Abstract: Friction-type bolted joints are widely used in both the civil and aerospace industries. Uncontrolled excessive bolt clamping force can cause damage to the laminated fiber-reinforced polymeric (FRP) composite through the thickness and damage the joint before applying the service loads. The effect of the friction coefficient (between 0 and 0.3), bolt clearance, joint type, and other parameters on failure modes and the maximum bolt clamping force of the carbon FRP lapped joint is studied. A three-dimensional finite element (FE) model consisting of a bolt, a washer, a laminate FRP composite plate, and steel plates was developed for the simulation of the double- (3DD) and single (3DS)-lapped bolted joint. The FE model was validated by using experimental results and was able to predict the experimental results by a difference of between 2.2 and 6.7%. The joint capacity of the clamping force was found to be greatly increased by adopting the double lap technique, which involves placing an FRP composite plate between two steel plates. Also, it was recommended to use an internal washer diameter less than or equal to the FRP composite plate hole diameter since a larger washer clearance can produce higher contact pressure and reduce the resistance by 22%. In addition, reducing the bolt head diameter can lead to a 65% reduction in the 3DS joint clamping strength.

Keywords: CFRP; bolted joint; clamping force; thickness damage; modeling

1. Introduction

Due to their advantages, such as the ease of disassembly and assembly, bolted joints are commonly used in several composite connecting applications. It is known that bolted joints are the weakest link in any structure. Bolt hole drilling could create microcracks, and even the existing hole produces a high stress concentration during the construction or the service of the structure [1–4]. The analysis and the design of fiber-reinforced polymeric (FRP) laminated composite bolted joints involve a high degree of complexity and require special attention because of their anisotropic, inhomogeneous, and viscoelastic properties. Bolt pre-load, joint geometry, fiber orientation, and stacking sequence are some of the control parameters that should be considered for a reliable joint design [5].

Clamping force is an important parameter; hence, the ultimate bearing strength of the bolted joints can be increased to a saturation level by increasing the clamping pressure. At the saturation level, no increase in clamping can be evaluated by fastening the bolt. In addition, the delamination bearing strength increases progressively due to the clamping pressure. Up to now, there are no criteria to control or prevent clamping force-induced damage in the design of FRP composite bolted joints [6]. Industrial specifications, bolt-specific applications, and bolt types affect the recommended bolt clamping stress values. When a bolt is tightened to join two assemblies, the bolt elongates, producing tension or
pre-load in the fastener, which then results in a compressive load on the components being mated. For brittle materials such as fiber-reinforced polymers, excessive clamping stress can cause damage and lead to a significant loss in strength.

There are no guidelines or recommendations for the bolt clamping stress values of the fasteners used to link the FRP composite component joints. Some experimental and numerical studies were performed, which focused mainly on the maximum clamping stress of some specific configurations [6–10]. Most previous studies focused on the effect of clamping force on joint tension strength [11–15]. Experimental works were conducted to study the effect of clamping force on the stiffness and strength of FRP composite single- and double-lapped joints [11–15]. Experimental results showed that tightening the bolt increased the initial bearing stress by 22% and the maximum bearing stress by 105% [11–14]. Increased clamping pressure significantly improved the post-peak stiffness, whereas the initial stiffness and the bolt hole elongation decreased simultaneously by 20 to 50% [11–14]. Parameters such as the washer size and friction coefficient were not deeply studied.

As the clamping force increased, the failure changed to the favorable progressive failure mode from the usual catastrophic fracture [12–15]. In progressive failure, the laminated composite element fails layer by layer or part by part, achieving a load deformation curve consisting of a linear part, nonlinear part, and post-peak part [16–19]. On the other hand, brittle failure is a catastrophic fracture at which most or all of the load displacement relation is the linear part.

The effect of lateral supports on the failure mode of FRP composite bolted joints was experimentally studied [13–15]. It was found that the clamp load increased with the applied tensile load as the delamination was suppressed by the clamping pressure and the closed inter-laminar cracks.

The effects of clamp-up pressure on the net tension failure of bolt-filled laminated FRP composite plates were studied experimentally [14,15]. For hole-filled samples, the higher the clamping pressure, the lower the tensile strength of the bolt filled-hole laminate. Unlike the bolt filled-hole laminate, the tensile strength of the bolted FRP composite joints increased with clamping pressure [14,15].

