A nonlocal sinusoidal shear deformation beam theory with application to bending, buckling, and vibration of nanobeams

Abstract
This paper presents a nonlocal sinusoidal shear deformation beam theory for the bending, buckling, and vibration of nanobeams. The present model is capable of capturing both small scale effect and transverse shear deformation effects of nanobeams, and does not require shear correction factors. Based on the nonlocal differential constitutive relations of Eringen, the equations of motion as well as the boundary conditions of the beam are derived using Hamilton’s principle. Analytical solutions for the deflection, buckling load, and natural frequency are presented for a simply supported beam, and the obtained results are compared with those predicted by the nonlocal Timoshenko beam theory. The comparison firmly establishes that the present beam theory can accurately predict the bending, buckling, and vibration responses of short nanobeams where the small scale and transverse shear deformation effects are significant.

Keywords: Nonlocal theory; Sinusoidal theory; Bending; Buckling; Vibration; Nanobeam

1. Introduction
Nanostructures are widely used in micro- and nano-scale devices and systems such as biosensors, atomic force microscopes, micro-electro-mechanical systems (MEMS) and nano-electro-mechanical systems (NEMS) due to their superior mechanical, chemical, and electronic properties [1]. In such applications, small scale effects are often observed. These effects can be captured using size-dependent continuum mechanics such as strain
gradient theory [2], modified couple stress theory [3], and nonlocal elasticity theory [4]. Among these theories, the nonlocal elasticity theory initiated by Eringen is the most commonly used theory. Unlike the local theories which assume that the stress at a point is a function of strain at that point, the nonlocal elasticity theory assumes that the stress at a point is a function of strains at all points in the continuum.

Based on the nonlocal constitutive relation of Eringen, a number of papers have been published attempting to develop nonlocal beam models for predicting the responses of carbon nanotubes. The nonlocal Euler-Bernoulli beam theory (EBT) and Timoshenko beam theory (TBT) first proposed by Peddieson et al. [5] and Wang [6], respectively, were adopted by many researchers to investigate bending [7-9], buckling [10-12], and vibration [13-15] responses of carbon nanotubes. A complete development of EBT and TBT was presented by Reddy and Pang [16] who provided the analytical solutions for the deflection, buckling load, and natural frequency of nanobeams with various boundary conditions. It should be noted that the EBT is only applicable for slender beams where the shear deformation effect is negligible and leads to underestimate deflection and overestimate buckling load as well as natural frequency for short beams. The TBT accounts for the shear deformation effect for short beams by assuming a constant shear strain through the height of the beam. Therefore, a shear correction factor is required to compensate for the difference between the actual stress state and the constant stress state. To avoid the use of shear correction factor, higher-order shear deformation theories were developed based on the assumption of the higher-order variation of axial displacement through the height of the beam, notable among them are the third-order theory of Reddy [17], generalized theory of Aydogdu [18], refined theory of Thai [19], and sinusoidal shear deformation theory of Touratier [20].
The sinusoidal shear deformation theory of Touratier [20] is based on the assumption that the transverse shear stress vanishes on the top and bottom surfaces of the beam and is nonzero elsewhere. Thus there is no need to use shear correction factors as in the case of TBT. This theory is also employed to predict the response of laminate plate [21] and functionally graded sandwich plates [22-24]. The aim of this paper is to propose a nonlocal sinusoidal theory which accounts for both small scale and shear deformation effects of nanobeams. The small scale effect is taken into account by using the nonlocal constitutive relations of Eringen, while the shear deformation effect is captured using the sinusoidal shear deformation theory [20]. The nonlocal equations of motion and boundary conditions are derived using Hamilton’s principle. Analytical solutions for the deflection, buckling load, and natural frequency are presented for simply supported nanobeams, and the obtained results are compared with those predicted by the TBT to verify the accuracy of the present solution.

