# Development of a Testing Approach to be Used in Determination of Load Displacement Loops of Suspension Universal Joints

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*Abstract* - **Ball joints located in the front suspension of the vehicles are universal joints that enable the vertical movement of the wheels to be applied to the rotation from the steering commands. Ball joints work within precise geometric tolerances regarding driving safety and comfort, as they take part in the vehicle's interaction with the driver. The manufacturers determine the force-displacement behavior of the ball joint under load and optimum clearance values are defined in terms of comfort and safety. Within the scope of quality control, the load-displacement cycles are measured by subjecting the samples fixed from the ball joint body to tension-compression cycles in the direction of the rod axis. This study, it is aimed to determine an appropriate methodology to be used in the application of these tests, which are applied in a very narrow displacement range and relatively high force values. Within the scope of the mentioned methodology, a unique test fixture was developed to test the force-displacement behavior of the ball joints in accordance with the operating condition in the suspension system. In this framework, necessary design optimizations and verifications have been applied in order to obtain reproducible measurements.**

*Index Terms* - **Suspension universal joint; load-displacement test; spring travel test; finite element analysis**

## I.INTRODUCTION

Suspension systems are chassis parts which are responsible from transfer forces coming different road conditions and driver commands. Ball joints in suspension systems provide rotational movement and connects control arms to wheels. A typical ball joint cross-sectional image is given in the Figure 1. Ball joints consist of ball stud, steel body and plastic bearing etc.

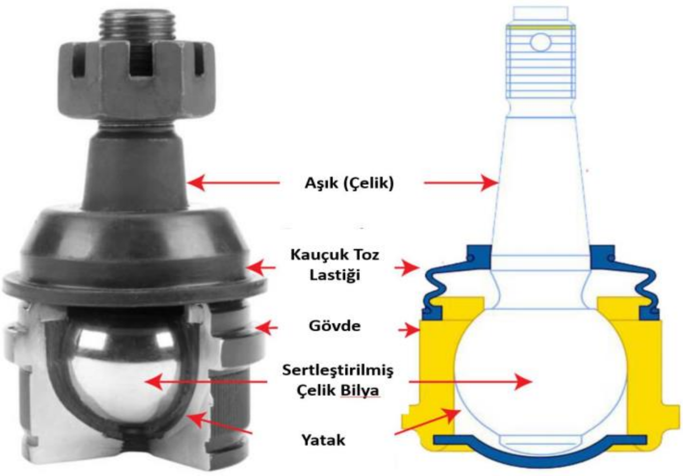



**Figure 1.** Cross-sectional view of a typical ball joint representing main components

Assembly of the ball joints is the one of the most critical factor affecting tolerances and tightness of the sliding surfaces [1]. Especially assembly of ball stud and body has a critical affect on service conditions and overall mechanical performance. Tight tolerances increase the resistance of ball joint to sliding. Loose tolerances, on the other hand, causes considerable loss in vehicle control [2]. This makes it essential to carry out the critical checks to ensure that the ball joints are manufactured and assembled properly. Testing of the ball joints have been subject of the many previous studies in the literature. There are studies researching the interactions between the ball stud spherical tip and the spherical bearing under different loading conditions to determine the friction and wear behavior of the tribological couple. Muscă et al. developed a test setup for testing ball joint friction to investigate how friction force changes with lubrication under multiaxial dynamic loading conditions. Their results showed that, difference between the static friction coefficient and the dynamic friction coefficient increases with increasing normal forces [3]. Baraou et al. studied changes in the properties of the ball joints mounted on the vehicle after a certain driving distance. According to the results of this study, it has been shown that axial clearances of the ball joints force in the pull-out test decreased significantly [4]. Gürsel & Çakır discussed the spinning process that applied while assembly of ball joint. They compared the crush force values which are used in the spinning process and real-time physical process results. The reasons for the deviations between the results evaluated.

