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WAVE PROPAGATION ANALYSIS OF EDGE CRACKED BEAMS RESTING ON ELASTIC FOUNDATION

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Abstract

This paper presents responses of an edge circular cantilever beam resting on Winkler-Pasternak foundation under the effect of an impact force. The beam is excited by a transverse triangular force impulse modulated by a harmonic motion. The Kelvin–Voigt model for the material of the beam is used. The cracked beam is modelled as an assembly of two sub-beams connected through a massless elastic rotational spring. The considered problem is investigated within the Bernoulli-Euler beam theory by using energy based finite element method. The system of equations of motion is derived by using Lagrange's equations. The obtained system of linear differential equations is reduced to a linear algebraic equation system and solved in the time domain by using Newmark average acceleration method. In the study, the effects of the foundation stiffness on the characteristics of the reflected waves and cracks are investigated in detail.

Keywords: Open edge crack, Wave propagation, Additional Wave, Winkler-Pasternak Foundation

1. Introduction

Elastic wave propagation through the monitored part is of considerable interest in many fields. The most striking example of the engineering applications is detection of damage or/and material difference in the investigated media. By investigating the character of waves, the type and position of damage or/and different material can be determined.

In the last decades, much more attention has been given to the elastic wave propagation of beam structures. Teh and Huang [1] studied an analytical model, based on the elasticity equations, to investigate wave propagation in generally orthotropic beams. A finite element technique is developed for studying the flexural wave propagation in elastic Timoshenko and Bernoulli-Euler beams by Yokoyama and Kishida [2]. Wave propagation in a split beam is analyzed by treating each section separately as a waveguide and imposing appropriate connectivities at their joints by Farris and Doyle [3]. A direct mathematical approach method is developed to study the problem of coupled longitudinal and flexural wave propagation in a periodically supported infinite long beam by Lee and Yeen [4]. A spectral super-element model was used in Gopalakrishnan and Doyle [5] to model transverse crack in isotropic beam and the dynamic stress intensity factor was obtained accurately under impact type loading. Palacz and Krawczuk [6] investigated longitudinal wave propagation in a cracked rod by using the spectral element method. The use of the wave propagation approach combined with a genetic algorithm and the gradient technique for damage detection in beam-like structure is investigated by Krawczuk [7]. Krawczuk et al. [8] studied a new finite spectral element of a cracked Timoshenko beam for modal and elastic wave propagation analysis. Usuki and Maki [9] formulated an equation of motion for a beam according to higher-order beam theory using Reissner's principle. They derived the Laplace transform of the equation and investigated wave-propagation behavior under transverse impact. A method of crack detection in beam is provided by wavelet analysis of transient flexural wave by Tian et al. [10]. Kang et al. [11] applied the wave approach based on the reflection, transmission and propagation of waves to obtain the natural frequencies of finite curved beams. A spectral finite element with embedded transverse crack is developed and implemented to simulate the diagnostic wave scattering in composite beams with various forms of transverse crack by Kumar et al. [12]. The wave propagation model investigated herein is based on the known fact that material discontinuities affect the propagation of elastic waves in solids by Ostachowicz et al. [13]. A spectral finite element model for analysis of flexural-shear coupled wave propagation in laminated and delaminated, multilayer composite beams is presented by Palacz et al. [14,15]. A new spectral element is formulated to analyse wave propagation in an anisotropic inhomogeneous beam by Chakraborty and Gopalakrishnan [16]. Watanabe and Sugimoto [17] studied flexural wave propagation in a spatially periodic structure consisting of identical beams of finite length. Vinod et al. [18] investigated a formulation of an approximate spectral element for uniform and tapered rotating Euler–Bernoulli beams. Sridhar et al. [19] investigated the development of an effective numerical tool in the form of pseudospectral method for wave propagation analysis in anisotropic and inhomogeneous structures. An experimental method of detecting damage using the flexural wave propagation characteristics is proposed by Park [20]. Chouvion et al. [21] studied a systematic wave propagation approach for the free vibration analysis of networks consisting of slender, straight and curved beam elements and complete rings. Frikha et al. [22] investigated physical analysis of the effect of axial load on the propagation of elastic waves in helical beams. Kocatürk et al.[23] studied wave propagation of a piecewise homegenous cantilever beam under impact force. Kocatürk and Akbas [24] investigated wave propagation of a microbeam with the modified couple stress theory. In a recent study, wave propagation and localization in periodic and randomly disordered periodic piezoelectric axial-bending coupled beams are studied by Zhu et al. [25].

