340, 1972.

APPENDIX I

Comparison of Dynamic Tests

Organisation Parameter	ECE	ISO	BSI	SAE
Sled velocity change *	50 ± 1km/h	not specified	not specified	not specified
Sled mass	400 ± 20kg	not specified	"Framework" 380 to 1000kg	not specified
Stopping distance *	400 ± 20 mm	not specified	not specified	not specified
Sled deceleration	26 to 34g max. (envelope provided)	30g max. (50g max. transients)	30g max. (50g max. transients)	30g max. (50g max transients)
Sled jerk	1300 to 2700g/s	2000g/s max. (transients up to 5 ms)	2000g/s max. (transients up to 5 ms)	2000g/s max. (transients up to 5 ms)
Sled pulse duration *	50 to 70 ms	50 to 150 ms	80 to 110 ms	50 to 150 ms
Calibration lap and sash loads	calibration load method not specified	17.64kN ± 0kN total (4000 lbf)	20kN ± 1kN total (4496 lbf)	6000 lbf min. total 50:50 ± 10%
Dummy mass	Dimensions etc. all specified	Complete design specified	75kg (165 lb) and 610 mm base to shoulder	125 to 175 pounds
Seat angle	+ 10 degrees	+ 10 degrees	+ 10 degrees	0 degrees
Footrest	40/45 deg.	None	40/45 deg.	None
Calibration webbing	not required	8 ± 1% at 4kN 17±2% at 11kN	8 ± 1% at 4kN 17±2% at 11kN	17 ± 3% at ?
Slack	25 mm board	25 mm board	25 mm board	see procedure
Calibration geometry	not required	Figure provided	Figure provided	As test
Test geometry	Figure or as required	Figure provided	Figure provided	Appropriate
Measurements (displacement etc.)	Chest 200/300 mm	Chest 200/300 mm	Pelvis 200 mm max. and Chest 300 mm	Record pulse, load, displacement

* Instants of exit and entry not defined. Convention in U.S.A. Military Specifications is to record duration between points 10% above base line.

APPENDIX 2

Draft Australian Specification for Dynamic Testing of seat belts (with retractors where appropriate)

1. REQUIREMENT

On completion of the test described in Clause 2, no total fracture of any component shall have occurred, no load bearing component shall have become separated from its mating part and the securing buckle shall be capable of release by an adult observer without mechanical assistance; the slip of webbing in the loaded direction through any adjuster (but not including the withdrawal of webbing from any retractor) shall not exceed 25 mm; the sum of all such slip for one assembly shall not exceed 50 mm. (Requirements limiting withdrawal from any retractor at any time during the test, are to be added later, with requirements for displacement of the dummy).

2. TEST METHOD - PROPOSAL A

2.1 Test equipment

The equipment shall consist of a dummy and a test frame, and means for producing acceleration and deceleration of the test frame and dummy.

(a) Dummy. The dummy shall have a seated height from seat to shoulder of 550 to 650 mm and a mass of 57 to 80 kg and shall be such as to comply with the calibration requirement of clause 2.2.

(b) Test frame. The test frame shall comply with Figure 1* and its effective mass shall be 380 to 1000kg. It shall include a rigid framework on which is mounted a fixed rigid seat for the test dummy. The framework shall be fitted with anchorage points suitable for the belt

* Figures not reproduced

under test. The structure carrying the anchorages must be rigid. The anchorages must not be displaced by more than 0.2 mm in the horizontal loading direction under a load of 980 N. (Editorial Note: This is taken from B.S.I. who will be asked to clarify). The test frame shall be designed and constructed so as to perform as a monolitic structure under decelerations employed in testing or calibration. Means shall be provided for accelerating the test frame and belted dummy to the required velocity and of applying the required deceleration. The test frame shall be contrained to move only along the nominal loading axis.

(c) Acceleration and deceleration. The pulse to produce belt loading shall be such as to decelerate (or accelerate backwards) the test frame relative to the dummy by not more than 30g for a period of 80 to 110 ms measured at the 10% level, except that transients lasting less than 3 ms above the 30g level may be tolerated provided none exceed 50g. The rate of change of deceleration during increase of belt loading shall be not more than 2000g/s when averaged over any 5 ms period.

(d) Instrumentation. Acceleration and load measurements shall be made with equipment having flat frequency response from 0 to 150 Hz and linear over the ranges of measurement. The degrees of flatness and linearity required are those needed to satisfy the performance requirements during system calibration.

2.2 <u>Calibration procedure</u>

Calibration shall be performed using a new pair of calibration seat belts, one lap and one sash constructed as shown in Figure 2*. The webbing used for calibration shall have the following characteristics:-

* Figures not reproduced

(a) Material. Polyamide or polyester continuous filament yarn.

(b) Elongation. 8±1% at 4kN and 17±2% at 11kN when tested by the method described in Australian Standard E47.

The belts shall be adjusted to the lengths specified in Figure 2 and attached to the anchorages shown thereon, with the dummy in place. The two belts shall be pulled tight to tension of about 40N (10 pounds), so that the dummy's back is in contact with the seat back and thighs in contact along its length with the seat base.

The test equipment shall be adjusted so as to produce the following loads measured by webbing tension transducers placed close to each of the four anchorages.

(i) In the lap belt. A total of 10±1kN which should preferably be distributed about equally between the anchorages.

(ii) In the sash belt. A total of 10±1kN which should preferably be distributed about equally between the anchorages.

The calibration procedure shall be carried out at sufficiently frequent intervals in order to check performance of the dummy. This may be performed on any testing equipment complying with this specification.

The calibration procedure shall also be carried out to establish initially the validity of the system and at sufficiently frequent intervals to assure its maintenance thereafter; for these purposes the entire system to be employed in testing shall be calibrated simultaneously.

2.3. Seat belt testing procedure

The seat belt shall be mounted on the test frame in the manner (including geometry) specified by the authority for whom the testing is to be done; this shall not differ significantly from the manner to be employed in the vehicle for which the belt is intended. In the preparation for the test, care shall be taken to lock a cam-action securing buckle only by the force of its springs, if any; it shall not be forced into the closed position. A controlled amount of slackness shall be introduced into the assembly by placing a 25 mm thick board at least 610 mm long and of a width about equal to that of the dummy's torso behind the back of the dummy before fastening it tightly. The board shall be removed and the dummy repositioned so that its back is in contact along its length with the seat back and its thighs in contact along its length with the seat base. When belts are tested with retractors which are automatically tensioning this controlled slackness shall not be introduced; in this case the strap shall be drawn fully from the reel and allowed to run back slowly under spring tension through any pulleys and sheaves. The strap shall be marked on the unloaded side of each buckle or adjuster in the assembly. No load transducers shall be mounted on the webbing.

The testing equipment shall be operated as in system calibration and the belt examined for compliance with requirements.

2. TEST METHOD - PROPOSAL B

2.1 Test equipment

The equipment shall consist of a dummy and a test frame, and means for producing acceleration and deceleration of the test frame and dummy.

(a) Dummy. The dummy shall comply with S.A.E. Recommended Practice J963 - Anthropometric Test Devices for Dynamic Testing, or TNO 10 (the Dutch dummy).

(b) Test frame. The test frame shall comply with Figure 1* and its effective mass shall be 400±20Kg. It shall include a rigid framework on which is mounted a fixed rigid seat for the test dummy. The framework shall be fitted with anchorage points suitable for the belt under test. The structure carrying the anchorages must be rigid. The anchorages must not be displaced by more than 0.2 mm in the horizontal loading direction under a load of 980 N (Editorial Note: This is taken from B.S.I. who will be asked to clarify). The test frame shall be designed and constructed so as to perform as a monolitic structure under decelerations employed in testing or calibration. Means shall be provided for accelerating the test frame and belted dummy to the required velocity and of applying the required deceleration. The test frame shall be constrained to move only along the nominal loading axis.

(c) The sled shall be propelled so that its change of velocity is 50±1km/h over a sled displacement of 400±50 mm. The sled shall remain horizontal throughout deceleration.

* Figures not reproduced

(d) Acceleration and deceleration. The pulse to produce belt loading shall be such as to decelerate (or accelerate backwards) the test frame relative to the dummy by at least 26g for a minimum period of 20 ms and by not more than 35g at any time, except that transients lasting less than 3 ms above the 35g level may be tolerated provided none exceeds 50g. The rate of change of deceleration during increase of belt loading shall be between 1000 and 2700 g/s when averaged over the first 5 ms period and shall not exceed 2700 g/s in any subsequent 5 ms period. The deceleration pulse shall be of duration 50 to 80 ms measured at the 10% level.

(e) Instrumentation. Acceleration and load measurements where required, shall be made with equipment having flat frequency response from 0 to 150 Hz and linear over the ranges of measurement. The degrees of flatness and linearity required are those needed to satisfy the performance requirements.

2.2 Seat belt testing procedure

The seat belt shall be mounted on the test frame in the manner (including geometry) specified by the authority for whom the testing is to be done; this shall not differ significantly from the manner to be employed in the vehicle for which the belt is intended. In the preparation for the test, care shall be taken to lock a cam-action securing buckle only by the force of its springs, if any; it shall not be forced into the closed position. A controlled amount of slackness shall be introduced into the assembly by placing a 25 mm thick board at least 610 mm long and of a width about equal to that of the dummy's torso behind the back of the dummy before fastening it tightly in. The board shall be removed and the dummy repositioned so that its back is in contact along its length with the seat back, and its thighs in contact along its length with the seat base.

When belts are tested with retractors which are automatically tensioning this controlled slackness shall not be introduced. In this case the strap shall be drawn fully from the reel and allowed to run back slowly under spring tension through any pulleys and sheaves. The strap shall be marked on the unloaded side of each buckle or adjuster in the assembly. No load transducers shall be mounted on the webbing.

The belt shall be examined for compliance with requirements.

TNO RUN NO. 442

Velocity 49.1 km/h (30.6 mph)

Stated 29.9g

Load 29.60 kN (6650 lbf)

TNO RUN NO. 443

Velocity 49.0 km/h (30.5 mph)

Stated 30.0g

Load 30.60 kN (6780 lbf)

TNO RUN NO. 444

Velocity 49.0 km/h (30.5 mph)

Stated 28.2g

Load 30.60 kN (6780 lbf)

TNO RUN NO. 446

Velocity 49.1 km/h (30.6 mph)

Stated 27.6g

Load 27.40 kN (6160 lbf)

TABLE 1: Details of TNO Tests

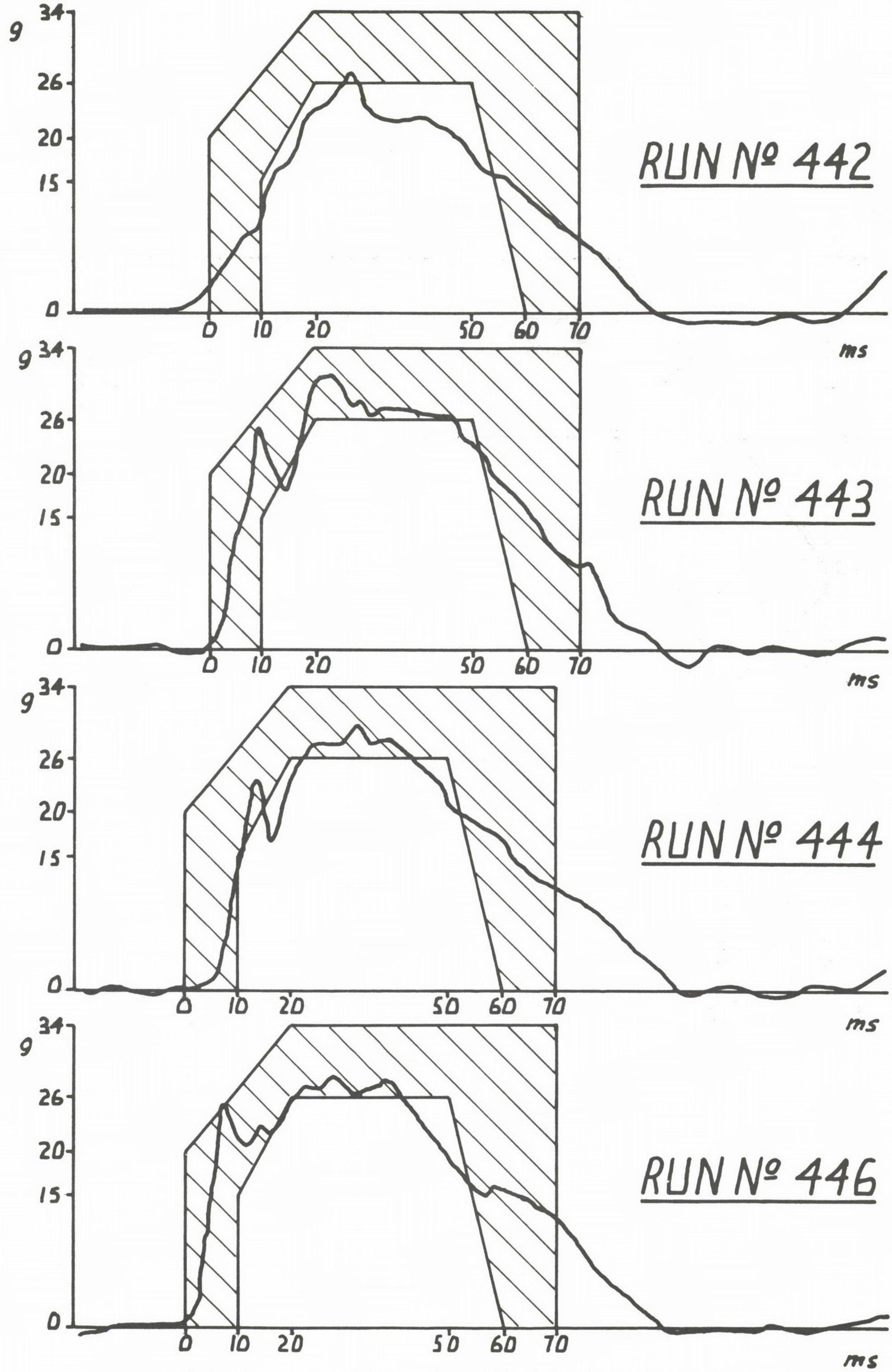
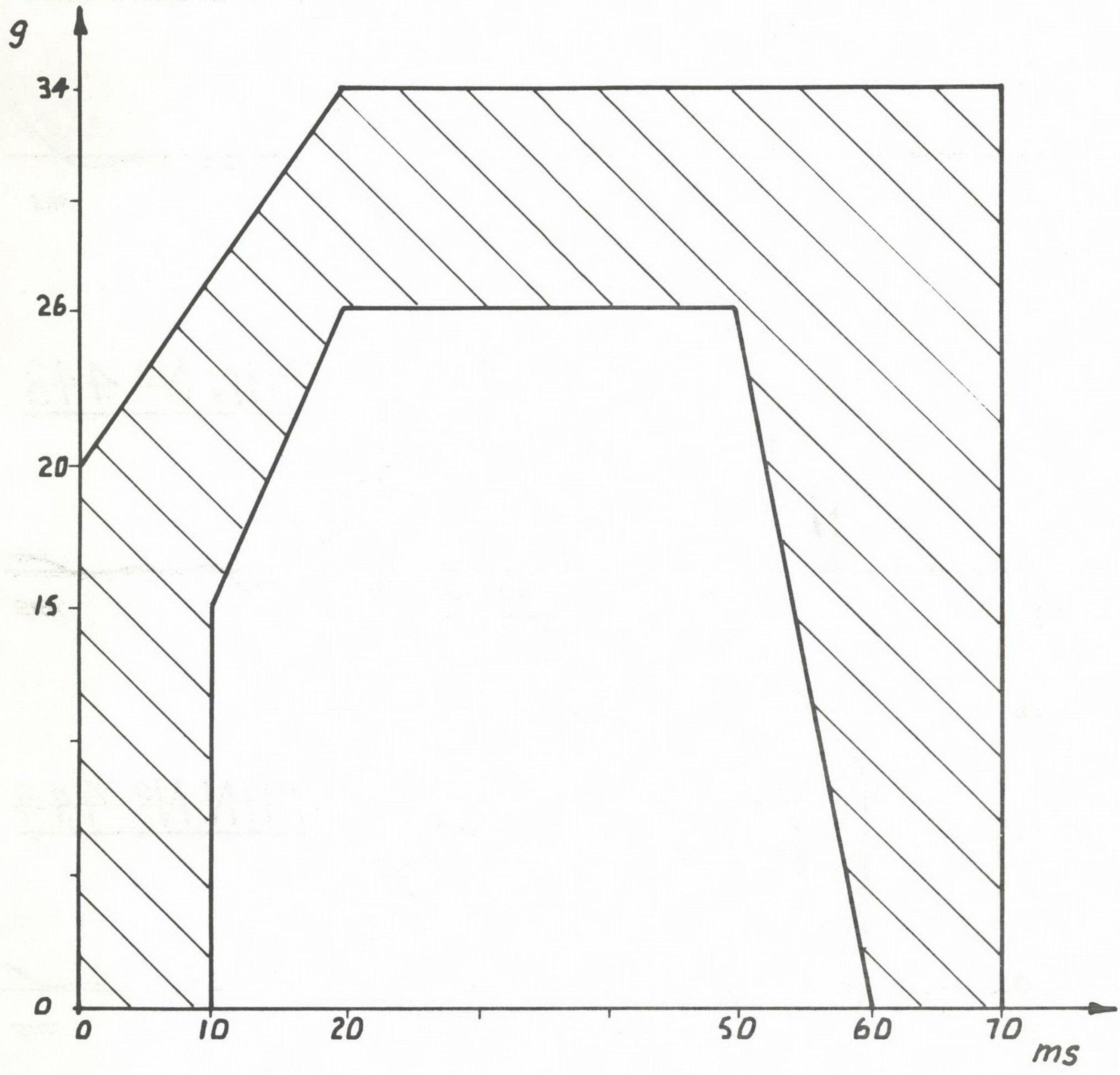


FIG. 1. T.N.O. & E.C.E. PULSES.



E.C.E. PULSE. FIG. 2.