Limited research focused on the effect of increasing the clamping force on inducing damage in FRP composite plates [6–10,20]. As the laminate through-the-thickness compression strength (TTCS) has a significant effect on the bolt clamping strength, it was experimentally investigated with different sample geometries and laminate layups [21,22]. It was found that the dominant parameter in the through-thickness stiffness modulus is the matrix cross-ply layup orientation [0°, 90°], which can double the stiffness of the unidirectional laminate. Therefore, the fibers played a role in the TTCS by constraining the material against Poisson’s transverse effect. The mixture rules did not give a good prediction of the modulus of the TTCS and strength because the orientation of the fiber was not considered but it is important in the simulation since most of the finite element modeling (FEM) codes implemented the rule of mixtures in calculating the TTCS of the laminate. Many experimental tests were performed to determine the single-lap joint fastener torque limit [7,9,22]. Different unidirectional and quasi-isotropic laminate configurations were examined in these tests. Tests were implemented with and without the friction effect. Grease was used in the interface between the bolt and nut and the nut and washer to eliminate friction; however, for the samples without grease, the friction coefficient was not identified. The results showed that the maximum clamping force was about 50% higher than the frictionless case. Also, the fasteners were not able to damage the investigated FRP composite system as the thread of the fastener failed first. Although the samples did not fail due to FRP composite damage, this study provides good information that can be used for validating finite element (FE) models [7].

Several works were adopted to analyze the progressive damage of FRP composite materials [23–33]. The simplest technique was the ply discount, in which an instantaneous factor of degradation was applied to the stiffness if any element met the failure criteria [23].
Although this model was easy to implement, in many cases, the numerical results did not compare well with the experimental results due to the sudden complete failure [5].

In this paper, a numerical study was performed to investigate the effect of different parameters on the maximum clamping force of FRP composite steel single- and double-lapped joints. The composite under investigation is a carbon FRP (CFRP) laminated composite plate. Nonlinear finite element (FE) modeling was performed in the bolted joint assembly. In these models, the joint was subjected to an excessive clamping force that caused damage to the composite material and pushed it to the post-peak softening stage. The models were validated by previously published experimental results. Parameters such as the bolt diameter (D_b), bolt head diameter (D_bh), friction coefficient (µ), bolt hole clearance (C), and washer hole clearance (CW) were studied to find their effects on the strength and damage of the single- and double-lapped joint.

2. Numerical Modeling

In this section, the problem under investigation will be discussed. The finite element modeling, i.e., element type, material type, and contact details, will be described.

2.1. Bolted Joint Model Description and Boundary Condition

To study the double- or single-lapped joint in the three-dimensional (3D) space, a set of finite element models was built. Figure 1 shows the joints under investigation. Each 3D model consists of two plates (PL_1 and PL_2) for the single-lap joint case (3DS) and three plates (PL_2, PL_1, and PL_2) for the double-lap joint case (3DD). This connection geometry is characterized by the plates’ width (W), the plates’ length (L), the plates’ thicknesses (t_{PL_1}) and (t_{PL_2}), the edge distance (e), the bolt diameter (D_b), the hole diameter (d_h), the washer’s internal (d_i) and external (d_o) diameters, and the bolt head diameter (D_{bh}).

![Figure 1](image_url)

Figure 1. Specimen components and geometry.

PL_1 is always a CFRP composite laminate, while PL_2 is a metal plate, according to the case study. ANSYS finite element code was used to perform the simulation [34]. Figure 2 shows the finite element mesh of the 3DD and 3DS models. Three-dimensional SOLID185 elements with eight nodes were used to model the plates and the bolts [34]. SOLID185 has orthotropic damage capability, which is used to handle the CFRP composite laminate layered medium. For the 3DD model, three symmetric boundary conditions (BCs) were used on the XY, XZ, and ZY planes, making the FE model stable and constrained in all directions. For the 3DS model, only two symmetric boundary conditions (BCs) were used on the XZ and ZY planes, making the FE model stable and constrained in the X and Z
elements with eight nodes were used to model the plates and the bolts [34]. SOLID185 was permitted to slide inside the nut to simulate the bolt-tightening process. The third one is the MPDG model. This model supports 3D solid failure model. The second model supports plane and shell elements only and it is built based on damage mechanics.

For the Y direction, the top surface of the bolt nut was restrained. The bolt shank was permitted to slide inside the nut to simulate the bolt-tightening process. The force in the bolt due to the incremental displacement was recorded and considered the clamping force (F).

After starting the loading of the 3DD model, the model will be symmetric around the XZ plane, except for the bolt shank because the bolt shank will be extended in one direction. It was assumed that this would not affect the results as the bolt is very rigid compared to the composite material and the only function of the bolt is to simulate the pressure exerted on the washer.

To simulate the laminated CFRP composite, a volume layer was built for every layer. Orthotropic material characteristics were defined and assigned to every lamina. A unique local coordinate system was assigned for every lamina with the proper fiber direction to consider the material directions. A mesh convergence study was performed to find the size that produces accurate results with the lowest computational efforts. The model was studied with different numbers of elements between 5000 and 25,000 elements. It was found that changing the number of elements from 20,000 to 22,000 elements changed the results by less than 0.5%. The number of solid elements was 22,000 and 25,000 elements for the 3DS and 3DD models. A very fine mesh was used in the CFRP composite plate, especially around the location of stress concentrations, and a larger element size was used near the edges of the plate. In the same model, the element size of the CFRP composite plate ranged from 0.06 to 1.5 mm.

