DESIGN AND CONSTRUCTION OF MECHANICAL DEVICE FOR PERFORMING TORSIONAL STRENGTH TESTS AT TENSILE TEST MACHINE: CASE STUDY AT ARISTOTLE'S UNIVERSITY OF THESSALONIKI

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ABSTRACT: The present paper describes, research and design of experimental device, which transforms a tensile test machine to a torsional test machine, followed be examination of experimental analysis of torsion. The experimental device for torsion assembled at existing tensile test device, transforming translational to rotational motion. With rack and pinion, we achieve the transformation of motion. Design of experimental device has been performed with 3D software. The study of critical design steps such as retaining screws, rolling bearings and the study of torque test specimens follows. For evaluation of the experiments and the constructed experimental device, same experiments performed in exist torque test device. Comparison of experimental results of two torque test devices, existed and constructed, allow us to modify and correct the investigated device.

KEY WORDS: Mechanical device, design, torsion test, FEM analysis.

1. INTRODUCTION

For torque experiments, in Laboratory for Machine Tools and Manufacturing Engineering, construction of mechanical device required. Design that can be assembled to exists tensile test device. The goal is to achieve that with a simple and economical device, at which metallic test specimen with relative small value of breaking point of torsion, for as small as possible strain on tensile device, can be tested. For motion transformation from translational to rotational, a rack and a pinion system has be designed, which is the most common solution for this transformation [1]. A direct motion transmission from rack to pinion without interference from other elements (belts, chains), achieves high performance coefficient (up to 99%) as well as a constant transmission ratio independent of the magnitude of load or rotational speed [2].

Torsion theory originally formulated by Coulomb is applicable only for specimens with circular cross sections. A theory formulated by Saint-Venant, allows analysis of torsion to test specimens with non-circular cross-section [3]. Torsion theory by St. Venant and the theory of the elastoplastic behavior [4] of the material during the torsional stresses were used. The specimens used for torsion

experiments are profiled bars with contoured square edges. Brass materials, like CuZn39Pb3 that achieve maximum breaking stress of 33 to 55 kp/mm2 [5]. Easily appears that the extreme values of stress equal to the maximum value of shear stress and acting at 45 degrees. For less durable at shear, materials, breaking occurs from shear stress at the same direction, with the directions of stresses, while for materials less durable to tensile, like brittle materials, breaking occurs at 45 degrees and the cross section has helical form [6].
The procedure of design and dimensioning of the device taking into account the stresses that receives each component separately is described. Aim is the stresses exerted on the test specimen to exceed breaking point of it while other elements of the structure not reach the limit of their elastic region. Therefore, our main interest is to study the test specimen and the breaking point in torsion.
To evaluate the constructed experimental device, experiments at existing torsion test device (Instron) performed and undergo a comparison in order to extract some conclusions.

2. THEORETICAL BACKGROUND AND CALCULATIONS

2.1. Rack-Pinion

During rack-pinion calculation important limitation constitutes the space in which can be placed. To achieve cooperation between them, diameter of gear must not exceed 110 mm. During torsional test, strain of the tooth is static, for this it was attempted to simulate the situation with the help of finite elements method. The main failures that are investigated are the failure at break of a gear. From analytical design we achieve safety coefficient greater than two. Due to experiments nature, device shows static loads. With FEM analysis, simulation of rack during maximum load is performed (Figure 1). Modeling and calculations of rack-pinion, performed with pre-processing software ANSA [7] and post-processing μETA [8]. The model solved with Abaqus [9].
Emphasis was given on advanced geometry of indentation. Through Matlab software a program was created, which with given module and for a given profile angle of 20 degrees according to DIN 866, creates an advanced curve that is the advanced geometry of the indentation. By introducing the advanced curve in Autodesk Inventor 2013 software the gear wheel and rack were designed. From μETA software we can extract conclusions about the maximum Von-Mises stresses of rack, which occurs breaking. In compression area the indentation stresses are around 180 N/mm2 i.e. 6.5 times lower than the breaking point (1170 N/mm2) of 15CrNi6 material for static stress.
For calculation in recesses by the contact forces diagram created by μETA software, showing the maximum value of force to the contact tooth. The value of force in a tooth is 4942.11 N and according to Hertz equation for linear contact curve and level are 121 kp/mm2, which is compared with the allowable surface pressure for recesses (152 kp/mm2) and is 1.25 times less.