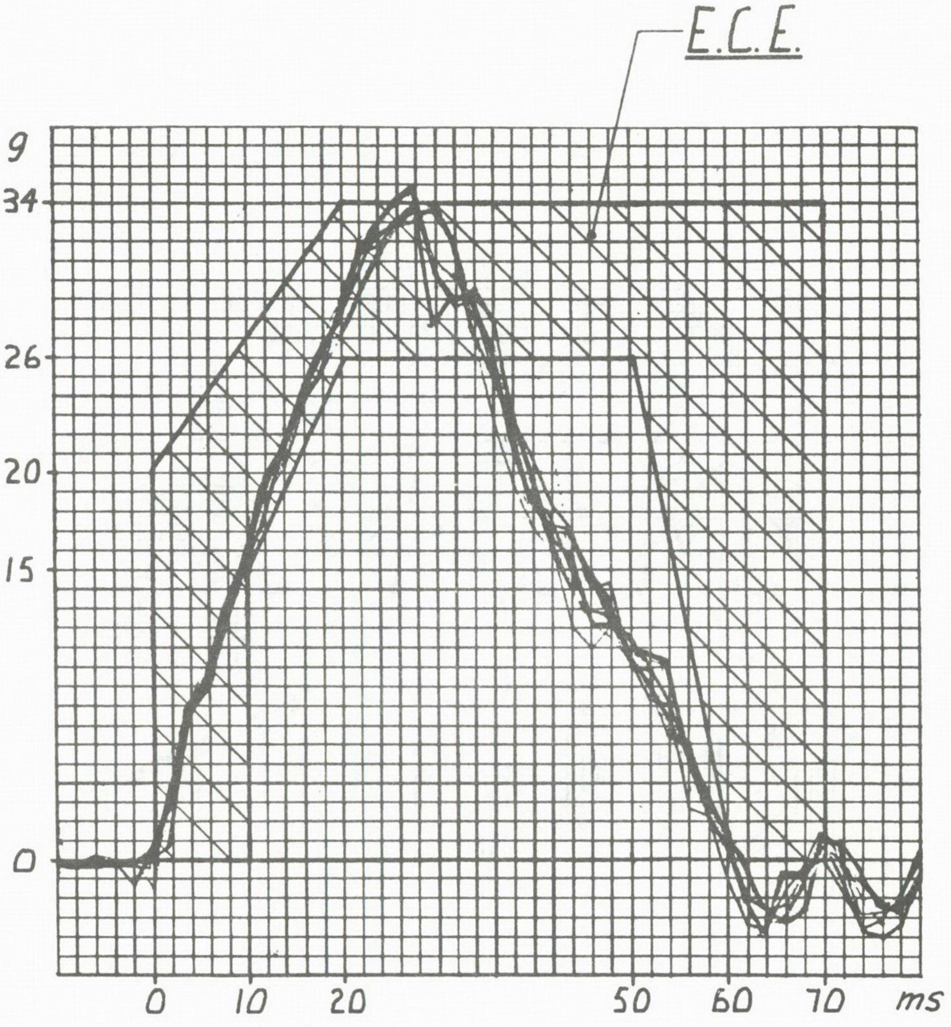


FIG. 3

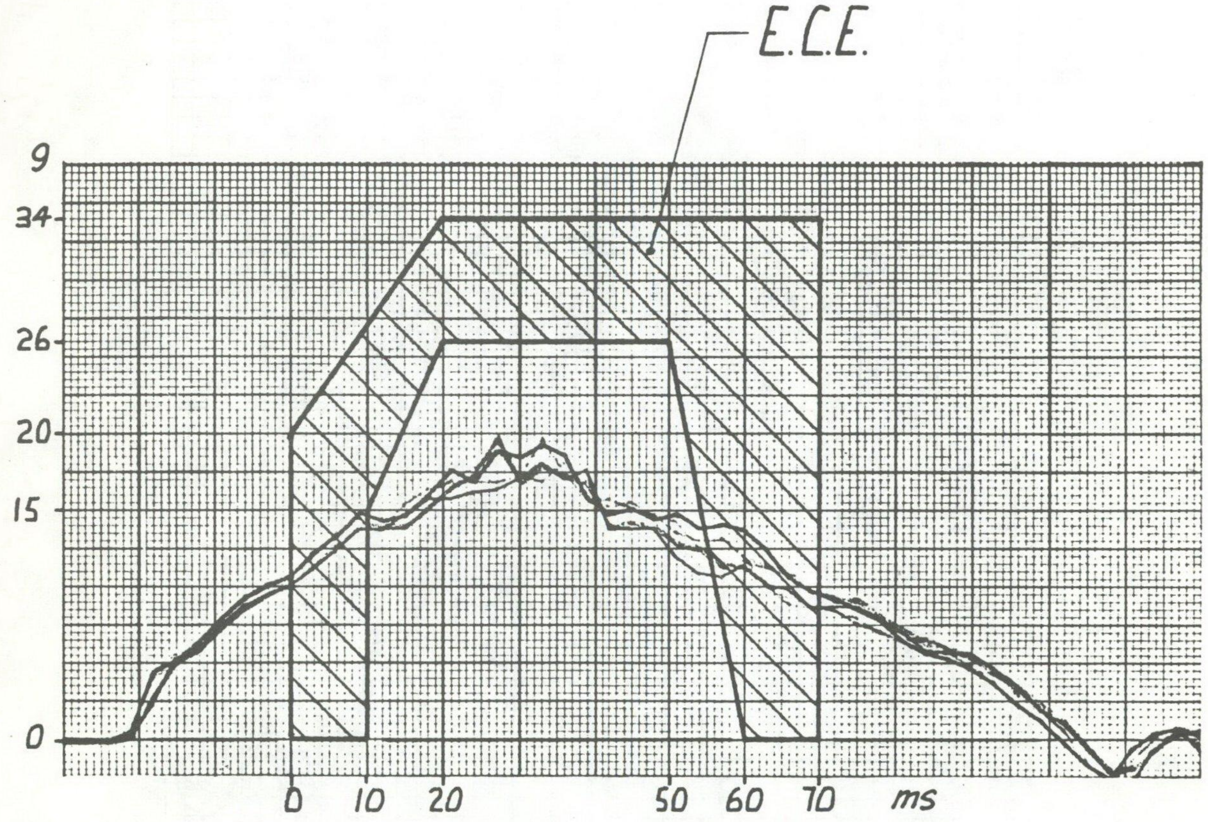


FIG. 4

2. SEAT BELT DYNAMICS

INTRODUCTION

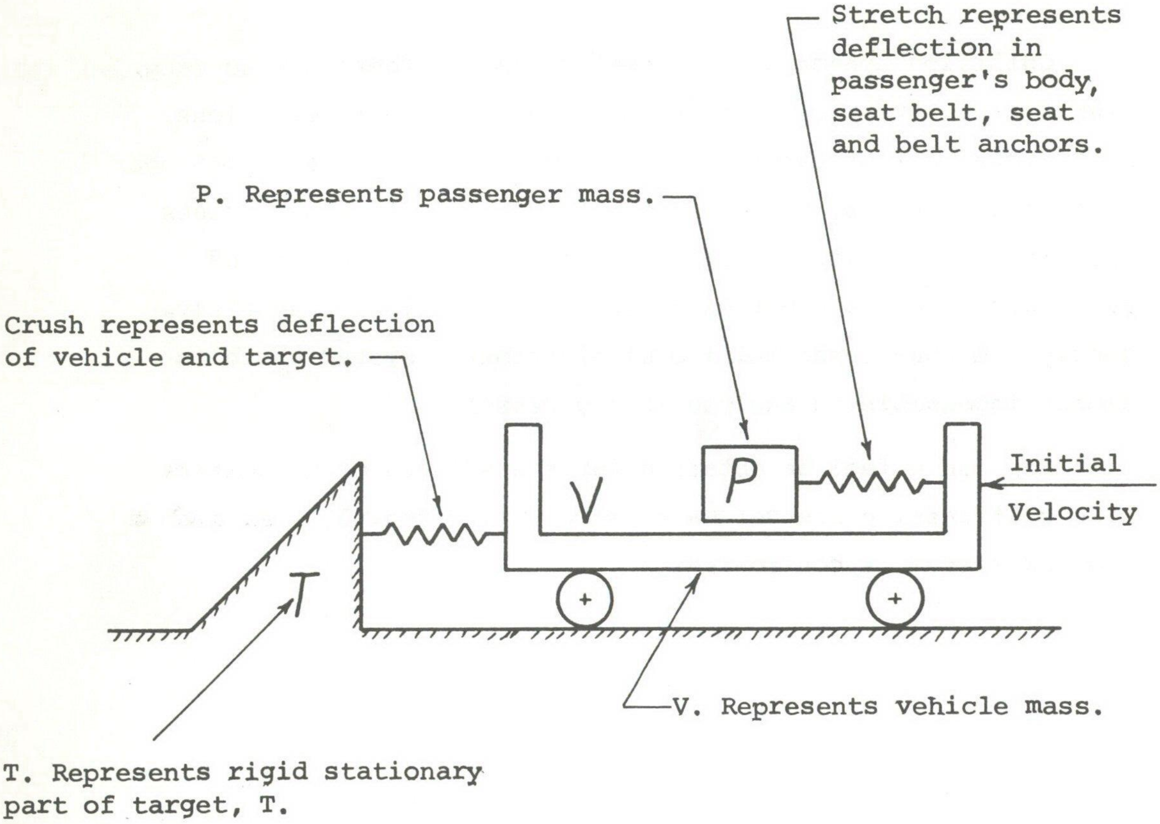
Collision dynamics is a complex topic. There are so many significant variables. And so many complicated interactions. Restraints pull at varying angles, springs bottom, the passenger yaws, sinks, rolls, bends and flails. The car pitches, rises up. Nothing is rigid. Different parts of the vehicle and passenger decelerate out of phase, quickly, slowly, severely, gently. A non-linear multimodel distributed system which is almost impossible to analyse in any detail.

How can a feel be obtained for the effects of parameters like stiffness, passenger mass, seat spring travel, when such a complex system is confronted?

MODELS

Several authors[1,2] have reported insight into collision dynamics which they have derived from mathematical models of the collision process. This chapter, being intended to indicate the simplest cause-effect relationships, has made use of a very simple model of an automobile collision.

The model presented here is simple to the point of being unrealistic. Yet, by covering the four main components roughly, it allows some feel for the interactions between the components in the system. The model is described in Figure 1.



SIMPLE COLLISION MODEL

Figure 1

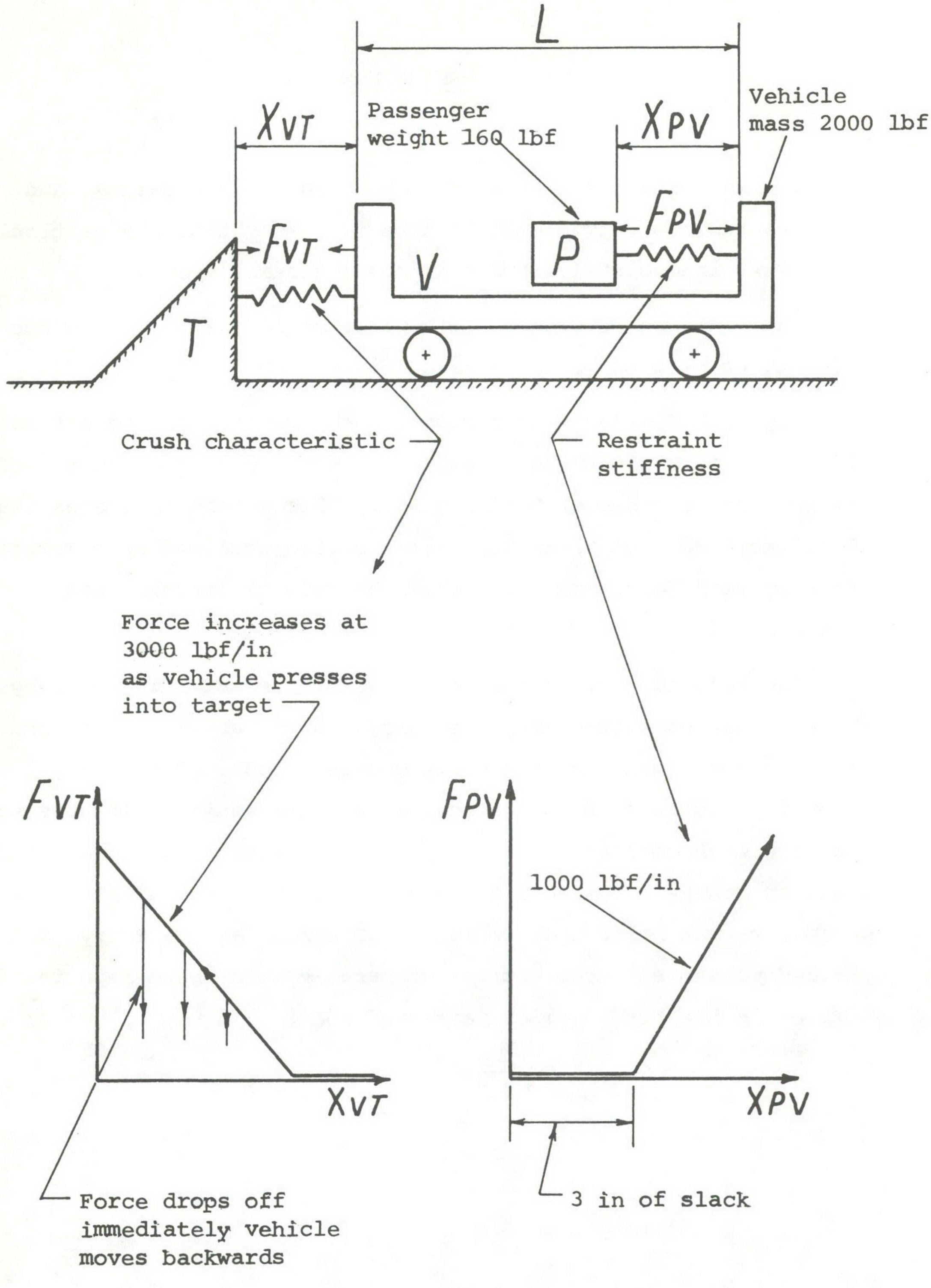
PARAMETERS

The masses and stiffnesses of passengers, restraints, and vehicles range widely enough to make it likely that any particular model will inadequately represent some existing systems.

The wide ranges also justify gross approximations when the model is to be a basis for general discussion.

Figure 2 displays the parameter selections. The selection of vehicle mass and crush parameters was based upon a report[3] of instrumented crashes of Ford Anglias. That report indicates that floor deceleration (behind the front seat) increased approximately linearly with deformation and that the rate of increase was approximately independent of impact velocity.

The selection of restraint parameters is made difficult by the articulated nature of passengers, and the absence of records of their performance during heavy decelerations. There are, however, good reasons for expecting the restraining stiffness to increase with deflection and the representation with a slack and a linear zone was intended to cover this. The guesstimation of parameters was based upon webbing stiffnesses and geometry, and yielded passenger decelerations comparable with those reported for dummies in the Ford Anglias mentioned above.



PARAMETERS

Figure 2

COMPUTATION

The relationships between the variables in the model are as follows

Deceleration = Force ÷ mass (by Newton)

i.e. $\frac{-dU}{dt} = F \div \frac{W}{g}$

i.e. $\lim_{\delta t \to 0} (U_{t+\delta t}) = U_t - \frac{g}{W} F_t \; \delta t$

For passenger:-

$$\lim_{\delta t \to 0} (U_{P_{t+\delta t}}) = U_{P_t} - \frac{g}{W_P} F_{PV_t} \; \delta t \; . \; . \; . \; . \; . \; . \; 1$$

where F_{PV} is the function of X_{PV} defined in Figure 2

For vehicle:-

$$\lim_{\delta t \to 0} (U_{V_{t+\delta t}}) = U_{V_t} + \frac{g}{W_V} (F_{PV_t} - F_{VT_t}) \; \delta t \; . \; . \; 2$$

where F_{PV} and F_{VT} are the functions of X_{PV} and X_{VT} defined in Figure 2

Since, by definition, $U = \frac{dX}{dt}$

For crush:-

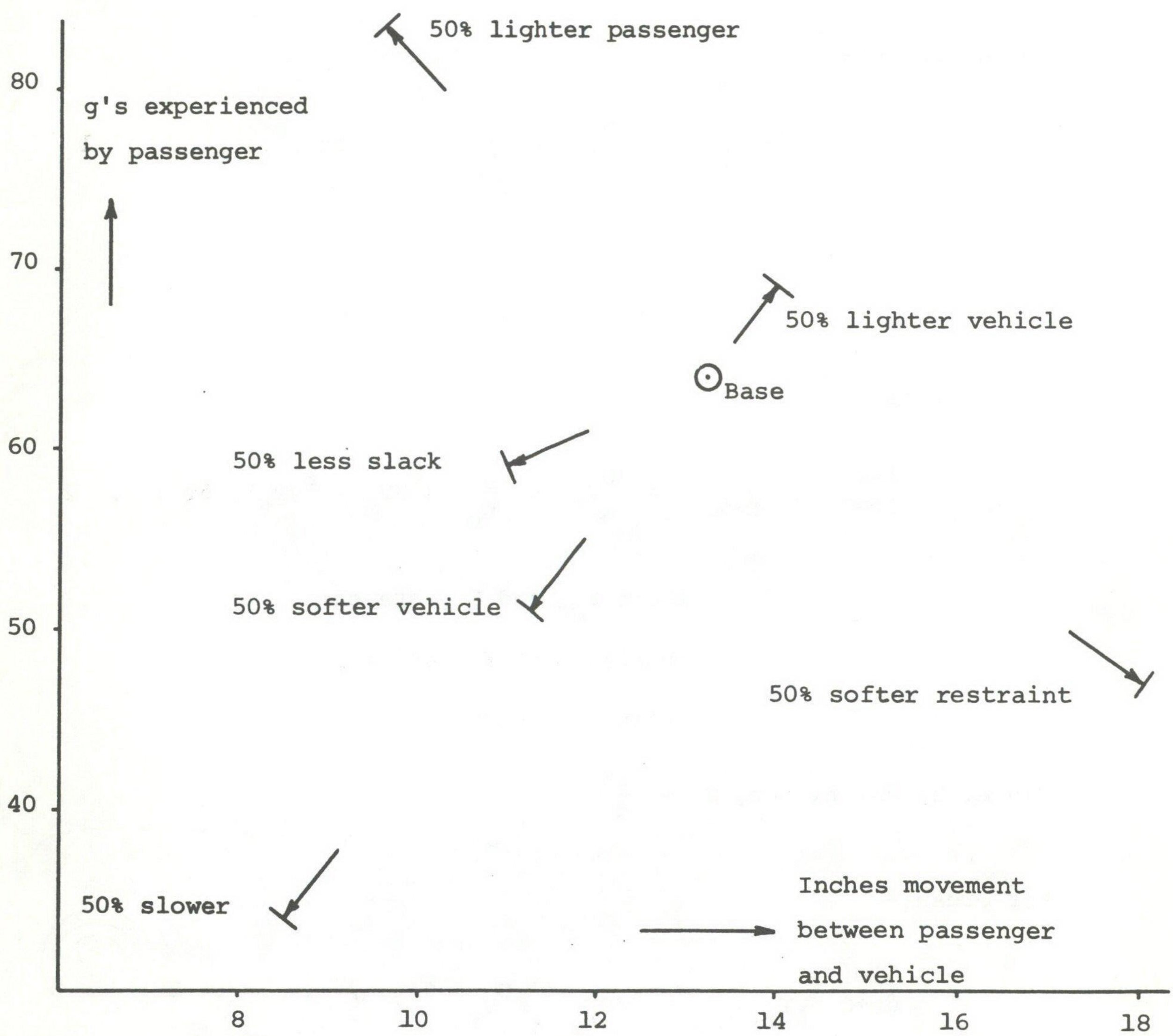
$$\lim_{\delta t \to 0} X_{VT_{t+\delta t}} = X_{VT_t} + U_{V_t} \; \delta t \; . \; . \; . \; . \; . \; 3$$

For restraint:-

$$\lim_{\delta t \to 0} X_{PV_{t+\delta t}} = X_{PV_t} + (U_{P_t} - U_{V_t})\,\delta t \quad . \; . \; . \; . \; 4$$

Given initial velocities and positions, equations 1 to 4 enable calculation of velocities and positions after a small time interval, δt. And then after a further δt. And so on.

