DYNAMIC BUCKLING TESTS OF CYLINDRICAL SHELLS IN COMPOSITE MATERIALS

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Abstract

This paper describes an experimental investigation of the elastic dynamic buckling of carbon fiber reinforced plastic cylindrical shells subjected to pulse axial compression. The critical impulse is applied using a horizontal crash sled, so to obtain axial compression shape similar to a half-sine and duration of the impact load of approximately 0.1 s.

The test results are reported and analyzed. The dynamic buckling load of the initially imperfect shell is related to its static buckling load and the ratio is less than unity. Since the common practice is to assume that dynamic buckling loads are higher than the static ones, which means that static design is safe, extra caution and careful design is recommended.

1 Introduction

The stability of composite cylindrical shells is of fundamental interest in aircraft and missile design as thin-walled cylindrical shells have constituted primary structural parts for many years and because the composite materials are nowadays extremely attractive due to their considerable strength-to-weight ratio. These structures are subjected to various loading both static and dynamic. The investigations in the past two decades were primarily confined to the static stability. This effort has led to a reasonably good understanding of the response of the composite shells to static loading, including the effects of initial geometric imperfections and lamina stacking sequence.

But there is also the need to design structures that have to withstand time dependent dynamic loads, sometimes quite severe, and thus may be susceptible to “dynamic buckling”.

Though many classes of phenomena are encompassed by the term “dynamic buckling” [1, 2], the most common, that is also used in the present study, is related to the behavior of the structures subjected to pulse loads, and represents the loss of stability or the deformation of a structure to unbounded amplitudes as the result of a transient response to an applied pulse, i.e. dynamic buckling under impact loads. In particular, according to the concept of Budiansky [3], the dynamic response is such that, for a fixed value of time duration, for less than a certain value, called dynamic buckling load, the response is bounded, whereas it is unbounded for a load greater than the dynamic buckling load. So, the dynamic buckling of a structure is associated with the state at which a small change in the magnitude of the loading leads to a large change in the structure response.

The dynamic buckling was investigated numerically in several studies, where the equations of motion were solved for various values of the loading and the value at which there was a significant jump in the response was assumed critical. Therefore implementation of this criterion required to solve the equations of motion for different values of the loading parameter, then plot the displacement amplitude versus loading curve from which the critical loading value was determined [4-7].

Instead, very few experimental data can be found in literature, regarding dynamic buckling. Abramovich and Grundwald [8] performed experimental studies with axially impacted laminated composite plates; Zimcik and

Tennyson [9] investigated experimentally the dynamic response of thin-walled circular cylindrical shells to transient dynamic square-pulse loading of varying time duration; Cui, Cheong and Hao [10] performed an experimental study of dynamic buckling of mild steel plates under fluid-solid slamming; Yaffe and Abramovich [11] investigated experimentally the dynamic buckling of aluminium cylindrical stringer stiffened shells under axial dynamic applied loading.
Although the static buckling of such structures is fairly well understood, a clear understanding of dynamic buckling is lacking. Existing analyses hardly provide any general guidelines for design against dynamic buckling and no test results were obtained to form a sound experimental database for this phenomenon.
In any case, in most of these numerical and experimental studies, it was shown that the permissible load intensity is load duration dependent which reveals a special feature of pulse buckling. Consequently, whereas in static buckling the maximum safe load is determined, in pulse buckling the load amplitude determines the maximum safe duration of its application.
It was shown, that in general the tested structures experience dynamic buckling loads larger than the relevant static buckling loads, provided their duration is very short. So, the ratio of the dynamic buckling to the static buckling of the structure is greater than unity. On the opposite, for longer duration, the dynamic buckling load is smaller than the static one, which means that taking the static buckling load as the design point for dynamic problems may be misleading.
This paper presents an experimental study of the dynamic buckling of carbon fiber reinforced plastic (CFRP) thin-walled cylindrical shells. In particular two cylindrical shells were tested using a horizontal crash sled.