Edward et al. presented the how driving safety was affected by number of cycles which are obtained from different loading conditions [5].

| OSKİM KODU | A | B | C | D |
|---|---|---|---|---|
| 50200 | 16 | 13 | R5 | 10,2 |
| 50237 | 16 | 13 | R5 | 10,2 |
| 47203 | 16 | 12,8 | R5 | 16,8 |
| 16201-48201 | 16 | 13 | R5 | 13 |
| 50204 | 16 | 13,2 | R5 | 15 |
| 56200 | 17 | 12,7 | R5 | 12 |
| 50201 | 18 | 15 | R5 | 11 |
| 51222 | 18 | 15,5 | R6 | 10,5 |
| 46201 | 18 | 14,4 | R6 | 17,5 |
| 17235 | 18 | 15,4 | R5 | 18 |
| 51255 | 18 | 15,5 | R6 | 14,7 |
| 16202-48210 | 18 | 15,5 | R5 | 14 |
| 48203 | 18 | 15,4 | R5 | 17,5 |
| 51268 | 18 | 15,5 | R6 | 11,3 |
| 47270 | 18 | 15 | R5 | 19,2 |
| 22227 | 19 | 15,9 | R6 | 17 |
| 50234 | 20 | 17 | R6 | 11,5 |
| 25228 | 21 | 18 | R6 | 14,5 |
| 64213 | 22 | 16,9 | R6 | 15,5 |
| 38213 | 22 | 16,9 | R7 | 20 |
| 51233 | 22 | 17,9 | R6 | 18 |
| 34238 | 25 | 19,7 | R6 | 19,8 |
| 25200 | 16,5/17,5 | 14 | R6 | 13 |

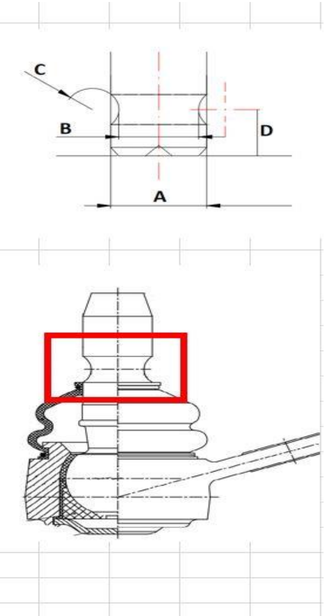



**Figure 2.** Different ball joint stud types using on passenger and commercial vehicles

When the studies are evaluated in general, it is noteworthy that different approaches have been developed for both manufacturing and test conditions of ball joints. The obtained results reveal the approaches followed in ball joint manufacturing significantly affect the mechanical performance. This relation needs to be considered in terms of both ball joint design and optimization of ball joint manufacturing processes. This requirement also brings the approaches and methods applied during the tests of ball joints to an important point. As a matter of fact, if the data obtained during the tests are not reproducible, it is obvious that the reliability of the studies will be questionable to a certain extent. On the other hand, this situation has the potential to cause misleading results in both design and manufacturing process development steps. At this point, the most important problem faced by manufacturers is that the result-oriented standards, directives or regulations for testing ball joint types summarized in Figure 2. Thus, in acceptance criteria, the limits of axial displacement of ball joints under tension-compression loading are expressed. However, as shown in Figure 2, the different geometrical properties of the ball stud and body of the ball joint severely limit the geometric freedom required to load application. As a result of this, the fact that the ball studs are designed with screw connection in some original models, while in some models they are designed with pin connection, the clamping approaches required for testing are limited. This situation also makes it questionable to what extent the test results meet the acceptance criteria.

Based on the problem explained above, the main purpose of this study is to systematically examine the design criteria of the fastening apparatus developed for testing the ball joints connected to the vehicle control arms with pins. In the studies, the geometric design and displacement under load of a clamping apparatus, in which models with different subjects and dimensions on the ball stud can be tested, are determined both computationally and experimentally, and compared with the responses of the same apparatus on the test bench.

## II. Test Procedure, Design and Design Verification

Compression and tensile tests defined for testing ball joints consist of measuring the displacement allowed by ball stud-body assembly according to applied forces. In the tests applied for testing ball joints used in passenger cars and light commercial vehicles, there are forces 5 cycles -2600 - +2600 N in the axial direction and the allowed difference between the maximum displacement of the ball joint in the tension and compression stages of the loading cycles is 0.3 mm. With this feature, the use of a universal tensile-compression testing machine in the application of the tests meets the expectations. In the design processes of the test fixture to be used for fixing the ball joints to the machine, it is prioritized that the ball joint to be fixed to the body with the basic pin should be used in both the body and stud connections, capable of providing the axial direction, and that the stud connection can be provided for different lengths and positions. For this purpose, the design given in Figure 3 was created. In this design, which can be used in a universal tension-pressure testing machine, it was developed to form a shape bond to the pin slots positioned on the stud stems of different sizes, in order to connect the lower jaw group ball joint body with the machine and to work in harmony with ball joint bodies with different heights and base geometries.

