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Original Research Article

Response Surface-Based Design Study of a Relay Lever for a Bus Independent Suspension Steering Mechanism

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Abstract

In the scope of this work, mechanical design stages and the structural optimization process of a relay lever that will be used as one of the main load carrying members of a passenger bus multi-link steering system are summarized. In the first stage of the study, design load of the steering mechanism was determined. For this reason, two different methods were used: the bore torque approach and the multibody dynamics (MBD) analysis of the steering mechanism. Therefore, a full-scaled multibody model of the passenger bus was built and analyzed for a chosen critical driving maneuver via Adams/Car™ module of MSC. Adams™ commercial software package. Primary mechanical design of the part was composed with the use of the load model which gives greater reaction forces. Finite element analysis (FEA) of the draft design was also implemented to determine the possible stress concentrated regions. In order to obtain the appropriate relay lever structure which satisfies minimum stress concentration and minimum deformation under the selected design load, a response surface methodology (RSM)-based optimization study was also carried out. Results of the optimization process showed that the final structure of the relay lever satisfies the strength requirements for the chosen critical load case.

Key Words: Multi-link steering mechanism, independent front suspension (IFS), multibody dynamics (MBD), mechanical design, design of experiments (DOE), response surface methodology (RSM), finite element analysis (FEA), optimization, fatigue

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1. Introduction

Because of its kinematic advantages, the multi-link mechanism is used as the steering system in the majority of the modern passenger busses equipped with independent front suspensions (IFS). An example of the multi-link steering system and its basic structural elements are seen in Fig.1. In this design type, the mechanism consists of two steering arms, two tie rods, an idler arm, a relay lever and a track rod. Maneuverability of a vehicle is closely related to the steering mechanism. Hence, a steering mechanism should be categorized as a safety system. As a result, mechanical components of the system should be resistant enough under the service loads.

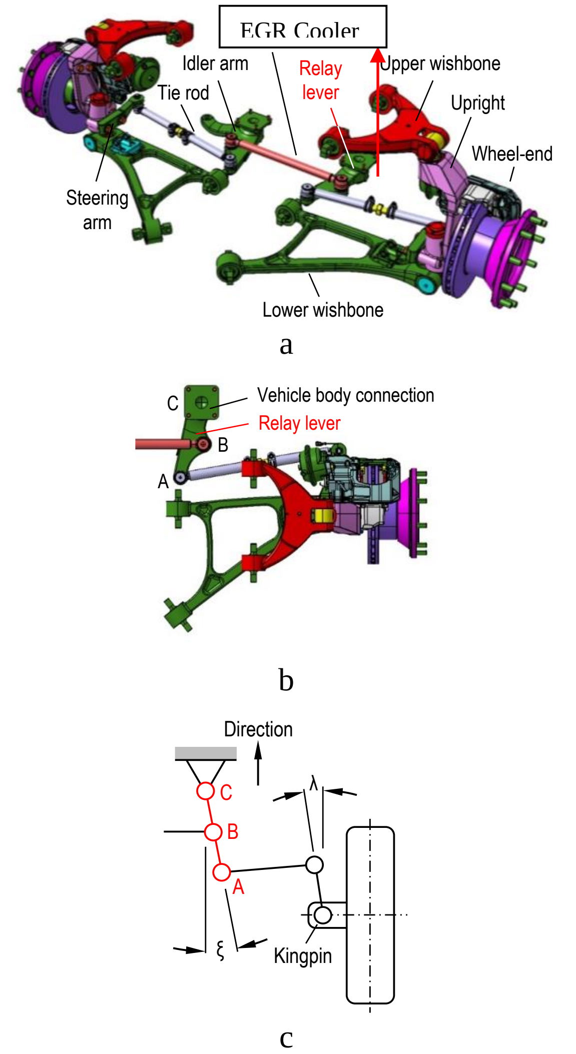


Figure 1. The multi-link steering system of a commercial vehicle IFS: a. General view [1] b. Plan view c. Schematic [2]

This study summarizes the design stages of the relay lever which is a basic structural element of a 17 metric tonnes capacity passenger bus steering mechanism. In the first stage, design load of the entire system was determined by using two different methods: the bore torque approximation and the multibody dynamics simulation of the full vehicle. By using the prior known kinematic hardpoint positions, joint forces and the mechanical properties of the material a draft design of the relay lever was composed. A primary finite element (FE) analysis of this design was also carried out. By this way, the stress concentrated regions which may cause fatigue failure of the component were determined. Three geometric parameters were selected as design factors. The final geometry of the relay lever which satisfies the design targets such as minimum equivalent stress concentration and minimum deformation was determined via a Response Surface Methodology (RSM) –based optimization study.

