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Big data analysis via computer and semi numerical simulations for dynamic responses of complex nanosystems

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Abstract. In the present research, for the first time, the vibrational as well as buckling characteristics of a three-layered curved nanobeam including a core made of functionally graded (FG) material and two layers of smart material—piezo-magneto-electric—resting on a Winkler Pasternak elastic foundation are examined. The displacement field for the nanobeam is chosen via Timoshenko beam theory. Also, the size dependency is taken into account by using nonlocal strain gradient theory, aka NSGT. Then, by employing Hamilton's principle, energy procedure, the governing equations together with the boundary conditions are achieved. The solution procedure is a numerical solution called generalized differential quadrature method, or GDQM. The accuracy and reliability of the formulation alongside solution method is examined by using other published articles. Lastly, the parameter which can alter and affect the buckling or vocational behavior of the curved nanobeam is investigated in details.
Keywords: big data analysis; complex nanosystems; computer and semi numerical simulations; dynamic responses

1. Introduction

The recent technological advances require the developments of nanoelectromechanical systems (Dai and Safarpour 2021, Forsat *et al*. 2021, Ghamkhar *et al*. 2021, Khadimallah *et al*. 2021a, b, Kumar *et al*. 2021, Madenci 2021, Tlidji *et al*. 2021). Given this, the researchers have carried out studies on devices such as nanosensors, nano-actuators, and nanoharvesters to name but a few (Habibi *et al*. 2016, 2018a, b, 2019d, f, g, Ebrahimi *et al*. 2019a, Esmailpoor Hajilak *et al*. 2019, Pourjabari *et al*. 2019, Safarpour *et al*. 2019a). By knowing this, it should be noted that the theoretical studies in this area is a vital step to investigate various parameters affecting the devices in order to enhance the designing these devices (Habibi *et al*. 2017, Safarpour *et al*. 2018, 2019b, 2020, Habibi *et al*. 2019b, e, Alipour *et al*. 2020, Ebrahimi *et al*. 2020a, Ghazanfari *et al*. 2020, Chen *et al*. 2022). Also, the other significant fact that should be considered is that the classical continuum mechanics are unable to through which explore the characteristics of small scaled structures. Thus, scholars have put a lot of effort to introduce and present new types of elasticities to capture the effect of small size. In this regard, integral nonlocal was presented (Ebrahimi *et al*. 2019b, c, 2020b, Hashemi *et al*. 2019, Moayedi *et al*. 2019, 2020a, b, Mohammadgholiha *et al*. 2019, Mohammadi *et al*. 2019, Habibi *et al*. 2020, Oyarhossein *et al*. 2020, Shariati *et al*. 2020a, b, Shokrgozar *et al*. 2020). It is interesting to mention that the differential form of this theory was introduced in order to reduce the complexity as well as costs related to the computations. Therefore, this elasticity has been utilized to study the vibration, buckling, as well as wave propagation of nanostructures (Hashemi *et al*. 2019, Al-Furjan *et al*. 2020e, o, q, s, Bai *et al*. 2020, Cheshmeh *et al*. 2020, Li *et al*. 2020b, Lori *et al*. 2020, Najaafi *et al*. 2020, Shariati *et al*. 2020c, Zhang *et al*. 2020, Guo *et al*. 2021b, Liu *et al*. 2021a).

Based on nonlocal elasticity alongside strain gradient theory, a new type of elasticity to capture the nonuniform strains as well as size effect was introduced, aka nonlocal strain gradient theory (Adamian *et al*. 2020, Al-Furjan *et al*. 2020c, d, Li *et al*. 2020c, Liu *et al*. 2020b, Zare *et al*. 2020, Dai *et al*. 2021b, Habibi *et al*. 2021, He *et al*. 2021, Huang *et al*. 2021a, Liu *et al*. 2021b, Zhang *et al*. 2021). It should be stated that this theory has been employed to study small scaled structures in the past decade, one of which is the work by Li *et al*. (2016) in which they studied the longitudinal vibrational behavior of rods in small scaled, by utilizing nonlocal strain gradient theory. in this paper, they extracted results analytically along with numerically using a finite element model. Also, the vibrational characteristics related to beams are formulated on the basis of Euler-Bernoulli beam theory as well as nonlocal strain gradient theory, or NSGT (Apuzzo *et al*. 2018). Next, it can refer to a paper in which Mehralian *et al*. (2017) investigated the vibration of nanotubes, by employing NSGT. They also tried to calibrate the factors associated with the nonlocal strain gradient theory by utilizing a molecular dynamic model. Additionally, the formulation associated with the buckling together with the vibrational analysis of shells, based on first order shear deformation theory as well as NSGT, were extracted, and then they were solved by means of GDQM (Moayedi *et al*. 2021). By using NSGT and HSDT, or higher order shear deformable theory, the vibrational analysis of nanodisks which are spinning and are made of piezoelectric materials was conducted (Al-Furjan

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et al. 2021b). The solution method which is utilized in the mentioned research was GDQM. In this area, the article by Oyarhossein *et al*. (2020) can be cited where they managed to present an investigation on the vibrational feature of tubes, on the basis of NSGT. In the article which is cited, they utilized first order shear deformation theory in order to obtain the displacement fields.

The pursue to design and manufacture better NEMS// MEMS devices has been one of the major fields of studies for the researcher. Given this, the smart materials are one of the inseparable parts of these structures and devices (Wu *et al*. 2018, 2020, Ning *et al*. 2021, Sheng *et al*. 2021, Liu *et al*. 2022a). Thus, the study on magnetoelectric and piezoelectric structures has been soared in the recent years. Scholars have conducted various studies such as the vibrational, buckling, etc. behavior of structures made of such smart materials (Wu *et al*. 2021, Zheng *et al*. 2021, 2022c, Liu *et al*. 2022c, Shen *et al*. 2022). In this regard, one of the studies which can refer to is the work by Arefi (Arefi 2020) in which the vibration analysis if a piezoelectric panel which is subjected to electrical as well as mechanical loading was investigated. The size dependency in the mentioned article was considered through nonlocal elasticity. Next, the paper in which the critical buckling voltage associated with a microdisk, the material of which is reinforced composite, with a coupled piezoelectric layer as an actuator can be cited (Shamsaddini Lori *et al*. 2021). Additionally, by employing modified couple stress theory, the vibrational behavior of nanobeams which are made of magnetoelectric material and subjected to thermal loading was explored by Habibi *et al*. (2019a). Also, the vibration control corresponded to reinforced composite shell incorporating a layer of piezoelectric was carried out by Al-Furjan *et al*. (2020k). Additionally, by incorporating the nonlocal elasticity together with GDQM as the numerical solution procedure, the nonlinear vibration of magneto-electric plates in the thermal environment was probed (Ansari and Gholami 2016). Next, it ca refer to the investigation on the vibration behavior of shells made of two layers, one of which is piezoelectric and the other is reinforced composites (Habibi *et al*. 2019c).

On the structural designs utilized by engineers in order to build the NEMS/MEMS devices are curved beam (Guo *et al*. 2022, Ju *et al*. 2022, Sun *et al*. 2022, Ye *et al*. 2022, Zhang *et al*. 2022a). There are quite a few studies in this field, one of which is the study carried out by Ganapathi and Polit (2017) where they studied the free as well as forced vibration of curved beam. In this study, the used nonlocal elasticity to investigate the effect of small size. Additionally, the formulation associated with the vibrational analysis of a functionally graded nanobeam made of piezoelectric material was obtained through nonlocal elasticity and was solved via Navier approach (Ebrahimi and Barati 2018). Also, Rahmani *et al*. (2018) explored the vibration of Timoshenko beam by incorporating the modified couple stress theory. A three-layered curved beam was modeled in order to study its vibration along with bending (Arshid *et al*. 2021). The beam was subjected to thermal as well as magnetic loads. Sobhy (2020) in a paper presented an investigation on the buckling together with vibrational response associated with a three-layered beam, the face sheets of which are made of reinforced composite materials. By employing higher order refined beam theory and NSGT, Ebrahimi and Barati (2017) examined the critical buckling loads of a FG curved beam.

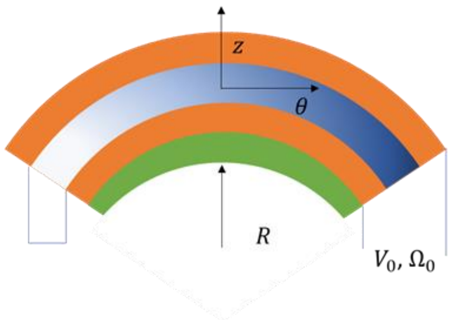


Fig. 1 Schematic of the three-layered curved beam on the two-parameter elastic foundation

In the presented investigation, on the basis of nonlocal strain gradient theory as well as first shear order beam theory, or Timoshenko, the buckling along with vibrational behavior of a three-layered nanobeam containing a functionally graded—FG—core as well as two face sheets made of piezo-magnetic materials is studied. The formulation together with the end conditions are acquired through energy method, or Hamilton's principle. Then, by incorporating Generalized differential quadrature method (GDQM) the governing equations are solved numerically. The presented formulation as well as solution method are validated by means of other studies. Lastly, the impact regarding various parameter which can influence the buckling as well as vibration of this system is explored in detail.

2. Problem formulation

In this part of this research, on the basis of Timoshenko beam theory as well as nonlocal strain gradient theory, the formulation associated with vibration as well as buckling of a curved functionally graded nanobeam coupled with two face sheets made of piezo-magnetoelectric material which is resting on a Winkler-Pasternak foundation is presented. Here, the schematic of the mentioned beam can be seen in Fig. 1.