2 Cylindrical shells

The examined cylindrical shells, manufactured and provided by AGUSTA, are made of carbon fiber reinforced plastics (CFRP) and are characterised by a mean radius of 350 *mm*, a length of 700 *mm* and a thickness of 1.32 *mm*, with a nominal radius-to-thickness ratio of 265. The stacking sequences is [0°/45°/-45°/0°], where 0° corresponds to the axial direction of the shells and where the stacking sequences is taken from outside to inside. Each ply is 0.33 *mm* thick and has the material properties reported in Table 1.

Elastic modulus E_{11} [N/mm^2]	52000
Elastic modulus E_{22} [N/mm^2]	52000
Shear modulus G_{12} [N/mm^2]	2350
Poisson's ratio ν_{12}	0.302
Density [kg/m^3]	1320
Thickness [*mm*]	0.33

Table 1. Mechanical properties of the CFRP ply.

The CFRP cylindrical shells were already investigated experimentally and numerically for static buckling under axial compression, under torsion and under combination of axial compression and torsion [12-15], during the European Research Program *"Design and validation of imperfection-tolerant laminated shell structures (DEVILS)"*. The static buckling tests were performed using an equipment that allows axial and torsion loading, applied separately and in combination, in a position control mode, and includes a laser scanning system for the measurement in situ of the geometric imperfection as well as of the progressive change in deformations.
The buckling of the shell occurred at a load of 140 *kN*, suddenly, intensively audibly and with the load dropping drastically at about 50 *kN*, as reported in Figure 1. The axial compression post-buckling mode was diamond-shaped, 10 half-waves around the circumference and 2 half-waves along the length. Figure 2 reports a photo of the post-buckling mode during the axial compression static test.

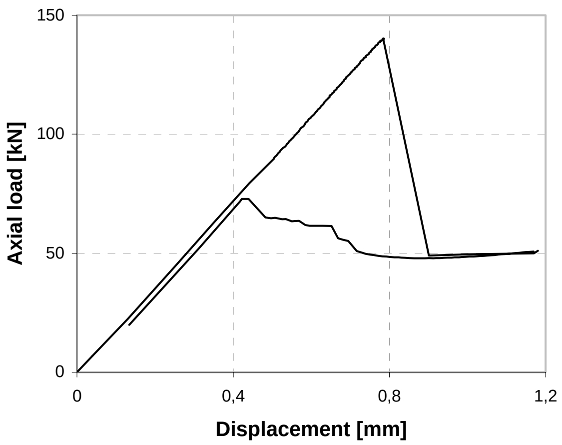

Fig. 1. Static test:
diagram of axial load versus displacement.

Fig. 2. Post-buckling mode of the
axial compression static test.

3 Equipment

To perform the dynamic tests here presented, a deceleration horizontal sled running on two horizontal rails was used. The equipment is normally used for crash tests and presents a maximum length equal to 100 *m*, a maximum mass of 2.5 *ton* and a maximum velocity of 20 *m/s*.

The idea to use a typical crash test equipment for the dynamic buckling investigation is due to the difficult to set-up an ad-hoc testing machine able to guarantee the load sequence requested for dynamic buckling tests, i.e. high loads coupled with a very short application time. Indeed, this practical difficulty is one of the reasons limiting the experimental investigations on the dynamic buckling phenomena.

A photo of the shell clamped on the experimental equipment before the test is reported in Figure 3, while a scheme of the set-up of the dynamic buckling test is reported in Figures 4 and 5. In each test, the whole moving system included the front sled, the specimen and the rear sled.

The initial velocity of the two sleds with the shell in the middle was reached by means of a compressed air piston and was decelerated by means of a pre-calibrated hydro-pneumatic brake that provided the requested pulse (Figure 6). When the front sled entered in the brake and was decelerated, the rear sled was free to move forward, given the axial compression pulse load to the cylindrical specimen.

The length of the loading period, which has a shape similar to a half-sine, can be altered by increasing/decreasing the deceleration of the system through the hydro-pneumatic brake, while the load intensity can be altered by increasing or decreasing the mass of the rear sled.

The behaviour of the shell, subjected to the axial compression pulse load, is represented by the deformation of the structure as result of a transient response to an applied pulse.