2. Equations of motion of the sinusoidal beam theory

Consider a beam length $L$ and rectangular cross section $b \times h$, with $b$ being the width and $h$ being the height. The $x$-, $y$-, and $z$-coordinates are taken along the length, width, and height of the beam, respectively. Equations of motion are derived using Hamilton’s principle. The principle can be stated in analytical form as [25]

$$0 = \int_0^T (\delta U + \delta V - \delta K) dt$$

(1)

where $\delta U$ is the variation of the strain energy; $\delta V$ is the variation of the potential energy; and $\delta K$ is the variation of the kinetic energy.

According to the sinusoidal theory, the displacement field is chosen based on the assumption that the transverse shear stress vanishes on the top and bottom surfaces of
the beam and is nonzero elsewhere. The displacement field is given as [20]

\[ u_1(x, z, t) = u(x, t) - \frac{z}{h} \frac{dw}{dx} + f \varphi \]
\[ u_2(x, z, t) = 0 \]
\[ u_3(x, z, t) = w(x, t) \]  

(2)

where \( f = \left( \frac{h}{\pi} \right) \sin \left( \pi z / h \right) \), \( u \) and \( w \) are the axial and transverse displacements, respectively, of a point on the midplane of the beam and \( \varphi \) is the rotation of the cross section about the y-axis. The only nonzero strains are

\[ \varepsilon_x = \frac{du}{dx} - \frac{z}{h} \frac{d^2w}{dx^2} + f \frac{d\varphi}{dx} \]  

(3a)

\[ \gamma_{xz} = \cos \left( \frac{\pi z}{h} \right) \varphi \]  

(3b)

It can be observed from Eq. (3b) that the transverse shear strain \( \gamma_{xz} \) is zero at the top \( (z = h/2) \) and bottom \( (z = -h/2) \) surfaces of the beam thus satisfying the traction free conditions for \( \sigma_{xz} \).

The variation of the strain energy of the beam can be stated as

\[ \delta U = \int_0^L \int_A \left( \sigma_x \delta \varepsilon_x + \sigma_{xz} \delta \gamma_{xz} \right) dA dx = \int_0^L \left( N \frac{d\delta u}{dx} - M \frac{d^2\delta w}{dx^2} + P \frac{d\delta \varphi}{dx} + Q \delta \varphi \right) dx \]  

(4)

where \( N, M, P, \) and \( Q \) are the stress resultants defined as

\[ (N, M, P) = \int_A (1, z, f) \sigma_x dA \quad \text{and} \quad Q = \int_A \cos(\pi z / h) \sigma_{xz} dA \]  

(5)

The variation of the potential energy of the applied loads can be expressed as

\[ \delta V = -\int_0^L q \delta w dx - \int_0^L N_0 \frac{dw}{dx} \frac{d\delta w}{dx} dx \]  

(6)

where \( q \) and \( N_0 \) are the transverse and axial loads, respectively.

The variation of the kinetic energy is obtained as
where dot-superscript convention indicates the differentiation with respect to the time variable $t$; $\rho$ is the mass density; and $(m_0, m_2)$ are mass inertias defined as
\[
(m_0, m_2) = \int_A (1, z^2) \rho dA
\]  
Substituting the expressions for $\delta U$, $\delta V$, and $\delta K$ from Eqs. (4), (6), and (7) into Eq. (1) and integrating by parts, and collecting the coefficients of $\delta u$, $\delta w$, and $\delta \phi$, the following equations of motion of the beam are obtained
\[
\delta u: \frac{dN}{dx} = m_0 \ddot{u}
\]  
\[
\delta \phi: \frac{dP}{dx} - Q = \frac{6m_2}{\pi^2} \ddot{\phi} - \frac{24m_2}{\pi^3} \dot{\phi}
\]  
\[
\delta w: \frac{d^2M}{dx^2} + q - N_0 \frac{d^2w}{dx^2} = \frac{24m_2}{\pi^3} \frac{d\phi}{dx} + m_0 \ddot{w} - m_2 \frac{d^2\dot{w}}{dx^2}
\]  
The boundary conditions of the present theory are of the form
\[
\text{specify } u \quad \text{or} \quad N
\]  
\[
\text{specify } \phi \quad \text{or} \quad P
\]  
\[
\text{specify } w \quad \text{or} \quad \frac{dM}{dx} - N_0 \frac{dw}{dx} + m_2 \frac{d\dot{w}}{dx} - \frac{24m_2}{\pi^3} \ddot{\phi}
\]  
\[
\text{specify } \frac{dw}{dx} \quad \text{or} \quad M
\]

