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EFFECT OF PULSE DURATION AND SHAPE ON DYNAMIC BUCKLING OF STIFFENED PANELS

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Abstract: Dynamic buckling of stiffened panels under axial compression loading having the form of finite duration pulse was analyzed by finite element modeling. Welding induced defects modifying the skin plate curvature were incorporated. Material degradation in the heat-affected zone was also taken into account. The Budiansky and Roth criterion was employed to predict the collapse load. Various pulse shapes were investigated. The obtained results have shown that the pulse period and profile have severe effects on the buckling strength. For the considered boundary conditions and load pulses up to 56% reduction of the strength was observed in comparison with static buckling.

Keywords: Dynamic buckling, Stiffened panel, Pulse duration, Pulse shape, Geometric imperfections, Heat affected zone

1. Introduction

Stability of stiffened panels is a main concern in many engineering applications like for example in marine and aeronautics sectors [1], [2]. As modern structures are more and more designed to be light and of thin-walled shape, buckling risk constitutes a real problem to undertake while working for increasing the strength-to-weight ratio. Stiffened panels are structures that enable improved strength for a given weight. But, they are subject in service life to various destabilizing loading conditions either static or dynamic and may undergo also various alterations resulting from material degradation or initial geometric imperfections.

Static buckling has been extensively investigated for many types of structures that are prone to undergo instabilities. This has been achieved under various loadings and is now a relatively good understood subject [3]. Buckling of structures under the action of dynamical loads that are suddenly applied has not yet received the same amount of attention, even if in practice this kind of loading occurs very frequently, especially for ships and aircrafts [4], [5].

In the literature different approaches have been presented by various authors to describe how the dynamic buckling load can be assessed. Simitses [6] classified the various concepts and methodologies used in estimating critical conditions for suddenly loaded elastic systems in two main approaches: equations of motion based methods and energy based methods.

Energy approach is applicable mostly to conservative systems having a low number of degrees of freedom, whereas the approach using the equations of motion [7] seems to be more suited for continuous structures like stiffened panels that are characterized by a huge number of intervening degrees of freedom. In this last approach, the equations governing the instability problem are solved for various values of parameters defining the loading to obtain the response of the system. The load parameter at which a large change happens in the response is called critical. This approach has become prominent in the field of dynamic buckling because of its ability to be easily adapted to computational methods such as the general methods based on finite element modeling.

Considering the case of an impacted beam Wooseok and Waas [8] have shown that, unlike the static case, dynamic buckling resulted in localized non-uniform buckle mode shapes due to the interactions between the in-plane and out-of-plane deformation responses. The authors concluded that dynamic buckling cannot be resolved by considering only static buckling analysis as it was found to be more severe.

Dynamic buckling of beams and plates subjected to axial impact was also investigated by Weller et al. [9]. They performed numerical calculations to determine the Dynamic Load Factor (DLF) of in-plane impacted beams and plates. The DLF is defined as the ratio between the dynamic buckling load and the static buckling load for the same structure and boundary conditions. Good agreement was observed between the predictions and experimental results. The DLF was in general larger than unity, both for beams and plates. However, in the presence of certain values of relatively large initial geometric imperfections and for pulse durations of the applied load that are close to the first period of natural flexural vibrations of either the beam or the plate, the DLF was found to be much smaller than unity which signifies that dynamic buckling could be more severe than static buckling. This observation has been found to be valid also for axially impacted composite plates [10]. The experimental tests these authors have performed on laminated composite plates have shown that a DLF smaller than unity can be obtained for a given composite plate which is axially impacted with a compressive load that has a period close to the first period of its natural lateral vibrations.

In the particular case of marine and aeronautic structures, there is a crucial need to determine how a dynamical load could affect the buckling strength in order to assess reliability of design. The particular case of stiffened panels that are loaded by a pulse impact compression load having a finite duration and acting axially in the direction of stiffeners is investigated in the following. The analysis is performed by using a nonlinear incremental formulation based on the ABAQUS/Explicit procedure under

Abaqus software package. Both initial geometric imperfections and material degradation associated to the Heat Affected Zone (HAZ) are included. The buckling state is determined according to Budiansky and Roth criterion [7]. This is based on fitting the curves giving end-shortening as function of time, for a given pulse durations and shape, while varying the load amplitude. The investigated pulses include rectangular, triangular, double triangular and half-sine shapes. The duration of these pulses is varied between the quarters and two times the natural period of vibration of the stiffened panel. Two kinds of analyses are performed: elastic plastic and purely elastic. The effects of load pulse characteristics on the dynamic buckling strength are analyzed.