To measure the response of the shells, accelerometers with 200 *g* full-scale were placed on the front sled and on the rear sled. The measured acceleration was filtered at 180 *Hz* using a CFC 180 filter [16].

To monitor the tests and to see the buckling shape, high speed film with 1000 photograms per second were taken during the tests.

Figure 7 and 8 show the configuration of the crash equipment prepared for the dynamic bucking tests and the loading device.

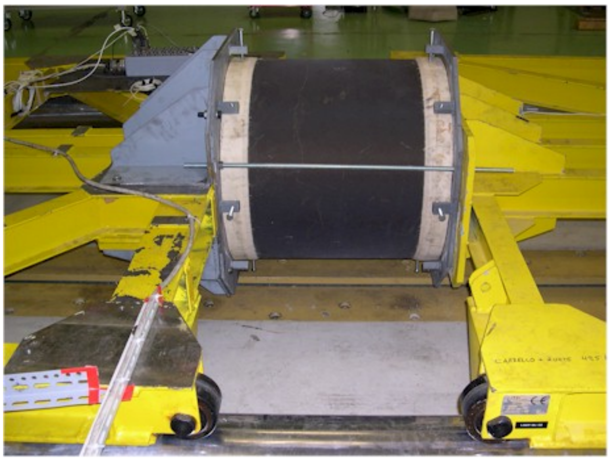

Fig. 3. Shell on the experimental equipment before the dynamic buckling test.

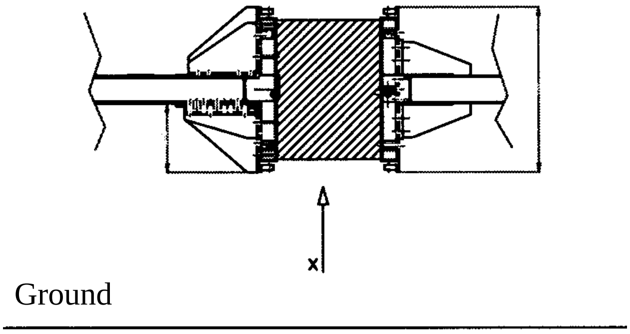

Fig. 4. Lateral view of the experimental set-up.

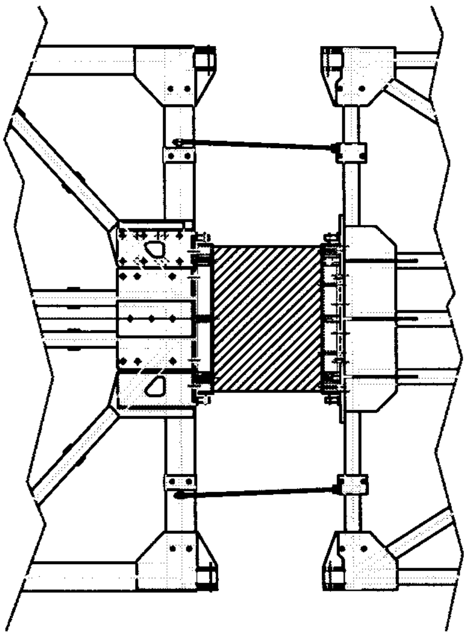

Fig. 5. Top view of the experimental set-up.

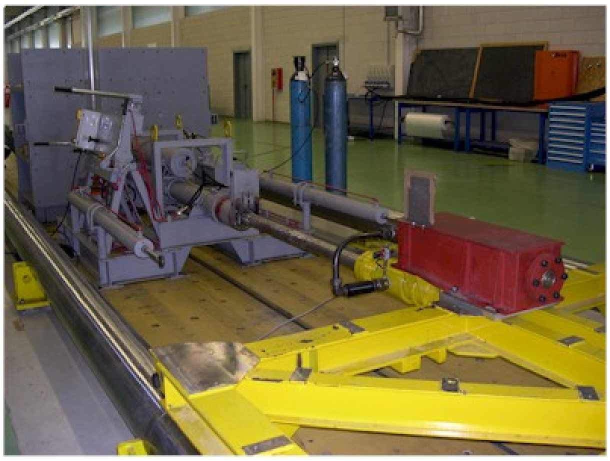

Fig. 6. Pre-calibrated hydro-pneumatic brake.