3. Nonlocal theory

3.1. Constitutive relations

Unlike the local theory, the nonlocal theory assumes that the stress at a point depends not only on the strain at that point but also on strains at all other points of the body.
According to Eringen [26], the nonlocal stress tensor $\sigma$ at point $x$ is expressed as

$$\sigma - \mu \nabla^2 \sigma = \tau$$

(11)

where $\tau$ is classical stress tensor at a point $x$ related to the strain by the Hooke’s law; $\mu = (e_0 a)^2$ is the nonlocal parameter which incorporates the small scale effect, $a$ is the internal characteristic length and $e_0$ is a constant appropriate to each material. The nonlocal parameter depends on the boundary conditions, chirality, mode shapes, number of walls, and type of motion [27]. So far, there is no rigorous study made on estimating the value of the nonlocal parameter. It is suggested that the value of nonlocal parameter can be determined by experiment or by conducting a comparison of dispersion curves from the nonlocal continuum mechanics and molecular dynamics simulation [6, 28]. In general, a conservative estimate of the nonlocal parameter is $e_0 a < 2.0$ nm for a single wall carbon nanotube [29].

3.2. Stress resultants

For an isotropic material in a one-dimensional case, the nonlocal constitutive relation in Eq. (11) takes the following forms

$$\sigma_x - \mu \frac{d^2 \sigma_x}{dx^2} = E \varepsilon_x$$

$$\sigma_{xz} - \mu \frac{d^2 \sigma_{xz}}{dx^2} = G \gamma_{xz}$$

(12)

where $E$ and $G$ are the elastic and shear modulus of the beam, respectively. By substituting Eq. (3) into Eq. (12) and the subsequent results into Eq. (5), the stress resultants are obtained as

$$N - \mu \frac{d^2 N}{dx^2} = EA \frac{du}{dx}$$

(13a)
\[ M - \mu \frac{d^2 M}{dx^2} = -EI \frac{d^2 w}{dx^2} + \frac{24EI}{\pi^3} \frac{d\phi}{dx} \]  
(13b)

\[ P - \mu \frac{d^2 P}{dx^2} = -\frac{24EI}{\pi^3} \frac{d^2 w}{dx^2} + \frac{6EI}{\pi^3} \frac{d\phi}{dx} \]  
(13c)

\[ Q - \mu \frac{d^2 Q}{dx^2} = \frac{GA}{2} \phi \]  
(13d)

where

\[ (A, I) = \int_A (1, z^2) dA \]  
(14)

4. Equations of motion in terms of displacements

The nonlocal equations of motion of the proposed beam theory can be expressed in terms of displacements \((u, w, \phi)\) by substituting stress resultants in Eq. (13) into Eq. (9) as

\[ EA \frac{d^2 u}{dx^2} = m_0 \left( \ddot{u} - \mu \frac{d^2 \ddot{u}}{dx^2} \right) \]  
(15a)

\[ -\frac{24EI}{\pi^3} \frac{d^3 w}{dx^3} + \frac{6EI}{\pi^3} \frac{d^2 \phi}{dx^2} - \frac{GA}{2} \phi = 6m_1 \frac{\phi}{\pi^3} \left( \dot{\phi} - \mu \frac{d^2 \phi}{dx^2} \right) - \frac{24m_2}{\pi^3} \left( \frac{d\ddot{w}}{dx} - \mu \frac{d^2 \ddot{w}}{dx^2} \right) \]  
(15b)

\[ -EI \frac{d^4 w}{dx^4} + \frac{24EI}{\pi^3} \frac{d^3 \phi}{dx^3} + q - \mu \frac{d^2 q}{dx^2} - N_0 \left( \frac{d^2 w}{dx^2} - \mu \frac{d^4 w}{dx^4} \right) \]  
(15c)

\[ = \frac{24m_3}{\pi^3} \left( \frac{d\ddot{\phi}}{dx} - \mu \frac{d^2 \ddot{\phi}}{dx^2} \right) + m_0 \left( \dddot{w} - \mu \frac{d^2 \dddot{w}}{dx^2} \right) - m_2 \left( \frac{d^2 \dddot{w}}{dx^2} - \mu \frac{d^4 \dddot{w}}{dx^4} \right) \]

