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Mechanical buckling of FG-CNTs reinforced composite plate with parabolic distribution using Hamilton's energy principle

Tayeb Si Tayeb [1,2], Mohamed Zidour*[1,3,4], Tayeb Bensattalah [3,4],
Houari Heireche [1,2], Abdelillah Benahmed [1,2] and E.A. Adda Bedia [5]

[1] *Laboratory of Modeling and Multi-Scale Simulation, Department of Physics, Faculty of Exact Science University of Sidi Bel Abbes, Algeria*
[2] *University Djillali Liabes of Sidi-Bel-Abbès, BP 89, Sidi Bel Abbès 22000, Algeria*
[3] *Université of tiaret, BP 78 Zaaroura, 14000 Tiaret, Algeria*
[4] *Laboratory of Geomatics and Sustainable Development, Ibn Khaldoun University of Tiaret, Algeria*
[5] *Department of Civil and Environmental Engineering, King Fahd University of Petroleum & Minerals, 31261 Dhahran, Eastern Province, Saudi Arabia*

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Abstract. The incorporation of carbon nanotubes in a polymer matrix makes it possible to obtain nanocomposite materials with exceptional properties. It's in this scientific background that this work was based. There are several theories that deal with the behavior of plates, in this research based on the Mindlin-Reissner theory that takes into account the transversal shear effect, for analysis of the critical buckling load of a reinforced polymer plate with parabolic distribution of carbon nanotubes. The equations of the model are derived and the critical loads of linear and parabolic distribution of carbon nanotubes are obtained. With different disposition of nanotubes of carbon in the polymer matrix, the effects of different parameters such as the volume fractions, the plate geometric ratios and the number of modes on the critical load buckling are analysed and discussed. The results show that the critical buckling load of parabolic distribution is larger than the linear distribution. This variation is attributed to the concentration of reinforcement (CNTs) at the top and bottom faces for the X-CNT type which make the plate more rigid against buckling.

Keywords: nanotubes; shear deformation; parabolic distribution; buckling; volume fractions

1. Introduction

Recently, a new class of promising materials has been discovered by Iijima (1991) known as single-walled carbon nanotube (SWNT) and multi-walled carbon nanotube (MWNT) has drawn considerable attention. Multitude studies indicated that the carbon nanotubes (CNTs) have an excellent candidate for the reinforcement of polymer composites due to their extraordinary Young's modulus (Van Lier *et al*. 2000), tensile strength (Yu *et al*. 2000), electrical conductivity (Thess *et al*. 1996), thermal properties (Biercuk *et al*. 2002) and flexibility (Ma *et al*. 1998). Several studies have focused on material properties of carbon nanotube-reinforced composites (CNTRCs) and have shown that the introduction of carbon nanotubes into polymers may improve their properties (Fidelus *et al*. 2005, Han and Elliott 2007, Bonnet *et al*. 2007, Semmah *et al*. 2019).

Functionally graded materials (FGMs) are a new types of composites developed recently, namely (FGM) has high potential to use as a structural material. Therefore, by changing the properties of the material it is possible to perform a certain function of material properties (Benahmed *et al*. 2019, Bourada *et al*. 2019, Tlidji *et al*. 2019, Avcar and Mohammed 2018, Ehyaei *et al*. 2017, Fahsi *et al*. 2017, Avcar 2019, Boussoula *et al*. 2019, Avcar and Alwan 2017, Boukhari *et al*. 2016, Daouadji and Adim 2016, Sofiyev and Avcar 2010, Sekkal *et al*. 2017, Bousahla *et al*. 2016, Bounouara *et al*. 2016). Reinforced composites are made up of a combination of fibers (Fellah *et al*. 2019), or particle in a matrix material composite material are increasingly being used in aircraft primary structures because of their superior strength properties over the traditional materials. Diamanti and Soutis (2010) studied the structural health monitoring techniques for aircraft composite structures. Advanced composites have been replacing traditional structural materials to repair the aircraft structures deflection and a nonlinear bending analysis of functionally graded carbon nanotube reinforced composite are presented by (Katnam *et al*. 2013), Bonnet *et al*. (2007) studied the thermal properties and percolation in carbon nanotube-polymer composites. Thus, this topic has been fascinating many researchers for recently years (Zhang *et al*. 2015, Mirzaei and Kiani 2016, Mehar *et al*. 2017, Wu *et al*. 2016, Mehar and Panda 2017, Asadi and Beheshti 2018, Karami *et al*. 2019a).

Due to difficulties encountered in experimental methods, the molecular dynamics (MD) simulations are used to predict the elastic properties of polymer/carbon nanotube composites (Griebel and Hamaekers 2004, Han and Elliott 2007) these studies are limited by systems calculation. The continuum mechanics methods are widely used to predict the responses of reinforced composites

∗Corresponding author, Ph.D.,
E-mail: zidour.m@univ-tiaret.dz

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structure and nanostructures (Kolahchi *et al*. 2017, Shokravi 2017a, Karami *et al*. 2017, 2018a, b, 2019b, c, Kolahchi 2017, Ould Youcef *et al*. 2015, Shokravi 2017b, Odegard *et al*. 2003, Pour *et al*. 2015, Hajmohammad *et al*. 2018, Chemi *et al*. 2015, Shahsavari and Janghorban 2017a, Bouadi *et al*. 2018, Berghouti *et al*. 2019). Vodenitcharova and Zhang (2006) developed a continuum model for pure bending of a straight nanocomposite beam with a circular cross section reinforced by a single-walled carbon nanotube.

The constitutive models and mechanical properties of carbon nanotube polymer composites have been studied analytically, experimentally, and numerically. Cooper *et al*. (2002) and Barber *et al*. (2003), which demonstrated that carbon nanotubes are effective in reinforcing a polymer due to remarkably high separation stress according to a series of pull-out tests of individual carbon nanotubes embedded within polymer matrix. (Wan *et al*. 2005 and Barati 2017) investigated the effective moduli of the CNT reinforced polymer composite, with emphasis on the influence of CNT length and CNT matrix interphase on the stiffening of the composite. The behavior of CNTRC can be considerably improved through the use of a functionally graded distribution of CNTs in the matrix (Civalek 2017, Shokravi 2017b).