2. Materials and Methods

In this work, DOE-RSM methods were utilized to obtain the optimal geometry of the relay lever which satisfies the desired ranges of equivalent stress and deformation under service loads. Optimization process was carried out by ANSYS® Workbench™ which includes DOE and RSM tools. The DOE approach is used for understanding the correlation between the design parameters of the system and its performance [3]. Essentially, RSM, which uses a polynomial type regression model [4] is one of the extended DOE methods. Principal target of the response surface experiments is to obtain a proper model to estimate and analyze the relationship between design variables and system response. For a second order response surface model, the regression model is defined in general form as [5]

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \sum_{i \le j}^{k} \beta_{ij} x_i x_j + \varepsilon \qquad (1)$$

This model can also be expressed in matrix form as:

$$\boldsymbol{y} = \boldsymbol{X\beta} + \boldsymbol{\varepsilon} \qquad (2)$$

Here, y is vector of observations, X is the model matrix, β is the vector which includes the interception parameter β_0 and the partial regression coefficients and ε is the vector of random errors [6]. Estimated value of β which minimizes ε can be expressed as:

$$\hat{\boldsymbol{\beta}} = \left(\boldsymbol{X}^T\boldsymbol{X}\right)^{-1}\boldsymbol{X}^T\boldsymbol{y} \qquad (3)$$

In order to collect the experimental data, Central Composite Design *(CCD)* type which is offered in the design specification table of ANSYS/Workbench™ was utilized.

3. Results and Discussion

In order to provide the wheel steering, steering moment which also creates the joint force F_B, should be equal or greater than M_B, the bore torque on the tyre contact patch caused by the tyre-road friction as seen in Fig.2.a. Here, F_A and F_C are the reaction forces of the relay lever joints. To calculate the bore torque under the vertical load P_Z, shape of the tyre contact patch was approximated by a circle as seen in Fig.2.b and c [7]. The radius of the circle, R_B was calculated as:

$$R_B = \frac{1}{2}\left(\frac{L}{2}+\frac{B}{2}\right) = \frac{1}{4}(L+B) \qquad (4)$$

where, the length of the tyre contact patch L was determined by using the vertical stiffness c_R of the tyre and P_Z. The scrub radius of the suspension mechanism was also neglected. During the pure bore motions, circumferential forces F are generated at each patch element at the radius r. Hence, the maximum bore torque acting to the contact patch may be expressed as:

$$M_{Bmaks} = \frac{1}{R_B{}^2\pi}F * \int_0^{R_B}\int_0^{2\pi} rrd\varphi dr = \frac{2}{R_B{}^2}F\int_0^{R_B} r^2 dr = \frac{2}{3}R_B F \qquad (5)$$

The bore torque approximation represents the static condition of the vehicle. In order to simulate the self aligning effect of the tyre under a certain sideslip angle α, lateral force F_y and the tyre caster $r_{\tau,T}$ (Fig.3.a) during a lane change manoeuvre, a full multibody dynamics (MBD) model of the vehicle was also composed by using Adams/Car™ commercial software as seen in Fig.3.b and c. Double lane change test manoeuvre was applied to the model. The velocity of the vehicle was chosen as 60 km/h. Test procedure can be found in [9]. Time-dependent force components obtained at the A joint from the MBD model are seen in Fig.4.a.