2. Modeling stiffened panels under dynamic buckling

2.1. Geometry and imperfections

The initial geometric imperfections are taken into account in the actual modeling of dynamically loaded stiffened panels. Use is made of the nonlinear finite element method. A detailed description regarding the appropriate finite element formulation to be used for the numerical model of shell buckling problems can be found in [11]. In the following, the shell element S4R available in Abaqus software package is used [12]. This element has four nodes with six degrees of freedom at each node (nodal translations in x, y and z directions and nodal rotations about these axes).

Residual stresses developing after welding process induce distortions that have the main following effects: shrinkage in the transverse direction to the weld line, longitudinal shrinkage parallel to the weld line and rotation around the weld line. The ultimate form and magnitude of welding induced distortions depend on the actual welding parameters, the materials used, the geometric design of the panel being assembled and also the preventive restraints applied during welding.

In order to identify the initial geometric imperfection that is really involved, distortion measurements are required for sufficiently representative samples of the stiffened panel. This data may then be used within finite element analysis to assess the effect of initial distortions on the buckling strength. Lillemäe et al. [13] have measured these initial geometric imperfections for two assembled panels by welding and have obtained, when considering the transverse direction, almost the same profile for both of them. However in the longitudinal direction the distortion patterns measured by these authors were quite different for the two panels they have tested.

In the present study, the welds are assumed to connect the extruded parts of the stiffened panel in the middle of distance between stiffeners. The HAZ corresponds to the central strip of the skin separating two L-shaped stiffeners.

Denoting u_x, u_y and u_z the displacement components and θ_x, θ_y and θ_z the rotations, the boundary conditions considered in the subsequent numerical simulations are as follows. The lateral edges have the boundary conditions $u_x = \theta_x = \theta_z = 0$. The edge $z = 0$ is assumed to be perfectly anchored, $u_x = u_y = u_z = 0$, $\theta_x = \theta_y = \theta_z = 0$, while a uniform distributed edge load P_z is applied on the edge $z = a$ in addition to the rigid wall boundary conditions $u_x = \theta_x = \theta_z = 0$.

The considered boundary conditions are intermediate between the two limit cases: lateral edges completely fixed and these edges fully free. So the static buckling load is expected to be greater than that of free edges configuration and lower than that of totally fixed edges. This statement could not however be extrapolated a priori to the dynamic buckling case.

In the following, imperfections resulting from welding in the transverse direction are taken into account and the longitudinal distortion is assumed to be negligible. The imperfect stiffened panel considered has the geometrical configuration shown in *Fig. 1*.

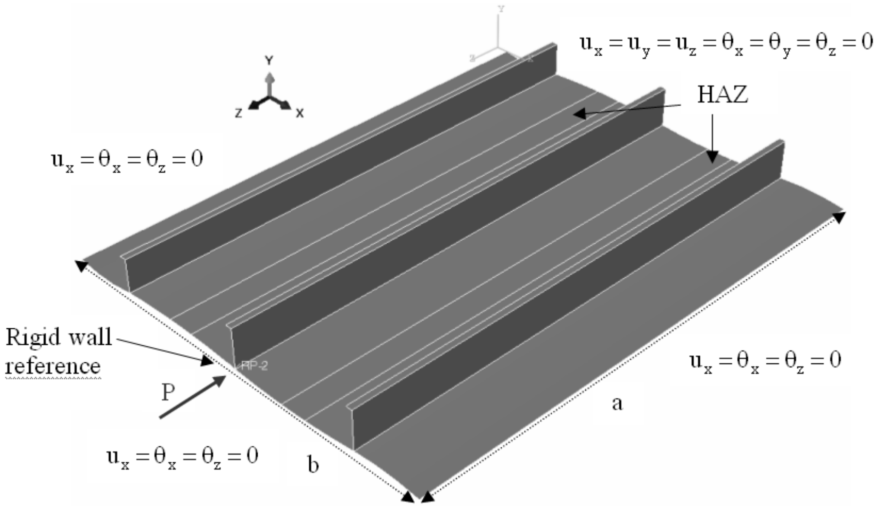

Fig. 1. Geometrical configuration of the stiffened panel and the considered boundary conditions