Next, the displacement field based on Timoshenko beam theory for curved beams are as follow.

$$u_1(x,\theta,z,t) = \left(1+\frac{z}{R}\right)U(\theta,t) + z\alpha(\theta,t)$$
$$u_3(x,\theta,z,t) = W(\theta,t) \tag{1}$$

Based on these displacement fields, the strains can be written.

$$\varepsilon_{\theta\theta} = \frac{\partial W}{\partial\theta} - \frac{U}{R} + z\frac{\partial\alpha}{\partial\theta}$$
$$\varepsilon_{\theta z} = \frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha \tag{2}$$

Now, the relation between stress and strain (Liu *et al*. 2020a, Wang *et al*. 2020, Zhou *et al*. 2020, Dai *et al*. 2021a, Guo *et al*. 2021a, Shao *et al*. 2021, Wu and Habibi 2021) of the FG core is

$$\begin{aligned}\sigma^c{}_{\theta\theta} &= Q_{11}\varepsilon_{\theta\theta}\\ \sigma^c{}_{\theta z} &= Q_{55}\varepsilon_{\theta z}\end{aligned} \tag{3}$$

Now, the FGM material properties are:

$$\begin{aligned}E(z) &= E_m + (E_{cm} - E_m)\left(\frac{z}{h}+\frac{1}{2}\right)^n\\ \rho(z) &= \rho_m + (\rho_{cm} - \rho_m)\left(\frac{z}{h}+\frac{1}{2}\right)^n\\ \nu(z) &= \nu\end{aligned} \tag{4}$$

And the component in Eq. (3) can be written as follow

$$\begin{aligned}Q_{11} &= \frac{E(z)}{1-\nu^2}\\ Q_{55} &= \frac{E(z)}{2+2\nu}\end{aligned} \tag{5}$$

Also, these relations corresponded to the face sheet made of piezo-magneto material can be written as the following equation.

$$H_{ij}(t) = \frac{1}{2}e_i^T(t_j)D_{ij}e_i(t) + \frac{1}{2}\int_0^{t_j} e_i^T(t_j)D_{ij}e_i(t)dt \tag{6}$$

Additionally, the electrical displacement as well as magnetic induction is

$$\begin{aligned}D_z &= e_{31}\varepsilon_{\theta\theta} + k_{33}E_z\\ D_\theta &= e_{15}\varepsilon_{\theta z} + k_{11}E_\theta\\ B_z &= m_{31}\varepsilon_{\theta\theta} + t_{33}E_z\\ B_\theta &= m_{15}\varepsilon_{\theta z} + t_{11}E_\theta\end{aligned} \tag{7}$$

In which the electrical and magnetic field can be defined as follow.

$$\begin{aligned}E_z &= \frac{\partial\Phi}{\partial z}\\ E_\theta &= \frac{\partial\Phi}{\partial\theta}\\ H_z &= \frac{\partial\Psi}{\partial z}\\ H_\theta &= \frac{\partial\Psi}{\partial\theta}\end{aligned} \tag{8}$$

where the magnetic along with the electrical potential can be written based on the condition that these two should satisfies the Maxwell's equation.

$$\begin{aligned}(x,z,t) &= -Cos\left(\beta\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)\right)\psi(\theta,t) + \frac{2z}{h_f}\Omega_0\\ \Phi(x,z,t) &= -Cos\left(\beta\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)\right)\phi(\theta,t) + \frac{2z}{h_f}V_0\end{aligned} \tag{9}$$

where $\beta = \frac{\pi}{h_f}$ Now, the strain energy (Bai *et al*. 2021, Yang *et al*. 2022b, Zhang *et al*. 2022b, Zheng *et al*. 2022a, b) associated with a curved nanobeam including an FG core as well as two face sheets made of piezo-magneto material can be introduced as

$$\begin{aligned}U &= \frac{1}{2}\int\int\int_{\frac{-h_c}{2}}^{\frac{h_c}{2}}(\varepsilon_{\theta\theta}\sigma^c{}_{\theta\theta} + \varepsilon_{\theta z}\sigma^c{}_{\theta z})\,d\theta dydz\\ &+ \int\int\int_{\frac{h_c}{2}}^{\frac{h_c}{2+h_f}}\begin{pmatrix}\varepsilon_{\theta\theta}\sigma^f{}_{\theta\theta}\\ +\varepsilon_{\theta z}\sigma^f{}_{\theta z}\\ -D_zE_z - D_\theta E_\theta\\ -B_zH_z - B_\theta H_\theta\end{pmatrix}d\theta dydz\end{aligned} \tag{10}$$

In addition, the kinetic energy (Chen *et al*. 2021, Lu *et al*. 2021, Meng *et al*. 2022, Wang *et al*. 2022b, Zhao *et al*. 2022) of the system is (Ma *et al*. 2022, Zhao *et al*. 2022, Hou *et al*. 2021, Huang *et al*. 2021b, c, Jiao *et al*. 2021, Liu *et al*. 2021c, Moradi *et al*. 2021, Xu *et al*. 2021, Dong *et al*. 2022, Luo *et al*. 2022, Yang *et al*. 2022a, Yu *et al*. 2022):

$$\Pi = \frac{1}{2}\iiint\rho\left(\frac{\partial u_1(\theta,z,t)}{\partial t} + \frac{\partial u_2(\theta,z,t)}{\partial t}\right)^2 dzd\theta dy \tag{11}$$

Now, the variation of these energy can be written in the following equation

$$\begin{aligned}\delta U &= \frac{1}{2}\int\int\begin{pmatrix}N\left(\delta\frac{\partial W}{\partial\theta} - \frac{\delta U}{R}\right) + M\delta\frac{\partial\alpha}{\partial\theta}\\ +Q\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right)\end{pmatrix}d\theta dy\\ &\quad + 2\int\int\int_{\frac{h_c}{2}}^{\frac{h_c}{2+h_f}}\zeta\,d\theta dydz\end{aligned} \tag{12}$$

where

$$\zeta = \begin{pmatrix}D_z\frac{\pi}{h_p}Sin\left(\frac{\pi\left(z-\frac{h_c}{2-\frac{h_f}{2}}\right)}{h_p}\right)\delta\phi - D_\theta\cos\left(\frac{\pi\left(z-\frac{h_c}{2-\frac{h_f}{2}}\right)}{h_p}\right)\delta\frac{\partial\phi}{\partial\theta}\\ +B_z\frac{\pi}{h_p}Sin\left(\frac{\pi\left(z-\frac{h_c}{2-\frac{h_f}{2}}\right)}{h_p}\right)\delta\psi - B_\theta\cos\left(\frac{\pi\left(z-\frac{h_c}{2-\frac{h_f}{2}}\right)}{h_p}\right)\delta\frac{\partial\psi}{\partial\theta}\end{pmatrix} \tag{13}$$

Also

$$\Pi = \frac{1}{2}\iint\begin{pmatrix}I_0\frac{\partial U}{\partial t}\delta\frac{\partial U}{\partial t} + \begin{pmatrix}I_0 +\\ \frac{I_2}{R^2}\\ +\frac{2I_1}{R}\end{pmatrix}\frac{\partial W}{\partial t}\delta\frac{\partial W}{\partial t} +\\ \left(I_1 + \frac{I_2}{R}\right)\begin{pmatrix}\frac{\partial\alpha}{\partial t}\delta\frac{\partial W}{\partial t}\\ +\frac{\partial W}{\partial t}\delta\frac{\partial\alpha}{\partial t}\end{pmatrix} + I_2\frac{\partial\alpha}{\partial t}\delta\frac{\partial\alpha}{\partial t}\end{pmatrix}d\theta dy \tag{14}$$

In which

$$\begin{aligned}\{I_0, I_1, I_2\} &= \int_{-h_c/2}^{h_c/2}\rho_c(z)\{1,z,z^2\}dz\\ &\quad + 2\int_{h_c/2}^{h_c/2+h_p}\rho_p\{1,0,z^2\}dz\end{aligned} \tag{15}$$

Also, resultant forces can be defined as

$$\begin{aligned}N &= b\left(\int_{-h_c/2}^{h_c/2}\sigma^c_{\theta\theta}dz + \int_{h_c/2}^{h_c/2+h_f}\sigma^f_{\theta\theta}dz + \int_{-h_c/2-h_f}^{-h_c/2}\sigma^f_{\theta\theta}dz\right)\\ M &= b\left(\int_{-h_c/2}^{h_c/2}z\sigma^c_{\theta\theta}dz + \int_{h_c/2}^{h_c/2+h_f}z\sigma^f_{\theta\theta}dz + \int_{-h_c/2-h_f}^{-h_c/2}z\sigma^f_{\theta\theta}dz\right)\end{aligned} \tag{16}$$

$$Q = k_s b\left(\int_{-h_c/2}^{h_c/2}\sigma_{\theta z}^{c}dz + \int_{h_c/2}^{h_c/2+h_f}\sigma_{\theta z}^{f}dz + \int_{-h_c/2-h_f}^{-h_c/2}\sigma_{\theta z}^{f}dz\right)$$

The external work (Shariati *et al*. 2012, 2016a, b, 2019, 2020d, e, f, g, h, i, j, 2021a, b) which is done by the elastic two-prameter foundation is

$$W_{ext} = \left(N_m + N_p\right)\frac{\partial W}{\partial\theta}\delta\frac{\partial W}{\partial\theta} - K_w W\delta W + K_p\frac{\partial W}{\partial\theta}\delta\frac{\partial W}{\partial\theta} \quad (17)$$

Here, by using the energy method, or Hamilton's principle (Al-Furjan *et al*. 2020a, b, h, i, m, p, t, 2021a, c), as follow

$$\int_{t_0}^{t_1}\delta(U - \Pi + W_{ext})\,dt = 0 \quad (18)$$

And setting the coefficient of to zero the governing equation of a three-layered curved beam with two piezo-magnetoelectric face sheets can be achieved as follow.

$$\delta U: \frac{N}{R} + \frac{\partial Q}{\partial\theta} = I_0\frac{\partial^2 U}{\partial t^2} \quad (19)$$

$$\delta W: -\frac{Q}{R} + \frac{\partial N}{\partial\theta} - \left(N_B + N_p + N_m\right)\frac{\partial^2 w}{\partial\theta^2} + K_p\frac{\partial^2 w}{\partial\theta^2} - K_w W = \left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2\alpha}{\partial t^2} + \left(I_0 + \frac{2I_1}{R} + \frac{I_2}{R^2}\right)\frac{\partial^2 W}{\partial t^2} \quad (20)$$

$$\delta\alpha: \frac{\partial M}{\partial\theta} + Q = I_2\frac{\partial^2\alpha}{\partial t^2} + \left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2 W}{\partial t^2} \quad (21)$$

$$\partial\phi: \int\int\int_{\frac{h_c}{2}}^{\frac{h_c}{2+h_f}}\left(D_z\frac{\pi}{h_f}Sin\left(\frac{\pi\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)}{h_f}\right) - \frac{\partial D_\theta}{\partial\theta}cos\left(\frac{\pi\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)}{h_f}\right)\right)d\theta dy dz = 0 \quad (22)$$

$$\partial\psi: \int\int\int_{\frac{h_c}{2}}^{\frac{h_c}{2+h_f}}\left(B_z\frac{\pi}{h_f}Sin\left(\frac{\pi\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)}{h_p}\right) - \frac{\partial B_\theta}{\partial\theta}cos\left(\frac{\pi\left(z - \frac{h_c}{2-\frac{h_f}{2}}\right)}{h_f}\right)\right)d\theta dy dz = 0 \quad (23)$$

