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Engineering optimization of a car frontal protection system component

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Abstract. The study focuses on the design of a car frontal protection system using response surface modelling and a genetic algorithm. The main attention was paid to optimal design of brackets. The analysis of car–pedestrian collision situation was performed using explicit FEA solver and the stiffness analysis of the bracket with implicit FEA solver. Multi-criteria optimal design problem was formulated and the necessary optimality conditions were derived. The obtained numerical results were validated by comparing with experimental test results. An alternative numerical approach was realized using optimization software package LS-OPT.

Key words: car frontal protection system, energy absorption, optimization, FEA, genetic algorithm, neural networks.

1. INTRODUCTION

In recent years, due to new regulations, engineers have begun to redesign vehicle parts, e.g. bumpers, hoods etc, with the aim to make them better energy absorbers in the case of car–pedestrian accidents. Various problems related to impact energy absorbers are considered in [1–5]. In [1], the crash behaviour of circular aluminium tubes, undergoing axial compressive loading, is studied experimentally and numerically. Static and dynamic behaviour of the circular thin-walled tubes is considered, the values of the mass and impact velocity of the tubes are varied. Axisymmetrical and mixed deformation modes are analysed. Numerical results obtained from FEA are validated against static and dynamic test results. In [2], common shapes of collapsible energy absorbers, including circular and square tubes, frusta, struts, honeycombs, and sandwich plates are reviewed. Axial crushing, lateral indentation, lateral flattening, inversion and splitting are considered as possible deformation modes. The viability of using structural foam in B-pillar and bumper designs is investigated by Gupta [3]. It was concluded that the B-pillar and the rear bumper are potential areas where structural foam could replace steel and other materials and allow weight reduction in comparison with respective base models. In [4], the impact energy absorption of the vehicles front structures during frontal crash is studied. The energy absorbing capabilities of columns under axial compression loading were analysed. It is pointed out that the degradation of the structures in old cars has significant influence on the energy absorbed. Capability to absorb up to three times more energy by new columns, compared to corroded columns in the old cars, is reported. Testing and numerical simulation of multi-material energy absorbers is performed by de Kanter [5]. The properties of the metallic and fiber reinforced plastic (FRP) cylinders in the crushing behaviour are discussed. It is stated that multi-material elements demonstrate both the crashing behaviour characteristics of metallic and composite materials. Three different possibilities for integrating metal and composite materials are analysed (reinforcement of a metal cylinder with FRP, reinforcement of a FRP cylinder with metal, creating a multi-material component as one).

The effects of vehicle bumper height and impact velocity on the type of lower extremity injury were studied by Matsui [6]. It is pointed out that the main injury at an impact velocity of around 20–30 km/h is to the knee ligament, but at an impact velocity of around 40 km/h, fracture of the lower extremities. The paper by Schuster [7] is devoted to the bumper system design for pedestrian impact. Different approaches for reducing the severity of pedestrian lower limb impacts are discussed. More popular cushioning methods are reviewed. Some trends in bumper design as use of lower stiffeners, alternative energy absorbers, beam face features, flexible beams and add-on structures are pointed out.

Employment of various optimization techniques is one of the trends in car frontal protection system design [8,9]. In [8], cylinder impact on a rigid wall is considered as an example. This paper is focusing on the description of the capabilities of LS-OPT software. Successive response surface methodology based optimization and its application to structural design is discussed in detail. Special attention is paid to crashworthiness analysis. Shape optimization of a crash box using HyperMorph and LS-OPT software is studied by Wang [9]. The smoothness of the force values during the folding process of the crash box is considered as the quality criterion. The difference between maximum and minimum force values of the force-intrusion is minimized.

The aim of the current study is to optimize the additional frontal protection system of a vehicle using tubular parts and brackets. The stiffness of the brackets is limited by pedestrian safety and required structural stiffness of the car parts. In order to obtain maximum energy absorption that is smooth enough, search for optimal configuration of support components of the structure is performed. The optimization procedures proposed in the current paper and in [10] are compared and validated against experimental test results.