2.2. Material Model

ANSYS 18.0 has three built-in materials that can simulate FRP composite damage. The first model is available only for explicit analysis which is a general FRP composite failure model. The second model supports plane and shell elements only and it is built based on damage mechanics. The third one is the MPDG model. This model supports 3D solid layered elements, such as SOLID185 and SOLID186. In this model, if the damage initiation criterion is met, the degradation factor can be multiplied directly by the corresponding material stiffness component. Since this paper is mainly focused on the 3D domain with implicit analysis, MPDG was used in this study [16]. The laminated plate that was used in this study is composed of carbon/epoxy composite plies with a stacking sequence of [0/(±45)/90]_{3s}, as listed in Table 1 [6,22]. $E_1$, $E_2$, and $E_3$ are the elastic moduli in the X, Y,
and Z directions, respectively. $G_{12}$ and $\nu_{12}$ are the in-plane shear modulus and Poisson’s ratio, respectively. $S_{t1}$ and $S_{t2}$ are the lamina tension strengths in the in-plane and out-of-plane directions, respectively. $S_{c1}$ and $S_{c2}$ are the lamina compression strengths in the in-plane and out-of-plane directions, respectively.

**Table 1.** Properties of IM7/8552 carbon/epoxy prepreg sheet [6,22].

<table>
<thead>
<tr>
<th>Stiffness Parameters</th>
<th>Strength Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1$</td>
<td>164 GPa</td>
</tr>
<tr>
<td>$E_2$, $E_3$</td>
<td>11.7 GPa</td>
</tr>
<tr>
<td>$G_{12}$, $G_{13}$</td>
<td>4.50 GPa</td>
</tr>
<tr>
<td>$\nu_{12}$, $\nu_{13}$</td>
<td>0.3</td>
</tr>
<tr>
<td>$S_{t1}$</td>
<td>2723 MPa</td>
</tr>
<tr>
<td>$S_{c1}$</td>
<td>1689 MPa</td>
</tr>
<tr>
<td>$S_{t2}$</td>
<td>64 MPa</td>
</tr>
<tr>
<td>$S_{c2}$</td>
<td>137 MPa</td>
</tr>
<tr>
<td>TTCS</td>
<td>1185 MPa</td>
</tr>
</tbody>
</table>

ANSYS supports several failure criteria such as maximum stress, maximum strain, Hashin, and Puck failure criteria. Initial simulations were performed by using all of these criteria. It was found that the maximum stress criteria give the closest result compared to the results from the experimental study. In this study, the maximum failure criteria were used.

The plate $P_{2}$, the washer, and the bolt were made of steel. A bilinear elastic isotropic model was used with Young’s modulus, Poisson’s ratio, and a tangent modulus of 200 GPa, 0.3, and 2000 MPa, respectively. The yield stress ($F_y$) of the washer and the bolts is 896 MPa, while it is equal to 500 MPa for the steel plate.

2.3. Contact Properties

Surface-to-surface contact pairs were used to simulate the contact between the joint parts. The parts include the interfaces between the bolt head and the plate, the washers and the plates, and the two plates, in addition to the bolt shank and the holes. The ANSYS contact pair consists of contact elements (CONTA174) and target elements (TARGE170). The Coulomb friction model was used which is supported by ANSYS [17–19]. Friction coefficient values of 0.1, 0.2, and 0.3 were considered in this study.

3. Validation of Finite Element Model

Previously published bolt torque experimental test results were utilized to validate the finite element model [6,22]. In this work, a 320-grit diamond circular saw was used to cut IM7/8552 test specimens to a nominal length $L = 75$ mm and width $W = 75$ mm to reduce the microdamage induced from the cutting process. The IM7/8552 sheet was manufactured from carbon fiber with an intermediate modulus that is encased in a high-strength, mid-toughened, 350 °F-cured, damage-resistant, structural epoxy [6,22]. The nominal thickness of the cured ply is $0.196$ mm, with a 60% fiber volume fraction.

In this study, the laminated configuration $[0/(\pm 45)/90]_{3s}$ will be used to verify the FE modeling. The plates had nominal thicknesses of $t_{P_{1}} = 4.70$ mm. All plates were cured in an autoclave according to the recommended Hexcel cure cycle. Table 1 shows the properties of the single lamina. Two different 1100 MPa bolts were used for investigation, with diameters $D_b = 6.35$ mm and $12.5$ mm. The bolt designations used in this study were NAS1958C-32 (12.5 mm) and NAS1954C-32 (6.35 mm), with corresponding self-locking nuts. The NASI587-8 washers ($d_o = 22.14$ mm and $d_i = 12.8524$ mm) and NAS1587-4 washers ($d_o = 11.2$ mm and $d_i = 6.5$ mm) with 1100 MPa strength were used.

In the experimental test, the steel plate is a rigid plate and the testing machine pushes the bolt assembly on the composite. The 3DS FE modeling technique described in the previous section was modified and used to simulate the experimental specimens. The bottom surface of the steel plate was restrained in the Y direction and the bolt nut was removed.
Figure 3 shows the quarter model of the samples, where symmetric boundary conditions were used at planes XY, ZX, and ZY. The model was solved under gradual displacement up to failure.