Also, the boundary conditions are

For clamped end:
$U = W = \alpha = \phi = \psi = 0$
For simply supported end:
$U = W = \phi = \psi = 0, M = 0$ (24)

Now, in order to capture the effect of size for the nanobeam, the nonlocal strain-gradient theory is incorporated. The stress and strain, based on this theory is

$$(1 - (e_0 a)^2\nabla^2)\sigma_{ij}^{f} = (1 - (l_m)^2\nabla^2)\left(c_{ijkl}\varepsilon_{kl}\right) - e_{mij}E_m - m_{mij}H_m \quad (25)$$

$$(1 - (e_0 a)^2\nabla^2)D_i = (1 - (l_m)^2\nabla^2)(e_{ikl}\varepsilon_{kl}) - k_{mij}E_m - t_{mij}H_m$$

By using the above equations, the resultant forces in Eq. (16) can be written in the frame work of nonlocal strain gradient elasticity as follow

$$\begin{aligned}
&(1 - (e_0 a)^2\nabla^2)N = (1 - (l_m)^2\nabla^2)\left(A\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + B\frac{\partial\alpha}{\partial\theta}\right)(1 - (e_0 a)^2\nabla^2)M \\
&= (1 - (l_m)^2\nabla^2)\left(B\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + C\frac{\partial\alpha}{\partial\theta}\right) \\
&+F_{31}\phi + J_{31}\psi(1 - (e_0 a)^2\nabla^2)Q \\
&= k_s\left((1 - (l_m)^2\nabla^2)k_s D\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right)\right) \\
&-k_s E_{15}\frac{\partial\phi}{\partial x} - k_s M_{15}\frac{\partial\psi}{\partial x} \\
&\int_{\frac{h}{2}}^{\frac{h}{2}+h_p}\left[(1 - (e_0 a)^2\nabla^2)D_\theta\cos\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)\right] \\
&= E_{15}(1 - (l_m)^2\nabla^2)\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right) + X_{11}\frac{\partial\phi}{\partial\theta} \\
&\int_{\frac{h}{2}}^{\frac{h}{2}+h_p}\left[(1 - (e_0 a)^2\nabla^2)B_\theta\cos\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)\right] \\
&= M_{15}(1 - (l_m)^2\nabla^2)\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right) + T_{11}\frac{\partial\psi}{\partial\theta} \\
&\int_{\frac{h}{2}}^{\frac{h}{2}+h_p}\left[(1 - (e_0 a)^2\nabla^2)D_z\beta\,Sin\,\beta\left(z - \frac{h_c}{2} - \frac{h_f}{2}\right)\right] \\
&= (1 - (l_m)^2\nabla^2)\left(F_{31}\frac{\partial\alpha}{\partial\theta}\right) \\
&-X_{33}\phi\int_{\frac{h}{2}}^{\frac{h}{2}+h_p}\left[(1 - (e_0 a)^2\nabla^2)B_z\beta\,Sin\,\beta\left(z - \frac{h_c}{2} - \frac{h_f}{2}\right)\right] \\
&= (1 - (l_m)^2\nabla^2)\left(J_{31}\frac{\partial\alpha}{\partial\theta}\right) - T_{33}\psi
\end{aligned} \quad (26)$$

In which the constants used can be defined as follow

$$\begin{aligned}
&\{A, B, C\} = \int_{\frac{-h}{2}}^{\frac{h}{2}}Q_{11}\{1, z, z^2\}dz + 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}C_{11}\{1, 0, z^2\}dz \\
&\{D\} = \int_{\frac{-h}{2}}^{\frac{h}{2}}Q_{55}dz + 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}C_{55}z^2dz \\
&F_{31} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}e_{31}\beta z\,Sin\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)zdz, \\
&E_{15} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}e_{15}\,Cos\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)dz, \\
&X_{11} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}k_{11}\,Cos^2\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)dz, X_{33} \\
&= 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}k_{33}\beta^2\,Sin^2\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)dz \\
&J_{31} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}}m_{31}\beta z\,Sin\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)zdz,
\end{aligned} \quad (27)$$

$$M_{15} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}} m_{15}\, Cos\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)dz,$$
$$T_{11} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}} t_{11}\, Cos^2\left(\beta\left(\begin{matrix} z - \\ \frac{h}{2} - \frac{h_f}{2}\end{matrix}\right)\right)dz,$$
$$T_{33} = 2\int_{\frac{h}{2}}^{\frac{h}{2+h_p}} t_{33}\beta^2\, Sin^2\left(\beta\left(z - \frac{h}{2} - \frac{h_f}{2}\right)\right)dz$$

where *A*, *B*, *C*, and *D* after the completion of integration can be written as follow.

$$\begin{aligned} A &= \frac{bh(E_{cm} + E_m n)}{1+n} + 2h_f C_{11} \\ B &= \frac{b(E_{cm} - E_m)h^2 n}{4 + 6n + 2n^2} \\ C &= \frac{bh^3\begin{pmatrix} 3E_{cm}(2+n+n^2) \\ +E_m n\big(8 + n(3+n)\big)\end{pmatrix}}{12(1+n)(2+n)(3+n)} \\ &+ \frac{1}{6}bh_f\begin{pmatrix} 3h^2 + \\ 6hh_f + 4{h_f}^2\end{pmatrix}C_{11} \\ D &= \frac{bh(E_{cm} + E_m n)}{1+n} + 2h_f C_{55} \end{aligned} \tag{28}$$

The governing equations can be rewritten in the framework of nonlocal strain gradient theory.

$$\begin{aligned} &\delta U: (1-(l_m)^2\nabla^2)\begin{pmatrix} \frac{1}{R}\left(A\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + B\frac{\partial\alpha}{\partial\theta}\right) + \\ \frac{\partial}{\partial\theta}\begin{pmatrix}\left(k_s D\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right)\right) \\ -k_s E_{15}\frac{\partial\phi}{\partial x} - k_s M_{15}\frac{\partial\psi}{\partial x}\end{pmatrix}\end{pmatrix} \\ &= (1-(e_0 a)^2\nabla^2)I_0\frac{\partial^2 U}{\partial t^2} \end{aligned} \tag{29}$$

$$\begin{aligned} &\delta W: \\ &(1-(l_m)^2\nabla^2)\begin{pmatrix} -\frac{1}{R}\begin{pmatrix}\left(k_s D\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right)\right) \\ -k_s E_{15}\frac{\partial\phi}{\partial x} \\ -k_s M_{15}\frac{\partial\psi}{\partial x}\end{pmatrix} \\ +\frac{\partial}{\partial\theta}\left(A\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + B\frac{\partial\alpha}{\partial\theta}\right) \\ -(N_B + N_p + N_m)\frac{\partial^2 w}{\partial\theta^2} \\ +K_p\frac{\partial^2 w}{\partial\theta^2} - K_w W\end{pmatrix} \\ &= (1-(e_0 a)^2\nabla^2)\left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2\alpha}{\partial t^2} + \left(I_0 + \frac{2I_1}{R} + \frac{I_2}{R^2}\right)\frac{\partial^2 W}{\partial t^2} \end{aligned} \tag{30}$$

$$\begin{aligned} &\delta\alpha: (1-(l_m)^2\nabla^2)\begin{pmatrix} \frac{\partial}{\partial\theta}\left(B\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + C\frac{\partial\alpha}{\partial\theta}\right) \\ +F_{31}\phi + J_{31}\psi \\ +\begin{pmatrix}\left(k_s D\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right)\right) \\ -k_s E_{15}\frac{\partial\phi}{\partial x} - k_s M_{15}\frac{\partial\psi}{\partial x}\end{pmatrix}\end{pmatrix} \\ &= (1-(e_0 a)^2\nabla^2)\left(I_2\frac{\partial^2\alpha}{\partial t^2} + \left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2 W}{\partial t^2}\right) \end{aligned} \tag{31}$$

$$(1-(l_m)^2\nabla^2)\left(F_{31}\frac{\partial\alpha}{\partial\theta}\right) - X_{33}\phi - \tag{32}$$

$$\frac{\partial}{\partial\theta}\left(E_{15}(1-(l_m)^2\nabla^2)\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right) + X_{11}\frac{\partial\phi}{\partial\theta}\right) = 0$$

$$\begin{aligned} &(1-(l_m)^2\nabla^2)\left(F_{31}\frac{\partial\alpha}{\partial\theta}\right) - X_{33}\phi \\ &-\frac{\partial}{\partial\theta}\left(M_{15}(1-(l_m)^2\nabla^2)\left(\frac{\partial U}{\partial\theta} + \frac{W}{R} - \alpha\right) + T_{11}\frac{\partial\psi}{\partial\theta}\right) = 0 \end{aligned} \tag{33}$$

