International Journal of Scientific & Engineering Research, Volume ƙ, Issueǁǁ, -ÖÝÌÔÉÌÜı2014
ISSN 2229-5518

Non linear dynamic analysis of vertical shaking conveyor under harmonic and parametric excitation

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Abstract— In this paper, we introduced a non linear dynamic analysis and mathematical study of the vibration behavior in vertical shaking conveyor under harmonic and parametric excitation. By studying the vibrating motion of vertical vibration conveyor, the equations of motion described by a coupled of nonlinear differential equations (two degree of freedom) including the linear and cubic nonlinear terms. Multiple scale perturbation method applied to study the approximate mathematical solutions up to the second order approximation and we studied the stability of the steady state solution mathematically at the worst different resonance cases using frequency response equations. The resulting different resonance cases reported and studied numerically. Also, the numerical solutions of vertical shaking conveyor investigated applying Runge-Kutta of fourth order. The stability of the steady state solution near the selected resonance cases investigated numerically using phase-plane technique. The effect of the different parameters of the vertical shaking conveyor studied numerically. Results compared to previously published work. In the future work, the system can be studied at another worst different resonance cases or we can use active and passive controller.

Index Terms— Vibro-impact system, Multiple time scale, Vibrations, Resonance, Stability.

1 Introduction

In the area of mechanics and electronics, the behaviors of mechanical systems under periodic loadings have been examined by many researchers. Vertical conveyors are effective examples observing various kinds of parameters of this problem. Vertical Vibratory conveyor is developed recently to convey the bulk materials. It has some advantages like simple structure, occupied less, long conveying road, low maintenances cost and energy consumed less. It uses to convey high temperature, wearing, poisoned and volatile materials if it is sealed. The screening, dryness, and cooling processes can be fulfilled at the conveying process. So it is used broadly in iron and steel industry, metallurgical industry, chemical plant. In vertical shaking conveyers, the load-carrying element performs double harmonic oscillations: linear along the vertical axis and rotational around that axis (i.e. longitudinal and torsional oscillations). Conveyer drives with centrifugal vibration exciters may have (1) a single unbalanced mass, (2) two equal unbalancing masses, (3) a pendulum-type unbalanced mass, (4) four unbalanced masses in two shafts, (5) four rotating unbalanced masses for three principal modes of oscillation, i.e. linear, elliptical, and circular.

Alısverisci [1] the transitional behavior across resonance, during the starting of a single degree of freedom vibratory system excited by crankand-rod. A loaded vibratory conveyor is safer to start than an empty one. Shaking conveyers with cubic nonlinear spring and ideal vibration exciter have been analyzed analytically for primary resonance by the Method of Multiple Scales, and numerically. The approximate analytical results obtained in this study have been compared with the numerical results, and have been found to be well matched. Comparing the results obtained by applying the approximate analytic method with those obtained numerically it is concluded that the difference is negligible, proving the correctness of the analytic procedure used. Bayıroglu [2] the nonlinear analysis for the change of the parameters of the motion, stability condition, and the jump phenomena has been shown graphically the transition over resonance of a nonlinear vibratory system, excited by unbalanced mass, is important in terms of the maximum vibrational amplitude produced on the drive for the cross-over. Alısverisci et al [3] The working ranges of oscillating shaking conveyers with a non-ideal vibration exciter have been analyzed analytically for primary resonance by the method of multiple scales with reconstitution, and numerically. The maximum amplitude of vibration is important in determining the structural safety of the vibrating members. The results of the numerical simulations, obtained from the analytical equations, showed that the important dynamic characteristics of the system such as damping, non-linearity and the amplitude excitations effects, and presented a periodic behavior for these situations. The jump phenomenon occurs in the motion of the system near resonance. The analytical results obtained in this study have been compared with the numerical results, and have been found to be well matched. Alısverisci et al [4], the vibrating system is analyzed, analytically, and nu-

International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

merically for superharmonic and subharmonic resonance by the method of multiple scales. Very often in the motion of the system near resonance the jump phenomenon occurs. The stable motions of the oscillator are shown with one peaks in the power spectrum for superharmonic resonance and with two peaks in the power spectrum for subharmonic resonance. Both analytical and numerical results that we have obtained are in good agreement. The system studied here exhibits chaotic behavior in case of strong nonlinearity. Yuejing Zhao et al [5] The configuration and force analysis of vertical vibratory conveyor are conducted. The model of system with considering the friction between the materials and the spiral conveying trough is developed. The numerical simulations are done and the dynamical responses curves are given. Suitable configuration parameters of vertical vibratory conveyor and parameters of materials can make it work normally. Bayıroglu [6] Primary, subharmonic, and superharmonic responses have been investigated with multiple scales along with numerical methods for vertical conveyors. The change in the parameters of motion, stability condition, and jump phenomena has been shown graphically by Mathematica software for comparing the results. Both analytical and numerical results obtained had good agreement. Systems, excited by unbalanced mass, are important in terms of the maximum vibration amplitude produced on the drive for the cross-over. The maximum amplitude of vibration is then of interest in determining the structural safety of the vibrating members. Eissa et al. [7-9] investigated saturation phenomena in non-linear oscillating systems subject to multi-parametric and/or external excitations. The system represents the vibration of a single-degree-of-freedom cantilever or the wing of an aircraft. They reported the occurrence of saturation phenomena at different parameters values. They applied saturation values of different parameters as optimum working conditions for vibration suppression of the cantilever. Hamed et al [10-12] showed how effective is the passive vibration control reduction at resonance under multi-external or both multi-external and multi-parametric and both multi-external and tuned excitation forces. They reported that the advantages of using multi-tools are to machine different materials and different shapes at the same time. This leads to saving the time and higher machining efficiency. Kamel and Hamed [13] studied the nonlinear behavior of an inclined cable subjected to harmonic excitation near the simultaneous primary and 1:1 internal resonance using multiple scale method. Hamed et al [14] presented the behavior of the nonlinear string beam coupled system subjected to external, parametric and tuned excitations for case 1:1 internal resonance. The stability of the system studied using frequency response equations and phase-plane method. It is found from numerical simulations that there are obvious jumping phenomena in the frequency response curves. Sayed and Hamed [15] studied the numerical response and stability analyses of the behavior of the pitch-roll ship model described by a two-degree-of-freedom system under harmonic and parametric excitation forces. They obtained the approximate solutions up to and including the second-order approximations using the method of multiple scale perturbation technique. Sayed et al. [16] investigated the non-linear dynamics of a two-degree-of freedom vibration system including quadratic and cubic nonlinearities subjected to external and parametric excitation forces. There exist multi-valued solutions which increase or decrease by the variation of some parameters. The numerical simulations show the system exhibits periodic motions and chaotic motions. Kamel et al. [17] studied a model subject to multi-external excitation forces. The model is represented by two-degree-of-freedom system consisting of the main system and absorber simulating ultrasonic machining. They used the passive vibration controller to suppress the vibration behavior of the system.