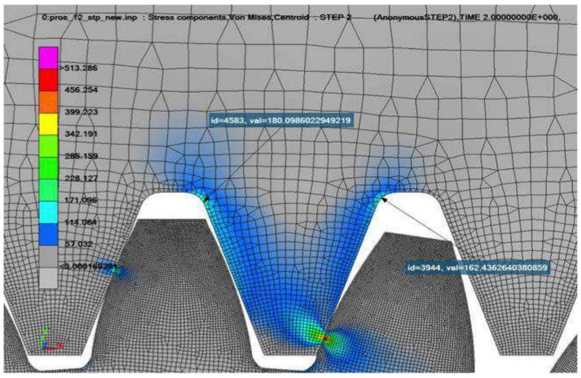


Figure 1. Von-Mises stresses for torque M=29,5 kpm.

2.2. Screws

The gear (pinion) of the construction is mounted through two bearings, which are mounted in two separate housings. Screws to a plate mount both housings. For maximum force of 6000N, applied from the rack, by free-body diagram and the static analysis we calculate the axial forces of the screws. After calculations based on theory of screws [2], M8 screws generate a 1,1 safety factor. M12 screws are selected for safety.

2.3. Ball Joint

When the trunk of a screw consists of fluctuating cross sections like the ball joint for the toothed rack then the calculation must determine the spring constant of each of the screw section. We perform finite elements analysis to solve the model of ball joint in ANSA program, deformation of the elements does not exceed 7μm at maximum value.

The model solved for maximum force of 1000 kp and as is clear by Von-Mises stresses in the following figure 3 (scale: N / mm2) that stresses are nowhere near the yield strength. The ball joint is a commercial car accessory therefore we do not know the material and the properties of the element. Low stresses give confidence to avoid failure probability.

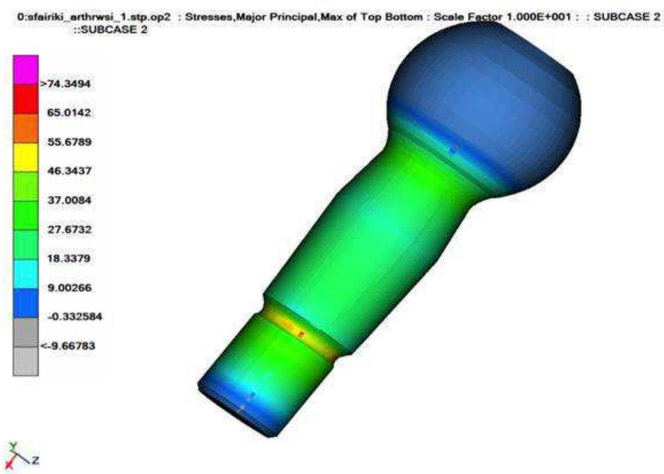


Figure 3. Von-Mises stresses at ball joint.

2.4. Static strength to ball bearings

To ball bearings that are not rotating or rotating with speed less than 20 rpm, computation of static strength is applied. This static calculation relies on comparing static strength Co and the equivalent static load of bearing [10].
Tables give the value of static strength for conventional bearings. The two rolling bearings of the construction are of series 6306 and 4207, which have greater value of static strength than the static load they receive.

3. DESIGN OF TORSION TEST DEVICE

For the design of the construction Autodesk Inventor was used. The model is based on actual dimensions of fixed and movable vice. Only basic and useful geometric dimensions designed mainly for the perception of space in which to assemble the mechanical torsion test device. Figure 4 shows the complete experimental torsion test device. The red color is the gear wheel in which the specimen is attached. Green is the rack that is hingedly connected to moving vice. The torsion shaft is based on two ball bearings in a fixed-mobile mounting.
The bearings mounted on two fixed rings that bind with screws in a specially shaped plate. Above the plate is shaped a centering groove so that the two fixed bearings and the plate to be perfectly aligned. The plate is mounted with screws on the constant vice. Identical a second plate fitted to the moving vice. Through a screw and a flange a ball joint is mounted on which the rack is screwed. The rack is rolling on the gear wheel and on a ring, which holds the rack constantly in engagement with the gear. Figure 5 shows a cross section of connection method of the rack with the moving vice of a tensile machine. A screw penetrates the vice and screwed on a special shaped flange as shown in the sectional shape in black color.

