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DEVELOPMENT OF A NEW BRAKING SYSTEM FOR A HORIZONTAL SLED USED IN CRASH TESTS WITH AN IMPOSED DECELERATION

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ABSTRACT

In crashworthiness, experimental tests with an imposed deceleration are customary. At LAST Crash Labs, Politecnico di Milano, Italia, an oleo-pneumatic braking system is currently used to arrest with a prescribed deceleration profile a trolley running on a horizontal sled. Unfortunately, this system is rather costly, difficult to regulate and complicated time profiles are out of reach. In effort to overcome these drawbacks, a different braking system is under development. The basic idea is to arrest the trolley by means of a number of irons bars (here called *deceleration beams*) with different lengths and placed at different distances the ones from the others. The concept is rather simple, but difficult to realise. Core of the system is the constraint system meant to slow down the trolley by deforming the deceleration beams. Static and dynamic tests initially carried out to develop the new braking system were also used to validate a numerical model that was eventually used to perfect the braking system. A numerical scheme was worked out so that, fixed an arbitrary deceleration profile, it were possible to decide the test set-up that guarantees a deceleration close to the fixed one.

1. INTRODUCTION

In crashworthiness, sled tests with an imposed deceleration are rather customary because allow reproducing the impact scenario of a crash event without requiring destroying the tested structure.
Sled tests are rather accurate and repeatable and it is possible to measure the quantities of interests.
Indeed, tests with an imposed deceleration are carried out with several purposes.
In automotive or dealing with flight safety or automotive, the requirements for the certification of the structures call for tests with an imposed deceleration.
In airworthiness, tests with an imposed deceleration are carried out to develop passive safety devices such as safety belts, restrain systems or energy absorption devices able to guarantee the passengers survive a crash landing and to verify the crash performance of the structures.
Different requirements call different deceleration profiles (Fig. 1).

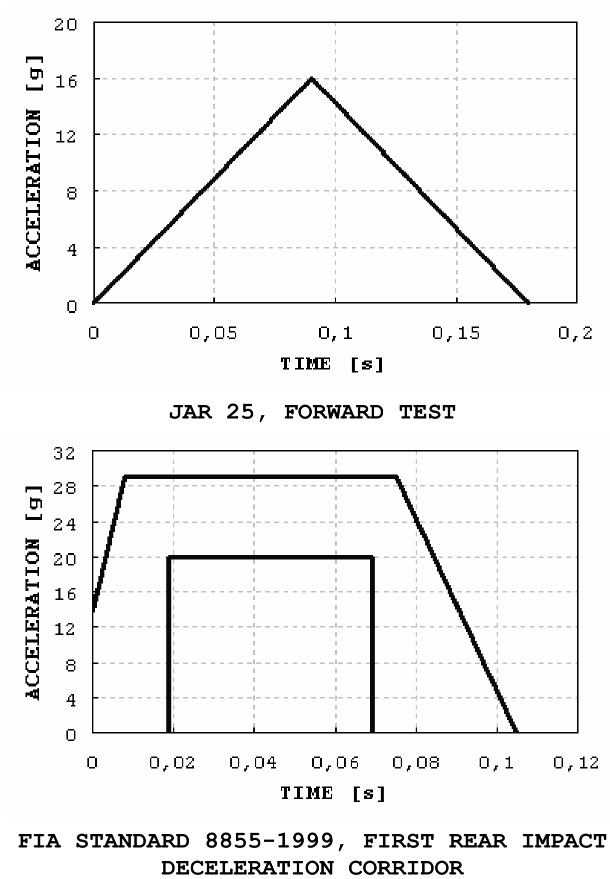


Figure 1. Typical deceleration profiles prescribed for different homologation tests.

Typical airworthiness requirements – FARs and corresponding JARs/EASAs 23, 25, 27 and 29 (Ref. 1, 2) – prescribe triangular deceleration profiles for seat certification. In JAR 25, for instance, an impact velocity of 13.41 m/s and a maximum deceleration peak of 16 g within 0.09 s for forward test are required (Fig. 1).
Requirements accordingly with FIA standard for competition seat – Standard 8855-1999 (Ref. 3) – call for trapezoidal deceleration profiles (Fig. 2).

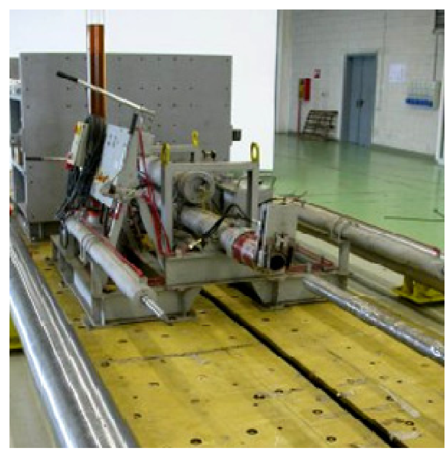
Imposed deceleration tests with impact scenarios similar to the ones described before are carried out at LAST Crash Labs, Politecnico di Milano.
The test article is mounted on a trolley running on a 40-m length horizontal sledge. The trolley is initially accelerated to the prescribed velocity and then the deceleration profile is imposed by means of an oleo-pneumatic braking system.
This braking system is rather efficient, but costly and difficult to use.
In view of that, a new braking system meant to substitute the oleo-pneumatic one is currently under development.
The underlying idea is to impose an arbitrary deceleration profile to the trolley using a number of iron bars – here called *deceleration beams*.
The core of the system is the beams constraint. The beams are placed on the trolley path and progressively engaged by an hooking system mounted beneath the trolley. The deceleration beams, after being engaged, run through the constraints undergoing severe deformations. Eventually, the kinetic energy of the trolley dissipates as plastic deformation of the beams.
In this research work the physical realization of the system and the experimental and numerical activities carried out to develop it are introduced.
Static tensile tests were carried out to evaluate the maximum restrain forces provided by one single deceleration beams when changing the kink in the constraints. Sled tests were carried out to evaluate the dynamic behaviour of the system, as well as measure the effective restrain force and trolley deceleration.
Numerical simulations were also carried out using LSTC LS-Dyna (Ref. 4) in order to perfect and further develop the system.
After having verified the feasibility of the new concept, a numerical technique was worked out so that, fixed an arbitrary deceleration profile, it were possible to decide the test set-up that guarantees a deceleration close to the fixed one.

