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DESIGN OPTIMIZATION OF STRUCTURAL COMPONENTS FOR FATIQUE LOADING

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Abstract: *The paper deals with structural optimization of a vehicle accessory subjected to high-cycle fatigue loads. Taking into account economical and time constraints a methodology of using finite element models to structurally optimize components subjected to fatigue loads was worked out. A three dimensional virtual model with corresponding loading case and boundary conditions has been developed. Experimental validation of the numerical model has been performed.*
Key words: high-cycle fatigue loads, structural optimization, FEM analysis.

1. INTRODUCTION

The aim of the durability analysis is prevention of field failures and optimization of vehicles and its components. As rule, the durability analysis performed, contains a combination of experimental tests and numerical simulation [1–2]. In order to save expenses on time and other recourses the amount of experimental tests can be reduced (replaced with simulations) [3-4].

In the current study the design of car rear tow hook subjected to high-cycle fatigue loadings is considered. The final product designed had to satisfy the requirements of the Directive 94/20/EU [5].

For the simulation of fatigue analysis the finite element model has been developed using ANSYS Workbench software. The fatigue test with equivalent terms and cyclical loads was performed to ensure the validity of the finite element model. After the validation, design variables were defined and parametric optimization model was created, followed by multicriteria structural optimization study. The optimization procedure concentrates on evaluating car accessories subjected to high-cycle fatigue loads with finite element models. As observed in the computational example, the proposed methodology allows effectively to optimize the studied structure against multiple criteria while maintaining its fatigue durability on an established level and reducing development time and expenditure. The multicriteria optimization problem posed has been solved by combining response modeling and global optimization techniques [6-11].

2. PROBLEM FORMULATION

The main aim of the current study is to design a car rear tow hook subjected to high-cycle fatigue loadings. The geometry and boundary conditions of the car rear tow hook are shown in Figure 1.

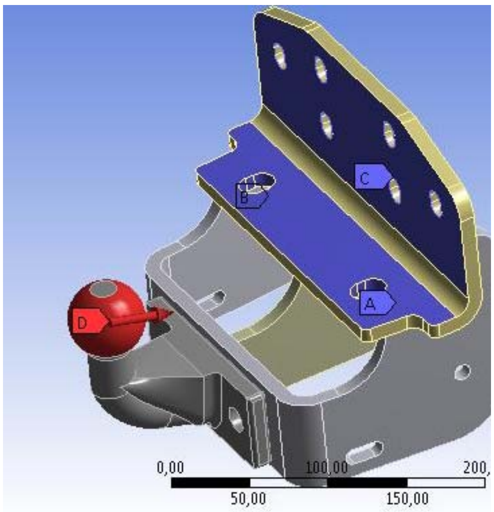

Figure 1. A car rear tow hook: geometry and boundary conditions.

In Figure 1 the frictionless support and loading force are denoted by A and D, respectively, C and D stand for fixed support. An engineering approach for determining the crack initiation in terms of bulk material properties and local stress or strain is characterized by the S–N curve [1-2]

$$\sigma_a = \sigma_f'(2N_f)^b. \qquad (1)$$

In (1) σ_a is the alternating stress, N_f is the number of cycles to failure, σ_f' is the fatigue strength coefficient, and b is the fatigue strength exponent. These S–N curves are obtained for zero mean stress. An equivalent alternating stress to a zero mean stress $\sigma_{eq,a}$ can be calculated as [2, 4]:

$$\sigma_{eq,a} = \frac{\sigma_a \sigma_u}{\sigma_u - \sigma_m}. \qquad (2)$$

In (2) σ_m and σ_u stand for the mean stress, respectively and the ultimate tensile strength, respectively. The above results are obtained assuming fatigue test results with constant amplitude loading. According to damage theories the fatigue damage can be expressed as follows

$$D_i = \frac{n_i}{N_{f,i}}. \qquad (3)$$

In (3) n_i and $N_{f,i}$ stand for the number of applied stress (strain) cycles and number of cycles to failure at a given stress amplitude, respectively. The damage accumulation D can be calculated by use of linear damage rule as

$$D = \sum_{i=1}^{n} D_i \leq 1. \qquad (4)$$

The value of the D less than one means that the component will have a residual fatigue life, which can be determined becoming relation (4) into equality.

