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Effects of Contact between Rough Surfaces on the Dynamic Responses of Bolted Composite Joints: Multiscale Modeling and Numerical Simulation

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Abstract: A multiscale numerical model, considering both microscopic interfacial properties and macroscopic composite properties, has been developed to model the dynamic responses of bolted composite structures using the elastic-viscoelastic correspondence principle. The complex contact modulus of the uneven interfaces of the joints, featuring multi-asperity contact at the micro perspective, was derived using the fractal contact theory. The frequency-dependent complex moduli of composite materials were ascertained through modal impact tests on unidirectional composites. The damping and stiffness of the composite specimens were solved within ABAQUS software using the complex eigenvalue method. The damping ratios of the joints decreased by over 8.5% while the resonant frequencies increased by more than 0.59% when the bolts of the joints were adjusted from fully loose to fully tightened. Results from the numerical prediction and the experiment were found to be in good agreement, whereby the influence of the interfacial contact conditions on the dynamic responses of the bolted composite structures was revealed through the multiscale analysis.

Keywords: fractal analysis; rough interface; bolted joint; damping ratio; numerical analysis

1. Introduction

Advanced composite materials have been widely adopted in various industrial fields for weight critical applications such as aerospace, automobile, and marine engineering, due to their advantageous

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properties typified by high strength-weight ratio and high stiffness-to-weight ratio [1]. The preferred method of assembling composite structures is to use bolted joints, in order to streamline manufacturing and facilitate system maintenance [2-4]. However, lightweight structures, including bolted composite joints, are prone to vibration, which may result in unexpected instability and can reduce structural integrity [4, 5]. Therefore damping capability plays a critical role in determining the long-term performance of a bolted composite structure subject to vibration [3, 6]. The authors' previous work revealed that the damping ratios of bolted composite joints increased when the joints were subject to vibration fatigue [3]. Therefore, it is of vital importance to develop an accurate model to predict the dynamic properties of bolted composite structures, including the damping ratio and the resonant frequency [7].

With the development of numerical techniques, the static and dynamic characteristics of composite structures have been successfully predicted using the numerical method. However, three main challenges still exist that must be solved prior to achieving an accurate prediction of the modal parameters of bolted composite structures: (1) theoretical derivation of the stiffness and damping properties for rough contact surfaces; (2) implementation of numerical modeling for the contact properties of rough interfaces; (3) implementation of numerical modelling of the frequency-dependent damping properties for the composite materials.

Rough surfaces that are a result of manufacturing tolerances usually lead to complex energy dissipation behaviors of a bolted joint [8, 9]. As a result, understanding the contact mechanisms at the rough surfaces is still a challenging problem, and inaccurate modeling of this may cause unacceptable errors during the dynamic analysis of bolted composite structures. Without loss of generality, the joints represent discontinuities in a bolted structure and may lead to high-stress concentrations and further structural failure. It has been reported that for a bolted metallic structure, up to 90% of the structural damping is generated by the joint interfaces [10], which dictate around 50% of the system stiffness [11, 12]. Therefore, the basic challenge for accurately predicting the interfacial energy dissipation and the structural stiffness of a bolted joint is to develop an efficient model that is capable of describing surface interactions within the bolted structure[7]. Fractal theory can characterize rough surfaces without dependence on the resolution and sampling length of the measuring instruments. Majumdar and Bhushan's (*MB*) fractal theory [13] has been widely adopted to derive the contact stiffness and damping of the rough interfaces of joints. For instance, Wang *et al.* [14] characterized contact surfaces as fractals and used fractal geometry to analyze temperature rises at the microcontacts in a slow sliding region. Zhang *et al.* [15] proposed a modified *MB* fractal model to investigate the mechanisms of energy dissipation at joint interfaces and further derived the tangential stiffness and damping loss factor of the joint interfaces. Zhao[12] established a modified three-dimensional fractal model to analyze the stiffness and damping of rough contact surfaces for a bolted metallic joint.