A digital computer was programmed to perform these iterative calculations and hence indicate the sequence of velocities and deflections implied by the model.



EFFECTS OF PARAMETERS

Figure 3

THE VEHICLE AND TARGET

The well known advantages to passengers in vehicles which collide more slowly, are heavier, or crush more softly are indicated by Figure 3. A trade-off which is not shown is the crush distance which increases from 22.5 in to 32 in in the softer vehicle, and from 16 in to 22.5 in in the heavier vehicle.

Provided that they would deflect comparable distances when comparably loaded, "soft" targets would also contribute a reduction to the passenger decelerations and relative movements.

When the target has a finite mass, and can be moved at a significant velocity by the impacting vehicle (e.g. another car), then the change of velocity of the impacting vehicle may be reduced, reducing the passenger loads and movements accordingly.

THE RESTRAINT

Figure 3 demonstrates the principle that the passenger must be decelerated as soon and as evenly as possible during the time available if his peak loading and movement are to be kept small.

This ideal is quite the opposite to most real systems in which the peak load is reached after many inches of movement. After slack in a seat belt is taken up, softer sprung components begin to deflect. As they "bottom", stiffer components start deflecting significantly until finally the belt webbing, passenger skeleton and vehicle structure probably contribute most of the final movement between the passenger and the vehicle.

The soft compliance contributed by the passenger should be minimised by ensuring that the restraint supports his strong, stiff, components. At least one manufacturer of child restraints has reduced the compliance problem by mounting the passenger backwards in a relatively stiff seat.

Sarrailhe reports[4] some developmental work with an energy absorber which significantly reduces the loads produced in a dummy wearing a lap/sash belt by yielding when the load becomes high.

PASSENGER WEIGHT

The deceleration to be expected by a child will be greater than that for an adult when the child is supported by an adult system. Figure 3 indicates that the lighter passenger uses less of the available space, indicating the desirability of "tailoring" restraint systems to the masses of their occupants.

RELATIVE VELOCITY

It will be noted as one observes dynamic testing facilities around Australia that they commence and conclude their impacts between different velocities.

Commencing and concluding velocities which might be observed are

Ford	30 mph before	0 mph after
GMH	0 mph before	-30 mph after
TARU	16 mph before	-14 mph after

Can these three facilities be considered equivalent?

Since all interactions are by way of forces, it is instructive to consider the relationships between the forces on the rigs. Relations are of three kinds:-

1. Inertial forces, $F_i = M_i \dfrac{dv_i}{dt}$

   where v is velocity and M is mass

2. Strain forces, $F_{ij} = f(x_{ij}, t)$

   where f is some function of distances x_{ij} between two components and time, t

3. Force interactions, $F_i = a_j F_j + a_k F_k + \ldots$

   where a_j, a_k, etc. are resolving coefficients which will vary as distances between rig components vary.

Provided that the initial distances between components on two rigs are identical, masses are identical, and all initial accelerations are zero, a given time sequence of force between each rig and the outside world should result in activity defined by the above equations.

The equations are all independent of absolute velocity. They determine velocity change, however.

Thus any given deceleration force sequence, when applied between a self contained rig and the outside world, will result in identical forces and identical changes of velocity irrespective of the initial velocity of the rig. It is therefore possible to represent identical conditions on the three rigs referred to above.

It must be noticed, however, that the distance between a point on the rig and a point on the ground is not independent of absolute velocity. In fact, 10in of impact deformation on the TARU rig is likely to represent 30in of crush in the other rigs and in a real collision.

IMPLICATIONS OF SLED PULSES

In Chapter 1 the two deceleration pulses which have been used extensively during the recent exploratory work of TARU have been shown (Figures 3 and 4).

The narrow pulse approximates that specified by ECE and the flatter pulse approximates that required by BSI.

The barrier crashes of Ford Anglias[3] indicate deceleration curves which are more triangular than both with about the peak deceleration of ECE but with a duration close to that required by BSI.

The model described earlier supports the expectation that the ECE pulse tends to represent a lighter stiffer vehicle, and hence a harsher test, than the BSI type of pulse.

CONCLUSION

A simple model draws attention to the effects of the main parameters in collisions.

The equivalence of different test rigs was shown, and the greater severity of the ECE type of pulse when compared to the BSI type of pulse was shown.

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3. SPECIFICATION OF A TEST DUMMY FOR DYNAMIC TESTING OF AUTOMOTIVE SEAT BELTS

INTRODUCTION

The purpose of this chapter is to derive from all relevant publications and from known unpublished work carried out in Australia, a technically satisfactory and a practical specification for a dummy that can be used for the purposes of both Rule and Standard. This chapter will assume that the test method and pulse are to be specified elsewhere and it should be noted that detailed proposals for such a method are made in Chapter 1.

LIMITS ON DUMMY SPECIFICATION

The specification should preferably permit the use of relatively cheap, simple and rugged dummies. One reason for this is that any dummy used in this work is not required to be a research tool for simulation of all human responses - it is only required as routine test equipment capable of repeatedly and reliably producing realistic dynamic crash loads on seat belts.

At the same time, the specification should preferably not exclude the use of more expensive and complex anthropometric dummies. This is particularly true where such dummies are already available and where they produce suitable seat belt loadings during crash simulation.

For economic reasons, it would be preferable for the specification to permit the use of certain specific dummies which are already available in Australia, or which will shortly be available. These dummies are listed in Table 1.

For the above reasons, it was decided that the specification should be written broadly in the form of a performance requirement, rather than as a specific hardward requirement.

SPECIFICATION DEVELOPMENT

The basic performance requirement for a suitable test dummy is that, ideally, it should load a seat belt in a given (calibration) crash in the same manner as a human being in the same situation - that is, the magnitude and distribution of the loads involved should be identical. It is intended that this loading should be that which would arise without any loads being taken by a footrest, because vehicle occupants in a car crash do not always have the benefit of such a support.

Since the dummy design is not dependent on the use of a particular pulse, the major areas of investigation before writing the dummy specification concern the determination of a dummy mass representative of humans; determination of actual load distributions and values induced in seat belts by humans in crashes; determination of a representative dummy geometry in the areas contacted by a combination seat belts, harnesses and lap belts; and determination of suitable calibration test conditions. These questions are examined in greater detail in the following paragraphs.

Dummy Mass

As far as could be ascertained, the basis for the determination of the masses of all the dummies listed in Table 1 has been the statistically 50th percentile American male. Whether or not the mass of the statistically 50th percentile Australian male is similar (which would seem a reasonable hypothesis) is rather academic, because the economic considerations mentioned in Section 2 above dictate the permitted use of the Table 1 dummies. Table 2 lists a number of additional dummies which have been specified by other organisations for the use in dynamic testing of restraint systems.

It may be noted from Tables 1 and 2 that, except for the ISO and TNO dummies, all dummies have specified masses having very similar values. The most common value is the one based on SAE Recommended Practice J963 "Anthropometric Test Device for Dynamic Testing"[1], that is, 164 pounds $\pm$ 3 pounds (74.4 kg $\pm$ 1.4 kg), and it would seem logical to adopt this value for the Australian test dummy specification, but with a wider tolerance so as not to exclude satisfactory dummies. As will be seen in the next section, the dummy mass does not need to be closely specified, provided that its performance is satisfactory.

Seat Belt Load Distribution

The only published reports on automotive crash simulation experiments with human subjects available to the writer concerned a series of test carried out at Holloman Air Force Base in the United States during the late 1960's[2,3].

The first programme, reported by Armstrong and Waters[2], involved 25 fully instrumented human tests at impacts up to approximately 15G, where type 2 seat belts were worn whilst the subjects were sitting on a hard wooden seat (with the seat pan having a 5 degree downward slope to the front). The subjects had widely varying statures and weights. A footrest was provided and loads were measured at the four extremities of the seat belt system and at the footrest. Throughout this series of tests, the proportioning of the total load to the torso and lap belts varied considerably - largely (apparently) due to the varying proportions of load taken by the footrest. The relevant test results are given in Table 3.

Following the programme described above, a further series of tests was carried out at Holloman Air Force Base, involving 17 subjects wearing combination seat belts in a production automobile bucket seat[3]. Once again the subjects had widely varying statures and weights, and once again a footrest was provided. The instrumentation was similar to that used in the first group of experiments. The relevant test results are given in Table 4.

In order to determine the proportions of the total load which would have been carried by the lap and torso belts, respectively, if no footrest had been used, the following procedure was adopted. The lap and torso loop loads and the horizontal leg forces in Tables 3 and 4 were summed to give a first approximation of the total horizontal load (it is only a first approximation since not all the forces were necessarily acting in exactly the same direction), and the percentage contribution of the three components to this total load were found. The percentage contribution of the lap and torso loop loads in Tables 3 and 4 were then each plotted against the percentage contribution of the footrest load (Figures 1 and 2, respectively) and regression lines fitted using least squares calculations.

From Figures 1 and 2, it may be seen that, when the contribution of the footrest forces to the total load is zero, the lap belt carries 48% and the torso belt carries 52% of the total loop load.

There are a number of published reports containing information on seat belt load distributions for a number of dummies restrained by combination seat belts in simulated crashes. Armstrong and Waters[2] carried out tests with a wooden torso block, a National Bureau of Standards sandbag dummy, two different types of Alderson dummies and a Sierra 292-850 dummy; Chandler and Christian[4] evaluated the performance of five dummy types (the simple Swedish Standard SIS 88 25 52E dummy and four unidentified anthropometric dummies) and Bastiaanse[5] listed performance data in his description of the TNO 10 dummy. The authors have undertaken some crash simulation work with Sierra 292-850 (with 292-325 pelvis) and 292-1050 dummies restrained by combination seat belts in the rear of a Morris Mini car body. These data are summarised in Table 5. The importance of dummy type as a variable in dynamic seat belt testing is illustrated by the wide ranging total belt loop loads (by a factor of two) found by Armstrong and Waters[2] with different dummies under identical test conditions (Table 6).

The authors have also carried out a series of tests with different dummies under identical test conditions (Table 7). Each dummy was tested at least twice: once each on a seat with the cushion sloping 10 degrees upward and 10 degrees downward, respectively, to the front. The joints of the various dummies

(Sierra 292-850 with original pelvis, Sierra 292-1050, Ogle M50/71, and Alderson F-50-AU) were adjusted before each test such that their limbs would just fall under gravity (that is, set for 1G loads).

Bearing in mind the limitations imposed by the scanty availability of suitable data, the following conclusions have been derived.

(a) Human beings restrained only by combination seat belts in a crash induce approximately equal loads in the lap and torso sections of the belt, respectively;

(b) Sierra 292-850 and 292-1050 dummies induce loop and torso sections of combination seat belts in a crash in the ratio of 2:3;

(c) Sierra 292-850 and 292-1050 dummies, Ogle M50/71 and Alderson F-50-AU dummies induce higher loads in the lap belt than in the diagonal belt when tested with separate belts in the same test and using the anchorage geometry specified by ECE[6] and BSI[7];

(d) When testing with separate lap and diagonal belts in the same test, in the BSI/ECE anchorage geometry, use of a downward sloping seat generally results in higher total loop loads and a greater proportion of the loop load being carried by the lap belt than is the case with an upward sloping seat; and

(e) Dummies tested so far impose greater total belt loop loads than do humans. The more advanced and articulated the dummy, the less the increase in load above the figure for humans.

The dummy specification must therefore include a calibration check of its loading characteristics with a standard seat belt and belt geometry using a standard crash pulse. The requirements for loading characteristics might be written in two ways. The first way would be to specify that the total loop load must be evenly distributed to the lap and torso belts and that either the total loop load, the lap belt loop load or the torso belt loop must have a specified value with upper and lower limits. The setting of a lower limit is necessary in order to prevent the use of a dummy that would reduce the total loop load to below that experienced by humans, through the dissipation of some of the crash energy through the resilience characteristics of the dummy structure, or by other means.

The other way of specifying loading characteristics would be by specifying the (equal) load values for the lap belt and torso belt loops. The advantage of this method would be that owner of dummies with poor load distribution characteristics (such as Sierra 292-850 dummies) could still use them, for dynamic testing, provided the specified minimum loop loads were achieved and provided that such dummies were used only for passing (and not failing) seat belts. Such users would, of course, be subjecting the unfavourably loaded strap of their seat belts to greater loads than absolutely necessary. Independent testing authorities and purchasing bodies would have to use dummies which load the lap and torso belts to the nominated values within specified tolerances. Seat belt manufacturers, on the other hand, could not be criticised for applying too much load.

The authors carried out the series of tests mentioned above (and detailed in Table 7) partly in order to determine the values which should be specified as the minimum loop loads to be achieved in the calibration check. As a result of these tests, it is suggested that a suitable dynamic testing dummy for seat belts should, when tested in a calibration test using calibration seat belts, belt geometry and crash pulse, produce loop loads of 10 kN each in the lap belt and torso belt, respectively.

Dummy Geometry

One factor which could be another source of variability in dynamic testing of seat belts is the dummy geometry in the areas where the seat belt contacts the dummy. To avoid this possibility, it is proposed that this geometry be restricted in the dummy specification.

The type of important variation most likely to be encountered between dummies is that of different seat shoulder heights. The most commonly specified dimension for seated shoulder heights of dummies appears to have been based on SAE J963[1], that is, a value of 23.6 inches (599.4 mm) ± 0.8 inches (20.3 mm). All Sierra and Ogle dummy dimensions are based on J963 and the ECE[6] dummy specifies a seated shoulder height of 600 mm. The ISO[8] dummy has a sloping shoulder with the seated height ranging from 590 mm to 665 mm, and the BSI dummy (from the rig drawing) appears to be approximately 600 mm in seated shoulder height.

It seems reasonable, therefore to specify a seat shoulder height of 600 mm ± 50 mm. (At the time of writing, the seated shoulder height of the TNO 10 dummy is unknown).

The design of the dummy pelvis is also important. Early model Sierra 292-805 dummies (Sierra Stan) do not sit properly and are not suitable for belt testing. A pelvis retrofit kit 292-325 was produced by Sierra in order to convert early 850 dummies to sitting dummies. The newer Sierra 292-1050 dummy (Sitting Sid) incorporates the new pelvis.

CALIBRATION TESTING OF DUMMIES

In order to check that any given dummy will perform satisfactorily in a dynamic test of a particular seat belt, the dummy characteristics listed in paragraph 3.2 and 3.3 above must be checked in a calibration test. The specification for this test must include the crash pulse (outside the scope of this report), the calibration seat belt, the belt anchorage geometry, the seat geometry and the pre-test adjustment of the seat belts.

Calibration Seat Belt

Since the loading characteristics of webbing can affect peak loop load magnitude and distribution, it is necessary to specify the type of seat belt webbing to be used in the calibration test. The ISO[8] and BSI[7] dynamic calibration test requirements specify a polyamide continuous filament yarn webbing, with elongation characteristics of 8±1% at 4kN and 17±2% at 11kN static load.

This webbing is not manufactured in Australia and it would be preferable, for ease of supply, to use Australian (polyester) webbing in the calibration seat. A dynamic test comparing polyamide and Australian webbing was carried out by the authors (Table 7). This test showed that Australian polyester webbing provided an acceptable alternative to the European polyamide webbing when stretch characteristics were similar.

The calibration seat belts must consist of separate lap and torso belts, respectively, because the use of a single combination type belt might permit load transfer (through webbing slippage) from one belt to the other. The belts must not have buckles because the dynamic forces generated by the mass of a buckle might affect the final load. Load transducers, for similar reasons, must be anchored to the test frame.

Calibration Geometry

The ECE[6] and BSI[7] dynamic tests for seat belts specify identical test geometry for calibration work and the ISO[8] specifies only slightly different dimensions. All three specifications require that the anchorages shall have sufficient stiffness so that they do not deflect more than 0.2 mm when subjected to a horizontal force of 980N. It is proposed that the ECE/BSI anchorage geometry and stiffness requirements be adopted for the Australian calibration test specification.

The three specifications also require the use of a rigid seat with the cushion sloping upwards to the front at an angle of 10 degrees. It is proposed that, in the Australian specification, this angle should, for calibration purposes, be changed to a downwards-sloping angle of 10 degrees, for the following reasons. (The seat angle in belt testing must be considered in the test method).