![Quarter FE model of maximum clamping force samples](image)

The maximum clamping force ($P_{\text{max}}$) result obtained by Kostreva [22] was compared to the numerical model results. The experimental and numerical results of $P_{\text{max}}$ of the IM7/8552 rectangular samples can be found in Table 2.

**Table 2. Comparison between experimental and numerical results.**

<table>
<thead>
<tr>
<th>Bolt Model</th>
<th>$D_b$, mm</th>
<th>Clamping Force, kN</th>
<th>Experimental</th>
<th>Numerical</th>
<th>Difference, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>NAS1958C-32</td>
<td>12.5</td>
<td>144.56</td>
<td>141.36</td>
<td>−2.2</td>
<td></td>
</tr>
<tr>
<td>NAS1954C-32</td>
<td>6.35</td>
<td>33.36</td>
<td>35.6</td>
<td>6.7</td>
<td></td>
</tr>
</tbody>
</table>

The model gives a good prediction for the experimental results. For example, the $P_{\text{max}}$ of the present model of the 6.35 mm diameter bolt $[0/(\pm 45)/90]_3s$ was 1069.98 MPa compared to the measured experimental value of 1072.82 MPa, which represents a good correlation between the experiment and the present model, with a percent difference of about −2.20%. For all tested samples, the maximum difference was +6.7%. It can be concluded that the current model can predict the bolt clamping strength of the laminated CFRP composite.

4. Determination of Maximum Clamping Force in Lapped Joints

Two FE models were created to evaluate the maximum accepted value of clamping force in 3DS and 3DD. The laminate that was considered is composed of CFRP composite plies made from carbon/epoxy IM7/8552 lamina, as given in Table 1. The stacking sequence of the composite PL1 is $[0/(\pm 45)/90]_3s$, with a 4.69 mm thickness for both 3DS and 3DD models, while PL2 is made of steel with a 4.69 mm thickness for 3DS and 2.345 mm for 3DD. The specimen geometries are typical 3DS and 3DD joints, as shown in Figure 3, $W = 25.4$ mm and $L = 25.4$ mm, with different values for the other parameters, as shown in Table 3. The effects of the friction coefficient and bolt hole clearance on the behavior of the single- and double-lapped joints will be demonstrated by comparing cases 1 through 24. The effects of washer bolt hole clearance on the behavior of the single-lapped joints will be demonstrated by comparing cases 9 and 25 through 28. The effects of bolt diameter on the behavior of the single-lapped joints will be demonstrated by comparing cases 9 and
29 through 32. Finally, the effects of bolt head diameter will be investigated by comparing cases 9, 11, and 33 through 34.

Table 3. Numerical parametric study.

<table>
<thead>
<tr>
<th>Case #</th>
<th>$D_b$</th>
<th>$D_{bh}$</th>
<th>$d_h$</th>
<th>$d_l$</th>
<th>$d_o$</th>
<th>$\mu$</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 to 8</td>
<td>6.35</td>
<td>9.5</td>
<td>6.35</td>
<td>6.5</td>
<td>11.2</td>
<td>0, 0.1, 0.2, and 0.3</td>
<td>3DS and 3DD</td>
</tr>
<tr>
<td>9 to 16</td>
<td>6.35</td>
<td>9.5</td>
<td>7</td>
<td>6.5</td>
<td>11.2</td>
<td>0, 0.1, 0.2, and 0.3</td>
<td>3DS and 3DD</td>
</tr>
<tr>
<td>17 to 24</td>
<td>6.35</td>
<td>9.5</td>
<td>8</td>
<td>6.5</td>
<td>11.2</td>
<td>0, 0.1, 0.2, and 0.3</td>
<td>3DS and 3DD</td>
</tr>
<tr>
<td>25</td>
<td>6.35</td>
<td>9.5</td>
<td>7</td>
<td>7.0</td>
<td>11.4</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>26</td>
<td>6.35</td>
<td>9.5</td>
<td>7</td>
<td>7.5</td>
<td>11.8</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>27</td>
<td>6.35</td>
<td>9.5</td>
<td>7</td>
<td>8.0</td>
<td>12.15</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>28</td>
<td>6.35</td>
<td>9.5</td>
<td>7</td>
<td>8.5</td>
<td>12.5</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>29</td>
<td>7</td>
<td>10.15</td>
<td>7.65</td>
<td>7.15</td>
<td>11.85</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>30</td>
<td>8</td>
<td>11.15</td>
<td>8.65</td>
<td>8.15</td>
<td>12.85</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>31</td>
<td>9</td>
<td>12.15</td>
<td>9.65</td>
<td>9.15</td>
<td>13.85</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>32</td>
<td>10</td>
<td>13.15</td>
<td>10.65</td>
<td>10.15</td>
<td>14.85</td>
<td>0.2</td>
<td>3DS</td>
</tr>
<tr>
<td>33 to 34</td>
<td>6.35</td>
<td>7.0</td>
<td>6.5</td>
<td>11.2</td>
<td>0 and 0.2</td>
<td>3DS</td>
<td></td>
</tr>
</tbody>
</table>

Note: bolt hole clearance (C) = $(d_h - D_b)/2$ and washer hole clearance (CW) = $(d_l - D_b)/2$.