The equations of motion of local beam theory can be recovered from Eq. (15) by setting the nonlocal parameter \(\mu\) equal to zero. Also, the equations of motion of the nonlocal EBT can be obtained from Eq. (15) by setting the rotation \(\phi\) equal to zero. It can be seen from Eq. (15) that the axial deformation \(u\) is uncoupled from transverse deformations \((\phi, w)\). Thus, the equations of motion for the transverse response of the
beam are reduced to Eqs. (15b) and (15c).

5. **Analytical solution of simply supported beams**

Consider a simply supported beam with length \( L \) subjected to transverse load \( q \) and axial load \( N_0 \). The simply supported boundary conditions of the beam are

\[
w = M = 0 \quad \text{at} \quad x = 0, L
\]  
(16)

The following expansions of the generalized displacements \( (\varphi, w) \) satisfy the boundary conditions in Eq. (16)

\[
\varphi(x,t) = \sum_{n=1}^{\infty} \phi_n e^{i\omega t} \cos \alpha x
\]

\[
w(x,t) = \sum_{n=1}^{\infty} W_n e^{i\omega t} \sin \alpha x
\]  
(17)

where \( i = \sqrt{-1} \), \( \alpha = n\pi / L \), \( (\phi_n, W_n) \) are coefficients, and \( \omega \) is the natural frequency.

The transverse load \( q \) is also expanded in the Fourier sine series as

\[
q(x) = \sum_{n=1}^{\infty} Q_n \sin \alpha x
\]  
(18)

where

\[
Q_n = \frac{2}{L} \int_0^L q(x) \sin \alpha x dx,
\]

\[
= \begin{cases} 
q_0 (n = 1) & \text{for sinusoidal load } q_0 \\
\frac{4q_0}{n\pi} (n = 1, 3, 5, ...) & \text{for uniform load } q_0 \\
\frac{2}{L} Q_0 \sin \frac{n\pi}{2} (n = 1, 2, 3, ...) & \text{for point load } Q_0 \text{ at the center}
\end{cases}
\]  
(19)

Substituting the expansions of \( \varphi \), \( w \), and \( q \) from Eqs. (17) and (18) into the equations of motion Eq. (15), the closed-form solutions can be obtained from the following equations

\[
\begin{bmatrix} s_{11} & s_{12} \\ s_{21} & s_{22} - k \end{bmatrix} - \lambda \omega^2 \begin{bmatrix} m_{11} & m_{12} \\ m_{12} & m_{22} \end{bmatrix} \begin{bmatrix} \phi_n \\ W_n \end{bmatrix} = \begin{bmatrix} 0 \\ \lambda Q_n \end{bmatrix}
\]  
(20)
where

\[
\begin{align*}
    s_{11} &= \frac{GA}{2} + \frac{6EI}{\pi^2} \alpha^2, \quad s_{12} = -\frac{24EI}{\pi^3} \alpha^3, \quad s_{22} = EI\alpha^4 \\
    m_{11} &= \frac{6m_2}{\pi^2}, \quad m_{12} = -\frac{24m_2}{\pi^3}, \quad m_{22} = m_0 + m_2\alpha^2, \quad \lambda = 1 + \mu\alpha^2, \quad k = \lambda N_0\alpha^2
\end{align*}
\]

The static deflection is obtained from Eq. (20) by setting \( N_0 \) and \( \omega \) equal to zero

\[
w(x) = \sum_{n=1}^{\infty} \frac{\lambda Q_n}{s_{22} - s_{12}^2/s_{11}} \sin \alpha x
\]

The buckling load is obtained from Eq. (20) by setting \( q \) and \( \omega \) equal to zero

\[
N_0 = \frac{s_{22} - s_{12}^2/s_{11}}{\lambda \alpha^2}
\]