2. PREVIOUS BRAKING SYSTEMS

In the years, different procedures for sled tests with an imposed acceleration/deceleration have been developed.
At LAST Crash Labs, Politecnico di Milano, Italia, tests with an imposed deceleration are carried out using a 45-m length horizontal sled. The test article is mounted on a trolley running on the sled. The trolley is accelerated to a *prescribed* velocity (up to 25 m/s) and hence stopped by means of a braking system.
In what follows, two of the braking systems developed at LAST Crash Labs are described.

2.1. Oleo-pneumatic braking system

The oleo-pneumatic braking system (Fig. 2) is the first developed at LAST.
In the years, it has demonstrated to be reliable: the deceleration profiles are accurately achieved and the tests are repeatable.



FRONT VIEW

DETAIL

Figure 2. The oleo-pneumatic braking system used at LAST Crash Labs, Politecnico di Milano.

Initially, the oleo-pneumatic braking system was developed to achieve triangular deceleration profiles accordingly with the FAA and JAA requirements for helicopter seats homologation (Ref. 1, 2). Subsequently, its was extended to sport cars seats homologation tests accordingly with FIA requirements (Ref. 3).
In the hands of expert technicians, the use of this braking system is not particularly troublesome – though the set-up is quite difficult.
Furthermore, the maintenance of the system is rather expensive and, due the low reactivity of the system, multifaceted deceleration profiles are out of reach.
Another serious concern is the anti spring-back system that causes a progressive deteriorating of the engage-system (shown in the detail in Fig. 2).
Nevertheless, these are not the main reasons why the feasibility of different braking system was investigated. Indeed, it was the need to achieve complex deceleration profiles.

2.2. Servo-assisted braking system

The first attempt to achieve complex deceleration profiles consisted of an onboard braking system mounted beneath the trolley and controlled by an active control.
The system consisted of three braking pistons and twelve lining tablets on six steel plates. The latter at the end of the run of the trolley engage some fixed pivot that stop them and hence the trolley. The braking pressure (i.e. the pressure exerted by the braking pistons on the plates) is controlled in feedback by an active control system that basically allows to obtain any desired deceleration time history.
Unfortunately, this braking system is heavy and rather difficult to use. The effectiveness of the control is limited by the low reactivity of oleo-pneumatic system that controls the braking pistons. Furthermore, plastic deformations and local erosion in the plates cause a fast deterioration of the braking system performances.
In view of that, the servo-assisted braking system was eventually abandoned.

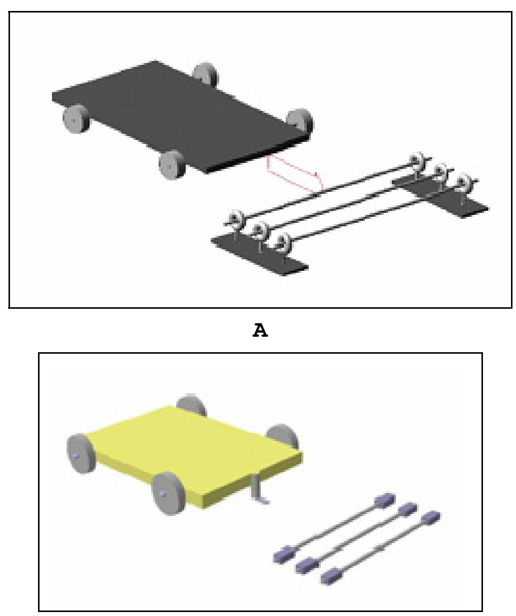
3. THE NEW BRAKING SYSTEMS

The concept of the new braking system is here introduced and the main subcomponents are described.

As mentioned, the underlying ideas are conceptually rather simple, though difficult to put into practice.

3.1. Basic ideas

The concept consists of a rather simple idea: a prescribed deceleration is imposed to the trolley running on the horizontal sled by means of a set of iron bars placed on the path of the trolley.
The same concept has already found applications in sled tests. The distinguish feature of the system here investigated is the aiming at developing a reliable and ease-to-use system which allows to achieve arbitrary deceleration profiles.
Furthermore, the new braking system is meant to guarantee high repeatability of the tests, low vibrations and negligible elastic energy restitution (i.e. a negligible rebound of the trolley).
In Fig. 3, the concept of the braking system is shown by means of two *multibody* models worked out in the preliminary analysis phase of the research.
The deceleration is accomplished by converting the kinetic energy of the trolley into deformation energy in the bars. In view of that, it is straightforward to figure out that the beams constraint is of paramount importance for the effectiveness of the overall braking system.



A

B

Figure 3. Two multibody schematic representation of the new braking system basic concept.