The total length of the base plate is $a = 958\,\text{mm}$ and its width is $b = 757.5\,\text{mm}$. The HAZ corresponds to the central strip of each segment, which is delimited with two straight lines. The plate and HAZ materials have the same depth, which is assumed to be uniform $t = 4.9\,\text{mm}$. The stiffeners are L-shaped stiffeners and have the constant depth $t_w = 2.95\,\text{mm}$, the height $h_w = 64\,\text{mm}$, the flange depth $t_f = 4.3\,\text{mm}$ and the flange height $b_f = 12\,\text{mm}$, see *Fig. 2*.

The skin plate is assumed to have an initial distortion due to welding. This distortion is supposed to be represented by a constant curvature in the transverse direction and which is symmetric about the welding line. The initial geometric imperfection resulting from welding process is modeled as shown in *Fig. 2*. The amplitude w_0 of this imperfection is fixed at the value $6\,\text{mm}$. This value is of the same order than the depth of the skin plate. It was intentionally fixed like this for this particular stiffened panel in order to emphasize dynamic buckling phenomenon. For amplitudes w_0 that are smaller than $5\,\text{mm}$, dynamic buckling will be in fact marginal as it will occur always with critical loads that are higher than those associated to the static case. This remark is quite general in the field of dynamic buckling affecting plate like structures as this phenomenon would be significant only in the presence of enough large initial geometric imperfections [14].

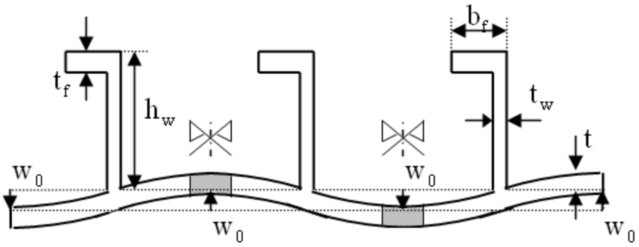

Fig. 2. Characteristics of the initial distortion in the transverse direction of the stiffened panel and stiffeners geometric parameters; the HAZ is located in the area adjacent to the welds

2.2. Material properties

The stiffened panel considered here is assumed to be made from aluminum alloy 6000 series, which is thermally treated and artificially aged after cooling from an elevated temperature shaping process. The numerical designation of this material is EN AW 6082 temper T6 according to EN 1999-1-1 [15]. Its chemical designation is AlSi1MgMn. Longitudinally stiffened panels made from this alloy were studied experimentally under axial compression in [16]. The elastic properties in the intact zone are taken from this last reference and correspond to Young's modulus $E = 64.5\,\text{GPa}$ and Poisson's coefficient $\nu = 0.3$. The plastic behavior is assumed to be described by an isotropic bilinear constant hardening law having the yield stress $\sigma_y = 265\,\text{MPa}$ and plastic modulus $E_p = 5.5\,\text{GPa}$.

In the heat affected zone the material properties are given in [14]. The Young's modulus is reduced to $E_{HAZ} = 51.6\,\text{GPa}$ while Poisson's coefficient is kept the same $\nu_{HAZ} = 0.3$. The HAZ plastic material loading curve is depicted in *Fig. 3.* The initial yield stress is $\sigma_{y,HAZ} = 135\,\text{MPa}$ and the maximum resistance stress is $\sigma_{R,HAZ} = 220\,\text{MPa}$. For both the stiffened panel intact area and HAZ stiffened panel, the material density was fixed at $\rho = 2700\,\text{kg}\cdot\text{m}^{-3}$.

To study sensitivity of dynamic buckling to material properties, both elastic plastic behaviors as described above and purely elastic behavior are considered. In the purely elastic case, the plastic part of the behavior is inactive while the other material constants are kept invariant.