The buckling instability problem is a common question in everyday life, in engineering, buckling is a sudden failure of a structure subjected to high compressive stresses and refers to loss of the load-carrying capacity of a component within a structure or of the structure itself. In reality, it is very easy to investigate the buckling instability of small systems like a single homogenous rod. Building an experimental setup one can consider different boundary conditions. There are two major categories leading to the sudden failure of a mechanical component: material failure and structural instability, (buckling). For material failures, you need to consider the yield stress for ductile materials and the ultimate stress for brittle materials. A research project on stability cannot start without recognizing the contribution of Euler to the problem of stability when he published his famous Euler's equation on the elastic stability of columns back in 1744. His original work consisted of determining the buckling load of a cantilever column that was fixed at the bottom and free at the top. Practical solutions are still not available for some types of structures.

Stability is one of the most critical limit states for structures during construction and during their service life. One of the most difficult challenges in structural stability is determining the critical load under which a structure collapses due to the loss of stability; this is because of the complexity of this phenomenon and the many material properties that are influenced by geometric and material imperfections and material nonlinearity. The critical buckling lead is the force that has to be exceeded to buckle the rod. The critical force is contour length-dependent and vanishes for an infinitely long rod.

In cases of critical buckling lead analyses of CNTRC structures, Mehar *et al*. (2019), investigated the buckling of graded CNT-reinforced composite sandwich shell structure under thermal loading. Using the Multiscale modeling approach Mehar and Panda (2019) gave the solutions for thermal critical buckling of nanocomposite curved structure. The nonlinear thermal buckling behaviour of laminated composite panel structure including the stretching effect and higher-order finite element was reported by Katariya *et al*. (2017b). Thus, the topic of buckling of functionally graded materials and laminated composite has been fascinating many researchers for recently years (Meziane *et al*. 2014, Al-Basyouni *et al*. 2015, Katariya and Panda 2016, Kar *et al*. 2017, Kar and Panda 2016, 2017, Kar *et al*. 2016, Bourada *et al*. 2016, Panda and Katariya 2015, Bellifa *et al*. 2017a, b, Panda *et al*. 2017, Panda and Singh 2009, Abdelaziz *et al*. 2017, El-Haina *et al*. 2017, Menasria *et al*. 2017, Chikh *et al*. 2017, Shokravi 2017b, Kaci *et al*. 2018, Mokhtar *et al*. 2018, Yazid *et al*. 2018, Kadari *et al*. 2018, Bourada *et al*. 2018, Hellal *et al*. 2019, Karami *et al*. 2019d, Alimirzaei *et al*. 2019, Meksi *et al*. 2019).

To the best of authors' knowledge, no many reports have been found in the literature on the buckling of sandwich plates and Post-buckling of laminated composite. However, Panda and Singh (2010) used the non-linear finite element method for optimization of Thermal post-buckling of a laminated composite spherical shell panel embedded with shape memory alloy fibres. Katariya *et al*. (2017a) investigated the thermal buckling strength of laminated sandwich composite panel structure embedded with shape memory alloy fibre. Panda and Singh (2013a) gave the post-buckling analysis of laminated composite doubly curved panel embedded with SMA fibers subjected to thermal environment. Buckling analysis of SMA bonded sandwich structure–using FEM has been developed by Katariya and Panda (2018). Panda and Singh (2013b) investigated the thermal post-buckling behavior of laminated composite spherical shell panel using NFEM. Katariya and Panda (2014) studied the Thermo-Mechanical Stability of Composite Cylindrical Panels.

Elastic stability must satisfy two basic criteria, the ability of the structure to support the imposed loading (strength) and the capacity of the structure to resist distortions (stiffness) (Nethercot and Kirby 1979). Elastic buckling instability is frequently associated with large changes of geometry which often occur quickly as the structural member moves from one geometrical position of equilibrium to another. For example, when a wood I-joist is loaded in the plane of its web by a gradually increasing load, an axial deflection is the first wood I-joist response to the applied load. This axial deflection lasts until a particular load is reached. Any further increase in the applied load will cause wood I-joist instability. This instability is characterized by the presence of additional wood I-joists responses to the applied load, including a lateral wood I-joist deflection and rotation of the wood I-joist with respect to its neutral axis. This condition of instability is called lateral-torsional buckling instability. The load at which lateral-torsional buckling instability occurs is known as the critical lateral-torsional buckling load (Nethercot and Trahair 1976).

This present paper attempts to show the critical buckling load of a functionally graded reinforced polymer plate with a parabolic distribution of carbon nanotubes using the shear deformation plate theory. This research seeks to analyze the influences of various parameters on the critical buckling load of plate such as plate thickness, aspect ratios, volume fraction and type of reinforcement.

2. Geometrical configuration and properties of CNTs plate

As shown in Figs. 1 and 2, consider the case of a uniform thickness, an FG-CNT reinforced polymer plate with linear and parabolic distribution of carbon nanotubes referring to coordinates (x, y, z) with length a, width b and thickness h.

Three different models of the distribution of reinforcements across the thickness are taken into consideration in this study such as uniformly distributed (referred to as UD-CNT), linear (referred to as CNT- L) and non-linear (referred to as CNT-NL) in the thickness direction (Fig. 2).

Several micromechanical models have been developed to predict the effective material properties of CNTRCs, The Mori-Tanaka model is applicable to micro particles (Seidel and Lagoudas 2006, Li *et al*. 2007) and the rule of mixture (Anumandla and Gibson 2006, Esawi and Farag 2007) is simple and convenient to predict the global material properties of the CNTRC.