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2. Mathematical Analysis

The elevator has a cylindrical casing. A helical open trough or closed pipe is attached to the outside or the inside of the vertical tubular casing along which the load can be transported from the bottom upward. A vibration-exciting drive is mounted at the top or bottom of the casing to impart directed vibrations along and around the vertical axis to the bottom, which cause the load to move upward along the helix. The rotating unbalanced masses develop centrifugal forces P in the vertical conveyor, as shown in Fig. 1. The vertical components of these forces Pz, induce vertical vibrations in the conveyor (along the vertical axis); the horizontal components Px are directed differently and form a moment, which causes angular (torsional) vibrations of the conveyor. With a particular combination of these oscillations at definite frequency and amplitude, the load moves upward along the helix in Fig. 1a.

International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

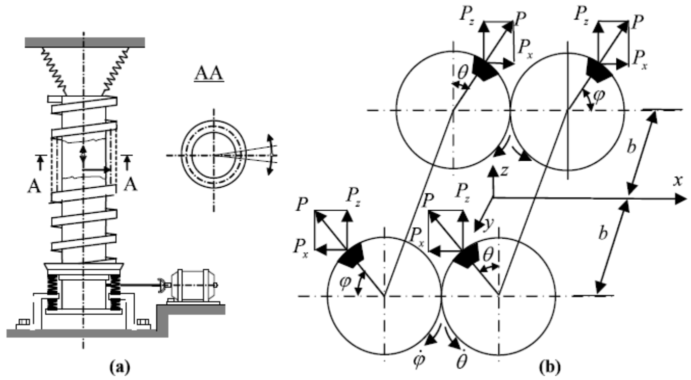


Fig. 1a Vertical shaking conveyor
Fig. 1b Unbalanced masses for the vertical shaking conveyor

The system is excited by linear and nonlinear external and parametric excitation forces. Proceeding as in Ref. [6], we can obtain the following nonlinear ordinary differential governing equation of motion for the Vertical shaking conveyor

$$\ddot{z} + \omega_1^2 z_1 = \varepsilon f_1 \left(\cos \Omega t + \sin \Omega t \right) + \varepsilon z_1 f_2 \left(\cos \Omega_1 t + \sin \Omega_1 t \right) - 2\varepsilon \mu_1 \dot{z}_1 - \varepsilon \alpha_1 z_1^3 \tag{1}$$

$$\ddot{z}_2 + \omega_2^2 z_2 = \varepsilon f_1 \left(\cos \Omega t + \sin \Omega t \right) + \varepsilon z_2 f_2 \left(\cos \Omega_1 t + \sin \Omega_1 t \right) - 2\varepsilon \mu_2 \dot{z}_2 - \varepsilon \alpha_2 z_2^3 \tag{2}$$

where μ_1, μ_2 are the damping coefficients of vertical shaking conveyor system and controller, α_1 and α_2 are nonlinear coefficients of the vertical shaking conveyor system, f_1, f_2 are the excitation forcing amplitudes and Ω, Ω_1 are external and parametric excitation frequencies. The vertical shaking conveyor system natural frequencies are ω_1 and ω_2, ε is a small perturbation parameter and $0 < \varepsilon << 1$.

3. Perturbation Analysis

The MSPT method is used to obtain a uniformly valid, asymptotic expansion of the solutions for (1)–(2) is in the form:

$$z_1(t;\varepsilon) = z_{10}(T_0, T_1) + \varepsilon z_{11}(T_0, T_1) + O(\varepsilon^2) \tag{3}$$

$$z_2(t;\varepsilon) = z_{20}(T_0, T_1) + \varepsilon z_{21}(T_0, T_1) + O(\varepsilon^2) \tag{4}$$

The derivatives will be in the form

$$\left. \begin{aligned} \frac{d}{dt} &= D_0 + \varepsilon D_1 + \ldots \\ \frac{d^2}{dt^2} &= D_0^2 + 2\varepsilon D_0 D_1 + \ldots \end{aligned} \right\} \tag{5}$$

For the first-order approximation, we introduce two time scales, where $T_n = \varepsilon^n t$ and the derivatives $D_n = \partial / \partial T_n$, (n= 0, 1). Substituting (3)–(5) into (1)–(2) and equating the coefficients of equal powers of (ε) leads to