In this study, wave propagation in a cantilever circular beam resting on Winkler-Pasternak foundation under the effect of an impact force is studied. The considered problem is investigated within the Bernoulli-Euler beam theory by using energy based finite element method. The Kelvin–Voigt model for the material of the beam is used. The cracked beam is modelled as an assembly of two sub-beams connected through a massless elastic rotational spring. The system of equations of motion is derived by using Lagrange's equations. The obtained system of linear differential equations is reduced to a linear algebraic equation system and solved in the time domain by using Newmark average acceleration method. The effects of the foundation stiffness on the characteristics of the reflected waves and cracks are investigated in detail.

2. Theory and Formulations

Consider a beam of length *L*, diameter *D*, containing an edge crack of depth *a* located at a distance L_1 from the left end, resting on Winkler-Pasternak foundation with spring constant k_w and k_p, as shown in Fig. 1. It is assumed that the crack is perpendicular to beam surface and always remains open. When the Pasternak foundation spring constant $k_p = 0$, the foundation model reduces to Winkler type. The beam is subjected to an impact force in the transverse direction as seen from Fig. 1.

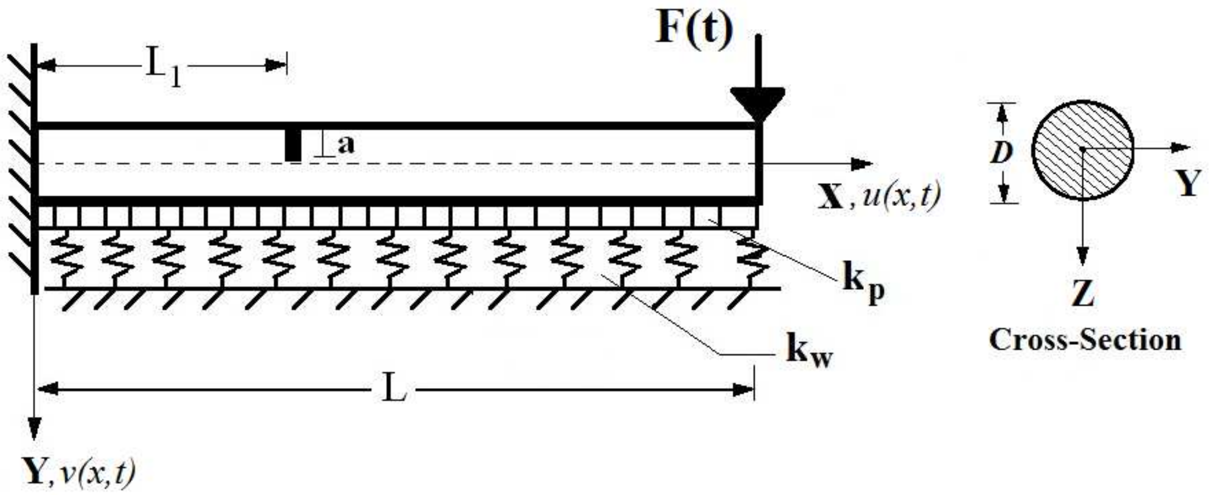


Fig. 1 A circular beam with an open edge crack resting on Winkler-Pasternak foundation and cross-section.

2.1 Governing equations of intact beam

The beam is modeled within the Euler-Bernoulli beam theory. According to the coordinate system *(X,Y,Z)* shown in Fig. 1, based on Euler-Bernoulli beam theory, the axial and the transverse displacement field are expressed as

$$u(X,Y,t) = -Y\frac{\partial v(X,t)}{\partial X} \tag{1}$$

$$v(X,Y,t) = v(X,t) \tag{2}$$

$$w(X,Y,t) = 0 \tag{3}$$

Where u,v and w are x,y and z components of the displacement vector q, respectively, and t indicates time.