In the present study, according to the rule of mixture by introducing the CNT efficiency parameters (η_1, η_2, η_3), the effective Young's modulus and shear modulus of the CNTRC layer can be expressed as (Shen 2009).

$$E_{11} = \eta_1 V_{cnt} E_{11}^{cnt} + V_p E^p \tag{1a}$$

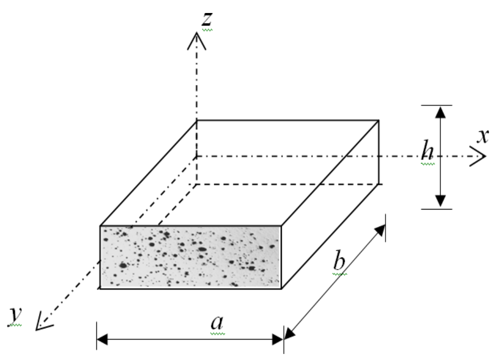

Fig. 1 Plate with nanotube of Carbone

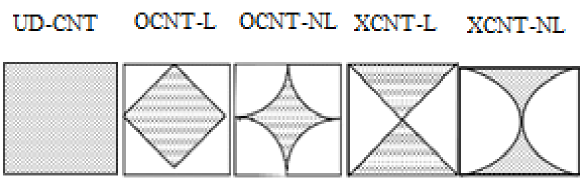

Fig. 2 The different models of the reinforcement provisions

Table 1 Distributions of reinforcements across the thickness

Uniformly distributed	UD- CNT	$V_{cnt} = V_{cnt}^*$
Linear functionally graded	O- CNT- L	$V_{cnt} = 2\left(1 - \frac{2\lvert z\rvert}{h}\right) V_{cnt}^*$
	X- CNT- L	$V_{cnt} = 2\left(\frac{2\lvert z\rvert}{h}\right) V_{cnt}^*$
Non-linear functionally graded	O- CNT- NL	$V_{cnt} = 2\left(1 - \frac{2\lvert z\rvert}{h}\right)^2 V_{cnt}^*$
	X- CNT- NL	$V_{cnt} = 2\left(\frac{2\lvert z\rvert}{h}\right)^2 V_{cnt}^*$

$$\frac{\eta_2}{E_{22}} = \frac{V_{cnt}}{E_{22}^{cnt}} + \frac{V_p}{E^p} \tag{1b}$$

$$\frac{\eta_3}{G_{12}} = \frac{V_{cnt}}{G_{12}^{cnt}} + \frac{V_p}{G^p} \tag{1c}$$

Where E_{11}^{cnt}, E_{22}^{cnt} and G_{12}^{cnt} indicate the Young's moduli and shear modulus of SWCNTs, respectively, and E^P and G^P represent the properties of the isotropic matrix. η_1, η_2 and η_3 are CNT/matrix efficiency parameters, the V_{cnt} and V_P are the volume fractions of the carbon nanotubes and matrix, respectively, and are related by $V_{cnt} + V_P = 1$. For other properties in terms of Poisson's ratio (ν) and mass density (ρ), these can be written as

$$\nu_{12} = V_{cnt}\nu_{12}^{cnt} + V_p\nu^p, \quad \rho = V_{cnt}\rho^{cnt} + V_p\rho^p \tag{2}$$

To consider the three distributions of reinforcements across the thickness (Table 1).

Where $V_{cnt}^*$is the given volume fraction of CNTs, which can be obtained from the following equation (Draoui *et al*. 2019)

$$V_{cnt}^* = \frac{W_{cnt}}{W_{cnt} + (\rho^{cnt}/\rho^m)(1 - W_{cnt})} \tag{3}$$

Where W_{cnt} is the mass fraction of the carbon nanotube in the nano-composite plate. in this study, we introduce the CNT efficiency parameter to consider the small-scale effect and other effects onthe material properties of CNTRCs. The CNT efficiency parameters (η) associated with the given volume fraction (V_{cnt}^*) (Zhu *et al*. 2012)

$\eta_1 = 0.149$ and $\eta_2 = \eta_3 = 0.934$
for the case of $V_{cnt}^* = 0.11$

$\eta_1 = 0.150$ and $\eta_2 = \eta_3 = 0.941$
for the case of $V_{cnt}^* = 0.14$

$\eta_1 = 0.149$ and $\eta_2 = \eta_3 = 1.381$
for the case of $V_{cnt}^* = 0.17$

3. Equations of motion

According to the high order shear deformation plate theory for describing the behaviour of the CNTRC plates. The displacements of any point in the plate along the x, y and z axes, denoted by $u(x, y, z, t)$, $v(x, y, z, t)$ and $w(x, y, t)$ respectively is given below (Reddy 2004, Mahi *et al*. 2015)

$$\begin{Bmatrix} u(x,y,z,t) \\ v(x,y,z,t) \\ w(x,y,t) \end{Bmatrix} = \begin{Bmatrix} u_0(x,y,t) \\ v_0(x,y,t) \\ w_0(x,y,t) \end{Bmatrix} - \begin{Bmatrix} zw_{,x} \\ zw_{,y} \\ 0 \end{Bmatrix} + \psi(z)\begin{Bmatrix} \phi_x \\ \phi_y \\ 0 \end{Bmatrix} \tag{4}$$

In which u_0, v_0 and w_0 are the displacements along the x, y and z directions in the mid plane of the plate, t is time and ϕx, ϕy are the total bending rotation of the cross-section at any point of the reference plane. If the last term in Eq. (4) is neglected, the displacements are reduced to the classical plate theory (*CPT*). Also, the first order shear deformation theory (FSDT) is obtained by setting, $\Psi(z) = z$.

In terms of higher order shear deformation theories, the corresponding shape functions are defined as follows.

Third order shear deformation theory (TSDT):

$$\psi(z) = z\left(1 - \frac{4z^2}{3\hbar^2}\right) \tag{5a}$$

Sinusoidal shear deformation theory (SSDT):

$$\Psi(z) = \frac{h}{\pi}\sin\left(\frac{\pi z}{h}\right) \tag{5b}$$

The linear normal strain and transverse shear strain are associated with the displacements via

$$\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial u_0}{\partial x} \\ \dfrac{\partial v_0}{\partial y} \\ \dfrac{\partial u_0}{\partial y} + \dfrac{\partial v_0}{\partial x} \end{Bmatrix} - z\begin{Bmatrix} \dfrac{\partial^2 w_0}{\partial x^2} \\ \dfrac{\partial^2 w_0}{\partial y^2} \\ 2\dfrac{\partial^2 w_0}{\partial x \partial y} \end{Bmatrix} + \psi(z)\begin{Bmatrix} \dfrac{\partial \varphi_x}{\partial x} \\ \dfrac{\partial \varphi_y}{\partial y} \\ \left(\dfrac{\partial \varphi_x}{\partial y} + \dfrac{\partial \varphi_y}{\partial x}\right) \end{Bmatrix} \tag{6a}$$

$$\begin{Bmatrix} \gamma_{xz} \\ \gamma_{yz} \end{Bmatrix} = \begin{Bmatrix} \varphi_x + \dfrac{\partial \psi(z)}{\partial z} \\ \varphi_y + \dfrac{\partial \psi(z)}{\partial z} \end{Bmatrix} \tag{6b}$$