2. PROBLEM FORMULATION

It was assumed that the height of the car frontal protection system designed is less than 500 mm and main attention was paid to the safety requirements proceeding from lower legform impact test, required by the directive of the European Parliament and the Council 2005/66/EC [11] (Fig. 1).

In the test, the impactor is shot at the speed of 11.1 m/s at the frontal protection system of the vehicle. There are three types of sensors mounted inside the impactor: acceleration sensor, bending angle sensor and shear displacement sensor. According to the directive [11]:

– the maximum dynamic knee bending angle shall not exceed 21.0°;
– the maximum dynamic knee shearing displacement shall not exceed 6.0 mm;
– the acceleration measured at the upper end of the tibia shall not exceed 200g (g = 9.81 m/s^2).

It was assumed that the total permissible mass of the vehicle is less than 2500 kg. Unfortunately, most of the structures absorb energy in an unstable manner. There will be a high peak of reaction force when impact loading starts, followed by smaller peaks or more constant level of reaction forces. More desirable situation would be if the reaction force increases steadily to some predefined level and remains constant at this level [12]. In order to decrease the acceleration, optimal design of tubular parts and fastening components has to be addressed.

The current study is focused on the design of brackets. The main energy absorbing component is shown in Fig. 2. Initial design of the energy absorbing component, depicted in Fig. 2, was given by the manufacturer. Thus, the topology was predefined to a certain extent by the manufacturer and the main task was to search for optimal set of design variables *a*, *b*, *c*, *d* and *e* (Fig. 2). However, some corrections in the topology could be made. The properties of the tubes were selected as appropriate as technologically possible (light structure, thin walls, etc), detailed optimization of the tubes was omitted.

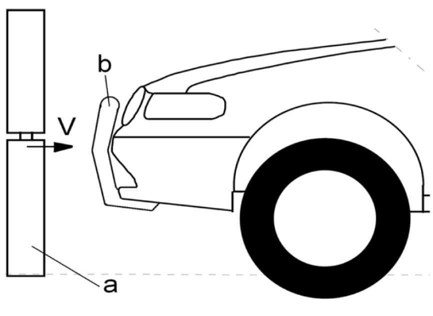

Fig. 1. Lower legform impact testing: a – legform impactor, b – frontal protection system, V – velocity of the impactor.

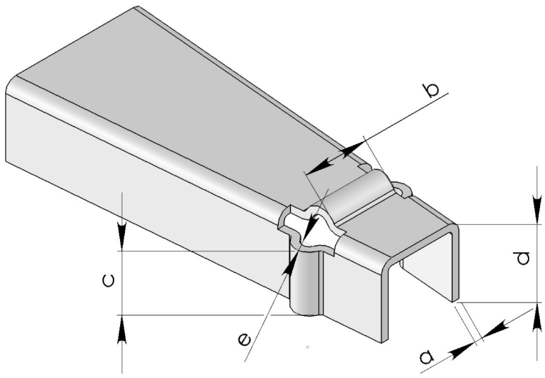

Fig. 2. Energy absorbing component (*a*, *b*, *c*, *d* and *e* are design variables).

3. ESTIMATE OF THE DEFORMATION ENERGY

The thoroughgoing theoretical analysis of the posed problem is too complicated due to its complexity (geometrical and physical non-linearity, impact loading, non-linear constraints, etc). However, a simple estimate of the deformation energy can be derived.

Let us proceed from the simplified model, used above, and assume that the velocity v is the same for the legform and car protection system. The kinetic energy before and during the crash can be expressed as

$$E_B = m\frac{v_0^2}{2}, \quad E_D = (M+m)\frac{v^2}{2}, \tag{1}$$

where m and M stand for masses of the legform and car protection system, respectively, and v_0 is the initial velocity of the legform. The deformation energy of the bracket D_D during the crash (energy absorption) can be calculated as

$$D_D = E_B - E_D. \tag{2}$$

Alternatively, the deformation energy can be computed as an integral of the reaction force F as

$$E_D = \int F ds, \qquad \text{or} \qquad E_D = \int F dt. \tag{3}$$

Formulas (1)–(3) determine the relation between the velocity and the reaction force F and give an estimate of the deformation energy of the bracket during the crash.

