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**ABSTRACT**
In the Netherlands, more troop carrying helicopters are flying more "passenger-miles" in peace time operations than before. These helicopters were designed before crashworthiness was a severe design issue. NLR is evaluating if the crashworthiness of existing helicopters can be upgraded significantly by retrofitting crashworthy troop seats. Within this framework a generic research investigation was undertaken to design and fabricate (NLR), test and analyze (TNO) a generic troop seat, to obtain a baseline configuration and characteristic to be used for the assessment mentioned above. The test results indicated that the seat design was satisfactory, but the energy absorbing crash tube needs to be optimized, including the break initiation mechanism. The MADYMO crash simulation package predicted the test results well and has potential for optimization studies.

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This paper will be presented at the 23rd European Rotorcraft Forum, 16-18 September 1997, Dresden, Germany.

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Abstract

In the Netherlands, more troop carrying helicopters are flying more "passenger-miles" in peace time operations than before. These helicopters were designed before crashworthiness was a severe design issue. NLR is evaluating if the crashworthiness of existing helicopters can be upgraded significantly by retrofitting crashworthy troop seats. Within this framework a generic research investigation was undertaken to design and fabricate (NLR), test and analyze (TNO) a generic troop seat, to obtain a baseline configuration and characteristic to be used for the assessment mentioned above. The test results indicated that the seat design was satisfactory, but the energy absorbing crash tube needs to be optimized, including the break initiation mechanism. The MADYMO crash simulation package predicted the test results well and has potential for optimization studies.

## Symbols

| | |
|---|---|
| $a_a$ | Acceleration of airframe, $m/s^2$ |
| $a_s$ | Acceleration of seat/occupant, $m/s^2$ |
| $F$ | Force in crash tube, N |
| $F_{dyn}$ | Estimated force in crash tube during dynamic test, N |
| $F_{st}$ | Force required to crash the tube in a static test, N |
| $g$ | Acceleration due to gravity, 9.80665 $m/s^2$ |
| $G_m$ | Maximum deceleration magnitude of crash pulse, g units |
| $G_l$ | Maximum deceleration magnitude allowed for the seat/occupant combination, g units |
| $m$ | Mass of seat/occupant, kg |
| $s_a$ | Displacement of airframe, m |
| $s_s$ | Displacement of seat/occupant, m |
| $t$ | Time, sec |
| $t_f$ | Time until the seat/occupant comes to rest, sec |
| $t_l$ | Time to reach seat pan maximum deceleration, sec |
| $t_m$ | Time until maximum deceleration, sec |
| $t_r$ | Maximum pulse duration, sec |
| $v_0$ | Initial impact velocity of airframe, m/s |

## Introduction

Since the concept of crashworthiness was introduced for helicopters in the early seventies, design requirements have been formulated and structural solutions to satisfy these requirements have been developed. Modern helicopters, such as the NH90 and the Tigre are now designed from the beginning to be crashworthy. The design for crashworthiness requires a "systems approach" where several structural elements act together to reduce the inertia loads on the passengers below specified values. Major structural elements involved in the absorption of impact energy are the specially designed landing gear, the sub-floor structure and the seats. However, most helicopters operated today by the military forces in peace time operations were designed before crashworthiness was such a severe design issue. Quite a few of these helicopters are troop carrying transport helicopters, in which considerable numbers of "passenger-miles" are being "produced". The question then arises whether the crashworthiness of such helicopters could be enhanced by retrofitting structural elements for this purpose. Improving the landing gear or the sub-floor structure to current crashworthy standards is an unrealistic option. However, retrofitting crashworthy seats in older types of helicopters would be feasible, technically as well as economically. With this in mind a generic research investigation was undertaken, to design and fabricate (NLR), test and analyze (TNO) a crashworthy troop seat, with the long term objective to evaluate the option of retrofitting such seats in existing helicopter frames.

Firstly, the design requirements are described for crashworthy seats, and troop seats in particular, as formulated in international standards. Secondly, the design of the seat is described and relevant details of the seat are discussed. The energy absorption mechanism in the seat consists of a crushable composite tube, which was designed specifically for the test conditions considered. The seat was tested in the TNO crash laboratory with a dummy sitting on it. The test setup is described, including the instrumentation, followed by a description of several sled test results. The seat was placed in a horizontal position, to simulate a drop test. A computer model of the seat and dummy was created. This model was analysed and optimized with MADYMO, which is a specially designed software tool for crash safety analysis. Relevant MADYMO features and simulation results are described. A comparison is made between the test and simulation results, and conclusions are drawn.

## Design requirements

Design and testing requirements for helicopter crashworthy seats were formulated for both military (Ref. 1 and 2) and civilian (Ref. 3 and 4) aircraft. In the early seventies, design and test methodologies for military aircraft were developed under sponsorship of the U.S. Army (Ref. 5). However, studies indicated that significant differences exist in the crash environments of military and civilian helicopters (Ref. 6). If the military crash resistance design criteria were applied to the civil fleet, a severe weight and cost penalty would be imposed on civilian helicopters. Therefore, a research program was initiated by the FAA Technical Center to define design and test criteria which are more realistic for the civil rotorcraft crash environment (Ref. 6 and 7).

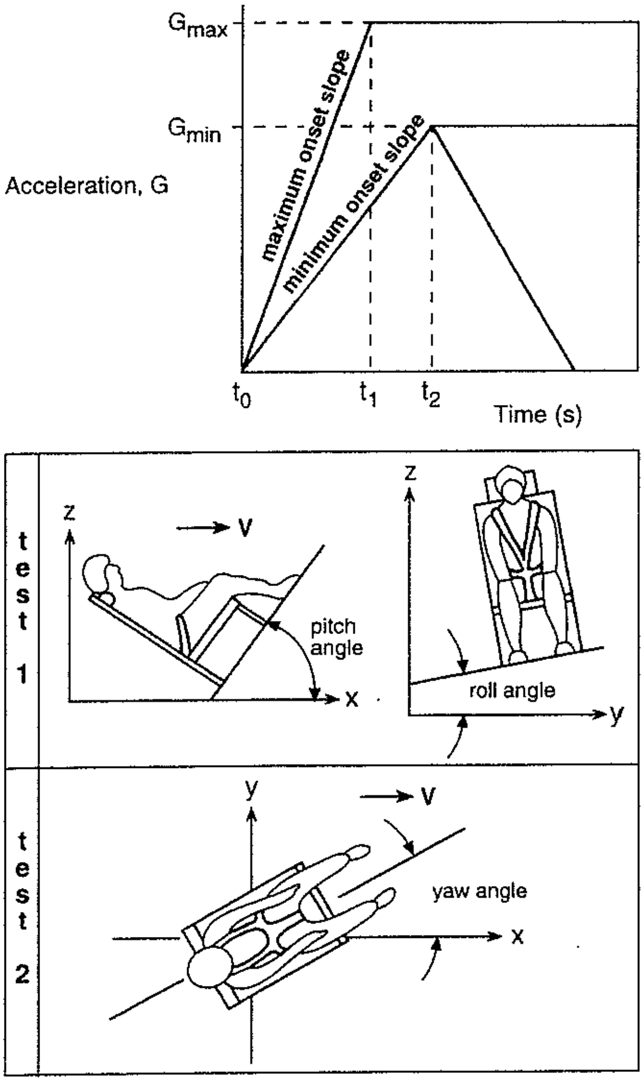



*Fig. 1 Dynamic tests for rotorcraft seats [Ref. 1]*

Both the military and civilian requirements define two dynamic tests for rotorcraft, as illustrated in figure 1, in which the first test represents a predominantly vertical impact and the second test a longitudinal impact with yaw angle. For both tests, the required impact velocity, peak accelerations and time to reach the peak are shown in table 1 for both military and civilian requirements. The design requirements for static inertial load factors are also identified in table 1.

