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Primary instability regions of square plates subjected to uniaxial in-plane periodic compression load

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Abstract
The purpose of this thesis is to investigate the dynamical behavior and the parametric primary instability of thin elastic plate subjected to uniform in-plane uniaxial periodic compression load. The plate is assumed to be simply supported along its four edges (SSSS). Based on small deflection theory of plates, the value of natural frequency of free vibration and buckling load are determined by assuming the mode shape of vibration of plates simply supported along all edges. If the plate subjected to parametric excitation (static and dynamic component of loading) the plate lose its stability dynamically. The instability regions are found according to Bolotin's concept. In the present research we studied the effect of different parameter such as effect of excitation parameter and load value. It found that increasing both of the excitation parameter and loading value cause an increase in the regions of instability. An experimental work is conducted to improve the accuracy of the mathematical model. Two sets of experiments are taken for the primary instability regions, the error which represent the difference between the theoretical and experimental results lies within 9%.

Keywords: Parametric instability; Small deflection theory; Buckling load; Primary instability regions; Excitation parameter.

1. Introduction

Thin plates are widely used in all fields of engineering which are subjected to static or dynamic loads like, mechanical, civil, aerospace engineering, architectural structures, marine, solar panels, bridges, hydraulic structures, containers, airplanes, missiles, ships, instruments and machine parts.

Plates used in structure elements which subjected to in-plane periodic compression loads (dynamic load) may undergo unstable transverse vibrations, leading to a parametric resonance, due to certain combinations of the values of load parameters and natural frequency of transverse vibration. The study of the static and dynamic behavior of plate under the in-plane compression load is of certain importance, because this load affect the response of the plate. If the plate subjected to static in -plane compression load, it may be unstable statically. As the magnitude of this load increases, the natural frequency of

vibration reduces and at the critical buckling load the natural frequency failed (becomes zero). If the plate subjected to a periodic in-plane compression dynamic loads, it will losses its stability dynamically in the form of resonant transverse vibrations. This phenomenon is known as **''parametric resonance''**.
The study of dynamic instability back to **(1831)**. This phenomenon was first observed by Faraday, when he noticed that when a fluid filled container vibrates vertically, fluid surface oscillates at half the frequency of the container**.**
K. Takahashu, et al., [1], determined the natural frequency of free vibration, static buckling load and boundaries of regions of dynamic instability of a cantilever rectangular plate. Thin plate small deflection theory is used to derive the governing equation of plates. The plate assumed to be subjected to an in-plane sinusoidally varying load applied along the free end. The vibration and buckling problems were solved according to Rayleigh-Ritz method , while the dynamic stability problem according to Hamilton's principle to derive time dependent variables. Unstable regions of dynamic instability are given for various loading conditions. Simple parametric resonances and combination parametric resonances are acquired for various loading conditions, static load and damping.
J. H. Kim, [2], studied the dynamic instability of plates based on Kirchhoff plate theory and midline plate theory subjected to follower force using Finite Element Method. Four, nine and sixteen -node quadrilateral elements mesh is used to study the dynamic stability analysis of a thin plate under an in-plane force and a follower force. The calculation results of these elements gave good agreement when compared with the analytical solutions. Also one can noted that sixteen -node quadrilateral mindlin elements is the best element for studying dynamic instability of a shear deformable plates.
A. K. L. Srivastava, et al., [3], used the Finite Element Method to study the vibration and dynamic instability behavior of stiffened plate under the effect of in-plane periodic edges load. Various boundary conditions, varying mass and stiffness properties, aspect ratios and varying number of stiffeners have been discussed for dynamic instability using the method of Hill's infinite determinants. The results explain that the principal instability regions have significant effect.
Q. Wang, et al., [4], studied the dynamic instability of variable stiffness thin-walled column under the effect of periodically alternating axial force neglecting the effect of damping. The variable stiffness means that it changes with periodically alternating axial force as for non-linear geometry stiffness matrix of thin-walled column. The second-order differential equations with period coefficients describes the dynamic instability of thin-walled columns were solved using Finite Element Method with the help of computer program represented by matlab. The results give good agreement when compared with other calculation method.
S. Z. Al-sarraf, et al., [5], used the equivalent grid-framework model to study the problem vibration and stability of plate behavior subjected to both dynamic and static load. Numerical results are offered for several example problems, and they show that the above method is reasonably accurate.
A.K.L. Srivastava, et al., [6], used the finite element method to study vibration and buckling characteristics of stiffened plates under the effect of in-plane uniform and non-uniform edge loading. Various boundary conditions, aspect ratios, varying mass and stiffness properties and varying number of stiffeners have been analyzed for buckling and vibration studies. The equation of motion used to determine the characteristic equations for the natural frequencies, buckling loads and their corresponding mode shapes. This study also include the effects of aspect ratio, boundary conditions, the position of stiffeners and number of stiffeners parameters upon the buckling load parameter and fundamental natural frequency of the stiffened plates. The above results solutions give good agreement with other solution.
Y. F. Zhou, et al., [7], used the exact solution to study the dynamic stability of viscoelastic rectangular plate subjected to tangential follower force. The boundary conditions used in this study one edge clamped and the other three edges are simply supported. The viscoelastic rectangular plate is supported by two opposite edges simply supported and other two edges clamped. Firstly, by assuming the mode shape of free vibration that satisfies the simply supported boundary conditions, the governing partial differential equation is transferred to an ordinary differential equation with variable coefficients. The results show that the aspect ratio is of great effects on the stability and the critical loads of the viscoelastic plates.
From the above literature, most of the research papers mentioned which studied parametric instabilities deals with the primary instability regions of plates subjected to parametric uniaxial compression load.
In the present work the primary instability regions of square plates are investigation using exact solutions. Also experimental work is added to improve the accuracy of the present theoretical work.

2. Theoretical analysis

2-1 Based Equation

When the load that applied on a plate is parametric (static and dynamic load), the plate will resonate (fluttering motion) in regions known as regions of instability. The parametric excitation can be define by:

$$N_x = N_s + \delta N_s cos\theta t \quad (1)$$

Where

N_s : Magnitude of static load (N)

δ : Excitation parameter

θ : Excitation frequency (rad/sec).