The expression of normal and shear stress are written by linear elastic constitutive law as

$$\begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau_{xy} \end{Bmatrix} = \begin{pmatrix} Q_{11} & Q_{12} & 0 \\ Q_{21} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{pmatrix}\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \gamma_{xy} \end{Bmatrix}; \tag{7a}$$

$$\begin{Bmatrix} \sigma_{yz} \\ \sigma_{xz} \end{Bmatrix} = \begin{pmatrix} Q_{44} & 0 \\ 0 & Q_{55} \end{pmatrix}\begin{Bmatrix} \gamma_{yz} \\ \gamma_{xz} \end{Bmatrix}, \tag{7a}$$

Where Q_{ij} are the transformed elastic constants

$$\begin{aligned} Q_{11} &= \frac{E_{11}}{1-\upsilon_{12}\upsilon_{21}}, & Q_{22} &= \frac{E_{22}}{1-\upsilon_{12}\upsilon_{21}}, \\ Q_{12} &= \frac{\upsilon_{21}E_{11}}{1-\upsilon_{12}\upsilon_{21}}, & Q_{66} &= G_{12} \\ Q_{55} &= G_{13} & Q_{44} &= G_{23} \end{aligned} \tag{7b}$$

The governing differential equations of motion can be derived from Hamilton's principle (Attia *et al*. 2015, Guessas *et al*. 2018, Belabed *et al*. 2018, Cherif *et al*. 2018, Fourn *et al*. 2018, Youcef *et al*. 2018, Karami *et al*. 2018c, Draiche *et al*. 2019, Chaabane *et al*. 2019).

$$\int_0^t (\delta U + \delta V)\, dt = 0 \tag{8}$$

Where δU and δV are the virtual variation of the strain energy and the virtual work done by external forces.

The expression of the virtual strain energy is (Beldjelili *et al*. 2016, Zine *et al*. 2018, Attia *et al*. 2018)

$$\begin{aligned} \delta U = \int_{-\frac{\hbar}{2}}^{\frac{\hbar}{2}}\int_A & \sigma_{xx}\delta\varepsilon_{xx} + \sigma_{yy}\delta\varepsilon_{yy} + \sigma_{xy}\delta\gamma_{xy} \\ & + \sigma_{yz}\delta\gamma_{yz} + \sigma_{xz}\delta\gamma_{xz}\big) dA dx \end{aligned} \tag{9}$$

By substituting Eq. (6) into Eq. (9), one obtains

$$\begin{aligned} \delta U = \int_A & \{N_{xx}\delta u_{0,x} - M_{xx}\delta w_{O,xx} + P_{xx}\delta\emptyset_{x,x} \\ & + N_{yy}\delta v_{0,y} M_{yy}\delta w_{0,yy} P_{yy}\delta\emptyset_{y,y} \\ & + N_{xy}\left(\delta u_{0,y} + \delta v_{0,x}\right) - 2M_{xy}\delta w_{0,xy} \\ & + P_{xy}\left(\delta\emptyset_{x,y} + \delta\emptyset_{y,x}\right) + R_{yz}\delta\emptyset_y + R_{xz}\delta\emptyset_x\} dx dy \end{aligned} \tag{10}$$

Where stress resultants can be defined as follows

$$\left(\mathrm{N_{xx}}, \mathrm{N_{yy}}, \mathrm{N_{xy}}\right) = \int_{-\hbar/2}^{\hbar/2}\left(\sigma_{xx}, \sigma_{yy}, \sigma_{xy}\right)\mathrm{d}z \tag{11a}$$

$$\left(\mathrm{M_{xx}}, \mathrm{M_{yy}}, \mathrm{M_{xy}}\right) = \int_{-\hbar/2}^{\hbar/2} z\left(\sigma_{xx}, \sigma_{yy}, \sigma_{xy}\right)\mathrm{d}z \tag{11b}$$

$$\begin{aligned} \left(\mathrm{P_{xx}}, \mathrm{P_{yy}}, \mathrm{P_{xy}}\right) &= \int_{-\frac{\hbar}{2}}^{\frac{\hbar}{2}} \psi(z)\left(\sigma_{xx}, \sigma_{yy}, \sigma_{xy}\right)\mathrm{d}z \\ R_{yz} &= \int_{-\frac{\hbar}{2}}^{\frac{\hbar}{2}} \frac{\partial\psi(z)}{\partial(z)}\sigma_{yz}\mathrm{d}z \\ R_{xz} &= \int_{-\hbar/2}^{\hbar/2} \frac{\partial\psi(z)}{\partial(z)}\sigma_{xz}\mathrm{d}z \end{aligned} \tag{11c}$$

The stress resultants in form of material stiffness and displacement components are obtained by substituting Eq.

(7) into Eq. (10).