$O(\varepsilon^0)$

$$(D_0^2 + \omega_1^2) z_{10} = 0 \tag{6a}$$

$$(D_0^2 + \omega_2^2) z_{20} = 0 \tag{6b}$$

$O(\varepsilon^1)$

$$(D_0^2 + \omega_1^2) z_{11} = f_1 \left(\cos \Omega t + \sin \Omega t \right) + z_{10} f_2 \left(\cos \Omega_1 t + \sin \Omega_1 t \right) - 2 D_0 D_1 z_{10} - 2\mu_1 D_0 z_{10} - \alpha_1 z_{10}^3 \tag{7a}$$

$$(D_0^2 + \omega_2^2) z_{21} = f_1 \left(\cos \Omega t + \sin \Omega t \right) + z_{20} f_2 \left(\cos \Omega_1 t + \sin \Omega_1 t \right) - 2 D_0 D_1 z_{20} - 2\mu_2 D_0 z_{20} - \alpha_2 z_{20}^3 \tag{7b}$$

The general solutions of (6) can be written in the form

$$z_{10} = A_1(T_1) \exp\left(i\omega_1 T_0 \right) + cc \tag{8a}$$

$$z_{20} = A_2(T_1) \exp\left(i\omega_2 T_0 \right) + cc. \tag{8b}$$

Where A_m (m=1, 2) are complex function in T_1, cc represents the complex conjugate of the previous terms. Substituting (8) into (7) and eliminating the secular terms, the particular solutions of (7) will be in the form:

$$\begin{aligned} z_{11} &= \left[\frac{\alpha_1 A_1^3}{8\omega_1^2} \right] \exp\left(3i\omega_1 T_0 \right) + \left[\frac{f_1}{2(\omega_1^2 - \Omega^2)} \right] \exp\left(i\Omega T_0 \right) \\ &+ \left[\frac{f_1}{2i(\omega_1^2 - \Omega^2)} \right] \exp\left(i\Omega T_0 \right) + \left[\frac{f_2 A_1}{2(\omega_1^2 - (\Omega_1 + \omega_1)^2)} \right] \\ &\times \exp\left(i(\Omega_1 + \omega_1) T_0 \right) + \left[\frac{f_2 A_1}{2i(\omega_1^2 - (\Omega_1 + \omega_1)^2)} \right] \\ &\times \exp\left(i(\Omega_1 + \omega_1) T_0 \right) + \left[\frac{f_2 A_1}{2(\omega_1^2 - (\Omega_1 - \omega_1)^2)} \right] \\ &\times \exp\left(i(\Omega_1 - \omega_1) T_0 \right) + \left[\frac{f_2 A_1}{2i(\omega_1^2 - (\Omega_1 - \omega_1)^2)} \right] \\ &\times \exp\left(i(\Omega_1 - \omega_1) T_0 \right) + c.c \end{aligned} \tag{9a}$$

$$\begin{aligned} z_{21} &= \left[\frac{\alpha_2 A_2^3}{8\omega_2^2} \right] \exp\left(3i\omega_2 T_0 \right) + \left[\frac{f_1}{2(\omega_2^2 - \Omega^2)} \right] \exp\left(i\Omega T_0 \right) \\ &+ \left[\frac{f_1}{2i(\omega_2^2 - \Omega^2)} \right] \exp\left(i\Omega T_0 \right) + \left[\frac{f_2 A_2}{2(\omega_2^2 - (\Omega_1 + \omega_2)^2)} \right] \end{aligned}$$

International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

$$\times\exp\left(i(\Omega_1+\omega_2)T_0\right)+\left[\frac{f_2A_2}{2i(\omega_2^2-(\Omega_1+\omega_2)^2)}\right]$$

$$\times\exp\left(i(\Omega_1+\omega_2)T_0\right)+\left[\frac{f_2A_2}{2(\omega_2^2-(\Omega_1-\omega_2)^2)}\right]$$

$$\times\exp\left(i(\Omega_1-\omega_2)T_0\right)+\left[\frac{f_2A_2}{2i(\omega_2^2-(\Omega_1-\omega_2)^2)}\right]$$

$$\times\exp\left(i(\Omega_1-\omega_2)T_0\right)+c.c \qquad (9b)$$

We can rewrite Eqs. (9) in the form

$$z_{11}=K_1\exp(3i\omega_1T_0)+K_2\exp\left(i\Omega T_0\right)+K_3\exp\left(i(\Omega_1+\omega_1)T_0\right)$$
$$+K_4\exp\left(i(\Omega_1-\omega_1)T_0\right)+cc \qquad (10a)$$

$$z_{21}=H_1\exp(3i\omega_2T_0)+H_2\exp\left(i\Omega T_0\right)+H_3\exp\left(i(\Omega_1+\omega_2)T_0\right)$$
$$+H_4\exp\left(i(\Omega_1-\omega_2)T_0\right)+cc \qquad (10b)$$

where $K_i,(i=1,...,4)$ and $H_i,(i=1,...,4)$ are a complex function in T_1,T_2 and *cc* represents the complex conjugates. From the above-derived solutions, many resonance cases can be deduced. The reported resonance cases are classified into:

(A) Primary Resonance: $\Omega\cong\omega_1,\ \Omega\cong\omega_2$.

(B) Sub-Harmonic Resonance: $\Omega_1\cong 2\omega_1,\ \Omega_1\cong 2\omega_2$.

(C) Simultaneous or Incident Resonance: Any combination of the above resonance cases is considered as simultaneous or incident resonance.