A constraint system to investigate the influence of the constraint geometry on the restrain force (shown in Fig. 4) was developed.
The beams are constrained in the direction of trolley motion, not along their own axis and therefore the beams can be disengaged.
The deceleration profile is obtained by deciding for the number (the number for each constraint and the overall number) and the characteristics (geometry and material) of beams and/or the distance between one constraint station and the others.

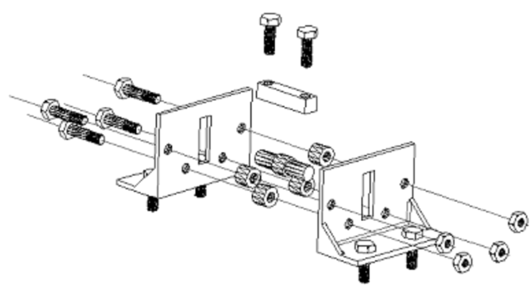


Figure 4. The exploded view of the braking system constraint block.

3.2. Overall physical description

The new braking system is meant to be a part of the horizontal sled facility.
Specific requirements for the new braking system exist – such as the ease-of-use and maintaining costs.
Furthermore, the facility in the within of which the system will operate poses severe constraints on the braking system features.

3.2.1. The horizontal sled facility

The horizontal sled facility (partly shown in Fig. 10) consists of:

- two solid parallel guides, 46-m length and 2.5-m distance;
- a fixed anvil;
- the launch system used to accelerate either a trolley running on the guides or a test article endowed with wheels between the guides.

3.2.2. *The deceleration beams*

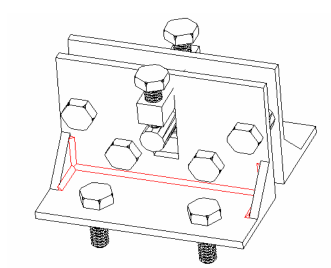
The features of the deceleration beams are fundamental. The effectiveness of the braking system depends on that: geometry and material of the deceleration beams.
If the circular cross-section is doubtless the most appropriate for the beams, the length and the diameter are parameter the importance of which was carefully investigated as well as the influence of the beams material.

3.2.3. The hooking system

In effort to achieve a prescribed deceleration, the beams are engaged by a hook fixed under the trolley.
The basic requirements for the hook are two: it must be solid and it must decelerate the trolley without transferring to the trolley significant vibrations and moments. The first hooking system was realised as a blunt cylinder welded to a plate. The plate is as long as the trolley and as width as the cylinder. During the test it was bolted to the bottom structure of the trolley.

3.2.4. The constraint system

The constraint system is the core of the braking system here introduced. The first constraint system realised consisted of two blocks (shown in Fig. 5) defined RHS and LHS block.

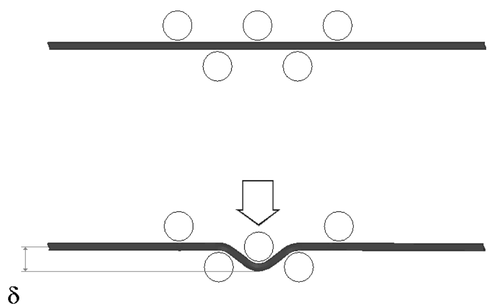


DRAFT OF THE CONSTRAINT BLOCK

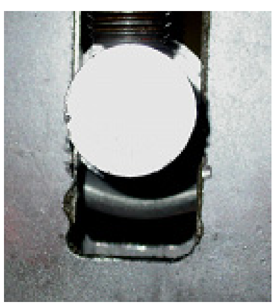
PHYSICAL CONSTRAINT BLOCK

Figure 5. The first constraint block – in detail.

The two blocks consists of two L plates, one I-beam and five cylindrical supports and wheels. Four screws lock together the two L plates and the five wheels (show in Fig. 4).
Screwing down the I-beam screws (Fig. 6), it is possible to impose a pre-deformation (the *kink*) to the beam in the region of the constraints and hence to modify the restrain force provided by the beam.
The diameter of the wheels was such to avoid beams failures during the disengagement phase. The size of screws was such to avoid shear, axial and combined shear and axial failures (Ref. 4).



KINK FORMING PROCESS



PHYSICAL KINK

Figure 6. The kink *forming process.*

4. EXPERIMENTAL ACTIVITY

The core of the new braking system is the constraint system and, therefore, the most part of the experimental activities regarded the development of this component.
Static and dynamic tests were carried out to verify the feasibility of the constraint system: *static test* to evaluate the restrain force when pulling the beam out of the constraints; *dynamic tests* to investigate the performances of the constraint system when mounted in the within of the horizontal sled facility.

The material used for the deceleration beams is a parameter of paramount importance.
In effort to acquire relevant knowledge on the mechanical properties of the beam material, preliminarily static tensile tests were carried out. The specimens were obtained from a beam of the ones used in the static and dynamic tests described in what follows.

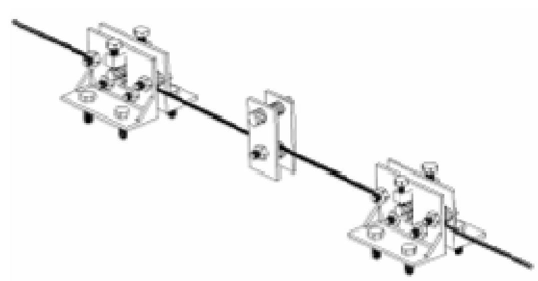
4.1. Static tests

Static tests were carried out considering different beam pre-deformations in effort to evaluate the maximum restrain force of a single beam as well as the influence of the beam local bending. Load-displacement curves were also acquired.