Three predominant methods are currently used to model the contact properties of rough interfaces in finite element analysis [16], namely node-to-node contact, thin-layer elements, and zero-thickness elements are currently adopted to address this issue. Of these methods, the virtual material model, using thin-layer elements can be easily integrated with commercial finite element software including ABAQUS [11, 17, 18]. For instance, Tian *et al.* [11] conducted virtual material-based numerical modeling of fixed joint interfaces and successfully improved the modeling accuracy of a machine tool that incorporated joints. Two approaches have been used to model damping properties of composite materials, namely the strain energy approach and the elastic-viscoelastic correspondence principle[19]. However, the frequency dependence or the viscoelastic properties of the composite materials are usually ignored in the former method. The authors have previously adopted the latter method in their research [20] to provide an efficient prediction of the modal damping properties of anisotropic materials using ABAQUS code.

In this study, a multi-scale numerical model developed in the ABAQUS software, consisting of composite materials, thin-layer interfaces (virtual materials), and fasteners has been proposed in order to quantitatively investigate the influence of microscale surface contact on the dynamic structural responses of a bolted composite joint using the elastic-viscoelastic correspondence principle. The research flowchart is shown in **Figure 1**. The complex modulus of the rough contact surfaces of the joint has been quantitatively linked to the contact pressure, by using the fractal contact theory in conjunction with microscopic observation. The frequency-dependent complex modulus of the composite materials was ascertained from modal tests on unidirectional composites. Subsequently, the damping and stiffness of the bolted composite structures were modeled by using the complex modulus of both the composite materials and the rough interfaces. The dynamic responses of the bolted composite joints with different leftover torque values were solved using the complex eigenvalue method, whereby a quantitative relationship between the modal parameters of the joints and the applied torque was built. Modal impact tests were conducted on cross-ply composite beams and bolted composite joints subject to different torques, to verify the efficiency of the proposed numerical analysis. The results calculated from numerical models with rough interfaces and perfect interfaces respectively were further compared in order to reveal the influence of rough interfacial contact on the dynamic responses of bolted composite structures.

2. Theoretical Analysis of the Contact Stiffness and Damping of Rough Surfaces

The surfaces of solids are usually uneven in the micro perspective and have randomly distributed asperities, as shown in **Figure 2**. Therefore, two contacting rough surfaces can be simplified as spherical asperities with different radii in contact. In the fractal contact model, asperity contacts with different contact areas are hypothesized to undergo different degrees of deformation[13]. Large contacting asperities form contact areas (a) that exceed the critical area (a_c) and therefore undergo elastic deformation. While asperities that have contact areas that are less than the critical area (a_c) are in plastic deformation. For asperities that undergo elastic deformation, the reliance of the contact load (p_e) on the truncated contact area ($a^{'}$) can be expressed as [15]:

$$p_e = 4/\left[3(2\pi)^{0.5}\right] E^{'} G^{D-1} a^{'(3-D)/2} \tag{1}$$

where D and G denote the fractal dimension and roughness, respectively. $E^{'}$ is the equivalent Young's modulus of two contact surfaces possessing Young's modulus (E_1 and E_2) and Poisson's ratio (ν_1 and ν_2):

$$1/E^{'}=(1-\nu_1^2)/E_1+(1-\nu_2^2)/E_2 \tag{2}$$

The relation between truncated contact area ($a^{'}$) and the contact area (a) in **Figure 3 (b)** can be expressed as [11]

$$a^{'}=\pi r^{'2}=2a=2\pi r^2 \tag{3}$$

The critical truncated area which distinguishes between elastic and plastic deformation of the asperities can be written as [15]:

$$a_c^{'}=2G^2/\left[(0.5HE_1\pi)^{2/(D-1)}\right] \tag{4}$$

where H is the hardness of the softer material of the two in contact.