Because the transversal surfaces of the beam is free of stress, then

$$\sigma_{ZZ} = \sigma_{YY} = 0 \tag{4}$$

By using Eqs. (1) and (2) the linear strain- displacement relation can be obtained:

$$\varepsilon_{xx} = -Y\frac{\partial^2 v(X,t)}{\partial X^2} \tag{5}$$

According to Hooke's law, constitutive equations of the beam are as follows:

$$\sigma_{xx} = E\,\varepsilon_{xx} = E\left(-Y\frac{\partial^2 v(X,t)}{\partial X^2}\right) \tag{6}$$

Where E is the Young's modulus of the beam, σ_{xx} and ε_{xx} are normal stresses and normal strains in the X direction, respectively. The potential energy of the beam is follows

$$U_i = \frac{1}{2}\int_0^L \int_A \sigma_{xx}\,\varepsilon_{xx}\,dA\,dX + \frac{1}{2}\int_0^L k_w\,(v(x,t))^2\,dX + \frac{1}{2}\int_0^L k_p\left(\frac{\partial v(x,t)}{\partial x}\right)^2 dX \tag{7}$$

The kinetic energy of the beam at any instant *t* is

$$T = \frac{1}{2}\int_0^L \int_A \rho\left(\frac{\partial v(X,t)}{\partial t}\right)^2 dA\,dX \tag{8}$$

Where ρ is the mass density of the beam. The potential energy of the external load can be written as

$$U_e = -\int_{x=0}^{L} F(X,t)\,v(X,t)\,\mathrm{d}x. \tag{9}$$

The Kelvin–Voigt model for the material is used. The constitutive relations for the Kelvin–Voigt model between the stresses and strains become

$$\sigma = E(\varepsilon + \eta\dot{\varepsilon}) \tag{10}$$

where η is the damping ratios, as follows

$$\eta = \frac{c}{E} \tag{11}$$

where c is the coefficient of damping of the beam. In this case, the dissipation function of the beam at any instant *t* is

$$R = \frac{1}{2}\int_0^L \eta E I\left(\frac{\partial^2 \dot{v}(X,t)}{\partial X^2}\right)^2 dx \tag{12}$$

Lagrangian functional of the problem is given as follows:

$$I = T - (U_i + U_e) \tag{13}$$

2.2 Solution method of the problem

The problem is solved by using Lagrange's equations and time integration method of Newmark [26]. In order to apply the Lagrange's equations, the displacements of nodes of the unknown functions *q* (X,t) which is written for a two-node beam element shown in Fig. 2 are defined as follows

$$\{q(t)\}_{(e)} = \left[v_i^{(e)}(t)\;\; \theta_i^{(e)}(t)\;\; v_j^{(e)}(t)\;\; \theta_j^{(e)}(t)\right]^T \tag{14}$$

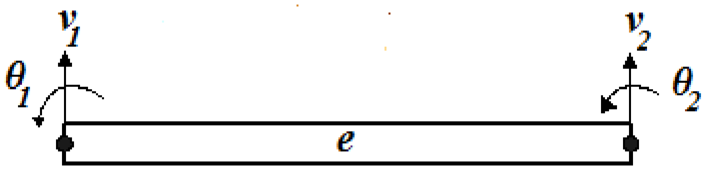


Fig. 2 A two-node beam element

The displacement field of the finite element is expressed in terms of nodal displacements as follows

$$v^{(e)}(X,t) = \varphi_1(X)v_1^{(e)}(t) + \varphi_2(X)\theta_1^{(e)}(t) + \varphi_3(X)v_2^{(e)}(t) + \varphi_4(X)\theta_2^{(e)}(t) \tag{15}$$

where $\varphi_1, \varphi_2, \varphi_3$ and φ_4 are interpolation functions and given as follows:

$$\begin{aligned}
\varphi_1(X) &= 1 - 3(X/L_e)^2 + 2(X/L_e)^3 \\
\varphi_2(X) &= L_e\left(-(X/L_e) + 2(X/L_e)^2 - (X/L_e)^3\right) \\
\varphi_3(X) &= 3(X/L_e)^2 - 2(X/L_e)^3 \\
\varphi_4(X) &= L_e\left((X/L_e)^2 - (X/L_e)^3\right)
\end{aligned} \tag{16}$$

where L_e is the length of the beam element. After substituting Equation (15) into Eq. (13) and then using the Lagrange's equations gives the following equation;

$$\frac{\partial I}{\partial q_k^{(e)}} - \frac{d}{dt}\frac{\partial I}{\partial \dot{q}_k^{(e)}} + Q_{D_k} = 0, \qquad k = 1, 2, 3, \ldots \tag{17}$$

where

$$Q_{D_k} = -\frac{\partial R}{\partial \dot{q}_k^{(e)}}, \qquad k = 1, 2, 3, \ldots \tag{18}$$

Q_{D_k} is the generalized damping load which can be obtained from the dissipation function by differentiating R with respect to $\dot{q}_k^{(e)}$.