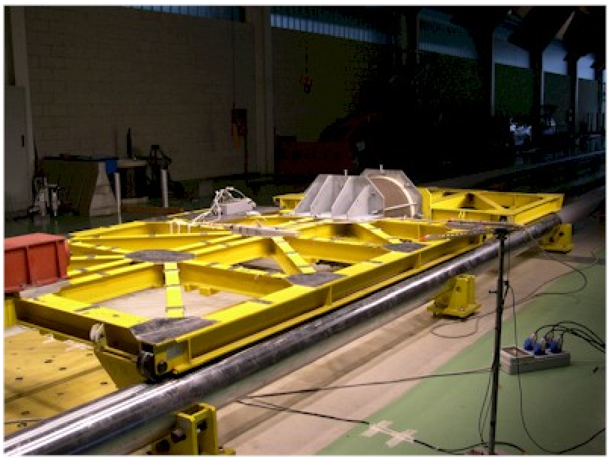

Fig. 7. Configuration of the crash equipment prepared for the dynamic bucking test.

Fig. 8. View of the experimental equipment.

4 Tests results

Two tests were performed on two nominally identical cylindrical shells.
In each test, three accelerometers were placed on the front sled and two on the rear sled, while the mass of the rear sled was equal to 598 *kg*.
The impact velocity was equal to 6.2 *m/s* and the pulse load duration was equal to 0.1 *s*.
Figure 9 presents the load versus time duration curve obtained from an accelerometer placed on the front sled, while Figure 10 the same curve obtained from the rear sled.

In the second test, the same equipment was used and two extensometers were added laterally to the cylinder to measure the shortening of the shell.
Figure 11 presents the load versus time duration curve obtained from an accelerometer placed on the front sled, while Figure 12 the same curve obtained from the rear sled. Figure 13 reports the specimen shortening that is equal to 6 *mm*.
The buckling load results equal to 74 *kN*.

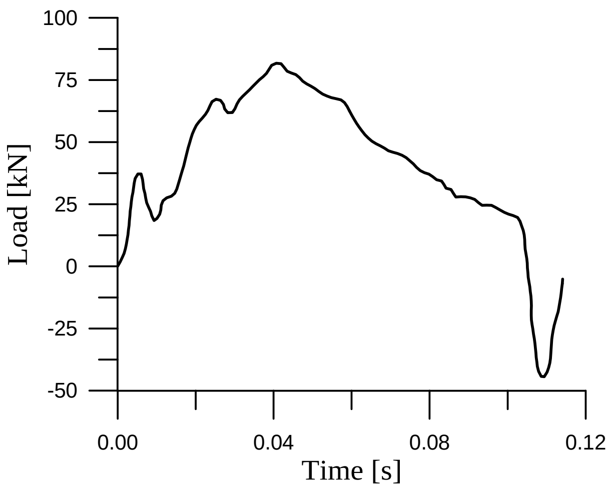

Fig. 9. First test: acceleration on the front sled.