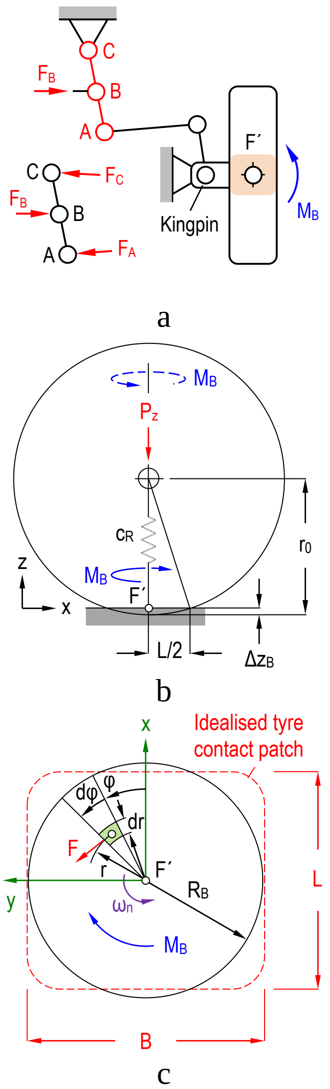
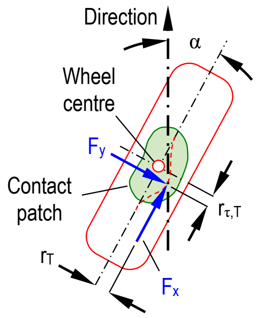


Figure 1. Schematics for a. Reaction forces at the spherical joints A and B b. Tyre deflection under the vertical load c. Idealized tyre contact patch

As can be seen from this diagram, maximum value of the reaction force acting on the ball joint A (F_A) does not exceed 5.3 kN. A comparison of the maximal forces provided from the bore torque approximation and the

MBD model is also seen in Fig.4b. It should be noted that calculated values of the aligning torque and the ball joint forces, F_{Ax}, F_{Ay} and F_{Az} depend on the selected tyre model. In this conceptual design study, results showed that the resultant ball joint force obtained from the bore torque is ~2.2 times greater than the reaction forces those calculated by the MBD software. For safety reasons, the bore torque was selected as the design load for the strength calculations of the draft design.



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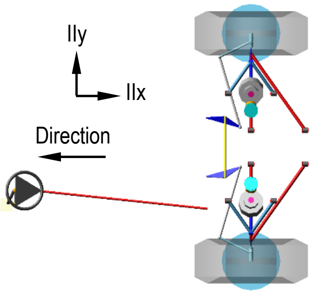

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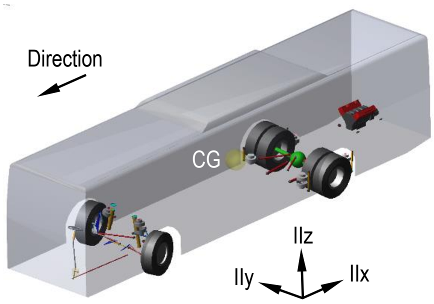


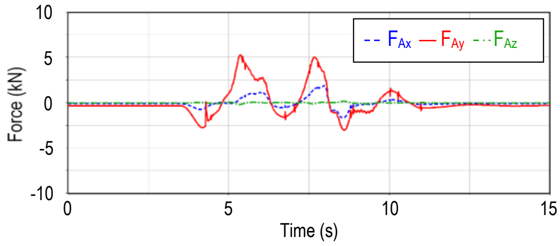
c

Figure 2. a. Contact patch forces [8] b. IFS and the steering mechanism c. MBD model of the full vehicle

4. Parametric Modelling and Optimization

4.1. Draft design

Structure of the vehicle body and the draft design of the relay lever are seen in Fig 5. Positions of the hardpoints A, B and C were determined with the use of a kinematic optimization method given in [1]. In order to avoid any penetration of the spherical joints and the structural elements of the body (Fig.5.a; detail D) during the steering motion of the mechanism, distance "h" should be taken into account at the primary stage of the draft design.



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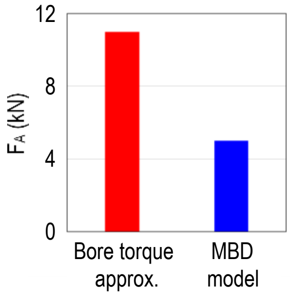

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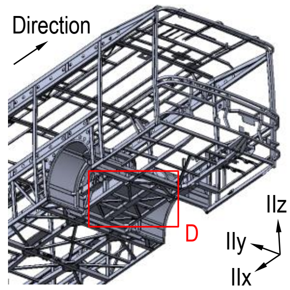
Figure 3. a. Time-dependent reaction forces at the joint A, b. Comparison of the maximum reaction forces



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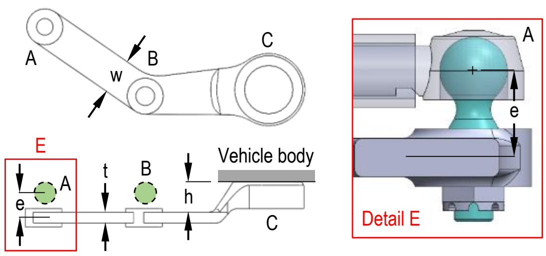

b

Figure 4. a. Body structure [1] b. Draft design and the design parameters of the relay lever

A primary strength analysis was carried out by using Maximum Distortion Energy Criterion to determine the width "w" considering the bending moment and torsion caused by the length "e" given in Fig.5.b.