By neglecting the effect of longitudinal vibration in the middle plane, then only the transverse vibration is of important interest and taking into account. Alos neglecting the effect of shear deformation and rotary inertia, then the basic equation of vibration subjected to in-plane parametric load can be summarized as :

$$D\nabla^4 w + \rho h\frac{\partial 2w}{\partial t^2} = -(N_s + \delta N_s cos\theta t)\frac{\partial 2w}{\partial x^2} \quad (2)$$

2-2 Primary Regions of Instability:

The primary regions of instability can be evaluated when the period of time is taken (2T) and the summation takes the odd values. If the plates are simply supported along all edges then the solution of Eq.(2) is written in the form **[8]:**

$$W_{km}(x,y,t) = A_{km}\sin\frac{k\pi x}{a}\sin\frac{m\pi y}{b}f_{km}(t) \quad (3)$$

Where

$f_{km}(t)$: Unknown function of time.

A_{km}: Constants.

k,m : Integer (k = m = 1,3,5,….)

Substituting Eq.(3) into Eq.(2) gives:

$$\rho h\frac{d2f}{dt^2} + D\left(\frac{k^2\pi^2}{a^2} + \frac{m^2\pi^2}{b^2}\right)^2 f - (N_s + \delta N_s cos\theta t)\frac{k^2\pi^2}{a^2}f = 0 \quad (4)$$

After some mathematical manipulations, Eq.(4) can be written as:

$$\frac{d2f}{dt^2} + \omega_{km}{}^2\left[1 - \left(\frac{(N_s + \delta N_s cos\theta t}{N_{cr}}\right)\right]f = 0 \quad (5)$$

Where

$\omega_{k,m}$: Natural frequency of free vibration (rad/sec).

Equation.(5) is written in the form:

$$\frac{d2f}{dt^2} + \omega_s{}^2(1 - 2\varphi_{km}cos\theta t) = 0 \quad (6)$$

Where

ω_s: Frequency of vibration of plates subjected to uniaxial in-plane compression static force (rad/sec).

$$\omega_s = \omega_{km}\sqrt{\left[1 - \frac{N_s}{N_{cr}}\right]} \quad (7a)$$

$$\varphi_{km} = \frac{\delta N_s}{2(N_{cr} - N_s)} \quad (7b)$$

The solution of Eq.(6) can be used to find the regions of dynamic instability, the function represent these regions is expressed by **[43]**;

$$f(t) = \sum_{m=1,3,5,..}^{\infty}\left(\Psi_m \sin\frac{m\theta t}{2} + \Phi_m \cos\frac{m\theta t}{2}\right) \quad (8)$$

Substitute Eq.(8) into eq.(6) results in the following equation:

$$\sum_{m=1,3,5,\ldots}^{\infty} -\left(\frac{m\theta}{2}\right)^2 \Psi_m \sin\frac{m\theta t}{2} - \left(\frac{m\theta}{2}\right)^2 \Phi_m \cos\frac{m\theta t}{2} + \omega_s{}^2\Psi_m \sin\frac{m\theta t}{2} + \omega_s{}^2\Phi_m \cos\frac{m\theta t}{2}$$
$$-2\,\varphi_{km}\,\omega_s{}^2\Psi_m \sin\frac{m\theta t}{2}\cos\theta t - 2\varphi_{km}\,\omega_s{}^2\Phi_m \cos\frac{m\theta t}{2}\cos\theta t = 0 \qquad (9)$$

Using the trigonometric relations and equating the coefficients of *(sin(mθt/2)*) and (*cos(mθt/2))* Eq.(9) leads to the following two equations in matrix form:

$$\begin{bmatrix} 1+\varphi_{km}-\frac{\theta^2}{4\omega_s{}^2} & -\varphi_{km} & 0 & \vdots \\ -\varphi_{km} & 1-\frac{9\theta^2}{4\omega_s{}^2} & -\varphi_{km} & \vdots \\ 0 & -\varphi_{km} & 1-\frac{25\theta^2}{4\omega_s{}^2} & \vdots \\ \cdots & \cdots & \cdots & \vdots \end{bmatrix} \begin{Bmatrix} \Psi_1 \\ \Psi_2 \\ \Psi_3 \\ \vdots \end{Bmatrix} = 0 \qquad (10)$$

$$\begin{bmatrix} 1-\varphi_{km}-\frac{\theta^2}{4\omega_s{}^2} & -\varphi_{km} & 0 & \vdots \\ -\varphi_{km} & 1-\frac{9\theta^2}{4\omega_s{}^2} & -\varphi_{km} & \vdots \\ 0 & -\varphi_{km} & 1-\frac{25\theta^2}{4\omega_s{}^2} & \vdots \\ \cdots & \cdots & \cdots & \vdots \end{bmatrix} \begin{Bmatrix} \Phi_1 \\ \Phi_2 \\ \Phi_3 \\ \vdots \end{Bmatrix} = 0 \qquad (11)$$

The order of the above two matrix equations are infinite and the size of the matrices is the same. Because of the solution related to the period 2T exists, then applying the condition of the existence of nontrivial solution (the determinant of the two matrices must be equated to zero). These two matrices may be combined together under the $\mp$ sign to calculate the determinant, in which the following equation result:

$$\begin{vmatrix} 1\pm\varphi_{km}-\frac{\theta^2}{4\omega_s{}^2} & -\varphi_{km} & 0 \\ -\varphi_{km} & 1-\frac{9\theta^2}{4\omega_s{}^2} & -\varphi_{km} \\ 0 & -\varphi_{km} & 1-\frac{25\theta^2}{4\omega_s{}^2} \end{vmatrix} = 0 \qquad (11)$$

According to **Bolotin's** concept **[8]**, using the first approximation, only the first diagonal element of the above matrix is taken and solved to give the boundaries of the upper and lower limits of the primary regions of dynamic instability;

$$1 \pm \varphi_{km} - \frac{\theta^2}{4\omega_s{}^2} = 0 \qquad (12)$$

Substitute Eqs.(7a) and (7b) into Eq.(12) gives:

$$\frac{\theta^2}{4\omega_{km}{}^2} = 1 - \emptyset \pm \frac{\delta}{2}\emptyset \qquad (13)$$

Where

$$\varnothing = \frac{N_s}{N_{cr}}$$

Equation (13) can be separated into two equations:

$$\frac{\theta^2}{4\omega_{km}{}^2} = 1 - \varnothing + \frac{\delta}{2}\varnothing \quad (14)$$

$$\frac{\theta^2}{4\omega_{km}{}^2} = 1 - \varnothing - \frac{\delta}{2}\varnothing \quad (15)$$

Equation(14) used to determine the upper limit of the primary instability region while Eq.(15) used to determine the lower limit.

