iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH
IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 11 Issue: XI Month of publication: November 2023

DOI: https://doi.org/10.22214/ijraset.2023.56656

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.538
Volume 11 Issue XI Nov 2023- Available at www.ijraset.com

# Analysis of Torsen Differential Using Autodesk Inventor Nastran

Valentin Mereuta
*Faculty of Engineering, University "Dunarea de Jos", Galati, Romania*

***Abstract: In this work the 3D model of the Torsen differential assembly was designed using Autodesk Inventor 2023 and finite element analysis is performed using Nastran software by applying restrictions, contact conditions and loads conditions. The materials taken for static analysis are chosen in concordance with the literature data. Maximum Von Mises stress values were obtained at the base of the gear tooth. After running the fatigue analysis, the tendency to reduce the life of the gear was observed, by breaking the teeth due to fatigue, which occurs due to repeated bending of the tooth, which results in fatigue cracks and finally tooth breakage.***

***Keywords: Torsen differential, Static analysis, Fatigue analysis, Autodesk Inventor Nastran***

## I. INTRODUCTION

Differentials are used in every car because they are needed in simple operations such as turning. When a car turns, the inside wheel turns more slowly than the outside wheel. Without a differential, there would be wheel slippage or drive shaft breakage due to differences in wheel speed. The differential allows the wheels to rotate at different speeds while transmitting torque to the ground. Differentials also affect vehicle handling, stability and traction. The wheels of a vehicle rotate at different speeds, mainly when cornering. The differential is designed to drive a pair of wheels with equal torque, while allowing them to rotate at different speeds [1]. There are two main types of differentials: open differentials and limited slip differentials. The open differential allows equal torque to both wheels despite the relative speed of the wheels and does not prevent the inside wheel from spinning ahead of the outside wheel as the car begins to understeer and reduced engine torque must be applied to stop wheel spin which, leading to when limiting acceleration [2]. In the limited slip differential (LSD), the side gears are coupled to the rack via a multi-disc clutch, which allows additional torque to be transmitted to the high resistance wheel when the friction limit is reached at the other wheel. Below the friction limit, more torque reaches the slower wheel. If there is no load on one wheel, then no torque is transferred to the other [1]. The limited-slip differential provides no torque except under spring loading, but an additional effect can be obtained by partially applying the vehicle's parking brake when one wheel is spinning, which provides some resistance to increase total torque. [1]. This only works when the handbrake is acting on the drive wheels, as in a traditional rear-wheel drive configuration, with the handbrake released as soon as the vehicle is moving again. A locking differential, such as those using an electrically controlled mechanical system, allows no difference in speed between the two wheels on the same axle when locked. They use a mechanism to allow the planetary gears to lock against each other, causing both wheels to rotate at the same speed, regardless of which has more traction [1]. This is equivalent to effectively bypassing the differential gears entirely. Other locking systems may not even use differential gears, but drive or wheel or both, depending on torque value and direction [1]. An Automatic Torque Biasing (ATB) friction differential, such as the Torsen differential, practices friction between the gear teeth, resulting in more torque being applied to the driven wheel with the most resistance, grip or traction than is available to the other driven wheel when the friction limit in the differential is reached at the other wheel [3]. In an all-wheel drive vehicle, a viscous coupling unit can completely replace a center differential or be used to limit slip in a conventional "open" differential [3]. It works on the principle of allowing the two output shafts to rotate in the opposite direction to each other through a system of plates running in a viscous fluid, often silicone. The fluid allows relatively slow movements of the shafts, such as those caused by cornering, but will strongly resist high-speed movements, such as those caused by turning a single wheel [3]. This system is similar to a limited slip differential. A four-wheel drive (4WD) vehicle will have at least two differentials (one on each axle for each pair of driven wheels) and possibly a center differential to distribute torque between the front and rear axles [3]. The Torsen T-1 differential uses an Invex gear to accomplish all of the aforementioned features. The "T-1" worm gear arrangement allows the differential to move as a solid unit, transferring equal power to both axles when traveling in a straight line and there is no slippage [3].

