A DESIGN SOLUTION FOR A CRASHWORTHY LANDING GEAR WITH A NEW TRIGGER MECHANISM FOR THE PLASTIC COLLAPSE OF METALLIC TUBES

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Abstract

The paper presents the basic guidelines for the design of a landing gear adopting a crash tube as an energy absorbing device in crash conditions. In the considered landing gear lay-out, a light alloy thin walled tube is mounted coaxially to the shock absorber cylinder and, in severe impact condition, is allowed to collapse in order to enhance the energy absorption performance of the landing system.
The basic aspects of the collapse of axially loaded metallic tubes are described. A novel triggering mechanism, activated in crash impact conditions, has been developed in order to eliminate the initial load peak in the tube collapse process. The device allows to study the possible design solutions for an additional shock absorbing stage that can be integrated in a landing gear structure without requiring the introduction of frangible attachments.
The characteristics of the triggering device are presented and the structural lay-out of a crashworthy landing gear adopting the developed additional energy absorbing stage is outlined.
The triggering mechanism validation tests confirm the device efficiency in the reduction of the initial load peak in the crash tube collapse. Experimental results are reported and correlated with the results of numerical analyses performed with a finite element explicit codes.
The potential performances of a landing gear adopting the crash tube, the triggering mechanism and a properly designed shock absorber, are analysed by means of a simplified numerical model, showing that appreciable energy absorbing capabilities and efficiencies can be obtained in crash conditions.

List of symbols

M_0	crash tube unit plastic collapse moment
σ_0	average stress of crash tube material in plastic range
σ_y	yield stress of crash tube material
D	crash tube diameter
t	crash tube thickness
λ	wavelength of the crash tube force oscillations during the collapse
P_m	crash tube average collapse load
P_A	maximum (initial) crash tube load at the collapse onset
P_o	maximum tube collapse force during the stationary collapse phase
W_s,	landing gear static load
n_c	average crash load factor
x_{SA}	piston stroke
x_{CT}	crash tube stroke
X_H	design hard landing shock absorber stroke
X_{CT}	crash tube available stroke
V_H	hard landing vertical velocity
V_{CO}	viscous force cut-off velocity
F_γ	adiabatic compression contribution to shock absorber reaction
F_V	viscous contributions to shock absorber reaction

Introduction

A main issue in the design of a crashworthy aircraft structure is the absorption of the impact energy at controlled load levels, in order to prevent human injuries derived from excessive accelerations and to reduce the strength required to obtain a protective shell in the occupant compartments.

An adequately designed landing system can give a significant contribution to improve the structural crashworthiness and can represent one the basic locations, together with the aircraft subfloor and the occupant seats, where energy absorption capabilities can be integrated in the aircraft (Ref. 1).

The performance required to a crashworthy landing gear can be evaluated considering the crash scenarios describing potentially survivable impacts, that are characterised by significantly higher vertical velocities than in normal landing operations, up to 12.8 ms^{-1} (*42 fts^{-1}*). In order to obtain a significant energy absorption contribution, the shock absorbing capabilities of the landing systems have thus to be significantly enhanced with respect to limit or hard landing conditions requirements.

It is worth noting, moreover, that the admissible load factors in crash conditions have to be defined through a trade-off between the energy absorbing capabilities and the structural strength required to transmit the ground loads in order to decelerate the whole aircraft mass.

Basing on the aforementioned considerations, crashworthy landing gear can be realised by introducing a specifically designed energy absorption stage, activated in crash impact conditions. For helicopter landing gears, a suitable solution is represented by the adoption of a mechanical energy absorbing device, integrated in the landing gear lay-out (Refs. 3,4,5,6). In the solutions presented in the technical literature, an energy absorbing element is mounted coaxially to the shock absorber cylinder that, in crash conditions, is allowed to slide through the trunnion fitting. A frangible element connects the shock absorber to the trunnion fitting and is designed to fail at a given load level in order to activate the crash energy absorbing stage.

The present work is focused on the development of a landing gear structural lay-out adopting an additional energy absorbing stage, activated in crash conditions, based on the progressive plastic buckling of a metallic tube. The solution investigated is aimed to obtain an adequate energy absorption performance, with limited overall dimensions and without requiring the introduction of frangible fittings between the gear components, thus increasing the structural reliability and simplifying some aspects of the design.

The energy absorption capabilities of thin walled metallic tubes are well reported in the existing literature (Refs. 7,9,10,11,12,13,14,15,16) and are characterised by a typical collapse process, with a force vs. tube shortening curve that presents, in the stationary phase of the process, characteristic oscillations about a constant average load.

Before the onset of the stationary phase, the absorber behaviour, however, is characterised by a typical load peak, at force levels significantly higher than the subsequent averagely constant energy absorption load.

The paper describes briefly the general lay-out of landing gears adopting cylindrical mechanical working energy absorbing elements and reviews the basic characteristics of the collapse processes for axially loaded light alloy tubes.

To activate the energy absorbing stage a specific patent pending triggering mechanism has been developed. The trigger, that plays a key role in the development of a possible design solution, has been validated with experimental tests and studied with numerical analyses, performed with an explicit finite element code.

The possible performances of a landing system adopting the triggered energy absorption stage are evaluated by means of a simplified dynamic model.

Configurations of landing gears adopting energy absorbing crash tubes

The introduction of a cylindrical energy absorbing element in the lay-out of a landing gear can be accomplished by mounting the absorber coaxially to the shock absorber cylinder. This solution allows the integration of the additional shock absorbing stage without increasing the gear overall dimensions.

As shown in Fig. 1-A, the introduction of the cylindrical absorber leads to a landing gear structure that is constituted of five main subassemblies, namely the wheel-axle assembly (WA), the shock absorber cylinder (Sac), the piston (Sap), the crash tube (CT), representing the energy absorption stage activated in crash conditions, and the trunnion fitting (TF) connected to the fuselage structure.