By setting \( q \) and \( N_0 \) in Eq. (20) equal to zero, the natural frequency can be obtained from the following equation

\[
\left(m_{11}m_{22} - m_{12}^2\right)\lambda^2\omega^4 - (m_{11}s_{22} + m_{22}s_{11} - 2s_{12}m_{12})\lambda\omega^2 + (s_{11}s_{22} - s_{12}^2) = 0
\]

The closed-form solutions of the EBT can be obtained from Eq. (20) by setting \( \phi_n \) equal to zero. Thus, the deflection \( w \), buckling load \( N_0 \), and frequency \( \omega \) of the EBT are expressed as

\[
w(x) = \sum_{n=1}^{\infty} \frac{\lambda Q_n}{s_{22}} \sin \alpha x = \sum_{n=1}^{\infty} \frac{\lambda Q_n}{EI\alpha^2} \sin \alpha x
\]

\[
N_0 = \frac{s_{22}}{\lambda \alpha^2} = \frac{EI\alpha^2}{\lambda}
\]

\[
\omega = \sqrt{\frac{s_{22}}{\lambda m_{22}}} = \sqrt{\frac{EI\alpha^4}{\lambda \left(m_0 + m_2\alpha^2\right)}}
\]

6. Results and discussions

Table 1 shows the nondimensional deflection, critical buckling load, and fundamental
frequency of a simply supported beam subjected to uniform load. The obtained results are compared with those reported by Thai [19] based on nonlocal TBT for a wide range of small scale coefficient and length-to-depth ratio \( L/h \). Shear correction factor and Poisson’s ratio are taken as 5/6 and 0.3, respectively. The side of nanobeam \( L \) is assumed to be 10 nm. Nondimensional deflection, buckling load, and natural frequency are defined as

\[
\bar{w} = \frac{100EI}{q_0L^4}, \quad \bar{N} = \frac{N_{cr}L^3}{EI}, \quad \bar{\omega} = \omega L^2 \sqrt{\frac{m_0}{EI}} \quad (28)
\]

It can be seen that the results of present theory are in excellent agreement with those predicted by TBT for all values of small scale coefficient and length-to-depth ratio even for short beams at the higher vibration modes where the effects of transverse shear deformation and rotary inertia are significant (see Table 2). It is worth noting that the TBT requires a shear correction factor to satisfy the free transverse shear stress conditions on the top and bottom surfaces of the beam, whereas the present theory satisfies the free transverse shear stress conditions on the top and bottom surfaces of the beam without using any shear correction factors.

The effect of shear deformation on the bending, buckling, and vibration responses of nanobeams is shown in Fig. 1 for a simply supported beam with \( c_{\text{eff}} = 1 \text{nm} \). In this figure, the deflection, buckling load, and frequency ratios are defined as the ratios of those by present theory to the correspondences by EBT where the shear deformation effect is omitted. It is observed that, the effect of shear deformation is to increase the deflections and decrease the buckling loads and natural frequencies, and this effect is significant for short beams at higher vibration modes (see Fig. 2). This indicates that the shear deformation effect results in a reduction of the beam stiffness. Another illustration
of the shear deformation effect is also shown in Table 3 for a simply supported single-walled carbon nanotube (SWCNT) with diameter \( d = 0.678 \text{ nm} \) and effective tube thickness \( t = 0.066 \text{ nm} \) [30]. The material properties of SWCNT are [13]: \( E = 5.5 \text{ TPa} \), \( \nu = 0.19 \), and \( \rho = 2.3 \text{ g/cm}^3 \). The results of present theory obtained from Eq. (24) are compared with those of EBT from Eq. (27) for various values of length-to-diameter ratio \( L/d \). It can be seen from Table 3 that the frequencies obtained by present theory are smaller than those given by EBT. The difference between present theory and EBT is significant for higher modes and small \( L/d \). For example, for the fundamental mode \( n = 1 \), the differences are only 5.23\% and 0.45\% for short tube \( (L/d = 5) \) and long tube \( (L/d = 20) \), respectively. However, for the higher mode, say \( n = 5 \), the differences are 43.20\% and 7.77\% for short tube \( (L/d = 5) \) and long tube \( (L/d = 20) \), respectively. So the present beam should be used to predict the responses of short beams at higher modes where the shear deformation effect is significant.