4.1.1. Test facility

Static tests were carried out fixing the constraint system at a distance of 0.6 m on a massive box I-beam positioned below a 50-t hydraulic press machine (Fig. 7).

TEST FACILITY



CONSTRAINT SET-UP

Figure 7. The static test facility and the constraint set-up.

The beams were engaged in the constraint blocks and pulled by the press machine using a specific device also shown in Fig. 6.
The load/displacement curve of the beam was acquired by means of a 10-t force transducer and a LVDT connected to a PC via a Pacific Instruments data acquiring system with a sample frequency of 10 Hz.

4.1.2. Data acquired

Three test settings were considered, different in depth of the local bending of the beams in the constraint block (here also indicated as *kink*) δ: (1) δ = 0 mm, (2) δ = 0 mm, (3) δ = 15 mm.
The local bending at the constraint was achieved by screwing down the constraint I-beam (see the detail in Fig. 7).
In that, the aim was to obtain a quantitative evaluation of the maximum restraint force as well as the influence of the kink on the shape of the load/displacement curve.
In all the tests a final displacement of 370 mm was imposed by arresting the hydraulic press machine.
Several tests were carried out with different offsets. Data acquired were reliable and tests repeatable.
In Fig. 11, force/displacement mean curves obtained after the tests are shown with regard to the different test settings.
Eventually, it was observed that the absorbed energy and, hence, the deceleration obtainable is approximately proportional to the kink in the constraint.

4.2. Dynamic tests

Dynamic tests were carried out to evaluate the deceleration profile of trolley using the new braking system.

4.2.1. Test facility

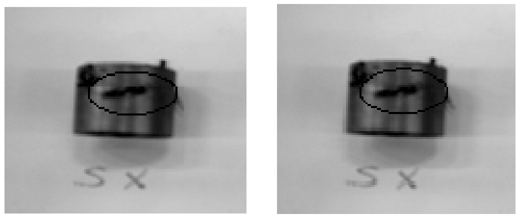
The dynamic tests were carried out using the horizontal sled facility (Fig. 9). A provisional hook system was mounted on the forward plate of the trolley. The overall trolley and the hook system had a mass of 520 kg.
The constraint system used in static tests (Fig. 7) was adapted accordingly with the facility characteristics. The distance between the two constraint blocks was increased to 0.865 m.
The deceleration of the trolley was measured by means of piezoresistive accelerometers. The impact velocity was measured by means of Tag-Heuer photoelectric cells.
Data were acquired with a Pacific Instruments data acquiring system – sample frequency: 12,500 Hz.

4.2.2. Data acquired

Dynamic tests were carried out launching the trolley at a prescribed velocity against one single beam (Fig. 9).
Four test settings were considered. The settings differed in the sled impact velocity: (1) V_o = 8.7 m/s, (2) V_o = 10.5 m/s, (3) V_o = 11.6 m/s and (4) V_o = 10.3 m/s.
Impact velocities were the only difference between the tests. The impacting mass, 520 kg, the beam material, SIGY 400, the length of the beam, 2 m (except in the fourth test in which length of the beam was 4 m) and the beam kink, 15 mm, were the same.
In second and third test the decelerating bar was completely disengaged while in the first and fourth trial the bar stopped the trolley without disengaging.
Several tests were carried out with the different impact velocities. The data acquired were reliable and the tests repeatable.
In Fig. 12, the force/displacement mean curves obtained after the tests are shown with regard to the different test settings.

Figure 8. The horizontal sled and the test facilities used in the dynamic tests.

In Fig. 9, the wheel at the kink (i.e. the central wheel in Fig. 4) of the LHS and RHS constraint blocks after a dynamic test are shown.
The friction of the beam sliding on the constraint wheels caused a local rising in temperature and wheel melting. This occurrence was not unexpected, in fact the wheels are meant to be substitutable, but undesired and therefore a new constraint system is currently under development.



LHS RHS

Figure 9. LHS and RHS constraint block wheels at the end of the test programme.

4.3. Remarks

With regard to the static tests, a significant increase in the restrain force was observed when considering different depth of the kink at the constraint.
Comparing the measures from static and dynamic tests, a difference in the peak force of about 20% due to dynamics effects (strain rate and friction) was observed.

5. NUMERICAL ACTIVITY

Moving from the results of the experimental tests, a simplified finite element model was realised.
The aim was to exploit the potentiality of the numerical approach to further develop the braking system.
The simulations were carried out using LSTC LS-Dyna 970 (Ref. 4) a proven nonlinear explicit FE code widely diffused in crash analyses.

5.1. Finite element model

The FE model (Fig. 10) was developed referring to a simplified geometry of the braking system so that it was possible to use the same model for both static and dynamic tests.
The FE model consisted of three parts: the constraint wheels, the hook and the beam.
The constraint wheels were modelled with 1270 four-nodes shell elements (127 elements for each wheel). The wheels were considered rigid – because the deformations were not relevant. The wheels were fixed.
The hook was modelled with 188 shell elements. The piecewise linear plasticity material was used. Translation in a direction perpendicular to the axis of the beam was the only degree of freedom.
The beam was modelled with 6000 eight-nodes solid elements.
In order to avoid inaccuracy in the solution due to zero-energy modes, a rather fine mesh was built and a hourglass control was defined.
The piecewise linear plasticity material model was used and the mechanical property defined referring to the previous experimental tests.
Two different contact interfaces, between the constraint wheels and the deceleration beams and between the hook and the deceleration beams, were defined.
Static and dynamic friction coefficients and exponential decay coefficients have been determined correlating output results of numerical simulation with experimental data.