3. Experimental work

The experimental work is carried out according to the following steps:

1. Determining the mechanical properties of the plate and its chemical composition.
2. Determining the stiffness of spring .
3. Measuring the limits of regions of dynamic instability of the plate.

3.1 Mechanical properties of plates

The tensile test specimens are picked from the tested plate. This test is used to determine the tensile strength, yield strength and elongation of plates. This test is done in the standardization and quality control center depending on the British Standard European Norm (BSEN) as shown in table 1.

.

Table 1. Results of tensile test

Properties	Values	Units
Tensile strength	494	N/mm^2
Yield strength	300	N/mm^2
Elongation% (L_o =80mm)	26	-
Modulus of elasticity	183	Gpa

2-2 Instrumentation

Figure 1 shows a sketch of the instruments used in stability test and consist of:

1. Exciter used to apply harmonic load as shown in figure 2, this exciter consist of:
    a) three phase induction motor with speed (**378 rpm**).
    b) two pulleys of diameter values **(8, 12 cm)**.
    c) belt for transferring motion.
    d) shaft used to apply dynamic load.
    e) a spring used to apply the static load and anoter one used to apply dynamic load.
    f) regulator used to vary speed of motor from **(0 - 650 rpm)**.
    g) structure rig.
2. 3-axis accelerometer type ADXL335 and a microcontroller used is to receive the response of tested model (amplitude of vibration) as shown in figure 3. The accelerometer measures the transverse acceleration of the plate due to dynamic load in the form of voltage signal. This accelerometer is connected to the computer through a microcontroller interface, this microcontroller show in figure 4.

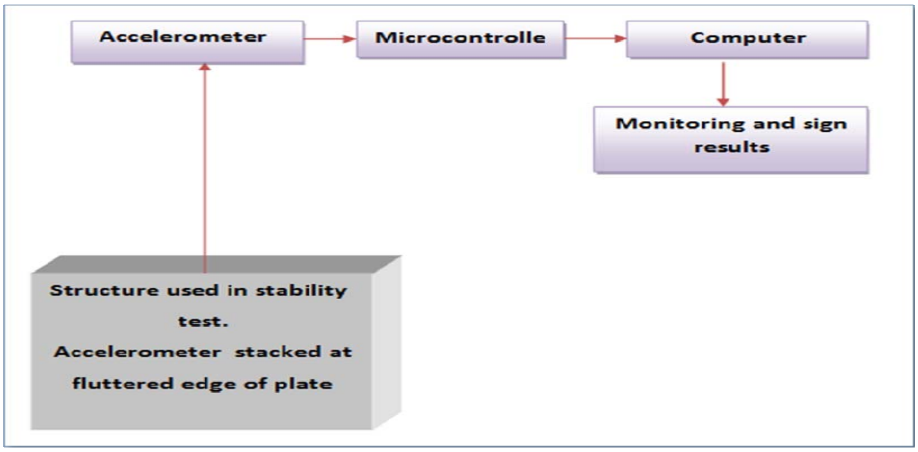

Figure 1. Steps used in stability test

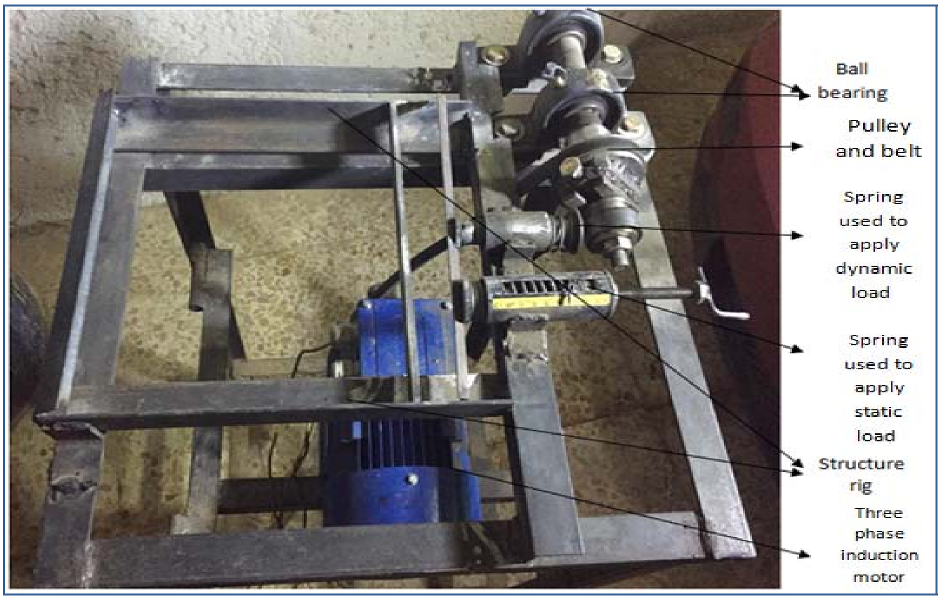

Figure 2. Exciter used in vibration and stability test.

Figure 3. Accelerometer.

Figure 4. Microcontroller.

3-3 Stiffness of Spring

The stiffness of spring used to apply static and dynamic loads is determined by measuring the deflection of spring at loaded point under various load (2, 4, 6, 8, 10 kg force). Based on slope of curve fitting the value of stiffness can be measured through different load conditions. This procedure is repeated to minimize error.

3-4 In-plane Constant Compression Load

The static force is apply through a spring of known stiffness which compression by a screw with known distance as shown in figure 5.

Figure 5: Spring used to apply static load

3-5 In-plane Parametric Compression Load

To apply the in-plane periodic compression load, the shaft which prepared for this purpose is connected with small pulley as shown in figure 6. This shaft is constructed so that different amplitudes of excitation can be satisfied. The amplitude of excitation is controlled by shifting the shaft through the slot as illustrated in figure 7. When the shaft imposed at slot center, the amplitude of excitation is zero (no excitation), while the maximum excitation is improved when the shaft imposed at the end of the slot.