The Lagrange's equations yield the system of equations of motion for the finite element and by use of usual assemblage procedure the following system of equations of motion for the whole system can be obtained as follows

$$[K]\{q(t)\} + [D]\{\dot{q}(t)\} + [M]\{\ddot{q}(t)\} = \{F(t)\} \tag{19}$$

where

$$[M] = \int_{x=0}^{L} \rho A \{\varphi(X)\}^T \{\varphi(X)\}\, dX \tag{20}$$

$$[D] = \int_{x=0}^{L} \eta \, EI \, \{\varphi''(X)\}^T \{\varphi''(X)\} \, dX \qquad (21)$$

$$\{F(t)\} = \int_{x=0}^{L} \{\varphi(X)\}^T F(X,t) \, dX \qquad (22)$$

$$[K] = \int_{x=0}^{L} \{\varphi''(X)\}^T EI \{\varphi''(X)\}^T \, dX \qquad (23)$$

where, $[K]$ is the stiffness matrix, $[D]$ is the damping matrix, $[M]$ is mass matrix and $\{F(t)\}$ is the load vector. The motion equations which is given by Eq. (19), are solved in the time domain by using Newmark average acceleration method (Newmark [26]).

2.3. Crack modeling

The cracked beam is modeled as an assembly of two sub-beams connected through a massless elastic rotational spring shown in Fig. 3.

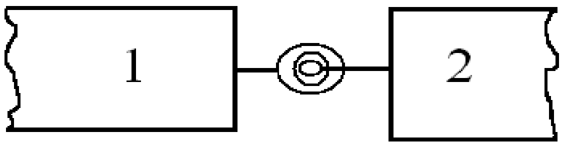


Fig. 3 Rotational spring model

The bending stiffness of the cracked section k_T is related to the flexibility G by

$$k_T = \frac{1}{G} \qquad (24)$$

Cracked section's flexibility G can be derived from Broek's approximation (Broek [27]):

$$\frac{(1-\nu^2)K_I^2}{E} = \frac{M^2}{2}\frac{dG}{da} \qquad (25)$$

where M is the bending moment at the cracked section, K_I is the stress intensity factor (SIF) under mode I bending load and is a function of the geometry and the loading properties as well. ν indicates Poisson's ratio. For circular cross section, the stress intensity factor for K_I a single edge cracked beam specimen under pure bending M can be written as follow (Tada et al. [28])

$$K_I = \frac{4M}{\pi R^4}\frac{h'_x}{2}\sqrt{\pi a}\, F(a/h'_x) \qquad (26)$$

Where

$$F(a/h'_x) = \sqrt{\frac{2h'_x}{\pi a}\,\mathrm{tg}\left(\frac{\pi a}{2h'_x}\right)}\;\frac{0.923 + 0.199\left(1-\sin\left(\frac{\pi a}{2h'_x}\right)\right)^4}{\cos\left(\frac{\pi a}{2h'_x}\right)} \qquad (27)$$

Where a is crack of depth and h'_x is the height of the strip, is shown Fig. 4, and written as

$$h'_x = 2\sqrt{R^2 - x^2} \tag{28}$$

where R is the radius of the cross section of the beam.

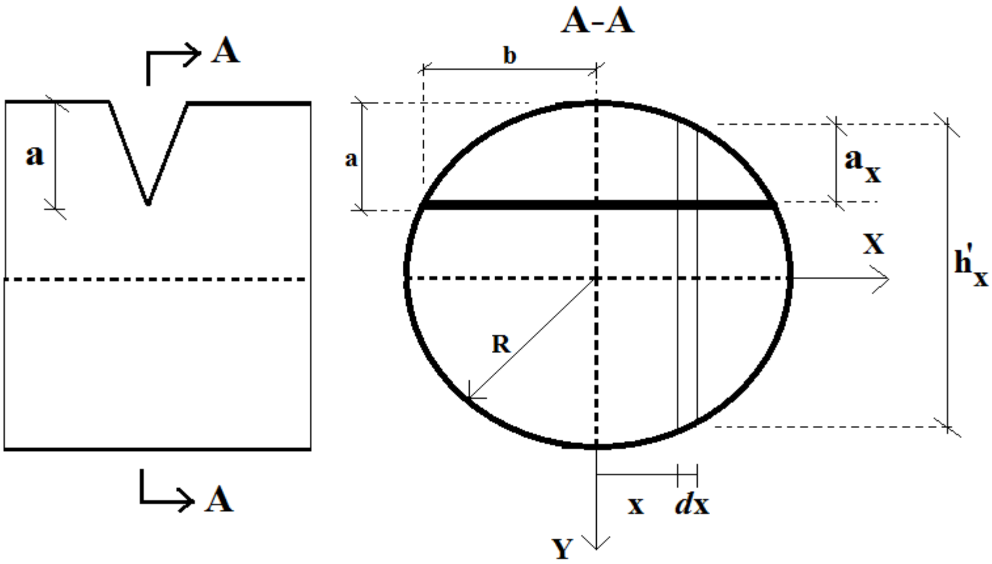


Fig. 4 The geometry of the cracked circular cross section

After substituting Eq. (26) into Eq. (25) and by integrating Eq. (25), the flexibility coefficient of the crack section G is obtained as