For both military and civilian test requirements, floor or bulkhead deformation that may occur in an accident should be taken into account. For a floor-mounted seat, one of the seat tracks must be misaligned by 10 degrees in pitch and the other must be 10 degrees in roll. For a bulkhead-mounted seat, attachments should be distorted with a misalignment of 5 degrees in the plane of the bulkhead (see Fig. 2). Attachments of the seat and restraint system must remain intact. Each seat should be tested as a complete unit, including the occupant and occupant restraint. The occupant is an anthropomorphic dummy (ATD), which must include a provision for measurement of "pelvic force", the force that is transmitted to the dummy pelvis through the spinal column.

Regarding the seat attachment, the seats should be mounted in a suitable fixture by using the normal seat system to aircraft structure tie-downs. The fixture should be representative of the aircraft surrounding structure and spring rate to a degree which is economically feasible for test purposes. To prevent seat connection failures induced by fuselage distortion, structural joints should be capable of large angular displacements in all directions without failure.

For all troop seats, seat or mounting provisions should not interfere with rapid ingress of egress. Braces, legs, cables, straps and other structures should be designed to prevent snagging or tripping. Loops should not be formed when the restraint system is in the unbuckled position.

Troop seats should be so designed that they may be quickly removed or folded and secured. Tools should not be required for this operation. Each single occupant seat should be capable of being folded, stowed and secured or unstowed quickly and easily by one person in a period not to exceed 20 seconds. The mass of the complete single occupant troop seat, including the restraint, should not exceed 6.8 kg (15 lbs).

Table 1 Static and dynamic test requirements for crashworthy seats

| | Civil | | Military | |
|---|---|---|---|---|
| Test requirements of: | JAR/FAR Part 27 | JAR/FAR Part 29 | MIL-S-58095A(AV) | MIL-S-85510(AS) |
| Applicapable for: | Normal category rotorcraft | Transport category rotorcraft | cockpit seats | troop seats |
| Static tests | | | | |
| Forward | | | | |
| minimum load factor, G | 16 | 16 | 35 | 30 |
| Mass ATD[4], kg (lb) | 98 (216) | 98 (216) | 114 (250) | 110 (242.2) |
| Aftward | | | | |
| minimum load factor, G | | 1.5 | 12 | 12 |
| Mass ATD, kg (lb) | | 98 (216) | 114 (250) | 110 (242.2) |
| Lateral | | | | |
| minimum load factor, G | 8 | 8 | 20 | 20/23[1] |
| Mass ATD, kg (lb) | 98 (216) | 98 (216) | 114 (250) | 110 (242.2) |
| Downward | | | | |
| minimum load factor, G | 20 | 20 | 25 | 14.5 ± 1 |
| Mass ATD, kg (lb) | 75 (165) | 75 (165) | 91 (200) | 89 (196.6) |
| Upward | | | | |
| minimum load factor, G | 4 | 4 | 8 | 8 |
| Mass ATD, kg (lb) | 98 (216) | 98 (216) | 114 (250) | 110 (242.2) |
| Dynamic tests | | | | |
| Test 1 (vertical) | | | | |
| max. peak deceleration, G | | | 51 | 37 |
| min. peak deceleration, G | 30 | 30 | 46 | 32 |
| time to max. peak, sec | | | 0.043 | 0.059 (0.034)[2] |
| time to min. peak, sec | 0.031 | 0.031 | 0.061 | 0.087 |
| velocity, m/s (ft/s) | 9.14 (30) | 9.14 (30) | 15.2 (50) | 15.2 (50) |
| roll angle, degrees | 0 | 0 | 10 | 10 |
| pitch angle, degrees | 60 | 60 | 60 | 60 |
| yaw angle, degrees | 0 | 0 | 0 | 0 |
| Mass ATD, kg (lb) | 77 (170) | 77 (170) | 105 (230) | 89 (196.6) |
| percentile | 50 | 50 | 95 | 50 |
| minimum stroke, m (in) | | | 0.3048 (12)[3] | 0.3556 (14)[3] |
| limit load factor, G | 12 | 12 | 14.5 | 14.5 |
| Test 2 (longitudinal) | | | | |
| max. peak deceleration, G | | | 33 | 27 |
| min. peak deceleration, G | 18.4 | 18.4 | 28 | 22 |
| time to max. peak, sec | | | 0.066 | 0.081 |
| time to min. peak, sec | 0.071 | 0.071 | 0.100 | 0.127 |
| velocity, m/s (ft/s) | 12.8 (42) | 12.8 (42) | 15.2 (50) | 15.2 (50) |
| roll angle, degrees | 0 | 0 | 0 | 0 |
| pitch angle, degrees | 0 | 0 | 0 | 0 |
| yaw angle, degrees | 10 | 10 | 30 | 30 |
| Mass ATD, kg (lb) | 77 (170) | 77 (170) | 105 (230) | 110 (242.2) |

Notes:

1 20 for light fixed-wing, attack, and cargo helicopters.
  23 for utility and observation helicopters.

2 Requirement for research and development testing.

3 According to the Aircraft Crash Survival Design Guide:
  - Cockpit seats: 6 inch minimum.
  - Cabin seats: 12 inch minimum.

4 ATD = Anthropomorphic Dummy.

The troop seats should have an integral restraint system, with lap belt and shoulder harness for each seating position. The restraint should be comfortable, light in weight, and easy for the occupant to put on and remove even in the dark. Reduction in support of the occupant should not occur due to stroking of the energy absorbers or deformation of the seat. A restraint system consisting of a two-strap shoulder harness and a lab belt assembly is preferred.

The troop seat should contain an appropriate headrest assembly consisting of a padded structure or as a minimum, a fabric or netting configuration which should be designed to provide among others protection of the occupant from potential head/neck injury.