4. Stability of Motion

4.1. For the First Mode of Vertical Shaking Conveyor System

Stability of the considered system is investigated at the simultaneous primary $\Omega\cong\omega_1$ and principle parametric $\Omega_1\cong 2\omega_1$ are considered. Two detuning parameters σ_1 and σ_2 such that

$$\Omega=\omega_1+\varepsilon\sigma_1 \text{ and } \Omega_1-\omega_1=\omega_1+\varepsilon\sigma_2 \qquad (11)$$

This case represents the system worst case. Substituting Eq. (11) into Eq. (7a) and eliminating the secular terms, leads to the solvability conditions for the first order approximation, we get

$$2i\omega_1D_1A_1=-2i\,\omega_1\mu_1A_1-3\alpha_1A_1^2\bar{A}_1+\left(\frac{1-i}{2}\right)f_1\exp(i\sigma_1T_1)$$
$$+\left(\frac{1-i}{2}\right)f_2\bar{A}_1\exp(i\sigma_2T_1) \qquad (12)$$

To analyze the solutions of Eq. (12), we express $A_1(T_1,T_2)$ and $A_2(T_1,T_2)$ in the polar form

$$A_1(T_1,T_2)=\frac{a_1}{2}e^{i\varphi_1} \qquad (13)$$

where a_1 and φ_1 are the steady state amplitude and phase of the motion of the first mode. Substituting Eq. (13) into Eq. (12) and equating the real and imaginary parts we obtain the following equations describing the modulation of the amplitude and phase of the first modes of vertical shaking conveyor response:

$$\dot{a}_1=\mu_1a_1-\frac{f_1}{2\omega_1}\cos\theta_1+\frac{f_1}{2\omega_1}\sin\theta_1-\frac{f_2a_1}{4\omega_1}\cos\theta_2$$
$$+\frac{f_1a_1}{4\omega_1}\sin\theta_2 \qquad (14a)$$

$$a_1\dot{\varphi}_1=\frac{3\alpha_1a_1^3}{8\omega_1}-\frac{f_1}{2\omega_1}\cos\theta_1-\frac{f_1}{2\omega_1}\sin\theta_1-\frac{f_2a_1}{4\omega_1}\cos\theta_2$$
$$-\frac{f_1a_1}{4\omega_1}\sin\theta_2 \qquad (14b)$$

Where

$$\theta_1=\sigma_1T_1-\varphi_1 \text{ and } \theta_2=\sigma_2T_1-2\varphi_1 \qquad (15)$$

Form the system of Eqs. (14) to have stationary solutions, the following conditions must be satisfied:

$$\dot{a}_1=\dot{\theta}_1=\dot{\theta}_2=0 \qquad (16)$$

It follows from Eq. (15) that $\dot{\varphi}_1=\sigma_2-\sigma_1$

Hence, the steady state solutions of Eqs. (14) are given by
Hence, the fixed points of Eqs. (22)- (23) are given by

$$\mu_1a_1-\frac{f_1}{2\omega_1}\cos\theta_1+\frac{f_1}{2\omega_1}\sin\theta_1-\frac{f_2a_1}{4\omega_1}\cos\theta_2$$
$$+\frac{f_1a_1}{4\omega_1}\sin\theta_2=0 \qquad (17a)$$

International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

$$a_1(\sigma_1-\sigma_2)+\frac{3\alpha_1 a_1^3}{8\omega_1}-\frac{f_1}{2\omega_1}\cos\theta_1-\frac{f_1}{2\omega_1}\sin\theta_1-\frac{f_2 a_1}{4\omega_1}\cos\theta_2-\frac{f_1 a_1}{4\omega_1}\sin\theta_2=0 \quad (17b)$$

Solving the resulting algebraic equations for the fixed points, we obtained

$$\sigma_1^2+\left(\frac{3\alpha_1 a_1^2}{4\omega_1}-2\sigma_2\right)\sigma_1+\left(\mu_1^2+\sigma_2^2+\frac{9\alpha_1^2 a_1^4}{64\omega_1^2}-\frac{3\alpha_1\sigma_2 a_1^2}{4\omega_1}-\frac{f_1^2}{2a_1^2\omega_1^2}-\frac{f_2^2}{8\omega_1^2}-\frac{f_1 f_2}{2a_1\omega_1^2}\right)=0 \quad (18)$$

4.2. For the second mode of vertical shaking conveyor system

The stability is investigated at the simultaneous primary $\Omega\cong\omega_2$ and sub-harmonic $\Omega_1\cong 2\omega_2$ are considered. We introduce detuning parameters σ_3 and σ_4 such that

$$\Omega=\omega_2+\varepsilon\sigma_3 \quad \text{and} \quad \Omega_1-\omega_2=\omega_2+\varepsilon\sigma_4 \quad (19)$$

This case represents the system worst case. Substituting Eq. (19) into Eq. (7b) and eliminating the secular terms, leads to the solvability conditions for the first order approximation, we get