$$\begin{Bmatrix} N_{xx} \\ N_{yy} \\ N_{xy} \end{Bmatrix} = \begin{bmatrix} A_{11} & A_{12} & 0 \\ A_{21} & A_{22} & 0 \\ 0 & 0 & A_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(0)}{}_{xx} \\ \varepsilon^{(0)}{}_{yy} \\ \gamma^{(0)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} B_{11} & B_{12} & 0 \\ B_{21} & B_{22} & 0 \\ 0 & 0 & B_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(1)}{}_{xx} \\ \varepsilon^{(1)}{}_{yy} \\ \gamma^{(1)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} C_{11} & C_{12} & 0 \\ C_{21} & C_{22} & 0 \\ 0 & 0 & C_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(\psi)}{}_{xx} \\ \varepsilon^{(\psi)}{}_{yy} \\ \gamma^{(\psi)}{}_{xy} \end{Bmatrix} \quad (12a)$$

$$\begin{Bmatrix} M_{xx} \\ M_{yy} \\ M_{xy} \end{Bmatrix} = \begin{bmatrix} B_{11} & B_{12} & 0 \\ B_{21} & B_{22} & 0 \\ 0 & 0 & B_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(0)}{}_{xx} \\ \varepsilon^{(0)}{}_{yy} \\ \gamma^{(0)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} D_{11} & D_{12} & 0 \\ D_{21} & D_{22} & 0 \\ 0 & 0 & D_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(1)}{}_{xx} \\ \varepsilon^{(1)}{}_{yy} \\ \gamma^{(1)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} E_{11} & E_{12} & 0 \\ E_{21} & E_{22} & 0 \\ 0 & 0 & E_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(\psi)}{}_{xx} \\ \varepsilon^{(\psi)}{}_{yy} \\ \gamma^{(\psi)}{}_{xy} \end{Bmatrix} \quad (12b)$$

$$\begin{Bmatrix} P_{xx} \\ P_{yy} \\ P_{xy} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & 0 \\ C_{21} & C_{22} & 0 \\ 0 & 0 & C_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(0)}{}_{xx} \\ \varepsilon^{(0)}{}_{yy} \\ \gamma^{(0)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} E_{11} & E_{12} & 0 \\ E_{21} & E_{22} & 0 \\ 0 & 0 & E_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(1)}{}_{xx} \\ \varepsilon^{(1)}{}_{yy} \\ \gamma^{(1)}{}_{xy} \end{Bmatrix} + \begin{bmatrix} F_{11} & F_{12} & 0 \\ F_{21} & F_{22} & 0 \\ 0 & 0 & F_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon^{(\psi)}{}_{xx} \\ \varepsilon^{(\psi)}{}_{yy} \\ \gamma^{(\psi)}{}_{xy} \end{Bmatrix} \quad (12c)$$

$$\begin{Bmatrix} R_{yz} \\ R_{xz} \end{Bmatrix} = \begin{bmatrix} H_{44} & 0 \\ 0 & H_{55} \end{bmatrix} \begin{Bmatrix} \gamma_{yz}^{(0)} \\ \gamma_{xz}^{(0)} \end{Bmatrix} \quad (12d)$$

Where A_{ij}, B_{ij}, C_{ij}, D_{ij}, E_{ij}, F_{ij}, are the plate stiffness, defined by

$$[A_{ij}, B_{ij}, D_{ij}] = \int_{-\frac{h}{2}}^{\frac{h}{2}} Q_{ij}[1, z, z^2]\, dz; \quad i,j = 1,2,6 \quad (13a)$$

$$[C_{ij,}E_{ij}, F_{ij}] = \int_{-\frac{h}{2}}^{\frac{h}{2}} \psi\,(z) Q_{ij}[1, z, \psi(z)] dz; \quad i,j = 1,2,6 \quad (13b)$$

For the CNTRC plates, the principle of virtual work done by external loadingsis

$$\delta V = \int_A \left(N_x^0 \frac{\partial w_0}{\partial x}\frac{\partial \delta w_0}{\partial x} + N_y^0 \frac{\partial w_0}{\partial y}\frac{\partial \delta w_0}{\partial y} \right) dxdy \quad (14)$$

By substituting the equations of the strain energy and the virtual work done by external forces into Hamilton's principle, Then, integrating by parts and collecting the coefficients of δu_0, δv_0, δw_0, $\delta\varphi_x$ and $\delta\varphi_y$, leads to the following equations of motion.

$$\delta u_0: \quad \frac{\partial N_{xx}}{\partial x} + \frac{\partial N_{xy}}{\partial y} = 0 \quad (15a)$$

$$\delta v_0: \quad \frac{\partial N_{yy}}{\partial y} + \frac{\partial N_{xy}}{\partial x} = 0 \quad (15b)$$

$$\delta w_0: \quad \frac{\partial^2 M_{xx}}{\partial x^2} + \frac{\partial^2 M_{yy}}{\partial y^2} + 2\frac{\partial^2 M_{xy}}{\partial x \partial y} + N_x^0 \frac{\upsilon^2 w_0}{\upsilon^2 x} + N_y^0 \frac{\upsilon^2 w_0}{\upsilon^2 y} = 0 \quad (15c)$$

$$\delta\phi_x: \quad \frac{\partial P_{xx}}{\partial x} + \frac{\partial P_{xy}}{\partial y} - R_{yz} = 0 \quad (15d)$$

$$\delta\phi_y: \quad \frac{\partial P_{yy}}{\partial y} + \frac{\partial P_{xy}}{\partial x} - R_{yz} = 0 \quad (15e)$$

The Navier solution procedure is employed to formulate the closed-form solutions for buckling problems of simply supported CNTRC plates (Bakhadda *et al*. 2018, Medani *et al*. 2019).

$$\begin{aligned} u_0(x,y,t) &= \sum_{M=1}^{\infty}\sum_{N=1}^{\infty} U_{MN} e^{i\omega t} cos(\alpha x)\, sin(\zeta y) \\ v_0(x,y,t) &= \sum_{M=1}^{\infty}\sum_{N=1}^{\infty} V_{MN} e^{i\omega t} sin(\alpha x)\, cos(\zeta y) \\ w_0(x,y,t) &= \sum_{M=1}^{\infty}\sum_{N=1}^{\infty} W_{MN} e^{i\omega t} sin(\alpha x)\, sin(\zeta y) \\ \varphi_x(x,y,t) &= \sum_{M=1}^{\infty}\sum_{N=1}^{\infty} \Theta x_{MN} e^{i\omega t} cos(\alpha x)\, sin(\zeta y) \\ \varphi_y(x,y,t) &= \sum_{M=1}^{\infty}\sum_{N=1}^{\infty} \Theta y_{MN} e^{i\omega t} sin(\alpha x)\, cos(\zeta y) \end{aligned} \quad (16)$$

Where $\alpha = \frac{M\pi}{a}$ and $\zeta = \frac{N\pi}{b}. i = \sqrt{-1}$.

Where U_{MN}, and V_{MN}, W_{MN}, Θx_{MN} *and* Θy_{MN} are arbitrary parameters and ω , the frequency of free vibration.