Factor of safety was chosen as 1.8 [10]. For the primary design, thickness of the component was chosen as t = 15 mm. SAE 4140 (DIN 42CrMo4) forging steel was chosen as the relay lever material. Basic mechanical properties include the modulus of elasticity (E), Poisson's ratio (ν) ultimate tensile strength (S_{ut}), yield strength (S_m) and the maximum elongation (A) are seen in Table 1.

Table 1. Mechanical properties of the relay lever material [11]

Material	SAE 4140 (42CrMo4)
E (GPa)	210
ν (-)	0.3
S_{ut} (MPa)	1300
S_m (MPa)	877
A (%)	10

4.2. FE analysis

FE analysis of the primary design was carried by using ANSYS Workbench V16. For the load conditions obtained from the bore torque approximation, maximum von Mises stress values were calculated at the regions G and H as $\sigma_{V,G}$= 380 MPa and $\sigma_{V,H}$= 313 MPa respectively as seen in Fig.6.a. Analysis was also repeated for the loads achieved from the MBD model. Results obtained from this analysis are $\sigma_{V,G}$= 168 MPa and $\sigma_{V,H}$= 140 MPa. Since the steering mechanism is actually subjected to the dynamic loading during the service, fatigue life considerations should also be taken into account. In order to do that, a basic fatigue calculation was also carried out with the use of the method given in [12, 13]. Stress life endurance limit of the material is given as S_e'= 739 MPa for SAE 4140 [14]. The true fatigue strength S_e was calculated as 124.15 MPa by taking the Marin Factors into account. The surface factor k_a was obtained for "as forged" surface condition as 0.21 [15]. With the use of Fig.6.b, stress concentration factor was found for this geometry as K_t= 1.1 [16]. Hence, fatigue factor of safety n_f was calculated as 0.83 by using the Goodman equation [15]:

$$n_f = \frac{1}{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}}} \tag{6}$$

where, σ_a and σ_m the stress amplitude and the mean stress respectively. σ_a can be represented by using the maximum (σ_{max}) and minimum (σ_{min}) values of the equivalent stress during a steering motion as:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{7}$$

Loading type was idealized as "fully reversed" (σ_m= 0). Hence, σ_{max} was assumed as $\sigma_{V,G}$= 168 MPa.

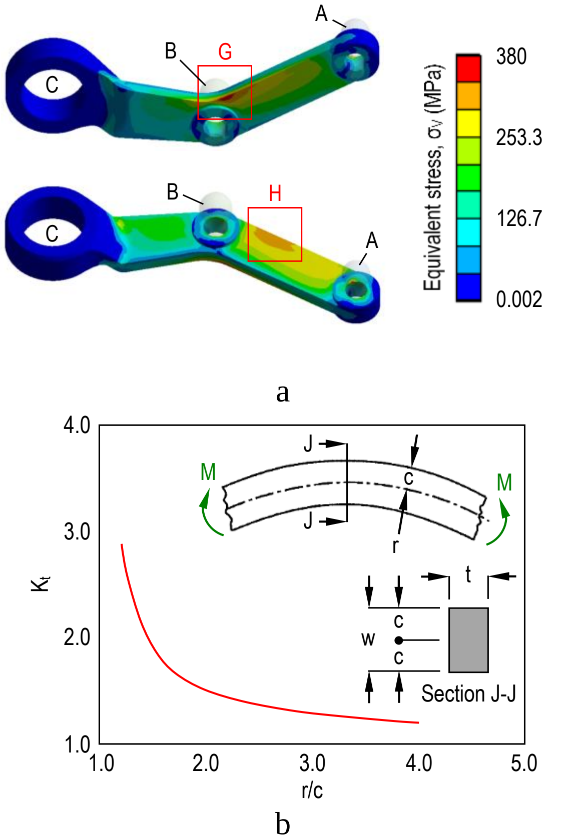

Figure 5. a. Equivalent stress distribution of the primary model b. Stress concentration factor for curved bar under bending [16]

4.3. Parametric Optimization

In order to find out the optimal shape of the relay lever which gives minimum equivalent stress concentration and infinite fatigue life at the critical regions, DesignXplorer™ optimization module of ANSYS® Workbench™ V16.0 commercial finite element software was utilized. For this reason, a parametric model of the primary design was built via SolidWorks® commercial software. Three design parameters; R_1 (r), R_2 and t, the thickness of the component were selected as the design factors as seen in Fig.7.a. Initial values and the variation ranges of the design factors are given in Table 2.