4. EXPERIMENTAL STUDY

In the current study two kinds of tests have been carried out: the static compression tests of the brackets, performed in the Laboratory of Mechanical Testing and Metrology of the Tallinn University of Technology, and the impact tests, performed in TÜV Rheinland (Germany).

4.1. Static compression tests

The static compression tests have been performed in order to obtain initial estimate of the deformation process of the bracket. The expenses and time needed for preparation and performing static compression test are several orders lower than those for the impact test. This allows examining a number of brackets with different geometry and performing initial selection.

However, the results obtained from static compression tests can be used as approximations only, in order to obtain more reliable results the dynamic impact tests are necessary. Also, for certain configurations of the brackets, where deformation process is highly deformation speed dependent, the static compression tests can be omitted (brackets, where energy absorption is based on friction, for example).

4.2. Impact tests

The general description of the impact test set-up is given in Section 2. The fastening components (brackets) are prepared by the authors of the current study in cooperation with the producer company. In order to save expenses and time, only part of fastening components prepared were tested, depending on the results of the first tests.

The acceleration plot of the lower legform impact test is given in Fig. 3. The bracket configuration used is similar to that given in Fig. 2.

Figure 3 shows that the acceleration, measured at the upper end of the tibia, remains within the limits, prescribed by EU directive [11]. Proceeding from the test results obtained, some modifications to the design data and the topology of the bracket were made.

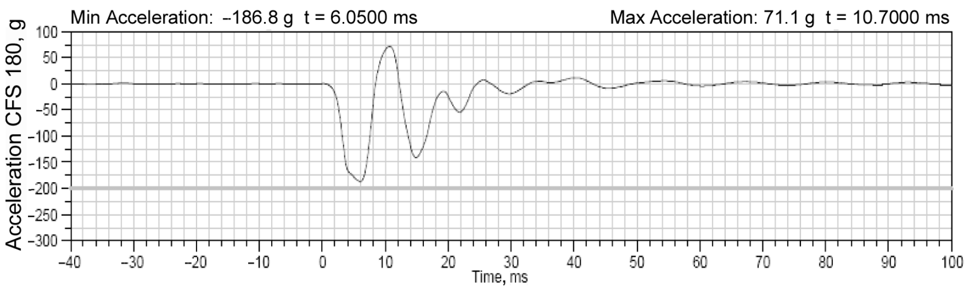

Fig. 3. Acceleration plot of the lower legform impact test.

5. FINITE ELEMENT ANALYSIS

LS-DYNA software was utilized for numerical analysis. Fully integrated shell elements were used. The stress–strain behaviour was modelled with multi-linear approximation. In order to consider plastic anisotropy, the Hill's second order yield criterion was employed. As mentioned above, FEA was performed separately for crash simulation and stiffness analysis. The total number of simulations depends on the number of design variables and on grid density, fixed in the stage of simulation data design. The dynamic and static analysis were performed with the same sets of the simulation data in order to get complete set of output data. The output data used in the further optimization procedure contained extreme values of the frontal force component (obtained from the dynamic analysis) and displacements in the *y*-*z* plane.

In order to validate the FEA models, an experimental study was carried out. Several versions of the component, shown in Fig. 2, were tested (the number of design variables used in different approaches was from 4 to 8). The preliminary estimates of the force components and deformation modes were obtained from the compression tests of the brackets, performed on universal testing equipment. In Fig. 4 the load displacement curves, obtained from the experimental tests and FEA, are compared. The design parameters are taken as $a = 1.6$, $b = 12$, $c = 6$ and $d = 10$ mm. The folds with a triangular shape (instead of a convex arc) are considered and the bend angle with the value of 6 deg is used instead of the design parameter e given in Fig. 2.

