TECHNICAL ARTICLE

Simple Fatigue Testing Machine for Fiber-Reinforced Polymer Composite

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Abstract

Fiber-reinforced polymer (FRP) composites seem to be the best options in many structural applications. Such applications are often exposed to fatigue loads, and therefore, the fatigue behavior of the composites must be studied in order to achieve a proper design. This is fulfilled by means of an experimental characterization, in which a lot of specimens are tested because of the several parameters involved (i.e., fiber/matrix ratio, fiber orientation, staking sequence, etc.). Besides, the fatigue tests must be carried out at low frequencies, in order to avoid temperature increments in the polymer matrix, which would change the mechanical properties of the composite. Consequently, considerable time is consumed to perform a complete set of tests and, when using conventional servohydraulic testing machines, costs rise notably. A machine to perform fatigue tests of composite materials under constant amplitude load cycles and a wide range of load ratios is presented in this paper. This machine exhibits as main goals the fulfillment of the corresponding standard requirements, a very low cost compared to conventional servohydraulic testing machines and, consequently, makes reasonably priced to have several machines testing specimens simultaneously, in order to reduce the necessary time to complete the whole characterization.

Introduction

Fiber-reinforced polymer (FRP) composites seem to be the best option in many engineering applications, as aeronautic components, wind turbines, and sport equipment, among others. The fatigue loads play an important role in most of these applications, generating the need to know the fatigue behavior to fulfill a correct design.[1]

Characterizing the fatigue behavior of high-modulus FRP composites is not a simple task due to the particular characteristics of these materials: anisotropy, temperature sensitivity, aging, sensitivity to stress ratios with compression loads, etc.

Every obtained fatigue test result concerns only to the material tested, related to their characteristics (i.e., type of fiber, type of matrix, fiber/matrix weight ratio, fiber orientations, ply stacking sequence, manufacture and cure methods, etc.). Any modification of these parameters demands a new series of tests to characterize the material.[2–5]

Rectangular specimens are generally tested under a previously specified stress cycle given by the maximum stress ($S_{\max}$), the stress ratio ($R =$ minimum stress/maximum stress $= S_{\min}/S_{\max}$), and the load frequency (f). Test results are usually displayed as S–N curves and constant amplitude lifetime (CAL) diagrams.

An S–N curve provides information about the amount of cycles at which the material will fail for different maximum stress levels and a unique value of stress ratio. The failure can be stated as the complete specimen failure, a given stiffness reduction, damage presence, or any other criteria that the designer considers suitable.[1]

A CAL diagram (also named master diagram, or CLD, constant life diagram) indicates the material fatigue life for different load configurations, and joins points of equal number of cycles to failure.[4] The average stress of the cycle [$S_{ave} = (S_{max} + S_{min})/2$] is plotted against the stress amplitude [$S_{cyc} = (S_{max} - S_{min})/2$] as Fig. 1 shows. Note that constant R curves correspond to straight lines. This diagram may be considered as a generalization of a Goodman diagram, although considering R values larger than 1, as FRP composites are sensitive to fiber buckling damage when they are exposed to compression loads.

Thirty ordered pairs were necessary to build the diagram of the figure (the points of null amplitude are from static tests and were not considered), and each of them is an average of at least three tests. It means that 90 tests are approximately needed to build a CAL diagram similar to that shown in the figure.

On the other hand, polymeric matrix properties are temperature sensitive and they are also poor heat conductors, limiting the test frequency up to values lower than 5 Hz,[4] being 1 Hz a usual value.[6] Considering that each specimen generally endures between 10^3 and 10^7 cycles, lasting up to several weeks, the construction of a CAL diagram can demand more than 1 year. All this invested time (and money) is used to characterize only one laminate at the test conditions. Consequently, the final cost of a test is greatly affected by the machine amortization, and therefore diminishing the cost of the testing machine will result in cheaper S–N curves and CAL diagrams.

Fatigue tests can be performed either under cyclic controlled displacement or cyclic controlled load. In the first case, the load on the specimen diminishes, as the test runs, because of specimen stiffness loss, crack growth, or specimen slipping on the grips, etc. Cyclic controlled load tests maintain peak and valley load values, independent of changes that could experience the specimen stiffness. Constant load cycles are typically employed in fatigue tests of composites since they suppose a more critical load situation.