According to Hertzian contact theory and micro slip theory, the equivalent normal (e_e) and shear modulus (g_e) of single-asperity contact in elastic deformation can be expressed as[11]

$$e_e=4/(3\pi)E^{'}\sqrt{R/\delta};\ g_e=16/\pi[1-t/(up)]^{1/3}G^{'} \tag{5}$$

Here, p and t are the normal and tangential loads carried by a single-asperity contact. R is the radius of the asperity and δ denotes the asperity interference (see **Figure 3(b)**)[10].

$$R=a^{'D/2}/(2\pi G^{D-1});\ \delta=G^{D-1}a^{'(D-1)/2} \tag{6}$$

$G^{'}$ is the equivalent shear modulus of the two rough surfaces with shear modulus G_1 and G_2, respectively, as:

$$1/G^{'}=(2-\nu_1)/G_1+(2-\nu_2)/G_2 \tag{7}$$

While for asperities in plastic deformation, the contact load p_p and the contact modulus (Young's modulus e_p, and shear modulus g_p) can be expressed with the truncated area as [11] :

$$p_p=0.5Ha^{'};e_p=g_p=0 \tag{8}$$

The size distribution of the truncated area $n(a^{'})$ of the asperity contacts takes the following form [14]

$$n(a') = 0.5D\psi^{(2-D)/2} a_l'^{D/2} a'^{-(D+2)/2} \quad (0 < a' < a_l') \tag{9}$$

Here a_l' represents the maximum truncated elastic area. ψ denotes the domain extension factor and can be expressed with fractal dimension D as :

$$\psi^{(2-D)/2} - (1+\psi^{-D/2})^{(D-2)/D} = (2-D)/D \tag{10}$$

In the above analysis, the contact properties of the asperities were expressed with the truncated contact area, and the area distribution of a contact interface can be characterized with fractal parameters (i.e., D and G). Then the normal contact load P of the whole interface, with multi-asperity contacts, can be obtained by using the truncated area a' as an integral parameter to calculate the contact loads carried by single-asperity contacts in both elastic and plastic deformation as

$$\begin{aligned} P &= \int_{a_c'}^{a_l'} p_e n(a')\,\mathrm{d}a' + \int_{a_s'}^{a_c'} p_p n(a')\,\mathrm{d}a' \\ &= \begin{cases} 2^{0.25}/\left(\sqrt{\pi}\right) EG^{0.5}\psi^{0.25} a_l^{0.75} \ln(a_l/a_c) + 6H\psi^{0.25} a_l^{0.75} a_c^{0.25} & (\mathrm{D}=1.5) \\ \psi^{1-0.5D}\left[\dfrac{2^{3-0.5D} EDG^{(\mathrm{D}-1)}}{3(3-2D)\sqrt{\pi}} a_l^{0.5D}(a_l^{1.5-D} - a_c^{1.5-D}) + \dfrac{2DH}{2-D} a_l^{0.5D} a_c^{1-0.5D}\right] & (\mathrm{D}\neq 1.5) \end{cases} \end{aligned} \tag{11}$$

Young's modulus E_n and the shear modulus G_t of the whole interface can be obtained as

$$E_n = \int_{a_c'}^{a_l'} [0.5 e_e n(a') a']\,\mathrm{d}a' / A_0 = 3D\psi^{1-0.5D}/(2A_0\pi^2) EG^{1-D} a_l^{D/2} [a_l^{0.5} - a_c^{0.5}] \tag{12}$$

$$\begin{aligned} G_t &= \int_{a_c'}^{a_l'} [0.5 g_e n(a') a']\,\mathrm{d}a' / A_0 \\ &= 16 a_l^{0.5D} (a_c^{1-0.5D} - a_l^{1-0.5D}) \psi^{1-0.5D} / \left[A_0 \pi (D-2)\right][1 - F_t/(\mu P)]^{1/3} G' D \end{aligned} \tag{13}$$

F_t is the tangential load carried by single-asperity contacts. The relation between the loads (i.e., F_t and P) carried by the whole interface with multi-asperity contacts, and the loads (i.e., t and p) carried by single-asperity contacts can be written as[15]

$$F_t / P = t / p \tag{14}$$

The equivalent Poisson's ratio of the rough interface can be calculated from[11]

$$v_e = E_n / (2G_t) - 1 \tag{15}$$

When a single-asperity contact is subject to both normal and tangential loads, both the adherence region and the microslip region exist at the whole contact surface in the case that the tangential load is less than the limiting friction. It has been reported that microslip in the tangential direction is the main mechanism of energy dissipation for rough contact interfaces[11, 15, 17]. The dissipated energy w_d arising from the microslip, and the input energy w induced by the tangential force can be expressed as[15]

$$w_d = 9\mu^2 p^2 / (10rG')\left\{1-[1-t/(\mu p)]^{5/3} - 5t/(6\mu p)\left[1+(1-t/(\mu p))^{2/3}\right]\right\} \tag{16}$$

$$w = 3\pi\mu^2 a^2 p^2 / (8A_r^2 G' r)\left\{3/5+2/5[1-t/(\mu p)]^{5/3} - [1-t/(\mu p)]^{2/3}\right\} \tag{17}$$