Symmetric BCs were added at the mid-plane of the joint. The 3DD model that is seen in Figure 2a was created to simulate the bolt-tightening process in the case of the double-lapped joint model. The second model that is found in Figure 2b was created to simulate the bolt-tightening process in the case of the 3DS model. These models were subjected to incremental clamping forces up to failure. In this paper, we focused on the failure of the CFRP composite plate; so, an elastic material model was used for the bolt to disable its failure possibility.

5. Results and Discussion

In this section, the results of the parametric study will be presented. The effect of the bolt hole clearance, washer hole clearance, friction coefficient, and bolt head size on the maximum clamping force of the joint will be highlighted.

5.1. Load vs. Thickness Displacement Behavior

The results of all 3DS and 3DD lapped joint cases are shown in Figures 4 and 5. For the 3DS samples, it can be seen that the load–deformation curve is composed of four parts, as shown in Figure 4a–c. The first part is the linear elastic part up to the joint elastic limit. The second part is a small nonlinear hardening part up to the peak load, and the third part is the post-peak softening part, where in the last stage, the load starts to increase gradually again. For the zero-clearance samples shown in Figure 4a, it was observed that the elastic stage has an initial stiff part that increases by increasing the friction coefficient; after this part, the initial stiffness suddenly decreases. This happens due to the initial sliding that happens between the steel and composite parts, causing the stiffness to drop. For the three cases, in Figure 4, it can be observed that the size of the nonlinear hardening stage is small compared to the linear stage due to the direct contact between the washer and composite material that causes early damage due to the high stress concentration on the edges of the washer. Due to the displacement control analysis, the model was able to predict the post-peak part. The post-peak stage drop is highly affected by the friction in the case of no clearance, shown in Figure 4a; however, the effect is lower in the case of joints with clearance, as seen in Figure 4b,c. The post-peak performance is out of the scope of this paper and will be discussed in more detail in future works.
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For the initial stiffness, no significant difference between the curves was observed for $C = 0.0 \, \mu\text{m}$. However, by increasing the clearance, the relations become more divergent. For the case with $\mu = 0$, by increasing the clearance to 325 and 825 $\mu\text{m}$, the stiffness decreased by 9.95 and 33.1%, respectively. For the case of $\mu = 0.1$, the stiffness decreased by 4.25 and 28.31% by increasing the clearance from 0 to 325 and 825 $\mu\text{m}$, respectively.

It was also observed that increasing the friction coefficient leads to increases in the stiffness. By increasing the friction coefficient from 0 to 0.3, the initial stiffness increased by 1.71, 18.95, and 29.97% for $C = 0.0, 325$, and 825 $\mu\text{m}$, respectively.

For the 3DD samples, the load–deformation curve is composed of three parts, as seen in Figure 5a–c. The first three stages are similar to the 3DS case, while the fourth stage in the 3DS models disappears here. Unlike the 3DS case, the size of the nonlinear hardening stage is larger here due to the absence of direct contact between the washer and the composite plate which produces a lower stress concentration.

The third part is the post-peak part which is very small and sometimes disappears in 3DD cases. In general, it can be seen that the maximum clamping strengths of the 3DD joints are higher than the strength of the 3DS joints due to the more uniform stress distribution produced by the steel plate.

**Figure 4.** Load–deformation curves for 3DS joints; (a) $C = 0$, (b) $C = 325 \, \mu\text{m}$, and (c) $C = 825 \, \mu\text{m}$. 

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Figure 5. Load–deformation curves for 3DD joints; (a) $C = 0$, (b) $C = 325 \, \mu m$, and (c) $C = 825 \, \mu m$.

To demonstrate the difference between the load–displacement curves of the single- and double-lapped joints, Figure 6a,b show a comparison between the bolt clamping force and CFRP composite thickness deformation of the cases of 3DD and 3DS joints. For clearance values of 0 and 325 $\mu m$, it can be seen that the 3DS joint shows greater progressive failure behavior than the 3DD case. However, the strength of the double-lapped joints is much higher with pure brittle behavior in the 3DD case with no clearance. The failure load of the 3DD case was 50% larger than the failure load of 3DS in the case of $C = 0$, while the ratio was 56% in the case of $C = 325$. In terms of strength, it can be concluded that using double-lapped joints is much more effective than using single-lapped joints.

As mentioned before, there is a small stiff part in the case of $C = 0$ with friction that will be neglected from the stiffness comparison. The initial through-thickness stiffness of the 3DD model is larger than that of the 3DS model in both clearance cases. The initial stiffness of the 3DD sample was 53% larger than the stiffness of 3DS in the case of $C = 0$, while the ratio was 78% in the case of $C = 325$. 
Figure 6. Bolt force vs. CFRP composite laminate thickness deformation.