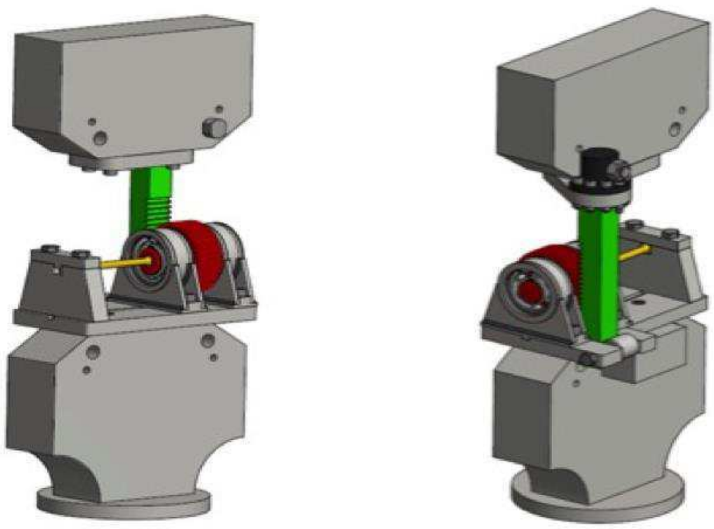


Figure 4. 3D drawings of torsion test device.

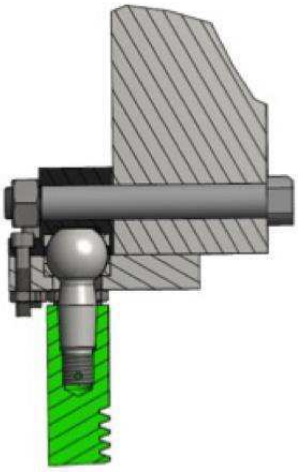


Figure 5. Cross section of connection method

4. EXPERIMETNAL

To evaluate the constructed torsion experimental device, we performed experiments on INSTRON torsion machine.
Through USB port, machine is connected to computer, from where we acquire digital data for torsion, force and steering angle diagrams. Two brass specimens (CuZn39Pb3) were used.
Similarly experiments were contacted in constructed device, which was adapted to Zwick tensile machine of laboratory. In following figure 6, the assembled torque test device is presented. Two plates, which are formatted in milling CNC, are attached on tensile machine vice with four M12 screws each. The two plates are mounting of torsion shaft and rack.
The tensile testing machine (Zwick) of laboratory connected to a computer through which records the forces measured by a sensor that is positioned in the upper vice, as well as the linear displacement of the pinion. Therefore we are able to record force and displacement and thus the corresponding force and displacement diagrams. To convert data of force and displacement in torque and steering angle diagrams we take into account the geometric characteristics of construction.

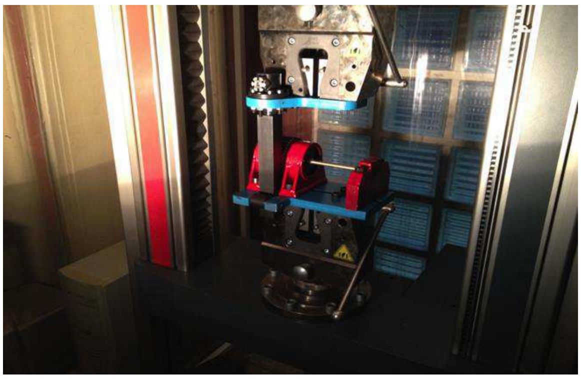


Figure 6. Assembled torque test device.

5. RESULTS AND DISCUSSION

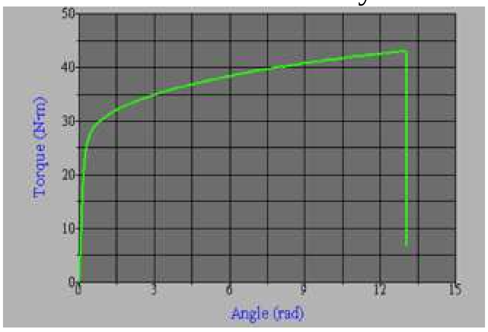
The results of torsion tests performed are torque (Nm) - steering angle (rad) and stress (N/mm2) – strain curve diagrams [11]. Below three figures 7 and 8 where four diagrams, two for each specimen, are presented. Maximum torque distinguished, is the breaking torque of specimens while the abscissa point is the total steering angle until the break occurred.