Using of fractal theory, the dissipated energy W_d and the input energy W of the whole interface can be integrated as:

$$\begin{aligned} W_d &= \int_{a_c'}^{a_l'} w_d n(a') da' \\ &= \frac{9(2\pi)^{\frac{1}{2}}\mu^2 P^2}{10G'} \frac{(2-D)^2 a_l'^{(D-4)/2}(a_l'^{(3-D)/2} - a_c'^{(D-4)/2})}{\psi^{(2-D)/2} D(3-D)} \left\{1-(1-\frac{F_t}{\mu P})^{\frac{5}{3}} - \frac{5F_t}{6\mu P}\left[1+(1-\frac{F_t}{\mu P})^{\frac{2}{3}}\right]\right\} \end{aligned} \tag{18}$$

$$\begin{aligned} W &= \int_{a_c'}^{a_l'} w n(a') da' \\ &= \frac{(2-D)^2 a_l'^{(D-4)/2}(a_c'^{(3-D)/2} - a_l'^{(3-D)/2})}{\psi^{(2-D)/2} D(3-D)} 2\pi \frac{3(2\pi)^{1/2}\mu^2 P^2}{16G'} \left\{\frac{3}{5} + \frac{2}{5}(1-\frac{F_t}{\mu P})^{\frac{5}{3}} - (1-\frac{F_t}{\mu P})^{\frac{2}{3}}\right\} \end{aligned} \tag{19}$$

The damping loss factor η_c of the interface can be expressed with the dissipated energy W_d and input energy W during the microslip as

$$\eta_c = \frac{1}{2\pi}\frac{W_d}{W} = \left\{1-(1-\frac{F_t}{\mu P})^{\frac{5}{3}} - \frac{5F_t}{6\mu P}\left[1+(1-\frac{F_t}{\mu P})^{\frac{2}{3}}\right]\right\} / \left[\frac{3}{5}+\frac{2}{5}(1-\frac{F_t}{\mu P})^{\frac{5}{3}} - (1-\frac{F_t}{\mu P})^{\frac{2}{3}}\right] \tag{20}$$

The elastic-viscoelastic correspondence principle was subsequently used to describe the stiffness (real part of the complex modulus) and the damping properties (imaginary part of the complex modulus) of the rough interface. The complex Young's modulus E_n^* and the shear modulus G_t^* of the rough interface are written as

$$E_n^* = E_n; \quad G_t^* = G_t + i\eta_c G_t \tag{21}$$

Then the orthotropic constitutive matrix ($[S_{Interface}^*]$) of the rough interface can be written in the following form (interfacial stiffness matrix can be obtained as $[K_{Interface}^*] = [S_{Interface}^*]^{'}$) as:

$$\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \\ \varepsilon_{xy} \\ \varepsilon_{yz} \\ \varepsilon_{zx} \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & & & \\ & S_{22} & S_{23} & & & \\ & & S_{33} & & & \\ & & & S_{44} & & \\ & & & & S_{55} & \\ & & & & & S_{66} \end{bmatrix} \begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \sigma_{xy} \\ \sigma_{yz} \\ \sigma_{zx} \end{Bmatrix} \tag{22}$$

Here, S_{33} represents the normal flexibility, while S_{55} and S_{66} define the tangential flexibility of the rough interface. The off-diagonal terms in the matrix are zero because there is no transversal contraction or in-plane shearing at the interface [16].

$$S_{33} = 1/E_n^*; \quad S_{55} = S_{66} = 1/G_t^* \tag{23}$$

3. Numerical Analysis of the Dynamic Responses of Bolted Composite Joints

A multi-scale numerical model, as shown in **Figure 4**, was developed in the commercial numerical software ABAQUS to quantitatively investigate the effect of rough surface contact on the dynamic responses of a bolted composite joint, based on the fractal analysis of interfacial contact stiffness of rough surfaces and modal analysis of composite materials.