$$G = \frac{32(1-\nu^2)}{E\pi R^8} \int_{-b}^{b} \int_{0}^{a_X} y\,(R^2 - x^2)\,F^2\,(a/h'_x)\,dy\,dx \tag{29}$$

where b and a_X are the boundary of the strip and the local crack depth respectively, are shown in Fig. 4, respectively, and written as

$$b = \sqrt{R^2 - (R-a)^2} \tag{30}$$

$$a_x = \sqrt{R^2 - x^2} - (R - a) \tag{31}$$

The spring connects the adjacent left and right elements and couples the slopes of the two beam elements at the crack location. In the massless spring model, the compatibility conditions enforce the continuities of the axial displacement, transverse deflection, axial force and bending moment across the crack at the cracked section ($X = L_1$), that is,

$$v_1 = v_2\,,\quad M_1 = M_2 \tag{32}$$

The discontinuity in the slope is as follows:

$$k_T\left(\frac{dv_1}{dX} - \frac{dv_2}{dX}\right) = k_T(\theta_1 - \theta_2) = M_1 \tag{33}$$

Based on the massless spring model, the stiffness matrix of the cracked section as follows:

$$[K]_{(Cr)} = \begin{bmatrix} 1/G & -1/G \\ 1/G & 1/G \end{bmatrix} = \begin{bmatrix} k_T & -k_T \\ k_T & k_T \end{bmatrix} \tag{34}$$

The stiffness matrix of the cracked section is written according to the displacement vector:

$$\{q\}_{(Cr)} = \{\theta_1, \theta_2\}^T \tag{35}$$

Where θ_1 and θ_2 are the angles of the cracked section. With adding crack model, the equations of motion for the finite element and by use of usual assemblage procedure the following system of equations of motion for the whole system can be obtained as follows:

$$([K] + [K]_{(Cr)})\{q(t)\} + [D]\{\dot{q}(t)\} + [M]\{\ddot{q}(t)\} = \{F(t)\} \tag{37}$$

The dimensionless quantities can be expressed as

$$\bar{k}_w = \frac{k_w L^4}{EI}, \quad \bar{k}_p = \frac{k_p L^2}{EI} \tag{38}$$

$\bar{k}_w$ is the dimensionless Winkler parameter and $\bar{k}_p$ is the dimensionless Pasternak parameter,

3. Numerical Results

In the numerical examples, the effects of the foundation stiffness on the characteristics of the reflected waves and cracks are presented. In the numerical study, the physical properties of the pile are Young’s modulus E=70 *GPa*, Poisson’s ratio v=0,3 and mass density ρ=2700 *kg/m3*. The geometrical properties of the pile are length L=3*m* and the diameter D= 2 cm. The problem is analyzed within the framework of the Bernoulli–Euler beam theory. Numerical calculations in the time domain are made by using Newmark average acceleration method. The system of linear diferantial equations which are given by Equation (19), is reduced to a linear algebraic system of equations by using average acceleration method. In the numerical calculations, the number of finite elements is taken as n = 100. The beam is excited by a transverse triangular force impulse (with a peak value 1 N) modulated by a harmonic function (Fig. 5) (Ostachowicz et al., [13]). In this study, higher frequency excitation impulse is used for detection of the cracks.