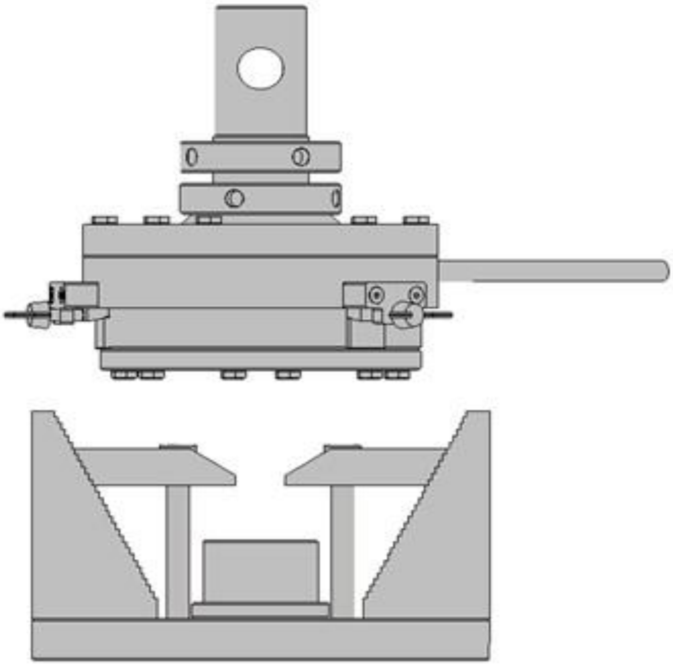


**Figure 3.** Ball joint gap test performed with the parts of the test apparatus developed within the scope of the study (a) to connect the ball joint body in the lower jaw of the tensile-compression test device and (b) to connect the stud stem pin slots on the upper jaw of the tensile-compression test device (sample to be added to the ball joint)

The image of the connection approach developed within the scope of the study, connected to a universal tensile-compression test machine, is shown in Figure 4. In the mentioned connection, the full axial position of the ball joint (stud axis, ball joint body axis and force direction coaxial position) is tested by making use of the coaxiality of the lower jaw and upper jaw provided in universal testing machines.

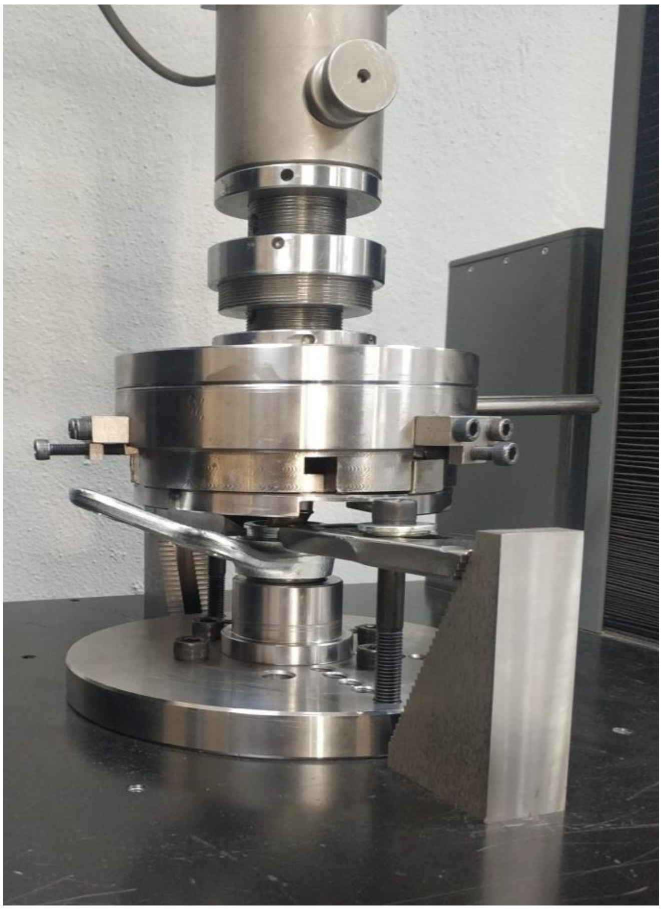


**Figure 4.** Photo showing the working condition in which the apparatus developed in the study is connected to the universal tension-pressure test machine.