3.1. Microscopic Analysis of the Complex Modulus of Rough Surfaces

The surface roughness profile of the composite material was measured using a scanning electron microscope (*SEM*) in order to obtain the interfacial properties of rough contact surfaces in bolted composite joints, as depicted in **Figure 2**. The results obtained from the metallographic tests (*MGT*) showed that a thin resin layer (approximately $5\mu m$), with randomly distributed asperities, covered the surface of the composite. The fractal dimension (D) and the fractal roughness parameter

(G) were subsequently ascertained as D = 1.22, G = 9.9852×10^{-9} through power spectrum analysis[12] of the resin surfaces.

Two types of contact interfaces, as shown in **Figure 4**, were considered in the numerical model, namely the interface between the composite material and the metallic fastener (i.e., the *M-C* interface), along with the interface between the composites (i.e., the *C-C* interface). The interfacial properties (the normal storage modulus E_n, the tangential storage modulus G_t, and tangential loss factor η_c) of the two interfaces were calculated using fractal contact analysis under different normal pressures P, induced by the applied torque T (subject to $P = T/(\tau d)$, where d and τ are the bolt diameter and the friction coefficient between the nut and bolt, respectively), as displayed in **Figure 5**.

It should be noted that the damping loss factor of the interfaces is independent of the excitation frequencies[10]. The material properties of the composite and the steel fastener that were utilized for the simulation are shown in **Table 1**. In order to consider the thread friction between the bolt and nut and simplify the analysis, the damping loss factor η_f, describing the energy dissipation between the fasteners, was assumed to be proportional to the applied torque. The equivalent loss factor η, a sum of η_c and η_f ($\eta = \eta_c + \eta_f$), was used to describe the interfacial energy dissipation of the joint.

3.2. Macroscopic Analysis of the Frequency-dependent Complex Modulus of Composites

The storage moduli of the composites were measured in static tensile tests on the 0°, 45°, and 90° unidirectional composites. The reliance of the loss modulus of the composites on the frequency was determined with the damping ratios (ξ) of the above unidirectional composites of different lengths, as displayed in **Figure 6**. It was observed that the damping ratios of the 45° and 90° unidirectional composites, which were dominated by the resin, were much higher than that of the 0° unidirectional composite, which was dominated by the fiber. The frequency-dependent complex modulus of the composites consisting of storage and loss moduli, given as $\eta = \xi / 2$ [21], can be written as:

$$E_{11}^{*} = E_{11} + E_{11}^{''} = E_{11}(1+\eta_0 i) \tag{24}$$

$$E_{22}^{*} = E_{22} + E_{22}^{"} = E_{22}(1+\eta_{90} i) \tag{25}$$

$$G_{12}^{*} = G_{12} + G_{12}^{"} = G_{12}(1+\eta_{45} i) \tag{26}$$

Using the classical lamination theory, in conjunction with the elastic-viscoelastic correspondence principle, the complex stiffness matrix ($[K_{Composite}^{*}]$) of laminated composites of different geometric sizes can be obtained.

3.3. Multiscale Modeling of Joint Specimens

The equation motion of a bolted composite joint with the inertia matrix($[M_{Interface}]$ and $[M_{Composite}]$) and complex stiffness matrix ($[K_{Interface}^{*}]$and $[K_{Composite}^{*}]$) of rough interfaces and composite materials can be written as

$$([M_{Composite}]+[M_{Interface}])\{\ddot{x}\}+([K_{Composite}^{*}]+[K_{Interface}^{*}])\{x\}=\{F\} \tag{27}$$

Using the complex eigenvalue method, the resonant frequency (ω) and the loss factor (η) of the composite joint can be determined as

$$([M_{Composite}]+[M_{Interface}])\omega^{*2}+([K_{Composite}]+[K_{Interface}])=0 \tag{28}$$

$$\omega = \mathrm{Re}(\omega^{*});\ \eta = \mathrm{Im}(\omega^{*2})/\mathrm{Re}(\omega^{*2}) \tag{29}$$