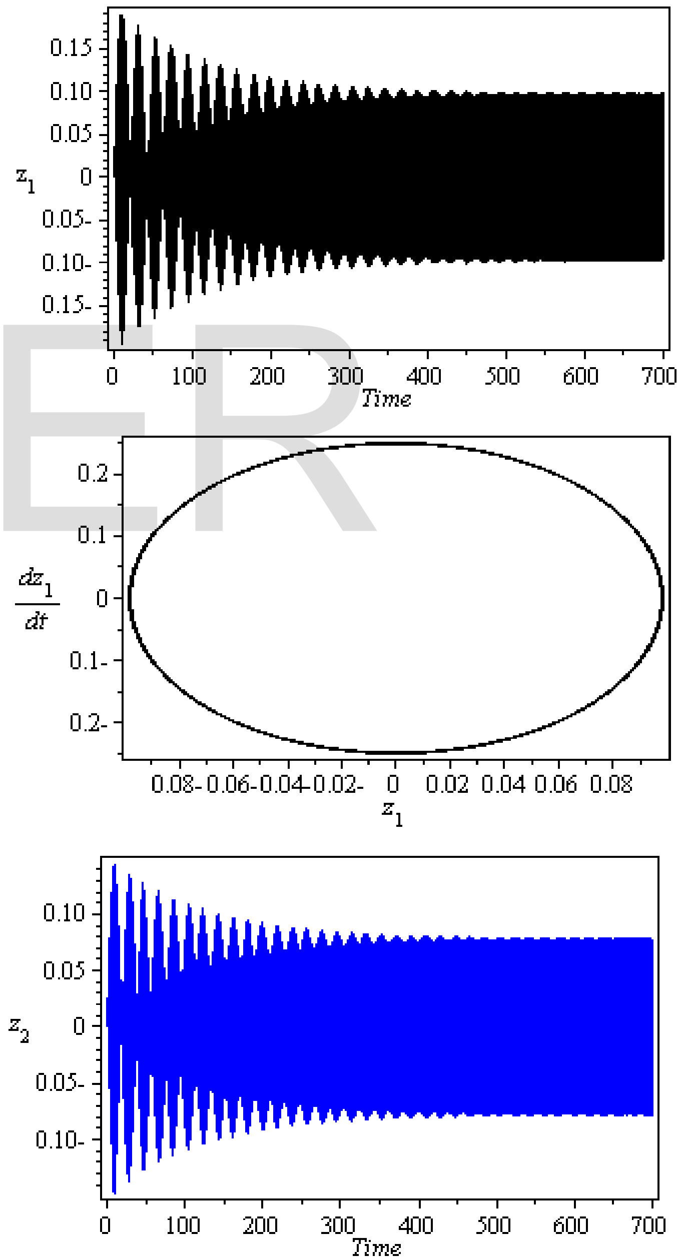
$$2i\omega_2 D_1 A_2=-2i\,\omega_2\mu_2 A_2-3\alpha_2 A_2^2\bar{A}_2+\left(\frac{1-i}{2}\right)f_1\exp(i\sigma_3 T_1)+\left(\frac{1-i}{2}\right)f_2\bar{A}_1\exp(i\sigma_4 T_1) \quad (20)$$

Appling the same process as the stability of the first mode to Eq. (20), the frequency equations for angular vibration can be obtained as

$$\sigma_3^2+\left(\frac{3\alpha_2 a_2^2}{4\omega_2}-2\sigma_4\right)\sigma_3+\left(\mu_2^2+\sigma_4^2+\frac{9\alpha_2^2 a_2^4}{64\omega_2^2}-\frac{3\alpha_2\sigma_4 a_2^2}{4\omega_2}-\frac{f_1^2}{2a_2^2\omega_2^2}-\frac{f_2^2}{8\omega_2^2}-\frac{f_1 f_2}{2a_2\omega_2^2}\right)=0 \quad (21)$$

5. Numerical results

To determine the numerical solution and response of the given system of equations (1) and (2), the Runge-Kutta of fourth order method was applied. Fig. 2 illustrates the response and the phase-plane for the non-resonant system (basic case) where $\Omega\neq\Omega_1\neq\omega_1\neq\omega_2$ at some practical values of the equation parameters $\mu_1=0.00825$, $\mu_2=0.00825$, $\alpha_1=0.05$, $\alpha_2=0.05$, $f_1=0.1$, $f_2=0.002$, $\Omega=2.55$, $\Omega_1=2.44$, $\omega_1=2.25$, $\omega_2=2.88$. It is observed from this figure that the response of the first and second modes of the vertical shaking conveyor system start with increasing amplitude with some chaotic and tuned oscillation respectively, the oscillation of the two modes becomes stable and the steady state amplitudes z_1 and z_2 are about **0.1** and **0.08** respectively and the phase plane shows limit cycle, denoting that the system is free from chaos.



International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

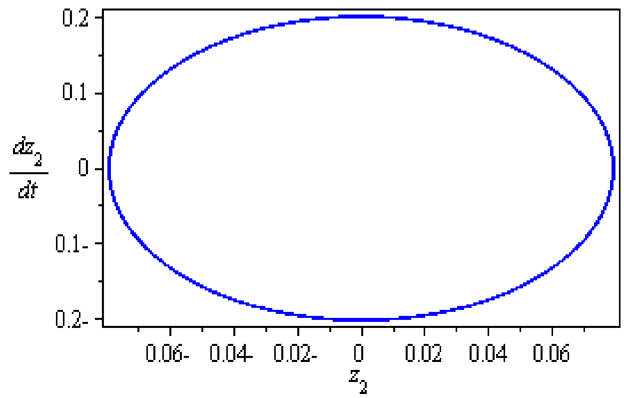
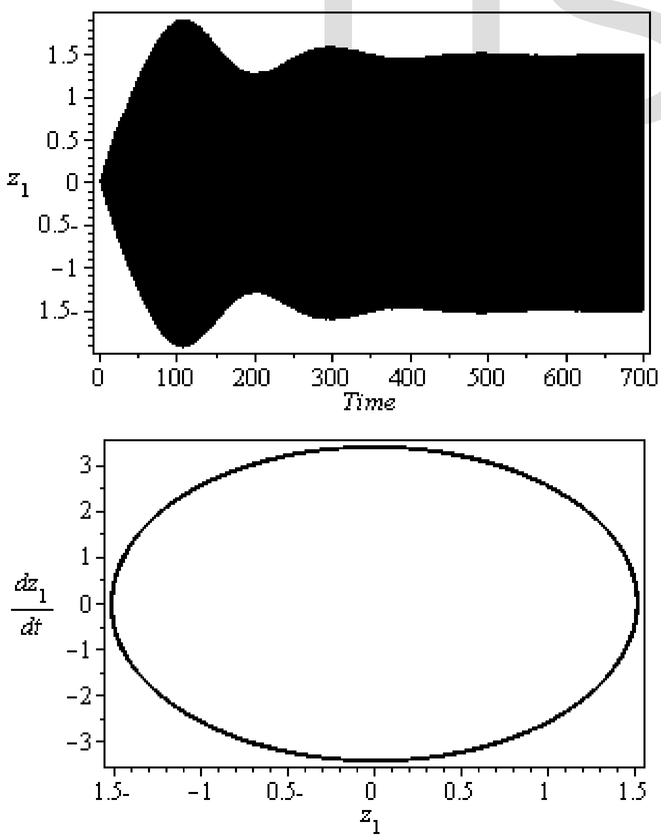