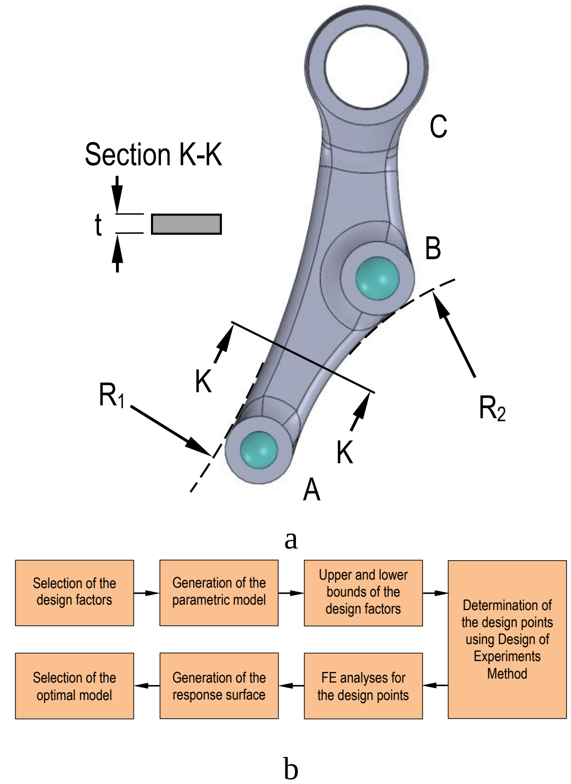

Figure 6. a. Design factors b. Steps of the optimization process

Table 2. Design limitations

Parameter	Initial value (mm)	Range (mm)
R_1	170	120-220
R_2	550	500-700
t	15	10-25

Table 3. Design points

Design point	1	2	3	4	5
R_1	600	600	600	500	700
R_2	170	170	170	170	170
t	17.5	10	25	17.5	17.5
Design point	**6**	**7**	**8**	**9**	**10**
R_1	600	600	518.7	518.7	681.3
R_2	120	220	129.35	129.35	129.35
t	17.5	17.5	11.4	23.6	11.4
Design point	**11**	**12**	**13**	**14**	**15**
R_1	681.3	518.7	518.7	681.3	681.3
R_2	129.35	210.65	210.65	201.65	210.65
t	23.6	11.4	23.6	11.4	23.6

At the next step, DesignXplorer™ module was started and Design of Experiments (DOE) method was chosen. Fifteen automatic design points were generated for three design factors. These points are given in Table 3.Stress analyses that correspond to these points were carried out. By using the results of the FE analyses, 3-D response surfaces for maximum von Mises stress and the deformation under the design load were generated by the FEA software. Design targets were selected as minimum stress concentration and minimum deformation. Flowchart of the optimization process is also seen in Fig.7.b.

5. Results and Discussion

Equivalent stress distributions obtained for the design points are given as example in Fig.9. Response surfaces for equivalents stress are also seen in Fig.10. In the light of the design targets (minimum values of σ_V and δ) and limitations, Goal Driven Optimization (GDO) approach which is offered in ANSYS® Workbench, was utilized to select the best combination of the design factors. Selected optimal values of the design factors are given in Table 4. For this model, σ_{max} was obtained as 145.96 MPa.

Table 4. Optimal values of the design factors

Parameter	R_1 (mm)	R_2 (mm)	t (mm)
Optimal value	216.53	581.15	24.98

In order to simulate the effect of the double lane change maneuver, a full FE model of the suspension mechanism including the steering mechanism was also carried out with the use of the forces obtained from the MBD model (Fig. 11.a). Results of this analysis showed that the maximum value of the σ_V does not exceed 67 MPa as seen in Fig. 11.b. Maximum deformation is obtained for the optimal model as δ= 1.4 mm (Fig. 11.c).