In the late 1960's, Armstrong and Waters[2] carried out simulated crash tests with Alderson 175 pound 50th percentile adult dummies restrained by combination seat belts in securely anchored 1965 model bench and bucket seats. They found that the dummies hips moved down into the seat cushions, as the dummies moved forward during the deceleration of the seat. The angle of movement was roughly 10 degrees. They repeated the tests using rigid seats inclined at various angles and found that the loop loads in simple lap belts and in the lap sections of combination belts increased as the seat base angle was changed gradually from upward-sloping to downward-sloping; however, the loop loads in the sash parts of combination belts decreased so that the total loop loads in combination belts remained unchanged with seat belt angle.

They concluded that the seat base angle for a smooth, rigid sled seat should be -10 degrees.

In a dynamic test, part of the total horizontal force on the dummy generated by the sled deceleration may be taken by friction between the dummy and the seat. Downward-inclined smooth seats cannot exert any friction load during a test, provided that the angle exceeds the angle of friction between the dummy and the seat.

If the object in specifying seat angle is to remove its influence (as it must be for calibration of dummies), then -10 degrees would be a suitable angle, provided that the dummy were so constructed as to have a smooth pelvic base free of projections that might raise the angle of friction to 10 degrees or more. Seats having greater negative angles might be necessary with some dummies that do have projections.

Armstrong and Waters'[2] work on test seat angles is supported by Chandler and Christian's[3] tests with human subjects restrained by combination seat belts in production bucket seats. The latter researchers found that the subjects' hips moved forwards and downwards during crash simulation.

In a real crash, the car seat takes some of the crash forces exerted by a restrained occupant, by compression of the seat cushion. At the present time, car seat cushions are not required by Design Rules to withstand these forces and, therefore, it must be assumed that in many crashes, the seat will not withstand the forces, but will fail. Under these circumstances, a vehicle occupant's seat will be required to withstand all of the crash forces generated by the wearer. It therefore follows that, when dynamically testing seat belts, the test conditions should be such that the test seat should not take any of the crash forces generated by the seat occupant. A test seat with a cushion sloping downwards to the front satisfies these conditions. (Failure of the seat might, of course, apply to the belt additional loads from the seat's kinetic energy.) This subject will be referred to again in the report on dynamic test procedures.

In view of the above, it is proposed that the calibration test seat should permit the dummy hip to move naturally during the crash simulation and that the test seat should be such that it does not absorb any energy by friction between the dummy and the seat. It is concluded that the seat should slope 10 degrees downwards to the front, because with this angle,

(a) the changes in dummy and belt attitude during a sled run are representative of what happens on car seats;

(b) The influence of friction should be removed; and

(c) small differences of sled seat and dummy construction and of test procedures are less important with downward than with upward sloping seats.

The proposed calibration geometry is shown in Figure 3.

Adjustment of calibration seat belts

The amount of slack or pre-test tension in a seat belt can have a significant effect on belt loadings. For the purposes of calibration of dummies, it is therefore necessary to remove this source of variability. For calibration purposes, it is proposed to adopt the I.S.O. recommended 44N (10 pounds) initial tension, although the I.S.O. tolerances of ±8N(2±pounds) may prove to be too close to be practicable.

CONCLUSIONS

Based on the data currently available, the following specification for a test dummy for dynamic testing of automotive seat belts has been evolved.

(a) The dummy shall have a mass of 57 to 80 kg.

(b) The dummy shall have a seated height from seat to shoulder of 600 mm ± 50 mm.

(c) When subjected to a calibration test, the dummy shall load the lap belt to 10 ± 1kN and the diagonal belt to 10 ± 1kN. In each case, the belt loads should preferably be evenly distributed between the anchorages.

ACKNOWLEDGEMENT

The authors are indebted to the Ford Motor Company of Australia for making an Alderson F-50-AU anthropometric dummy available for use in the crash simulation experiments.

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Make and Model	Dummy Mass	
	Pounds	Kilograms
Sierra 292-850	162.9	73.9
Sierra 292-1050	164 ± 3	74.4 ± 1.4
Ogle-Mira M50/71	164 ± 3	74.4 ± 1.4
Alderson F-50-AU	164	74.4
TNO (Bastiaanse) 10	Adjustable	

TABLE 1: Dummies available in Australia (currently or in the immediate future).

Make and Model	Dummy Mass	
	Pounds	Kilograms
International Standards Organisation[8] (in ISO/TC 94/SC)	154.32	70
Economic Commission for Europe[6] WP29/419	163.8 ± 3.1	74.3 ± 1.40
N.H.T.S.A.[10]	164 ± 3	74.4 ± 1.4
Swedish Standard SIS 88 25 52E[9]	163	74
British Standards Institution[7] (in AU160a:1971)	165.35	75

TABLE 2: Some dummies specified for dynamic testing of seat belts.

Run No.	Loop Load (lbf) lap belt	torso belt	Horizontal footrest force (lbf)	Total force (lbf)	Percentage distribution of force lap belt	torso belt	footrest
3928	510	170	1138.4	1818.4	28.1	9.4	62.5
3930	600	380	721.2	1701.2	35.3	22.3	42.4
3931	960	360	693.0	2013.0	47.7	17.9	34.4
3933	580	190	385.4	1155.4	50.1	16.5	33.4
3934	790	320	445.5	1555.5	50.8	20.6	28.6
3935	800	680	788.4	2268.4	35.3	30.0	34.7
3936	780	660	434.9	1874.9	41.6	35.2	23.2
3937	680	480	845.0	2005.0	34.0	24.0	42.0
3940	840	440	604.6	1884.6	44.5	23.4	32.1
3941	620	650	1145.5	2415.5	25.7	27.0	47.3
3942	580	540	374.8	1494.8	38.8	36.1	25.1
3943	620	180	583.4	1383.4	44.9	13.0	42.1
4059	740	470	251.8	1461.8	50.6	32.2	17.2
4062	310	60	395.4	765.4	40.5	7.8	51.7
4065	460	530	377.3	1367.3	33.7	38.8	27.5
4068	370	270	497.3	1137.3	32.6	23.8	43.6
4070	710	200	475.3	1385.3	51.2	14.5	34.3
4071	550	530	619.9	1699.9	32.4	31.2	36.4
4072	730	860	491.6	2081.6	35.1	41.3	23.6
4073	540	530	822.2	1892.2	28.6	28.0	43.4
4074	330	580	772.0	1682.0	19.6	34.5	45.9
4075	550	350	507.9	1407.9	39.0	24.9	36.1
4076	1120	700	695.4	2515.4	44.5	27.8	27.7
4077	600	570	692.1	1862.1	32.2	30.6	37.2
4085	570	690	395.2	1655.2	34.4	41.7	23.9

TABLE 3: Load distributions in human subject experiments by Armstrong and Waters[2]

Subject	Loop load (lbf) lap belt	torso belt	Horizontal footrest force (lbf)	Total Force (lbf)	Percentage distribution of force lap belt	torso belt	footrest
A	1356	1231	863	3450	39.3	35.7	25.0
B	1087	1080	1081	3248	33.4	33.3	33.3
C	407	558	932	1897	21.4	29.4	49.2
D	1570	1600	1188	4358	36.0	36.8	27.2
E	1638	1256	634	3528	46.4	35.6	18.0
F	1371	1208	844	3423	40.0	35.3	24.7
G	637	914	637	2188	29.1	41.8	29.1
H	1019	835	865	2719	37.5	30.7	31.8
J	849	1033	968	2850	29.8	36.2	34.0
K	769	1010	802	2581	29.8	39.1	31.1
L	1260	1457	1306	4023	31.4	36.2	32.4
M	996	781	914	2691	37.0	29.0	34.0
N	800	1488	715	3003	26.6	49.6	23.8
O	1077	988	780	2845	37.8	34.7	27.5
Q	923	1038	672	2633	35.1	39.4	25.5
R	996	1198	666	2860	34.8	42.9	23.3
S	1533	1429	478	3440	44.6	41.5	13.9

TABLE 4: Load distribution in human subject experiments by Chandler and Christian[3]

Data Source	Dummy	No. of tests	Percentage of total belt loop loads (and standard deviation) lap belt	torso belt
Armstrong & Waters[2]	Wooden torso block	3	54.2 (1.6)	45.8 (1.6)
"	N.B.S. sandbag dummy	3	46.5 (6.6)	53.5 (6.6)
"	Alderson F-50	2	50.1 (6.2)	49.9 (6.2)
"	Alderson VI-50	3	46.3 (0.2)	53.7 (0.2)
"	Sierra 292-850	6	40.5 (3.0)	59.5 (3.0)
Chandler & Christian[4]	Swedish	12	50.0	50.0
"	No. 2 (162 lb)	17	58.0	42.0
"	No. 3 (163 lb)	19	48.0	52.0
"	No. 4 (170 lb)	30	50.0	50.0
"	No. 5 (217 lb)	14	50.2	49.8
Bastiaanse[5]	TNO 10	10	49.8 (1.6)	50.2 (1.6)
The authors	Sierra 292-850 (325)	2	35.3 (1.0)	64.7 (1.0)
"	Sierra 292-1050	13*	37.4 (7.4)	62.6 (7.4)
"	Sierra 292-1050	10**	42.2 (6.6)	57.8 (6.6)

TABLE 5: Proportions of total loop load carried by lap and torso belts using various types of dummies.

* Initial slack clearance of 3½ inches on torso belt.

** Initial slack clearance of 1 inch on torso belt and angle or torso belt altered slightly from *

Dummy	Total belt loop load (lbf)
Wooden torso block	9760
Alderson F-50	8370
National Bureau of Standards sand bag dummy	7530
Alderson VI-50	6570
Sierra 292-850	5630
Human (extrapolated)	4250

TABLE 6: Total loop loads in lap and torso belts found by Armstrong and Waters[2] using different dummies under identical test conditions (30g sled deceleration, ½ sine wave pulse, 40 ft/sec, 19 inches stopping distance).

Seat Cushion Slope	Dummy	Peak webbing loads (kN) F_1 L.H. (top) diagonal	F_2 R.H. (bot.) diagonal	F_3 L.H. lap	F_4 R.H. lap	Total peak loop loads (kN)	Total load corrected to 18g pulse (kN)	Sled Run Number
10° up	Sierra 292-850	5.50	4.47	5.61	5.80	21.38	20.58	DHD5
"	"	6.40	3.92	5.37	5.68	21.37	20.46	DHD6
"	"	5.99	3.97	5.51	5.63	21.10	20.42	DHD8
"	Sierra 292-1050	5.79	4.03	5.84	5.77	21.43	20.85	DHD3
"	Ogle M50/71	6.30	4.86	6.25	6.06	23.47	21.66	DHD4
"	Alderson F-50-AU	5.32	3.97	5.76	5.83	20.88	20.76	DHD7
10° down	Sierra 292-850	6.65	4.06	6.51	6.06	23.28	21.71[1]	DHF4
"	"	6.51	4.34	7.24	7.25	25.34	23.63[2]	DHF5
"	Sierra 292-1050	6.31	4.59	7.11	6.57	24.58	24.86[3]	DHF2
"	Ogle M50/71	5.75	4.30	6.86	6.42	23.33	23.73[4]	DHF3
"	Alderson F-50-AU	4.86	3.80	6.13	5.83	20.62	19.74[5]	DHF1
10° up	Sierra 292-850	6.48	4.22	5.56	5.77	22.03	22.03[6]	DHH1
"	Ogle M50/71	6.06	4.50	6.35	5.29	22.20	23.20	DHH2

TABLE 7: (continued on next page)

Seat Cushion Slope	Dummy	Percentage load distribution: diagonal belt	Percentage load distribution: lap belt	Actual sled pulse: acceleration (g)	Actual sled pulse: duration (ms)	Sled run number
10° up	Sierra 292-850	47	53	18.7	115	DHD5
"	"	48	52	18.8	118	DHD6
"	"	47	53	18.6	119	DHD8
"	Sierra 292-1050	46	54	18.5	119	DHD3
"	Ogle M50/71	48	52	19.5	118	DHD4
"	Alderson F-50-AU	45	55	18.1	118	DHD7
10° down	Sierra 292-850	46	54	19.3	119	DHF4
"	"	43	57	19.3	120	DHF5
"	Sierra 292-1050	44	56	17.8	118	DHF2
"	Ogle M50/71	43	57	17.7	118	DHF3
"	Alderson F-50-AU	42	58	18.8	117	DHF1
10° up	Sierra 292-850	49	51	18.0	120	DHH1
"	Ogle M50/71	48	52	17.2	119	DHH2

TABLE 7: (continued on next page)

TABLE 7: The authors' test results with different dummies and two different seat cushion angles, under the same test conditions.

Test Conditions: ECE[6] seat and anchorage geometry using two separate belts (lap and diagonal) in the same test and without buckles. The webbing used was Australian polyester material with the exception of the last test (DHH1), which was carried out using European polyamide webbing. The belts were firmly tightened before each test. The tests were carried out on an M.T.S. rebound-type crash simulator using a half sine wave pulse (nominally 18g, 120 ms, duration, 30 kmh impact velocity, 275 mm programmer stroke).

Footnotes:

1. The total peak loop load is 6% higher than the average for DHD5, 6, 8.
2. The total peak loop load is 15% higher than the average for DHD5, 6, 8.
3. The total peak loop load is 19% higher than for DHD3.
4. The total peak loop load is 9% higher than for DHD4.
5. The total peak loop load is 5% lower than for DHD7.
6. The total peak loop load (using the polyamide webbing) was 8% higher than the average for DHD5, 6, 8 - see Table 8.

Webbing Type	Breaking Load (kN)	Stretch at 4kN load	Stretch at 11kN load
Imported Polyamide MT 326	24.78	11%	18.8%
Australian Polyester MT 338	25.13	7.8%	21.8%

TABLE 8: Comparison of load-stretch characteristics of a sample of polyamide and polyester webbing.

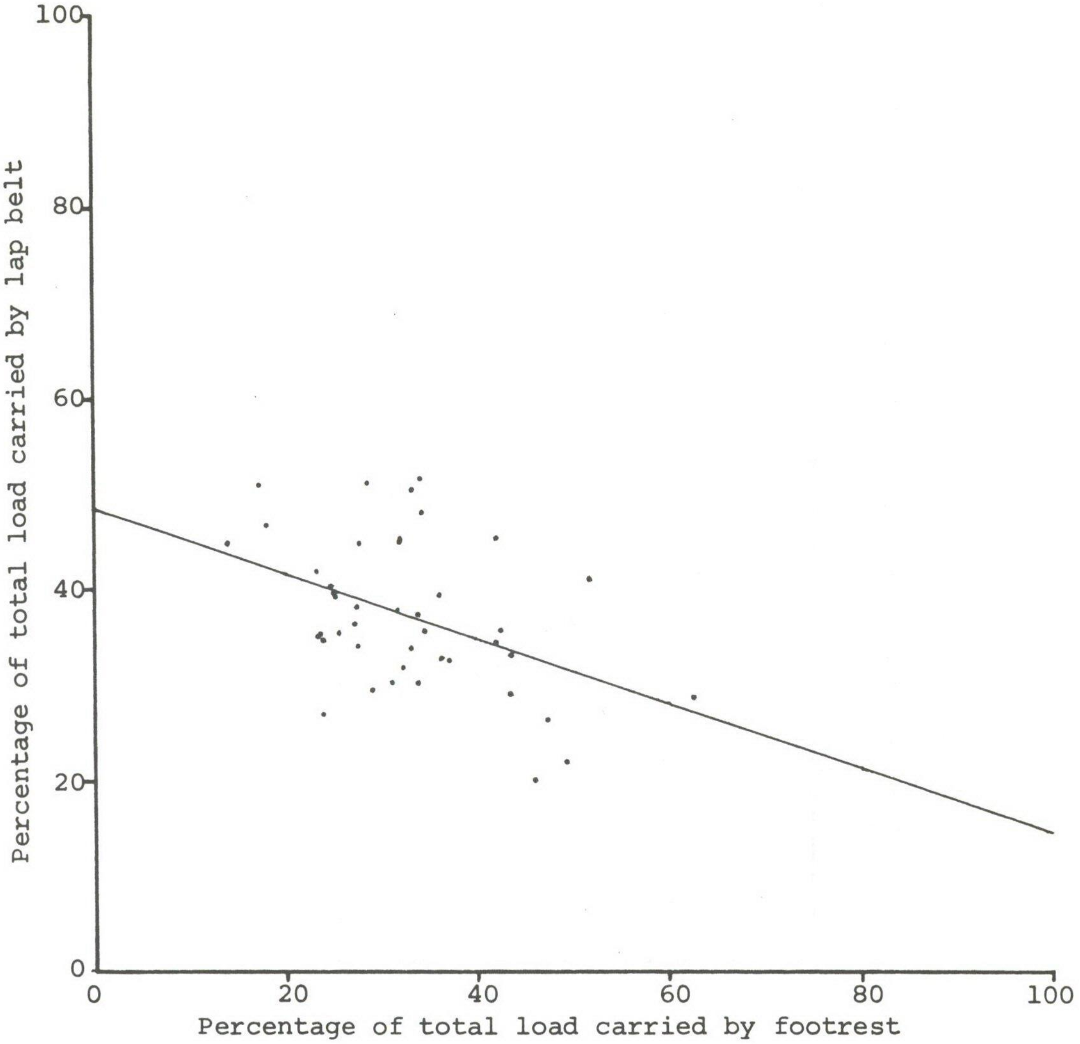


Figure 1 : Lap belt loop load (as % of total load) plotted against footrest load (as % of total load) - after Armstrong and Waters[2], and Chandler and Christian[3].