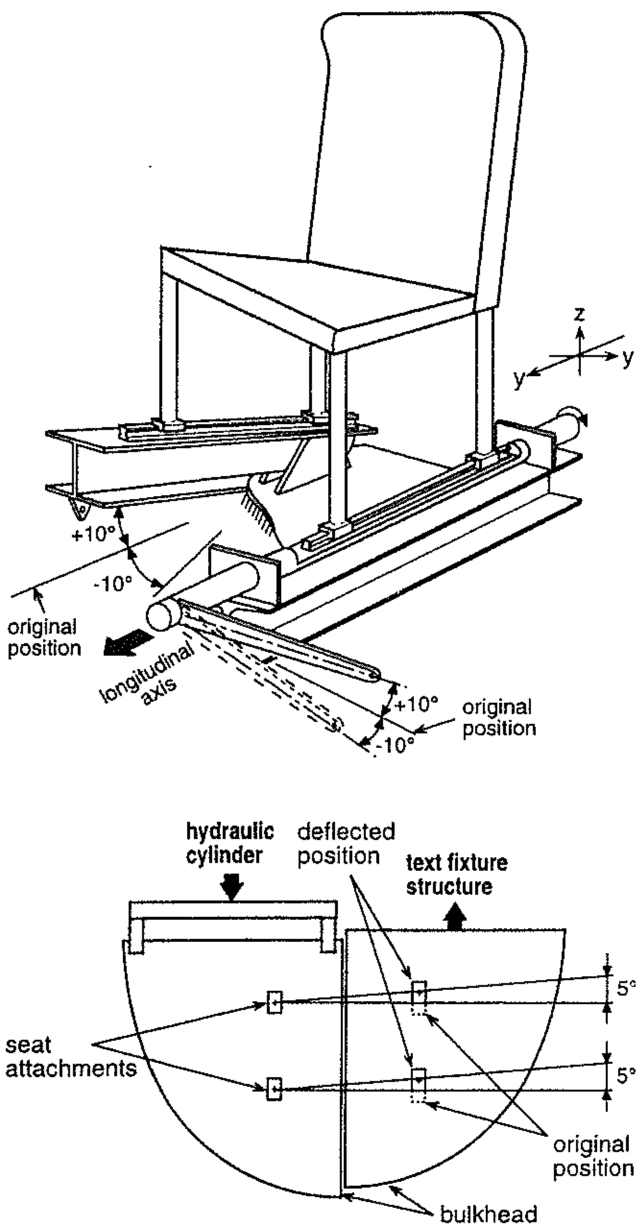



*Fig. 2 Floor or bulkhead warpage requirements [Ref. 1]*

Seat description

The troop seat as developed is shown in figure 3 and 4. The global lay-out is based on an existing troop seat of Simula (type number: P/N 104700-1, Ref. 8). The main differences between both seats can be found in the different materials used and the energy absorbing crash system as well as in the detailed design. The troop seat may be oriented to face to the side, forward of aft. It is designed to be mounted to the aircraft sidewall or bulkhead at 4 points without floor attachments.

The backside of the seat system is manufactured of carbon/epoxy. This backside is made of an H-shaped frame which moves vertically in one piece with the seat itself. The profiles used as vertical members are two I-shaped profiles, while the cross connection is formed by one U-shaped profile. The energy absorbing crash system is situated between this U-shaped profile and a second U-shaped profile which is connected between two fixed fuselage/cabin frame connection points.

The seatpan is made from aluminum tubing that supports a seat bottom and the seat back made of polyester fabric. The polyester fabric headrest is connected to the H-shaped seat frame. The backrest is quickly adjustable to accommodate an occupant with or without a backpack. There are no cables or struts beneath the seat, permitting the seat bottom to be rapidly stowed upright for easy cabin reconfiguration.

The seat is equipped with a 4-point safety-belt system which was made available by the RNLAF. In case of crash the harness will protect the occupant in an optimal way because it is attached directly to the seat bottom, reducing the change that "submarining" will occur.

Development of the energy absorbing crash tube

For the energy absorbing crash system a composite crash tube is used. This tube was developed and tested at NLR. The advantage of a composite crash tube compared to other energy absorbing systems is that the specific energy absorption is high.

To determine the force, F, required to crash the crash tube, the procedure described below is used.

Consider the schematic representation of the seat-airframe system in figure 5. The combined seat and occupant mass equals m. The force-displacement diagram of the crash-tube is idealized as a constant force over the entire stroke of the crash tube from 0 to 0.25 m.

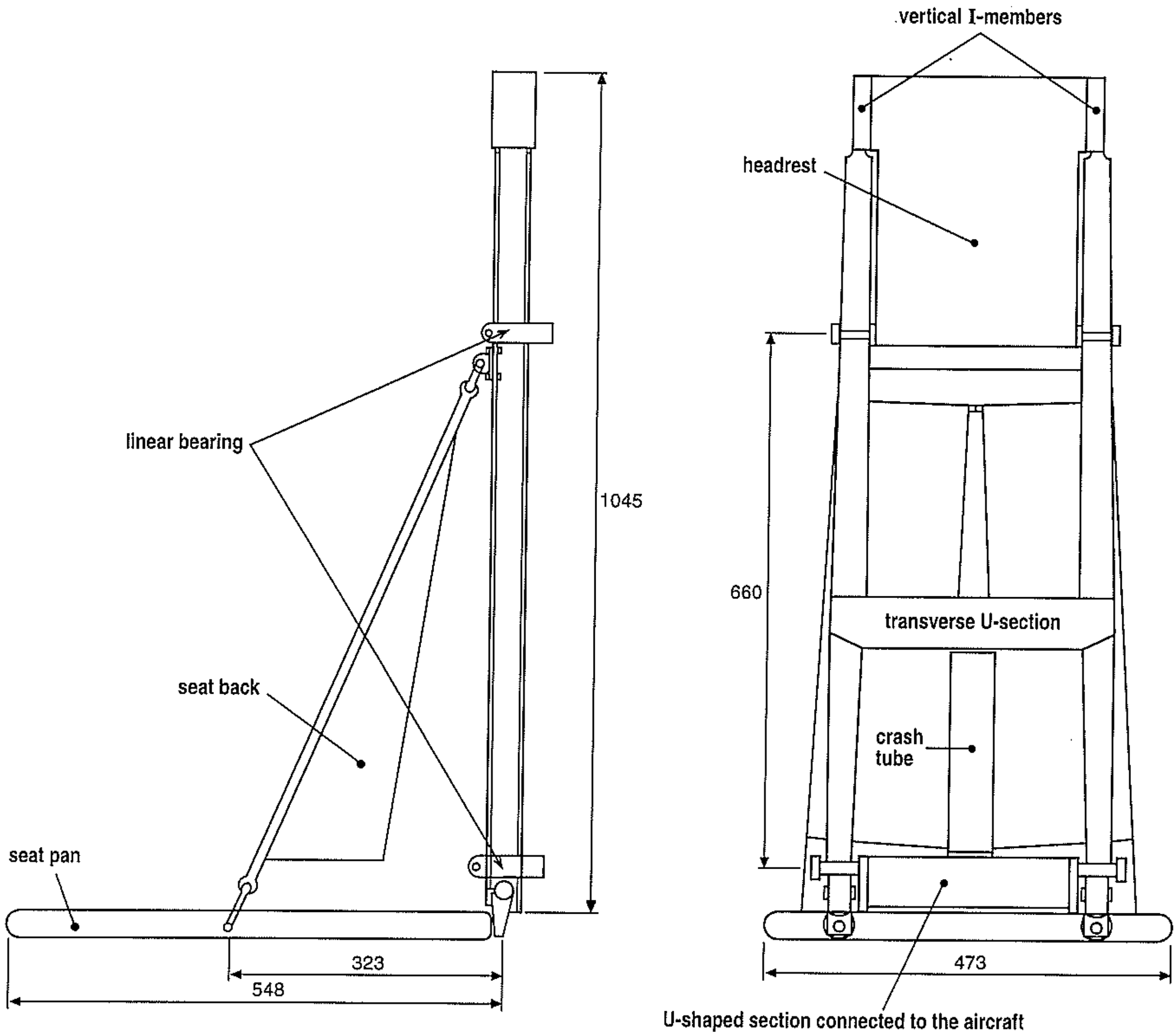



*Fig. 3 Troop seat outline*

In the dynamic requirements the acceleration on the airframe ($a_a$) is defined by a triangular pulse (see Fig. 1). The acceleration of the seat/occupant ($a_s$) is following the same slope until the threshold value of the break connections is reached and the acceleration is limited by the crash tube. So according to reference 5 the accelerations can be defined by

$$a_a = \begin{cases} -G_m g \dfrac{t}{t_m} & \text{for } 0 \le t \le t_m \\ -G_m g + G_m g \dfrac{(t-t_m)}{(t_r-t_m)} & \text{for } t_m \le t \le t_r \end{cases}$$