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.538
Volume 11 Issue XI Nov 2023- Available at www.ijraset.com

The differential housing transfers torque to the worm gear. With the development of modern industry, mechanical devices have expanded their dimensional range from subminiature units to large-scale units. Vibration and impact analysis of gears has been studied with great attention, especially in large-scale devices where gears play an important role.

A number of papers are present in the scientific literature on gear stress and strain analysis using the FEM method since 1970. For example, Chabert [4] analyzed the stresses and strain of spur gear teeth by the two-dimensional finite element method with a single tooth model.

Ramamurti and Ananda Rao [5] calculated the dynamic stress variations with time using two-dimensional finite elements and a concept of cyclic symmetry.

Wallace and Seireg [6] studied the stress variation with time under impact stress at three different points of the profile. Sahir Arilan and Kaftanoglu [7] studied the dynamic load and stresses at the tooth base in the case of gears with straight teeth, using the mesh stiffness concept.

Vijayarangan and Ganesan [8] studied the dynamic analysis of stresses in a spur gear under a moving load and three-dimensional finite element impact conditions.

Gawande et al. [9], designed by classical calculations the pinion gear using the given data and modeled the gear in PRO-E (wildfire4.0), which is an excellent CAD software, then transferred to IGES format and exported the model to the software analysis tool ANSYS 13.0. and studied the results obtained for stresses and strains.

## II. TORSEN DIFFERENTIAL DESIGN

Solid modeling of the Torsen differential assembly was done using Autodesk Inventor version 2023 with the literature data.

The solid model of the Torsen differential are shown in Fig. 1.

Fig. 1 A geometrical model of the Torsen differential

## III.DESIGN AND SIZING OF MACHINE PARTS

The design and dimensioning of machine parts was carried out with the special module included in the Autodesk Inventor program, Power Transmission, and aimed to dimension and verify certain parts that were obtained through automatic software generation. In Fig. 2 shows the summary of the analysis for the parallel key assembly between the differential shaft and the hub of the bevel gear.

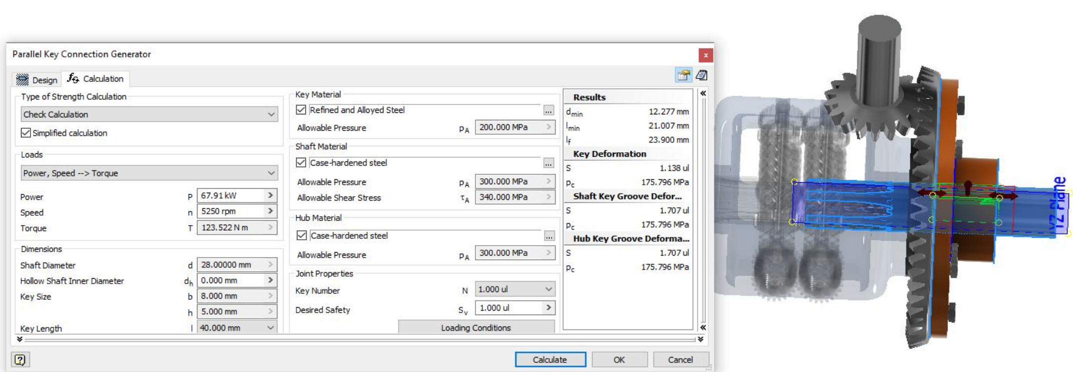
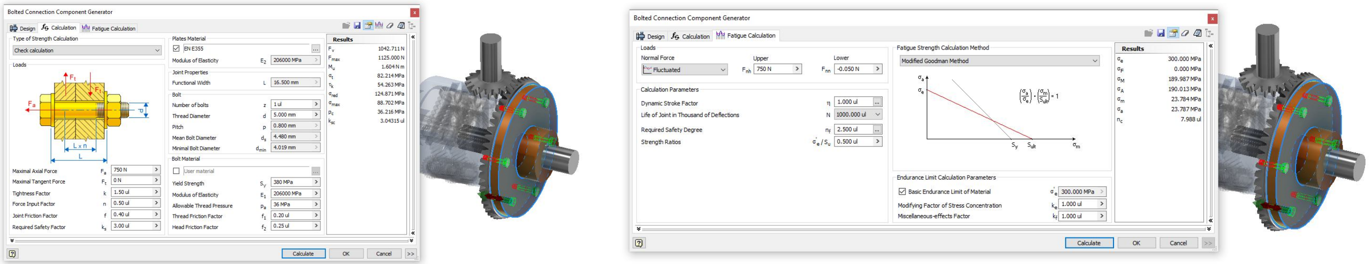