In normal landing operation configuration, the ground loads can be transmitted to the trunnion fitting through different connection elements, as a shear pins (a) (Ref. 3) or collars (Refs. 5,6) (b). These frangible attachments are designed to fail at a given load level, to allow the sliding of the cylinder through the fitting. An alternative solution, followed in this work, can be obtained by transmitting the forces directly trough the cylindrical absorber, thus avoiding the introduction of frangible attachments in the landing gear structural lay-out (c).

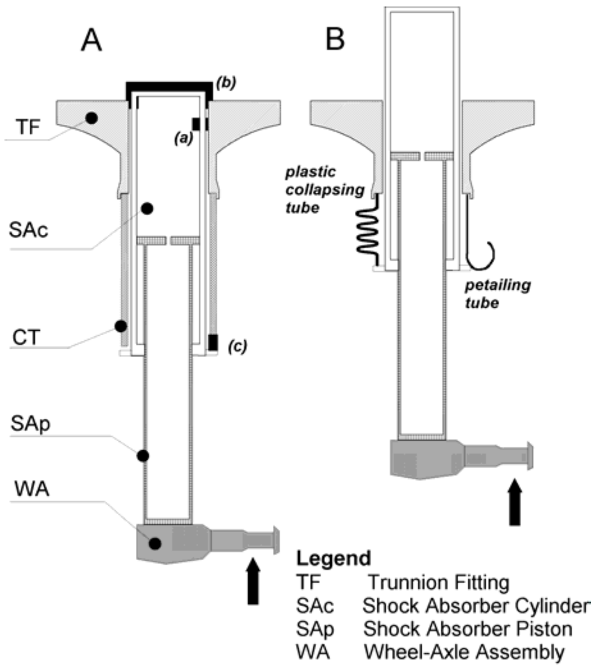


Fig. 1 – Landing system configurations in normal landing operations (A) and in crash conditions (B)

Fig. 1-B show the crash configuration of the landing system. The cylinder slides through the trunnion fitting while the additional energy absorbing stage collapse progressively. Solutions adopting a longitudinally fracturing petailing tube or a carbon composite elements are described in literature (Refs. 3,5,6). A light alloy crash tube undergoing plastic buckling has been chosen as energy absorbing member in the solution proposed in this work.

Collapse process of axially loaded metallic tubes:

simplified models and experiments

The energy absorbing capabilities of thin walled axially loaded metallic tubes have been accurately described by many authors in scientific and technical literature.
The collapse mode is characterised by the progressive development of folds according to a axi-symmetric (concertina mode) or non-symmetric (diamond mode) pattern along the tube walls. The resulting force vs. tube shortening curve oscillates about a well defined average value (Ref. 4).
The specific absorbed energy, defined as the ratio of the absorbed energy to the mass involved in the progressive deformation of the element, can reach values in the 20÷50 KJ/Kg range for aluminium alloy cylindrical tubes. This level is exceeded only by absorbers realised in composite material (Ref 8), and can be thus considered adequate to realise light and efficient energy absorber elements.

To predict the average collapse load, analytical or empirical formulations have been developed by many authors (Ref. 9,10,11,12,13,14,15,16). Most of the approaches are based on simplified mechanical models developed for the axi-symmetric or the non-symmetric collapse mode. Few other authors developed empirical models [6, 11]. In these formulations, generally, the average collapse load is expressed as a function of the plastic collapse unit moment of the thin wall, M_0, that can be estimated by knowing the stress in plastic range, σ_0, of the material (Eq. 1).

$$M_0 = \frac{1}{4}\sigma_0 t^2 \qquad \text{Eq. 1}$$

The ratio of the average load, P_m, to the moment M_0 is typically found to be a function of the diameter to thickness ratio, D/t. The three following expression are reported from Ref. 10 (Eq. 2), Ref. 8 (Eq. 3) and Ref. 11 (Eq. 4), respectively.

$$P_m/M_0 = \frac{8\pi^{1.5}(0.5\,D/t)^{0.5}}{3^{0.25}\cdot\left(0.86 - 0.37(0.5\,D/t)^{-0.5}\right)} \qquad \text{Eq. 2}$$

$$P_m/M_0 = 72.3(D/t)^{0.32} \qquad \text{Eq. 3}$$

$$P_m/M_0 = 86.14(D/t)^{0.33} \qquad \text{Eq. 4}$$

Fig. 2 compares the averaged collapse load predicted by the formulations given in Eqs. 2-4, with a series of experimental data collected from Ref. 6, Ref. 11 and Ref. 14, referred to Al/Si light alloy tube. Some experiments carried out at the Dipartimento di Ingegneria Aerospaziale of Politecnico di Milano are also considered.

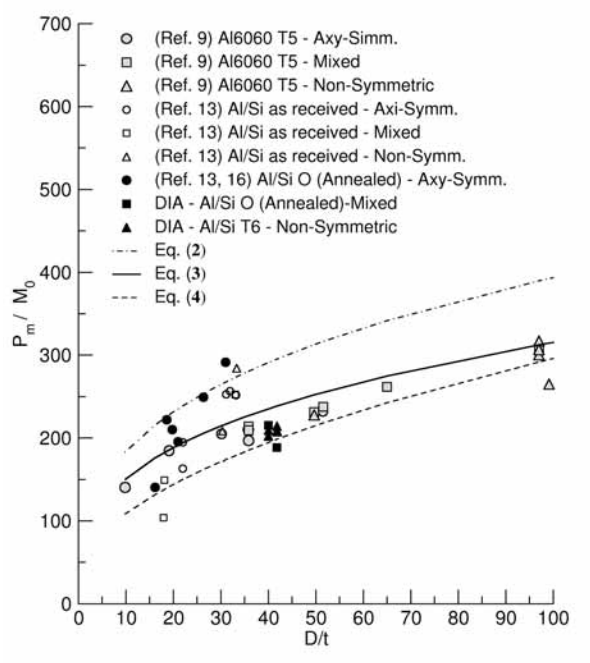


Fig. 2 – P_m/M_0 experimental values (static and dynamic tests) and predictions of the formulations available in literature for Al/Si crash tube