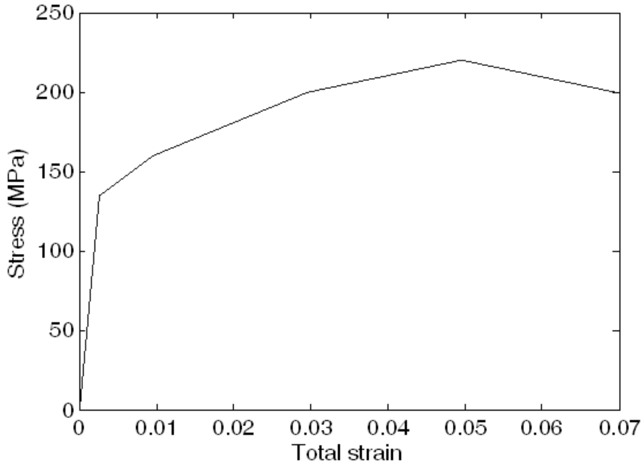

Fig. 3. Elastic-plastic loading curve of the HAZ material

2.3. Dynamic buckling

A finite element based modal model was developed at first. Convergence of this model was reached with a set of 2496 SR4 elements and a total number of 14200 free degrees of freedom. The obtained first frequency of natural vibrations is $f_1 = 104.44\,\text{Hz}$. *Fig. 4* shows the first mode of natural vibrations of the stiffened panel structure. The first mode can be seen to correspond essentially to global flexure of the panel.

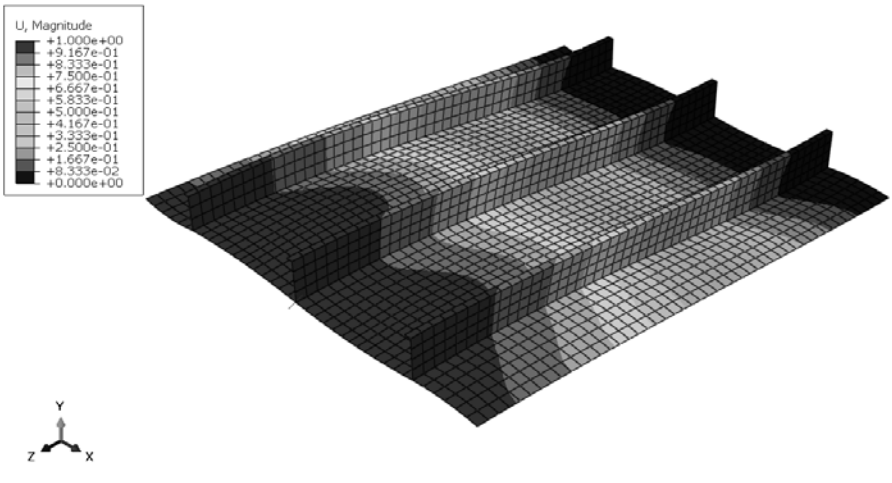

Fig. 4. First natural mode of vibrations of the stiffened panel in terms of displacement magnitude; the associated first modal frequency is f_1=104.44 Hz

Fig. 5 shows the second mode for which the natural frequency is $f_2 = 284.81\,\text{Hz}$. This mode is essentially local with strong coupling between the L-shaped stiffener and the plate skin.

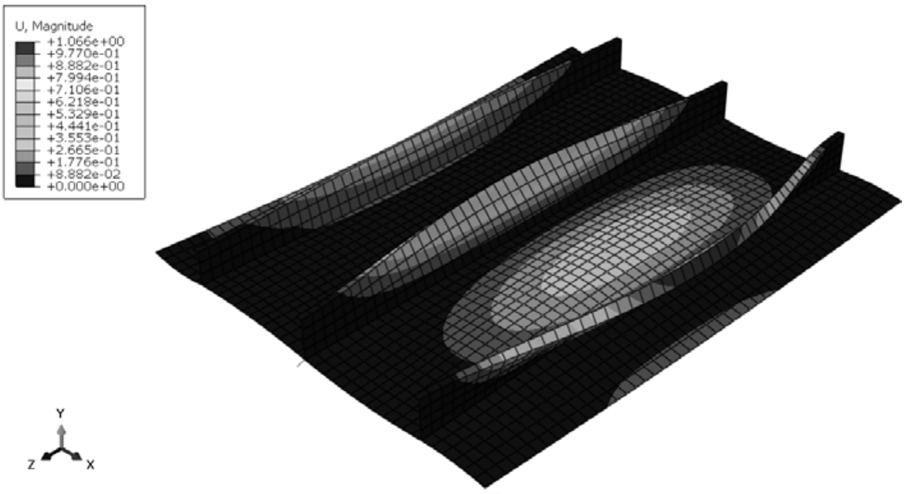

Fig. 5. Second natural mode of vibrations of the stiffened panel in terms of displacement magnitude; the associated second modal frequency is $f_2 = 284.81\,\text{Hz}$