Regression line has equation

$$Y = 48.1208 - 0.3477X$$

Correlation Coefficient = -0.4449

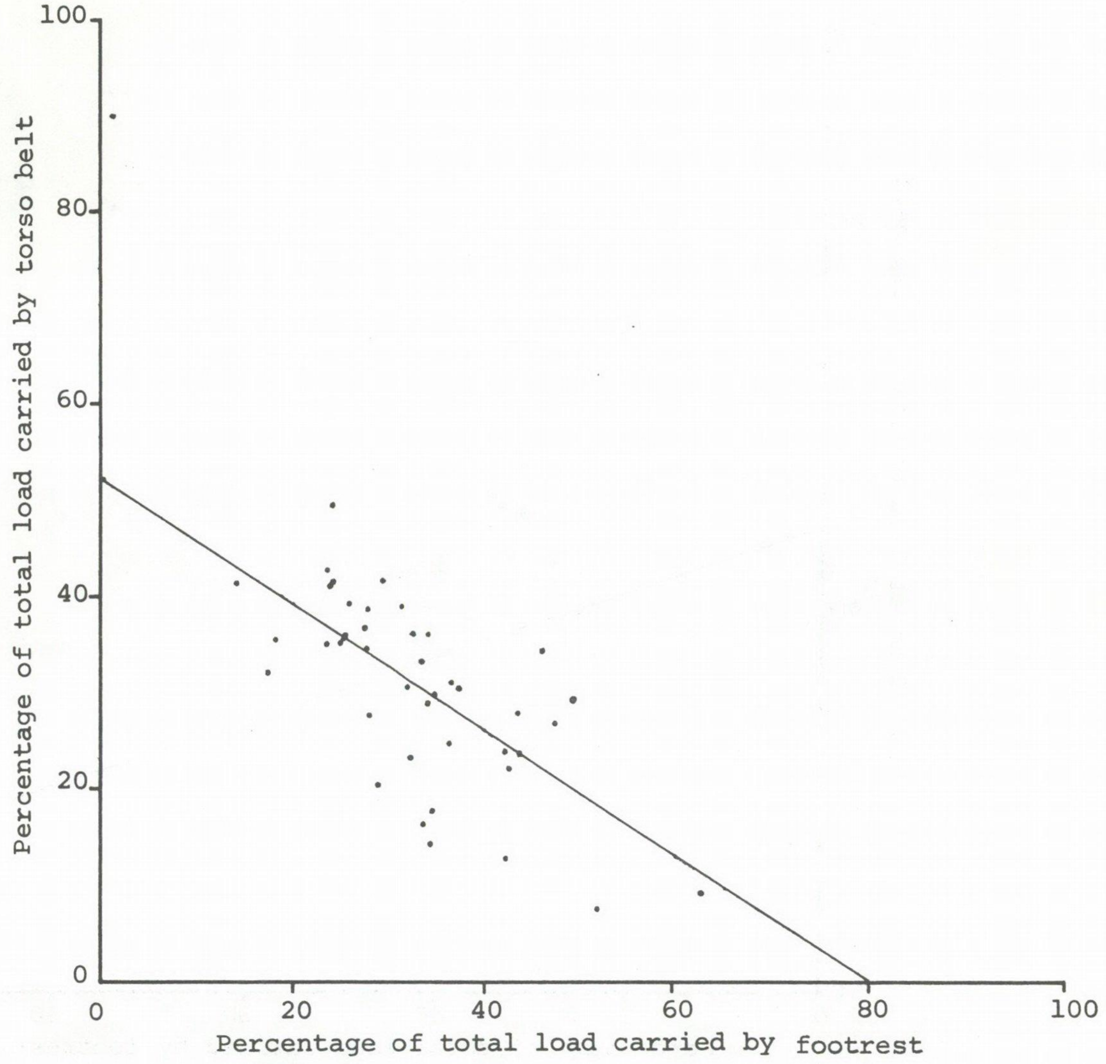


Figure 2 : Torso belt loop load (as % of total load) plotted against footrest load (as % of total load) - after Armstrong and Waters[2], and Chandler and Christian[3].

Regression line has equation

$$Y = 51.9801 - 0.6546X$$

Correlation coefficient = -0.6821

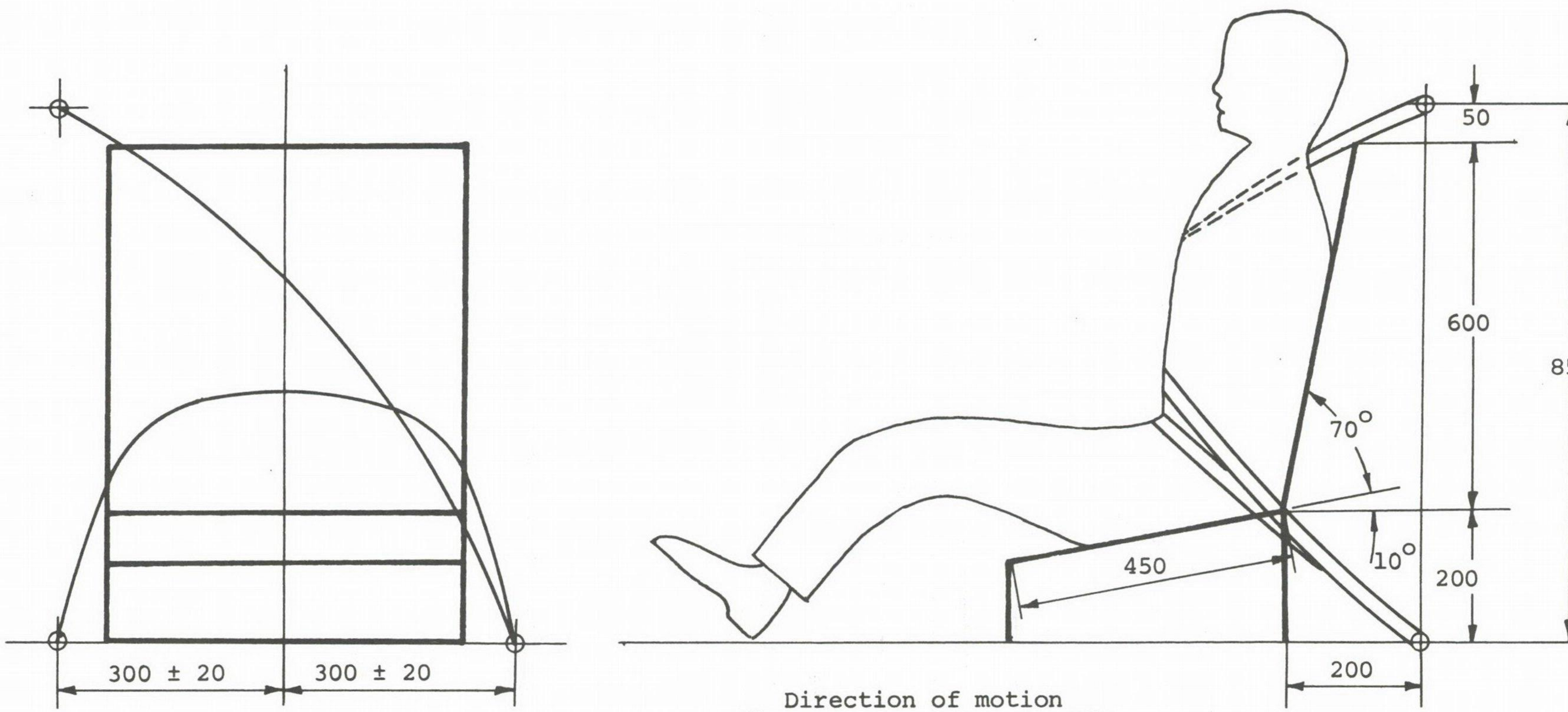


FIGURE 3: Location of anchorage points and relevant seat dimensions for calibration test (Dimensions in millimetres).

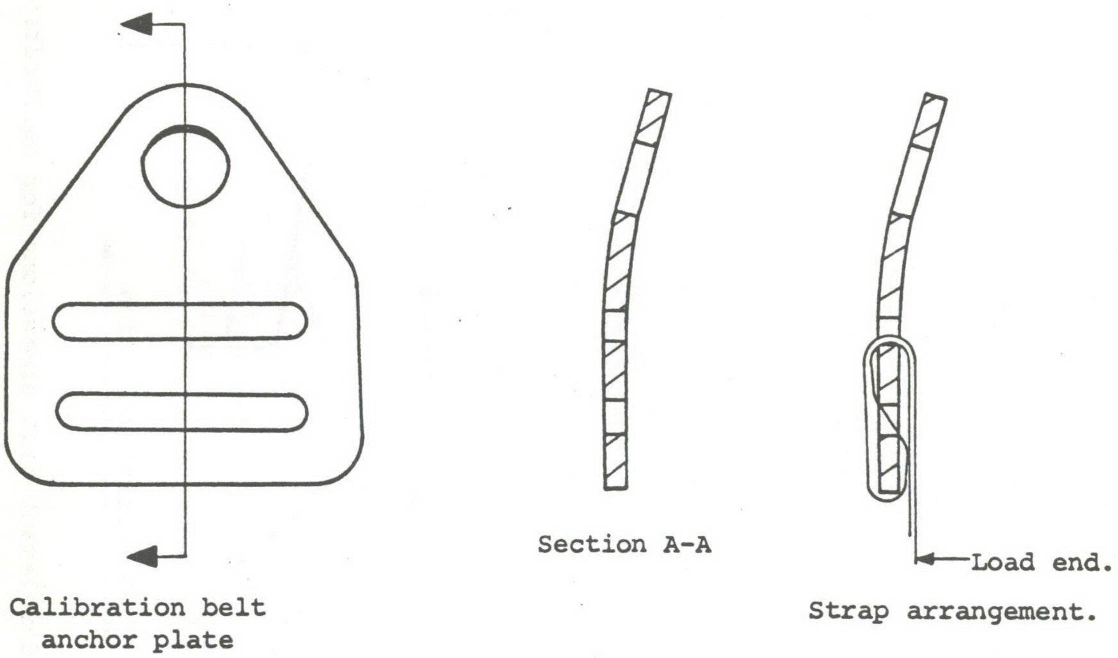


FIGURE 4: Typical strap anchorage fitting.

4. THE DESIGN AND OPERATION OF SEAT BELT BUCKLES

INTRODUCTION

The safety and reliability of a seat belt is only as good as its weakest and most unreliable component. The buckle represents probably the most complex single component of a (non-retracting) seat belt and is potentially one of the likely sources of unreliability.

This paper sets out to evaluate the mechanism of automotive seat belt buckles, from the viewpoints of safety and reliability. It covers both lever and push-button operated latching buckles and is relevant to the requirements of Australian Standard E35, "Seat Belt Assemblies for Motor Vehicles".

Consideration will be given in the paper to the general design of buckle latching mechanisms, the reliance on buckle covers for the retention of mechanisms and the accidental opening of buckles by inertial forces and off-set forces resulting from belt geometry.

LATCHING ACTIONS

For convenience of evaluation, seat-belt buckle latching mechanisms can be divided into three general types:

(a) Positive Blocking Type;

(b) Compressive Latching Member Type; and

(c) Tensioned Latching Member Type.

There are also one or more special types of latch, and one of these is described later.

Because of the limitations imposed by available space and by mass production methods, buckle latching and releasing mechanisms generally comprise rotating or tilting members rather than sliding ones. An exception is the sliding blocking bar in one design of positive blocking type latch.

The three forms of buckle latch listed are now described. Various makes of seat belt are listed in Appendix A, under the type of latch mechanism characterising the buckle.

Positive Blocking Type Latch

With this latch, the seat belt tongue engages in a projection or, possibly, a recess within the buckle after insertion, whereupon a rotating or sliding blocking bar is inserted by a spring between the tongue and a fixed part of the buckle, positively locking the tongue in engagement (Figure 1). When the release button is actuated, the blocking bar is withdrawn and the tongue is free to be withdrawn from the buckle. In another design, the blocking bar is a rotating member, withdrawn by a releasing lever.

The projection, which the tongue engages, may have a latching surface at slightly more than 90^0 to the plane of the buckle base. This facilitates freeing of the tongue without impairing the holding power of the latch, as the tongue is positively held by the blocking bar in the latched position over the buckle projection. There is little ill-effect from wear of the corners of the blocking bar, of the tongue where it engages the blocking bar or buckle projection, or of the projection itself. While under intensive load, the friction of the system makes the latching effect independent of the latch spring.

Indentation of the various mating surfaces under loading in these types of buckles is not considered likely to be sufficient to cause any difficulty in meeting the AS E35 unlatching test (at the conclusion of the Strength of Assembly Test). The test requirement is that the force needed to release the buckle must not exceed 25lbf while the seat belt assembly is still supporting a load of 250lbf.

Compressive Latching Member Type Latch

After the tongue is inserted into a buckle of this type, the latching surface on the tongue (formed by either a projection on, or opening in the tongue) is engaged by a spring-loaded latch. The latch pivot axis is so located that the latch is subjected to compression forces when the belt is loaded.

The angle between the buckle base and line A-A' in Figure 2 (passing through the latch pivot axis and the line of contact on the latching surface) is not critical, ranging (amongst brands) from about 12° to 45°. This angle will be called the latch thrust angle. When load is applied through the tongue, friction between the latching surfaces of the latch and the tongue tends to cause rotation of the latch member in the latching direction, so that, while under intensive load, the latch does not depend on its spring to keep is engaged. The greater the latch thrust angle, the greater is the tendency for the latch to self-energise. Thus, this latching principle has inherent stability.

The latching surface of the latch is preferably arcuate about the pivot axis. The accuracy of this relationship is of less importance for the higher values of latch thrust angle. For this reason, wear of latching surfaces of the latch and the tongue does not have a critical effect on the latching function. Indentation of the latching surface under intensive load would assist the self retaining property of the latch, although it might cause a difficulty during the AS E35 unlatching test. However, one maker successfully uses an aluminium alloy, much softer than steel, for both the tongue and the latch in a compressive latching member type of buckle.

While in most of the examples noted the latch pivot axis was parallel to the plane of the tongue, in one make of buckle (a rigidly mounted model), the pivot axis was at right angles to the plane of the tongue; the latch engages in a notch in the side of the tongue, which is unusually thick.

2.3 Tensioned Latching Member Type Latch

In a typical buckle incorporating this latch (Figure 3), the tongue is held in the buckle by a spring-loaded latch. The latch pivot is so located that when the seat belt is loaded, there is a tensile load in the latch, between the latching surface and the pivot.

The direction of line A-A' (passing through the pivot axis and the line of contact on the latching surface) determines whether the latch tends to engage itself or disengage under load. If this line slopes away from the buckle base at the tongue end, the latch will tend to self-engage. If the line slopes toward the buckle base (as in Figure 3) the latch will tend to self-open by rotating about the pivot in an anti-clockwise direction until equilibrium is reached (when the line of contact on the latching surface approaches the line of load B-B' passing through the pivot axis). Thus, in the example shown, the latch mechanism does not have inherent stability.

The angle between the sloping line and the buckle base will be called the latch tension angle. If it is assumed that a coefficient of friction of 0.1 exists between the latching surfaces shown in Figure 3, for a tongue load of 2000 lbf, the frictional holding force would be 200 lbf.

The unlatching force is proportional to the sine of the latch tension angle and would be 240 lbf for the angle of 7 degrees shown, reducing to 140 lbf if the angle were reduced to 4 degrees and to 70 lbf at 2 degrees.

The situation could be improved by undercutting the latching surface of the tongue projection, to match that of the latch. It could be further improved by displacing the tongue relative to the latch pivot axis, as shown in Figure 4. This would make the latch tension angle slope in the opposite direction. A maximum value of 8 degrees would occur in the case shown if the latch were to be rotated slightly so that its latching surface bore on the upper corner of the tongue projection latching surface. This would have a restoring effect on the latch. Moreover, this would be augmented by a slight undercutting effect, since, as the latch pivots to unlatch, the lower corner of the latching surface moves at a shorter radius than the upper corner.

Only sparing use can be made of undercutting in the cases shown in the Figures 3 and 4, because of the limitations on unlatching force imposed under AS E35.

The finely poised geometrical relationship of this type of latch can be distributed by only slight wear of the latching sufraces, particularly at the corners. The material used should therefore have good wearing qualities, as well as adequate strength.

Special Types of Latches

One type of seat belt fitted to a popular make of Australian-built car uses a buckle with a special type of latch. The buckle mechanism utilises an unusually shaped "tongue", in the form of a deep hook made from sheet steel. The tongue engages with the radiused heel of a moulded plastic release lever pivoted on a pin mounted in a webbing adjuster. The hook is deep enough to ensure stable engagement with the release lever heel. The lever and heel are kept in the latched position by a permanent magnet and any load on the belt causes the lever to be urged further into the latched position. When the release lever is actuated, a step adjacent to the radiused surface bears against the tip of the hook and lifts the heel out of the hook, thus releasing the belt.

The purely latching action of this buckle appears to be especially effective, although release can be effected by rotation of the tongue in the plane of the webbing. However, unintended release does not appear to be a problem in ordinary service (no complaints have been received). Nevertheless, it is possible that, under dynamic conditions, the tongue might rotate under the influence of an upward pull of the diagonal belt webbing.

MANUAL UNLATCHING OF BUCKLES

In Ordinary Service

The unlatching of any buckle should be straightforward and not involve undue effort. Lever type releases should be designed so as to avoid pinching an operator's fingers during unlatching of the buckle. It should not be possible to tilt the tongue in the buckle in such a way as to hinder or prevent its withdrawal.

After Strength of Assembly Test

If more than the specified force is needed to release a buckle at the conclusion of the Strength of Assembly Test of AS E35, the cause may be excessive friction between latching surfaces due to the remaining load. This is why the load may not be returned to zero before carrying out the unlatching test in AS E35. On the other hand, it would be advisable to inspect the latching surfaces for indentations and to check for any distortion of base, latch, pivot pin or other latch component subject to loading, and for distortion of the tongue.

UNINTENDED OPENING OF BUCKLES

For every buckle design, there is always the possibility that, after being properly latched, the buckle may unintentionally be opened. This may arise through inertial effects, wear, deflection or excessive clearances of components, failures of springs and rotation of buckle tongues (due to seat belt geometry). Inertial effects are examined in detail later in this report. The other aspects are considered as follows.

Unintended release of tensioned latching members, or compressive latching members with small latch thrust angles, could occur if wear of the latching surfaces caused a change in their angle of latching. Thus, durability testing of buckle mechanisms is essential to the continued safe and reliable operation of seat belts.

If the geometry of a buckle mechanism does not afford a good margin of latching reliability, the buckle could become unlatched under load, through distortion of the parts involved. The Strength of Assembly Test of AS E35 should resolve any questions of weakness in this regard.

In theory, a latch could be opened if clearance between latching surfaces (in the latched position) were excessive and the latch bounced off the tongue when load was suddenly applied. However, this potential source of trouble appears to be well recognised, and no buckles with excessive clearance of this kind have been seen in practice.

Continued integrity of latch springs is essential for reliability of latching. Springs should be held in position in such a way that they cannot be dislodged, either in ordinary service or under crash conditions. Durability testing of buckle mechanisms should ensure the resistance of latch springs to permanent deformation or fracture through fatigue.