Fig. 2 Non-resonance system behavior (basic case)
$\Omega \neq \Omega_1 \neq \omega_1 \neq \omega_2$

Fig. 3 shows that the time response and phase-plane of the simultaneous primary and sub-harmonic resonance $\Omega \cong \omega_n$ and $\Omega_1 \cong 2\omega_n$, n= (1, 2) which is one of the worst resonance cases. From this figure we have that the amplitude of the first mode of the vertical shaking conveyor system is increased to about **1500%** of that values shown in Fig. 2., while the amplitude of the second mode is increased to about **1875%** and becomes stable and the phase plane shows multi-limit cycle.



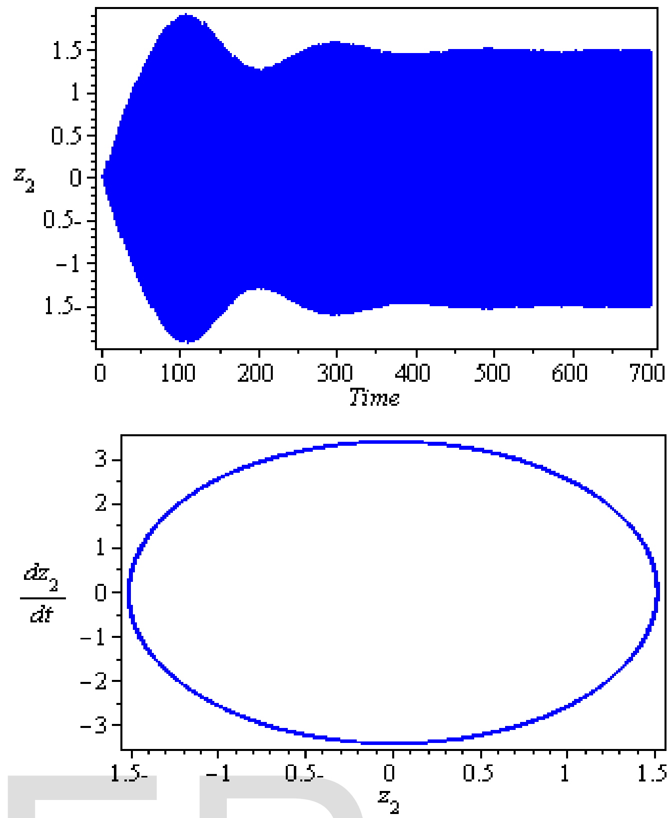


Fig. 3. Simultaneous primary and sub-harmonic resonance case ($\Omega \cong \omega_n$ and $\Omega_1 \cong 2\omega_n$, n= (1, 2)).

5.1. Response Curves and Effects of Different Parameters

In this section, the frequency response equations given by Eqs. (18) and (21) are solved numerically at the same values of the parameters shown in Fig. 2. Fig. 4a shows the steady state amplitudes of the first mode of the vertical shaking conveyor system against the detuning parameter σ_1 at the practical case, where $a_1 \neq 0, a_2 \neq 0$. Fig. 4b shows that the steady state amplitude of the first mode of the vertical shaking conveyor system is a monotonic decreasing function in the linear damping coefficient μ_1. For negative and positive values of the nonlinear parameter α_1 the curves are bent to the right and left and have harding and softing spring type and there exists jump phenomena and multi-valued amplitudes as shown in Fig. 4c. Fig. 4d shows that the steady state amplitude of the first mode is a monotonic decreasing function in the natural frequency ω_1, in this Figure, the response curves are bent to the right and have harding spring type and there exists jump phenomena. The steady state amplitude of the first mode is a monotonic increasing function in the excitation amplitudes f_1 and f_2 as shown in Figs. 4 (e, f).

International Journal of Scientific & Engineering Research Volume ,5, Issue 11, November-2014
ISSN 2229-5518