Fig. 2 The analysis for the parallel key assembly

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.538
Volume 11 Issue XI Nov 2023- Available at www.ijraset.com

Also during this stage of the work, the fastening screws of the bevel gear flange were dimensioned and checked. The power screws have been selected from the Content Center. In Fig. 3.a shows the static analysis of the assembly, and in Fig. 3.b fatigue analysis of threaded assembly.



a) b)

Fig. 3 The analysis for the of threaded assembly

## IV. STATIC ANALYSIS

The finite element analysis was carried out in order to determine the state of stresses and strains during the maximum stress of the shaft. Finite element analysis was performed with the dedicated Autodesk Inventor Nastran module included in the Autodesk Inventor software.

In order to obtain the best possible results and at the same time in order to be able to run the simulation with as few system requirements as possible, only the subassembly consisting of the bevel gear and the bevel gear flange was maintained. The static analysis is chosen as the type of analysis.

### *A. Choosing Material*

In order to choose the materials, it was necessary to specify the materials for each component of the assembly one by one. The materials with the properties specified in Table 1.

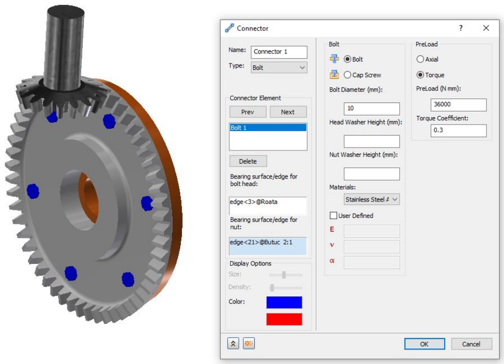
TABLE I

MATERIAL PROPERTIES

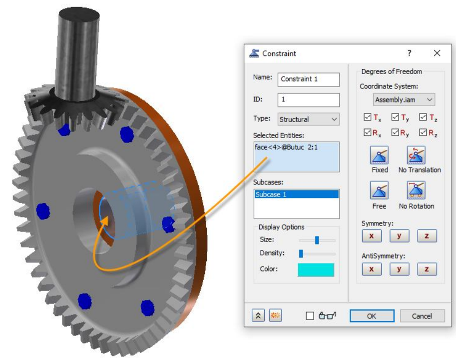
| Parameters | *Steel, High Stength, Low Alloy* | *Stainless Steel AISI 310* | *Iron Gray Cast ASTM A48 Grade 20* |
|---|---|---|---|
| Young's Modulus [MPa] | | | |
| Poisson's Ratio | 200000 | 199947 | 81495.6 |
| Shear Modulus [MPa] | 0.29 | 0.29 | 0.23 |
| Mass Density [$Ns^2/mm^4$] | 128700 | 77220 | 33025.7 |
| Tensile Strength [MPa] | 7.85E-09 | 8E-09 | 7.395E-09 |
| Yield Strength [MPa] | 448 | 399.394 | 179.263 |
| Thermal Expansion | 275 | 248.210 | 151.684 |
| Coefficient | 1.2E-05 | 1.170E-05 | 1.3E-05 |
| Thermal Conductivity N/(sec | 4.7E+01 | 1.62E+01 | 4.804E+01 |
| $^oC$) | 4.8E+08 | 5E+08 | 4.5E+08 |
| Specific Heat $mm^2/(sec^2\ ^oC)$ | | | |