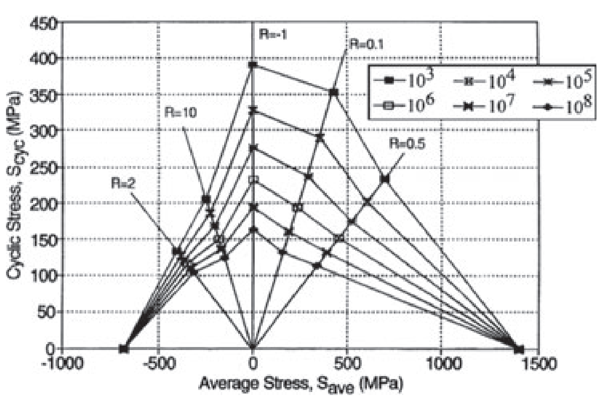

Figure 1 Schematic CAL diagram.

The most used stress ratios, R, are 0.1, 0.5, 2, 10, and −1. ASTM D 3479/D 3479M "*Standard test method for tension - tension fatigue of polymer matrix composite materials*" only refers to tension–tension load tests. For tests with compression loads, and even more with tension–compression loads, there are no universally accepted criteria about the effects of different factors, especially fiber buckling.[1] Therefore, laboratories apply different methodologies, being the most common the use of short specimens or lateral guides to avoid buckling.[6]

A machine to perform fatigue tests of composite materials under constant amplitude load cycles and a wide range of load ratios is presented in this work. This machine presents as main advantages the fulfillment of the standard requirements related to fatigue test of composite materials and a very low cost compared to conventional servohydraulic testing machines. Consequently, it makes reasonably priced to have several machines testing specimens simultaneously in order to reduce the necessary time to complete the whole characterization.

Machine Description

The maximum load values, frequency ranges, and load ratios of the machine, and even the specimen dimensions, have been adopted following the standard ASTM D 3479/D 3479M and a database developed by Sandia[7] with more than 2500 fatigue tests of FRP composite materials.

Considering that this machine is the first prototype, the maximum load was limited up to 10 kN in order to obtain the required power and dimensions. Nevertheless, this capability enables a great variety of test conditions, even more if we bear in mind that the specimen width can be varied. The standard ASTM D 3479/D 3479M recommends a width of 15 mm for specimens composed only of unidirectional fibrous laminae,with the fibers oriented parallel to the test axis, and 25 mm for the remaining ones. The recommended thicknesses range between 1 and 2.5 mm. The machine allows gage lengths up to 100 mm.

The mechanism used in the machine to generate the load cycles consists of a set of helical springs that are compressed (or expanded) by means of a reciprocating mechanism driven by an electrical motor. The deformation of these springs generates a force applied on the seesaw of the machine, which can pivot on a pair of bearings placed at one extreme (see Fig. 2) and in this way the load to the grips is transmitted.

The grips are attached to the seesaw and to the frame of the machine by means of guides that allow changing the horizontal distance up to the point of the seesaw pivot. The force on the specimen (attached to the grips) is inversely proportional to the above-mentioned distance, and therefore the guides are used to govern the load magnitude before beginning the test.

Once the specimen was placed, the minimum load of the cycle was generated compressing or expanding the springs by means of two threaded rods and a nut with right-handed and left-handed threads. This preload system was located up the springs and under the seesaw, as Fig. 2 shows. For $0 < R < 1$ (tension–tension cycle), the threaded piece was turned in the direction in which the threaded rods move away and therefore they compress the spring. For $R > 1$ (compression–compression cycle) the threaded piece must turn in such way that the rods come closer and therefore they expand the springs. The preload can be measured directly with the acquisition system, or also, the necessary displacement of the threaded rods can be calculated by means of a simple equation.

Once the preload value was adjusted, the threaded component was blocked by two locking nuts. Every piece had levers to avoid the use of additional tools.

All critical parts of the machine were analyzed in order to corroborate their strength under fatigue load. Soderberg criterion was used to obtain *infinite life* in all the components, where the involved stress states were calculated analytically by means of equations given by elasticity theory, and also numerically using finite elements analysis (FEA).[8] Figure 3 shows an FEA performed in the inferior springs' support.