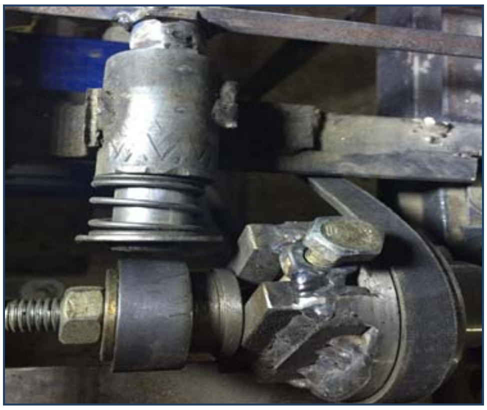

Figure 6: Shaft connected in slot used to apply dynamic load

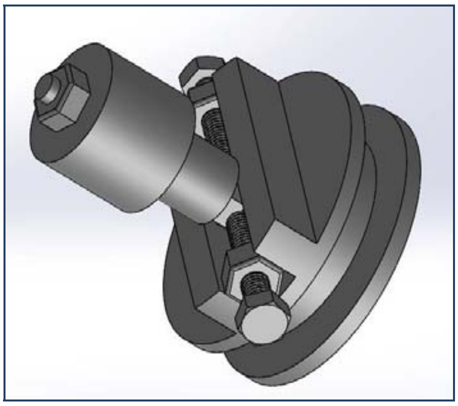

Figure 7: Illustration of the slot used to apply dynamic load.

3-6 Accelerometer Measurements

The signal that picked from the accelerometer of the tested model is processed by the microcontroller is a voltage signal. A calibration must be carried out to convert the readings to amplitude.

3-7 Stability Test

Table 2 show the properties of the plates using in the stability test. This study confirm the effect of periodical load and static load (spring under prescribed compression) on dynamic stability. To measure the frequency at which the plate lose its stability by flutter, the accelerometer is placed at the center of

edge under excitation as shown in figure 8. The frequency of excitation increased until maximum vibration flutter is achieved.
This test is repeated for three values of excitation parameter so this gives three values of fluttering frequencies for the upper limit and three values of fluttering frequencies for the lower limit. These values of fluttering frequencies forming the regions of primary instability.

Table 2. Properties of the plates used in the experimental work

Parameter	Value	Units
Length (a)	0.3	m
Width (b)	0.3	m
Thickness (h)	0.0008	m
Density (ρ)	7800	Kg/m^3
Modulus of Elasticity (E)	183	Gpa

Figure 8. Measured limits of dynamic instability of plates

3-7 Calibration of Instruments
3-7-1 Calibration of Three Phase Induction Motor
The speed of the motor is controlled by a frequency regulator. This speed is calibrated by a tachometer to improve the motor speed accuracy.

3-5-2 Calibration of Accelerometer
Calibration is done according to the relations shown below **[9]** and displayed on computer in the form of acceleration reading.

$$v_{ref} = \frac{v_1 + v_2}{2} \quad (16)$$

$$se = \frac{v_1 - v_2}{2g} \quad (17)$$

$$acc = \frac{v_{in} - v_{ref}}{se} - g \quad (18)$$

where
v_{in}: Output voltage of accelerometer (v).
v_1: Upper voltage (v).
v_2: Lower voltage (v).
se : Sensitivity (s^2.v/m).
v_{ref}: Reference voltage (v)

acc : Acceleration (m/s^2).
g : Earth gravitation (m/s^2).
The upper voltage is measured by fixing the accelerometer at top position and recording the value of voltage from matlab. while the lower voltage is measured by fixing the accelerometer at bottom position and recording the value of voltage **[9]**.

4. Results and discussions

This section explains the results obtained from the analytical model used to determine the primary instability regions of plates having aspect ratis (a/b = 1) and simply supported boundary conditions along all edges (SSSS). The results obtained from the mathematical model of (SSSS) square plate is proved by the experimental study where good agreement is obtained.

4-1 Natural Frequency

Square plates are modeled by considering the boundary conditions (SSSS) with aspect ratios 1. In order to check the values of natural frequencies determined in the present study comparison is used with **Chakraverty [10]** and give good agreement. If the plate is (SSSS), then exact values of natural frequency can be achieved without any error.

4-3 Primary instability regions

The values of limits of the primary instability regions depend on the excitation parameter (δ) , load ratio (ratio of static load to bucking load (Ø)), aspect ratio (a/b) and natural frequency of free vibration**.** If the value of natural frequency increased then the primary regions of instability shifted to higher values and if the natural frequency of free vibration decreased then the primary regions of instability lowered as shown in figure 9. The enclosed area between the upper and lower limits refer to unstable regions as shown in figure.