Additionally, the nonlocal type of resultant forces can be found in the following equation

$$\begin{aligned} Q &= \frac{R}{(e_0 a)^2 + R^2} \\ &\times\begin{pmatrix} -(e_0 a)^2\begin{pmatrix} +\begin{pmatrix} N_B + N_p \\ +N_m\end{pmatrix}\frac{\partial^2 w}{\partial\theta^2} \\ -K_p\frac{\partial^2 w}{\partial\theta^2} + K_w W \\ +\left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2\alpha}{\partial t^2} \\ +\begin{pmatrix} I_0 + \frac{2I_1}{R} \\ +\frac{I_2}{R^2}\end{pmatrix}\frac{\partial^2 W}{\partial t^2}\end{pmatrix} R\begin{pmatrix} k_s\begin{pmatrix}(1-(l_m)^2\nabla^2) \\ k_s D\begin{pmatrix}\frac{\partial U}{\partial\theta} \\ +\frac{W}{R} \\ -\alpha\end{pmatrix}\end{pmatrix} \\ -k_s E_{15}\frac{\partial\phi}{\partial x} - k_s M_{15}\frac{\partial\psi}{\partial x}\end{pmatrix} \\ +(e_0 a)^2 R\left(I_0\frac{\partial^2 U}{\partial t^2}\right)\end{pmatrix} \end{aligned} \tag{34}$$

$$\begin{aligned} N &= \\ &\frac{R}{ea + R^2}\begin{pmatrix} (e_0 a)^2 R\left(I_0\frac{\partial^2 U}{\partial t^2}\right) + R(1-(l_m)^2\nabla^2) \\ \left(A\left(\frac{\partial W}{\partial\theta} - \frac{U}{R}\right) + B\frac{\partial\alpha}{\partial\theta}\right) \\ +R(e_0 a)^2\begin{pmatrix} +\begin{pmatrix} N_B + N_p \\ +N_m\end{pmatrix}\frac{\partial^2 w}{\partial\theta^2} \\ -K_p\frac{\partial^2 w}{\partial\theta^2} + K_w W \\ +\left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2\alpha}{\partial t^2} \\ +\begin{pmatrix} I_0 + \frac{2I_1}{R} \\ +\frac{I_2}{R^2}\end{pmatrix}\frac{\partial^2 W}{\partial t^2}\end{pmatrix}\end{pmatrix} \end{aligned} \tag{35}$$

$$\begin{aligned} M &= \\ &\frac{1}{R}\begin{pmatrix} (e_0 a)^2 R\left(I_0\frac{\partial^2 U}{\partial t^2}\right) \\ +R\begin{pmatrix} (1-(l_m)^2\nabla^2)\begin{pmatrix} B\begin{pmatrix}\frac{\partial W}{\partial\theta} \\ -\frac{U}{R}\end{pmatrix} \\ +C\frac{\partial\alpha}{\partial\theta}\end{pmatrix} \\ +F_{31}\phi + J_{31}\psi\end{pmatrix} \\ -(e_0 a)^2 N - (e_0 a)^2 R I_2\frac{\partial^2\alpha}{\partial t^2} \\ +\left(I_1 + \frac{I_2}{R}\right)\frac{\partial^2 W}{\partial t^2}\end{pmatrix} \end{aligned} \tag{36}$$

3. Solution procedure

Due to the complexity of the current study, a numerical solution procedure, generalized differential quadrature method, with very high precision is utilized (Li *et al*. 2020a, b, Wang *et al*. 2022a). As the highest order of the formulation is four, the fourth order kind of this solution method is employed. Based on this theory the s-th order derivative of a function can be written through the following equation (Al-Furjan *et al*. 2020g, f, j, k, l, n, r, u, v).

$$\frac{\partial^s \Psi(\theta_i)}{\partial x^s} = \sum_{j=1}^{ns} y_{j0}^{(s)}(\theta_i)\Psi_j + y_{11}^{(s)}(\theta_i)\Psi_1^{(1)} + y_{ns1}^{(s)}(\theta_i)\Psi_{ns}^{(1)} = \sum_{j=1}^{ns+2} \gamma_{ij}^{(s)} V_j \; (i = 1,2,\ldots,ns) \tag{37}$$

By using the above equations, the resultant forces in Eq. (16) can be written in the frame work of nonlocal strain gradient elasticity as follow

$$\gamma_{ijl}^{(r)} = y_{jl}^{(r)}(\theta_i) = \frac{d^r y_{jl}(\theta_i)}{dx^r} \tag{38}$$

$$y_{jl}^{(r)}(\theta_i) = \begin{cases} 1 \text{if} i = j \quad l = r \\ 0 \text{otherwise} \end{cases} \tag{39}$$

$$\begin{gathered} y_{pi}(\theta) = \left(a_{pi}\theta^2 + b_{pi}\theta + c_{pi}\right) l_p(\theta) \\ (\text{p=1,nsand } i = 0,1) \\ y_{j0}(\theta) = \frac{(\theta - \theta_1)(\theta - \theta_{ns})}{(\theta_j - \theta_1)(\theta_j - \theta_{ns})} l_j(\theta) \\ (j = 2,3,\ldots,ns-1) \end{gathered} \tag{40}$$

Additionally, the constant component in Eq. (40) are

$$\begin{gathered} \begin{cases} a_{10} = \dfrac{-1}{(\theta_1 - \theta_{ns})^2} + \dfrac{-l_1^{(1)}(\theta_1)}{(\theta_1 - \theta_{ns})} \\ b_{10} = \dfrac{1}{(\theta_1 - \theta_{ns})} - a_{10}(\theta_1 + \theta_{ns}) \\ c_{10} = 1 - a_{10}\theta_1^2 - b_{10}\theta_1 \end{cases} \\ \begin{cases} a_{11} = \dfrac{1}{(\theta_1 - \theta_{ns})} \\ b_{11} = \dfrac{-(\theta_1 + \theta_{ns})}{(\theta_1 - \theta_{ns})} \\ c_{11} = \dfrac{\theta_1 \theta_{ns}}{(\theta_1 - \theta_{ns})} \end{cases} \\ \begin{cases} a_{ns0} = \dfrac{-1}{(\theta_1 - \theta_{ns})^2} + \dfrac{-l_{ns}^{(1)}(\theta_1)}{(\theta_1 - \theta_{ns})} \\ b_{ns0} = \dfrac{-1}{(\theta_1 - \theta_{ns})} - a_{ns0}(\theta_1 + \theta_{ns}) \\ c_{ns0} = 1 - a_{ns0}\theta_{ns}^2 - b_{ns0}\theta_{ns} \end{cases} \\ \begin{cases} a_{ns1} = \dfrac{-1}{(\theta_1 - \theta_{ns})} \\ b_{ns1} = \dfrac{(\theta_1 + \theta_{ns})}{(\theta_1 - \theta_{ns})} \\ c_{ns1} = \dfrac{-\theta_1 \theta_{ns}}{(\theta_1 - \theta_{ns})} \end{cases} \end{gathered} \tag{41}$$

Additionally, Lagrange interpolation which is employed in Eq. (40) are as follow.

$$l_j^{(1)}(\theta_i) = \begin{cases} \dfrac{R^{(1)}(\theta_i)}{(\theta_i - \theta_j) R^{(1)}(\theta_j)} \; (\text{for } i,j = 1,2,\ldots,ns, i \neq j) \\ -\displaystyle\sum_{j=1, i\neq j}^{ns} l_j^{(1)}(\theta_i) \; (\text{for } i,j = 1,2,\ldots,ns) \end{cases} \tag{42}$$

$$R^{(1)}(\theta_i) = \prod_{m=1, m\neq i}^{ns} (\theta_i - \theta_m) \tag{43}$$

The Lagrange interpolation corresponded to higher order derivatives are:

$$l_j^{(r)}(\theta_i) = \begin{cases} r\left(l_j^{(r-1)}(\theta_i) l_j^{(1)}(\theta_i) - \dfrac{l_j^{(r-1)}(\theta_i)}{(\theta_i - \theta_j)} \right) \\ (\text{for } i,j = 1,2,\ldots,ns) \\ -\displaystyle\sum_{j=1, i\neq j}^{ns} l_j^{(r)}(\theta_i) \; (\text{for } i,j = 1,2,\ldots,ns) \end{cases} \tag{44}$$

Chebyshev Gauss Lobatto is utilized to obtain the sample points as:

$$\theta_i = \frac{L}{2}\left(1 - cos\left(\frac{(i-1)}{(ns-1)}\pi\right)\right) \quad (i = 1,2,\ldots,ns) \tag{45}$$

Now, by incorporation of separation of variables, the independent variables can be defined as follow.

$$\begin{cases} W(\theta,t) = w(\theta)e^{i\lambda t} \\ U(\theta,t) = u(\theta)e^{i\lambda t} \\ a(\theta,t) = \bar{a}(\theta)e^{i\lambda t} \\ \phi(x,,y,t) = \bar{\phi}(\theta)e^{i\lambda t} \\ \psi(x,,y,t) = \bar{\psi}(\theta)e^{i\lambda t} \end{cases} \tag{46}$$

The discretized form of these variables can be rewritten in the below equation.

$$\{V\}^T = \begin{Bmatrix} \{w_1,,\ldots,w_{ns+2}\},\{u_1,,\ldots,u_{ns+2}\},\{\bar{\alpha}_1,,\ldots,\bar{\alpha}_{ns+2}\}, \\ \{\bar{\phi}_1,,\ldots,\bar{\phi}_{ns+2}\},\{\bar{\psi}_1,,\ldots,\bar{\psi}_{ns+2}\} \end{Bmatrix}^T \tag{47}$$

Now, with the help of GDQM, the governing equations related to the buckling analysis as well as vibration analysis of the system, Eqs. (29)-(33) can be rewritten in a matrix form respectively. It should be mentioned that, the formulation related to the buckling can be attained through setting $\lambda = 0$ in the equations of motions of the system.

$$\begin{bmatrix} [K_{bb}]_{12\times12} & [K_{bd}]_{12\times(3ns-6)} \\ [K_{db}]_{(3ns-6)\times12} & [K_{dd}]_{(3ns-6)\times(3ns-6)} \end{bmatrix} \cdot \begin{Bmatrix} \{V_b\} \\ \{V_d\} \end{Bmatrix} - N_B \begin{bmatrix} [M_{bb}]_{12\times12} & [M_{bd}]_{12\times(3ns-6)} \\ [M_{db}]_{(3ns-6)\times12} & [M_{dd}]_{(3ns-6)\times(3ns-6)} \end{bmatrix} \cdot \begin{Bmatrix} \{V_b\} \\ \{V_d\} \end{Bmatrix} = 0. \tag{48}$$

$$\begin{bmatrix} [K_{bb}]_{12\times12} & [K_{bd}]_{12\times(3ns-6)} \\ [K_{db}]_{(3ns-6)\times12} & [K_{dd}]_{(3ns-6)\times(3ns-6)} \end{bmatrix} \cdot \begin{Bmatrix} \{V_b\} \\ \{V_d\} \end{Bmatrix} - \lambda^2 \begin{bmatrix} [M_{bb}]_{12\times12} & [M_{bd}]_{12\times(3ns-6)} \\ [M_{db}]_{(3ns-6)\times12} & [M_{dd}]_{(3ns-6)\times(3ns-6)} \end{bmatrix} \begin{Bmatrix} \{V_b\} \\ \{V_d\} \end{Bmatrix} = 0. \tag{49}$$

Here, the eigenvalues can be acquired by solving the above matrix equation. And the eigen values of first matrix represents the buckling loads and the second matrix represents the vibrational frequencies of the system.