## III. MATERIAL AND METHOD

Finite element models prepared with the Siemens SimCenter3D and were used for the design and post-design verification studies of the test fixture. The designs were given in the Figure 3 and the basic operating systematics in Figure 4. These models were solved using the commercial Nastran software on the same platform. Since transitions to the plastic deformation region are not desired during the solution, material-related non-linear behaviors are ignored. With this feature, a steel material with a modulus of elasticity of 207E3 MPa and a Poisson ratio of 0.3 was defined for the entire assembly. A sliding contact algorithm was defined between parts in contact with each other. In this algorithm, the friction between two surfaces sliding on each other was calculated according to Coulomb's law. The friction coefficient for these calculations was defined as 0.11. Triangular pyramid elements were used in the preparation of the finite element model. In the loading analyzes of the model, the product with model number 38254 produced within Oskim Automotive was carried out considering the geometric boundaries (Figure 3). The 38254 model number product was chosen because it has the smallest diameter of the ball joint, the shortest distance from the connection position to the end of the ball joint stem, the highest loading value in terms of test requirements, and finally the product with the smallest tensile-compression displacement range.

While dimensioning the design, both the stresses caused by the test loads and the axial displacement values exhibited by the apparatus under these stresses were taken into account. In this framework, in order to reduce the deformation of the system during the measurement to the lowest possible level, all dimensioning criteria were applied with a stress safety factor of 2.3 times. Accordingly, the upper and lower parts of the system were applied separately with 6000 N distributed static (Figure 5). It was assumed that the force and the drop points of the elements in the loaded regions are equally distributed. Within the said assumption, discontinuities and manufacturing errors that may occur in the geometric properties of the tested product were ignored. In the computational studies, the geometry of the test product was not included in the model. With this feature, the results obtained were purified from the shape changes that occurred in the product during the test (Figure 5). In order to understand the effects of the compression force applied by tightening the bolts during the placement of the sample in the lower part of the apparatus, the force applied to the two loading columns on the lower apparatus was assumed to be equal to the analysis loading. In order, the loading for the lower apparatus was applied to the single fastener with a size of 6000 N (Figure 5). With this loading, it was aimed to investigate the stress state of the connection. In another analysis, the load applied to the loading column was limited to 3000 N. With this analysis, the degree of displacement during the test was determined. In order to fix the models, the movement of all the joint points on the lower surface of the main carrier table in the lower section that comes into contact with the machine was prevented in six degrees of freedom. Similarly, the movement of all the joint points on the inner surface of the connecting pin with the machine in the upper part of the test apparatus was fixed at six degrees of freedom.