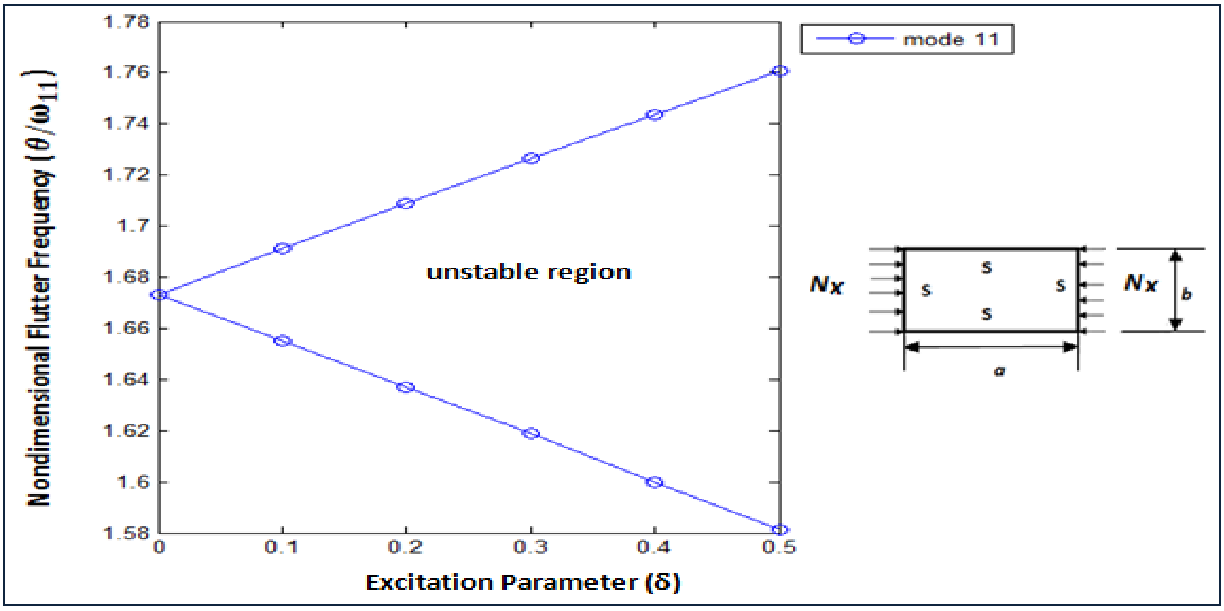

Figure 9. Primary instability region of square plate

4-3-1 Effect of excitation parameter on primary instability regions

To investigate the effect of the excitation parameter (δ) on regions of instability, this parameter is changed in a suitable range while the other affecting parameters are fixed at suitable values. In general the excitation parameter δ is changed from 0 to 0.5 and the value of nondimensional load is fixed at 0.3 (Ø = 0.3). It is shown that the width of instability regions increased with increasing δ as shown in figure 10. This attribution is due to the fact that the energy transferred from the load source to the plate increasing with increase the excitation parameter. This transferred energy will be translated in as an increase in the frequency because the amplitude of vibration is constant.

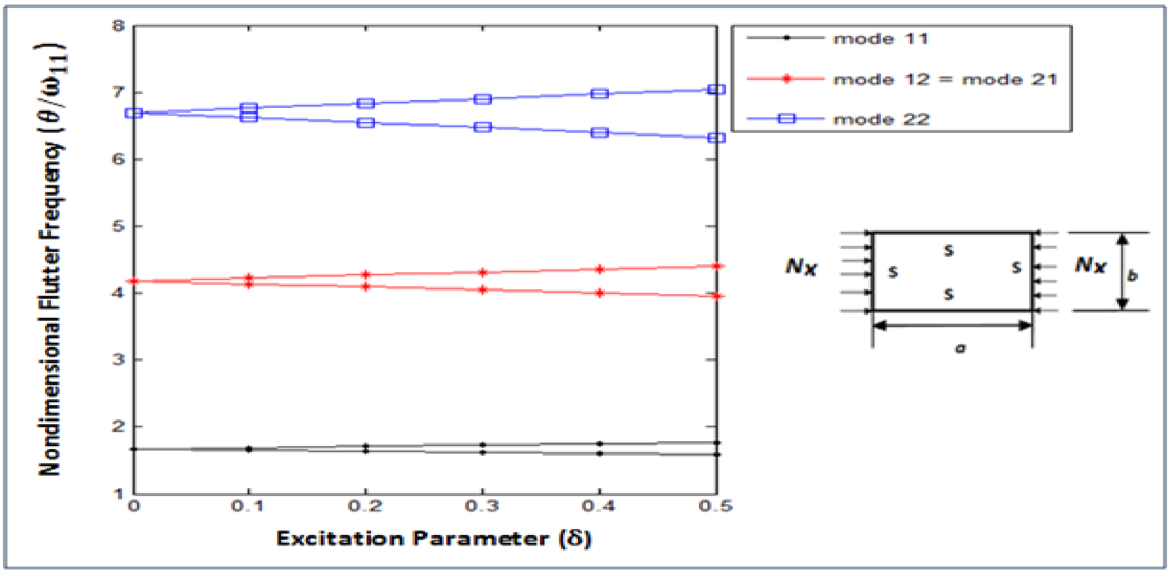

Figure 10: Effect of excitation parameter on primary instability regions of SSSS plate, a/b = 1 and Ø = 0.3.

4-3-2 Effect of the Loading on Primary and Secondary Instability Regions

The effect of the in-plane compression fluctuating load on primary instability regions illustrated in figure 11. In this figure the value of loading is changed in the range 0 - 0.5. As shown in this figure, the band of frequencies (between upper and lower limits) that the plate loses its stability by flutter increased as the load increased. This behavior can be elaborated by noticing that the time period of the harmonic fluctuation of the uniaxial in-plane compression load occurs at period which are 2T. The flutter that appears at this period can be caused by:

1- The time duration of variation of the force from the maximum value (N_{max}) to the minimum value ($N_{min} = N_s$) increased with increasing the loading in which this value will cause the size of unstable regions will be wider.

2- Increasing the load cause an increase in the inertia forces delivered in the structure (plate), therefore the size of the unstable region becomes wider.

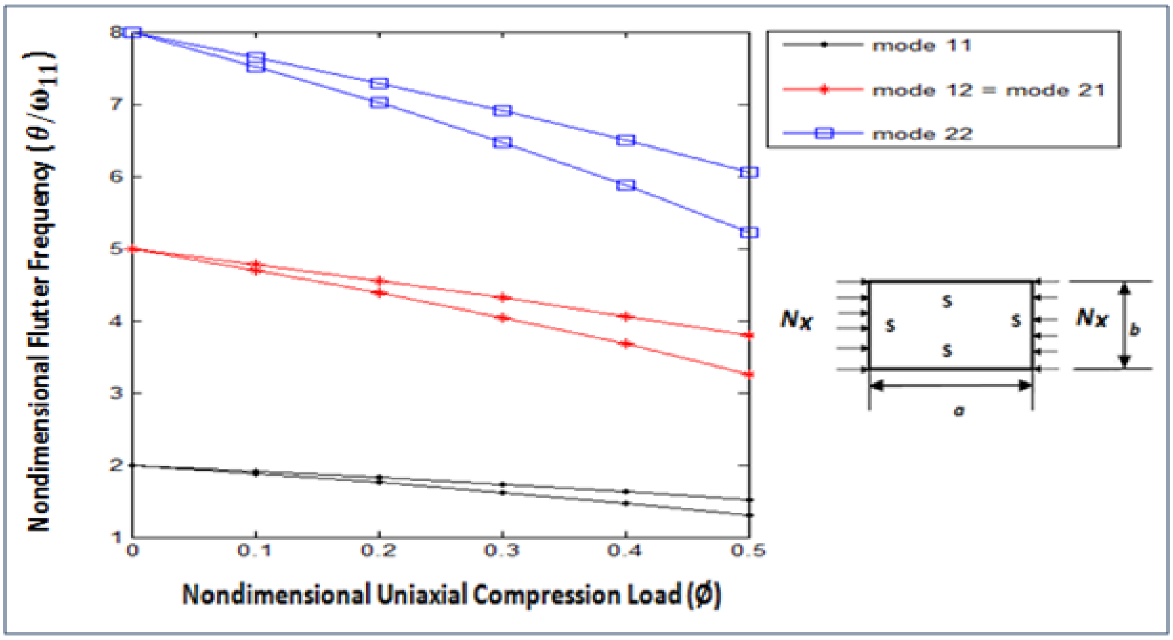

Figure 11: Effect of nondimensional uniaxial compression load on primary instability region of SSSS plate, a/b = 1 and δ = 0.3.