The first modal frequency f_1 yields the characteristic time $T_0 = 1/f_1$. This period is used in order to fix the pulse duration for the dynamic loading to be applied to the stiffened panel in the interesting zone. In the following pulse durations T that are belonging to the following set $\{0.25T_0, 0.5T_0, 0.75T_0, T_0, 2T_0\}$ and four pulse shapes are investigated. These are defined as

$$P_1(t) = P_0 \begin{cases} 1, & \text{if } t \in [0, T], \\ 0, & \text{otherwise}, \end{cases} \tag{1}$$

$$P_2(t) = P_0 \begin{cases} 2t/T, & \text{if } t \in [0, T/2], \\ 2(1 - t/T), & \text{if } t \in [T/2, T], \\ 0, & \text{otherwise}, \end{cases} \tag{2}$$

$$P_3(t) = P_0 \begin{cases} \sin^2(\pi t/T), & \text{if } t \in [0, T], \\ 0, & \text{otherwise}, \end{cases} \tag{3}$$

$$P_4(t) = P_0 \begin{cases} 4t/T, & \text{if } t \in [0, T/4], \\ 2(1-2t/T), & \text{if } t \in [T/4, T/2], \\ 2(2t/T-1), & \text{if } t \in [T/2, 3T/4], \\ 4(1-t/T), & \text{if } t \in [3T/4, T], \\ 0, & \text{otherwise,} \end{cases} \tag{4}$$

where t is the time and P_0 is the magnitude of the applied dynamic load. *Fig. 6* shows these various pulse shapes when fixing the same duration for all of them.

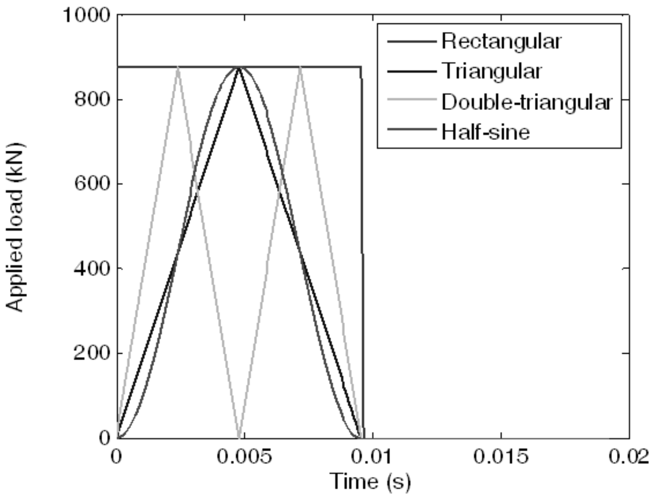

Fig. 6. Considered pulse shapes for the applied dynamic loading; $T = T_0 = 9.575$ ms and $P_0 = P^{stat} = 875$ kN

2.4. Dynamic buckling criterion

In the literature various criteria have been proposed for assessing dynamic buckling stability. The most widely used is however the condition of Budiansky-Roth [7]. In this criterion, it is assumed that the instability occurs when the displacement rate is the highest for a fixed force increment. This can also be identified as the lowest load at which there is a large sudden change in the transient response. The critical value of dynamic load corresponding to loss of stability can then be found by drawing parametric curves giving the end-shortening as a function of time for various load steps.

In the following, the critical conditions for dynamic buckling are estimated according to Budiansky and Roth judgment. To compare the dynamic buckling load to the static buckling load, the dynamic load is divided by the lowest static load as computed by means of the non-linear incremental method provided by the standard Static/Riks procedure of Abaqus software. The same model is used as in the dynamic problem regarding geometry, materials and boundary conditions

The static buckling bifurcation load is found to be $P^{stat} = 875\,\text{kN}$.

3. Results and discussion

The procedure ABAQUS/Explicit of Abaqus software package is used to solve the equations of motion with the automatic stepping option activated. The dynamic response in terms of end-shortening is then obtained for any load parameter P_0 and pulse shape of dynamical loading. The pulse duration was varied between $0.25T_0$ and $2T_0$ with $T_0 = 9.575\,\text{ms}$.