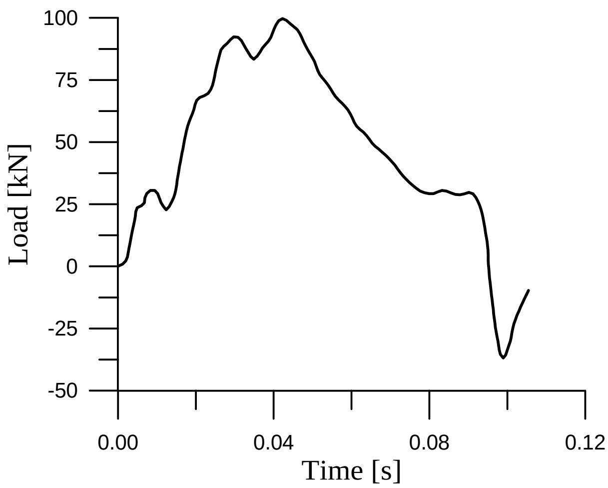

Fig.11.Second test:acceleration on the front sled

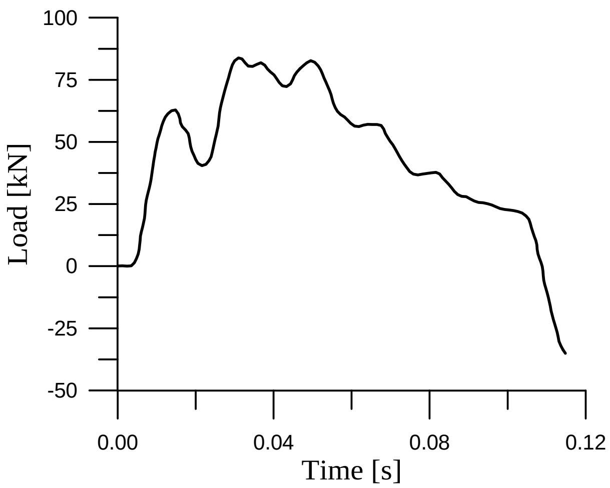

Fig. 10. First test: acceleration on the rear sled.

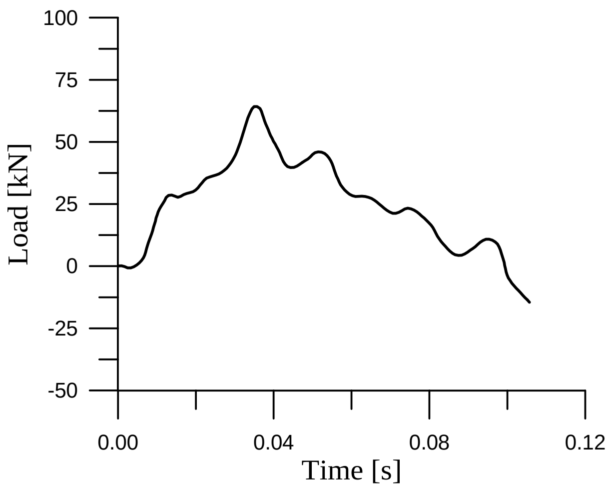

Fig.12. Second test: acceleration on the rear sled

The buckling load is taken equal to the maximum acceleration measured on the rear sled multiplied by the mass of the sled and results equal to 83 *kN*.

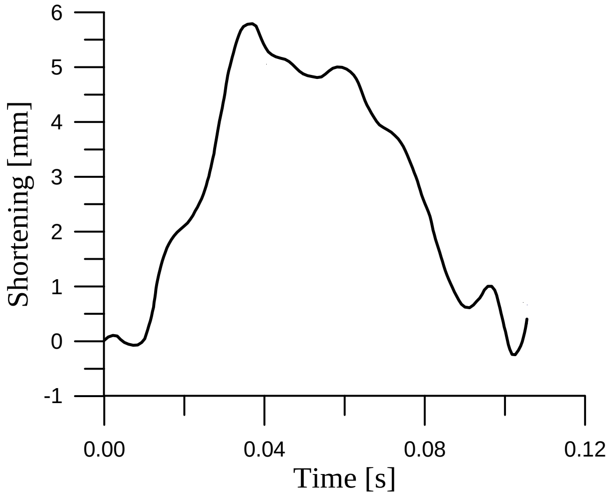

Fig. 13. Second test: specimen shortening.

The mode shape evolution taken during the second dynamic test is reported in Figure 14.
The results indicate that dynamic buckling always take place elastically. The duration of the impact load is measured to be approximately 0.1 *s*. So, the considered pulse load is a typical intermediate velocity impact and the occurrence of buckling take place within the time interval of load application.
The dynamic buckling loads are then compared to the static buckling loads of the shells. The ratio of the dynamic buckling to the static buckling of the shell is equal to 0.58-0.52.
As the cylindrical shells are very sensitive to initial geometric imperfections, the results are presented in a form such that the dynamic buckling load of the initially imperfect model is related to its static buckling load. Thus, explicit dependence on its initial imperfection is bypassed.