In addition, the existence of clearance can severely decrease the strength of the single-lapped joints, although it did not affect the case of double-lapped joints significantly. The main difference between the two cases is that in the case of 3DD, the load is transferred from the bolt head to the washer and then from the washer to the steel plate, and finally distributed over the CFRP composite plate. For 3DS, the load is transferred from the bolt head to the washer and then from the washer to the CFRP composite plate directly, with a stress value much larger than in the case of 3DD. It can be concluded that for friction-type connections or combined friction and bearing connections with CFRP composite plates, it is preferable to use the double-lapped joint as it will be more efficient.

5.2. Bolt Clamping-Induced Damage

In ANSYS, the maximum failure criterion (MFC) is computed based on the maximum of the fiber tensile failure criterion, fiber compressive failure criterion, matrix tensile failure criterion, and matrix compressive failure criterion. The damage index is calculated for every mode of failure based on the failure criteria used in the material definition and the maximum index will be presented in the MFC results. If the value of the MFC is higher than one, the element is considered damaged or partially damaged. This parameter was used to show the effect of clamping force on the CFRP composite. Figure 7 shows the MFC for all of the 3DS models and selected 3DD models at the same value of bolt contraction, i.e., δ. At the same δ value, it can be seen that the damage-affected area of the single-lapped joints is deeper than in the case of double-lapped joints, as shown in Figure 7a,d. It can also be seen that the damage extends through the thickness in the case of C = 325 more than in the case of C = 0, as shown in Figure 7c. This might be due to the lateral constraint that is provided by the bolt shank in the case of C = 0. In addition, the damage is concentrated more around the outer edge of the washers, as shown in Figure 7b,c. Finally, the damage-affected area decreased by increasing the friction coefficient, as shown in Figure 7a,b. The maximum value of the damage index is the maximum value of the contour bar. It can be seen that the damage index is higher in the cases without clearance than in the cases with clearance. In addition, the single-lapped joint produced much greater damage than the double-lapped joint due to the direct contact between the washer and the composite material in the single-lapped joint.
5.3. Effect of Bolted Joint Type on Stress Distribution

Figure 8a shows a comparison between the elastic stress distributions in the 3DS model before damage at 30% of $P_{\text{max}}$ and after damage at 100% of $P_{\text{max}}$. The stress values that are shown in the vertical axis are normalized by the maximum stress value over the curve. The distance values that are shown in the horizontal axis are normalized by the bolt diameter. It can be observed that after the damage was enforced, the peak point shifted toward the direction outside the plate due to the softening effect. This observation is more apparent in the 3DD case, as shown in Figure 8b.

5.4. Effect of Bolt Hole Clearance on Joint Clamping Strength

Figure 9 shows the relation between $P_{\text{max}}$, i.e., the maximum clamping force, and several bolt hole clearance values for the 3DS and 3DD models. In general, it can be seen that increasing the bolt hole clearance decreases the clamping strength. In the 3DS case, increasing the clearance from 0 to 825 µm led to a decrease in the maximum clamping force by 49.35, 41.78, 47.54, and 59.90% for joints with friction coefficients of 0, 0.1, 0.2, and 0.3, respectively. For the 3DD models, the reduction ratio was 23.21, 24.68, 30.74, and 28.07% for joints with friction coefficients of 0, 0.1, 0.2, and 0.3, respectively.
Increasing the clearance decreases the contact area between the washer or the steel plate and the CFRP composite plate, which increases the contact stresses and causes premature failure. In addition, for 3DS, decreasing this contact area from one side of the washer produces more nonuniform stress on the CFRP composite, causing a greater stress concentration. This explains why increasing the clearance values decreased the clamping strength of the 3DS joints more than in the 3DD cases.

5.5. Effect of Friction Coefficient on Joint Clamping Strength

The friction coefficient also affected the value of $P_{\text{max}}$, as seen in Figure 10. For 3DD, increasing $\mu$ from 0 to 0.3 led to an increase in $P_{\text{max}}$ by 14.39 and 28.07% for clearance values of 325 and 825 $\mu$m, respectively, as seen in Figure 10a. This increase can be due to the better distribution of $P_{\text{max}}$ due to the existence of the steel plate. This behavior was also observed in the 3DS case with zero clearance. However, for clearance values of 325 and 825 $\mu$m, $P_{\text{max}}$ increased and then decreased by increasing the friction, as seen in Figure 10b. Increasing the friction coefficient from 0 to 0.1 led to an increase in $P_{\text{max}}$; however, for...
friction coefficients higher than 0.1, $P_{\text{max}}$ decreased by increasing the friction coefficient. The stress concentration under the washer in the 3DS joints combined with the shear stress developed by the friction caused this premature failure in the outer layer. This finding is very interesting as it is well known that the friction coefficient enhances the behavior of bolted joints under tension loads. For the single-lapped joint, it is recommended to adjust $P_{\text{max}}$ at levels that do not reduce the strength of the joint.