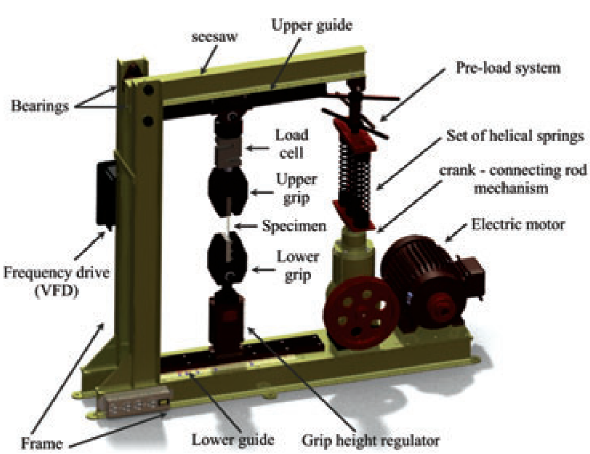

Figure 2 Projected machine sketch.

A load cell was placed at the specimen load line (up to the top grip). Its signal is amplified and then converted to digital by an analog-to-digital (A/D) converter; which allows to visualize the load in real time and to record the information in a file.

Grips corresponding to a resonance fatigue machine (100 kN, *AMSLER* Vibrophone) are in use nowadays. Their design seems to contemplate a suitable pressure distribution on the specimen grip zone, since it possesses a low-stiffness section, capable of bending with the load generated by the pair of vertically displaced screws (see Fig. 4). In this way, the grips put more pressure on the region near to the ends of the specimen, achieving a progressive transition from the uniaxial stress state to the triaxial stress state in the grips, and therefore avoiding to damage the specimen at the grips.

Analysis of Load Variation

As ASTM D 3479/D3479M demands an allowable amplitude load reduction up to 2% during the fatigue test, an analysis of load variation was performed. According to the results, this limit in load variation admits a specimen stiffness reduction up to 10%.

The load on the specimen is proportional to the sum of the displacement of the mechanism that compresses (or expands) the springs (d), plus the preload displacement (x_p), multiplied by the spring elastic constant (k) and by the lever ratio (P):

$$F \approx kP(d + x_p) \tag{1}$$

The latter is defined as the ratio between the horizontal distance from the bearings to the springs, and the horizontal distance between the bearings and the grips. The value of the preload displacement, x_p, represents the displacement that is applied to the springs in order to generate the minimum load value.

The load is diminished due to the specimen stretching, which reduces the effective displacement of the mechanism.

In the case of pure elastic behavior, the stiffness is $S = EA/l_0$, where A is the specimen section, E its elastic modulus, and l_0 the initial length. Note that stiffness can also be influenced by localized damage and that E is heavily influenced by temperature in polymer matrix composites.

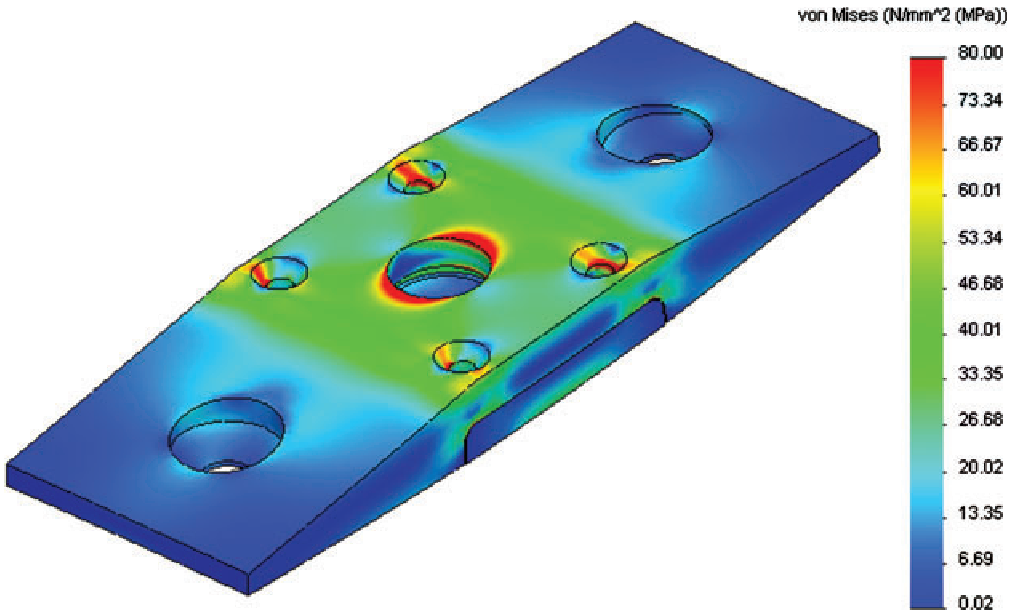

Figure 3 FEA model results corresponding to the lower springs' support.