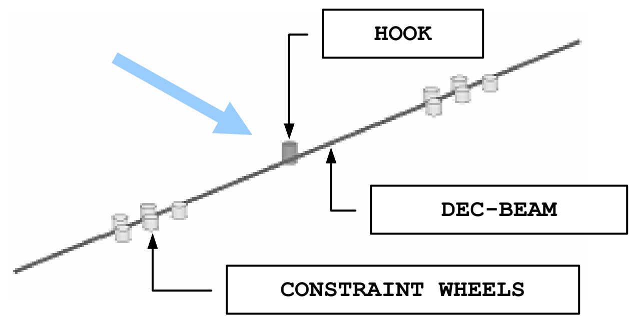


Figure 10. New braking system simplified FE model.

5.2. Static test simulations

In effort to reproduce the static tests, the simulations were divided into two phases.
In a first phase, the local bending at the constraint, the *kink*, is obtained by a prescribed motion of the central constraint wheel. The motion assigned to rigid central wheel was such to produce the three different test settings: (1) $\delta = 0$ mm, (2) $\delta = 0$ mm, (3) $\delta = 15$ mm.
In a second phase, the actual test is reproduced imposing a prescribed motion to the hook. In order to avoid inertial effects, the hook moved with a constant velocity of 5 m/s. The total displacement was 0.400 m.
In Fig. 11, the load/displacement curves from numerical simulations for the three test settings described are shown.

5.3. Dynamic test simulations

The simulations of the dynamic tests were divided into two phase, too.
In a first phase, the 15-mm kink at the constraint blocks is created. In the second phase, the trolley is launched with an (imposed) initial velocity equal to the impact velocity.
The mass of hook is the main difference between the FE model used in static and dynamic simulations: in dynamic simulations the entire mass of the trolley (i.e. 520 kg) was given to the hook.
Cowper-Symonds coefficients were defined to model the influence of the strain rate. The distance of the constraint blocks was increased to 0.860 m.
In Fig. 12, the load/displacement curves from the numerical simulations for four test settings described are shown.

6. NUMERICAL-EXPERIMENTAL CORRELATION

The data acquired during the tests and the results obtained after the simulations were compared. The close numerical-experimental correlation obtained justifies the use of the numerical model as a means to further develop the new braking system.

6.1. Static tests

With regard to the static tests, the numerical-experimental correlation is satisfactory: not only the initial slope but also the peak value and time profile are close.
In Fig. 11, the numerical-experimental correlation for the static tests is shown.

6.2. Dynamic tests

With regard to the dynamic tests, both the acceleration profile and the force/displacement curve were considered.
In both the case, the numerical-experimental correlation is satisfactory: not only the initial slope but also the peak value and time profile are close.
In Fig. 12, the numerical-experimental correlation for the dynamic tests is shown.

6.3. Remarks

When considering both of the static and dynamic tests, the numerical-experimental correlation is satisfactory.
In view of that, the numerical model is demonstrated to be a reliable tool to further develop the concept of the new braking system.

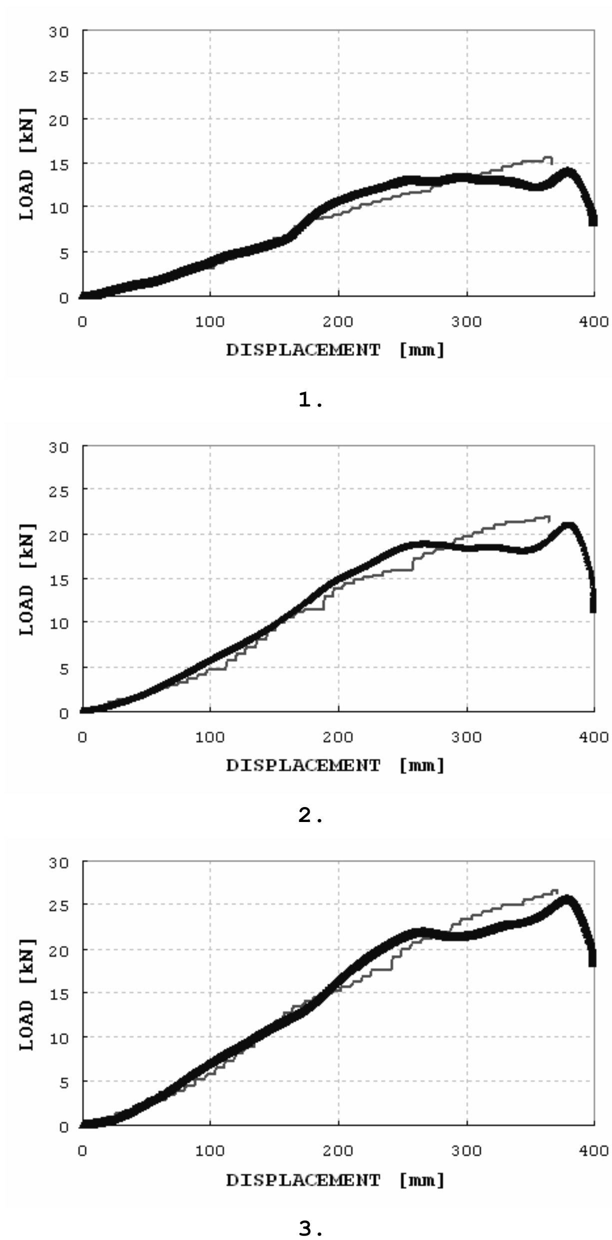
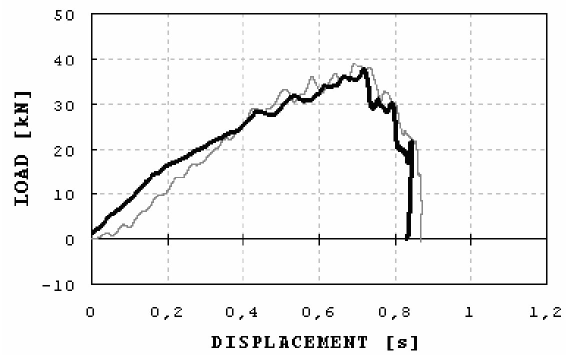