![Figure 10. Effect of friction coefficient on $P_{\text{max}}$.](image)

### 5.6. Effect of Washer Clearance ($C_W$)

For the single-lapped joint, the washers are rested directly on the CFRP composite. The bolt $P_{\text{max}}$ is transferred to the washer using the bolt head, which produces bearing stresses on the CFRP composite plate around the bolt hole. The geometric characteristics of the washer, such as the internal and external diameters, control this bearing stress and consequently control the joint strength. In addition, the position of the washer relative to the bolt head controls the stress distribution under the washer. The effects of the washer bolt hole clearance on the behavior of the single-lapped joint will be demonstrated by comparing cases 9 and 25 through 27, as shown in Table 3. The washer bolt hole clearance was controlled by changing the inner diameter of the washer; however, the washer’s external diameter was designed to have the same contact area with the surface of the CFRP composite plate. The other parameters, such as the bolt diameter, the bolt head size, and the friction coefficient, are found in Table 3. Figure 11a shows the normalized $P_{\text{max}}$ vs. the thickness deformation of all values of washer hole clearance. The $P_{\text{max}}$ was normalized by the $P_{\text{max}}$ value of the case with 75 $\mu$m clearance.

It can be found that increasing the $C_W$ from 75 to 325 $\mu$m led to an increase in $P_{\text{max}}$ by 7%. Beyond the $C_W$ value of 325 $\mu$m, $P_{\text{max}}$ started to decrease by increasing the value of $C_W$. Changing the $C_W$ value from 325 to 575, 825, and 1075 $\mu$m led to a decrease in $P_{\text{max}}$ by $-2.12$, $-9.93$, and $-20.24\%$, respectively. It can be concluded that there exists transition behavior by increasing the $C_W$ values. If the $C_W$ values are less than the bolt hole diameter, increasing the $C_W$ will lead to an increasing $P_{\text{max}}$; however, increasing the $C_W$ more than the C will lead to a decrease in $P_{\text{max}}$. This might be due to the position of the washer’s internal edge relative to the bolt head. Increasing the $C_W$ might cause eccentricity on the washer, which causes nonuniform bearing stress on the CFRP composite surface. Figure 11b shows the damage index distribution on the surface and through the thickness of the CFRP composite plate. It can be seen that the area under the outer diameter of the washer has maximum damage. The damage also spreads through the CFRP composite layers at a plane of 45° degrees for the case of $C_W = 325 \mu$m.
Figure 11. Effect of washer hole clearance: (a) normalized clamping force vs. thickness deformation relation and (b) damage index (MFC).

5.7. Effect of Bolt Diameter ($D_b$)

Increasing the bolt diameter will require a larger washer, which will lead to an increase in the contact area between the washer and the CFRP composite plate. To study the effect of bolt diameter on the maximum $P_{\text{max}}$ of the single-lapped bolted joint, five FE models were built. The same bolt hole clearance was maintained; however, other parameters such as the bolt head and the washer dimensions were modified to fit the new bolt size. The details of these models are found in Table 3 for cases 9 and 29 through 32. Figure 12a shows the normalized $P_{\text{max}}$ vs. the thickness deformation of all values of bolt diameters. The maximum clamping force $P_{\text{max}}$ was normalized by the $P_{\text{max}}$ of the case with a 6.35 mm diameter. It can be found that by increasing the bolt diameter from 6.35 mm to 7, 8, 9, and 10 mm, $P_{\text{max}}$ increased by 8, 17, 30, and 9%, respectively, due to the increase in the washer area that reduces the contact stress between the washer and the CFRP composite plate. Figure 12b shows the damage index distribution on the surface and through the thickness of the CFRP composite plate for the joint with a 10 mm diameter.

5.8. Effect of Bolt Head Diameter ($D_{bh}$)

Decreasing the bolt head diameter will lead to smaller contact area between the bolt head and the washer. For a small diameter bolt head, the bolt head will be rested near the internal side of the washer, which might make the washer apply nonuniform stress on the CFRP composite, causing premature failure. To show how the bolt size affects the value of $P_{\text{max}}$, two models with very small bolt head sizes ($D_{bh} = 7$ mm) were built (cases 33 and 44), as given in Table 3. The bolts were loaded until CFRP composite failure and compared with the same models with the normal bolt head sizes ($D_{bh} = 9.5$ mm). The internal diameter of the washer $d_i$ is 6.5 mm; so, the overlapped part of the diameter between the washer and the bolt head is 0.5 mm in the case of a small washer and 1.5 mm in the case of a large washer. Figure 13 shows a comparison between $P_{\text{max}}$ and the thickness displacement for both the large and small washers at two values of $\mu$, i.e., 0.0 and 0.2. It can be seen that the small washer significantly reduces $P_{\text{max}}$ by 65% at $\mu = 0.0$ and by 46% at $\mu = 0.2$. 