The predictions derived from Eq. 2 appear to underestimate, with few exceptions, the experimental values. It has to be considered that the correlation shown in Fig. 2 does not consider the material strain rate sensitivity, for dynamic tests, and adopt the yield value, σ_Y, as mean value of the stress in plastic range. Both approximation leads to underestimate the P_m/M_0 value attributed to the experimental tests. By introducing the proper corrections Eq. 2, derived from Ref. 10, can give a appreciable preliminary estimation of the collapse load.

The simplified mechanical model in Ref. 10, can give also an estimation of the wavelength, λ, of the oscillations about the mean value in stationary collapse phase:

$$\lambda = \left(\pi t D / 2\sqrt{3}\right)^{\frac{1}{2}} \qquad \text{Eq. 5}$$

The amplitude of the oscillations can vary according to the tube geometrical characteristics and the obtained collapse mode. The maximum loads during the stationary phase, P_o, is determined by the amplitude of oscillations and can be estimated, from the experimental results available, between 1.3 and 1.6 times the average collapse load value.

On the other hand, the initial phase of the collapse is characterised by a load peak that can be significantly higher than the average collapse load. This peak value is determined by the critical buckling load of the element and is more difficulty predictable than the average load, as it is sensitive to the constraint conditions and to the shape and thickness imperfections of the specimens, as well as to other parameters relevant to the test conditions. By applying the empirical expression given in Ref. 13, the ratio of the load peak to the average load, P_A/P_m, for axi-symmetric collapse modes, rises linearly with the D/t parameter, from 1.3 to over 3.5 in the range of D/t between 10 and 100. According to Ref. 8, light aluminium alloy cylindrical tubes can obtain load peak to average load ratios up to 2.5.

A design solution developed with a novel triggering mechanism

Basing on the considerations presented in the previous review of metallic tubes failure mechanisms, the force vs. tube shortening curve of these elements can be divided in an initial phase, characterised by a load peak of hardly predictable amplitude and a stationary phase, where the sustained force oscillates about a well defined and, at some extent, predictable load value. The energy absorbing capabilities observed in stationary phase of the collapse indicate that these elements can be efficiently adopted to realise an additional shock absorbing stage to be integrated into a landing gear structure.

On the other hand, the initial load peak, characterising the tube collapse and that is also shown by other energy absorber typologies (8), can affect significantly the performance of the landing system in crash conditions.

A typical solution to reduce the initial load peaks implies the realisation of cut-outs or bevels at one end of the energy absorber.

These solutions reduce the critical load of the tube and are to be carefully verified in order to be adopted if the ground loads in normal landing operations have to be transmitted through the energy absorbing member.

To develop a solution not requiring the introduction of frangible attachments and not reducing the crash tube strength a triggering mechanism has been developed.

The mechanism has to be located at the lower end of the shock absorber cylinder and is based on two rings mounted coaxially to the cylinder, shown in Fig.3.

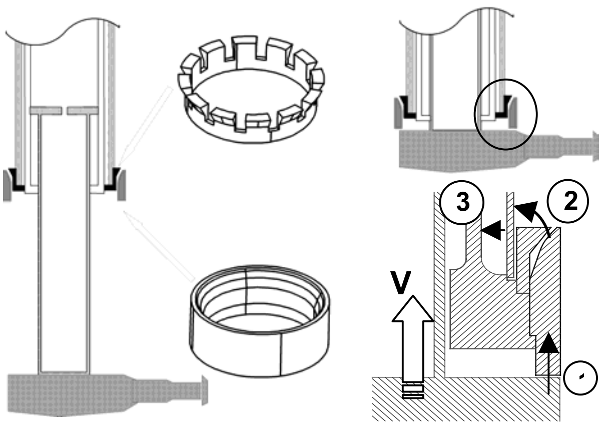


Fig. 3 – Triggering system

The crash tube has to be inserted between the lower border of the cylinder and the trunnion fitting, so that it turns out to be surrounded by the wedged flanges at the lower end. The moveable ring, with a properly shaped inner surface, is mounted coaxially to the previously described flanged collar.

In the considered application, the trigger is activated at the piston bottoming, by a surface fixed to the piston or to the wheel-axle assembly that impacts the lower end of the moveable ring and pushes it upwards (Fig. 3-1). The inner shape of the movable ring is designed to force the rotation of the wedged flanges around the border of the collar (Fig. 3-2) and the wedges squeeze the crash tube promoting the first fold development (Fig. 3-3).

Fig. 4 shows how the triggering mechanism and the energy absorption tube can be introduced in the structural layout of a typical landing gear. Cut-outs can be realised in the moveable ring to house the torque arm and the drag strut lugs.