It can be seen from Fig. 4 that the experimental and FEA results are in good agreement, the peak values of the reaction force and also the shapes of the curves are close. Differences that exceeded 10% were observed in the results in the case of brackets with inner folds (directed to the inner side of the bracket). In the latter case the folded parts of the bracket run into contact during the deformation process (assuming the folded area is long enough). In the experimental tests,

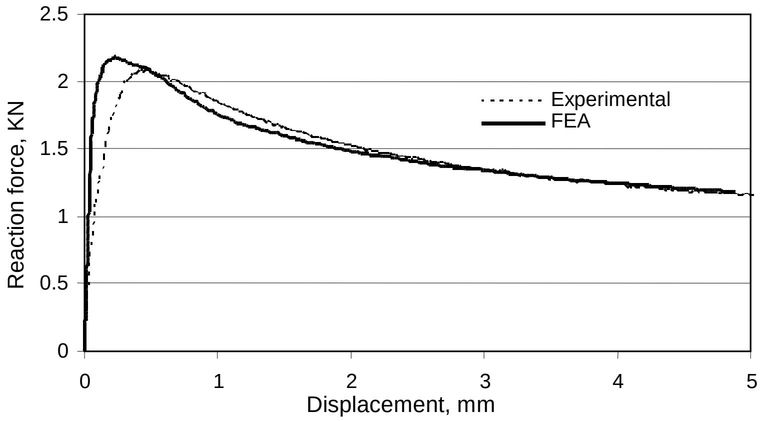

Fig. 4. Load–displacement curves: experimental and FEA.

sliding between the contacting parts was observed, since it is complicated to fulfill the symmetry conditions ideally in an experimental test. However, in some tests experimental and numerical results were close in the first stage of the deformation process, corresponding to the first peak of the reaction force. Considerable differences in the results appeared during the second stage, corresponding to the second peak of the reaction force (caused by contact between the folded parts). In terms of pedestrian safety, the first peak of the reaction force is most critical.

The influence of the design parameters a, b, c, e and their interactions on the value of the reaction force were analysed. The change of the thickness of the metal sheet from 0.5 to 2 mm resulted in the increment of the reaction force approximately for 5 times, from $F = 2213$ to $F = 10\,295$ N (other design parameters were fixed as $b = 16$, $c = 7$, $d = 10$ mm, $e = 0°$). The change of the folding angle e from 5° to 0° resulted in the increment of the reaction force for more than 3 times (from 2847 to 10 295 N, other design parameters were fixed: $a = 2$, $b = 16$, $c = 7$, $d = 10$ mm). Variation of all design parameters may result from the changes in the reaction force for more than 10 times. Some changes of the topology of the bracket also resulted in the expansion of the range of objectives. The initial design space was restricted substantially by the non-linear constraint on displacements in *y* and *z* direction.

6. RESPONSE SURFACE MODELLING

In the current paper, the generalized regression neural networks (NN) were used for the response surface modelling. The surface constructed by use of NN do not normally contain the given response values (similarity to the least-squares method in this respect). An approach proposed was based on the use of the MATLAB NN Toolbox. In MATLAB NN Toolbox a two-layer network is generated by use of the function *newgrnn*. The first layer has *radbas* neurons and the second layer has *purelin* neurons. The response surfaces were generated simultaneously (with one call to *newgrnn*) for all response quantities (frontal force component and y-z displacement).

7. OPTIMIZATION PROBLEM

The multi-criteria optimization problem was formulated as

$$\min[w_1 f_1 + w_2 f_2], \tag{4}$$

subject to

$$g_j(\overline{x}) \le 0, \quad j = 1, \dots, m, \tag{5}$$

$$h_k(\overline{x}) = 0, \quad k = 1, \dots, l. \tag{6}$$

The constraints (5)–(6) are given in terms of design variables $x_1, x_2, \ldots, x_n$. Both, the peak force f_1 and difference between maximal and minimal force values f_2 are subjected to minimization. The proportions of the functions f_1 and f_2 in the objective function are determined by the weight coefficients w_1 and w_2.