(a) Force displacement relation. (b) MFC.
5.8. Effect of Bolt Head Diameter (Dbh)

Decreasing the bolt head diameter will lead to smaller contact area between the bolt head and the washer. For a small diameter bolt head, the bolt head will be rested near the internal side of the washer, which might make the washer apply nonuniform stress on the CFRP composite, causing premature failure. To show how the bolt size affects the value of $P_{\text{max}}$, two models with very small bolt head sizes ($D_{bh} = 7$ mm) were built (cases 33 and 44), as given in Table 3. The bolts were loaded until CFRP composite failure and compared with the same models with the normal bolt head sizes ($D_{bh} = 9.5$ mm). The internal diameter of the washer $d_i$ is 6.5 mm; so, the overlapped part of the diameter between the washer and the bolt head is 0.5 mm in the case of a small washer and 1.5 mm in the case of a large washer. Figure 13 shows a comparison between $P_{\text{max}}$ and the thickness displacement for both the large and small washers at two values of $\mu$, i.e., 0.0 and 0.2. It can be seen that the small washer significantly reduces $P_{\text{max}}$ by 65% at $\mu = 0.0$ and by 46% at $\mu = 0.2$.

In addition, although the friction coefficient reduced the value of $P_{\text{max}}$ for the large washer, it caused an increase in the case of the small washer. To understand this, the deformation of the through-thickness plane of the joint was plotted, as seen in Figure 14. It can be seen that the small overlap between the washer and bolt head caused significant rotation for the washer, as shown in Figure 14a,c, while the rotation was un-notable in the case of the large washer, as shown in Figure 14b,d. This rotation causes nonuniform stresses and a greater stress concentration on the CFRP composite. It is recommended to use bolts with large bolt heads in CFRP composite bolted joints.
Figure 13. Effect of bolt head diameter.

In addition, although the friction coefficient reduced the value of $P_{\text{max}}$ for the large washer, it caused an increase in the case of the small washer. To understand this, the deformation of the through-thickness plane of the joint was plotted, as seen in Figure 14.

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(a) $\mu = 0.0$ and $D_{bh} = 7.0$ mm  
(b) $\mu = 0.0$ and $D_{bh} = 9.5$ mm  
(c) $\mu = 0.2$ and $D_{bh} = 7.0$ mm  
(d) $\mu = 0.2$ and $D_{bh} = 9.5$ mm

Figure 14. Washer deformation: (a) $\mu = 0.0$ and $D_{bh} = 7.0$ mm, (b) $\mu = 0.0$ and $D_{bh} = 9.5$ mm, (c) $\mu = 0.2$ and $D_{bh} = 7.0$ mm, and (d) $\mu = 0.2$ and $D_{bh} = 9.5$ mm.

6. Conclusions

In this paper, the maximum clamping force that can be induced on the single- and double-lapped CFRP composite plates was studied. For this purpose, a 3D FE model was developed to study the force–displacement behavior of lapped joints. The effect of different parameters, such as the bolt hole clearance, washer hole clearance, and friction coefficient, on the maximum failure load and stress distribution was studied. From this study, several recommendations that could be beneficial for both industry and future research can be drawn as follows.

- The proposed FE model was able to predict the experimental results with a maximum difference of 6.7%.
- The failure mechanism of the 3DD sample was brittle failure, while progressive failure was evaluated in the 3DS samples.
- The load–deformation curve of the 3DS models is composed of three parts. The first part is the linear elastic part up to the joint elastic limit. The second part is the nonlinear hardening part up to the peak load, and the third part is the post-peak softening part.
- The values of stiffness and $P_{\text{max}}$ of the 3DD samples are greater than the values of 3DS by more than 50%. It is recommended to design the CFRP joints as double-lapped joints.
- Increasing the friction coefficient of the 3DS samples decreases the $P_{\text{max}}$ value; on the other hand, for 3DD, increasing $\mu$ leads to an increase in $P_{\text{max}}$. The high friction stresses under the washer could cause early failure of the 3DS joints.
- For the 3DS joints, increasing the washer clearance affects the maximum clamping force value. It can reduce the joint strength by 20%. It is recommended to use an internal diameter less than the bolt hole diameter.
- Increasing the bolt diameter can enhance the $P_{\text{max}}$ value of the 3DS joints by 36% for 10 mm bolts. This might be due to the use of larger washers.
• Decreasing the bolt head diameter can cause high rotational deformation for the washer, which could cause premature failure for the underneath CFRP composite and reduce the maximum clamping force values. It is recommended to use bolts with a large bolt head diameter.

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**References**


20. Ramkumar, R.L.; Saether, E.S.; Cheng, D. Design Guide for Bolted Joints in Composite Structures; Northrop Corporation: Falls Church, VA, USA, 1986. [CrossRef]
23. Pietropaoli, E. Progressive failure analysis of composite structures using a constitutive material model (USERMAT) developed and implemented in ANSYS. Appl. Compos. Mater. 2012, 19, 657–668. [CrossRef]
34. ANSYS Inc. ANSYS Release 18.1 Documentation; ANSYS Inc.: Canonsburg, PA, USA, 2017.

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