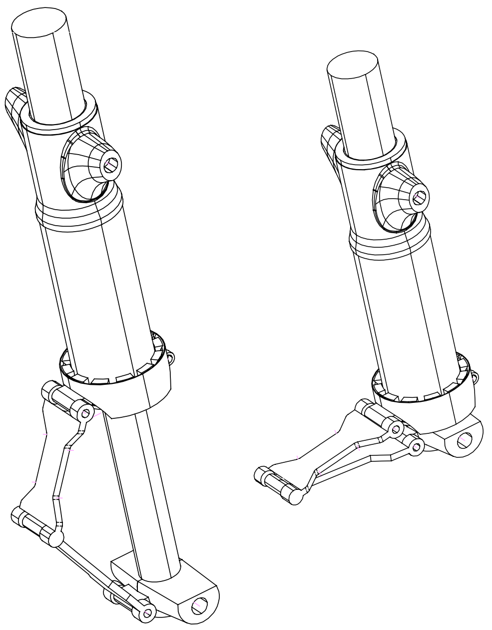


Fig. 4 – Installation of the additional energy absorbing stage

In the solution investigated in this work, the triggering device is thus activated only at piston bottoming, after the full hard landing stroke of the shock absorber has been exploited.

The landing gear structural lay-out does not require the introduction of frangible attachments, as the axial load, before the trigger activation, can be reliably introduced in the fuselage by means of the energy absorbing member.

The shock absorber can fully contributes to the energy dissipation before any failure is induced in the landing system.

On the other hand, this solution requires that the shock absorber response has to be designed taking into account the piston velocities experienced in a crash impact condition. At high velocities, the viscous contribution to the shock absorber response has therefore to be cut-off at a given force level. Without a cut-off provided by a relief valve or a blow-out plug, introduced in the shock absorber design, forces would rapidly increase over the maximum admissible levels in crash conditions and would eventually induce the tube to collapse before the piston bottoming.

Experimental validation of the triggering mechanism and numerical analyses

Experimental methodology. The trigger working mechanism and the obtainable initial peak reduction have been evaluated by means of an experimental test set up to reproduce the basic working conditions of the device. A test arrangement and a test fixture have been designed to meet this requirement.

The flanged collar has been realised on a aluminium alloy disk, to allow a simple connection with the other parts of the test arrangement. The moveable ring has been as well obtained in aluminium alloy with a properly machined inner surface. A dummy shock absorber cylinder has been realised in mild steel and bolted to the collar through a fixing plate. The crash tube is mounted coaxially to the dummy shock absorber and housed between the flanges at the lower end (Fig. 5).

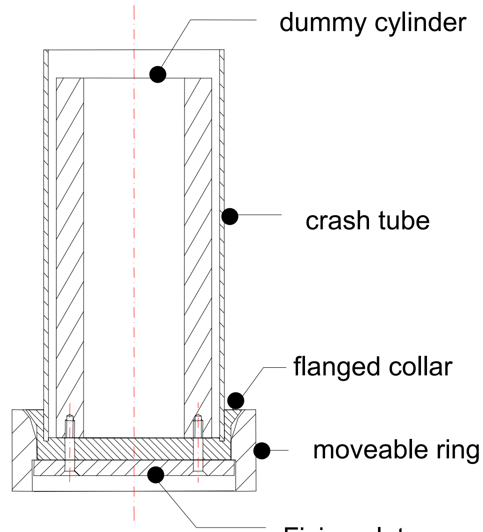


Fig. 5- Test arrangement

The test fixture, shown in Fig. 6, has been designed to carry out the validation test with a drop tower apparatus. The impacting mass lower surface (Fig. 6-a) plays the role of the impacting surface fixed to the piston in the landing gear. To allow the sliding of the dummy cylinder, the other end of the crash tube is connected to a steel hollow cylindrical base (Fig. 6-c). The crash tube end opposite to the trigger is mounted on a shoulder realised on the cylindrical base and fixed with a ring (Fig. 6-b). The forces transmitted by the crash tube to the cylindrical base are measured by a set of four load cells (Fig. 6-d).

The tests have been performed on an Al/Si solution heat-treated and artificially aged tube with 112 mm average diameter (*4.4 in*) and a thickness of 2.75 mm (*0.11 in*). A characterisation test performed on a specimen after an identical heat treatment has given a yield stress, σ_0, of 215 Mpa (*31.18 Ksi*). An impacting mass of 250 kg (*551.2* *lbs*) has been used with a drop height of 4.5 m (*14.76 ft*).

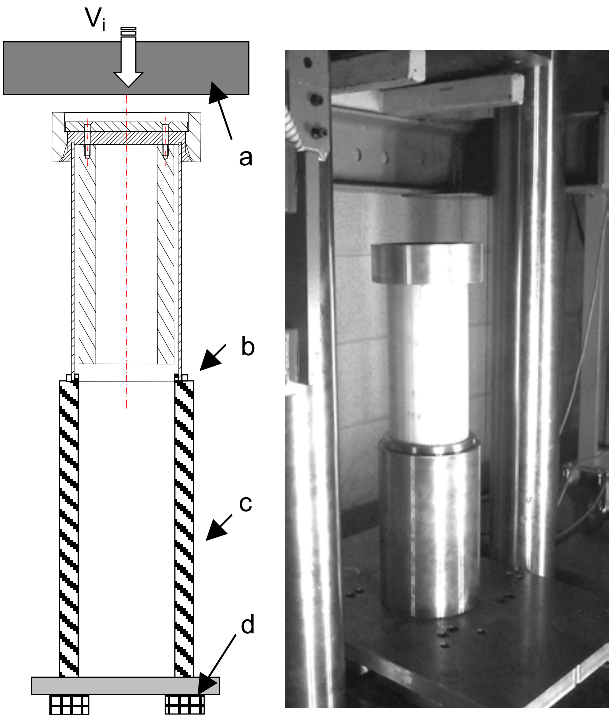


Fig. 6 – Test fixture scheme and set-up

Test results. Two reference dynamic tests have been performed on the same tube without the triggering mechanism with identical test conditions.

Fig. 7 reports the force vs. tube shortening curves, obtained from the load cell signals sampled at 12500 Hz and digitally filtered with 1000 Hz cut-off frequency. Fig 11 reports, as well, the curve obtained in a quasi-static test performed on an identical tube typology by means of an hydraulic test fixture.

