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DACIA LOGAN LIVE AXLE OPTIMISATION USING COMPUTER GRAPHICS

Abstract: *The paper presents some contributions to the calculus and optimisation of a live axle used at Dacia Logan using computer graphics software for creating the model and afterwards using FEA evaluation to determine the effectiveness of the optimisation. Thus using specialized computer software, a simulation is made and the results were compared to the measured real prototype.*

Key words: *Live axle, optimisation, computer graphics, simulation, calculus.*

1. INTRODUCTION

The role of front axle is to transmit the forces and moments that occur from interaction between the wheels and the road allowing changing the direction of the vehicle.
At cars with front propulsion, the live axles transmit also the torque from the gearbox to the wheels. At the front live axle, we can distinguish torque transmission mechanisms and the wheel guide mechanism.
Structurally, the transmission to the front wheels comprises angular couplings, axial clutches and axial angular- couplings.
If the case of Dacia Logan the transmission uses a double tripod obtained by, coupling serially an angular-axial tripod with spherical rollers with a tripod coupling angularly connected to the shaft.
The guidance mechanism is with quadrilateral articulation. The wheel bearing uses two axial bearings mounted on the wheel hub and the wheel is driven using shaft grooves.

2. PRESENTATION AND ANALYSIS OF FRONT LIVE AXLE STRESS

Consider the loading forces scheme of the front train shown in Figure 1.

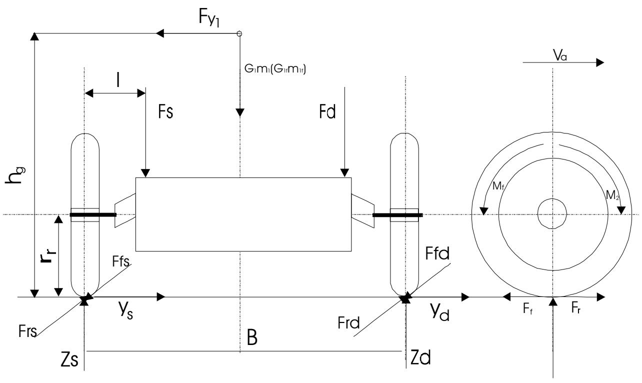

Fig. 1 Loading forces of the front train

The scheme for calculating the live axle is shown in Figure 2. Calculation of strength forces in live axle will be taken considering three modes of moving the car: in braking mode, skidding regime and crossing obstacles regime[3],[6]. If the front end is driving, the tangential reaction force $F_{rs} = F_{rd}$ is always smaller than the braking tangential reaction force $F_{fs} = F_{fd}$ as m1 <m_{1f}:

$$F_n = \frac{m_1 \times G_1}{2} < \frac{m_{1f} \times G_1}{2} = F_{fs} \qquad (1)$$

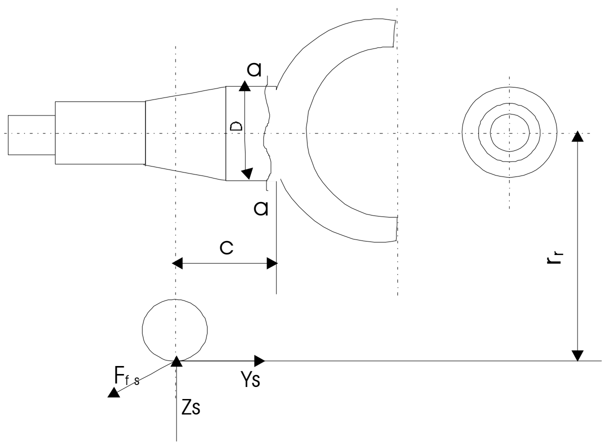

Fig. 2 Live axle geometry

Where F_{fs} – the drive force on the wheel
Z_S – vertical wheel reaction
Y_S – lateral reaction of the wheel

2.1. Calculus of loading forces in braking regime

In this mode, the live axle is calculated in the bending section under the action of vertical reaction Zs = Zd and braking force Ffs = Ffd.

The bending moment is:

$$M_i = c\sqrt{Z_s^2 + F_{fe}^2} = c \times Z_s \sqrt{1+\varphi^2} \qquad (2)$$

The tensile unit stress:

$$\sigma_i = \frac{M_i}{W_i} \qquad (3)$$

Where $W_i \cong 0,1 D^3$ for the driving live axle;

$$W_i = \frac{\pi\left(D^4 - d^4\right)}{32D} \quad (4)$$

2.2. Calculus of loading forces in skidding regime

This regime is characterised by the action of Z_s, Z_d, Y_s, D_d forces.