Table 1 The geometry as well as material properties of the beams in the current study

Piezo-magneto		Core	
$C_{11}(GPa)$	226	$E_m(GPa)$	70
$C_{55}(GPa)$	44.2	$E_{cm}(GPa)$	200
$\kappa_{11}(C(mV)^{-1})$	5.64×10^{-9}	$\nu_{cm}=\nu_m$	0.3
$e_{31}(Cm^{-2})$	2.2	$\rho_m\left(\frac{Kg}{m^3}\right)$	2.7
$e_{15}(Cm^{-2})$	5.8	$\rho_{cm}\left(\frac{Kg}{m^3}\right)$	5.7
$\kappa_{11}(C(mV)^{-1})$	6.35×10^{-9}		
$m_{31}\left(\frac{N}{Am}\right)$	290.1		
$m_{15}\left(\frac{N}{Am}\right)$	275		
$t_{11}\left(\frac{Ns^2}{C^2}\right)$	-297×10^{-6}		
$t_{33}\left(\frac{Ns^2}{C^2}\right)$	83.5×10^{-6}		
$\rho\left(\frac{Kg}{m^3}\right)$	5.55		

Table 2 The first dimensionless frequencies of a curved nanobeam for various slenderness and nonlocality

	$\frac{L}{h}$								
	20			50			100		
$e_0a(10^{-9})$	0	1	2	0	1	2	0	1	2
Present	9.8437	9.3809	8.9946	9.8653	9.4117	9.0155	9.8686	9.4148	9.0184
(Rahmani and Pedram 2014)	9.8296	9.3777	8.9829	9.8631	9.4097	9.0136	9.8680	9.4143	9.0180

Table 3 The first-three nondimensional vibration frequency of a curved beam for open angel of $\frac{\pi}{3}$ and various value of nonlocality along with $\frac{L}{h}$

		$e_0a(10^{-18})$					
		0		1		2	
$\frac{L}{h}$	Mode	Present	(Hosseini and Rahmani 2016)	Present	(Hosseini and Rahmani 2016)	Present	(Hosseini and Rahmani 2016)
	1	8.19913	8.19913	7.82946	7.8222	7.48566	7.4929
10	2	35.7451	35.7451	30.3376	30.2666	26.6545	26.7204
	3	77.3997	77.3993	56.5245	56.3256	46.77	46.45
	1	8.29123	8.29123	7.90868	7.91007	7.57905	7.57706
20	2	37.2875	37.2875	31.5688	31.5725	27.8965	27.8733
	3	84.3181	84.318	61.3393	61.3605	50.6444	50.6022
	1	8.3177	8.3177	7.93581	7.93532	7.60362	7.60125
50	2	37.7658	37.7658	31.9821	31.9776	28.2358	28.2309
	3	86.7081	86.7084	63.1216	63.1	52.0704	52.0367

4. Numerical results

Here, by employing GDQM as the numerical solution procedure, the results for the vibration as well as buckling behavior of a three-layered curved beam is investigated. These results are presented in two sections, first, the accuracy as well as reliability of the formulation together with the numerical solution is examined by comparing the results obtained through resent study with those from other published papers.

Also, it is vital to be stated that the material properties which is used in the current report can be seen in Table.1.

Now, to begging with, by eliminating the magneto-electro-elastic layers, the first nondimensional vibrational frequency, $\bar{\omega}=\omega L^2\sqrt{\frac{I_0}{C}}$, of the system by setting $l_m=0, n=0$, for a beam with a very large radius, $\frac{L}{R}=\frac{\pi}{360}$, is obtained and compared with those of Ref. (Rahmani and Pedram 2014). The results for this table are obtained for simply supported beams.

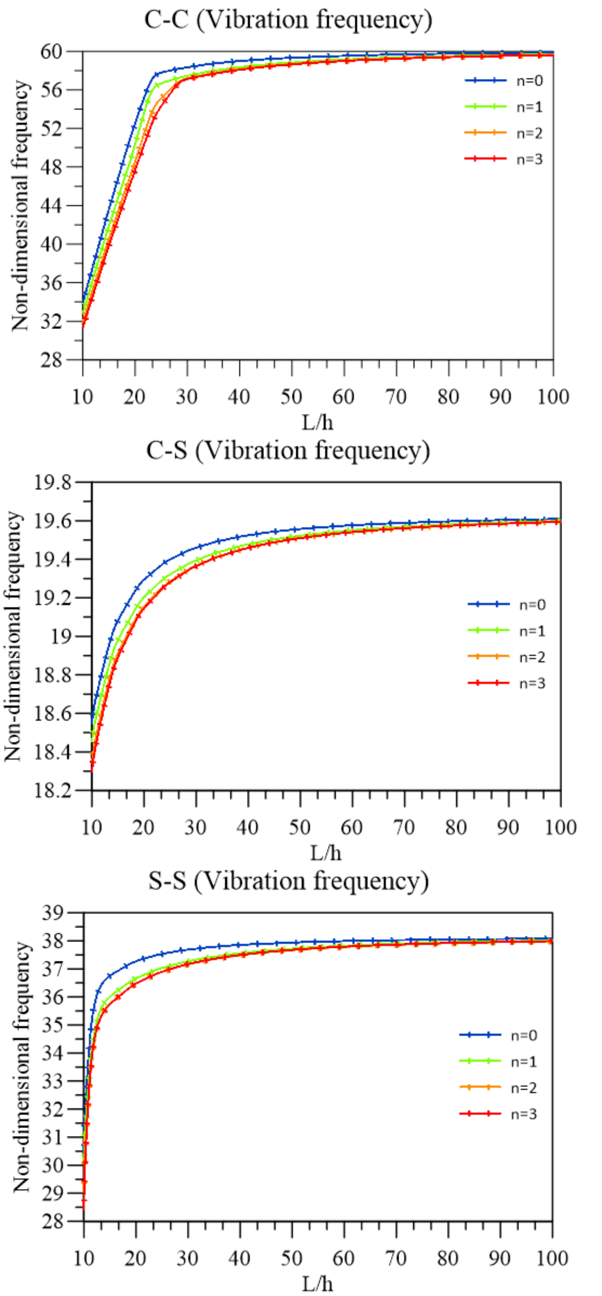


Fig. 2 Variation of nondimensional vibration frequency against $\frac{L}{h_c}$ for various index number of FG core and different boundary conditions

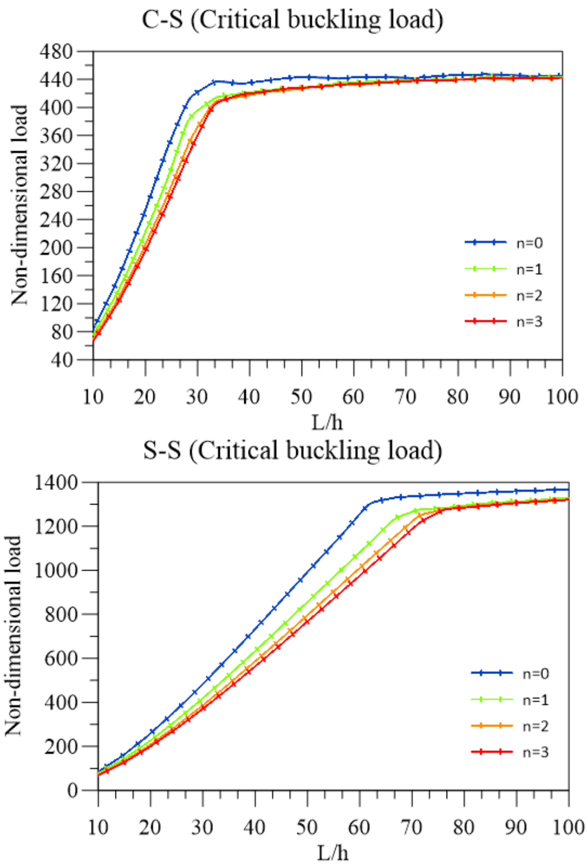
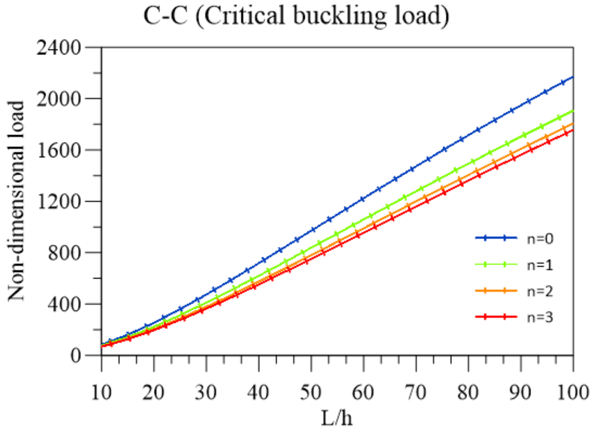
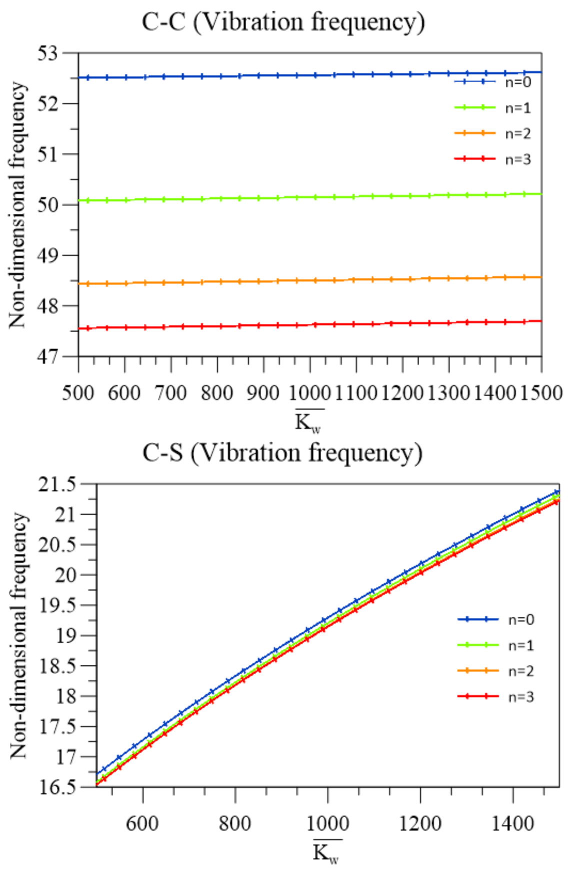


Fig. 3 Variation of nondimensional buckling load against $\frac{L}{h_c}$ for various index number of FG core and different boundary conditions