### *B. The Restrictions Condition*

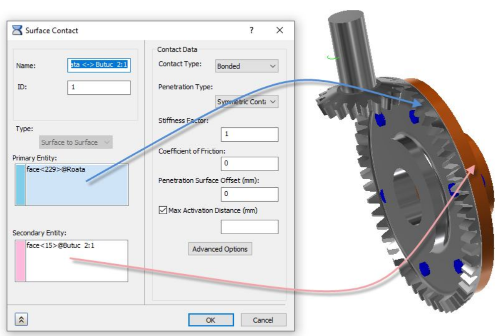
Another step necessary to obtain the state of stresses and strains was the imposition of restrictions and stresses. Autodesk Inventor Nastran offers the possibility of imposing bolt assembly connections, bolt type connector, Fig. 4.a, specifying the elements between which the assembly is made and the type of assembly. The rotation of the gear wheel was blocked and the torque was imposed Fig. 4.b. At the same time, the conditions for achieving contact between the surfaces of the toothed wheel and the pinion were imposed, Fig. 4.c.



a) Conector type bolt



b) Constrain



c) Surface contact

Fig. 4 Restrictions

### C. Load Conditions

The next step was to apply loads to the assembly. The paper analyzed the Renault Sandero model where the maximum torque is 206.78 [Nm] at a speed of 2536 [rot/min], and the maximum power is 67.91 [kW] which is reached at a nominal speed of 5250 [rot /min]

Determination of the calculation moment for cars with a motor axle, Mc, is considered the maximum moment of the engine $M_M$, reduced to the gear calculated by the relation:

$$M_c = M_M \cdot i_{cv1} \cdot \eta_{cv1} = 614.873 \text{ [N·m]} \quad (1)$$

where:

$M_M$ – maximum moment;

$i_{cv1}$ – the transmission ratio of the first gear

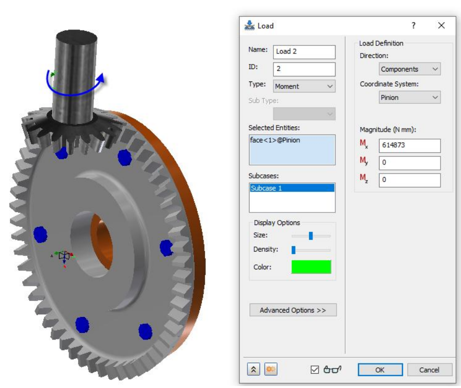


Fig. 5 Load Conditions

### *D. Generate Meshing*

To generate the mesh, the automatic generation mode was used with parabolic element order, the parts of the Torsen differential being meshed into 21392 elements and 38585 nodes.