To illustrate the small scale effect on the responses of nanobeams, Fig. 3 plots the deflection, buckling load, and frequency ratios with respect to the small scale coefficient \( e_0 a \) for a simply supported beam with \( L/h = 10 \). It is noted that the values of \( e_0 a \) should be smaller than 2.0 nm for a SWCNT as pointed out by Wang and Wang [29]. The deflection, buckling load, and frequency ratios are defined as the ratios of those by the nonlocal theory to the correspondences by the local theory (i.e., \( e_0 a = 0 \)). It can be seen that the deflection ratio is greater than unity, whereas the buckling load and frequency ratios are smaller than unity. It means that the local theory underestimates the deflections and overestimates the buckling loads as well as natural frequencies of the nanobeams compared to the nonlocal one. This is due to the fact that the local theory is unable to capture the small scale effect of the nanobeams. The difference between the
local and nonlocal theories is especially significant for the higher modes (see Fig. 4). In general, the effects of small scale and shear deformation are similar, and the inclusion of the small scale and shear deformation effects will reduce the stiffness of the beam, and consequently, leads to an increase in the deflections and a reduction of the buckling loads and natural frequencies.

7. Conclusions

A nonlocal sinusoidal beam theory is developed for the bending, buckling, and vibration of nanobeams. The present model is capable of capturing both small scale and shear deformation effects of nanobeams, and does not require shear correction factors. Based on the nonlocal differential constitutive relations of Eringen, the equations of motion as well as the boundary conditions of the beam are derived using Hamilton’s principle. Analytical solutions for deflection, buckling load, and natural frequency are presented for a simply supported beam, and the obtained results are compared well with those predicted by the TBT. It is observed that the inclusion of the small scale and shear deformation effects lead to an increase in the deflections and a reduction of the buckling loads and natural frequencies of nanobeams.

References


Fig. 1. Effect of transverse shear deformation on the deflection, critical buckling load, and fundamental frequency ratios for a simply supported beam with $e_0a = 1\text{ nm}$

Fig. 2. Effect of transverse shear deformation on higher frequency ratios for a simply supported beam with $e_0a = 1\text{ nm}$
Fig. 3. Effect of small scale on the deflection, critical buckling load, and fundamental frequency ratios for a simply supported beam with $L/h = 10$

Fig. 4. Effect of small scale on higher frequency ratios for a simply supported beam with $L/h = 10$
Table 1. Nondimensional deflection $\bar{w}$, critical buckling load $\bar{N}$, and fundamental frequency $\bar{\omega}$ of simply supported beams