The obtained results in terms of the DLF are given in *Fig. 7* for the elastic-plastic analysis and *Fig. 8* for the elastic case. *Fig. 7* and *Fig. 8* show that there are intervals of the pulse duration for which the DLF is lesser than unity, thus the dynamic buckling load is more severe than the static buckling load. This happens for periods that are close to the fundamental free vibration period of the stiffened panel $T_0 = 9.575\,\text{ms}$. The triangular pulse shape does not appear in these two figures as it has always given a dynamic buckling load with a DLF greater than 2. The other pulse shapes yield huge reduction of the stiffened panel strength in comparison with the static case. The DLF is largely smaller that unity for some values of the pulse duration.

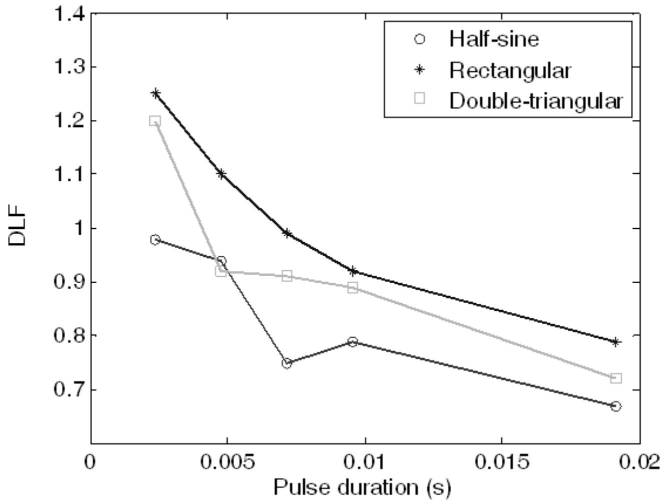

Fig. 7. Elastic-plastic analysis; the DLF as function of the pulse duration for the considered pulse shapes

Fig. 7 shows, in the case of elastic-plastic analysis, that the half-sine pulse gives the most severe reduction of the buckling load. The maximum reduction of the buckling load in dynamic conditions, i.e. DLF variation, reached in this case 33% of the buckling load which corresponds to static conditions. The double-triangular pulse shape gives the second more important reduction of the dynamic buckling load as compared to the static buckling load and a decrease of 28% of the DLF is obtained. The rectangular pulse yields the smallest variation of the DLF and the dynamic buckling load is only 21% lower than the static buckling load.

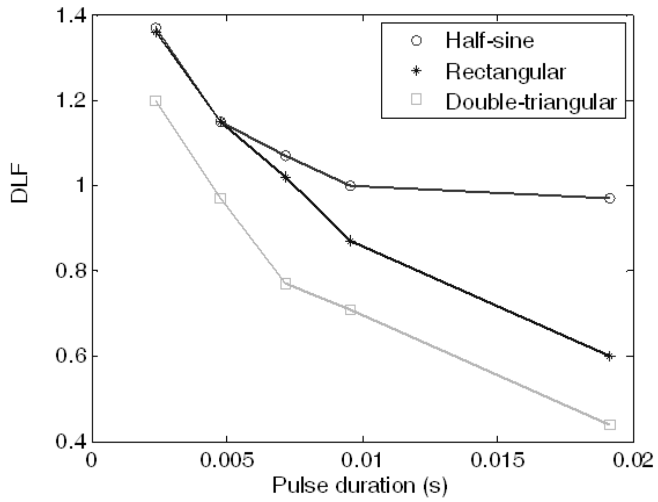

Fig. 8. Purely elastic analysis; the DLF as function of the pulse duration for the considered pulse shapes

Fig. 8 shows, in the case of elastic analysis, that the double triangular pulse is the most critical one. The reduction as evaluated from DLF variation reached 30% for the half-sine pulse. It reached 40% for the rectangular pulse which appeared to be more severe than the half-sine pulse. However the double-triangular pulse showed up to 56% reduction of the dynamic buckling load.

These results show that approximating the dynamic load by a rectangular pulse shape does not yield always the largest reduction of the buckling strength. On the other hand, material behavior affects largely the buckling load results. Plasticity appears to moderate buckling load reduction in the dynamical range as it passes from 56% to only 33%.