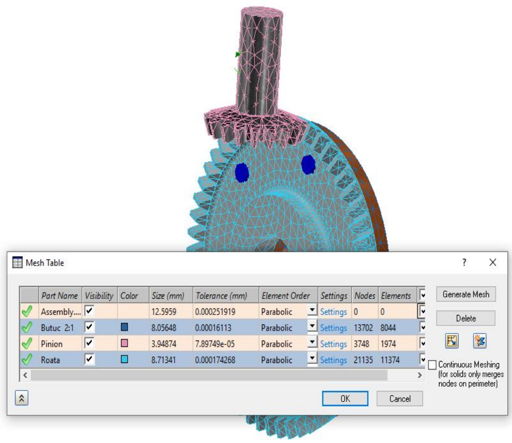
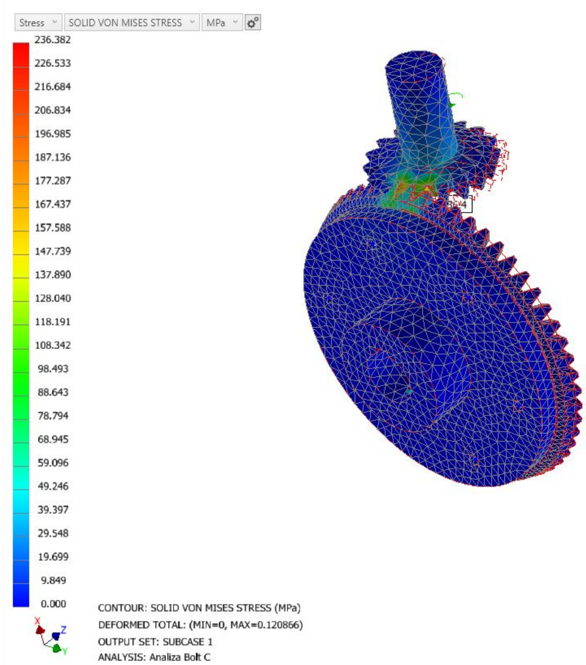



Fig. 6 The assembly meshing

### *E. Static Analysis Results*

The following figures show the results obtained from the FEM analysis for the bevel gear assembly. After running the analysis, Von Mises stress values, (236.362 MPa), were obtained, below the admissible limit of the material, values recorded at the base of the tooth, Fig. 7.



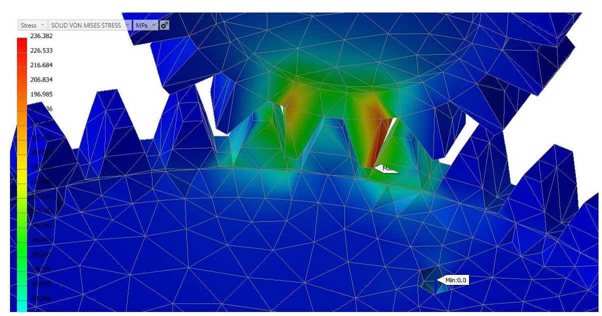



Fig. 7 Von Mises stress – steel rim

Fig. 8 Total deformation - steel rim

Fig. 7 Tire pressure

The following figure show the results obtained, following the FEM analysis, for the maximum deformations (0.121 mm), Fig. 8.

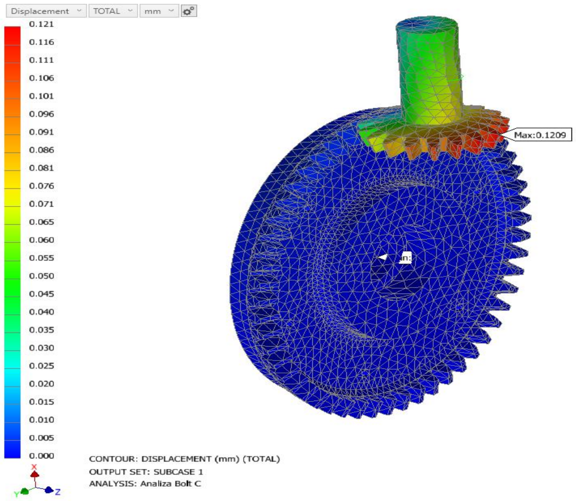



Fig. 9 Von Mises stress – aluminum rim

After running the fatigue analysis, the tendency to reduce the life of the gear was observed, by breaking the teeth due to fatigue, which is the main cause of the removal from use of gear wheels made of hard materials.

This phenomenon is due to the repeated bending of the tooth, which leads to the formation of fatigue cracks that eventually lead to tooth breakage. The crack usually starts in the connection area of the tooth, as can be seen in fig. 4.10, where a strong stress concentration occurs.