The sequences shown in Fig. 7, referred to the two reference dynamic tests indicate that non-symmetric collapse mode have been obtained. The average collapse loads have been respectively of 84300 N (*18951 lbf*) and 85700 N (*19270 lbf*). The specific energy absorbed is about 35 KJ/kg. Initial peaks of about 180000 N (*41600 lbf*) have been obtained.

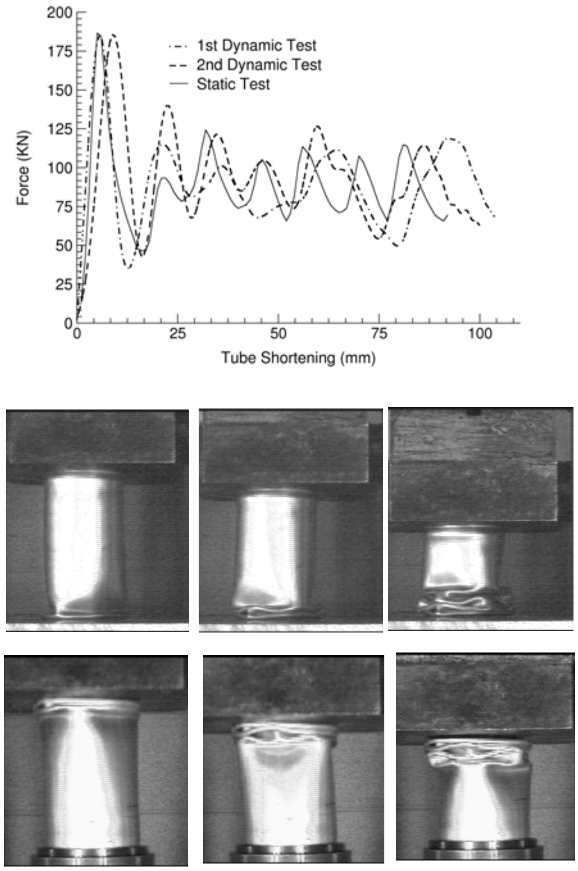
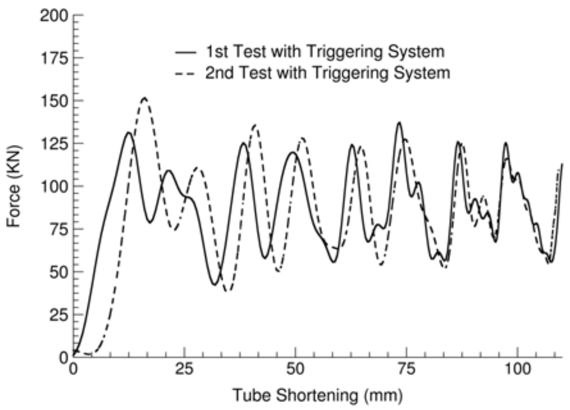


Fig. 7 – Reference tests performed without triggering mechanism

Two tests have been performed with the triggering mechanism and the designed text fixture. Load cell signals have been sampled and then digitally filtered as in the reference tests.

The obtained curves and collapse mode are shown in Fig. 8.

It can be observed that the trigger promoted a regular axi-symmetric collapse mode in both cases, with average collapse loads of 81800 N (*18400 lbf*) and of 83200 N (*18700 lbf*), slightly lower than in the reference tests. The maximum value of the force obtained in the tests are 131400 N (*29500 lbf*) and 151000 N (*33900 lbf*), respectively in the first and in the second test.



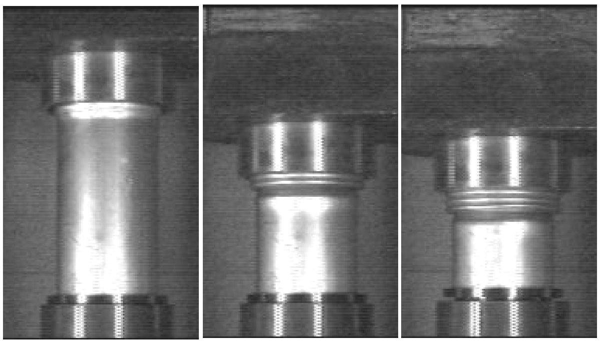


Fig. 8 – Trigger validation tests

Tab. 1 summarises the averaged dynamic tests results. Without trigger an initial force peak of 185000 N (*41600 lbf*) has been obtained, 48% percent higher than the maximum force in the stationary phase, while the tests performed with the trigger obtained, averagely, a peak force 141000 N (*31700 lbf*), that is only 12% higher than the maximum force in the stationary phase. The triggering mechanism reduced thus the load peak and obtained an initial load only slightly higher with respect to the maximum load in the stationary collapse phase.

Fig. 9 shows the regular collapse mode promoted, for the considered tube material and geometric characteristics, by the triggering device.

Tab. 1 – Trigger efficiency evaluation

	Reference tests	Triggering mechanism
P_m (N) (*lbf*)	85000 (*19100*)	82500 (*18550*)
Pa/P_m	2.2	1.7
P_o/P_m	1.47	1.51

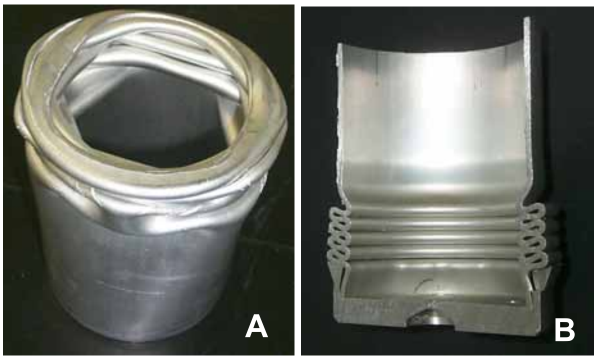


Fig. 9 – Collapse modes in the reference tests (A) and with the triggering mechanism (B)

Numerical analysis. Explicit finite element simulations of the experimental tests has been performed in order provide a better insight of the trigger working mechanism as well as to make available a modelling technique to study in detail the installation conditions, to analyse the adequacy of the structural design and to evaluate the landing gear performances with different landing attitudes.