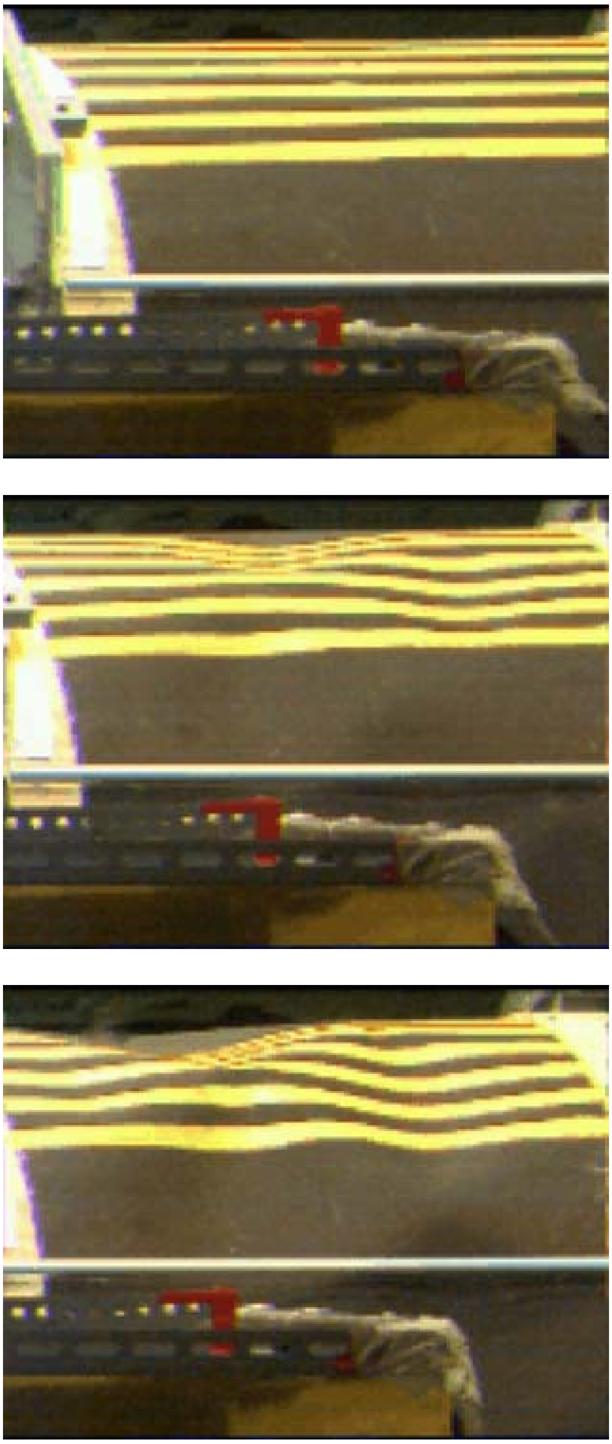

Fig. 14. Mode shape evolution during a dynamic buckling test.

5 Conclusions

This paper presents the preliminary results of an experimental study of the dynamic buckling of carbon fiber reinforced plastic cylindrical shells. In particular, two cylindrical shells were subjected to pulse axial compression. The critical impulse was applied using a horizontal crash sled, so to obtain axial compression shape similar to a half-sine and duration of the impact load of approximately 0.1 *s*.

The test results are reported and analyzed. The dynamic buckling load of the initially imperfect shell is related to its static buckling load and the ratio is less than unity. Since the common practice is to assume that dynamic buckling loads are higher than the static ones, which means that static design is safe, extra caution and careful design is recommended. Indeed, taking the static buckling load as the design point for dynamic problems might be misleading.

The reported results are not complete yet, but they must be considered as a feasibility study of the quite unusual testing set-up here proposed.

In the near future, further tests will be performed changing the loading duration. Pairs of strain gauges will be bonded face to face circumferentially at the mid-height to measure the response and to clearly identify the buckling load.

Besides, as the shape of the duration almost resembled the one of a half-sine, numerical predictions will be computed and a parametric investigation will be performed to find the dynamic buckling loads of the impacted shells and their respective loading durations to yield a ratio less than unity.

The results of the further tests and of the numerical finite element analyses will be reported in due time.

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