Considering skidding to the left, the live axles will be solicited in the a-a section, by the following torques:

For the left axle: $M_{is}=Z_s \ x \ c - Y_s \ x \ r_r$ (5)

For the right axle: $M_{id}=Z_d \ x \ c + Y_d \ x \ r_r$; (6)

Unit bending stress will be:

$$\sigma_i = \frac{M}{W_i} \quad (7)$$

Where M is the biggest between M_{is} *şi* M_{id}

2.3. Calculus of loading forces in crossing obstacles regime

In this regime the live axle is solicited by bending in *a-a* cross section by the vertical reaction force $Zs = Zd$, namely: $M_i=Z_s \cdot c$

Where c is the distance to the soliciting plane, (see figure 2). It results an effective unit stress:

$$\sigma_i = \frac{M_i}{W_i} \quad (8)$$

Recommended materials used to manufacture live axles must have maximum bending effort limit values: σ_{ai}=4500...6000 daN/cm^2 [5].

3. MODELLING, AND FEA SIMULATION OF THE LIVE AXLE

In order to make a behaviour simulation of the axle using the finite element method we must take the following steps [2]:

- Geometric Modelling
- Meshing.
- Specification of loads and boundary conditions
- Materials choosing
- Running the calculus
- Post-processing data

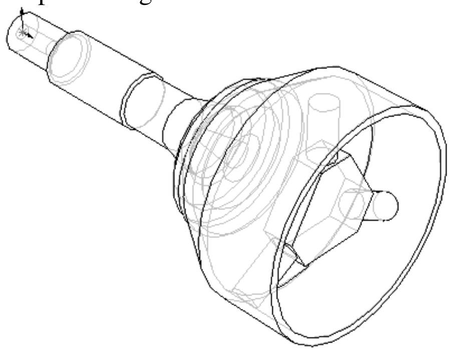

Fig. 3 Live axle 3D model

3.1. Geometric modelling

Although the live axle is, actually, a welded assembly consists of three parts; it modelled as a single part (see Figure 3 and 4).

In addition, the rubber seal and a series of fillets were suppressed considering that their presence does not significantly influence the outcome of the analysis.

In these areas, the meshing is particularly difficult and requires computer effort and especially additional processing time.

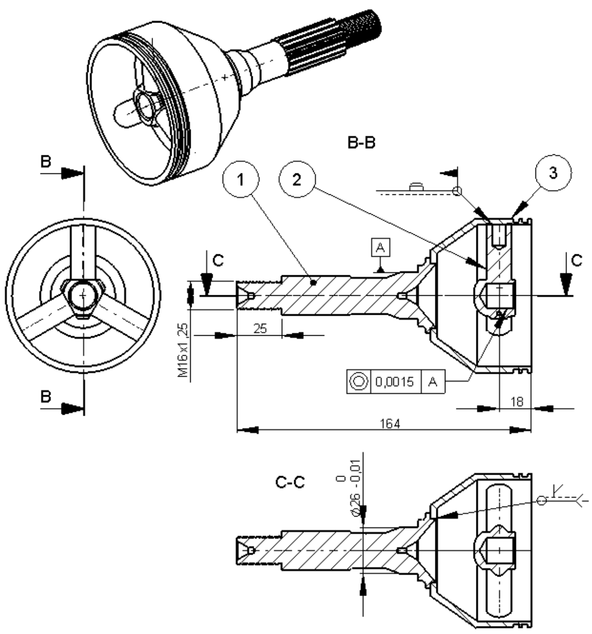

Fig. 4 Original live axle

The CSG (Constructive Solid Geometry)Tree of the model was developed mainly by printing a revolutions a plan profile (Base-Revolve), extrusion borehole (Cut-Extrude), choosing a plan convenient (Plane1) of that one of the arms extruded tripod (Boss extrusion) after that it has been multiplied in a circular pattern (CirPattern1) to obtain the other two arms[1].

The construction of the central portion of the tripod is done by extrusion (Boss Extrude2) from a plane (Plane2). See figure 5 for the CSG tree.

Bevel is applied and most important connections (Cahmfer1 and Fillet1).