Substituting the Eq. (13) into the Eq. (14), one obtains the closed-form solutions which are presented in the following matrix form.

$$\left(\begin{bmatrix} s_{11} & s_{12} & s_{13} & s_{14} & s_{15} \\ s_{12} & s_{22} & s_{23} & s_{24} & s_{25} \\ s_{13} & s_{23} & s_{33} & s_{34} & s_{35} \\ s_{14} & s_{24} & s_{34} & s_{44} & s_{45} \\ s_{15} & s_{25} & s_{35} & s_{45} & s_{55} \end{bmatrix} \right) \begin{Bmatrix} U_{MN} \\ V_{MN} \\ W_{MN} \\ \Theta_{x_{MN}} \\ \Theta_{y_{MN}} \end{Bmatrix} = 0 \quad (17)$$

Where

$$
\begin{aligned}
&s_{11} = -A_{11}\alpha^2 + A_{66}\zeta^2, \quad s_{12} = -\alpha\zeta(A_{12} + A_{66}),\\
&s_{13} = 0, \quad s_{14} = -B_{11}\alpha^3 - B_{66}\zeta^2,\\
&s_{15} = -B_{12}\alpha\zeta - B_{66}\alpha\zeta,\\
&s_{21} = s_{12}, \quad s_{22} = -A_{66}\alpha^2 - A_{22}\zeta^2,\\
&s_{23} = 0, \quad s_{24} = -B_{12}\alpha\zeta - B_{66}\alpha\zeta,\\
&s_{25} = -B_{66}\alpha^2 - B_{22}\zeta^2, \quad s_{31} = s_{13},\\
&s_{32} = s_{23},\\
&s_{33} = -D_{55}\alpha^2 - D_{44}\zeta^2 - Nx\alpha^2 - Ny\zeta^2,\\
&s_{34} = -D_{55}\alpha, \quad s_{35} = -D_{44}\zeta,\\
&s_{41} = s_{14}, \quad s_{42} = s_{24},\\
&s_{43} = s_{34},\\
&s_{44} = -C_{11}\alpha^2 - C_{66}\zeta^2 - D_{55},\\
&s_{45} = -\alpha\zeta(C_{12} + C_{66}), \quad s_{51} = s_{15},\\
&s_{52} = s_{25}, \quad s_{53} = s_{35},\\
&s_{54} = s_{45},\\
&s_{55} = -D_{44} - C_{66}\alpha^2 - C_{22}\zeta^2
\end{aligned} \tag{18}
$$

The following dimensionless parameter is used to present the numerical results for buckling analyses of CNTRC plates.

$$\overline{N}_{cr} = \frac{N_{cr}\alpha^2}{\pi D_0} \quad \text{Where} \quad D_0 = \frac{E^p h^3}{12[1-(\nu^P)^2]} \tag{19}$$

4. Results and discussions

Numerical results are presented and discussed in this section for FG-CNTRC plates. We first need to determine the effective material characteristics of CNTRCs plates employed throughout this work are given as follows.

PMPV (Polymer) is used as the matrix in which material properties are: $\nu^P = 0.34$, $\rho^P = 1150\ \text{kg/m}^3$ and $E^P = 2.1$ GPa. For reinforcement material, the armchair (10,10) SWCNTs is chosen with the following properties according to the study of Zhu *et al*. (2012)

$$\nu_{12}^{cnt} = 0.175;\ \rho^{cnt} = 1400\ \text{kg/m}^3;\ E_{11}^{cnt} = 5.6466\ TPa;$$
$$E_{22}^{cnt} = 7.0800\ \text{TPa};\quad G_{12}^{cnt} = G_{13}^{cnt} = G_{23}^{cnt} = 1.9445\ \text{TPa}$$

So as to check the mathematical formulation in previous sections of the present theory, Table 1 demonstrates a comparison between the results obtained by the present model and the results of TSDT (Wattanasakulpong and Chaikittiratana 2015) which is based on the first-order theory Mindlin- Reissner, this examination is made only for linear distribution under uniaxial and biaxial loading, with different values of carbon nanotube volume fraction and various reinforcement plate are considered in this table with thickness ratio of plate (a/h = 10). It can be seen the good agreement between the results.

Table 2 shows a comparison between the distribution of the linear and non-linear reinforcement linear inside the polymer matrix. We note from this table that the form non linear (X-CNT-NL) gives higher critical loads than the other forms (X-CNT-L, O-CNT-L, O-CNT-NL) under uniaxial and biaxial loading. This variation shows that the nonlinear distribution of the reinforcement makes the plate stiffer that will resist better against buckling. The high variation of

Table 2 The comparison results of dimensionless critical buckling load of present square plate with (Wattanasakulpong and Chaikittiratana 2015) results under different loading

Uniaxial loading γ_x = -1, γ_y = 0				
Reinforcement type	Source	$V_{cnt}^* = 0.11$	$V_{cnt}^* = 0.14$	$V_{cnt}^* = 0.17$
UD-CNT	TSDT	20.6814	23.3559	32.3180
	SSDT	20.7286	23.4229	32.3890
	Present	20.6788	23.3520	32.3142
X-CNT	TSDT	24.2864	26.8941	37.6943
	SSDT	24.3943	27.0177	37.8069
	Present	24.2791	26.8860	37.6881
O-CNT	TSDT	14.4990	16.6984	22.6823
	SSDT	14.4515	16.6451	22.6276
	Present	14.5040	16.7041	22.6883
Biaxial loading γ_x = -1, γ_y = -1				
UD-CNT	TSDT	10.3407	11.6780	16.1590
	SSDT	10.3643	11.7115	16.1945
	Present	10.3394	11.6760	16.1571
X-CNT	TSDT	12.1432	13.4471	18.8472
	SSDT	12.1972	13.5089	18.9035
	Present	12.1396	13.4430	18.8440
O-CNT	TSDT	7.2495	8.3492	11.3411
	SSDT	7.2257	8.3225	11.3138
	Present	7.2520	8.3521	11.3442

Table 3 Dimensionless critical buckling loads of CNTRC square plates with different types of distributions (linear and nonlinear) and various (a/h) ratios