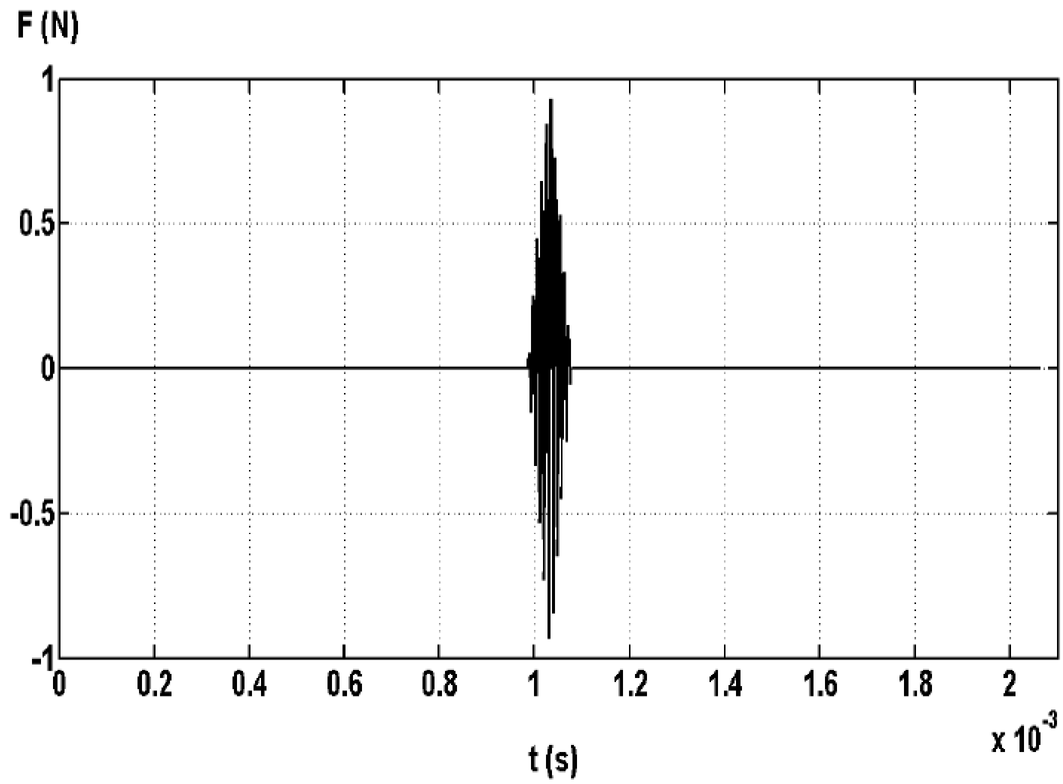
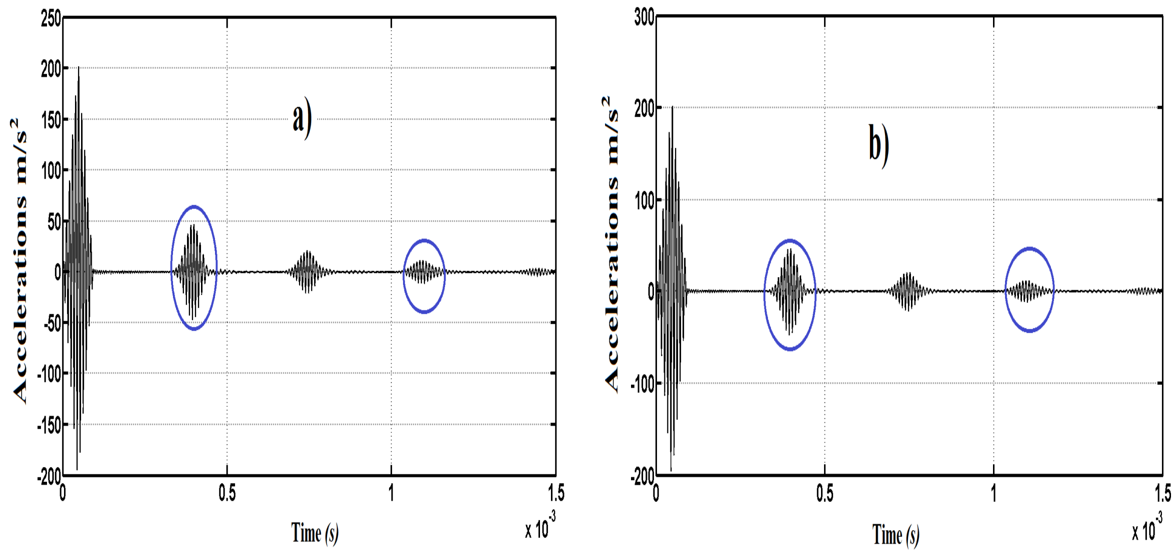


Fig. 5 The shape of the excitation impulse in the time domain

Figure 6 illustrates the effect of the dimensionless Winkler parameter $\overline{k}_w$ on the transverse accelerations at the free end of the cantilever beam for the crack depth ratio *a/D*=0.4, the crack location L_1/L=0.5 and dimensionless Pasternak parameter $\overline{k}_p = 0$.