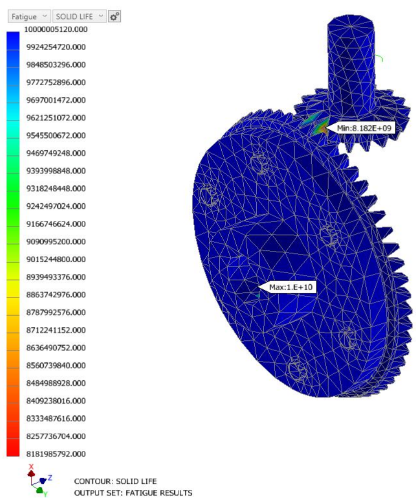



Fig. 10 Fatigue analysis

## V. CONCLUSION

The purpose of this paper was to carry out a study of the behavior under the action of loads of a Torsen differential. In order to determine the state of stresses and deformations, the 3D models of the component parts of the differential were made with the help of the Autodesk Inventor program version 2023. The finite element analysis, both static and fatigue analysis, was performed with the dedicated Autodesk Inventor Nastran module. After running the static analysis, Von Mises stress values were obtained, below the admissible limit of the material, values recorded at the base of the gear tooth. After running the fatigue analysis, the tendency to reduce the life of the gear was observed, by breaking the teeth due to fatigue, which is the main cause of the decommissioning of gear wheels made of hard materials. Fatigue occurs due to repeated bending of the tooth which results in fatigue cracks and finally tooth breakage. After the analysis, it was observed that the crack appears in the connection area of the tooth due to stress concentrators.

## REFERENCES

[1] J. Koo, C. Choi, Y. Huh and C. Seok, "Development of a Hydraulic Limited Slip Differential System Using a Pressure Generator," Int. Journal of Automotive Technology, pp. 323-327, 2008.
[2] D. A. Crolla, Automotive Engineering Powertrain , Chassis System and Vehicle Body, Butterworth-Heinemann, imprint of Elsevier, Linacre House, Jordan Hill, Oxford OX2 8DP, UK, 30 Corporate Drive, Suite 400, Burlington, MA 01803, USA, ISBN: 978-1-85617-577-7, 2009.
[3] R. Morselli, R. Zanasi, G. Sandoni, "Detailed and Reduced Dynamic Models of Passive and Active Limited-Slip Car Differentials," Mathematic and Computer Modelling of Dynamical Systems, vol. 12, 347–362,.
[4] G. Chabert, T. D. Tran, R. Mathis, "An evaluation of stresses and deflection of spur gear teeth under strain", ASME Journal of Engineering for Industry, 96, 85-93, 1974.
[5] V. Ramamurti, Rao M. Ananda, "Dynamic analysis of spur gear teeth", Computers and Structures, Vol. 29, No. 5, pp. 832-843, 1988.
[6] D. B. Wallace, A. Seireg, "Computer simulation of dynamic stress, deformation, and fracture of gear teeth", ASME Journal of Engineering for Industry, 95, 1108-1114, 1973.
[7] M. A. Sahir Arilan, B. Kaftanoglu, "Dynamic load and root stress analysis of spur gear", Annals of the CIRP, Vol. 38, No. 1, pp. 171-174, 1989.
[8] S. Vijayarangan, N. Ganesan, "A study of dynamic stresses in a spur gear under a moving line load and impact load conditions by a three-dimensional finite element method", Journal of Sound and Vibration, Vol. 162, No. 1, pp. 185-189, 1993.
[9] S. H. Gawande, S.V. Khandagale, V. T. Jadhav, V. D. Patil, D. J. Thorat, "Design, Manufacturing & Analysis of Differential Crown Gear and Pinion for MFWD Axle", IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) ISSN(e) : 2278-1684, ISSN(p) : 2320–334X, PP : 59-66, 2013.