4-5 Experimental Results

The experimental verification on the dynamic stabili ty of square plate simply supported along all edges is carried out to improve the accuracy of the theoretical values. The properties of plates used in this study are listed in table 2. The stiffness of spring used to applied static load is (k_s = 2368.134 N/m) and the stiffness of the spring that used to apply dynamic load is (k_d = 1367.514 N/m). The experimental results may be summarized as follows:

1. The effect of excitation parameter (δ = 0.129, 0.288, 0.433) on the regions of dynamic instability of square plate is illustrated in figure 12. This study is done by fixing the value of the ratio of static load (*Ns* = 47.362 N) when a spring used to applied static load is compressed at distance 2cm) to buckling load (*Ncr* = 3763.609 N) at (Ø = 0.012). The differences between experimental results and theoretical results gives very good agreement with error less than 8%. Figures 13 through 15 shwo the acceleration curve (amplitude of vibration) of experimental primary instability regions.
2. The effect of loading (Ø = 0.012, 0.025, 0.037) on the theoretical and experimental regions of dynamic instability of square plate are illustrated in figure 16. This experiment is done by fixing the value of the excitation parameter at (δ = 0.129). The differences between experimental results and theoretical results gives very good agreement with error less than 9%. Figures 17 through 19 shwo the acceleration curve (amplitude of vibration) of experimental primary instability regions.

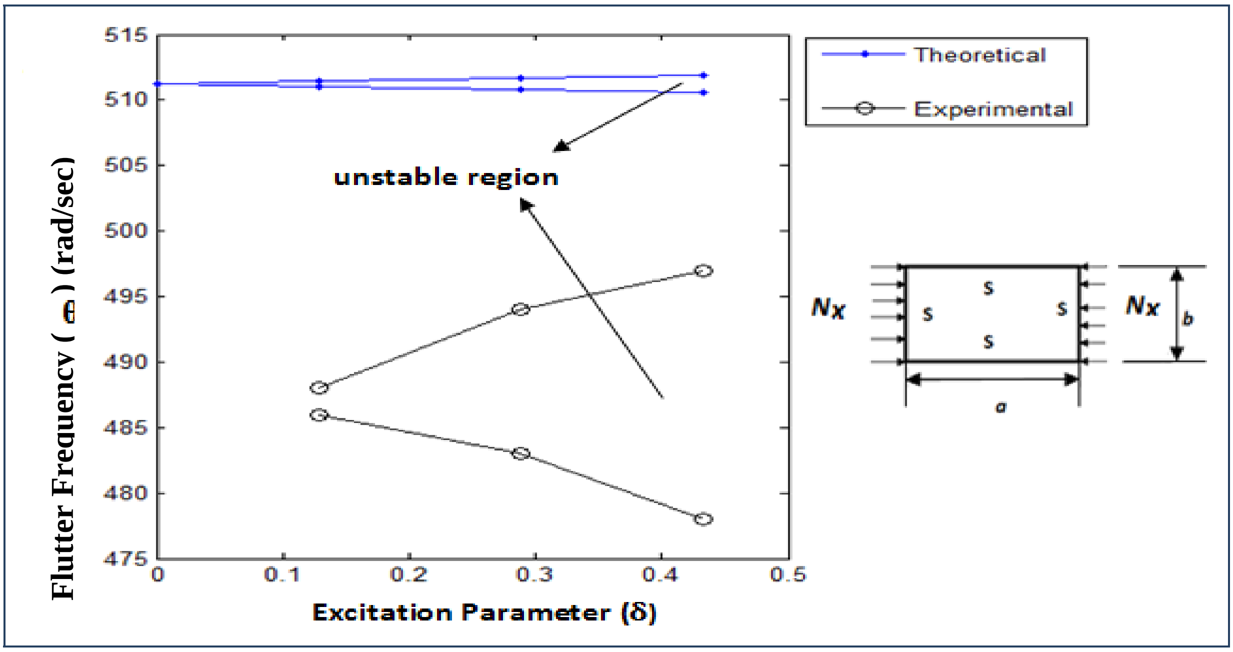

Figure 12. Theoretical and experimental effects of excitation parameter on primary instability region of SSSS plate.

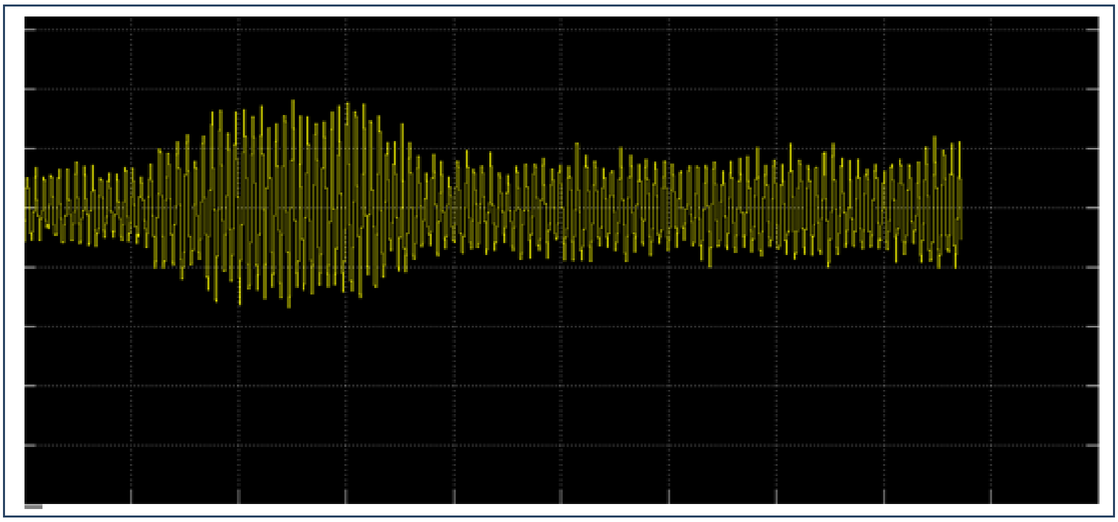

Figure 13. Acceleration curve for dynamic instability with excitation parameter (δ = 0.129)