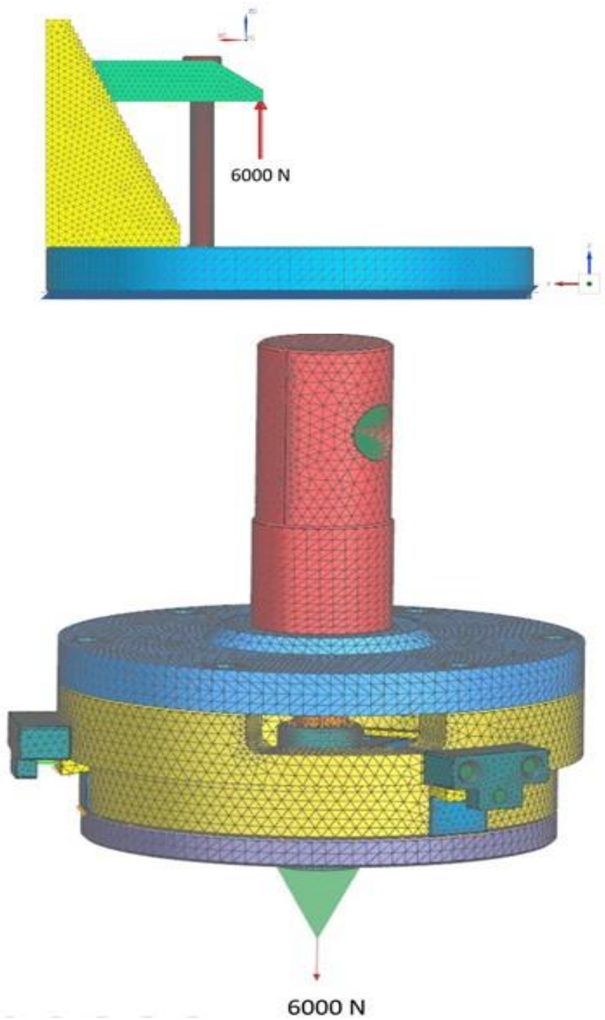


**Figure 5.** Boundary conditions of the finite element model used in the structural analysis of the apparatus.

## IV.RESULTS AND DISCUSSION

### *a) Basis principles of design according to aims*

The lower part of the apparatus is connected to the lower table of the compression tensile testing machine. A surface on which the lower surface of the ball joint sits is designed for centering the ball joint body, and the design of the lower part of the apparatus, which is designed with two tabs fixing the body to prevent its movement under load, has been completed as in Figure 5 below. In the lower group, it is essential to tighten the ball joint body by centering. In this framework, ladder-type holders can be used on the lower table in three positionable ways (Figure 3). With this feature, it is ensured that the part to be tested is loaded symmetrically. Thanks to the central cushioning piece on the lower table, it is possible to test ball joint housing that are not in cylindirical geometry (Figure 3). The cushioning part is used to provide stiffness under compression loads and the retaining claws under tensile loads. With this feature, as can be predicted, the stiffness of the sub-table assembly under tensile loads is lower than that it will exhibit under compression loads. This behavior can be considered as the main performance indicator of the lower jaw group of the system. The relationship of the assembly with the ball pin is provided by positioning the pins arranged at an equal angle around the pin slot in the handle section. In order for the pins to contact the socket together, the cam tool in the mechanism driven by a lever is used (Figure 3). In order to prevent the pins from loosening during loading by creating pressure on the slot, the curved surfaces of the carrier arms to which the pins are attached are connected using cams (Figure 3). The radial movement and fixation of the cams are provided by bolts. On the other hand, an abutment piece that presses on the forehead surface of the ball joint by turning the bolt thread has been added (Figure 3). Thanks to this feature of the upper group of the assembly, the ball joint stem is easily fixed to the assembly and a rigid connection is provided under both compression and tensile loading.