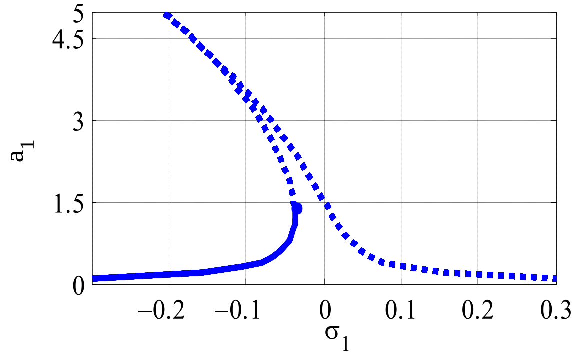


Fig.4a. Effects of the detuning parameter σ_1

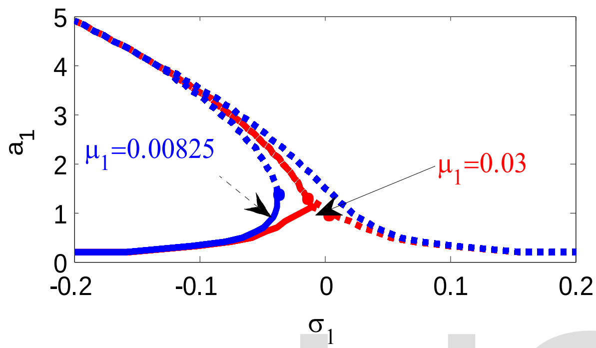


Fig.4b. Effects of the damping coefficient μ_1

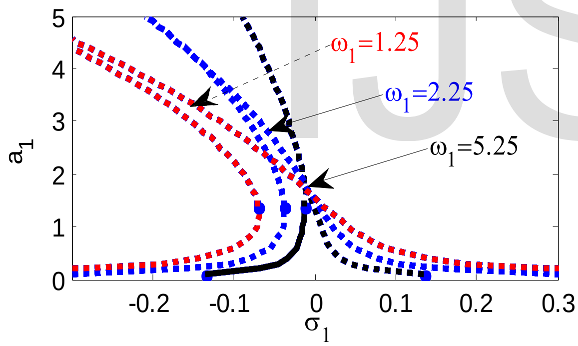


Fig. 4c. Effect of the natural frequency ω_1

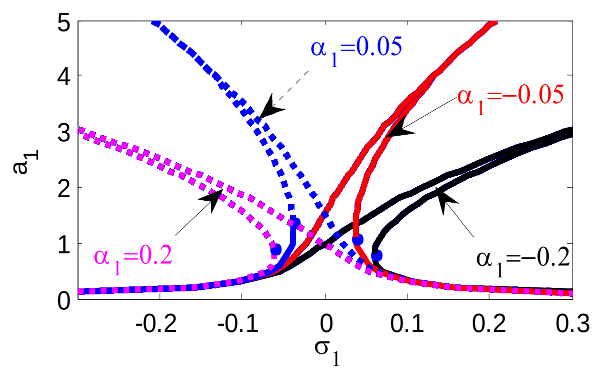


Fig. 4d. Effects of the non-linear parameter α_1

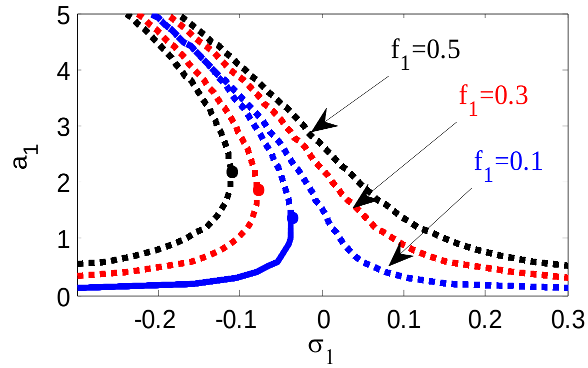


Fig.4e. Effects of the excitation amplitude f_1

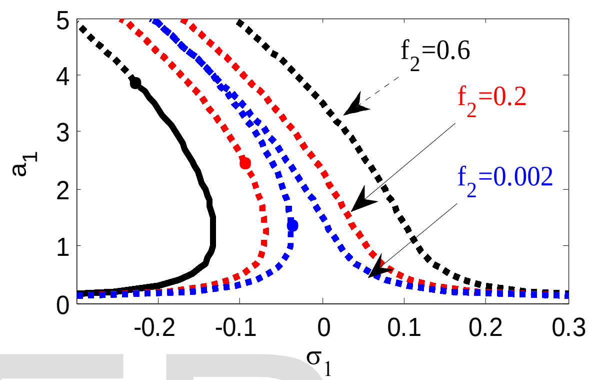


Fig. 4f. Effects of the excitation amplitude f_2

Fig. 5a shows the steady state amplitudes of the second mode of the vertical shaking conveyor system against the detuning parameter σ_3 at the practical case, where $a_1 \neq 0, a_2 \neq 0$. Fig. 5b shows that the steady state amplitude of the second mode of the vertical shaking conveyor system is a monotonic decreasing function in the linear damping coefficient μ_2. For negative and positive values of the nonlinear parameter α_2, the curves are bent to the right and left and have harding and softing spring type and there exists jump phenomena and multi-valued amplitudes as shown in Fig. 5c. Fig. 5d shows that the steady state amplitude of the second mode is a monotonic decreasing function in the natural frequency ω_2, in this Figure, the response curves are bent to the right and have harding spring type and there exists jump phenomena. The steady state amplitude of the second mode is a monotonic increasing function in the excitation amplitudes f_1 and f_2 as shown in Figs. 5 (e, f).