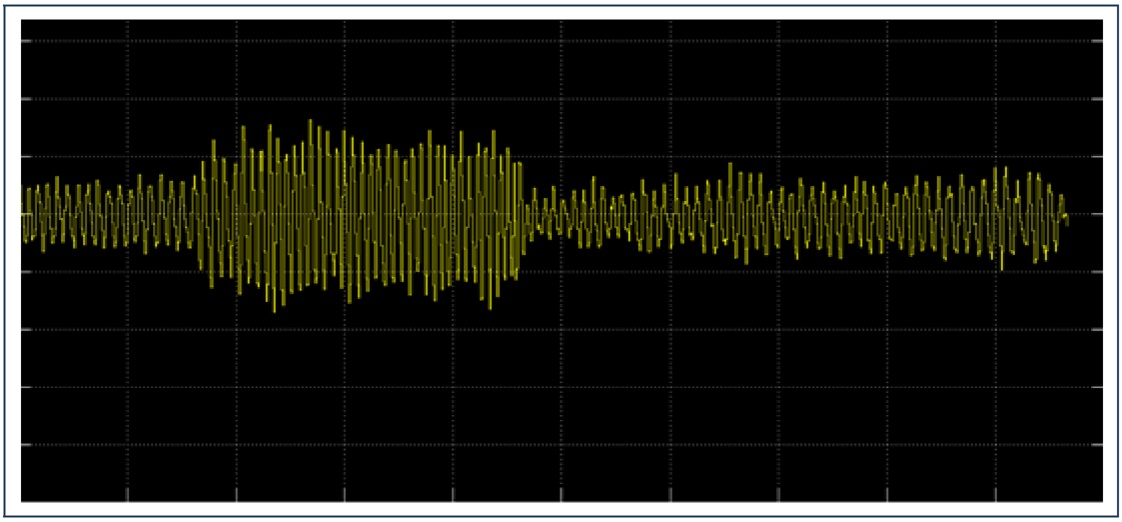

Figure 14. Acceleration curve for dynamic instability with excitation parameter (δ = 0.288)

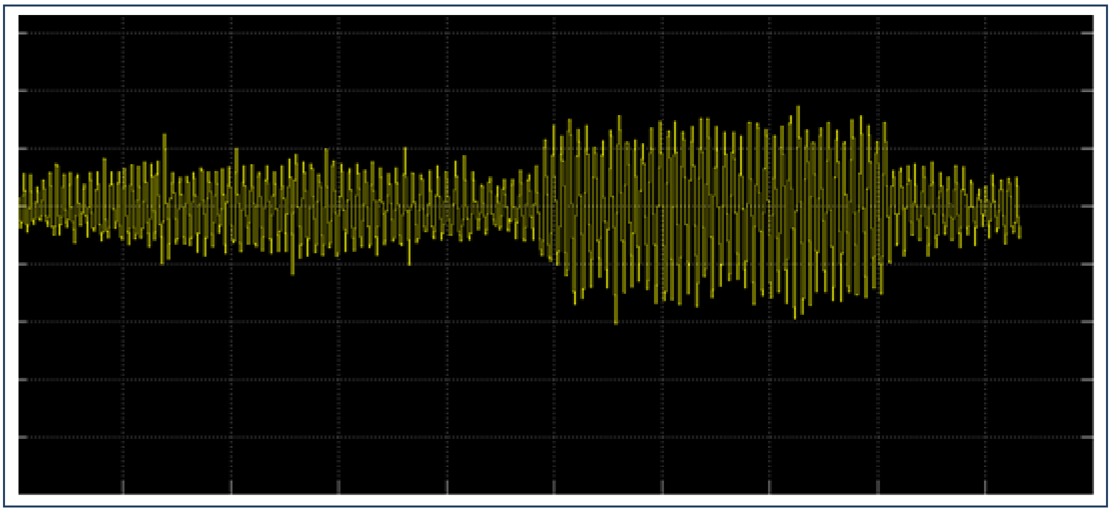

Figure 15. Acceleration curve for dynamic instability with excitation parameter (δ = 0.433).

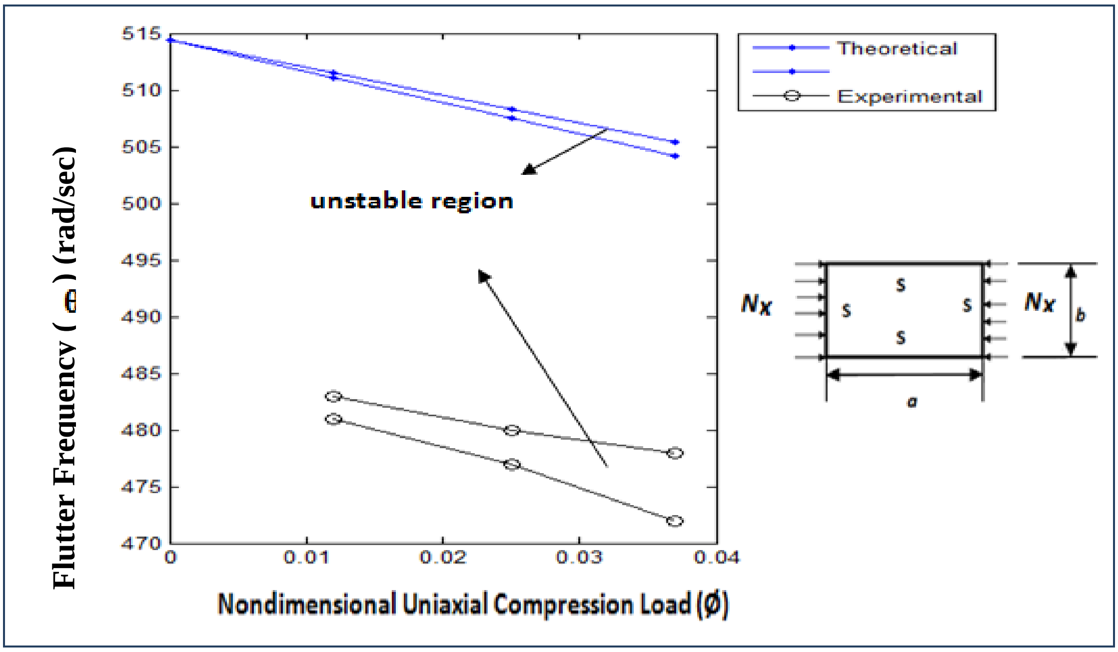

Figure 16. Theoretical and experimental effects of loading on primary instability region of SSSS plate.