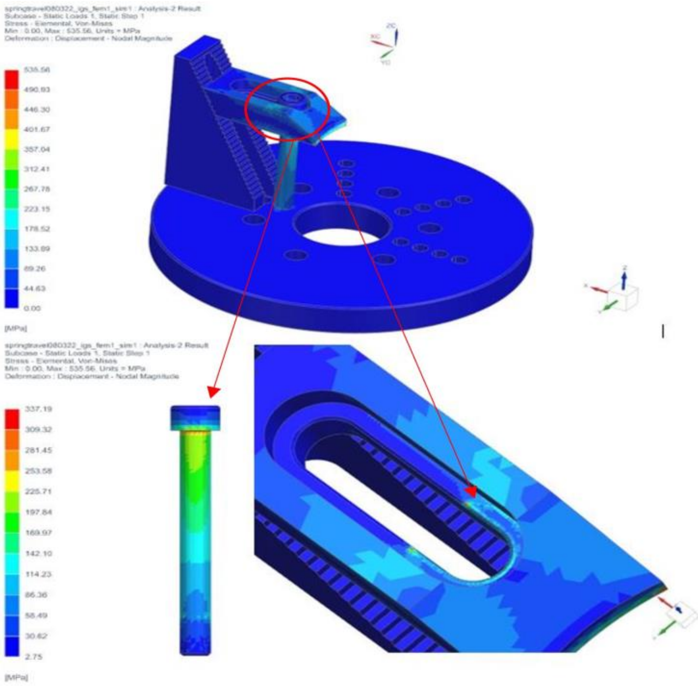
### *b. Structural Analysis of Design*

The deformation distribution under 6000 N load applied to the part of the fixture placed on the lower jaw of the universal tensile-compression test machine is given in Figure 6 and Figure 7 for the lower and upper fixture groups, respectively. Figure 6 shows the Von-Mises (V-M) equivalent stresses caused by the 6000 N force, corresponding to the sum of the clamping and test loads applied to the loading column of the lower section of the test fixture, on the loading beam, loading arm and connecting bolt. According to the results, the greatest stresses under the applied loading occur in the area where the loading beam contacts the part with a size of approximately 290 MPa. In addition, it is noteworthy that high stresses occur locally between the loading beam and the nodal points in the bolt contact area. This situation suggested that the risk of corrosion may arise under high pressures that will occur in the contact areas of the system. Although the stresses are at a level that can be managed within the limits of the structural steels, it is understood that it will be necessary to take measures to carry the abrasion and local contact pressures with the hardening heat treatment. When the displacement level caused by the test force after loading is examined, it is remarkable that the tip of the loading beam produces a displacement of **0.082 mm** under the test force of 3000 N.

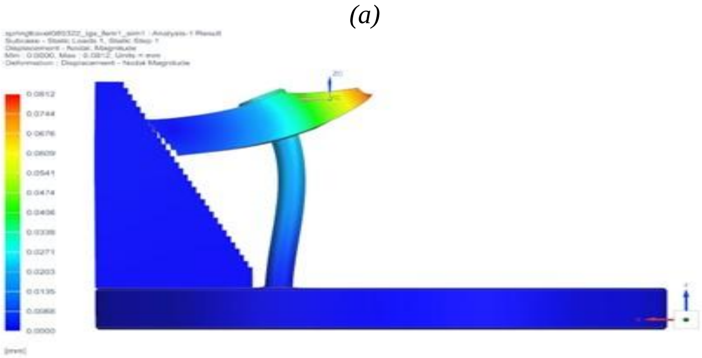
When Figure 7 is examined, it is indicated that the distribution of the stress values exhibited by the upper part of the connection fixture under a force of 6000 N is concentrated in the ball pin slot. It can be said that the maximum value of V-M stresses in the region in question reached 272 MPa. This value is considered to indicate an acceptable level of sizing with stresses. When the displacement values of the upper part of the system are examined (Figure 7(b)), it is understood that the largest displacement occurred on the lateral side of the ball pin handle with a size of **0.05 mm**.

In order to ensure the stability of the upper part of the fixture under load, the load values on the inclined cams used to minimize the sliding movement of the retaining pins in the ball pin socket and to carry the moments caused by the test loads to the body are shown in Figure 8. In order for the said cams to fulfill their duties stated above, it is essential to minimize the geometric gaps by tightening them with bolts (Figure 8). In this framework, the magnitude of the axial force that must be applied in order not to tighten the cams with inclined surfaces is an important operating parameter. In order to determine this parameter, the magnitudes of the axial forces occurring in the bar elements used in the modeling of the bolts between the curved cams and the body were calculated as 312 N. The tightening torque required for an M6 normal threaded bolt to produce the said axial force has been calculated as at least 0.15 Nm. It has been evaluated that if the tightening torque of the bolts to be used for loading the inclined cams is higher than this value, it will not be possible to obtain sufficient operating rigidity, and applying a tightening force more than this value will not provide a significant benefit.

When the displacement values are examined, the largest total displacement value produced by the gripper system occurs at the level of 0.132 mm within the safety factor. This is expected to be around 50-55 µm under normal test conditions. This value represents the magnitude of the axial uncertainty of the developed test fixture.