A finite element model of a quarter of the tube and of the test arrangement, described in Fig. 5, has been developed and solved by using the HKS/Abaqus Explicit code. The model consists of four deformable bodies, mutually interacting with properly calibrated contact algorithms, namely the *crash tube*, the *flanged disk*, the *moveable ring* and the *fixing plate* (Fig. 10-a). No sign of interference between the collapsing tube and the dummy shock absorber cylinder was detected in the tests, so that the model has been realised not including the inner tube. The *crash tube* has been modelled with 2340 shells, with a mesh refined towards the end introduced in the triggering mechanism. Globally, 4605 solid bricks have been used to model the other three deformable bodies. The plastic hinges zones, at the flange roots on the flanged ring, have been modelled with four bricks across the thickness.

An appreciable numerical-experimental correlation has been obtained either with respect to the deformed shape of the crash tube and of the triggering mechanism flanges (Fig. 10-B,C) , as well as considering the force time history transmitted at the end of the tube opposite to the trigger (Fig. 11).

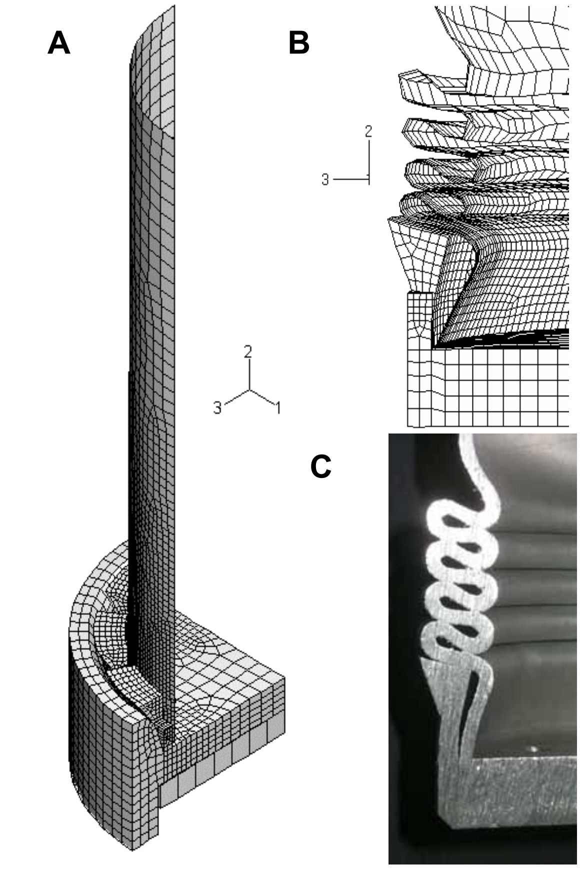


Fig. 10 – HKS/Abaqus explicit finite element model (A) and numerical-experimental correlation of the collapse mode (B-C)

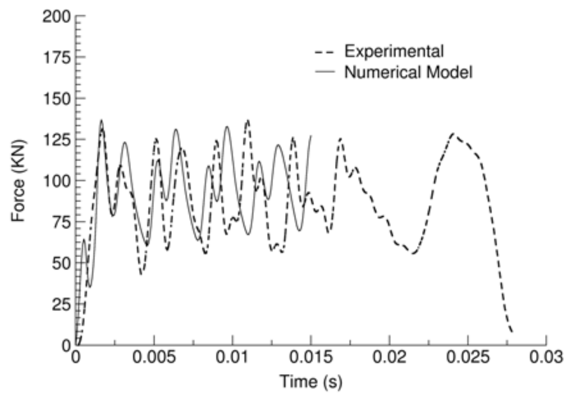


Fig. 11 – Numerical and experimental force vs. time curve for the crash tube test with the triggering device

As it can be observed from Fig. 12, referred to two time instants during the first peak oscillation, the numerical analysis indicate that the trigger works during the whole duration of the first fold initial development. The reduction of the initial load peak value is accomplished by promoting the development of plastic strain on the crash tube walls.

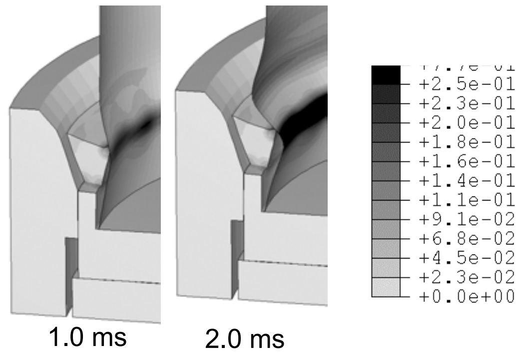


Fig. 12 – Contour of plastic strains during the first fold development

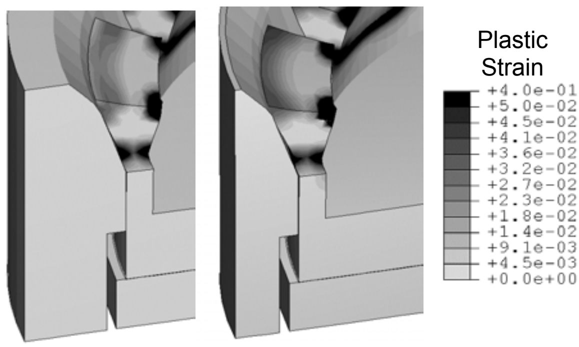


Fig. 13 – Contour of the plastic strain at 15 ms obtained with the tested configuration simulation and a thinner moveable ring

Fig. 13 compares the final plastic strain contour, at 15 ms, of two simulations performed by modelling the moveable ring adopted in the tested configuration and a thinner moveable ring.