Figure 4 Grips system used at the moment, which belong to a resonant testing machine.

The spring displacement due to the elastic specimen deformation is:

$$\Delta l = P\frac{F}{S} \tag{2}$$

Then maximum load is:

$$F = kP\left(d + x_{\mathrm{p}} - \frac{F}{S}P\right) \tag{3}$$

The preload displacement can be represented depending on both the stress ratio, R, and the maximum displacement of the mechanism that deforms the springs, d.

From Fig. 5, the preload displacement can be inferred like Eq. 4, where specimen deformation is neglected:

$$R = \frac{F_{\min}}{F_{\max}} = \frac{kx_{\mathrm{p}}}{k(d + x_{\mathrm{p}})} = \frac{x_{\mathrm{p}}}{d + x_{\mathrm{p}}}$$
$$x_{\mathrm{p}} = \frac{Rd}{1 - R} \tag{4}$$

Then, by substituting it in Eq. 3, the load can be obtained:

$$F = \frac{1}{1 - R}\frac{kdPS}{S + kP^2} \tag{5}$$

Equation 5 represents the load on the specimen depending on the stress ratio, the lever ratio, the elastic constant of the spring, the specimen stiffness, and the displacement of the system.

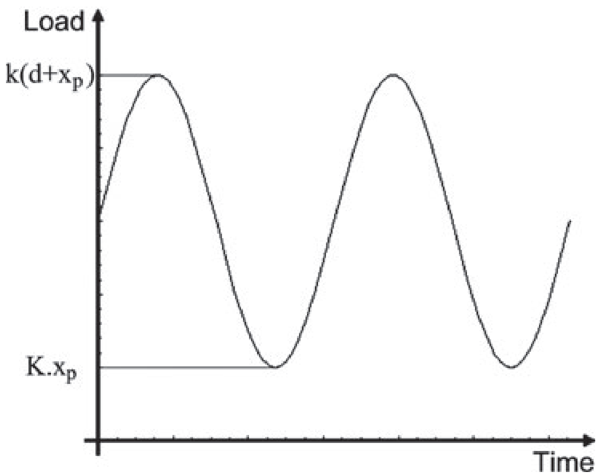

Figure 5 Load cycle applied to the specimen.

The condition of constant load amplitude can be analyzed quantifying the relative variation of the load, expressed as follows:

$$\Delta F = \left| \frac{F_{\text{final}} - F_{\text{initial}}}{F_{\text{initial}}} \right| = \frac{F_{\text{initial}} - F_{\text{final}}}{F_{\text{initial}}} = \left(1 - \frac{F_{\text{final}}}{F_{\text{initial}}} \right) \tag{6}$$

If no slippage occurs between the grips and the specimen conditions, the load value during a test will be affected only by the reduction in the specimen stiffness. To carry out the analysis, the ratio between the final and the initial stiffness will be termed "V":

$$V = \frac{S_{\text{f}}}{S} \tag{7}$$

So the final load will be:

$$F_{\text{final}} = \frac{1}{1-R} \frac{kdPVS}{VS + kP^2} \tag{8}$$

Substituting Eqs. 5 and 8 in 6:

$$\Delta F = 1 - V \frac{S + kP^2}{VS + kP^2} \tag{9}$$

Therefore, the load variation depends on the parameters of the machine and on the properties of the specimen.

To assign values to the product $\text{AE} = Sl_0$, the Sandia database was consulted once again in order to obtain possible values. Figure 6 shows the statistical frequency of the product AE.

The selection of the values that must take the lever ratio, the spring stiffness, and the system displacement, to satisfy the load variation, involves many factors as motor power, springs dimensions, load on the mechanism, machine dimensions, transmission ratio, frequency range, etc. ASTM D

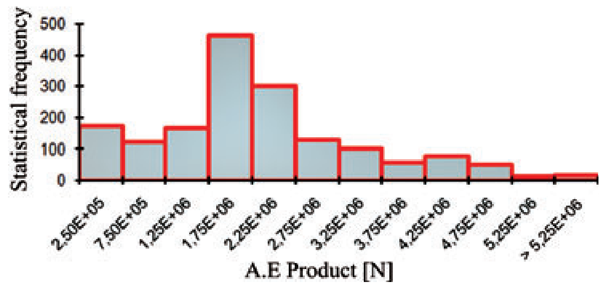

Figure 6 Statistical frequency of the product AE (Area × Young's Modulus).