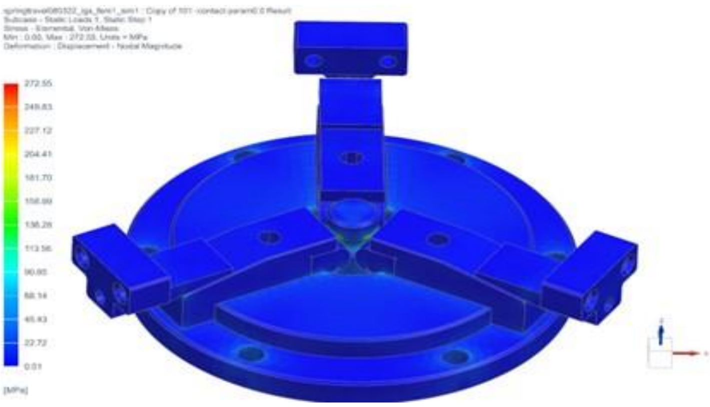
*(a)*

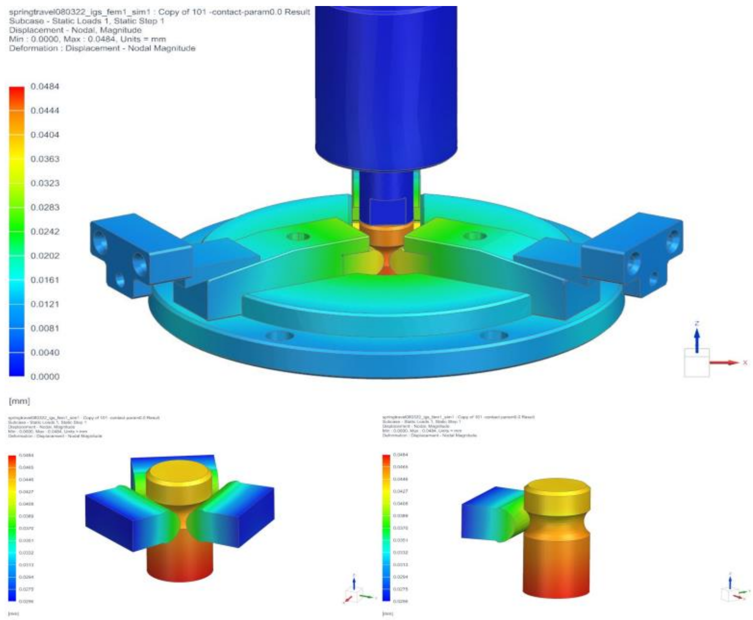


*(b)*

**Figure 6**. (a) The distribution of the magnitude of the stresses occurring in the lower part of the test fixture under the loading and clamping forces (b) The magnitude of the displacement values of the system and the distribution of the components under the test load of 3000 N (the displacements are photographed by enlarging for traceability of the results.)

*(a)*






*(b)*

**Figure 7**. (a) The distribution of the magnitude of the stresses occurring in the upper part of the test fixture (b) The magnitude of the displacement values of the system under test loading and the distribution in the components (displacements are photographed with enlargement for the traceability of the results.)