4. Conclusions

In this work, dynamic buckling of a longitudinally stiffened panel has been analyzed for both elastic and elastic-plastic behaviors by using non-linear finite element modeling. The panel was assumed to be subjected to in-plane axial compression, which is produced by a short pulse load applied on one transverse edge of the stiffened panel. The opposite edge was anchored and the three others were assigned symmetry about the transverse direction to the skin plate. Applying various dynamic loading conditions by using half-sine, rectangular, triangular or double triangular shapes, and with various pulse duration has enabled to assess dynamic buckling strength of the considered stiffened panel. This was achieved by means of the Budiansky and Roth stability criterion. The obtained results have shown that the most severe dynamic buckling case occurs when the pulse duration is close to two times the period of the first natural mode of vibrations. The dynamic load factor was found to be as low as 56% for the elastic range and about 33% for the elastic-plastic range of deformations. Therefore, dynamic buckling can be catastrophic for stiffened plates. This phenomenon cannot be undertaken by means of static analysis, or by fixing a priori pulse shape or pulse duration. Thorough parametric studies should be considered to mitigate this risk and guarantee structural integrity.

References

[1] Ronning L., Aalberg A., Larsen P. K. An experimental study of ultimate compressive strength of transversely stiffened aluminum panels, *Thin Walled Structures* Vol. 48, 2010, pp. 357–472.

[2] Sadovsky Z., Teixeira A. P., Guedes Soares C. Degradation of the compressive strength of rectangular plates due to initial deflection, *Thin-Walled Structures*, Vol. 43, 2005, pp. 65–82.

[3] Singer J., Arbocz J., Weller T. *Buckling experiments, experimental methods in buckling of thin walled structures,* Shells, built-up structures, composites and additional topics, Vol. 2. New York, John Wiley & Sons, Inc, 2002.

[4] Lindberg H. E., Florence A. L. *Dynamic pulse buckling, Theory and experiment*, M. Nijhoff, Boston, 1987.

[5] Bisagni C. Dynamic buckling of fiber composite shells under impulsive axial compression, *Thin-Walled Structures*, Vol. 43, No. 3, 2005, pp. 499–514.

[6] Simitses G. J. *Dynamic stability of suddenly loaded structures,* Springer-Verlag, New York, 1990.

[7] Budiansky B., Roth R. S. Axisymmetric dynamic buckling of clamped shallow spherical shells, *Collected Papers on Instability of Shell Structures*, NASA TN-D-1510; 1962; pp. 597–606.

[8] Wooseok J., Waas A. M. Dynamic bifurcation buckling of an impacted column, *International Journal of Engineering Science*, Vol. 46, 2008, pp. 958–967.

[9] Weller T., Abramovich H., Yaffe R. Dynamic buckling of beams and plates subjected to axial impact, *Computers & Structures*, Vol. 32, 1989, pp. 835–851.

[10] Abramovich H., Grunwald A. Stability of axially impacted composite plates, *Composites Structures*, Vol. 32, 1995, pp. 151–158.

[11] Paulo R. M. F., Teixeira-Dias F., Valente R. A. F. Numerical simulation of aluminum stiffened panels subjected to axial compression, Sensitivity analyses to initial geometrical imperfections and material properties, *Thin-Walled Structures*, Vol. 62, 2013, pp. 65–74.

[12] *Abaqus/Standard user's manual, ver. 6.10*. Hibbitt, Karlsson and Sorensen; 2010.

[13] Lillemäe I., Remes H., Romanoff J. Influence of initial distortion on the structural stress in 3 mm thick stiffened panels, *Thin-Walled Structures*, Vol. 72, 2013, pp. 121–127.

[14] Kubiak T. Dynamic buckling of thin-walled composite plates with varying width wise material properties, *Int J Solids Struct*, Vol. 45, 2005, pp. 5555–5567.
[15] EN 1999-1-1, Eurocode 9, Design of aluminum structures, Part 1-1, General structural rules, *European Committee for Standardization*, Brussels, 2007.
[16] Aalberg A., Langseth M., Larsen P. K. Stiffened aluminum panels subjected to axial compression, *Thin-Walled Structures*, Vol. 39, 2001, pp. 861–885.