In both cases the numerical simulations predict that plastic strain develop only on the tube walls and flanges (at the root, where the plastic hinge forms, and at the sharp edges that are in contact, respectively, with the moveable ring inner surface and with the outer tube surface). Either the moveable ring, as well as the flanged disk shoulder remain in elastic range during the impact, so that permanent strain or damage development have not to be expected in the moveable ring and in the fitting of the trigger to the cylinder.

The results shown in Fig 13 indicate that the radial dimensions of the device can thus be significantly reduced with respect to the tested configuration.

Preliminary evaluation of the crashworthy landing gear performances

The performances obtainable with the triggering mechanism have been evaluated by means of a simplified dynamic model realised considering a basic landing gear layout, shown in Fig. 14.

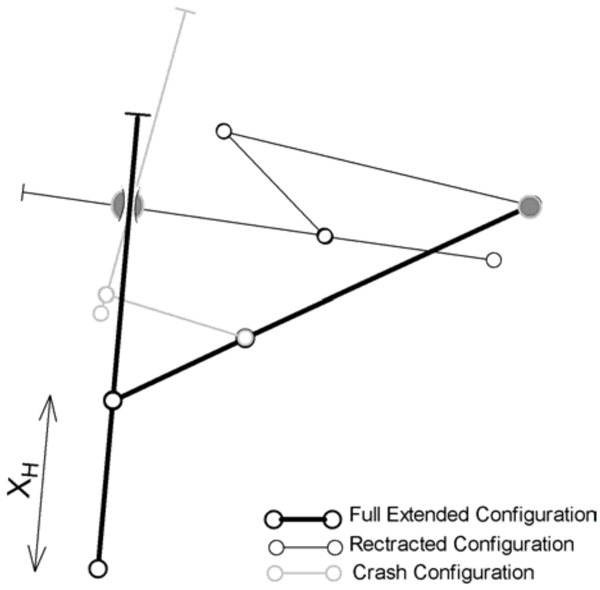


Fig. 14 – Kinematical layout of the landing system considered for the performance preliminary evaluation

The values reported in Tab. 2 have been chosen as representative requirements for an helicopter landing gear. The chosen crash load factor, n_c, is referred to the average collapse load of the crash tube.

Tab. 2 –Reference requirements for the landing system

	Limit Landing	Hard Landing	Crash Landing
Sink Speed, ms^{-1} (*fts^{-1}*)	3.05 (*10*)	4.57 (*15*)	12.8 (*42*)
Residual Lift Ratio Considered	0.67	1.0	1.0
Landing Gear Load Factor	2.33	3.5	4.4 (averaged)

Considering the role played by the triggering mechanism, the safety margin corresponding at the original load peak of the tube has been exploited to adopt a crash tube with an average collapse load relatively low, in order to reduce the maximum loads introduced in the fuselage.

By assuming a shock absorber efficiency of 0.85 in hard landing conditions and a tyre efficiency of 0.47 (Ref. 17), a design hard landing stroke, X_H, of 330 mm (*13 in*) has been estimated and has been adopted as a reference stroke for the gear.

The dynamic model used for the evaluation of the performances is schematised in Fig. 15. The equations of motion of the dynamic system have been solved by adopting the Matlab© ordinary differential equation solvers.

A basic response has been adopted for the shock absorber.

Fig. 16-A shows the shock absorber reaction in isothermal and adiabatic compression processes. The curves report the ratio of the reaction to the static load and are given as functions of the piston stroke to reference stroke ratio, x_{SA}/X_H.

Fig. 16-B shows the ratio of the viscous response to the static load, plotted as a function of the ratio of the piston velocity to the hard landing velocity, V_H. A quadratic relation describes the behaviour up to a cut-off velocity.

The viscous cut-off, that is needed to limit the viscous reactions at high velocity, has been set at a piston velocity higher than the maximum level evaluated by means of the simplified model in hard landing conditions.

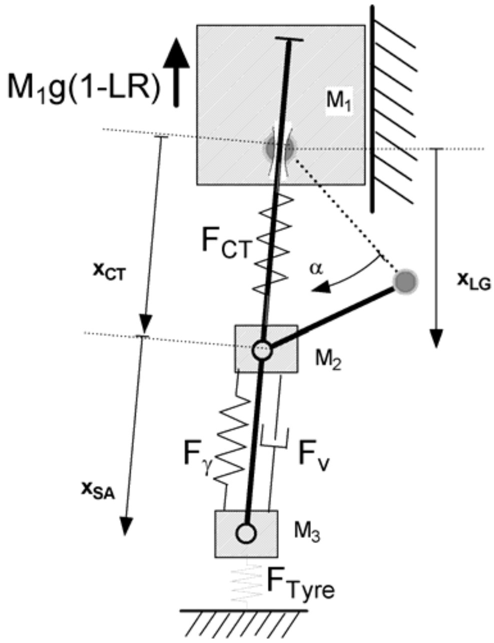


Fig. 15 – Simplified dynamic model of the landing system drop test

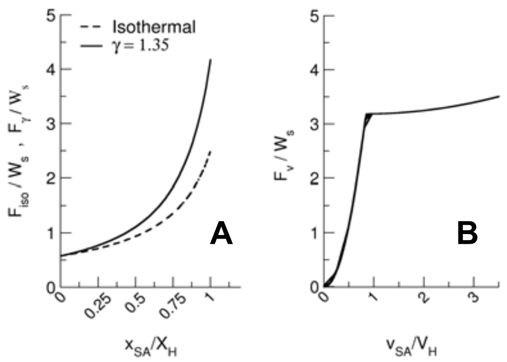


Fig. 16 – Shock absorber responses introduced in the numerical model

A crash tube representative response has been introduced, with a sinusoidal curve and a maximum load in the stationary phase, P_o, set at 1.5 times the average load P_m. The peak load has been eliminated, considering the trigger activation that has been set to 0.95 times the design hard landing stroke X_H. A penalty force decelerates the piston at the trigger activation. A crash tube maximum available stroke has been set to $0.6X_H$.