<table>
<thead>
<tr>
<th>L/h</th>
<th>$\mu , (nm^2)$</th>
<th>Deflection $\bar{w}$</th>
<th>Buckling load $\bar{N}$</th>
<th>Frequency $\bar{\omega}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TBT [19]</td>
<td>Present</td>
<td>TBT [19]</td>
<td>Present</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td>1.4321</td>
<td>8.9509</td>
<td>9.2740</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>1.5674</td>
<td>8.1468</td>
<td>8.8477</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1.7028</td>
<td>7.4753</td>
<td>8.4752</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1.8381</td>
<td>6.9061</td>
<td>8.1461</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1.9734</td>
<td>6.4174</td>
<td>7.8526</td>
</tr>
<tr>
<td>10</td>
<td>0</td>
<td>1.3346</td>
<td>9.6227</td>
<td>9.7075</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>1.4622</td>
<td>8.7583</td>
<td>9.2612</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1.5898</td>
<td>8.0364</td>
<td>8.8713</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1.7173</td>
<td>7.4244</td>
<td>8.5269</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1.8449</td>
<td>6.8990</td>
<td>8.2196</td>
</tr>
<tr>
<td>20</td>
<td>0</td>
<td>1.3102</td>
<td>9.8067</td>
<td>9.8281</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>1.4359</td>
<td>8.9258</td>
<td>9.3763</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1.5615</td>
<td>8.1900</td>
<td>8.9816</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1.6871</td>
<td>7.5664</td>
<td>8.6328</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1.8128</td>
<td>7.0310</td>
<td>8.3218</td>
</tr>
<tr>
<td>100</td>
<td>0</td>
<td>1.3024</td>
<td>9.8671</td>
<td>9.8679</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>1.4274</td>
<td>8.9807</td>
<td>9.4143</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1.5525</td>
<td>8.2405</td>
<td>9.0180</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1.6775</td>
<td>7.6130</td>
<td>8.6678</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1.8025</td>
<td>7.0743</td>
<td>8.3555</td>
</tr>
<tr>
<td>L/h</td>
<td>μ (nm²)</td>
<td>( \bar{\omega}_1 )</td>
<td>( \bar{\omega}_2 )</td>
<td>( \bar{\omega}_3 )</td>
</tr>
<tr>
<td>-----</td>
<td>--------</td>
<td>----------------</td>
<td>----------------</td>
<td>----------------</td>
</tr>
<tr>
<td></td>
<td>Present</td>
<td>TBT [19]</td>
<td>Present</td>
<td>TBT [19]</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td>9.2740</td>
<td>9.2752</td>
<td>32.1665</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>8.8477</td>
<td>8.8488</td>
<td>27.2364</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>8.4752</td>
<td>8.4763</td>
<td>24.0453</td>
</tr>
<tr>
<td>10</td>
<td>0</td>
<td>9.7075</td>
<td>9.7077</td>
<td>37.0962</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>9.2612</td>
<td>9.2614</td>
<td>31.4105</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>8.8713</td>
<td>8.8715</td>
<td>27.7303</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>8.5269</td>
<td>8.5271</td>
<td>25.0996</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>8.2196</td>
<td>8.2198</td>
<td>23.0989</td>
</tr>
<tr>
<td>20</td>
<td>0</td>
<td>9.8281</td>
<td>9.8282</td>
<td>38.8299</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>9.3763</td>
<td>9.3764</td>
<td>32.8786</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>8.9816</td>
<td>8.9816</td>
<td>29.0263</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>8.6328</td>
<td>8.6329</td>
<td>26.2727</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>8.3218</td>
<td>8.3218</td>
<td>24.1785</td>
</tr>
<tr>
<td>100</td>
<td>0</td>
<td>9.8679</td>
<td>9.8679</td>
<td>39.4517</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>9.4143</td>
<td>9.4143</td>
<td>33.4051</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>8.6678</td>
<td>8.6678</td>
<td>26.6934</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>8.3555</td>
<td>8.3555</td>
<td>24.5657</td>
</tr>
</tbody>
</table>

Table 3. First five frequencies (THz) of a simply supported single-walled carbon nanotube with \( e_i a = 1.0 \) nm

<table>
<thead>
<tr>
<th>Modes</th>
<th>( L/d = 5 )</th>
<th>( L/d = 10 )</th>
<th>( L/d = 20 )</th>
<th>( L/d = 100 )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>EBT</td>
<td>Present</td>
<td>EBT</td>
<td>Present</td>
</tr>
<tr>
<td>1</td>
<td>1.0459</td>
<td>0.9939</td>
<td>0.3283</td>
<td>0.3237</td>
</tr>
<tr>
<td>2</td>
<td>2.5622</td>
<td>2.2006</td>
<td>1.0459</td>
<td>0.9939</td>
</tr>
<tr>
<td>3</td>
<td>3.7911</td>
<td>2.9689</td>
<td>1.8294</td>
<td>1.6550</td>
</tr>
<tr>
<td>4</td>
<td>4.7180</td>
<td>3.4516</td>
<td>2.5622</td>
<td>2.2006</td>
</tr>
<tr>
<td>5</td>
<td>5.4019</td>
<td>3.7723</td>
<td>3.2171</td>
<td>2.6310</td>
</tr>
</tbody>
</table>