$$a_s = \begin{cases} -G_m g \dfrac{t}{t_m} & \text{for } 0 \le t \le t_l \\ -\dfrac{F}{m} & \text{for } t_l \le t \le t_f \, . \end{cases} \qquad (1)$$

$$\text{with } t_l = t_m \frac{G_l}{G_m} \, .$$

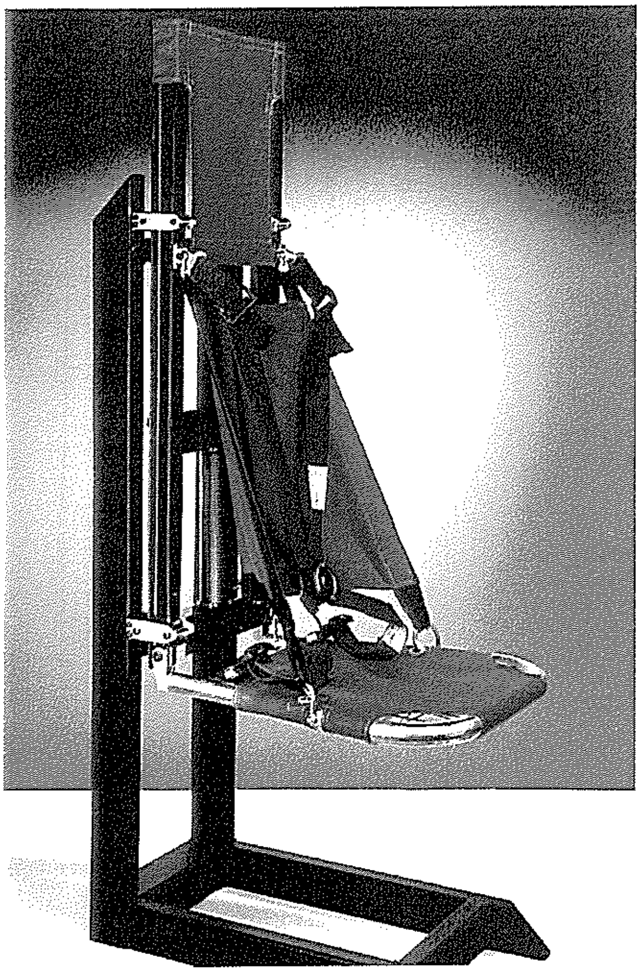


*Fig. 4 Picture of the troop seat*

When the difference between the displacement of the seat/occupant ($s_s$) and the displacement of the airframe ($s_a$) remains smaller than the maximum stroke of the crash tube (0.25 m), the acceleration of the seat/occupant is limited to the acceleration of (1), so

$$s_s - s_a \leq 0.25 \text{ m} \; . \qquad (2)$$

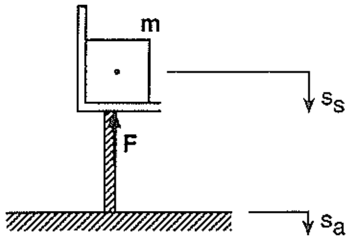



*Fig. 5 Schematic representation of the seat-airframe system*

A difference greater than 0.25 results in an increasing acceleration of the seat/occupant due to the velocity difference between the seat and airframe.

Twice integrating the accelerations and substitution of the time for which the velocity of the airframe ($t_r$) or seat/occupant ($t_f$) are zero gives the total displacements

$$v_s = 0 \quad \Rightarrow \quad t_f = \frac{G_m g m}{2F}\left[t_r - \frac{t_l^2}{t_m}\right] + t_l$$

$$s_a(t_r) = \frac{G_m g}{6}\left[t_r t_m + t_r^2\right]$$

$$s_s(t_f) = \frac{G_m g}{6 t_m}\left[(3 t_r t_m t_f - 3 t_f t_l^2 + 2 t_l^3)\right] - \frac{F}{2m}(t_f - t_l)^2 . \qquad (3)$$

Substitution of (3) in (2) gives two linear equations with two unknowns ($t_f$ and F) from which the resulting expression of the crash tube force, required to limit the seat acceleration to $G_l$ can be determined:

Table 2 Crash tube configuration

| Requirement | Tube id. | $G_m$ | $t_m$ [sec] | $v_o$ [m/s] | Mass ATD [kg] | $F_{req}$ (5) [kN] | Lay-up | Test result: $F_{st}$ [kN] | $F_{dyn}$ (0.85*$F_{st}$) [kN] |
|---|---|---|---|---|---|---|---|---|---|
| JAR 27 | 935 | 26 | 0.031 | 7.9 | 89 | 5.8 | $45_{fa}, 90_{UD}, 45_{fa}$ | 6.7 | 5.7 |
| - | 938 | 27 | 0.059 | 10.5 | 89 | 7.2 | $45_{fa}, 0_{fa}, 45_{fa}$ | N.T. | N.T. |
| MIL 85510 | 940 | 28 | 0.085 | 13.0 | 89 | 8.8 | $45_{fa}, 0_{UD}, 90_{UD}, 45_{fa}$ | 22.0 | 18.7 |

N.T. = Not Tested

$$F = \frac{\frac{m}{2}\left[\frac{G_m g(t_r - t_l^2)}{2t_m}\right]^2}{0.25 + \frac{G_m g}{6}\left[t_m t_r + t_r^2 - 3t_r t_l + \frac{t_l^3}{t_m}\right]} . \qquad (4)$$

This force is the idealized force in the crash tube during the dynamic tests and is assumed to be 85 % of the average force during crashing as observed in a static compression test of the crash tube. Based on NLR experience, crash tube configurations were defined and are presented in table 2.

Experimental set-up

Two sled tests have been carried out on TNO's sled facility in order to test the energy absorbing capabilities of the troop seat during a vertical crash.
The sled is propelled by rubber bands. The deceleration of the sled is controlled by pushing two cones into two rubber tubes. The seat was mounted with its vertical axis placed horizontally on the sled to simulate a drop test with the impact forces in the required direction. The pitch angle was 90 degrees and the roll angle 0 degrees. The total mass of the sled and the frame was 500 kg. A Hybrid II 95th percentile dummy was placed on the seat to represent a typical person with some luggage. A Hybrid II was used instead of the more advanced Hybrid III, because a 95th percentile Hybrid III dummy was not available. The total mass of the dummy and the moveable part of the seat was 110 kg. The dummy was restrained by a four-point belt-system. A support was mounted on the seat frame to support the lower legs simulating a realistic seat position.

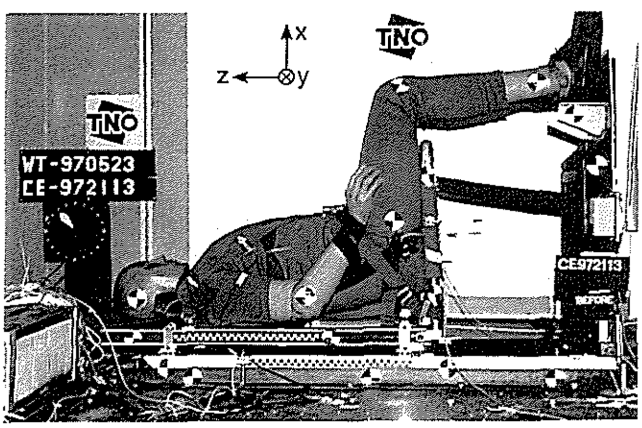



*Fig. 6 The experimental set-up*

Apart from high speed video, the following data was recorded during the tests:
- Sled and seat accelerations (-Z).
- Head, chest and pelvis accelerations (X, -Y, -Z).
- Shoulder and seat base belt forces.
- Front and rear seat base displacement (Z).