Figure 11. Static tests: experimental data (thin) and numerical results (thick).



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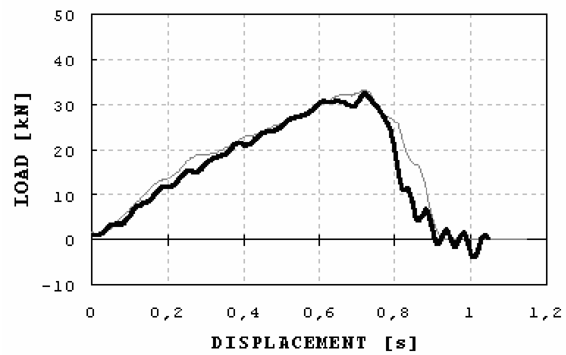

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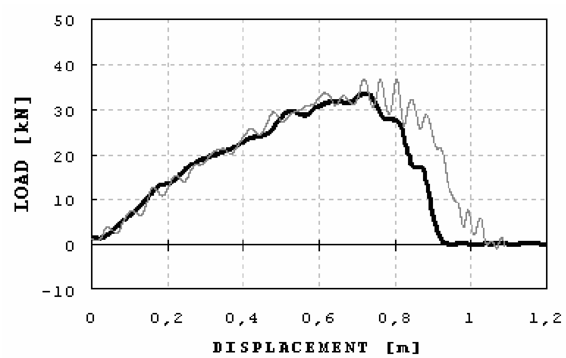

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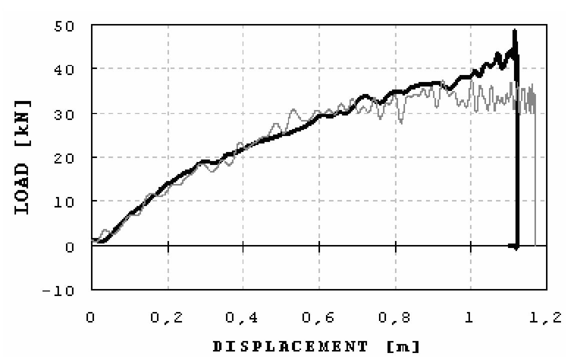


4.

Figure 12. Dynamic tests: experimental data (thin) and numerical results (thick).

7. FINAL ARRANGEMENT

The first braking system developed and previously described was meant to prove the feasibility of the concept ideas.

Eventually, a new constraint system (shown in Fig. 13) was developed. It allows to place up to sixteen iron bars in four different stations.

The constraint system consists of two massive plates (Fig. 13) fixed to the ground.

The four constraint wheels (i.e. cylinders with a diameter of 40 mm) are mounted on these plates so that a 25-mm kink is obtained at the constraint.

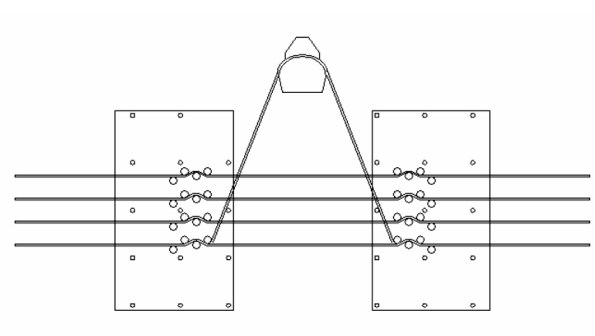
Different constraint geometry can be achieved mounting the wheels in different position.

The distance between two constraint blocks (inter-axis) depends on the distance between the two constraint plates.

The kink is fixed to 25 mm. The deceleration beams must be pre-deformed before being positioned.

The constraint system consists of four stations at relative distances of 120 mm. Each of them can house up to four deceleration beams simultaneously.

The diameter of deceleration beams is fixed to 10 mm.



DRAFT OF NEW CONSTRAINT SET-UP

PHYSICAL CONSTRAINT

Figure 13. The final arrangement of the constraint system.

The configuration of the braking system here described is in theory suitable for impact scenarios featuring a 2,000 kg impacting mass at a velocity of 9.1 m/s – to which corresponds deceleration peak of 30g in 0.031 seconds.

8. A METHOD TO OBTAIN ARBITRARY DECELERATION PROFILES

Since the early beginning of its development, the new braking system was meant to achieve arbitrary deceleration profiles in a rather simple way.
Experimental tests and numerical simulations show the feasibility of the underlying ideas.
Finally, a numerical tool to build an inverse relation between an arbitrary deceleration profile and the corresponding test set-up was developed.