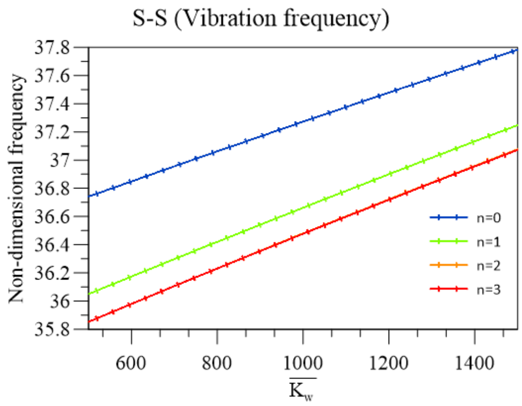


Fig. 4 Variation of nondimensional vibration frequency against $\bar{K}_w$ for various index number of FG core and different boundary conditions

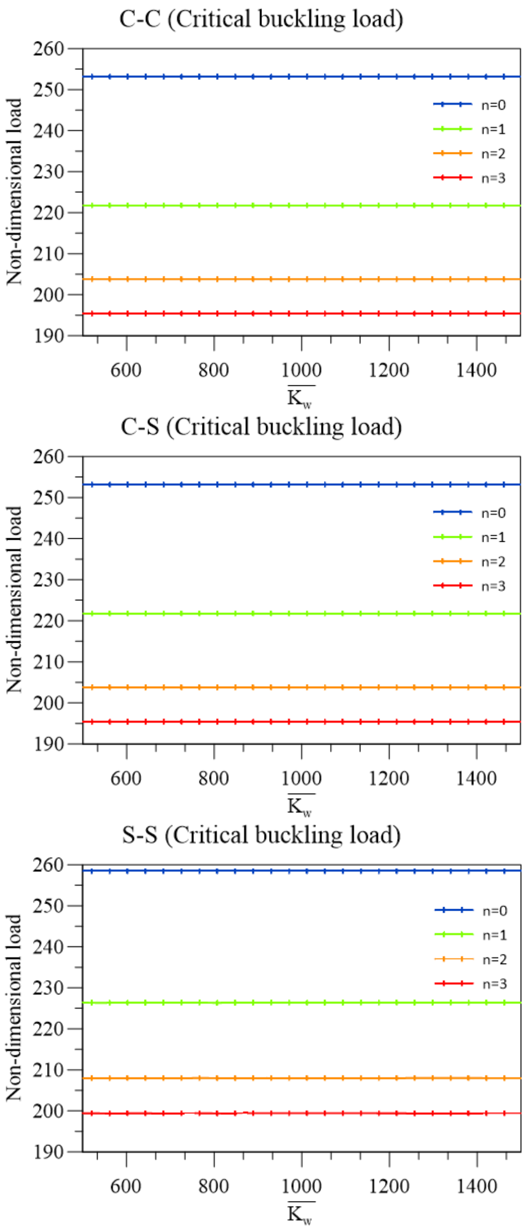


Fig. 5 Variation of nondimensional buckling load against $\bar{K}_w$ for various index number of FG core and different boundary conditions

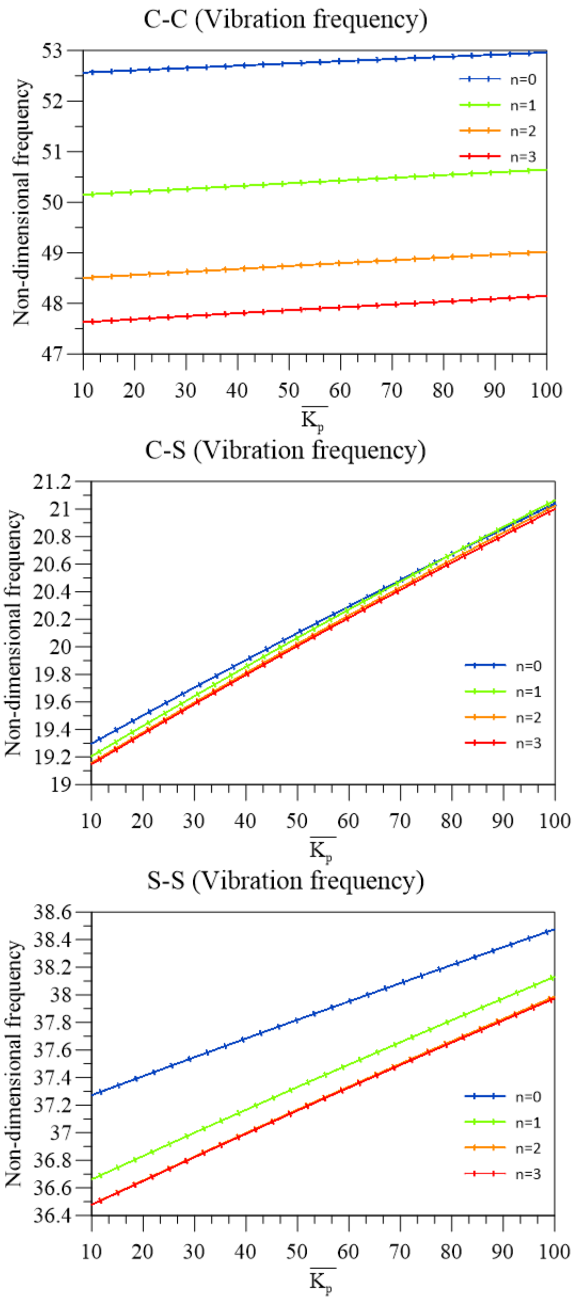
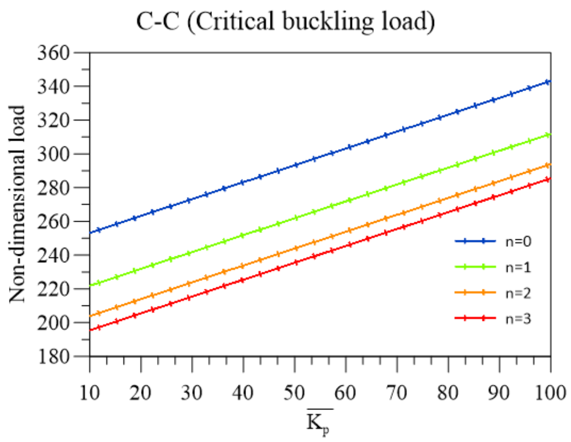


Fig. 6 Variation of nondimensional vibration frequency against $\bar{K}_p$ for various index number of FG core and different boundary conditions



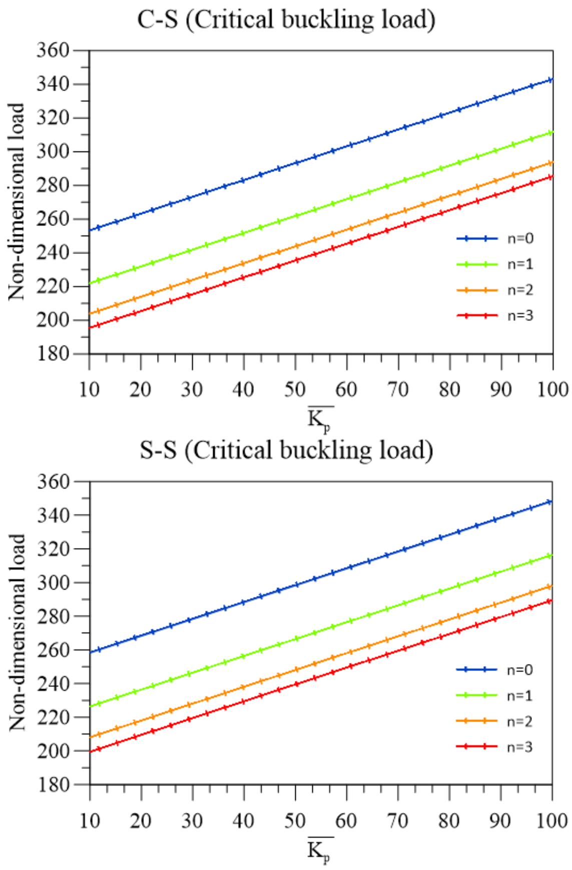
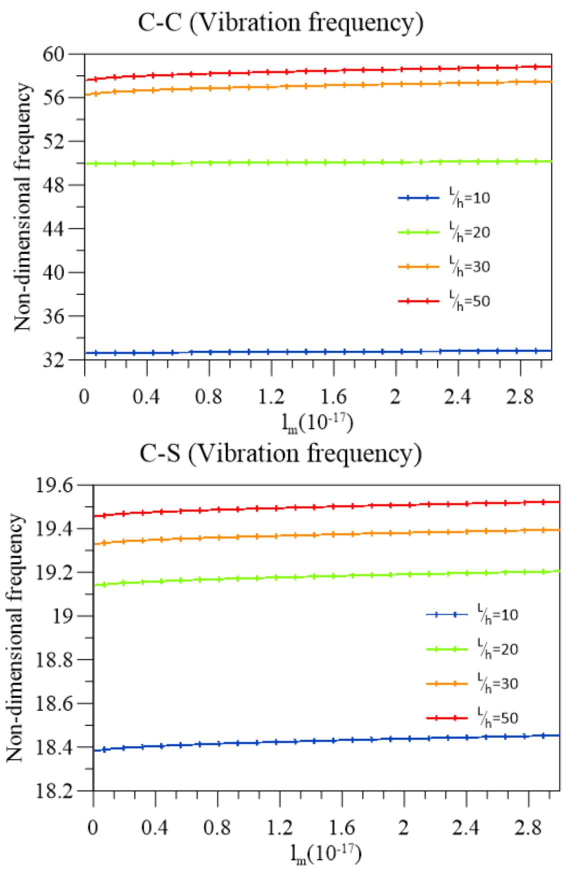


Fig. 7 Variation of nondimensional buckling load against $\bar{K}_p$ for various index number of FG core and different boundary conditions



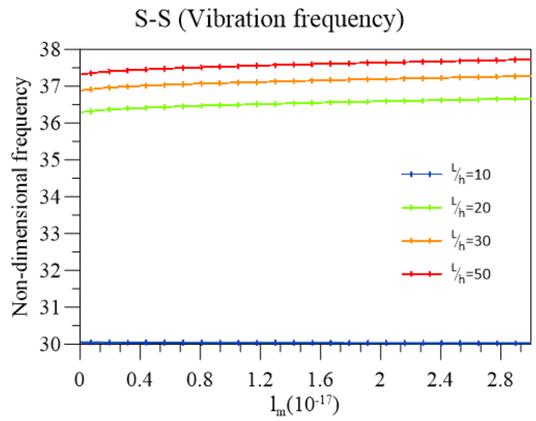


Fig. 8 Variation of nondimensional vibration frequency against l_m for various $\frac{L}{h_c}$ and different boundary conditions