For the present study, a bolted composite specimen (see **Figure 2**) was simulated as a 1/2 scale model using eight-node solid elements (C3D8). A schematic representation of the finite-element meshes and boundary conditions is shown in **Figure 4**. The specimen analyzed included two laminated beams (joint materials), thin-layer virtual interfaces and the metallic fasteners. The left end of the joint was fixed in place to form a cantilever beam. Two types of interface were adopted in the numerical model, namely a composite-composite (*C-C*) interface and a metal-composite (*M-C*) interface. A tie was used in the ABAQUS model for the contact interfaces. An aspect ratio of 100 was selected for the mesh of the virtual material.

The thin-layer virtual material, which possessed both a normal and tangential complex modulus and was defined by a UMAT subroutine for ABAQUS Standard, was used to simulate contact conditions of the *M-C* and *C-C* interfaces. When modeling the joint with different residual torques remained on the bolt, the complex normal and tangential moduli of the interfaces were adjusted according to the results shown in **Figure 5**. The complex stiffness matrix of the composite materials was defined using the frequency-dependent complex modulus measured from the experimental data as shown in **Figure 6**. When the complex stiffness matrix of the virtual material interfaces and composite materials had been defined, the complex frequency analysis implemented in ABAQUS, was adopted to calculate the resonant frequency and damping ratio of the bolted composite joint subject to different applied torques. The displacement decay curves of the joint, when subjected to an impact force, were calculated using modal dynamic analysis.

4. Experimental Validation

In order to validate the numerical analysis with regards with the dynamic responses of the bolted composite joints with distinct leftover torque values, the vibration modal parameters including both the resonant frequency and the damping ratio were comparatively measured from modal impact tests carried out on the joints. The beam specimens cut from sheets of laminated T700/7901 carbon-fiber-reinforced epoxy, which were obtained using hot pressing with a stacking sequence $[90, 0]_{4s}$. The specimens were 2 mm thick and 30 mm wide. Two composite beams with lengths of 100 mm and 130 mm, called C100 and C130 beams, were prepared for modal tests on the composite materials. Composite beams with a length of 50 mm were separately assembled with two beams with respective lengths of 70 and 100 mm, using an M6 bolt with a lap length of 20 mm, to form two bolted composite joints, termed as J100 and J130 joints. It should be noted that the total lengths of the C100 beam and J100 joint were the same and this was also true for the C130 beam and J130 joint.

The setup for the modal impact test is shown in **Figure 2**, which consisted of test specimens, fixtures, an impact hammer, an eddy current displacement meter (Donghua 5E106), and a dynamic strain indicator (Donghua 5923N). The joint was clamped vertically and one end was secured (i.e., the beam with a length of 50 mm) by jaws of the fixtures. Impact forces were applied at different points on the specimens during the test. The displacement meter was used to measure the displacement responses of the specimens with a sampling frequency of 10 kHz. The damping ratios were measured by fitting the displacement decay curves with the logarithmic decay method and the 1^{st}- order resonant frequencies were determined in the frequency spectra. To minimize the influence of higher-order vibration modes which quickly decay in the free decay signal, the fitting interval was set when the response displacement was between 1/3 of the maximum signal displacement and 0.05 mm which was introduced by environmental noise.

5. Results and Discussion

To verify the efficiency of the proposed finite element method in predicting the modal parameters of the composite materials, the 1^{st}-order damping ratios and resonant frequencies of the cross-ply composites (i.e., C100 and C130), obtained from both the experiment and the simulation were compared as shown in **Table 2**. It was observed that both the resonant frequencies and the damping ratios increased when the specimen length increased. A comparison of two methods showed that the numerical predictions were found to closely approach the experimental data with a relative error of less than 5%.

Figure 7 shows the damping ratios of the two joints (i.e., J100 and J130) subject to typical torques resulting from the different impact forces. Due to the influence of the damping effect of the air, the damping ratios of the joints displayed a linear dependence on the maximum displacement response upon being subjected to the impact forces. The intercept of the curves, which represents the damping ratios of a joint measured from a “zero” excitation magnitude, was taken as the equivalent damping ratios of the joints in this study. For both joints, it can be seen that the damping ratios decreased as the applied torque value increased.