3479/D 3479M standard indicates that the values of maximum and minimum loads must not change more than 2%; nevertheless, it allows to correct the load and to continue with the test. In order to have a safety margin, a maximum load variation of 1.3% was adopted for the project.

After a time-consuming iterative process, in search of the best configuration, lever ratios varying between 1.4 and 7 were chosen. Spring stiffness values of 15, 30, and 45 N/mm were selected, using one, two, or three parallel springs.

The displacement that compresses the springs is generated by means of a crank-connecting rod mechanism extracted from a go-cart engine with a stroke of 50 mm (displacement of the mechanism, d). This mechanism generates a sinusoidal load cycle, and is powered by an electric motor of 1.5 HP and 1420 nominal rpm by means of belts with a transmission ratio of 3:1 (conductive/driven). The test frequency is controlled by a variable frequency drive (VFD) that supplies energy to the electric motor, and can be set to move the mechanism between 1 and 9 Hz.

Results and Discussion

Figure 7 shows a photograph of the machine that was built with the following dimensions: 1100 × 1040 × 300 mm (width × height × depth). As mentioned previously, its maximum load capacity is 10 kN, and it was designed to perform cyclic controlled load with $0 < R < 1$ (tension–tension), $R > 1$ (compression–compression), and $R < 0$ (tension–compression); although only $0 < R < 1$ load ratios have been used up to the present time, due to the final gripping system for compression tests has not been completed yet.

The machine is very simple, with low-cost pieces which are easy to manufacture. Its simplicity also assures a great reliability, being capable of performing long-time tests (of several weeks) with no maintenance. On the other hand, the configuration of

Figure 7 Photograph of the built machine working with only one spring.

the load cycle is straightforward, and its changes—for R, load amplitude, or frequency variations—can be executed quickly. Only the material elastic modulus and the specimen dimensions must be known in order to calibrate the machine, though it does not mean a limitation, since static tests are usually performed before those of fatigue.

The acquisition system displays the load in real time, which is of great aid to recalibrate the load when specimen slides in the grips, or when a loss of stiffness is generated. It is also possible to count the number of cycles of the test by means of an adequate software. Switches have been placed to halt the machine when the specimen fractures, and software is under development to stop the machine when the load amplitude decreases more than 2%.

Figure 8 shows tested specimens, where failures that occurred out of grips can be observed. The corresponding results are represented as an S–N curve in Fig. 9.

The grips must apply a lateral compression force at the specimen ends to hold the specimen in the grips and prevent glide. The grip design is very important because FRP composites tested in fatigue many times fail at or near them, which invalidates the test. The failure occurs because the stress state changes from a uniaxial tensile state to a triaxial stress state in the grips.

Lowering the tightening force can produce specimen sliding, which causes possible variation in the

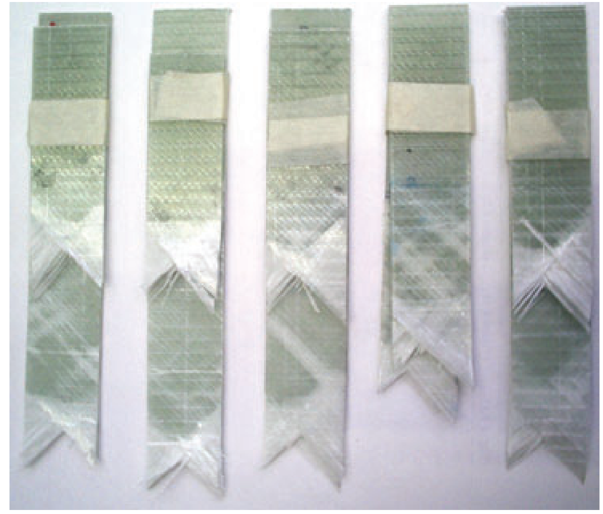

Figure 8 Tested specimens after failure.

axial load applied to the specimen (particularly for this kind of systems where the load is generated by springs). On the contrary, if a large force is applied to gripp the specimen, it can fail at the grips. To find a balance and solve this situation is not an easy task, and, additionally, general solutions do not exist since a good gripping system can be unsatisfactory for another configuration. The used grips have shown an excellent behavior in the tests performed, although new grips have been designed for the further implementation. They have three series of fasteners at each side. The group of bolts farest to the gage can put more pressure than the near series, making a gradual transition of pressure as Fig. 10 shows.