Figure 6 shows the experimental set-up and the orientation of the axes. The signals were measured according to ISO 6487, 1987. The acceleration transducers are accurate within 5 %.
The first test was carried out with a calibrated JAR 27 pulse and a crash tube (L=270 mm, Ø=50 mm, t=1 mm) with a relatively high crush force, corresponding to a specific fibre lay-up. This tube was actually designed for a crash test with a more severe pulse, but was used for this test to limit the severity.

The results of the first test were used to determine the second test set-up. This test was carried out with the same JAR 27 pulse, but with a crash tube with a different fibre lay-up, resulting in a lower crush force.

MADYMO models

Numerical tools allow the designer to study more alternative concepts, reduce the number of tests for design validation and thus reduce the costs and risks of the complete design cycle. The crash simulation package MADYMO (MAthematical DYnamic MOdels) combines multi-body and finite element techniques and advanced models for restraint systems, e.g. seatbelts and airbags, in one code. This allows the user to create models with the desired level of detail for the different design phases. An effective use of computer models in the field of occupant safety simulations requires that well-validated models for crash dummies are available. MADYMO includes models for most child, frontal and side impact adult dummies. Figure 7 shows the MADYMO3D structure (Ref. 9 and 10).

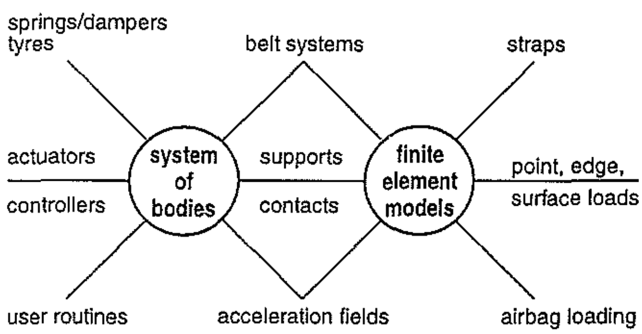



*Fig. 7 The MADYMO3D structure*

1. Sled test model with composite crash cylinder. The dummy has been modelled with the standard 95th percentile Hybrid III dummy, because no validated computer model is available yet for a 95th percentile Hybrid II dummy, which was used in the tests. The dummy model consists of a complex linkage system of 32 rigid bodies. The bodies are interconnected by kinematic joints including revolute joints, translational joints, universal joints and spherical joints. For every joint a dynamic restraint model that defines elastic, damping and friction loads depending on the relative motion in the joint is specified. The external geometry is represented by 51 (hyper)ellipsoids that allows the modelling of contact interactions.
The sled and rigidly connected seat frame are represented by one body. The geometry of that body is modelled by 9 hyperellipsoids and 3 planes. It is fixed to the inertial space.
The seat back and vertical I-members are represented by a second body with 2 hyperellipsoids and 5 planes attached to it. This body is connected to the seat frame by a translational joint. The joint is locked at the start of the simulation and unlocked at the moment that the vertical force in the joint exceeds 10 kN. This force level corresponds to the estimated threshold for the two break connections.
A third body with 2 hyperellipsoids represents the seat pan. It is connected to the seat back body by a revolute joint. Two belts are used to represent the right and left support straps. A non-linear stiffness with hysteresis has been defined for these belts.
The feet support is modelled by a fourth body, to which 2 hyperellipsoids are attached. The feet support is connected by a revolute joint to the seat frame body. A non-linear stiffness with hysteresis has been defined for this joint in order to model the plastic bending of the feet support plate. Contact between the dummy and the seat surfaces has been defined using non-linear contact stiffnesses, damping and friction.
The 4-point safety-belt system is represented by 2 belt systems of 6 segments each. The first belt is connected to the left hand side of the seat back, led over the left clavicle, the ribs and the abdomen and finally connected to the right hand side of the seat pan. The other belt connects in the same way the right hand side of the seat back with the left hand side of the seat pan. The belt model allows slip between two adjacent belt segments until a force equilibrium is reached in the harness system. For all the 12 belt segments a identical non-linear stiffness with hysteresis and some initial slack has been specified.
Finally, the composite crash cylinder is modelled by a finite element model consisting of 1440 quadrilateral shell elements. The length and diameter of the tube are 270 mm and 50 mm respectively. A uniform shell thickness of 1 mm is defined except for the shell elements at the location of the damage initiator, which have a thickness of 0.95 mm. An elasto-plastic material model with damage is used, which proved suitable to characterise the crushing behaviour of composites in earlier studies (Ref. 11).

The material parameters for the tube are chosen as follows:

| | |
|---|---|
| Young's modulus | 4000 MPa |
| Poisson ratio | 0.05 |
| Yield Stress | 285 MPa |
| Damage parameters | p1 = 0.2, p2 = 0.0, p3 = 4.0 |
| Density | 1300 kg/m$^3$. |

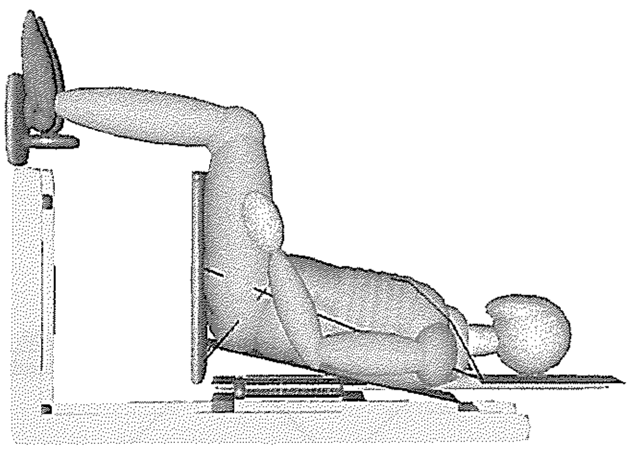


*Fig. 8 MADYMO3D model kinematics with FEM crash tube at t = 0.0 (s). (One hyperellipsoid is not shown to get a clear view at the FEM tube)*

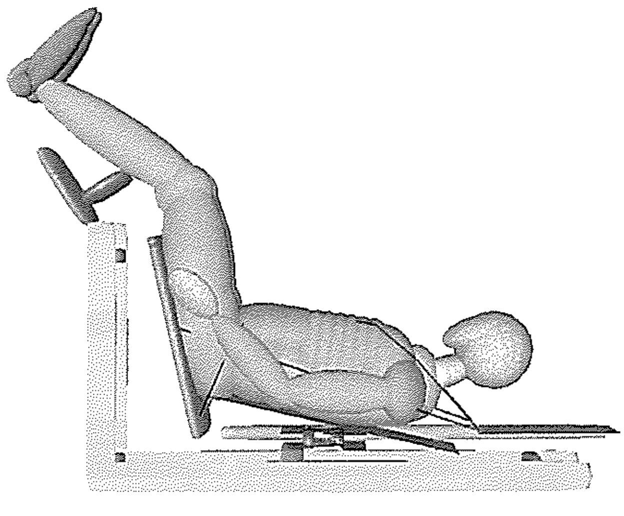


*Fig. 9 MADYMO3D model kinematics with FEM crash tube at t = 0.1 (s). (One hyperellipsoid is not shown to get a clear view at the FEM tube)*

The elastic behaviour of the tube has no significant effect on the crushing behaviour as long as the collapse mode is not effected. Therefore in order to reduce CPU-time, a much lower value than the statically measured value (124 GPa) for the Young's modulus has been used.
The lowest row of nodes at the lower end of the cylinder is in contact with a plane fixed to the seat frame. A friction coefficient of 0.7 is used for this contact. The upper end of the cylinder is initially almost in contact with a plane connected to the back seat, which is moving downward during the crash. Contact interactions between the nodal points and this plane have been defined.
Figure 8 and 9 show the complete MADYMO model in the initial state and after 0.1 seconds, respectively. By comparing these, the rotation of the seat pan and the bending of the feet support as well as the reducing length of the crash cylinder is obvious.