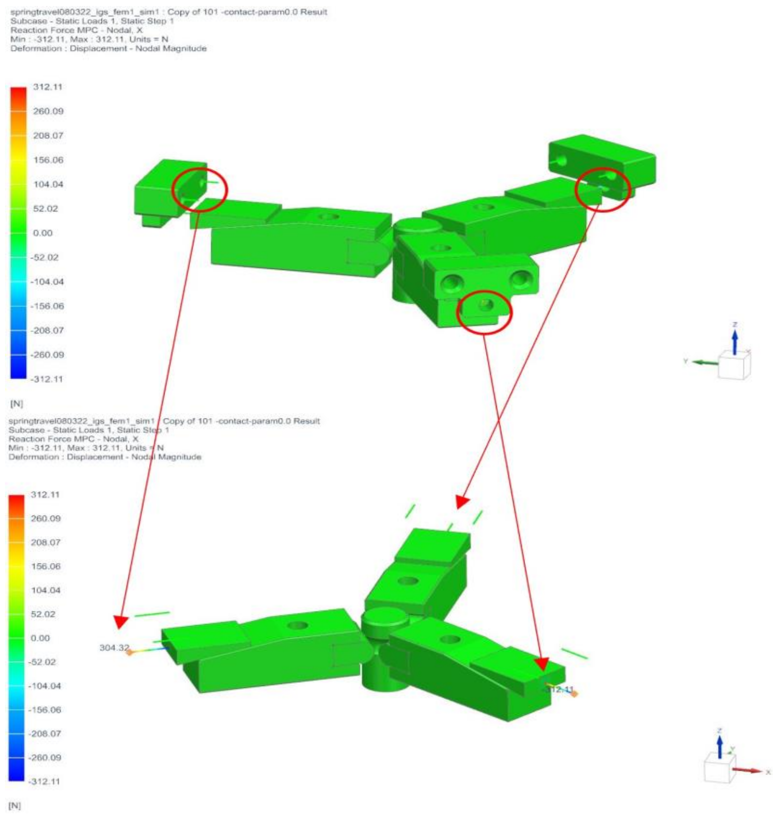



**Figure 8.** The magnitudes of the axial forces to which the inclined cams are subjected

**c. Experimental Validation**

The force-displacement graph obtained from the cyclic tensile-compression tests applied in the developed test system is given in Figure 9. In this way, the approved product, which does not contain any geometrical defects and can be installed in the vehicle, has been tested. Within the scope of the behavior, which is also expressed as spring-travel in its industrial application, the variation of the rigidity of the product under load with displacement is monitored. When Figure 9 is examined, it is seen that the maximum displacement caused by the force of 2300 N applied to the ideally produced product in the tensile and compression directions reaches a value of approximately 0.125 mm.

When the results obtained in the developed test fixture are evaluated, it is desired that the amplitude of the hysteresis in the displacement axis of the product, which is produced in accordance with the geometric and manufacturing infrastructure, in other words, which can be attached to the vehicle, should be at most 0.3 mm. The largest displacement difference exhibited by the hysteresis curve obtained from the test system of the accredited product was measured as 0.23 mm, smaller than this limit. Considering the fixture loss of approximately 0.06 mm in this measurement, it is noteworthy that the total spring travel of the product was determined as 0.17 mm.

Similarly, it is understood from Figure 9 that the test of a product with a deliberate void can be achieved within the expected results. As a matter of fact, it is important that if a gap of approximately 0.07 mm in the total tightness value to be obtained during the closing of the ball joint is left, traceable results can be obtained at the level of approximately 0.05 mm in the jaw and compression regions of the empty movement displacement measured from the system. In this case, it is understood that a faulty product can be detected with the developed fixture.

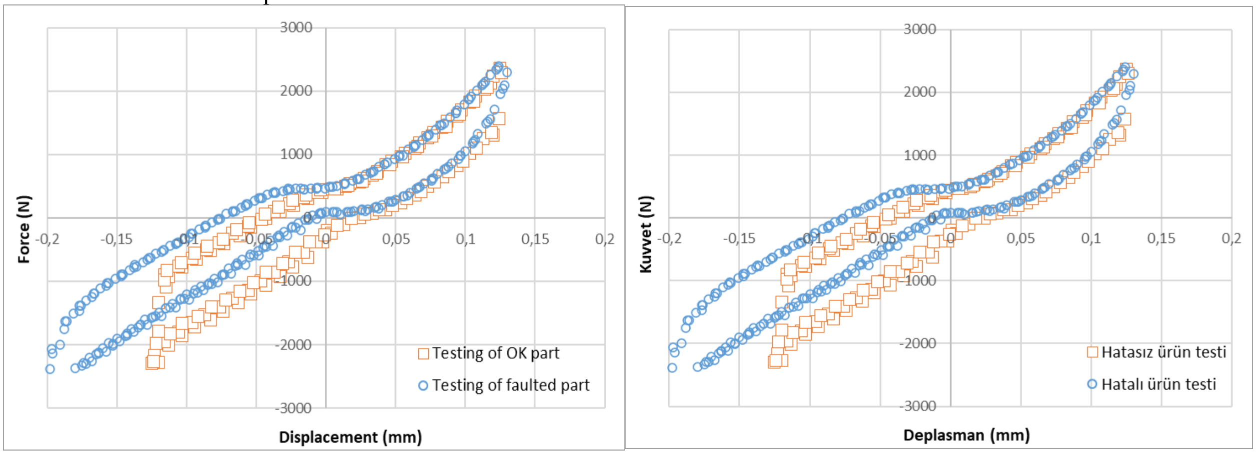



**Figure 9.** Experimental load-displacement curves of approved (OK) and faulted parts.

## V.CONCLUSION

It is aimed to present an experimental measuring fixture developed within the scope of the study, which can be evaluated in testing ball joints with different geometric properties. In the design in question, it exhibited rigidity that can be used to measure the total displacement of 0.3 mm. In addition, the system has demonstrated a design flexibility that allows testing different types and sizes of ball joints. By introducing a repeatability of the system that allows the separation of faulty products in commercial measurements, it has made it possible to implement quality control with high precision.

## VI.ACKNOWLEDGEMENT

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