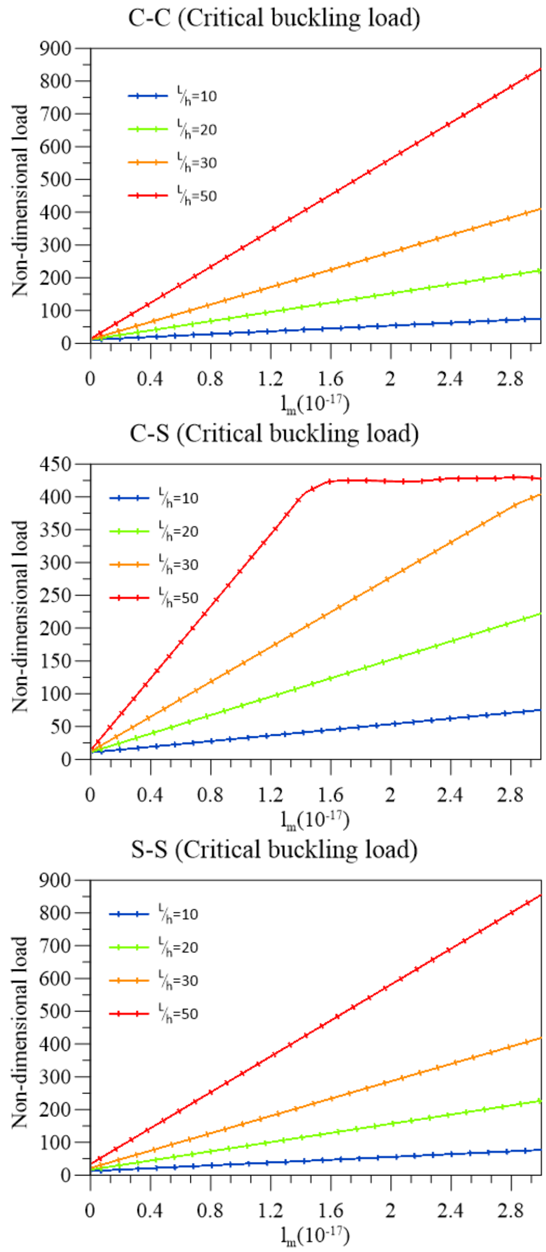


Fig. 9 Variation of nondimensional buckling load against l_m for various $\frac{L}{h_c}$ and different boundary conditions

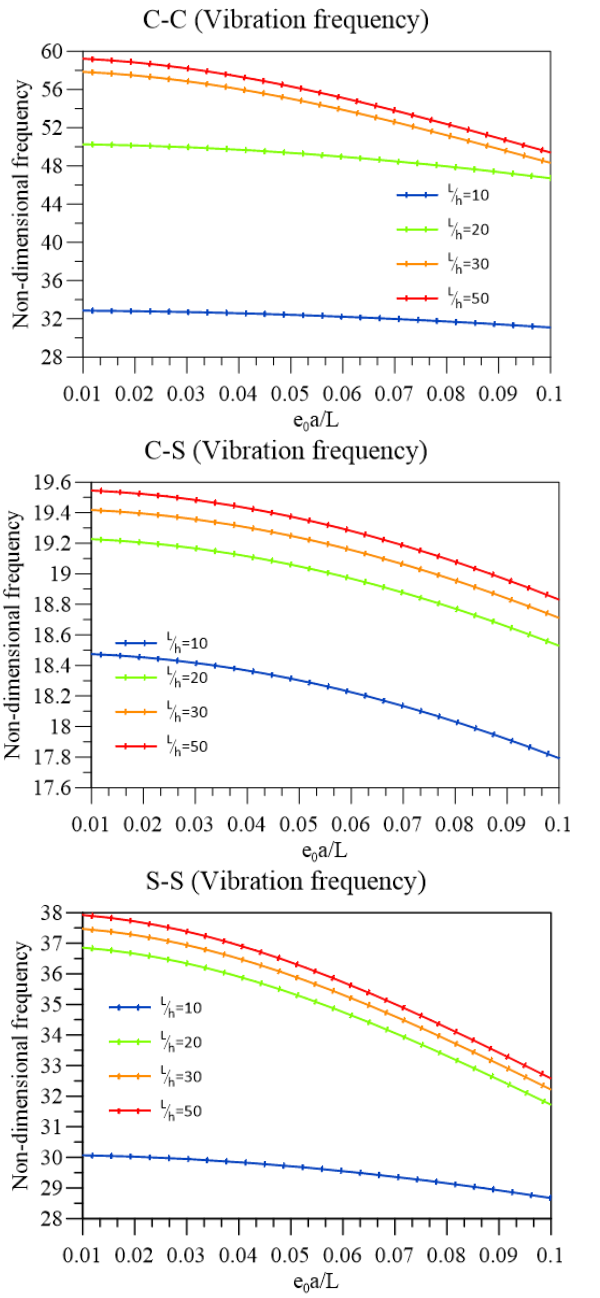


Fig. 10 Variation of nondimensional vibration frequency against $\frac{e_0 a}{L}$ for various $\frac{L}{h_c}$ and different boundary conditions

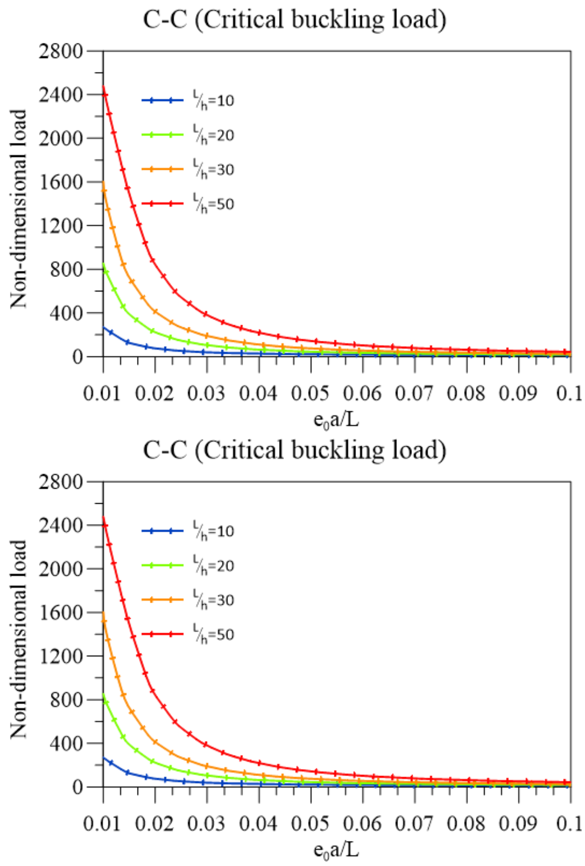
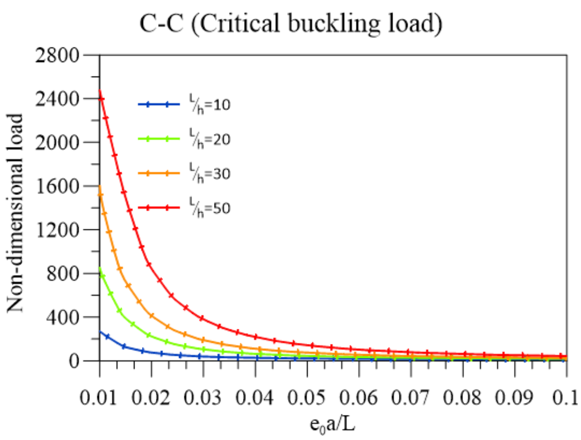
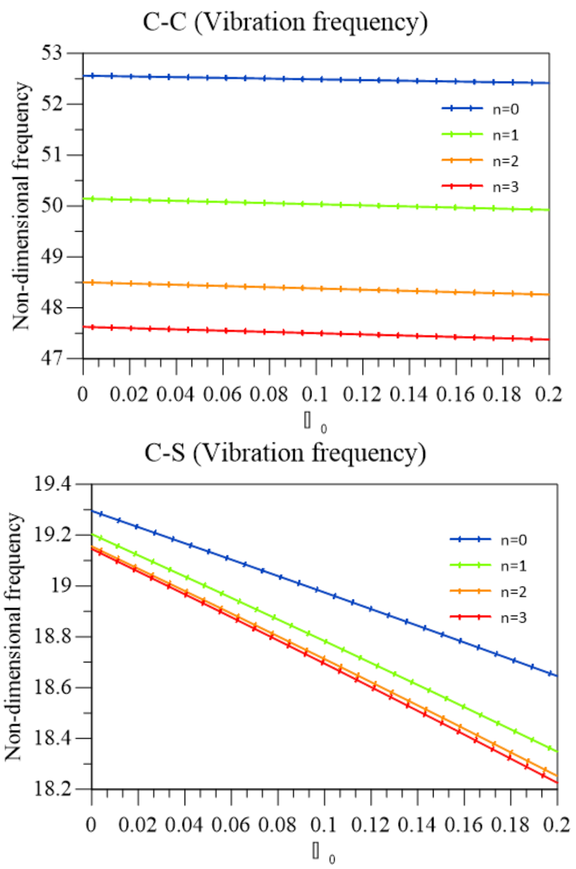


Fig. 11 Variation of nondimensional buckling load against $\frac{e_0 a}{L}$ for various $\frac{L}{h_c}$ and different boundary conditions





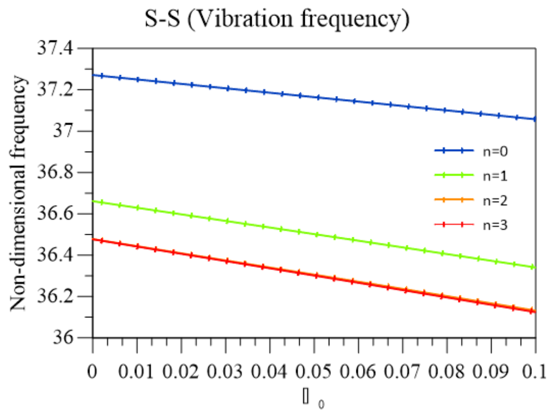


Fig. 12 Variation of nondimensional vibration frequency against Ω_0 for various index number of FG core and different boundary conditions