A main parameter is the motion of the seat relative to the frame. Therefore, instead of defining the acceleration of the seat frame equal to the deceleration pulse, the seat frame deceleration pulse is used as a fictitious acceleration field on the dummy, seat pan and seat back. The seat frame is fixed to the inertial space. The initial velocity of all bodies is zero. The accelerations of the dummy, seat pan and seat back are corrected afterward for this acceleration field. This reversed acceleration approach is allowed because the dummy rotations remain small. Accelerations due to gravity are taken into account by defining also an acceleration field (with a constant value of -9.81 $m/s^2$) in the X-direction.

2. Sled test model with non-linear spring. The finite element model of the composite crash cylinder in the previous model is replaced by a non-linear spring between the seat back and the seat frame. The force - relative-elongation characteristic of the spring is based on the computed force of the FEM tube in the previous model. This force is shown in figure 10 after using a CAE60 filter.

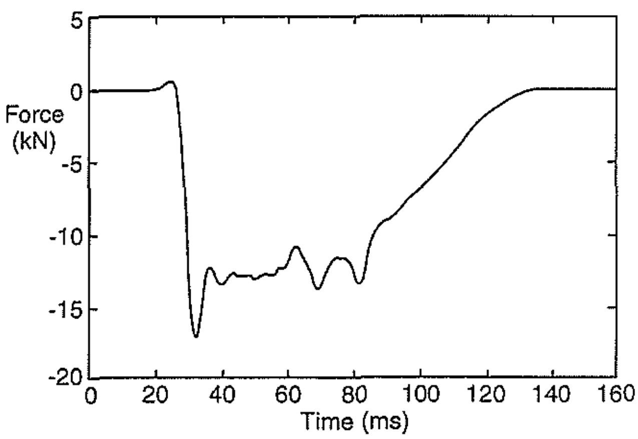



*Fig. 10 Tube crash force*

The rather high peak force is required for the initiation of the crushing process. During the actual crushing the force level is fairly constant. The stylized characteristic of the spring has a initial triangular peak followed by a constant force level.
The main advantage of this non-linear spring model is the CPU time. The new model can be analysed within a few minutes instead of a few hours, allowing optimization and sensitivity studies.

Test and simulation results

Four calibration tests were carried out to find the most suitable approximation, using standard crash pipes, of a JAR 27 crash pulse. Two sled test were carried out with this crash pulse, but with different tubes.

Sled test 1: The first test was carried out with tube 938. The length of the crushed part of the crash tube, measured statically after the test, was 17 cm. Approximately 80 % of the circumference of the tube had crushed, see figure 11. This means, that the tube has moved slightly in lateral direction. The crushing plane had contact with only a part of the tube.

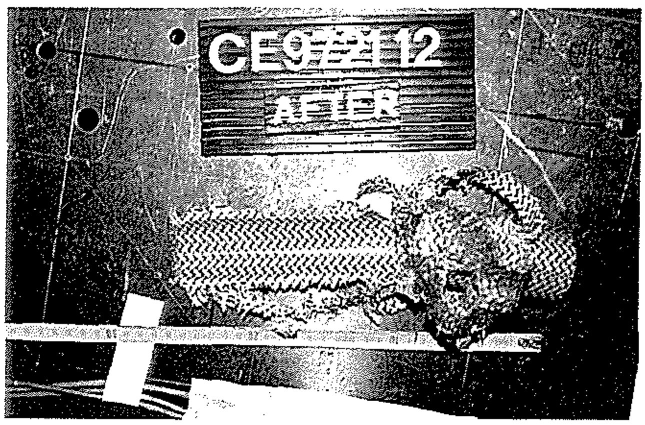
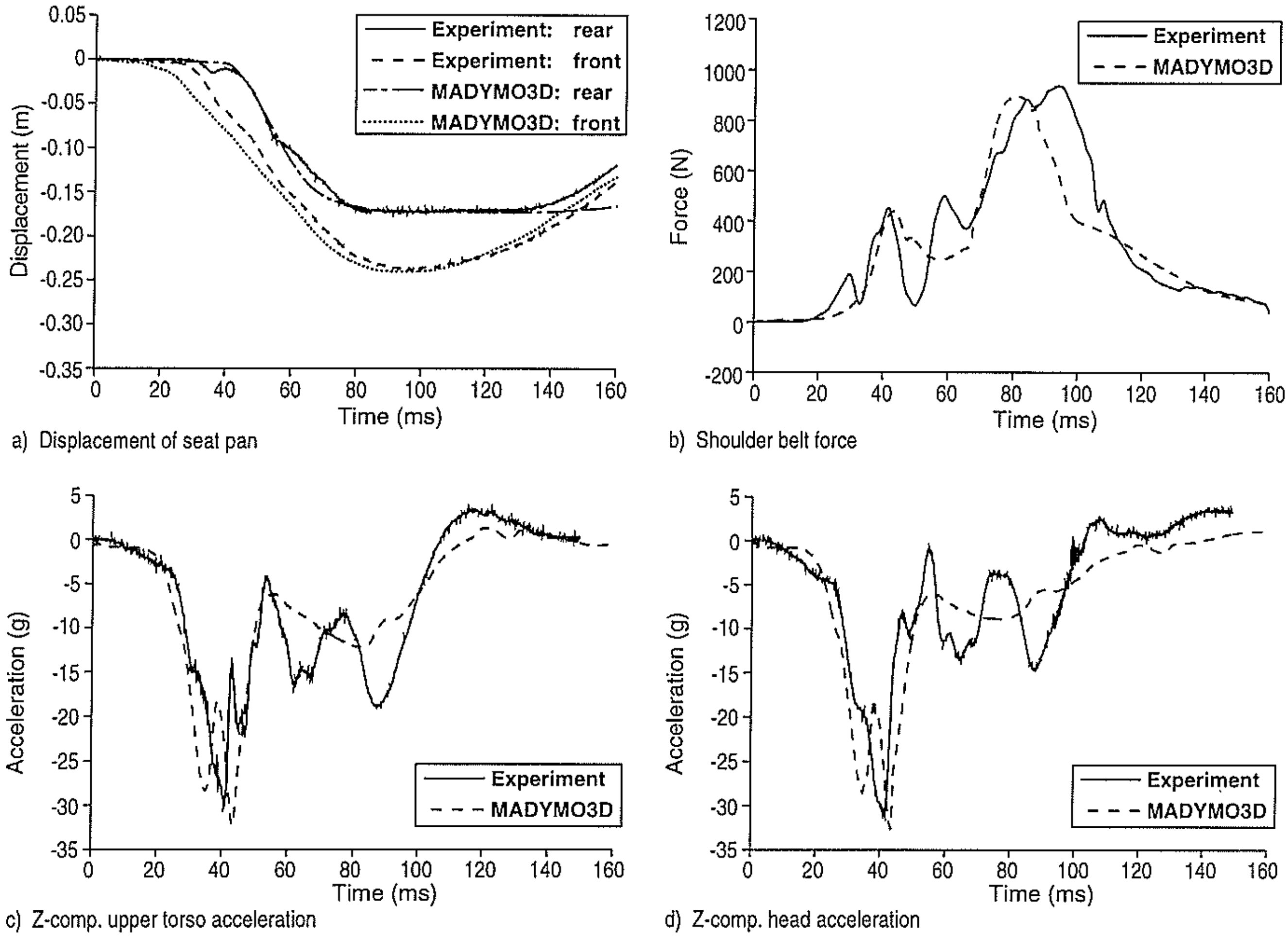