International Journal of Scientific & Engineering Research Volume 5, Issue 11, November-2014
ISSN 2229-5518

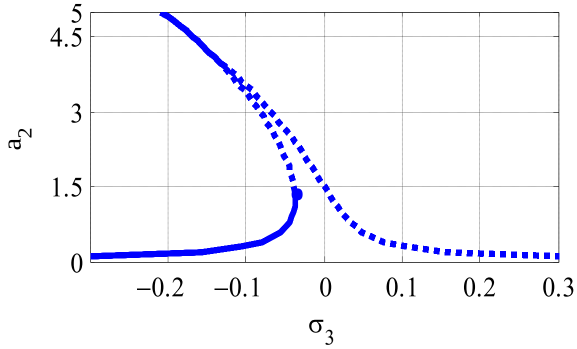


Fig.5a. Effects of the detuning parameter σ_3

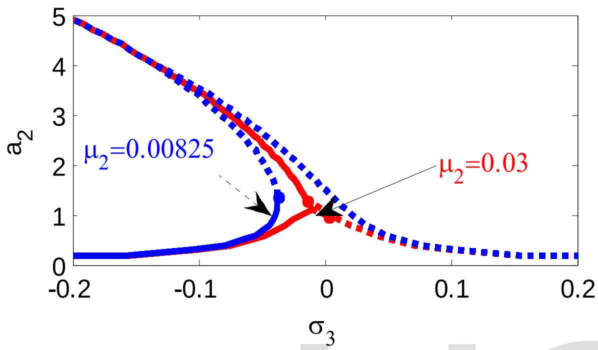


Fig.5b. Effects of the damping coefficient μ_2

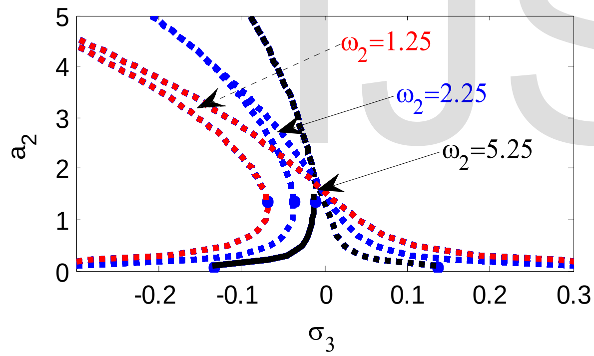


Fig. 5c. Effect of the natural frequency ω_2

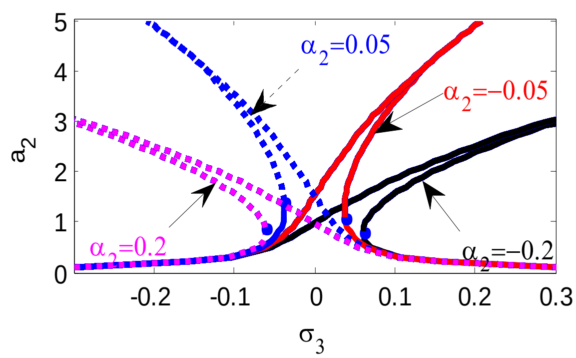


Fig. 5d. Effects of the non-linear parameter α_2

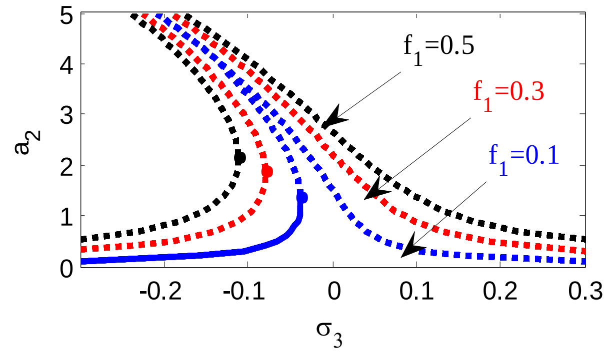


Fig.5e. Effects of the excitation amplitude f_1

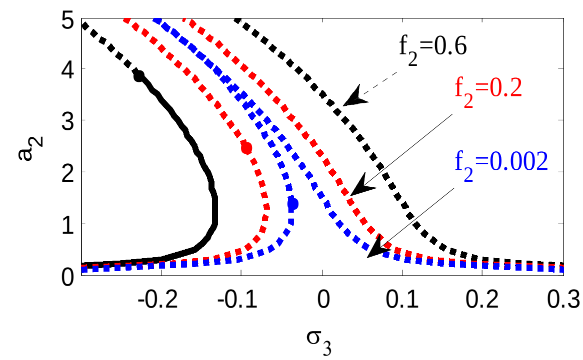


Fig. 5f. Effects of the excitation amplitude f_2

6. Conclusions

The nonlinear responses of the vertical shaking conveyor system subjected to external and parametric excitations have been studied. The problem is described by a two-degree-of-freedom system of nonlinear ordinary differential equations. The case of simultaneous primary and principle parametric resonance is studied by applying multiple time scale perturbation method. Both the frequency response equations and the phase-plane technique are applied to study the stability of the system. The effect of the different parameters of the system is studied numerically. From the above study the following may be concluded:

1. The oscillation of the two modes becomes stable and the steady state amplitudes z_1 and z_2 are about **0.1** and **0.08** at the non-resonant case ($\Omega \neq \Omega_1 \neq \omega_1 \neq \omega_2$).
2. the oscillation of the first mode of the vertical shaking conveyor system is increased to about **1500%,** while the amplitude of the second mode is increased to about **1875%** at the worst resonance case (the simultaneous primary and sub-harmonic resonance ($\Omega \cong \omega_n$ and $\Omega_1 \cong 2\omega_n$).
3. The steady state amplitudes of the first and second modes of the vertical shaking conveyor system are a monotonic decreasing function in the linear damping coefficients μ_1 and μ_2.

4. For negative and positive values of the nonlinear parameters α_1 and α_2, the curves are bent to the right and left and have harding and softing spring type and there exists jump phenomena and multi-valued amplitudes.
5. The steady state amplitude of the first and second modes of the vertical shaking conveyor system are monotonic decreasing functions in the natural frequencies ω_1 and ω_2, the response curves are bent to the right and have harding spring type and there exists jump phenomena.
6. The steady state amplitudes of the first and second modes are monotonic increasing functions in the excitation amplitudes f_1 and f_2.

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