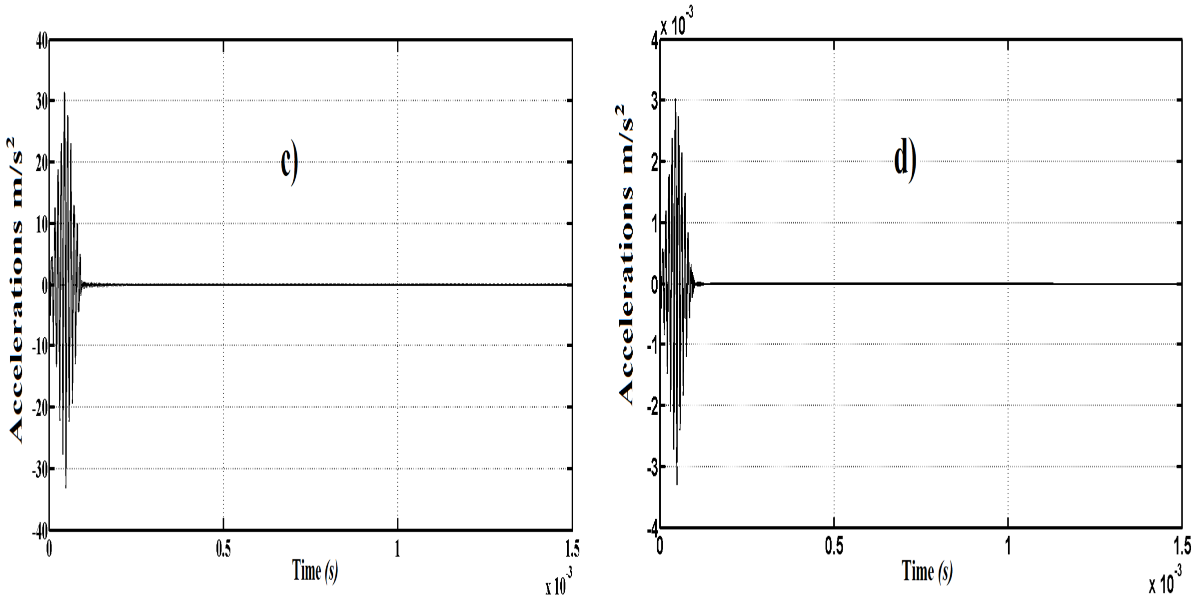


Fig. 6 Transverse accelerations at the free end of the beam. a) $\overline{k}_w = 0$, b) $\overline{k}_w = 10^8$, c) $\overline{k}_w = 10^{12}$ and d) $\overline{k}_w = 10^{16}$

It is seen from Figure 6 that additional waves occur in case of the cracked beam (see the circles) because of reflecting from the cracks. With the increase in the dimensionless Winkler parameter $\overline{k}_w$, amplitude of waves increases seriously. This is because by increasing in $\overline{k}_w$, the beam gets more stiffer. Also, it is observed from figure 6 that With the increase in the dimensionless Winkler parameter $\overline{k}_w$, the reflected waves disappear. It shows that the Winkler parameters $\bar{k}_w$ play important role on the wave propagation of the beam.

In figure 7, the effect of the dimensionless Pasternak parameter $\overline{k}_P$ on the transverse accelerations at the free end of the cantilever beam for the crack depth ratio *a/D*=0.4, the crack location L_1/L=0.5 and dimensionless Winkler parameter $\overline{k}_w = 10^8$.