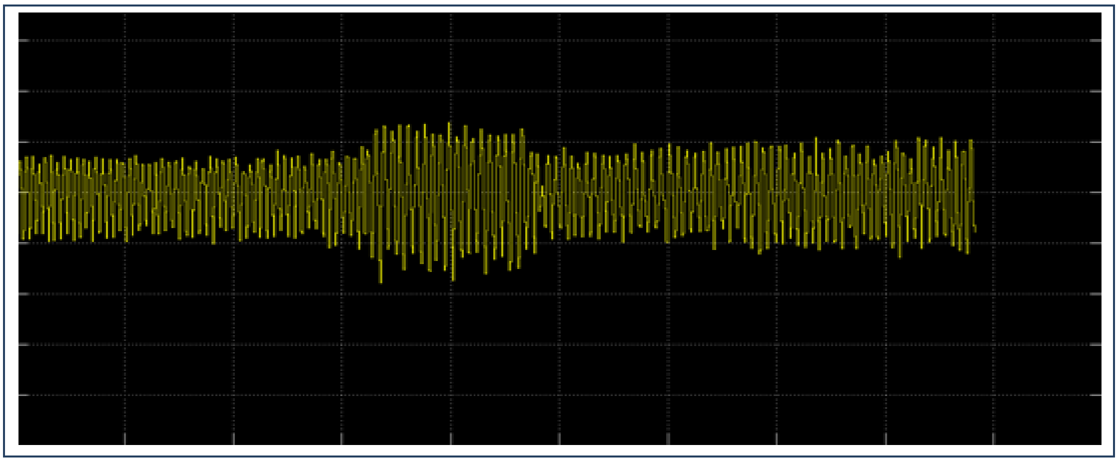

Figure 17. Acceleration curve for dynamic instability with loading ($\emptyset = 0.012$).

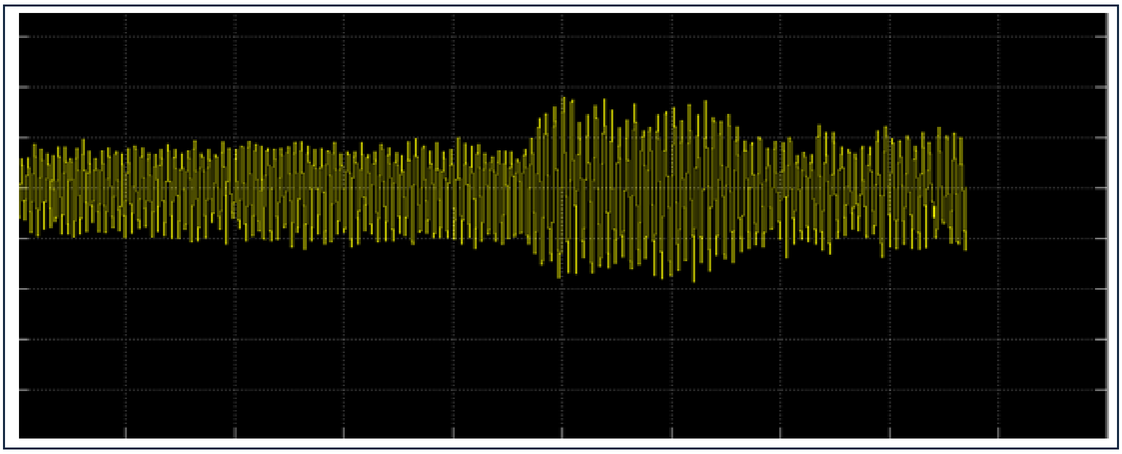

Figure 18. Acceleration curve for dynamic instability with loading ($\emptyset = 0.025$).

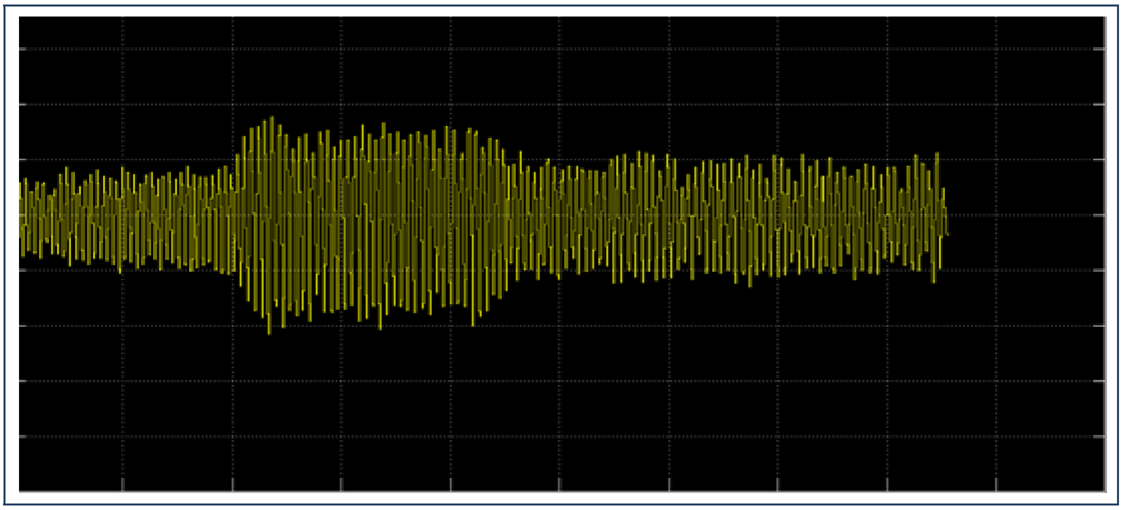

Figure 19. Acceleration curve for dynamic instability with loading ($\emptyset = 0.037$)

The differenc between experimental and theoretical results due to:

1. Neglecting the effect of shear strains (Υ_{xz} , Υ_{yz}) and the effect of normal strain (ϵ_z).
2. Neglecting the effect of stress in Z-direction (σ_z).
3. Neglecting the effect of structure damping (including internal friction or surrounding media).
4. Neglecting the higher terms of the slope.

5. Conclusions

Throughout the present study the following conclusion can be drawn:.

1. The band of frequencies that the plate loses its stability by flutter increased with increasing both of the excitation parameter and load value.
2. The regions of instability increase depended on the mode of vibration.

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