A comparison of maximum stress and deformation values obtained from the initial and optimized designs for different load models is seen in Fig.12. As can be clearly seen from these two diagrams, it is possible to decrease the equivalent stress and the deformation values by some 63.4% and 38.8% respectively. Final design of the relay lever that corresponds to the optimal values of the design factors can also be seen in Fig.13. For the optimized model, fatigue factor of safety n_f was calculated as 2.12 with the use of the method given in chapter 4.2.

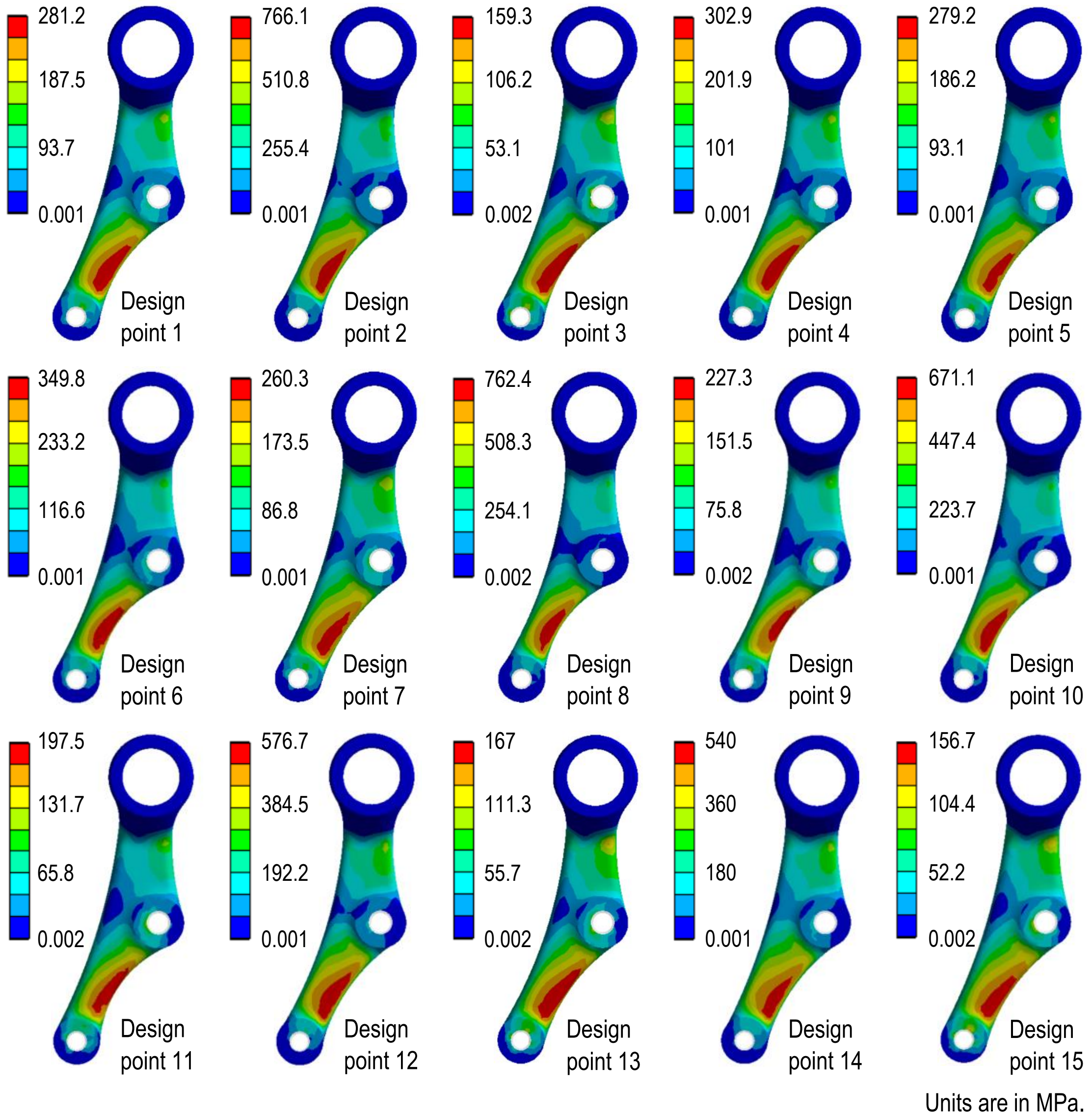

Figure 9. Equivalent stress distributions for design points

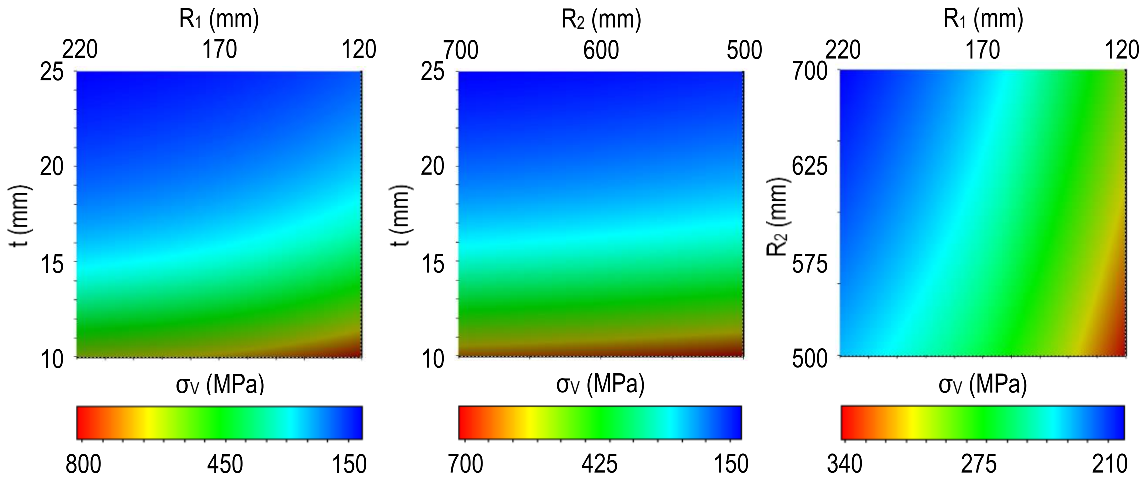

Figure 10. Response surfaces for equivalent stress

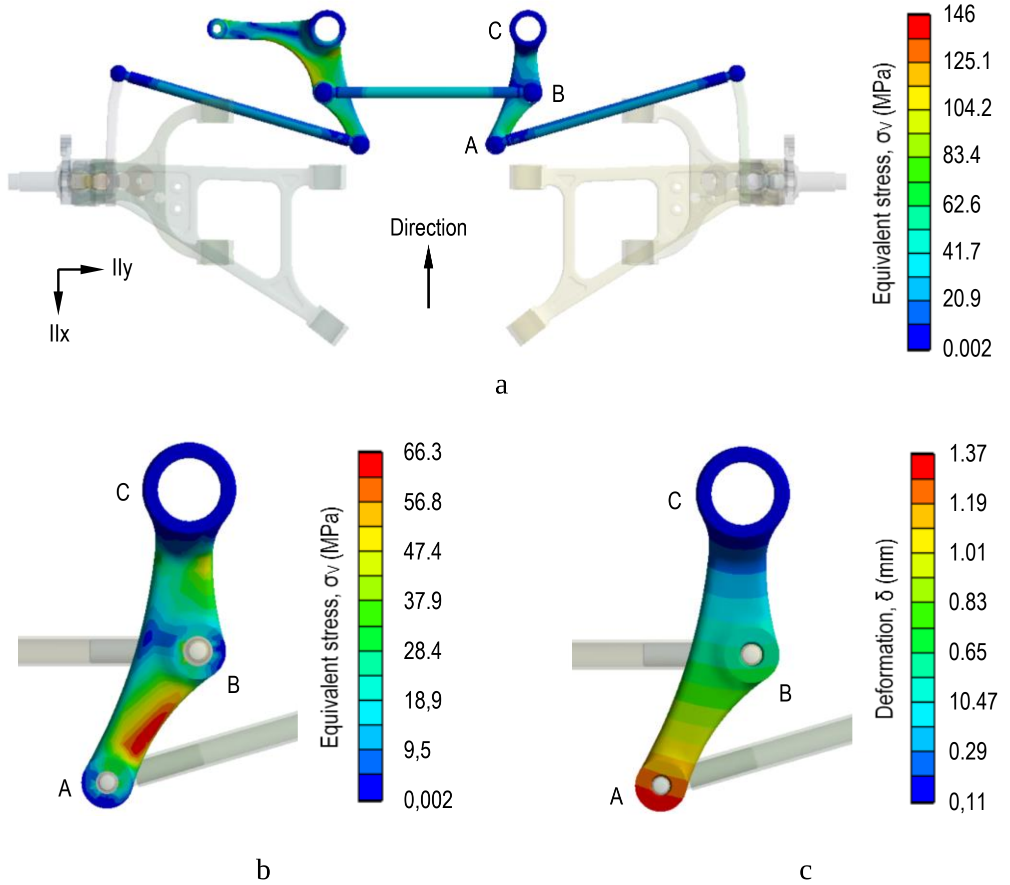

