The 5th International Conference of Euro Asia Civil Engineering Forum (EACEF-5)

Dynamic buckling of stiffened panels

Ouadia Mouhat a,*, Khamlichi Abdellatif b

aDepartment of Physics, Faculty of Sciences at Tetouan, Tetouan 93030, Morocco
bDepartment TITM, National School of Applied Sciences, Tetouan 93030, Morocco

Abstract

Design of stiffened panels requires evaluating their behavior under any loading circumstances by considering initial geometric imperfections and the altering effects resulting from eventual material degradation. Both static and dynamic loading scenarios are to be investigated in order to assess safety for buckling strength of these vital structures. In this work, dynamic buckling under in-plane uniform axial compression loading having the form of finite duration pulse is analyzed through nonlinear finite element modeling of the structure. Welding induced defects that consist of initial geometric imperfections modifying the skin plate curvature in the longitudinal direction were incorporated. Material degradation in the heat affected zone was also taken into account. The Budiansky and Roth buckling criterion was employed to predict instability under a given dynamic load pattern. Various profiles including rectangular, triangular, double-triangular and half-sine were considered. The obtained results have shown that both the pulse period and the pulse shape have a drastic effect on the buckling strength. For the considered boundary conditions, pulses having periods that are comparable to two times the period of the first natural mode of vibrations were found to reduce the static buckling strength up to 66% in the elastic regime and 33% in the elastic plastic regime.

© 2015 The Authors. Published by Elsevier Ltd.
Peer-review under responsibility of organizing committee of The 5th International Conference of Euro Asia Civil Engineering Forum (EACEF-5)

Keywords: Buckling, dynamic, stiffened panel, pulse duration, pulse shape, geometric imperfections;

1. Introduction

Stability of stiffened panels is a main concern in many engineering applications such as marine and aeronautics [1, 2]. As modern structures are more and more designed to be light with thin-walled shapes, buckling risk

* Corresponding author. Tel.: +212-679-148-263;
E-mail address: ouadie.mouhat@gmail.com
constitutes a real problem to undertake while managing to increase the strength-to-weight ratio. Stiffened panels are structures that enable optimizing strength for a given weight. But, they are subject in service life to various destablising loading conditions either static or dynamic and may suffer from various alterations resulting from material degradation or initial geometric imperfections.

Static buckling has been extensively investigated for various types of structures that are susceptible to undergo instabilities under the in-service applied loading and is now a relatively good stated 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 active 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 modelling.

Considering the case of an impacted beam, Wooseok and Waa [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 dynamic buckling 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.

In the particular case of marine and aeronautic structures, there is a crucial need to determine how a dynamical load could modify 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 dynamic 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 criterion 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. Period duration of these pulses is varied between the quarter 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 considered.

2. Modeling stiffened panels under dynamic buckling

2.1. Geometry and imperfections

The initial geometric imperfections are taken into account in the actual modelling of dynamically loaded stiffened panel. Use is made of the 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 [10]. In the following, the shell element S4R presented in Abaqus software package is used [11]. 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 on the preventive restraints applied during welding.

In order to identify the distribution of initial geometric imperfection which is really involved, distortion measurements are required for sufficiently representative samples of the stiffened panel. This data may then be used in finite element analysis to assess the effect of initial distortions on the buckling strength. Lillemäe et al. [12] 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 these panels. However in the longitudinal direction the measured distortion patterns were quite different for the two tested panels.

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

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 which is considered has the geometrical configuration shown in Fig. 1.

The chosen total length of the base plate is \( a = 958 \text{mm} \) and its width is \( b = 757.5 \text{mm} \). The HAZ correspond to the central strip of each segment, which is delimited by two straight lines. The plate and HAZ materials have the same thicknesses which are assumed to be uniform and equal to \( t = 4.9 \text{mm} \). The stiffeners are L-shaped webs and have constant thickness \( t_w = 2.95 \text{mm} \), height \( h_w = 64 \text{mm} \), flange thickness \( t_f = 4.3 \text{mm} \) and 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 is taken symmetric about the welding line. The initial geometric imperfection resulting from welding process is modelled as shown in Fig. 2. The amplitude $w_0$ of this imperfection is fixed at the value 6mm. This value is of the same order than the thickness 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 small then 5mm, dynamic buckling will be in fact marginal as it will occur always with critical loads that are higher than those obtained in the static case. This remark is quite general in the field of dynamic buckling affecting plate like structures as dynamic buckling would be significant only in the presence of enough large initial geometric imperfections [13].