*Fig. 11 Failure mode of tube 2 after sled test 1*

The canvas seat base was not able to carry the loads caused by the dummy and the canvas tore slightly. Moreover the feet support bent early during the test. This meant that for much of the time, the lower legs had no constraint, and they bounced around on the edge of the foot plate.
Figure 12 show the test and MADYMO results. The MADYMO results are taken from the model with the non-linear spring. The model predicts the trends in most cases very well, despite using different dummies in test and model.




a) Displacement of seat pan
b) Shoulder belt force
c) Z-comp. upper torso acceleration
d) Z-comp. head acceleration

*Fig. 12 Crash test 1 versus MADYMO model*

Sled test 2: The second test was carried out with tube 935. The seat base and the feet support were strengthened compared to the first test.
The tube in this test crumbled for its full length (25 cm) and approximately 80 % of its circumference (the same 80 % as for sled test 1). The seat base was not able to carry the loads and was completely torn apart. The feet plate bent a few degrees. Figure 13 show the test and MADYMO results. The trends are predicted very well by the model.

Discussion of the results

In general, differences between model and tests are caused by:
- No experimental data were available for the seat (cushion and back) canvas stiffness, belt stiffness, or support strap stiffness. Values were estimated based on experience.
- The belts that support the seat cushion also tension the seat back. This has been modelled in a relatively crude way.
- Usage of Hybrid III model instead of Hybrid II model.

The maximum belt force is less than 1.2 kN. No significant loads go through the belts compared to a typical frontal car crash, with peak loads of approximately 12 kN. A different harness design could possibly reduce the spine loads by carrying more loads in the (shoulder) belts.

The measured and calculated acceleration signals of test 1 show a peak at 40 ms. This peak is probably caused by an unwanted stiff behaviour due to the break connection and/or crash initiation. This behaviour is modelled by a peak in the force - relative-elongation characteristic of the spring modelling the tube. Design requirements formulate the maximum allowable acceleration level for the dummy during the crash. Due to the unwanted peak, this requirement is not satisfied.

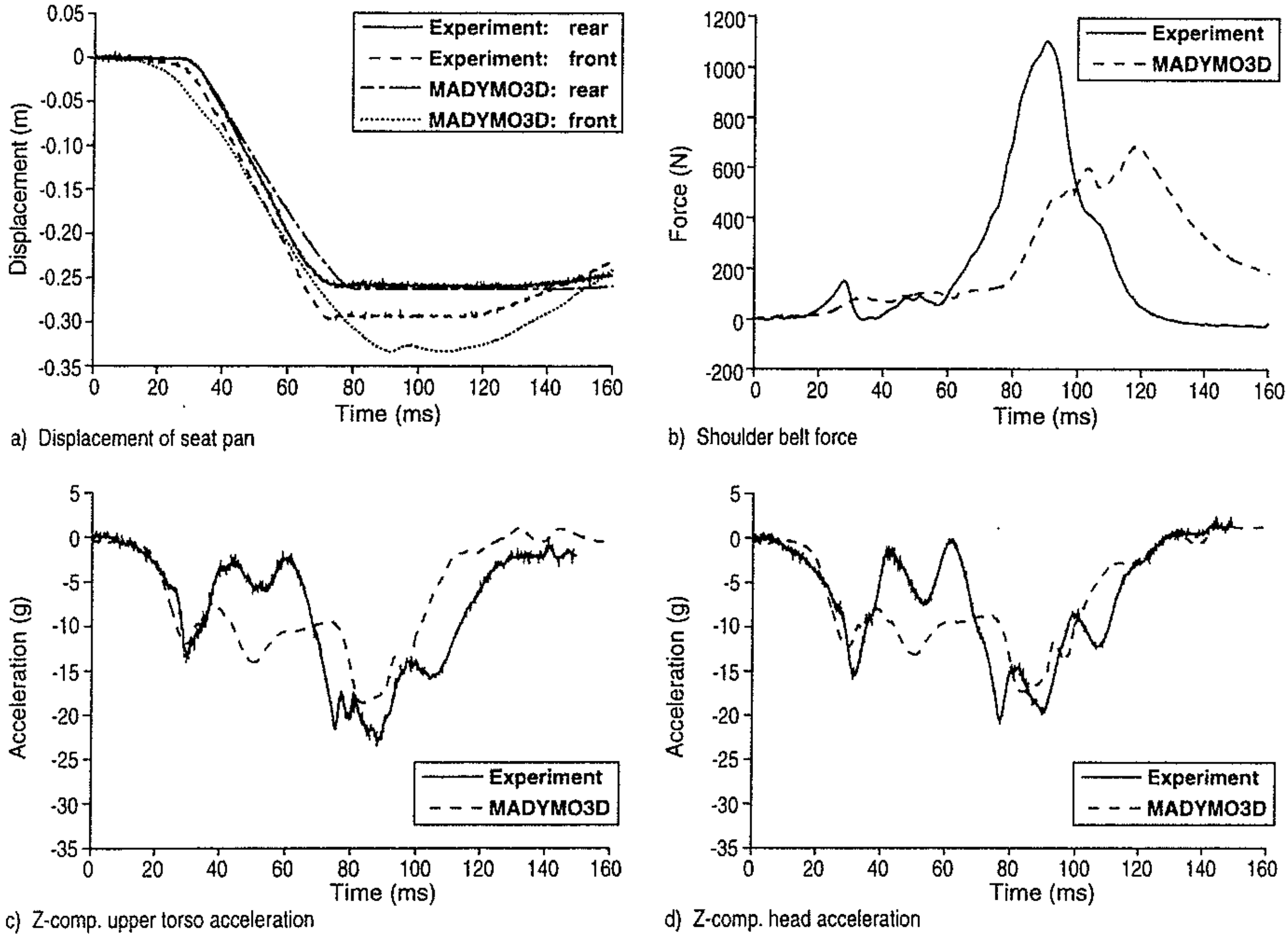



*Fig. 13 Crash test 2 versus MADYMO model*

In the second test, the seat base and the feet support were strengthened as compared to the first test. Likewise, in the MADYMO model the tube characteristics and the feet support were changed. The highest acceleration occurred at 80 ms, due to bottoming out of the tube, at that time the tube was fully crumbled. Moreover, the remaining kinetic energy caused rupture of the canvas part of the seat pan. The larger differences between test and model after 80 ms are mainly caused by not taking this rupture into account in the model.

The tube crushed for only 80 % in the test. We believe that this is due to some existing play in the seat connection points and the fact that the test is performed in horizontal direction. As the sled moves forward, the seat is still at rest due to its inertia.
Therefore the distance between the U-shaped profiles becomes larger due to the play and the crash tube is no longer fixed and moves slightly in lateral direction.
As a results, the crush mechanism was not the same as compared with a 100 % crushed tube. The result is, that the energy absorbing level of a 100 % crushed tube is not reached. So no explicit statement can be made about the energy absorbing capability of the crash tubes during good performance. In the actual situation this effect is less likely to occur.