Fig 11. a. Equivalent stress distribution of the full steering system b. equivalent stress distribution on the final relay lever design c. deformation of the final relay lever design

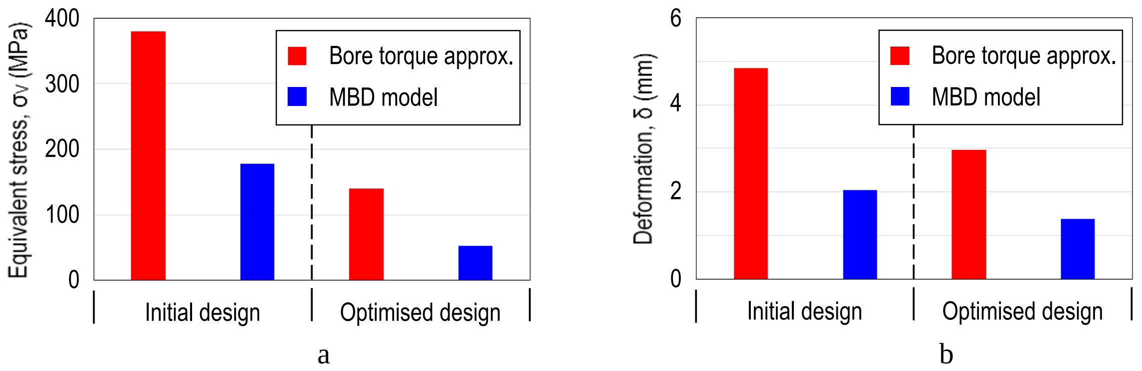

Figure 12. a. Maximum equivalent stress b. deformation for the initial and final relay lever designs

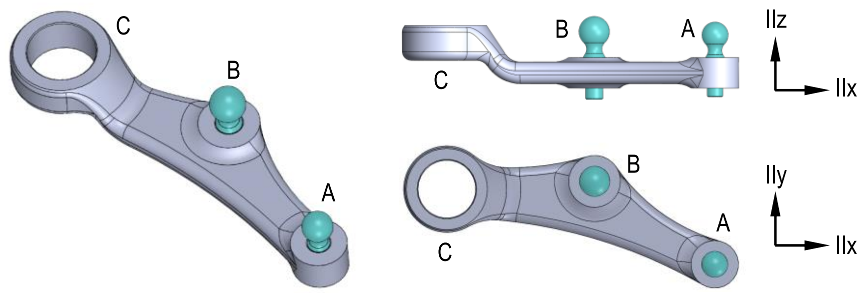

Figure 13. Final design of the relay lever

6. Conclusion

In this work, Response Surface Methodology-based conceptual design study of a relay lever for a bus independent front suspension steering mechanism was summarised. In the first stage, primary design of the relay lever was carried out by using the design load obtained from the bore torque approximation. In order to determine the reaction forces during a lane change manoeuvre, a full vehicle model of the passenger bus was also carried out with the use of the MBD approach. Maximum value of the time-dependent joint forces obtained from this model was utilised for the fatigue life calculations. FE analysis of the primary design showed that there are stress concentrated regions which may cause fatigue failure. In order to reduce the stress concentration, Response Surface Methodology (RSM)-based optimisation process was carried out by using ANSYS® Workbench commercial FE software package. The results of this study can be summarized as follows:

1. Maximum equivalent stress, σ_{Vmax} was reduced up to 63.4% in comparison with the base model.
2. Deformation of the relay lever, δ was decreased about 38.8% in comparison with the base model
3. For the optimised model, fatigue factor of safety, n_f was calculated as 2.12 which represents the infinite fatigue life.

The method given in this paper may be applied to the other components of the multi-link steering mechanism.

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