In some buckle designs, the tongue is not effectively guided within the buckle and unlatching can occur when the buckle is rotated in the plane of the webbing. The unlatching can result from a tongue latching surface of unsuitable angle coming into contact with the latch during the rotating movement. In one case noted, the latching portion of the tongue consisted of a dimple formed in it. The flat latching surface of the tongue merged into rounded shoulders, which were brought into contact with the buckle latch when the tongue was rotated. Latches embodying a tensioned latch member are peculiarly vulnerable to tongue rotation, which could possibly be caused in a crash by upward pull of the diagonal belt webbing. In one buckle example, any tendency for the tongue to rotate was minimised by providing the tongue with parallel edges fitting neatly in the buckle guides.

Where a buckle cover is relied upon to hold the parts of the latch mechanism in place, the cover should be secured by sufficiently reliable means to prevent it from being detached - especially during a crash. Particular attention should be given to arrangements where the stiffness of sheet metal or moulded plastic parts is relied upon to keep the buckle base and cover interlocked. Several complaints have been made to the Traffic Accident Research Unit concerning the accidental detachment of covers from buckle bodies in ordinary service.

Opening of Buckles under Inertial Forces

Buckles should not unlatch as a result of being subjected to inertial forces induced in their components in crashes. Deceleration of a buckle may be directly associated with that of the vehicle concerned, but short duration pulses of much higher amplitude can also occur if the buckle is flung against a hard object in the vehicle during the crash. Little specific information is available on the latter aspect at this time.

Vehicles in a crash can be subjected to impacts from many different directions. They can be struck at the front, at the sides (square-on and at various angles) and from the rear and they may be involved in rollovers. Thus, the seat belt buckles in such vehicles may be subjected to inertial loads acting in any direction. It is therefore important that, when considering dynamic loads on buckles, not only frontal collisions be considered, but that others also be taken into account.

The likelihood of inertial opening of a buckle may be examined in the design stages by calculation based on measurement of the latch components. It is highly desirable that the likelihood be also checked by dynamic tests.

The authors have carried out a program of dynamic testing on seat belt buckles and the program and test results are discussed in the following section.

DYNAMIC TESTING OF AUSTRALIAN SEAT BELT BUCKLES

The authors carried out a series of dynamic tests on a range of Australian automotive seat belt buckles. Two different types of tests, based on possible collision conditions, were separately carried out. The tests consisted of subjecting the buckles to

(a) deceleration forces representative of those imposed on vehicles in crashes, and

(b) acceleration forces such as might be encountered during impact with a hard object within a vehicle (that is, a vehicle fitting) under collision conditions.

It was recognised that suitable belt tensions would need to be chosen for the tests.

Test (a) - Equipment and Procedure

Six different brands of seat belts were purchased at random: three incorporating lever type buckles and three having push-button operated buckles. The tests were carried out using the Traffic Accident Research Unit crash simulator sled fitted with a programmer giving a generally square wave acceleration pulse (Figure 5).

A heavy (16kg) steel block was firmly bolted to the horizontal surface of the sled and the buckle test rig mounted on this block (Figures 6, 7 and 8). In the first few tests, an accelerometer was mounted on the block, but this was later removed (to avoid possible damage to it by flying belt components) after it had been established that the block and sled accelerations were identical.

The buckles were each oriented on the rig so that the acceleration forces on the mechanism in the simulated crashes would tend to open the buckle. In each case, the body of the buckle was attached to the leading or trailing edge of the metal block by bolts, screws or a clamp (according to its type of construction). The tongue was engaged and the belt fed around two rollers to terminate at a spring balance (which was used to set the belt tension). Care was taken to ensure that the tongue entered the buckle body with no lateral or transverse binding or rubbing.

Each buckle was subjected to 20g, 30g and 40g square wave pulses. Buckles which opened during these tests were also subjected to (nominally) 35g and 45g square wave pulses. All pulses were of (nominally) 35 millisecond duration.

Test (b) - Equipment and Procedure

The push-button operated examples of the six seat belts used in test (a) were used in these tests.

The tongue of the seat belt buckle under test was attached (via webbing) to a large, calibrated tension spring, which was in turn attached to one fork of a fork-lift truck. The buckle body was attached via webbing to a large (68kg) mass on the ground (Figure 9). The tension in the coupled assembly could be varied by raising or lowering the fork and its value determined by measuring the length of the calibrated spring. After some preliminary experiments, a tension of 334N (74lbf) was adopted for all tests.

A stiff (16 gauge brass) plate was screwed to the buckle under test and an accelerometer was attached to this plate. The accelerometer was oriented so that its output indicated the G force exerted in a direction perpendicular to the face of the buckle.

The buckles were accelerated by abruptly striking their exposed backs with the plastic handle of a screwdriver wielded by one of the laboratory staff. If a buckle remained latched in a test, it was released manually and then re-set for the next test.

The accelerations involved in this series of tests were recorded and measured by feeding the output of the accelerometer into a storage oscilloscope. The single-shot time-base of the oscilloscope was D.C. coupled and the trigger set to its most sensitive point on a positive going signal. Preliminary tests were carried out to determine the amplitudes that might be expected and then the vertical amplifier gain set to give a display of not less than one centimetre. The time-base then triggered between 0 and 3 or 4 mm.

In this way it was possible to measure the amplitude of the first pulse, since the accelerometer was orientated so as to give a positive going signal on impact. A fast time-base setting was used to enable easy measurement of the duration of the impact pulse.

Test Results

The results of the test described in 5.1 and 5.2 above are summarised in Tables 1 and 2 respectively.

Of the six types of seat belt buckle subjected to crash pulses on the simulator, only one (the buckle incorporating a magnetically-held latching system) showed any tendency to open. This buckle latching action is apparently more sensitive to a deceleration pulse than the "conventional" mechanical latching systems associated with other specimens.

The high level, short duration acceleration tests on the push button-operated buckles indicate that inadvertant release under these conditions is possible. The probability of these pulses occurring in the tested direction in accidents is unknown at this time.

DESIGN PRINCIPLES FOR RESISTANCE TO INERTIAL UNLATCHING OF BUCKLES

Seat belt buckle mechanisms incorporate sliding and/or rotating members which are held in position by springs.

In the case of a sliding member, the mass of the member and the friction in the slide are the most important factors. Minimising the mass will minimise inertial forces for a given acceleration. It should not be overlooked, however, that any resisting spring acting on the slide will also be subjected to inertial forces. These forces will reduce the resisting force exerted by the spring on the sliding member and thus increase the tendency for the mechanism to unlatch. The mass of any spring in the mechanism should therefore also be minimised. Friction in the slide will, of course, result in resistance to inertial unlatching, but its effect will be limited because, if the value is too high, problems could arise in meeting the unlatching force requirements of AS E35.

The most important factors affecting the inertial unlocking of latches with rotating members are (once again) the mass of return springs, and the moments of the rotating members about their pivots. Both the spring masses and the moments of the latch members should be minimised to reduce any tendencies for the buckles to open under inertial loadings.

CONCLUSIONS

From the view point of sensitivity to manufacturing tolerances, mechanism wear and inertial effects, the location of the latch pivot and the angles of latching surfaces are probably more important for tensioned member latching types than for the other two general types. For the first-named type, the choice of angles of latching surfaces is limited by the need to ensure that the unlatching force is within the specified limit when the AS E35

Strength of Assembly Test is performed. Effective quality control is necessary to ensure that the designed latching surface angles are maintained under the mass production methods used, while materials should be sufficiently durable to ensure that these angles are not degraded by wear. The tensioned latching member type of latch is especially sensitive to rotation of the buckle tongue in the plane of the webbing.

Latch mechanisms of the compressive latch member type are hardly affected by the foregoing factors. Where the latch thrust angle is large, the reliability and stability of this form of latch closely approaches that of the positive blocking type.

The positive blocking type latch is not sensitive to geometry or shape and hence is scarcely affected by wear. It is inherently the most stable in design and reliable in performance of the three types described.

The hook-type buckle with magnetic latch combines a positive action with simplicity and negligible wear effect. However, it is known from tests to be subject to inertial unlatching at a lower deceleration than is the case with most other types of buckles. Moreover, the tongue can be fairly easily rotated in the plane of the webbing, and dynamic testing of seat belt assemblies would be desirable to study this effect.

High level, short duration acceleration pulses will open push-button operated buckles, although the probability of such pulses occurring in accidents is unknown at this time.

Durability testing of seat belt buckle mechanisms should be of assistance in determining any ill effects on latching security of wear of latching surfaces, distortion of components or deformation or breakage of latch springs through fatigue.

Where a buckle cover is relied on to keep latch components in place, the cover should be secured by reliable means to prevent its being detached, especially in a crash.

It is considered that AS E35 should be amended to include a dynamic test on buckles (as distinct from dynamic test on seat belt assemblies) to verify their immunity from premature inertial opening. A 30 millisecond test pulse of 40g is proposed as a starting point.

In summary, designers of seat belt buckles not only have to consider wear and quality control problems, but also the effects of inertial loads. All of these effects may also interact, and this possibility must be taken into account.

APPENDIX A: REPRESENTATIVE MAKES OF BUCKLE, LISTED ACCORDING TO TYPE OF LATCH MECHANISM

Positive Blocking Type Latch

Rainsford stalk type (with retractor)

Early Britax Lyfelok

Compressive Latching Member Type

Bodigard

Dominion

Karinna

Star

Volvo fixed type (with retractor)

Tensioned Latching Member Type

Britax model BV

Cooldrive

Special Type

Tudor hook-type with magnetic latch

APPENDIX B: EQUIPMENT USED FOR DYNAMIC TESTING OF BUCKLES

A. Crash Simulator Tests

1) Horizontal Crash Simulator, Monterey model MRL6500

2) Accelerometer: C.E.C. (Bell & Howell) type 4-203-0001 unbonded, temperature compensated strain gauge $\pm$ 100g

3) Preamplifier: Hewlett Packard, Data Amplifier type 2470A

4) Oscilloscope:

   (a) Main frame: Tektronix type 564B

   (b) Time base: Tektronix type 3B3

   (c) Vertical ampl. Tektronix type 3A74

B. High Level, Short Duration Pulses

1) Accelerometer: (as above, but $\pm$ 500g)

2) Preamplifier: (as above)

3) Oscilloscope: (as above)

4) Fork lift truck and masses

Buckle Specimen and Latch type	Belt Tension (N)	Acceleration Pulse		No. of Tests in which	
		Amplitude (g)	Duration (ms)	Buckle remained latched	Buckle released
MT 033 Magnetic lever latch	44.5	20	35	4	-
	44.5	30	35	4	-
	44.5	35	33	2	1
	44.5	40	35	-	4
	44.5	46	32	-	3
MT 034 Lever operated position blocking type latch	44.5	20	35	4	-
	44.5	30	35	4	-
	44.5	40	35	4	-
MT 035 Push-button operated tensioned latching member type latch	44.5	20	35	4	-
	44.5	30	35	4	-
	44.5	40	35	4	-
MT 036 Push-button operated compressive latching member type latch	44.5	20	35	4	-
	44.5	30	35	4	-
	44.5	40	35	2	3*
	44.5	46	32	1	4*
MT 039 *	44.5	40	35	3	-
	44.5	46	32	3	-
	222.4	40	35	2	-
	222.4	44	30	2	-
	333.6	40	35	3	-
	333.6	46	32	3	-
MT 037 Lever operated compressive latching member type latch	44.5	20	35	4	-
	44.5	30	35	4	-
	44.5	40	35	4	-

TABLE 1: (continued on the next page)

Buckle Specimen and Latch type	Belt Tension (N)	Acceleration Pulse Amplitude (g)	Duration (ms)	No. of Tests in which Buckle remained latched	Buckle released
MT038	44.5	20	35	4	-
Push-button operated	44.5	30	35	4	-
compressive latching member type latch	44.5	40	35	4	-

TABLE 1: Test results for different seat belt buckles subjected to acceleration forces on the Traffic Accident Research Unit's crash simulator.

* The relatively poor performance of MT 036 prompted a close examination of the buckle body. This revealed slight distortion of the body, apparently caused by the method used to clamp it to the mounting block. A new example of this model of buckle (MT 039) was then tested after an improved, non-distorting method of clamping had been devised.

The test results of MT 036 should be disregarded (except insofar as they reflect an effect of buckle distortion) in favour of the test results of MT 039.

Buckle Specimen and Latch Type	Acceleration Pulse		Figure	Buckle opened
	Peak Value (g)	Peak Duration (ms)		
MT 035 Push-button operated tensioned latching member type latch	156	0.32	10(a)	Yes
	232	0.48	10(b)	Yes
	160	0.36	10(c)	Yes
	56	0.36	10(d)	No
MT 038 Push-button operated compressive latching member type latch	110	0.40	10(e)	No
	168	0.40	10(f)	Yes
	96	0.20	10(g)	Yes
	132	0.34	10(h)	Yes
	132	0.44	10(i)	No
MT 039 Push-button operated compressive latching member type latch	144	0.36	10(j)	Yes
	132	0.30	10(k)	No
	208	0.40	10(l)	No
	200	0.70	10(m)	Yes
	208	0.56	10(n)	Yes
	104	0.36	10(o)	No

TABLE 2: Test results for different push-button operated seat belt buckles subjected to high level, short duration acceleration pulses. In all cases, the seat belt was subjected to a tension of 334N.

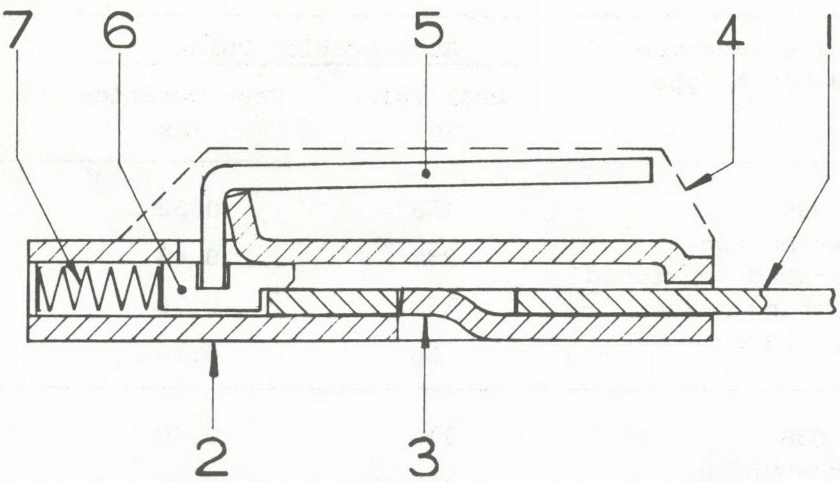
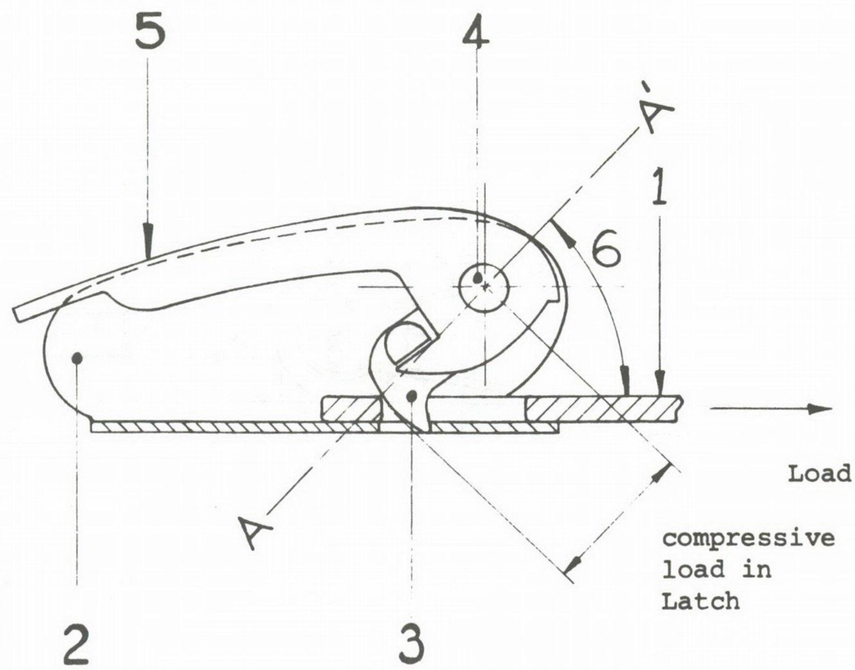


Figure 1: Positive Blocking Type Buckle Latch

1. Tongue
2. Buckle frame
3. Projection on buckle base
4. Buckle cover
5. Release button
6. Blocking bar
7. Latch spring



<u>Figure 2:</u> Compressive Latching Member Type Buckle Latch

1. Tongue
2. Buckle frame
3. Latch (biassed by moustrap type spring)
4. Pivot pin
5. Release lever (biassed by latch spring)
6. Latch thrust angle

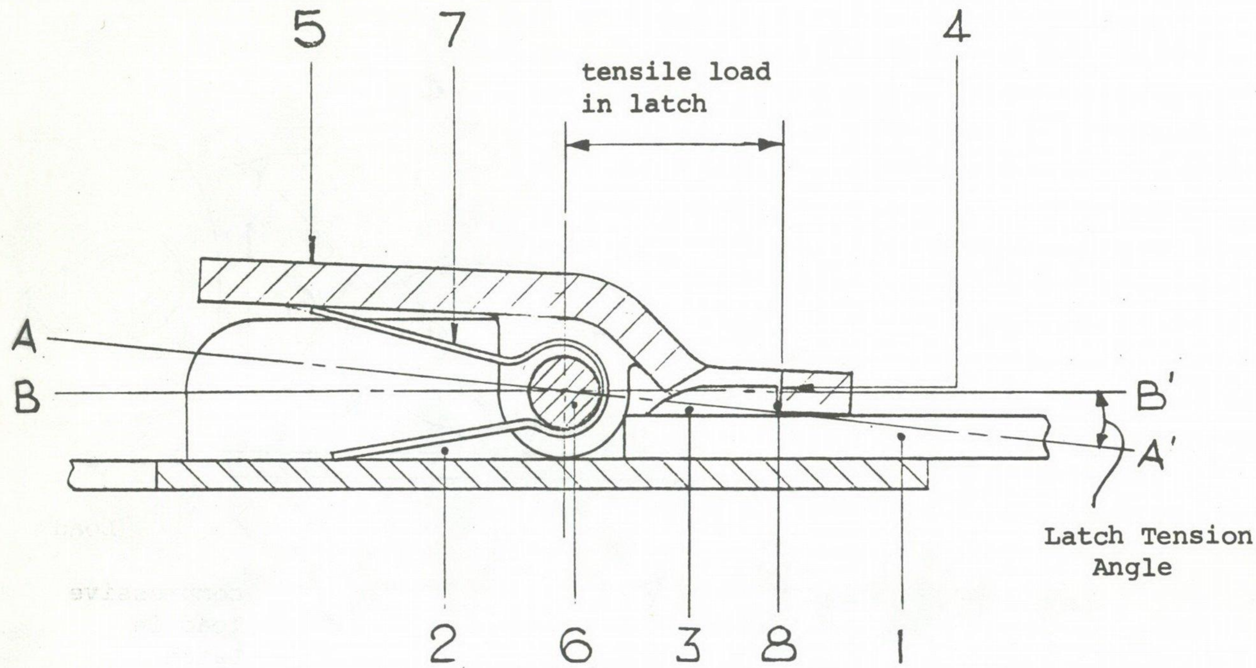


Figure 3: Tensioned Latching Member Type Buckle Latch

1. Tongue
2. Buckle frame
3. Projection on tongue
4. Latching surface of latch
5. Latch
6. Pivot pin
7. Latch spring
8. Section through line of contact of latch