8.1. Problem definition

From a mathematical standpoint, establishing a correlation (a sort of *transferring function*) between the deceleration profile (input) and the test set-up (output) can be regarded as a constraint optimisation problem: find a test-set-up such as the *distance* between the corresponding deceleration profile and the desired one tends to zero.
In that, two important remarks: the optimisation domain is not continuous and the problem is highly nonlinear.
Accordingly, heuristics optimisation schemes (such as Genetic Algorithms) are recommendable.
On the other hand, using an heuristic approach, it could happen that the number of iterations necessary to converge is larger than the number of allowable set-ups.
In view of that, it would be more efficient to draw a map of all the feasible deceleration profiles.
The success of the numerical tool depends on the reliability of the code used to obtain the deceleration from the test set-up. The use of explicit codes guarantees accurate solutions but is also time-consuming. Furthermore, an appropriate definition of *distance* is mandatory. The limits in the deceleration profiles in the requirements for the certification are usually given as inequalities. It is not uncommon that to good correlation factor correspond unacceptable deceleration profiles.

8.2. The sled test emulator

When considering complicated test set-ups, numerical simulations become time-consuming.
In view of that, the use of heuristic optimisation scheme seemed not recommendable.
In effort to overcome this problem, a simple sled test *emulator* was written – using FORTRAN 77. The emulator accepts in input the test set-up: number, type and positions of the deceleration beams, initial velocity, initial acceleration and mass of the trolley.
The braking system is modelled as a (one-dimensional) mechanical system consisting of the sum of a number of mass, dumpers and springs (Fig. 12).
The deceleration beams are modelled as *equivalent* nonlinear springs and dampers, the properties of which were obtained from numerical simulations carried out with LSTC LS-Dyna.
In particular, with regard to the nonlinear springs, the load/displacement curve is obtained dividing in eight fields and interpolating (least square error approximation) the results from numerical simulations similar to those previously described. Adopting this procedure, a library of different deceleration beam types was created.
Experimental tests are not necessary to further develop the system unless final verification of the system.
The dynamics of the system is computed using a simple explicit time integration scheme (Ref. 5).
The *emulator* was developed to reproduce all the feasible set-ups.
The set-ups are coded as a string of four groups of four words. The groups represent the stations and the word the deceleration beams – exactly as allowed by the rack (or roster in the detail in Fig. 12)
The value of the words represent the deceleration beam typology.
The accuracy and the efficiency of the emulator was evaluated considering a number of different test set-ups and comparing the results with the ones obtained with LS-Dyna.

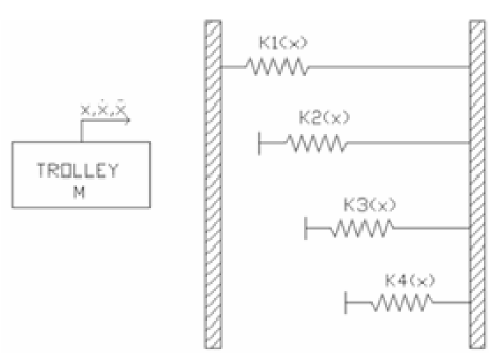


Figure 14. The schematic representation of the new braking system.

In particular, a test set-up for which also experimental data exist, is here presented.
The test set-up consists of eight identical beams: beams type 4 (i.e. beam length: 2 m, beam material: SIGY 275 MPa mild steel). The beams were positioned two-by-two in four different stations distant 200 mm. The trolley (1550 kg mass) had an initial velocity of 12.5 m/s.
The corresponding input string is given below.

$$\underbrace{[4][4][0][0]}_{\text{1st \_ station}}\underbrace{[4][4][0][0]}_{\text{2nd \_ station}}\underbrace{[4][4][0][0]}_{\text{3rd \_ station}}\underbrace{[4][4][0][0]}_{\text{4th \_ station}}$$

In Fig 15, the results of the simulations carried out with the emulator and LS-Dyna are compared. The correlation is close (APE index is 4.4, while the correlation index between results is 0.98).
In view of the results obtained, the developed sled test emulator was demonstrated to be both reliable and time-efficient.

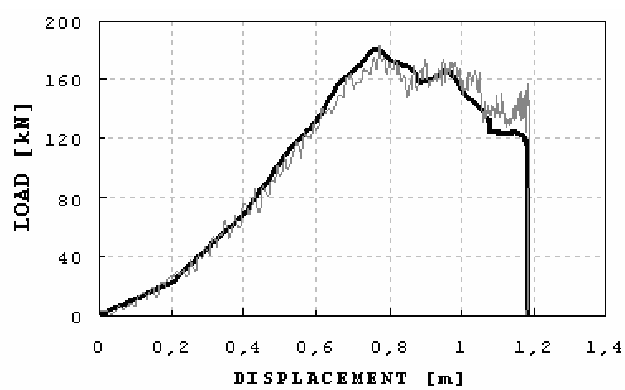


Figure 15. Comparison between LSTC Ls-Dyna 970 (thin) and sled test emulator (thick) results.