Uniaxial loading: γ_x = -1, γ_y = 0					
a/h	UD-CNT	X-CNT	O-CNT	X-CNTNL	O-CNTNL
5	13.9179	14.8082	11.5783	15.3194	10.2369
10	32.3142	37.6881	22.6883	40.3459	17.3376
20	51.8827	68.6094	30.9110	77.5797	21.3800
40	61.5705	87.3715	34.0940	102.3821	22.7389
Biaxial loading: γ_x = -1, γ_y = -1					
5	6.9590	7.4041	5.7892	7.6597	5.1184
10	16.1571	18.8440	11.3442	20.1730	8.6688
20	25.9414	34.3047	15.4555	38.7898	10.6900
40	30.7852	43.6858	17.0470	51.1910	11.3695

Table 4 The effect of different values of the volume fraction on the critical buckling load for CNTRC plate

	Uniaxial loading: γ_x = -1, γ_y = 0			Biaxial loading: γ_x = -1, γ_y = -1		
Volume fraction	UD-CNT	X-CNT-NL	O-CNT-NL	UD-CNT	X-CNT-NL	O-CNT-NL
$V_{cnt}^{*} = 0.11$	20.6788	26.0727	11.1411	10.3394	13.0363	5.5706
$V_{cnt}^{*} = 0.14$	23.3520	28.6197	12.9088	11.6760	14.3098	6.4544
$V_{cnt}^{*} = 0.17$	32.3142	40.3459	17.3376	16.1571	20.1730	8.6688

Table 5 Effect of mode number and type of loading on the variation of critical buckling load for different types of CNTRC plate

		Uniaxial loading: γ_x = -1, γ_y = 0			Biaxial loading: γ_x = -1, γ_y = -1		
(n; m)	V*cnt = 0.17	UD-CNT	X-CNT-NL	O-CNT-NL	UD-CNT	X-CNT-NL	O-CNT-NL
(1,1)		32.3142	40.3459	17.3376	16.1571	20.1730	8.6688
(1,2)		56.5915	60.7802	41.4465	50.9324	54.7022	37.3014
(1,3)		70.6534	77.1926	49.6396	67.9360	74.2237	47.7287
(n; m)	V*cnt = 0.14						
(1,1)		23.3520	28.6197	12.9088	11.6760	14.3098	6.4544
(1,2)		38.9258	42.3454	27.8884	35.0333	38.1109	25.0996
(1,3)		49.7685	55.5404	32.7098	47.8543	53.4043	31.4517
(n; m)	V*cnt = 0.11						
(1,1)		20.6788	26.0727	11.1411	10.3394	13.0363	5.5706
(1,2)		35.9232	39.6244	25.1726	32.3309	35.6619	22.6553
(1,3)		44.9805	50.3565	29.8728	43.2505	48.4197	28.7237

critical buckling load is esteemed at the high values of (a/h) ratios.

The dimensionless buckling critical loads of square reinforced plate under uniaxial and biaxial loads for various reinforcements are presented in Table 3 with different values of volume fraction. It is seen that the dimensionless critical load increases if the volume fraction of CNTs increases for al reinforcement type. On the other hand, the X-CNT-NL reinforced plate has a high resistance against buckling compared to other types of reinforcement because there is a concentration of the reinforcement at the top and bottom face of reinforced plate.

The effect of various mode number on the dimensionless critical buckling load of square reinforced plate are also presented in the Table 4 under two types of compressive load, uniaxial (γ_x = -1, γ_y = 0) and biaxial (γ_x = -1, γ_y = -1). It is deduced that the dimensionless critical buckling load increase by increasing of the number of modes. The increase of dimensionless critical buckling load is attributed to the deformation configuration concerning the vibration mode when the number of modes increases the wave length decreases and the plate supports better under

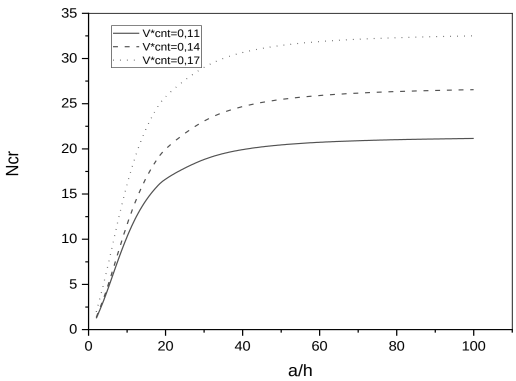

Fig. 3 The effect of aspect ratio (a/h) and values of volume fraction on the dimensionless critical buckling load of UD-NT under biaxial load

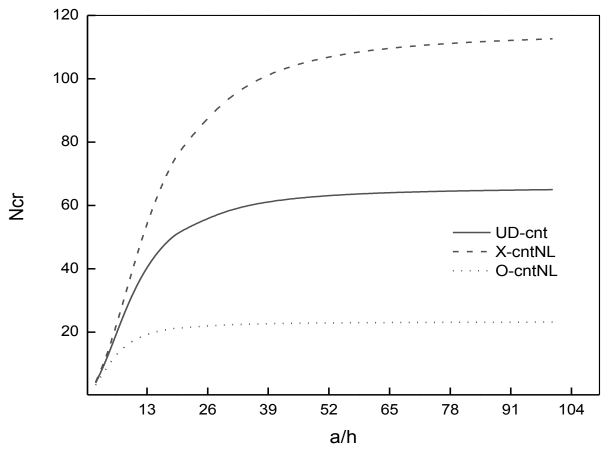

Fig. 5 The dimensionless critical buckling load of a square plate for different form of reinforcement distribution and geometric ratio with Vcnt* = 0.17