2.2. Material properties

The elastic material properties used for modelling the stiffened panel in the intact zone correspond to aluminium for which the Young’s modulus is $E=64.5\text{GPa}$ and Poisson’s coefficient is $\nu=0.3$. The plastic behaviour 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 Young’s modulus is reduced to $E_{\text{HAZ}}=51.6\text{GPa}$ while Poisson’s coefficient is taken to be also equal to $\nu_{\text{HAZ}}=0.3$. The HAZ plastic material loading curve is depicted in Fig. 3. The initial yield stress is $\sigma_{Y,\text{HAZ}}=135\text{MPa}$ and the maximum resistance stress is $\sigma_{R,\text{HAZ}}=220\text{MPa}$. For the intact and HAZ stiffened plate materials, the material density was fixed at $\rho=2700\text{kg.m}^{-3}$.

![Fig. 3. Elastic-plastic loading curve of the HAZ material](image)

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

2.3. Dynamic loading

A finite element based modal model was developed at first. Convergence of this model was assessed 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 $f=104.44\text{Hz}$. Fig. 4 shows the first mode of natural vibrations of the stiffened panel structure. One can see that the first mode is essentially a global flexure of the skin plate.
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. Are investigated 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. Fig. 5 depicts these various pulse shapes for the same duration $T = T_0$.

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 criterion 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 parameters.

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 one obtained under the same material properties and boundary conditions with applying the maximum load. The dynamic load is divided by the static load as computed by means of the nonlinear incremental method provided by the standard Static/Riks procedure of Abaqus software. The static buckling bifurcation load is found to be $P_{stat} = 875 \text{kN}$.
3. Results and conclusions

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

![Fig. 6. Elastic-plastic analysis; the DLF as function of the pulse duration for the considered pulse shapes](image)

The obtained results in terms of the DLF are given in Fig. 6 for the elastic-plastic analysis and Fig. 7 for the elastic case.

Fig. 6 and 7 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 one. This happens for periods that are close to the fundamental free vibration period of the stiffened panel \( T_0 = 0.009575 \text{ s} \). The triangular pulse shape does not appear in these figures as it has always given a dynamic buckling load with a DLF that is greater than 2. On the opposite, the other shown pulse shapes have yielded huge reduction of the static buckling load.

![Fig. 7. Purely elastic analysis; the DLF as function of the pulse duration for the considered pulse shapes](image)

Fig. 6 shows, in the case of elastic-plastic analysis, that the half-sine pulse gives the most severe reduction of the buckling load. It is followed by the double-triangular pulse shape, then the rectangular pulse. The reduction reached 33% for the half-sine pulse; 21% for the rectangular pulse and 28% for the double-triangular pulse.
Fig. 7 shows, in the case of elastic analysis, that the double triangular pulse is the most critical one. The reduction reached 3% for the half-sine pulse; 40% for the rectangular pulse and only 66% for the double-triangular pulse.

These results show that approximating the dynamic load by a rectangular pulse shape does not induce always the largest reduction of the buckling strength. On the other hand, material behaviour affects largely the buckling load results. Plasticity appears to moderate buckling load reduction as it passes from 66% to 33%.

4. Conclusions

In this work, dynamic buckling of a longitudinally stiffened panel has been analyzed for both elastic and elasctic-plastic behaviours by using nonlinear finite element modelling. The panel was assumed to be subjected to in-plane axial compression which is produced by a short pulse 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 having 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. The Budiansky and Roth stability criterion was used.

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 less as 66% for the elastic regime and about 33% for the elastic-plastic regime 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