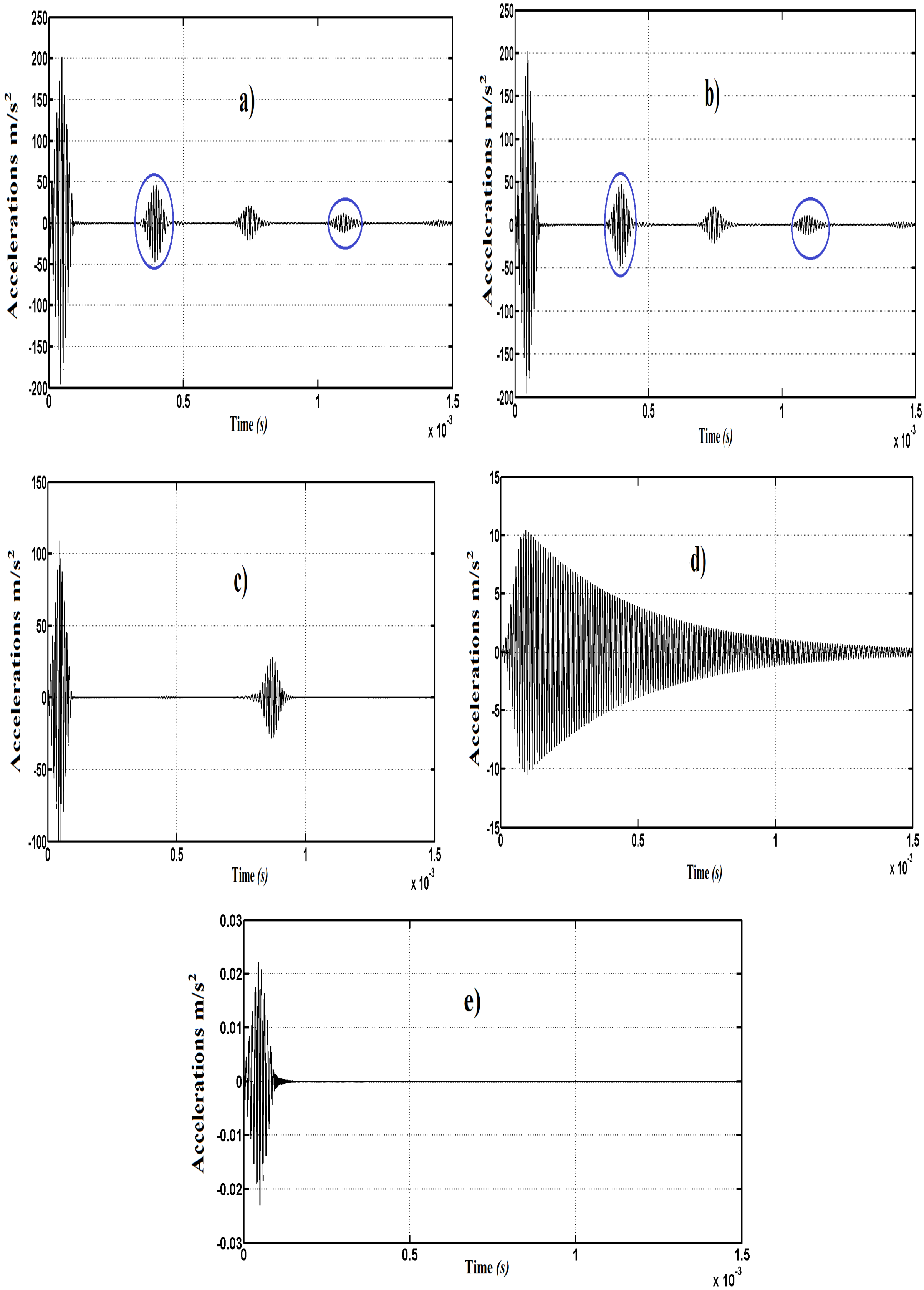


Fig. 7 Transverse accelerations at the free end of the beam. a) $\overline{k}_p = 0$, b) $\overline{k}_p = 10^3$, c) $\overline{k}_p = 10^6$, d) $\overline{k}_w = 10^9$ and e) $\overline{k}_w = 10^{12}$

It is seen from Fig. 7 that with increase in the dimensionless Pasternak parameter $\overline{k}_P$, amplitude of waves increases as expected. It is observed from figures 6 and 7 that Pasternak parameter $\overline{k}_P$ is more effective than Winkler parameter $\overline{k}_w$. It shows that Pasternak parameter is very effective for wave propagation.

It is deduced from figures that the stiffness parameters of the foundation are very effective for reducing the negative influence of the cracks. With the increase foundation parameters, the generation time and location of the primary and additional waves decreases.

4. Conclusions

Wave propagation in an edge circular cantilever beam resting on Winkler-Pasternak foundation under the effect of an impact force is investigated. The considered problem is investigated within the Bernoulli-Euler beam theory by using energy based finite element method. The system of equations of motion is derived by using Lagrange's equations. The obtained system of linear differential equations is reduced to a linear algebraic equation system and solved in the time domain by using Newmark average acceleration method. The stiffness parameters of the foundation have a great influence on wave propagation in the beam. It is observed from the investigations that the stiffness parameters of the foundation are very effective for reducing the negative influence of the cracks. It is seen from results that the Pasternak parameter is more effective than Winkler parameter on the wave propagation.

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