A servohydraulic machine costs over US$100,000, whereas the machine here presented costs less than US$3000. Amortizing both machines in the same period of time implies that building a CAL diagram with the last proposed machine is 97% cheaper. Moreover, as the cost of the proposed machine is very low, the fact that more than one can be built, in order to have a reduction in test time proportional to the number of working machines, can be highlighted.

Conclusions

A fatigue machine for composite materials of low cost and easy to manufacture was built. Its major characteristics are low cost, simplicity, reliability, and small size. These features allow laboratories—especially low-resource ones—to have

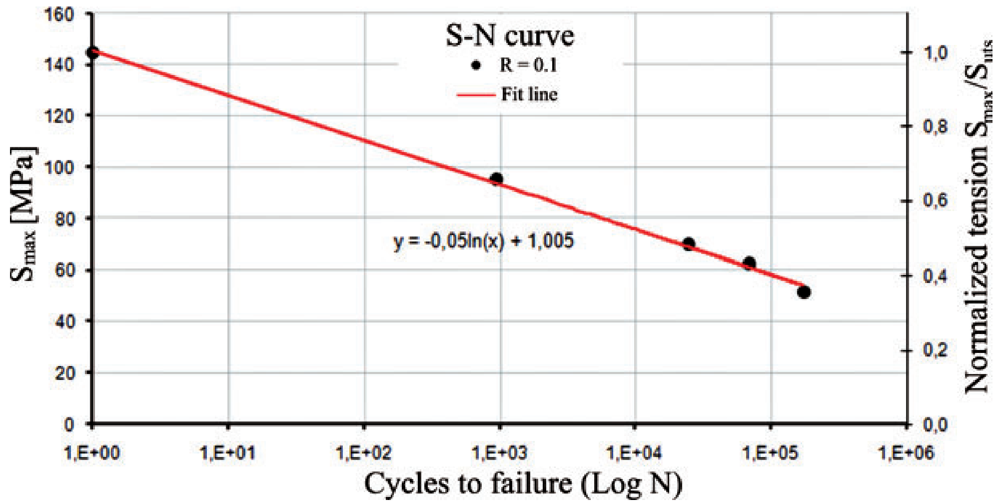

Figure 9 Obtained *S–N* curve.

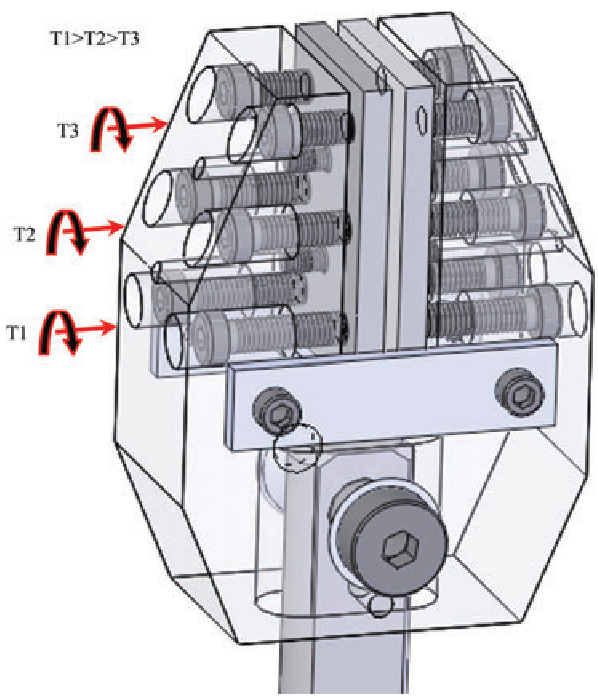

Figure 10 New grips designed.

a series of machines in order to carry out simultaneous tests obtaining results in shorter times.

The performed tests indicated that the machine works correctly, and the tested specimens broke out of grips, showing that the equipment is adequate to test glass fiber reinforced polymer (GFRP) composites.

Acknowledgment

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