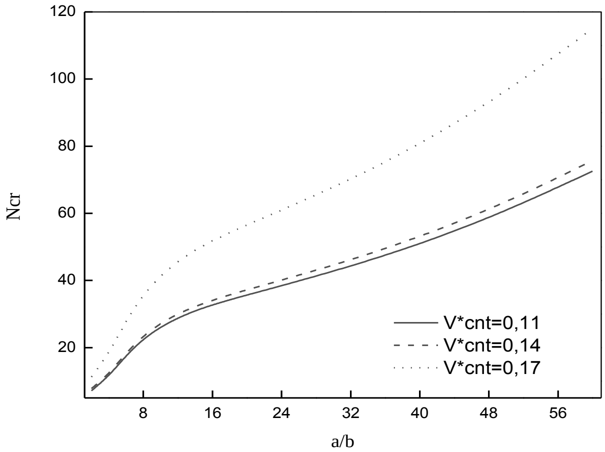

Fig. 4 The effect of geometric ratio (a/b) and values of volume fraction on the dimensionless critical buckling load of UD-CNT under biaxial load with (a/h = 10)

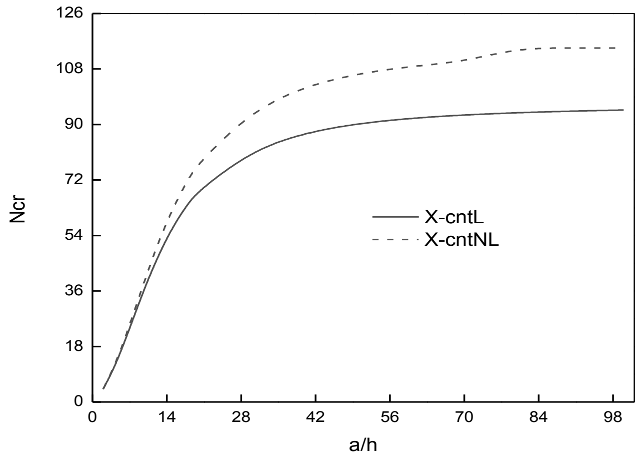

Fig. 6 The comparison between the linear X-CNT-L and nonlinear X-CNT-NL distribution for the plate type X-CNT

the applied axial load. In addition, the results reveal that the dimensionless critical buckling load results increase as the volume fraction increase.

Fig. 3 shows the effect of ratio a/h on the critical buckling load for different values of the volume fraction of CNT. Note that the increase in the ratio a/h leads to an increase in the critical buckling load. The increase in the ratio a/h makes the plate thin and diminish the transversal sheer effect taken into consideration by the Mindlin-Reissner theory. In addition, the critical load with a volume fraction equal 0.17 gives the largest load compared to the other fractions of carbon nanotube. The increase in the dimensionless critical buckling load is appropriate to the quantity of carbon nanotubes in the reinforced plate.

The effect of geometric ratio a/b on the dimensionless critical buckling load for different values of the volume fraction of CNT is presented in Fig. 4. Note that the increase in the ratio a/b leads to an increase in the critical buckling load. The effect of geometric ratio a/b is attributed to the increase of reinforced plate dimensions. Also, the dimensionless critical buckling load increase with increasing of the CNTs volume fraction.

Fig. 5 shows the influence of the geometric ratio parameter a/h and the distribution of carbon nanotubes in the polymer matrix on the critical buckling load. We observe that the critical buckling load increase with increasing of ratio a/h. Furthermore, the plate with distribution nonlinear X-CNTNL gives a Larger critical buckling load because has a high resistance against buckling compared to other reinforcement types UD -CNT and O-CNT-NL. The high resistance is attributing to the concentration of the CNTs reinforcement at the top and bottom face of reinforced plate.

The linear X-CNT-L and nonlinear X-CNT-NL distribution for the X-CNT reinforcement plate type is illustrated in Fig. 6. We see that the critical buckling load for both distribution increase when the ratio a/h increases. This increase is estimated on the little values of a/h ratio. We also note that the critical load of buckling of nonlinear distribution X-CNT-NL is larger than the linear distribution. This variation is attributed to the concentration of reinforcement (CNTs) at the top and bottom faces of the plate, which makes it more rigid against buckling.

5. Conclusions

In this paper, the influence of different parameters on the dimensionless critical buckling load of carbon nanotube-reinforced composite plates using the shear deformation theory is studied and discussed. The overseeing differential equations incorporate the various parameters that are solved by implementing Hamilton's principle and the dimensionless critical load analyses of linear and nonlinear distribution of CNTs are gotten. The exactness of the outcomes is analyzed utilizing the available date in the literature. Finally, through some parametric investigated study the results showed the dependence of buckling behavior on the different parameters such as aspect ratios, volume fraction, plate thickness and linear and nonlinear distribution.

From the numerical outcomes, it is presumed that the concentration of the nanotubes in the nonlinear distribution at the top and bottom face of plate conduce to high resistance against buckling compared with different types of reinforcement. In terms of critical buckling load, the results show:

- The form nonlinear (X-CNT-NL) gives higher critical loads than the other forms (UD –CNT,X-CNT-L, O-CNT-L, O-CNT-NL) under uniaxial and biaxial loading.
- It is seen that the dimensionless critical load increases if the volume fraction of CNTs increases for al reinforcement type.
- It is deduced that the dimensionless critical buckling load increase by increasing of the number of modes
- As the number of modes increases, the wavelength decreases and the plate supports better under the applied axial load.
- The results reveal that the dimensionless critical buckling load results increase as the volume fraction increase.
- The increase in the ratio a/h makes the plate thin and diminish the transversal sheer effect taken into consideration by the Mindlin-Reissner theory.
- The high resistance of the reinforced plate is attributing to the concentration of the CNTs reinforcement at the top and bottom face.

Finally, the results demonstrate the dependence of critical buckling load on the different parameters and the concentration of reinforcement (CNTs) at the top and bottom faces of nonlinear distribution X-CNT-NL plate make it more rigid compared with the linear distribution. An improvement of present formulation will be considered in the future work to consider the thickness stretching effect by using quasi-3D shear deformation models (Draiche *et al*. 2016, Ait Atmane *et al*. 2017, Abualnour *et al*. 2018, Benchohra *et al*. 2018, Younsi *et al*. 2018, Bouhadra *et al*. 2018, Karami *et al*. 2018d, e, Addou *et al*. 2019, Boukhlif *et al*. 2019, Boutaleb *et al*. 2019, Mahmoudi *et al*. 2019, Zarga *et al*. 2019, Bouanati *et al*. 2019, Zaoui *et al*. 2019, Khiloun *et al*. 2019, Boulefrakh *et al*. 2019).

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