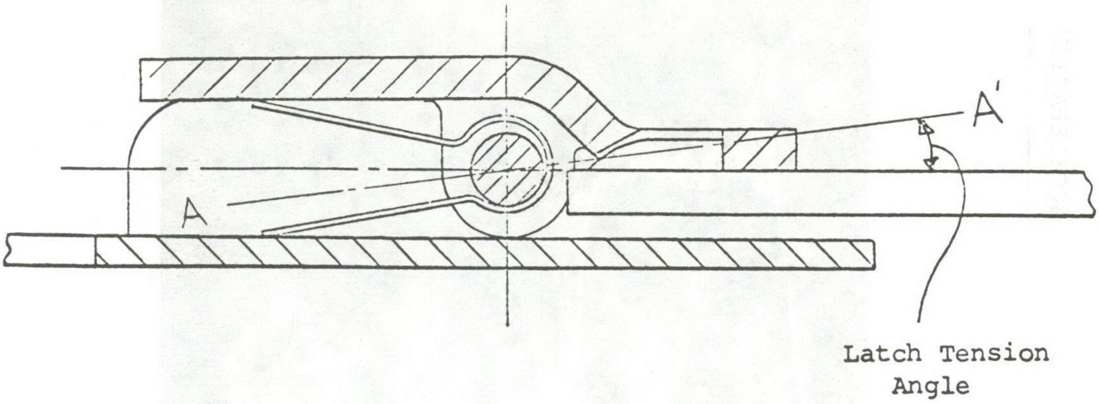
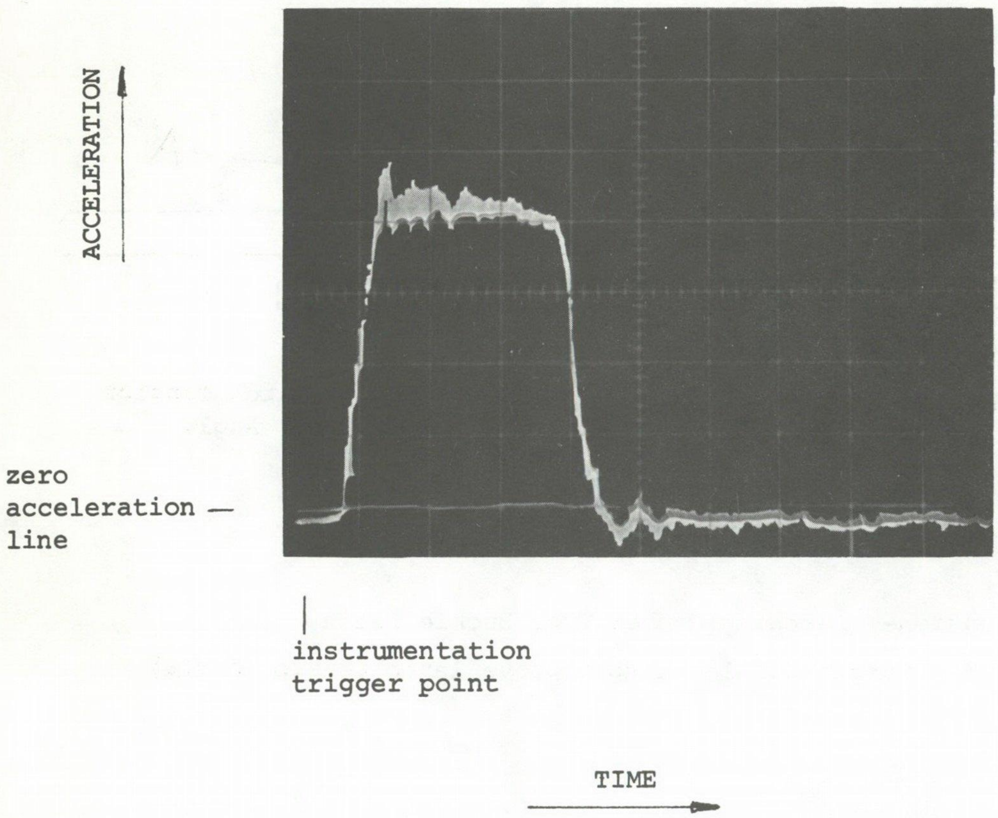
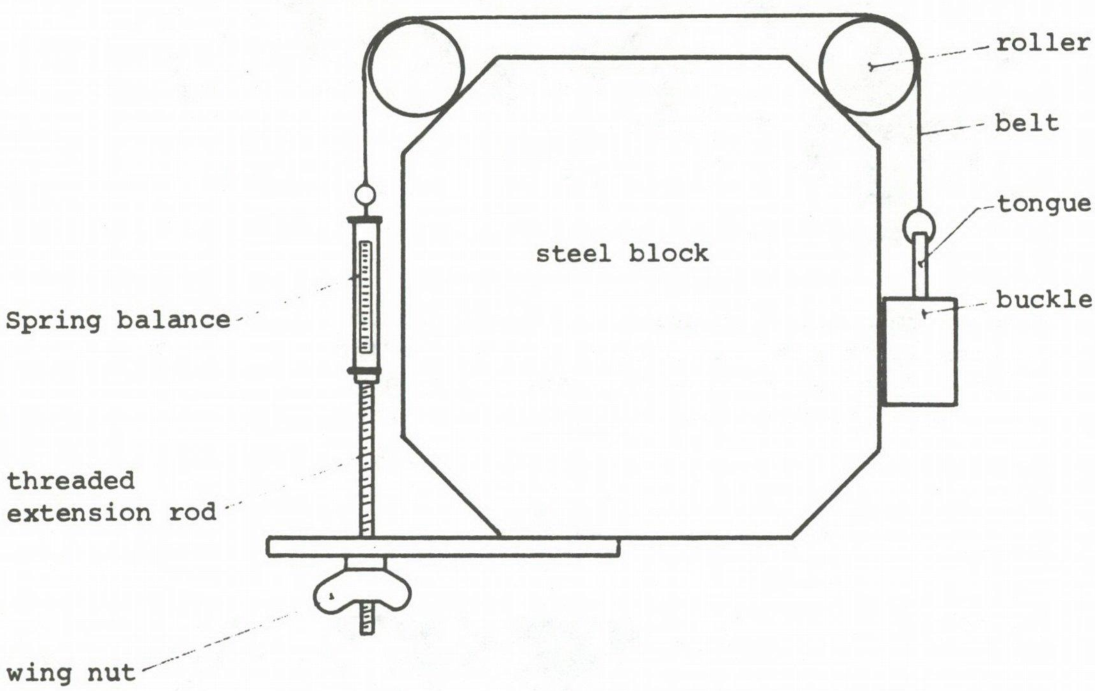


Figure 4: Tensioned Latching Member Type Buckle Latch
(with tongue displaced to improve latch tension angle)



<u>Figure 5</u>: Typical square wave acceleration pulse obtained during the dynamic testing of buckles on the TARU sled.
Time scale, 10mS per division
Acceleration scale, 8.92G per division



Press-button buckles	Lever buckles
←	→
Direction of initial travel of sled	Direction of initial travel of sled
→	←
Deceleration force on sled	Deceleration force on sled

Figure 6: Schematic plan of test rig used for dynamic testing of seat belt buckles on sled subjected to car crash accelerations.

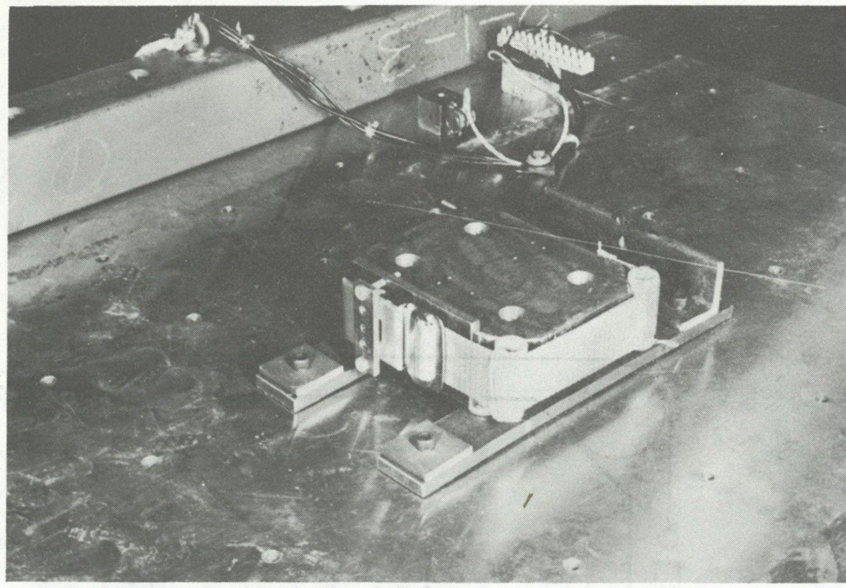


Figure 7: View of test rig used for dynamic testing of seat belt buckles (with push-button operated buckle fitted) on sled subjected to car crash accelerations.
(TARU Negative 008 - 10)

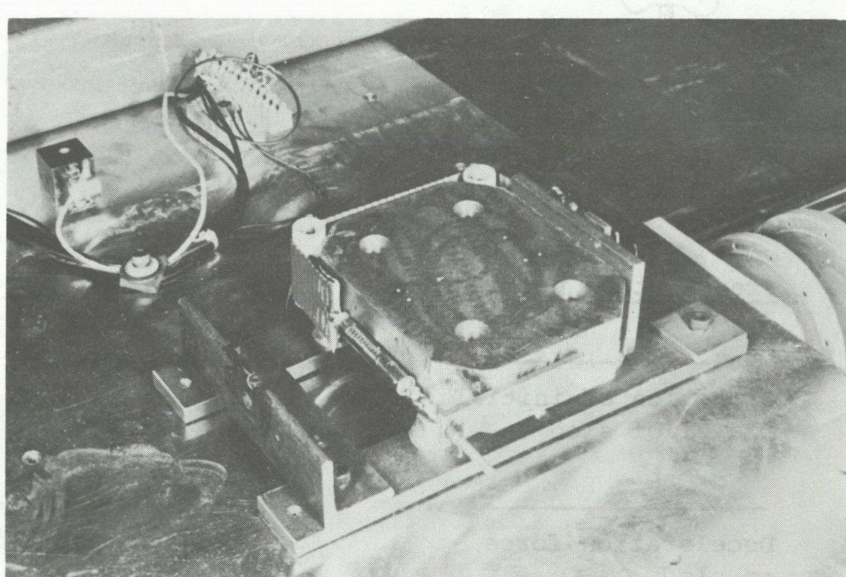
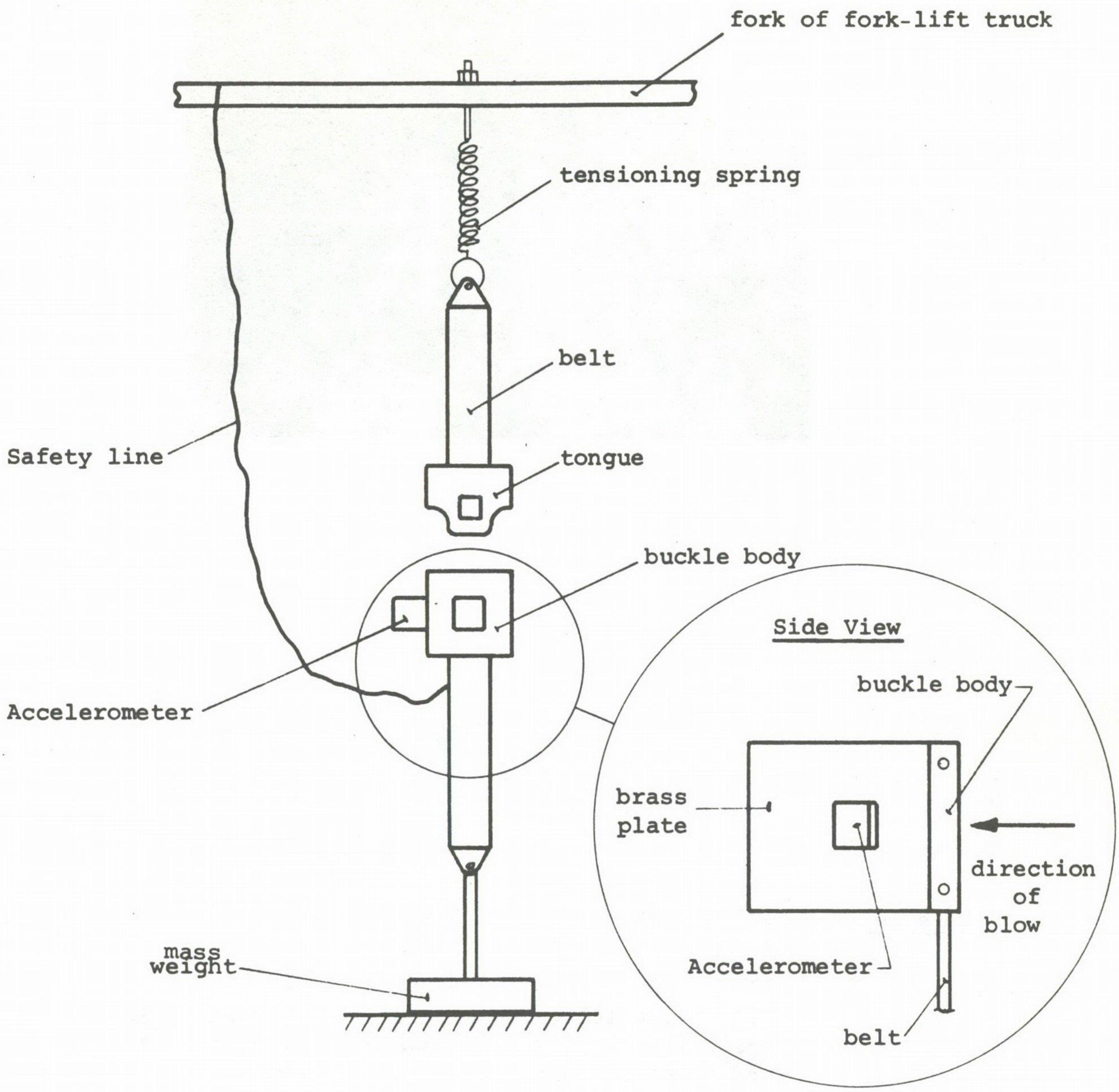


Figure 8: View of test rig used for dynamic testing of seat belt buckles (with lever operated buckle fitted) on sled subjected to car crash accelerations.
(TARU Negative 008 - 14)



<u>Figure 9</u>: Schematic view of test rig used for dynamic testing of seat belt buckles subjected to high value, short duration impact pulses.

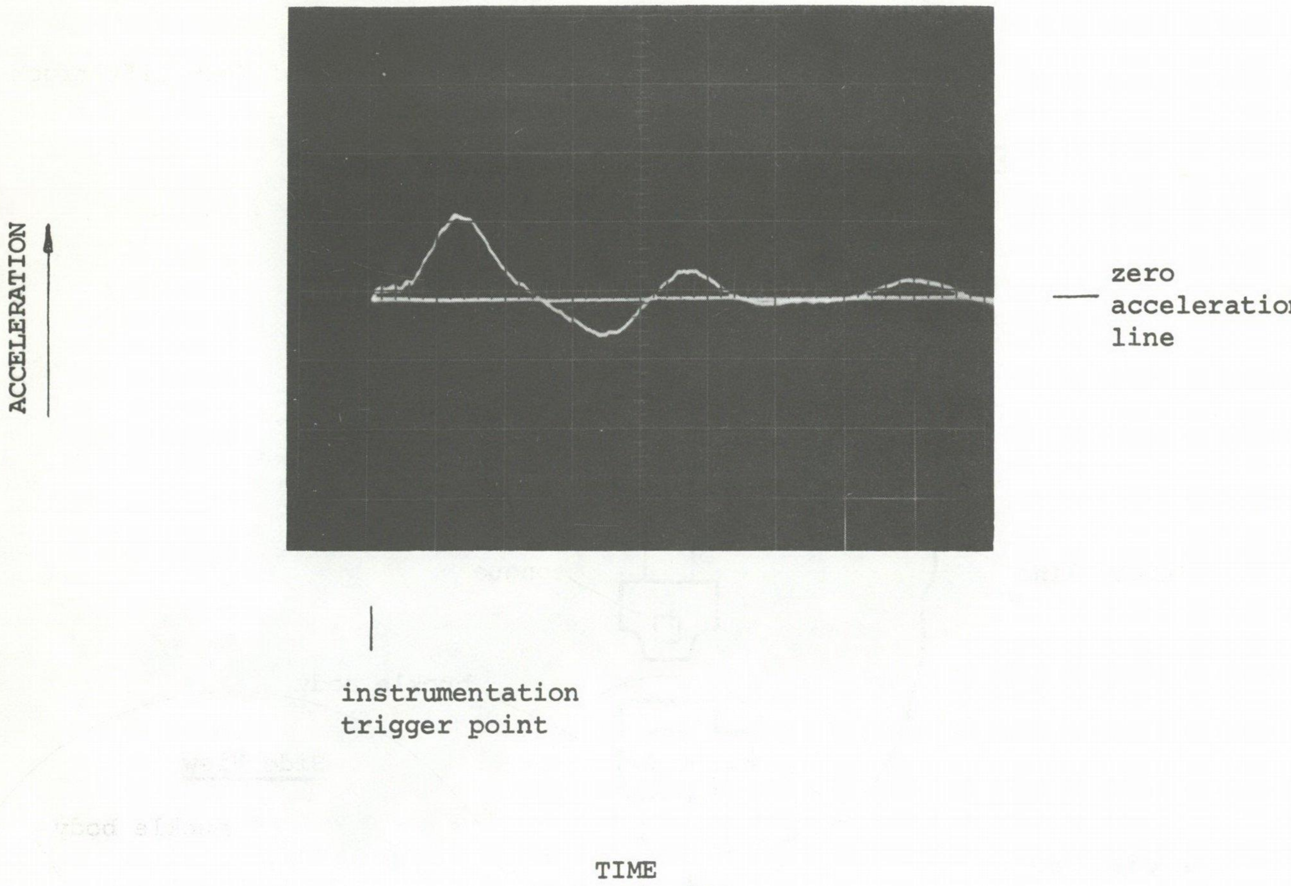


Figure 10(a): Pulse Peak of 156G, 0.32mS duration during inertial testing of buckles. Time scale, 0.2mS per division. Acceleration scale, 120G per division.