Figure 8 shows the displacement decay curves and frequency spectra of the J100 joint with typical residual torques. It can be observed that the displacement response decays more quickly for the joint with the smaller torque value, while the resonant frequencies decreased with an increase in the applied torques. A comparison of the results showed that numerical analysis was found to have a good consistency with the experimental findings.

To achieve a quantitative analysis, the graphs of the damping ratios and resonant frequencies of the two joints under increasing torques have been depicted in **Figure 9**. It can be concluded from the experimental investigation that for a certain torque range the damping ratios increased while the resonant frequencies decreased when the applied torque for both joints was increased. Taking the J100 joint as a representative result, starting at 1 N·m, the damping ratios decreased with some fluctuations, until the applied torque reached 13 N·m. Whereas, the resonant frequencies increased from 1 N·m until a torque of 11 N·m was reached, after which the resonant frequencies slightly decreased as the applied torque was increased. This implies that the pressure applied to the specimen, induced by a torque of 11 N·m was closed to or exceeded the compressive yield strength of the connecting material. The damping ratios of the J100 and J130 joints under the fully loose conditions (1 N·m) increased by 10.3% and 8.5%, respectively, compared to those under the fully tightened conditions (13 N·m). The resonant frequencies of the joints decreased by 0.59% and 0.91% when subject to the fully loose conditions, compared to those under the fully tightened conditions. A comparison of the methods showed that the numerical prediction reached a good agreement with the experimental investigation. In order to further study the effect of the contact conditions of rough surfaces on the modal parameters of bolted composite joints, joints with perfectly smooth surfaces were comparatively modelled. The contact properties (i.e., stiffness and energy dissipation) of the perfectly smooth interfaces were independent of the applied pressure due to the unchanging contact area. As a result, the resonant frequencies and damping ratios of the joints with perfect interfaces remained the same, regardless of applied torques. In addition, the results from the numerical analysis showed that the existence of rough surfaces leads to an increase in the damping ratio (induced by interfacial slip), and a decrease in the resonant frequency (caused by partial contact at the rough interface). From the theoretical analysis of the interfacial properties of rough contact surfaces under different pressures, as shown in **Figure 5**, it can be concluded that as the applied torques increase, more asperities will come into elastic contact at the interfaces and asperity sliding will decrease in the tangential direction, which will lead to an increase in the resonant frequency along with a decrease in the damping ratio of the bolted composite joints.

6. Concluding Remarks

A multiscale numerical model, that considers both microscopic interfacial properties and macroscopic composite properties, has been proposed to simulate the dynamic responses of bolted composite joints. The damping and stiffness of the bolted composite joints were modelled using the complex moduli of both the composites and the rough contact interfaces and also using elastic-viscoelastic correspondence principle. The modal parameters of the joints were solved using ABAQUS software through complex eigenvalue method. From a comparison of the modal parameters between the composite beams (C100 and C130) and the bolted composite joints (J100 and J130) of the same total length, the joints experienced a decrease in the resonant frequency and an increase in the damping ratio when the torque value was increased. Due to the surface roughness, the contact conditions of the interface of a joint were dependent on the applied torque. As the applied torque value was increased, more contact asperities came into elastic deformation at the interface, which resulted in an increase in the normal and tangential storage modulus of the contact interfaces. While energy dissipation of the interface decreased due to a reduction in asperity sliding in the tangential direction. Consequently, the resonant frequency of the bolted composite joint increased and the damping ratio decreased with increasing applied torques. The results obtained from the numerical predictions and the experimental investigations were found to have good consistency. This study has provided a theoretical contact model framework, which has relied on numerical analysis that can be accurately conducted to facilitate the task of predicting the dynamic responses for bolted composite structures subject to different clamping loads. Our current study has been dedicated to conducting first-order modal analysis of composite joints, and in the future the effect of contact conditions between rough surfaces on the higher-order modal properties of bolted composite structures will be further investigated.

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