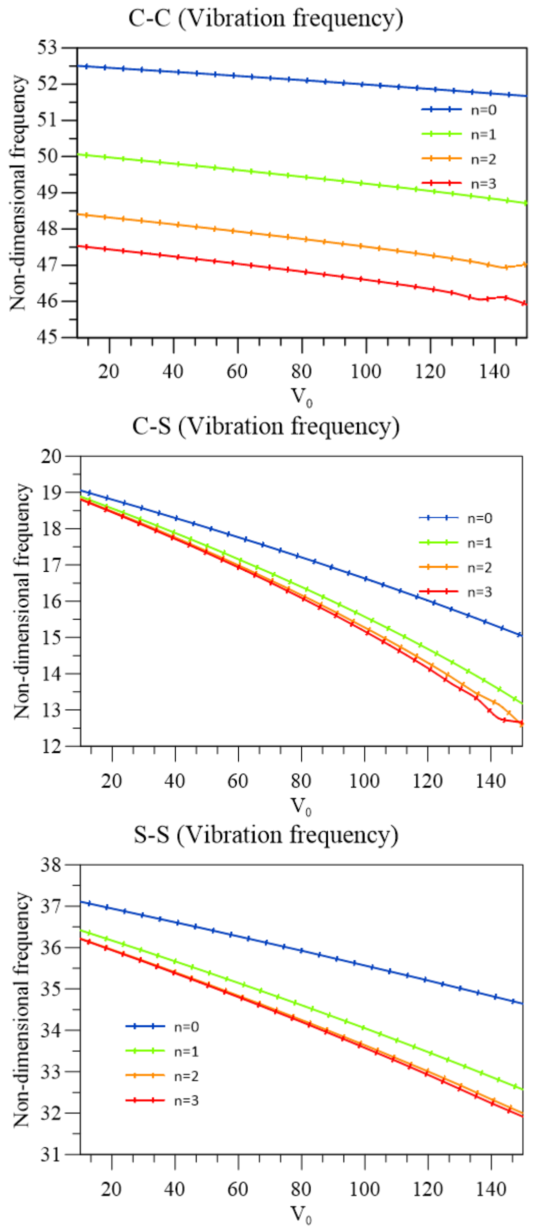


Fig. 13 Variation of nondimensional vibration frequency against V_0 for various index number of FG core and different boundary conditions

Additionally, the first-three vibrational frequencies for a curved beam modeled by Timoshenko beam theory, by omitting the effect of strain gradient, are extracted and presented next to the results of Ref. (Hosseini and Rahmani 2016) in Table 3.

As can we seen from Tables 2 and 3, the little difference between the results of present study and the results which are extracted from other papers indicate the accuracy as well as reliability of the presented formulation along with the solution procedure in examining the vibration and buckling behavior of a three-layered curved FG beam coupling with two layers of magneto-electro-elastic beams.

Thus, by showing the validity of the current study, now, the effect of different parameters that can have an impact on the characteristic of a three-layered nanobeam which is modeled by nonlocal strain gradient theory is investigated.

Here it is important to define the nondimensional factors related to the two-parameter elastic foundation as $\bar{K}_w = K_w\frac{L^4}{C}$ and $\bar{K}_p = K_p\frac{L^2}{C}$.

First, the impact of the slenderness of the three-layered curved beam on the nondimensional vibration frequency $\bar{\omega} = \omega L^2\sqrt{\frac{I_0}{C}}$, in Fig. 3, and on the nondimensional critical buckling load $\bar{N} = \frac{NL^2}{C}$ of the system, in Fig. 4, is investigated. In the both of the figures, the results are extracted for various boundary conditions as well as various index number related to the FG core. Additionally, the other constants which plays a role in determining the results are, $l_m = 30nm^2, \frac{e_0 a}{L} = 0$, $V_0 = 0, \Omega_0 = 0$, $\bar{K}_w = 1000$, $\bar{K}_p = 10$, $\frac{L}{R} = \frac{\pi}{360}$, and $h_p = 0.005L$.

The results in Fig. 2 shows that increasing the value of $\frac{L}{h_c}$, regardless of the type of the FG material which is in the core of the system and the type of end conditions, leads in an increase in the vibrational frequency. Also, it is interesting to mention that the increase is more observable in lower values of $\frac{L}{h_c}$, as this intensifying effect is little in thinner beams. Additionally, it should be pointed out that, the higher the index of FG core is, the lower the vibrational frequency can be.

Similar to the results for vibration analysis, the buckling load of the system escalate when the beam has thinner core. However, against the results in frequency response, the buckling load behavior is quite different in different boundary conditions, so much so that the increasing of the buckling load is almost stop in some point of increasing $\frac{L}{h_c}$ of the core for cases with SS and CS end conditions, while it is not the case for systems with CC boundary conditions. Also, it is vital to be mentioned that the core with higher index lead in lower critical buckling load.

Next, the impact of factors associated with the two-parameter elastic foundation are investigated on the vibration as well as buckling load of a three-layered curved beam made of a FG core and two layers of magneto-electro-elastic beams. To do so, the effect of Winkler coefficient, respectively, on the vibration and buckling response of the system in Figs. 4 and 5, and also the impact of Pasternak coefficient on the vibrational behavior along with buckling load of the system in Figs. 6 and 7 is investigated. Also, other constants are $l_m = 30nm^2$, $\frac{e_0 a}{L} = 0.02$, $V_0 = 0, \Omega_0 = 0$, $\frac{L}{h_c} = 20$, $\frac{L}{R} = \frac{\pi}{360}$,and $h_p = 0.005L$.

From Fig. 4, the thing which is worth mentioning is that this coefficient can increase the vibrational frequency, in cases with stiffer foundation. Also, this effect is more apparent in SS and CS boundary conditions, softer end conditions.

Unlike the results of previous figure, the result for critical buckling load exhibit that this parameter has little effect on the buckling load of the system. Additionally, similar to other figures, it can be stated that the buckling load decrease in systems with FG core in which the index number is higher.

Similar to the other coefficient of the foundation, the Pasternak factor can increase the vibrational frequency of the system, and this is more observable through beams with SS and CS boundary conditions.

However, against the results in Fig. 5, the Pasternak coefficient impact on the critical buckling load is significant. Accordingly, it can be seen that the intensifying this factor, stiffening the foundation, can cause the buckling load of the system to escalate.

Here, to investigate the impact of strain gradient length coefficient,l_m,on the vibration response and the critical buckling load of the system, Fig.8 and 9 are presented. In these figures, the variation of nondimensional frequency as well as buckling load against the value of l_m is presented for different values of slenderness of the beam and various end conditions. Additionally, the other constants in these figures are *n=1*,$\frac{e_0 a}{L} = 0.02$, $V_0 = 0, \Omega_0 = 0$, $\bar{K}_w = 1000$, $\bar{K}_p = 10$, $\frac{L}{R} = \frac{\pi}{360}$ and $h_p = 0.005L$.

The exhibited plots in Fig.8 shows that, however little, the increasing the values related to $l_m$can increase the vibrational frequency of the system, which is more apparent in the cases with thinner beams. Also, it is worth to be stated that, as expected, the thinner beams possess higher vibration frequencies.

Unlike the results for vibration response, the critical buckling load can increase dramatically by escalating the value for l_m. Additionally, this increasing effect is higher in cases with thinner core, so much so that these systems have higher slope in this figure.

Now, the vibrational frequencies as well as critical buckling loads of the system versus the various values for nonlocality of the system is plotted in Figs. 10 and 11, respectively. In these two figures, the values are extracted for different end condition as well as slenderness ratio. Also, the constants values are *n=1*, $l_m = 30nm^2$, $V_0 = 0, \Omega_0 = 0$, $\bar{K}_w = 1000$, $\bar{K}_p = 10$, $\frac{L}{R} = \frac{\pi}{360}$ and $h_p = 0.005L$.

As expected, the higher the nonlocality of the system is, the lower the vibration frequency should be, due to the nature of softening effect of nonlocal loads in nonlocal elasticity. Also, the other notable results which can be mentioned is that this softening effect is more observable in beams with thinner core, or higher values of $\frac{L}{h_c}$.

Similar to previous results, the system with higher nonlocality, as they are softer, requires lower loads to reach to critical buckling status. Thus, escalating the nonlocality of the system cause the beam to have lower buckling load.

Lastly, in Figs. 12 and 13, the effect of external magnetic potential as well as external voltage on the variation of the vibration frequency of the three-layered beam is studied for various type of the FG core. The other constants in these figures are $l_m = 30nm^2$, $\frac{L}{h_c} = 20$, $\bar{K}_w = 1000$, $\bar{K}_p = 10$, $\frac{L}{R} = \frac{\pi}{360}$ and $h_p = 0.005L$.

Fig. 12, shows that, by intensifying the value of Ω_0, the vibrational frequency of the system diminishes and lead the system towards buckling, due to the compressive form of positive magnetic potential. Also, this phenomenon can be seen better in systems with SS and CS boundary conditions.

Similar to the magnetic potential, the external voltage can reduce the vibrational frequency of the system. Also, this reduction is more significant in the softer type of end conditions. It worth mentioning that the impact of both external voltages together with magnetic potential is higher when the FG core index is higher.

5. Conclusions

This investigation deals with the vibration and buckling analysis of three-layered curved beam which is made of two magneto-electro-elastic together with an FG layer and is resting on a two-parameter elastic foundation. The size-dependent theory by which the impact of nanosize is taken into account is NSGT. By defining the displacement field via Timoshenko beam theory and employing energy method, the governing equations as well as end conditions are attained. Then, this formulation is solved through GDQM. By studying the various parameter which plays a role in determining the critical loading and frequency response of the system, the below conclusion can be drawn:

- Escalating the nonlocality of the system cause the beam to have lower buckling load.
- The higher the nonlocality of the system is, the lower the vibration frequency is.
- The critical buckling load can increase dramatically by escalating the value for l_m.
- Increasing the values related to l_m can increase the vibrational frequency of the system.
- The premiere associated with foundation can increase the vibrational frequency as well as critical buckling load of the system.
- Increasing the value of $\frac{L}{h_c}$ leads in an increase in the vibrational frequency as well as critical buckling load of the system.
- Intensifying the value of Ω_0 and V_0 diminish the vibrational frequency of the system.

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