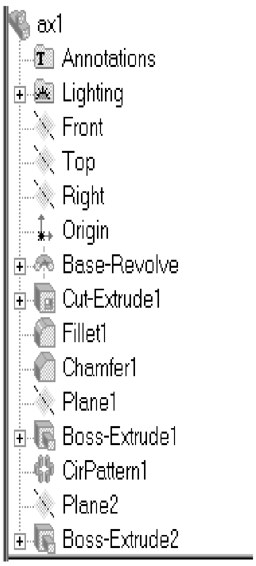

Fig. 5 CSG tree

3.2. Meshing

Meshing geometric consists in applying on the support of the part a network of interconnected nodes. Depending on the number of nodes and network structure formed, it approximates more or less closely the geometric model.
Mesh problem is one of compromise because the more the nodes they approximate better the reality, but in the same time increases exponentially the processing time, prohibitive for most PC computers.
In this case it was used an automatic adaptive mesh to optimize the bandwidth matrix.
The elements used were solid brick type with curved edges, which requires three refinements to stay within the 10% error originally required [4].
When applying the network it were automatically, checked and corrected the following parameters defining the structure(see table 1):

Table 1

Mesh final imposed characteristics

Distortions	< 0,1%	Deformation	< 0,2 %
Interior angles	< 90 ^{0}	Middle node migration	< 50 %

After meshing elements were obtained with 19 407 11 401 knots with an average size of 2.97 mm elements.

3.3. Loads and boundary conditions

As stated in literature [7], the calculus of the strength of the knuckle is made for three modes of displacement of the vehicle: brake mode, skid mode and crossing obstacles.
At Dacia cars, because the front train is driving the front axle tangential reaction forces Frs = Frd are always smaller than the reaction tangential braking forces F_{fs} = F_{fd} as m_1 <m_{1f}:

$$F_{rs} = \frac{m_1 \cdot G_1}{2} < \frac{m_{1f} \cdot G_1}{2} = F_{fs} \qquad (9)$$

This analysis will consider the worst possible case, namely braking skid. The live axle is not take part in the suspension of the front train so, passing over the obstacles do not substantially affect its behaviour [7].

3.4.Solving the problem

We imposed the calculation of the following parameters:
• Deformations
• Tensions
• Efforts
• Reaction Forces
It has also been chosen by adaptive calculation using the h factor, which refines the network from nodes error.
This is an iterative process that stops at the maximum prescribed number of iterations or if the maximum number of errors is exceeded.

3.5. Data post processing

If meshing error falls within the permissible may proceed to verifying employment tensions in yield analysis evidenced a scale showing a percentage ratio compared to the yield strength of the material.
Any overtaking, how small, invalidates the results of the final analysis coming out of the presumptive framework of linear elasticity.
In this case, it is advisable to adopt one of the following solutions:
• Changing the material chosen for the part
• Verification of load values and/or boundary conditions
• Changing the geometry of the model
• Use a program of nonlinear analysis
In Figure 6 shows that the maximum combined Von Mises stress (after breaking theory IV) does not exceed the yield limit, it is even almost half of it

$\sigma_{Von\ Mises.max}$ = 535.314 N / mm2

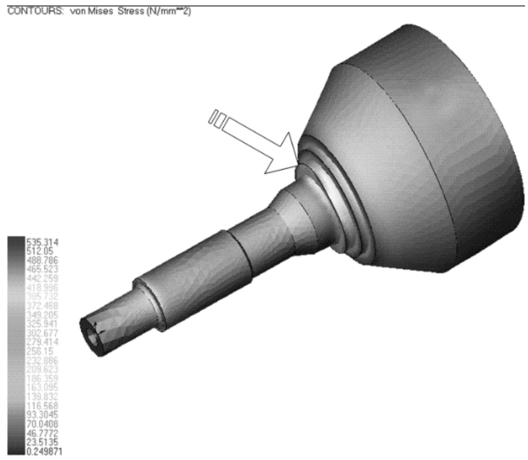

Fig. 6 Von Mises Stress – 535N/mm2 – max

Figure 7 shows that the nodal displacement of the nodes are extremely small being distributed around the cup circumference.

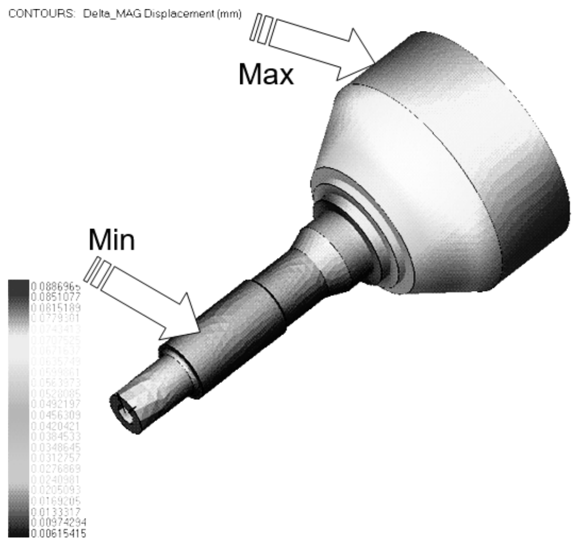

Fig. 7 Nodal displacements

This may be the sign of deterioration of the live axle housing that is the frequently the cause for tripod breaking from one of the three weldments. The nodal displacement values vary between:

Delta_MAGmax = 0,08869 mm (fig.7 - Max)
Delta_MAGmin = 0,00615 mm (fig.7 - Min)

4. OPTIMISATION OF THE LIVE AXLE

The results of the simulation using the finite element analysis method that shows that the axle remain in the linear elasticity boundaries even upon the maximum loads.