Torque-steering angle diagrams and stress-strain curve for experiments at constructed test device are shown in Figure 9.

6. CONCLUSIONS

The stress - strain diagram of the specimen until breaking from Figure X shows that the specimen breaking stress is 396 N/mm2. This value is close to the corresponding breaking stress from the results of Instron machine by which we accepted a mean breaking stress of 415 N/mm2. Such differences are not to admit any evidence of an error of torsion test device or any errors in the conduct of experiments. Instead deviations as above, shows the deviations of properties of material as well as geometrical errors in molding of test specimens. Similar differences are observed in the final breaking strain γ of the material 0,48 for Zwick and 0,42 for Instron machine.

Comparing results of experiments conducted in torsion Instron machine with those conducted at constructed device assembled on the tensile machine Zwick, we conclude that constructed test device has accuracy of 96% in terms of finding the torque breaking point of the material.



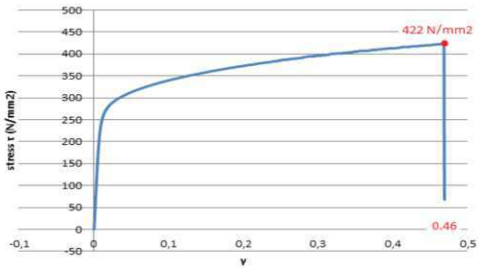
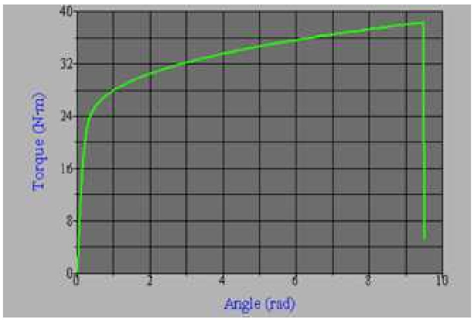


Figure 7. Torque - steering angle and stress – strain curve diagrams to the breaking of the specimen 1.



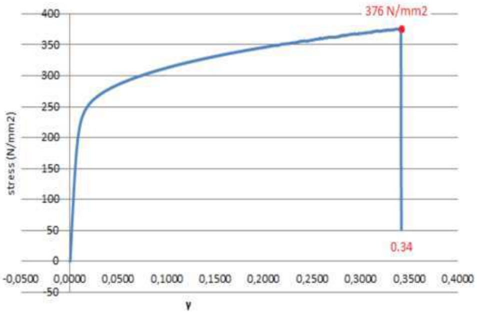
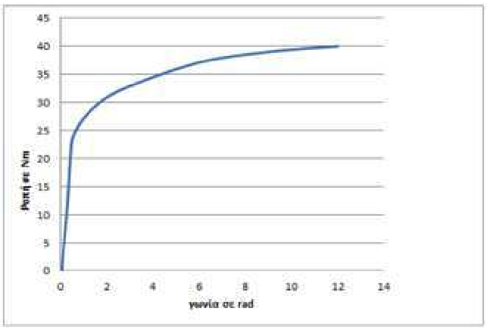


Figure 8. Torque - steering angle and stress – strain curve diagrams to the breaking of the specimen 2



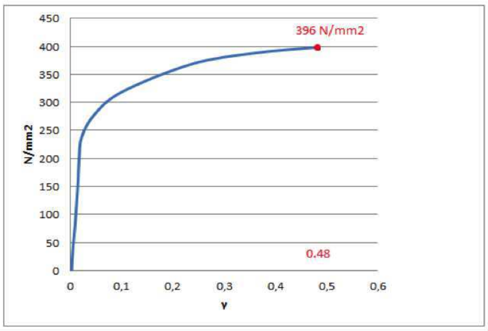


Figure 9. Torque - steering angle and stress – strain curve diagrams to the breaking of the specimen.

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