Fig. 17 shows the curve of the load factor vs. the ratio of the landing gear overall stroke to the reference stroke, x_{LG}/X_H. The response is compared with the hard landing performance. The curve is truncated at the available stroke of the crash tube.

Fig. 18-A reports the piston stroke, x_{SA}/x_{LG}, and the crash tube stroke, x_{CT}/x_{LG}, in crash conditions plotted vs. the overall landing gear stroke, x_{LG}/X_H. The piston stroke in hard landing is also reported.

Fig. 18-B shows the variation of the angle of the shock absorber axis with respect to the vertical axis. The angle variation is plotted vs. the overall landing gear stroke, x_{LG}/X_H .

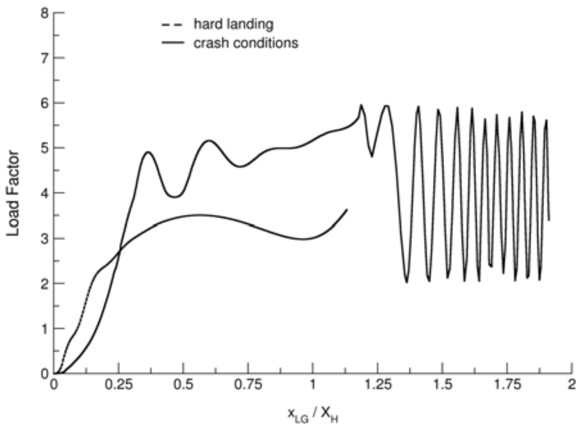


Fig. 17 – Numerical load factor vs. landing gear stroke in crash and hard landing conditions

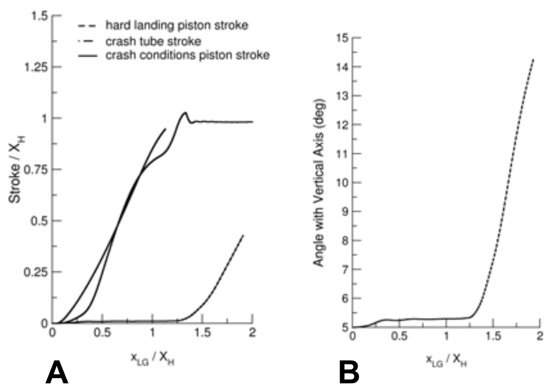


Fig. 18 – Piston and crash tube numerical strokes (A) and angle with vertical axis (B) vs. landing gear stroke

The numerical evaluation indicates that about the 30% of the whole impact energy in crash conditions is absorbed by the landing system.

The maximum load factor is 6.5 and occurs at the level of the maximum load oscillations during the stationary phase of the tube collapse process. It can be observed, from Fig. 18, that the hard landing stroke is lower than the trigger activation stroke, thus indicating that the trigger can be installed without interfering with the piston stroke in normal landing operations.

The viscous response shown in Fig. 16-B can be replaced by the cut-off provided by a blow-out plug. These devices can lead to a slightly lower energy absorption performance but can be, however, more easily integrated in the shock absorber layout. Fig. 19 is referred to the simulation in crash conditions obtained introducing an irreversible drop in the viscous coefficient characterising the shock absorber response. With this modification an energy absorption corresponding to the 27.0% of the whole impact energy is obtained.

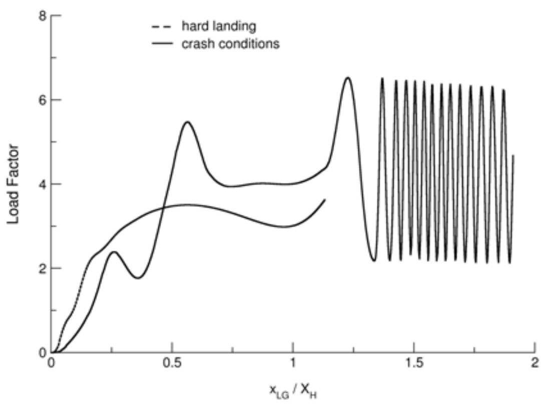


Fig. 19 – Numerical load factor vs. landing gear stroke in crash and hard landing condition with a different viscous response

Concluding Remarks

The developed triggering mechanism for inducing the collapse of axially loaded metallic tubes has been experimentally validated and investigated by means of numerical analyses. The peculiar feature of the triggering device is the capability of removing almost completely the initial load peak in the force vs. tube shortening response without requiring the realisation of cut-outs or bevels in the tube walls.

The obtained results has allowed to study a solution for a crashworthy landing gear, adopting a crash tube without requiring the introduction of frangible attachments in the gear lay-out. This solution can be considered particularly reliable from the structural point of view and presents a relatively simple structural layout with respect to other solutions adopting a cylindrical energy absorber device.

In crash landing conditions, the performances obtained with the adoption of a viscous cut-off provided by a relief valve or by blow-out plugs, can be appreciable and a significant contribution to the absorption of the whole impact energy in crash conditions is obtained. The significant margin of safety of the crash tube before the trigger activation can be exploited to reduce the average design load factor of the crash tube, so to reduce the influence of load oscillations on the maximum load levels introduced in the fuselage.

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