Therefore, we tried to attempt an optimization solution by practicing a Ø8 mm hole through the central rod shown in figure 8, 9 and 10.

Also it was reinforced the shoulder area, by making smooth transitions between the cup and the stem where in the initial analysis were observed increased Von Mises stress values.

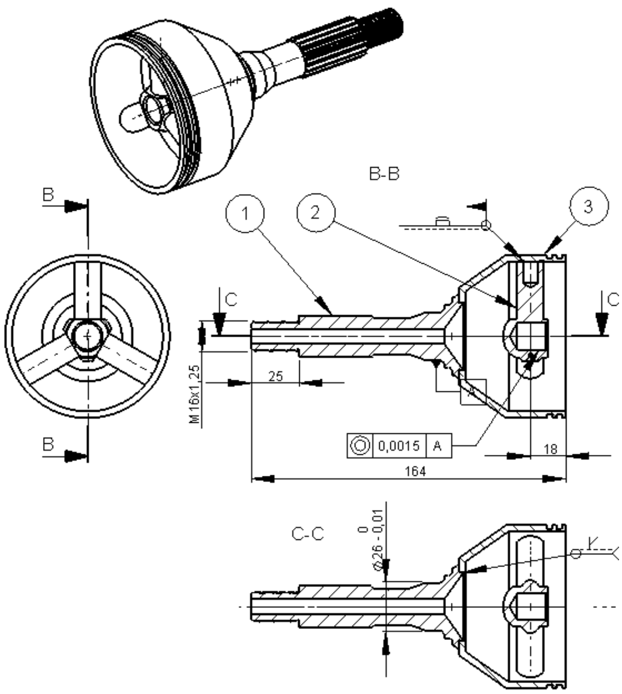

Fig. 8 Optimized live axle

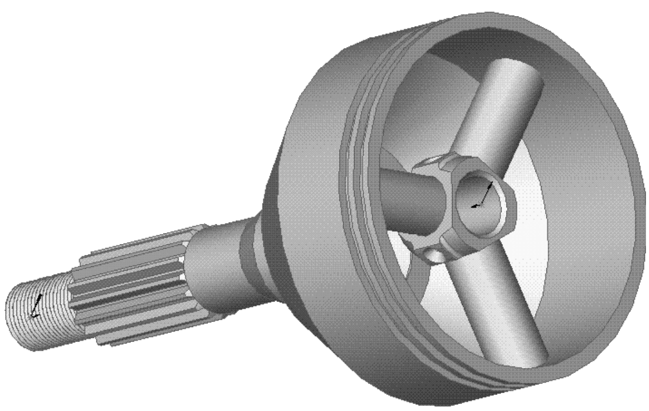

Fig. 9 3D Model of the optimized live axle

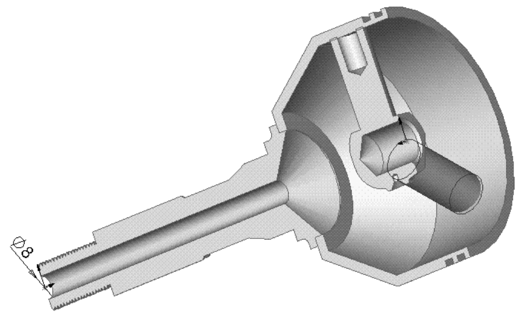

Fig. 10 Crossection through the optimized live axle

5. CONCLUSIONS

The optimized live axle by the practice of holes in the central rod Ø8 mm and the strengthening of the transition between the central rod and the cup has a superior behavior compared to classical solution at the same time causing material savings and technology [2].

By comparing the stress values obtained using the finite element method presented above, with the stress values resulting from the analytical calculus we have a difference of 7%. This difference between the two methods of calculation, the finite element method and the analytical method is acceptable and is considered that the finite element method can be validated.

Application of finite element method to estimate the live axle behavior provides a complete picture of the stresses and strains on the entire surface of the assembly.

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