8.3. Optimisation scheme

The developed sled test emulator was suitable to run a large number of simulations and therefore was integrated, as subroutine, in a numerical scheme based on Genetic Algorithm theory to obtain a test set-up moving from a desired arbitrary deceleration profile.
Genetic Algorithms employ Darwin's concept of natural selection by creating a population of candidate designs and applying probabilistic rules to simulate the evolution of the population. Individuals in the population are discrete encodings of candidate solutions to the problem being solved, and the evolutionary process searches for optimal designs using only objective function information.
Genetic Algorithms are less likely than conventional optimization techniques to get trapped in locally optimal areas of the search space, and the population structure makes it useful in exploring contemporarily many candidate designs.
The optimisation scheme developed was rather simple and implemented using FORTRAN 77. The population consists of sixteen individuals. Each one of these individuals represents a test set-up. The information on the set-up are coded up in a chromosome string (the string introduced in the previous section). The value given to a chromosome represents the deceleration beam type. The position in the string is representatives of the position of the deceleration beam in the roster.
The initial population is randomly chosen among all possible combinations of set-ups.
At each step, the deceleration profile corresponding to each individual is obtained by means of the sled test emulator. Hence, the fitting of each individual with the target deceleration profile is evaluated referring to the Average Percentage Error (APE) index.
Since an overall accuracy does not necessarily imply a good local accuracy, also a second estimator of accuracy related to the maximum local errors computed was considered: the Relative Maximum Absolute Error (RMAE).
Then the evolutionary process is applied to the current population and best individuals are used to produce the following population.

8.4. Application

The numerical scheme described was eventually used to find the test-up that allows a deceleration peak of 12 g in 0.071 s – that is the deceleration profile required in helicopter seat homologation tests (Ref. 1, 2).
Eventually, the set-up obtained was:

$$\underbrace{[4][4][0][0]}_{\text{1st \_ station}}\underbrace{[0][0][0][0]}_{\text{2nd \_ station}}\underbrace{[4][4][0][0]}_{\text{3rd \_ station}}\underbrace{[4][4][0][0]}_{\text{4th \_ station}}$$

With regard to the corresponding LSTC LS-Dyna 970 simulation, the correlation was good (APE index: 10.23, the RMAE index: 0.2740).
In Fig. 16, the deceleration profile obtained with the numerical scheme developed is compared with the desired one and the one obtained after a full-scale test.
The deceleration profile obtained with the numerical scheme has the same shape of the target deceleration profile, but the acceleration peak is smaller.

The optimisation process converged and therefore there is not doubt that the profile obtained is the one that better fit the target curve.
In that, the numerical scheme developed showed a weakness of the braking system that eventually is not able to reproduce the target profile.
Different deceleration beams and/or constraint systems need to be developed.
With regard to the experimental data, it should be noted that the difference in the descending part of the deceleration profile was due to an unexpected beams disengagement.
That suggested the use of a stochastic approach to include in the numerical model the influence of uncertainties and *unexpected* events.

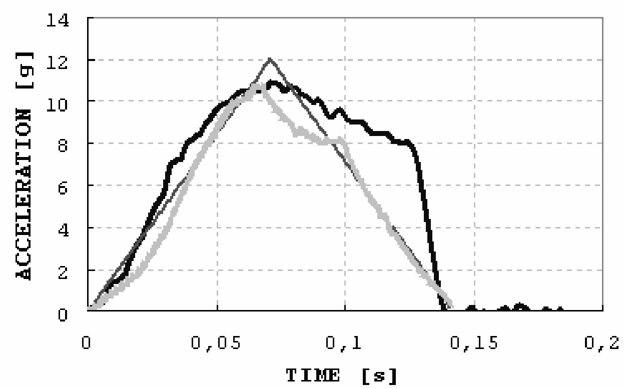


Figure 16. Comparison between numerical results (thick grey) and experimental data (thin black).

CONCLUSIONS

In crashworthiness, experimental tests with an imposed deceleration are customary.
At LAST Crash Labs, Politecnico di Milano, Italia, an oleo-pneumatic braking system is currently used to arrest with a prescribed deceleration profile a trolley running on a horizontal sled. Unfortunately, this system is rather costly and complicated time profiles are out of reach.
In effort to overcome these drawbacks, a new concept of braking system is under development.
The basic idea is to arrest the trolley by means of a number of iron bars (here called deceleration beams) having different lengths and placed at different distances the ones from the others.
A solution conceptually rather simple but difficult to put into practice.
The core of the system is the beam constraints meant to slow down the trolley by progressive deformation of the beams.
Initially, static and dynamic tests were carried out to verify the feasibility of the new braking system concept.
Subsequently, in effort to exploit the potentiality of the numerical approach to the problem, the tests were also numerically reproduced and the data collected in the tests compared with the results of the simulations.
The numerical-experimental correlation was satisfactory considering both of the static and dynamic tests. In view of that, the numerical model was demonstrated to be a reliable aided-design tool to further develop the new braking system concept.
Eventually, a numerical scheme based on Genetic Algorithm was worked out so that, fixed an arbitrary deceleration profile, it were possible to decide the test set-up that guarantees a deceleration close to the fixed one.
The scheme showed the need to improve the braking system and that the system need to be further developed before being used with confidence.

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REFERENCES

1. Federal Aviation Administration, Code of Federal Regulations, Part 27 “*Airworthiness Standards: Normal Category Rotorcraft*” and Part 29 “*Airworthiness Standards: Transport Category Rotorcraft*”, Apr 26, 2006
2. Joint Aviation Authorities, Joint Aviation Requirements, JAR-27 “*Small Rotorcraft*” and JAR-29 “*Large Rotorcraft*”, Nov 1, 2004
3. Federation Internationale de L'automobile, “*Norme 8855-1999 / Standard 8855-1999, Norme FIA Pour Sieges De Competition / FIA Standard For Competition Seats*”, Feb 14, 2003
4. E F Bruhn et al., “*Analysis and Design of Flight Vehicle Structures*”, III, Tri-State Offset Company, Ohio 1965. pp. 927
5. J O Hallquist, “*Ls-Dyna Theoretical Manual*”, Livermore Software Technology Corporation 1998.