3. FINITE ELEMENT ANALYSIS

ANSYS Workbench software was utilized for numerical analysis. Solid hexahedral elements (solid185), with 8 nodes were considered. Constant amplitude fully reversed loading without any mean stress correction theory was applied to the structure. For the fatigue analysis the safety factor was designated as the output data. This illustrates the safety margin to a fatigue failure at a given design life (10^9 cycles in the current model).Whereas the hook part of the product had been certified forehand, it was excluded from the solution part of the analysis. But for a more precise transfusion of the reaction forces included in the setup of the FEA model. The total number of simulations was dependent on the number of design variables. In all of the simulations grid density was fixed in order to get a complete set of output data. Gathered information was used for further optimization procedure.
In the first FEA model testing rig and nonlinear contacts (Figure 2) were included in the simulation model.

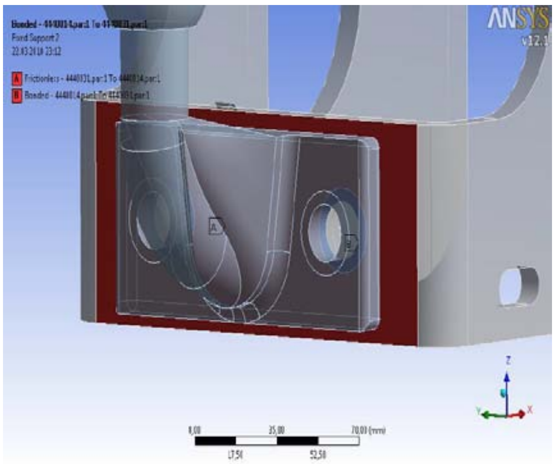

Figure 2 – Nonlinear contacts

Forenamed contacts were implemented for a precise description of processes that take place in a bolted connection. Three different hexahedral element side lengths were considered according to the general dimensions of the model objects. Stiffness ribs were meshed with 3 mm element side length, hook with 5 mm side length and weldments with 2 mm side length (Figure 3).

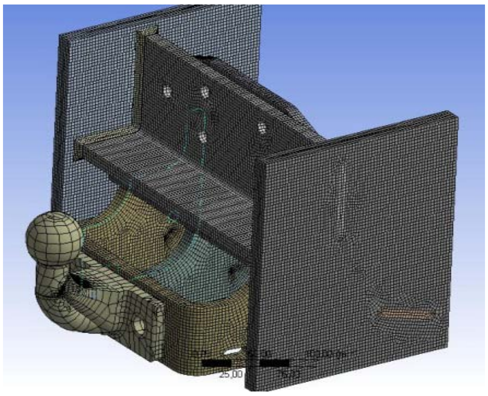

Figure 3. Mesh with testing rig.

Model with the centre stiffness rib, 12 mm material thickness and 10 mm bending radius consisted of 179 785 elements and resulted in safety factor of 1.5259 (Figure 4).

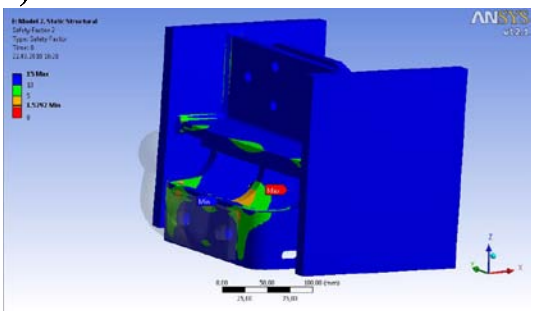

Figure 4. Results with testing rig and non-linear contacts.

FEA model without the testing rig and nonlinear contacts (with same grid and design parameters) consisted of 60 298 elements (Figure 5) and resulted in safety factor of 1.5206.

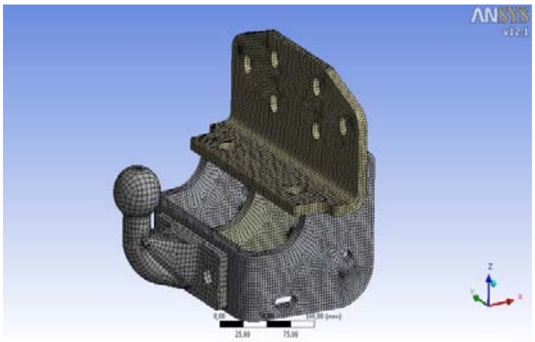

Figure 5. Mesh without testing rig.