Parameter study

Two major simplifications have been made in the analytical model to calculate the tube force:
- One rigid mass for the complete dummy and moveable part of the seat.
- Constant force characteristic for the tube without the initial peak.

A parameter study, using the MADYMO model with the non-linear spring, has been done to check the consequences of these two assumptions.

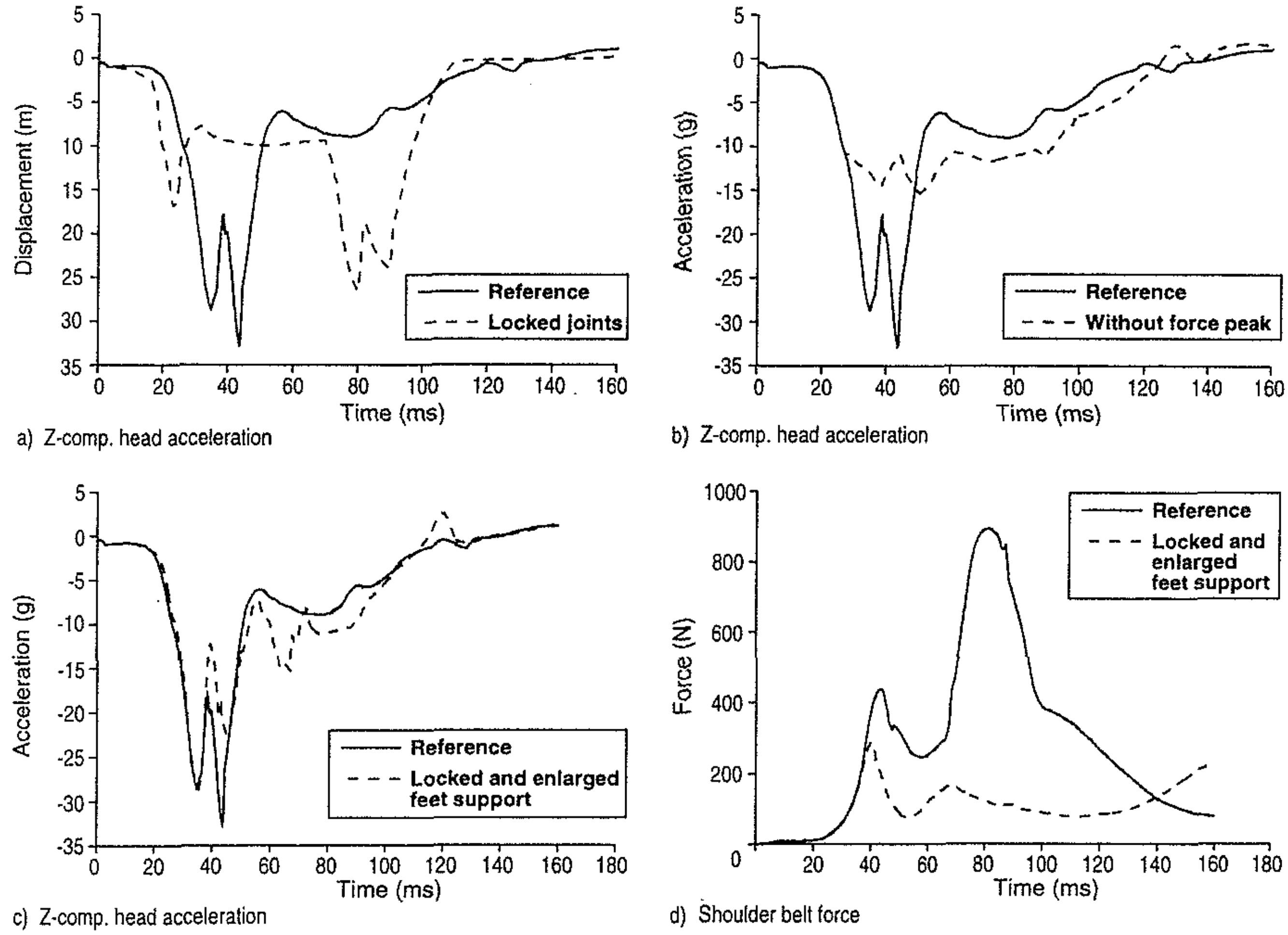



*Fig. 14 Parameter study*

To check the first assumption, the joints of the dummy and the revolute joint of the seat pan have been locked making it effectively two rigid bodies. The contact between the feet support and the dummy has been removed. Figure 14a shows the head acceleration in Z-direction in both cases, locked and unlocked joints.
Comparison of the two signals up to 80 ms leads to the conclusion, that the feet support is very important. A significant part of the load goes through the legs and the occupant motion is strongly effected by it. The theoretical calculation does not take this force path into account. Also the multi mass modelling of the occupant is important. Representing the dummy by one body leads to an underestimation of the accelerations.
The acceleration peak around 80 ms for the locked case is caused by bottoming out of the crush tube. The tube force was too low to decelerate the seat in time. This in spite of the fact that the tube force was much higher than necessary according to the analytical model. Bottoming out results in high acceleration peaks in the later stages of the crash similar to the peak caused by the crash initiation. Obviously this bottoming out should be avoided by using a longer tube or one with a higher crush force.

To check the second assumption, the peak in the force - relative-elongation characteristic has been removed resulting in a constant force characteristic. Figure 14b shows the head acceleration in Z-direction in both cases, with and without peak. The figure shows that a strong correlation exists between the peak around 40 ms and the peak in the crush characteristic of the tube. Obviously, the analytical model should take this peak into account in order to give a good approximation of the actual behaviour. Due to the importance of this peak on the accelerations experienced by the dummy, it is worthwhile to investigate options to reduce this peak. A redesign of the break connection and/or crumble initiator may be necessary.

A third parameter study has been done to analyze the influence of the bending of the small feet support. During the first test the feet plate bend and the lower legs bounced around the edge of the plate. In the model this support has been modelled using a contact ellipsoid. To check the influence of this, a modified support has also been used. In this model, the feet support joint has been locked and the size of the feet plate is enlarged in order to prevent uncontrolled motion of the legs. Figure 14c shows the head acceleration in Z-direction in both cases, unlocked small and locked large feet plate. The figure shows a reduced peak around 40 ms and an increased acceleration level between 50 and 110 ms. However, the locked enlarged feet support did significantly decrease the belt force. The forward motion of the legs, which caused the peak in the belt force around 80 ms (Fig. 14d), disappears if the locked large feet support model is used.

## Conclusions

Regarding to the design:

- The primary mechanism which controls the loads on the occupant are the crush (force-deflection) characteristics of the tube, and the stiffness of the seat cushion.
- The influence of the feet support on the belt forces implies also a large influence of the helicopter floor penetration on these forces.
- Redesign of the break connection and/or crumble initiator may be necessary to lower the correlated acceleration peak. Bottoming out should be avoided by using tubes with sufficient energy absorbing capabilities.
- No significant loads go through the belts. There is a potential to reduce the spine loads by carrying more vertical load in the belts. This would require a different harness design.

Regarding to the model:

- The MADYMO Hybrid III database is a good representation of the Hybrid II in vertical loading.
- The MADYMO model predicts experimental results very well, and is suitable for optimization and studying design alternatives.
- Representing the dummy by one body in the analytical model leads to underestimation of the accelerations. This is also true for neglecting the initial peak in the crush characteristic of the tube.

The present study has indicated the feasibility to design, fabricate, test and analyze crashworthy troop seats with the potential to upgrade the crashworthiness of older type military troop carrying helicopters. Future studies will be focused on the effect of retrofitting such seats on the chances for survivability of the troops carried.

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