All of the following Figures 10(b) - 10(o), inclusive, are of the type shown above and all of the ordinates and abscissa represent time and acceleration, respectively.

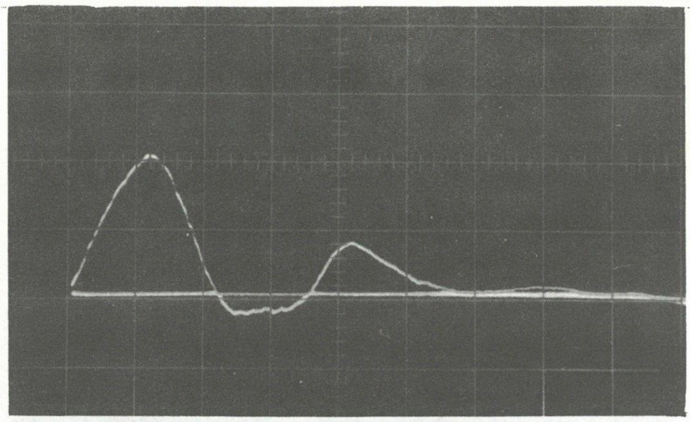


Figure 10(b)

232G, 0.48mS
0.2mS/div.
80G/div.

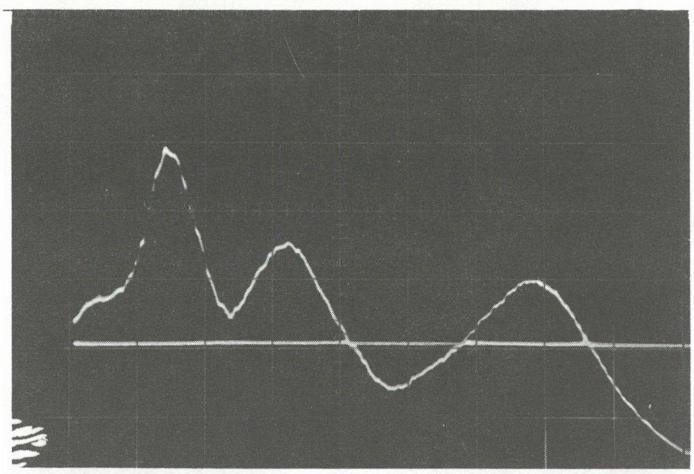


Figure 10(c)

160G, 0,38mS
0.2mS/div.
80G/div.

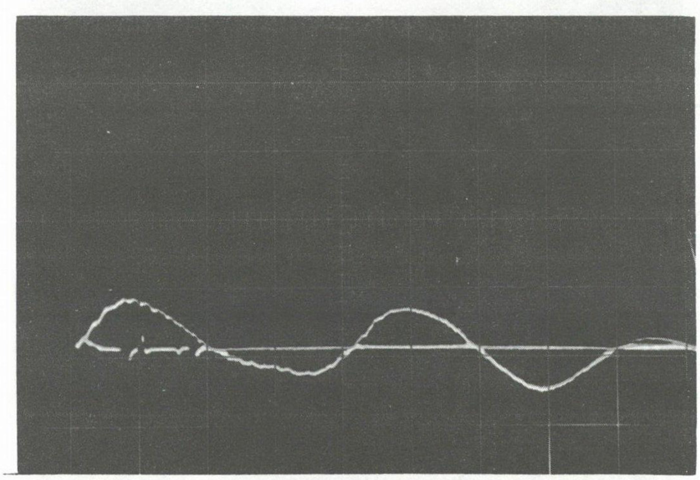


Figure 10(d)

56G, 0.36mS
0.2mS/div.
80G/div.

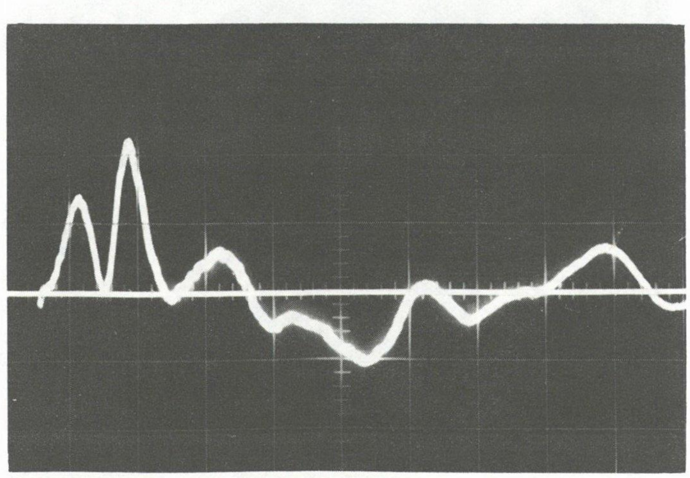


Figure 10(e)

110G, 0.40mS
0.5mS/div.
48G/div.

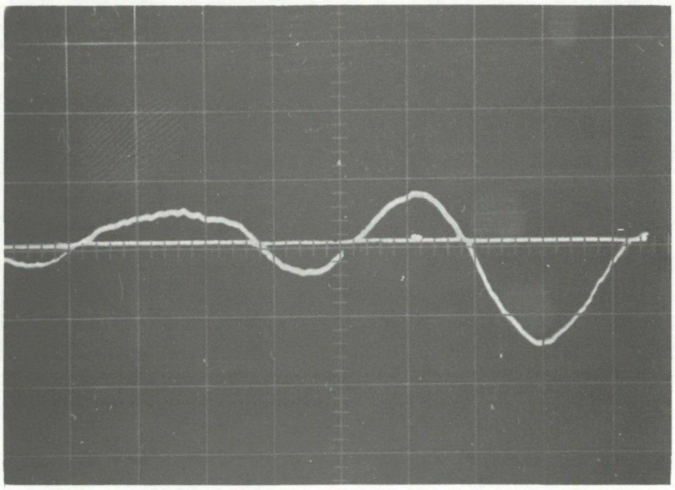


Figure 10(f)

168G, 0.40mS
0.2mS/div.
120G/div.

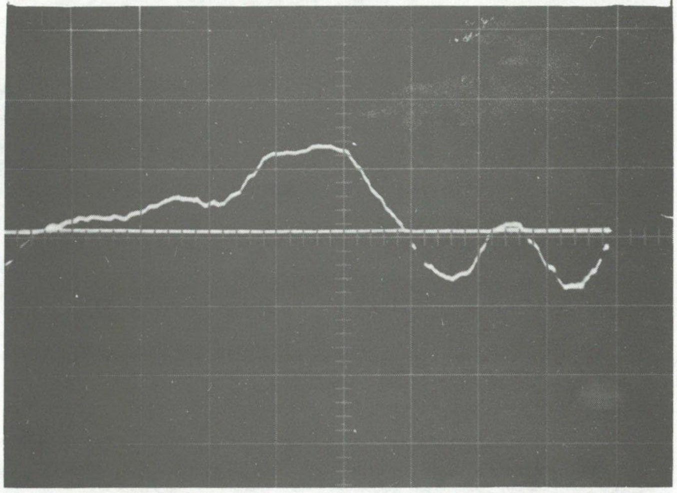


Figure 10(g)

96G, 0.20mS
0.2mS/div.
120G/div.

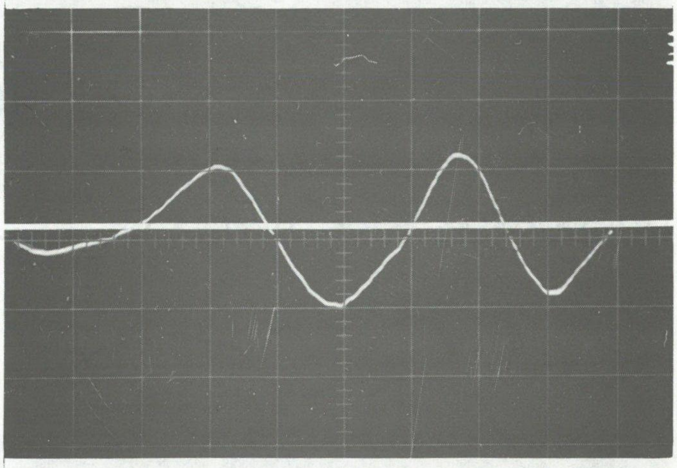


Figure 10(h)

132G, 0.34mS
0.2mS/div.
120G/div.

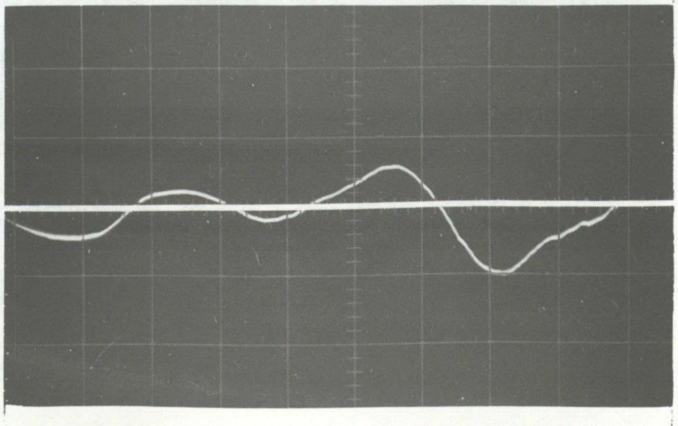


Figure 10(i)

132G, 0.44mS
0.2mS/div.
120G/div.

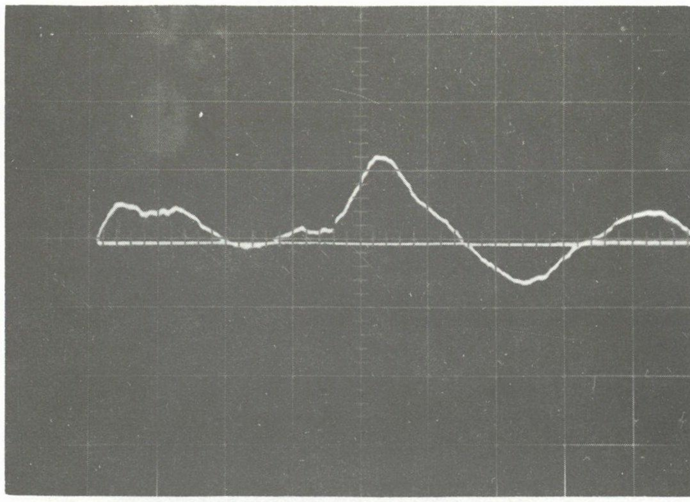


Figure 10(j)

144G, 0.36mS
0.2mS/div.
120G/div.

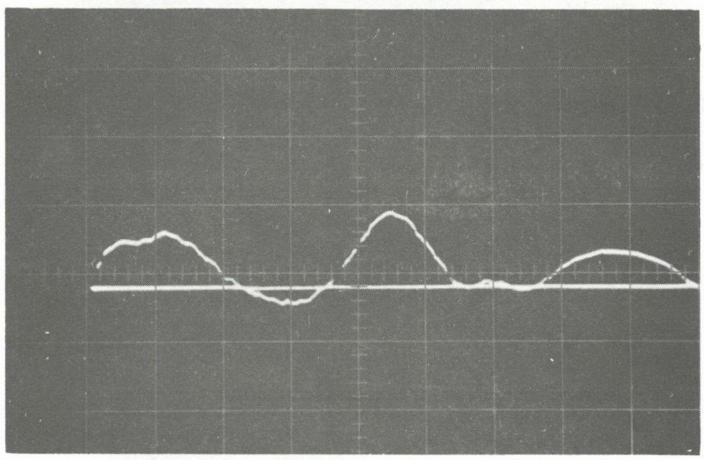


Figure 10(k)

132G, 0.30mS
0.2mS/div.
120G/div.

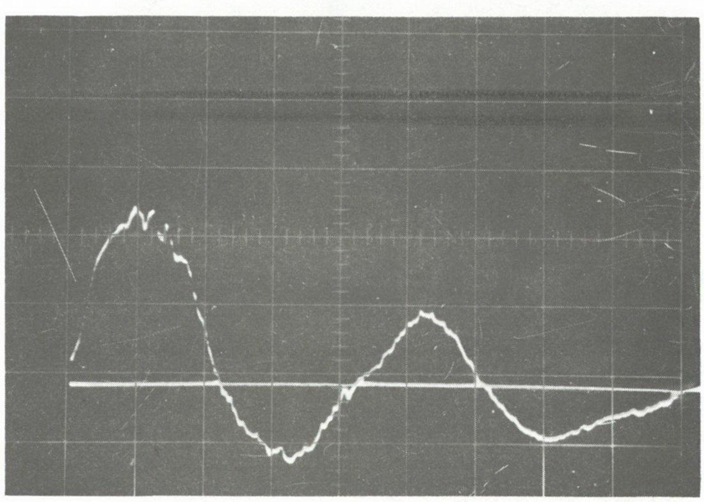


Figure 10(l)

208G, 0.40mS
0.2mS/div.
80G/div.

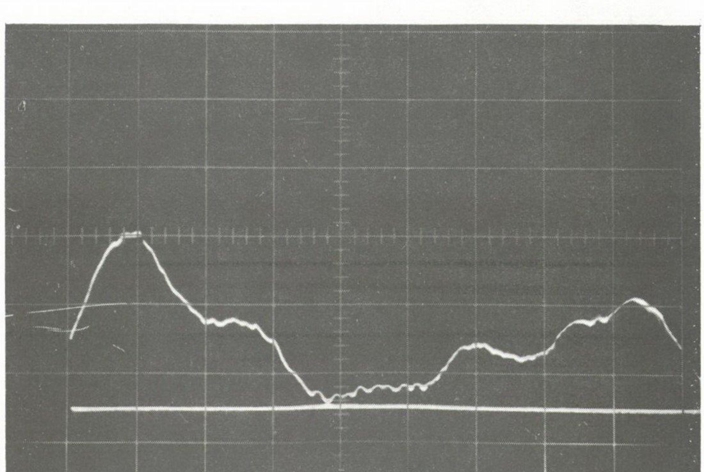


Figure 10(m)

200G, 0.70mS
0.2mS/div.
80G/div.

028495

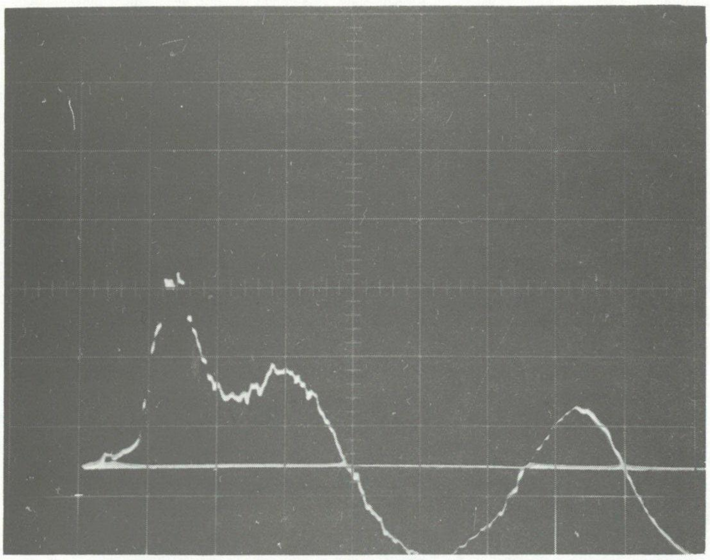


Figure 10(n)

208G, 0.56mS
0.2mS/div.
80G/div.

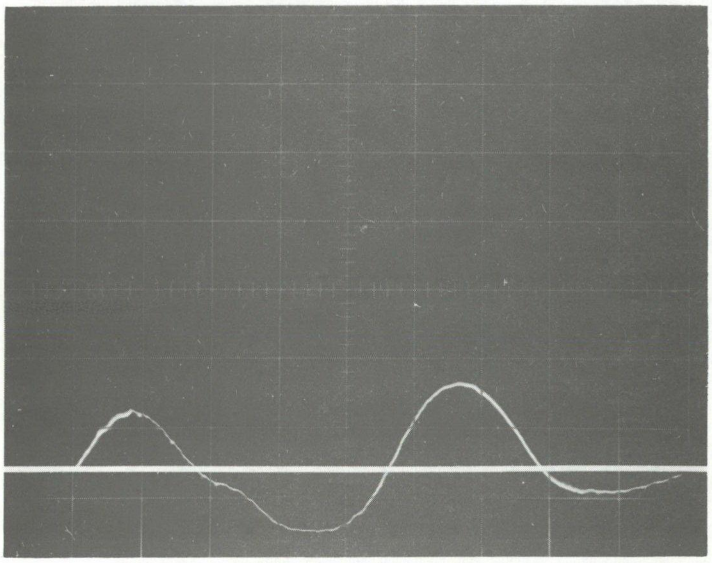


Figure 10(o)

104G, 0.36mS
0.2mS/div.
80G/div.