The overall difference in calculation results was considered as marginal and therefore nonlinear contacts and the testing rig were discarded form the FEA model. This became an important feature during the optimization process when design parameters were varied. The time to complete one simulation shortened approximately 5 times.

4. EXPERIMENTAL STUDY

In order to validate the FEA models an experimental study has been performed on testing machine Instron 8802 (Figure 6). All the experiments and certifications were carried out according to the European Union directive 94/20/EU (details given in [5]).

Figure 6. Fatigue test: constant amplitude loading.

Multiple FEM analysis with different boundary conditions were solved and compared to the experimental study performed with same values of parameters. This determined the level and nature of the approximations what were applied to the simulation model in order to reduce simulation time for the optimization process. Good agreement between test and simulation data has been observed, the differences in results remains in range of 10%.

5. DESIGN OPTIMIZATION

For problem considered above the multicriteria optimisation problem can be formulated as

$$F(\bar{x}) = \min(F_1(\bar{x}), F_2(\bar{x})) \,, \qquad (5)$$

subjected to

a) linear constraints on design variables

$$x_i \le x_i^{*}, \qquad -x_i \le x_{i*}, \;, \qquad i = 1,\dots,n \,, \qquad (6)$$

b) non linear constraints on displacements

$$g_j(\bar{x}) \le 0 \,, \qquad j = 1,\dots,m \,, \qquad (7)$$

where $F_1(\bar{x})$ and $F_2(\bar{x})$ stand for the safety factor (taken with minus sign) and manufacturing cost, respectively. The design variables x_i characterize the geometry of the car rear tow hook. Its initial design depicted in Figure 1 was given by the manufacturer. Thus, the topology is predefined to a certain extent by the manufacturer and the main task is to search for an optimal set of design variables (thickness, radius etc.). However, some corrections in topology are available.

In the following the objectives $F_1(\bar{x})$ and $F_2(\bar{x})$ are normalized since the magnitudes and the units used to measure the objectives are different

$$f_1(x) = \frac{\max F_1(x) - F_1(x)}{\max F_1(x) - \min F_1(x)} \,, \qquad (8)$$

$$f_2(x) = \frac{\max F_2(x) - F_2(x)}{\max F_2(x) - \min F_2(x)} \,. \qquad (9)$$

The simplest techniques used for solving multicriteria optimisation problems are physical programming techniques [6-8]. In the current study the weighted summation and compromise programming techniques are considered. The evaluation of the objective and constraint functions described above includes time consuming FE simulations. A common technique for reducing computational cost in optimal design problems is to use surrogate models for the approximation of the objective and constraint functions. In the current paper the surrogate models are used to guide the search towards a global optimum. (6)The output data obtained from the FE analysis are treated as response values. The generalized regression neural networks (NN) are used for the surface fitting [11]. An approach proposed in this paper is based on the use of the MATLAB neural network toolbox.

6. RESULTS

(7)

Behavior of objective functions (fatigue safety factor and manufacturing cost) with respect to design variables has been studied. The design appears sensitive with respect to thickness of metal sheet and folding radius.

Change of the thickness of metal sheet from 8 mm to 12 mm resulted in the increment of the safety factor approximately for 1.4 times (from 1.0790 to 1.5351 other design parameters were fixed). Change of the folding radius from 8 mm to 14 mm resulted in the increment of the safety factor approximately for 1.1 times (from 1.5351 to 1.3775, other design parameters were fixed).

Note, that due to practical considerations (vehicle safety first), the following weights

a) w_1=0.9; w_2=0.1;
b) w_1=0.8; w_2=0.2;

are used for the objective functions.

7. CONCLUSION

(8)

Design optimization of the structural components for fatigue loading has been performed. The study was focused on design of a car rear tow hook. The artificial neural networks and global optimization techniques are combined for solving the engineering problem posed above. It can be concluded that:

a) The fatigue safety factor is sensitive with respect to certain design variables (thickness of metal sheet and folding radius) and its dependence on these variables is linear. Thus, the best result can be obtained in the case of boundary values of design variables (maximal thickness, maximal folding radius).

b) The manufacturing cost is not sensitive with respect to design variables considered (dependence on design variables is weak).

The final conclusion is that the car rear tow hook satisfying the requirements of the Directive 94/20/EU can be designed by use of existing equipment and materials.

8. ACKNOWLEDGEMENTS

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