The constraints laid on design variables depend on the structure and topology of the car frontal protection system considered. In the general form, the linear constraints can be written as

$$x_i \le x_i^*, \quad -x_i \le -x_i^{**}, \quad (i = 1, \ldots, n), \tag{7}$$

where x_i^* and x_i^{**} stand for upper and lower bounds of the design variable, respectively. As mentioned above, the stiffness of the components of the car frontal protection system is limited by pedestrian safety and required structural stiffness of the car accessories. The constraint, providing the required stiffness of the car frontal protection system, can be expressed in terms of displacements as

$$u_c = \sqrt{u_y^2 + u_z^2} \le u^*, \tag{8}$$

where u_y and u_z stand for the displacements in y and z direction, respectively, and u^* is a given limit value, determined experimentally. The extended functional J_* is introduced as

$$J_* = w_1 f_1(\overline{x}) + w_2 f_2(\overline{x}) + \lambda^T g(\overline{x}), \tag{9}$$

where λ stands for the Lagrange multiplier vector and all constraints are included in the inequality constraint vector $g(\overline{x})$. In order to obtain the necessary optimality conditions, the total variation of the functional J_* is equalized to zero $(\delta J_* = 0)$.

8. OPTIMIZATION PROCEDURE

Global optimization was performed by use of the two-stage and hybrid genetic algorithms (GA).

8.1. Search for optimal solution

In order to determine the minimal value of the objective function (4), the MATLAB function *ga* has been utilized (Genetic Algorithm and Direct Search Toolbox). The response surfaces for frontal force component F and *y-z* displacement u_c were treated as objective and constraints functions, respectively. In order to achieve higher accuracy, the real-coded approach of GA was considered. An alternative solution was realized by use of LS-OPT software. Latter solution is based on the use of the leap-frog algorithm [13].

As it can be expected, optimization via genetic algorithms utilizes natural selection as a means of finding the optimal solution in the global domain, the computed solution is not the global extreme, but a close value to it. Thus, further refinement of the design is still necessary.

8.2. Design improvement

In the case of the two-stage GA, both global and local search was performed. The domain for local search is taken as

$$x_i^g - \delta_i <= x_i <= x_i^g + \delta_i, \quad (i = 1, \ldots, n), \tag{10}$$

where x_i^g stands for the value of the design variable obtained from global search and δ_i is a given deviation for the *i*-th variable. In the case of hybrid approach, the global search was performed by GA, but the steepest descent method was applied for local search using the same domain (10) for design variables. Use of the two-stage GA is justified in cases when several extreme values of the objective function are expected in local search domain. Otherwise, the hybrid approach is preferred, since it is assumed to converge faster and exactly to extreme value (not only close to extreme value as GA) [14,15]. The hybrid GA developed, has been applied by the authors with success for solving a number of optimization problems of practical and theoretical character (modelling of a new composite from recycled GFP, optimal design of a composite bathtub, optimal material orientation of 3D orthotropic materials, etc).

Since GA is a stochastic search method, which converges close to the global extreme value "in better case", the results obtained in different subsequent runs of the algorithm may differ slightly.

9. RESULTS AND DISCUSSION

As mentioned above, the limitation on acceleration (or corresponding force component) appears to be the most critical. For that reason f_1 is considered as dominating term in the objective function (4). As a result of the design process, the maximal value of the frontal force component $F(t, \bar{x})$ was reduced more than 4 times in comparison with the initial design. In Fig. 5, the frontal force component $F(t, \bar{x})$, corresponding to initial and optimal sets of design variables, is given. All constraints are fulfilled in the case of both designs. Note that energy absorption is twice higher in the case of the initial design. The latter fact can be explained by reduced dimensions of the component. The lower energy absorption should not cause any problems, because the excessive energy will be absorbed by the absorbers of the car (bumper, crash-box, etc).

In Table 1, the optimal values of the frontal force component, obtained by use of different optimization algorithms, are compared.

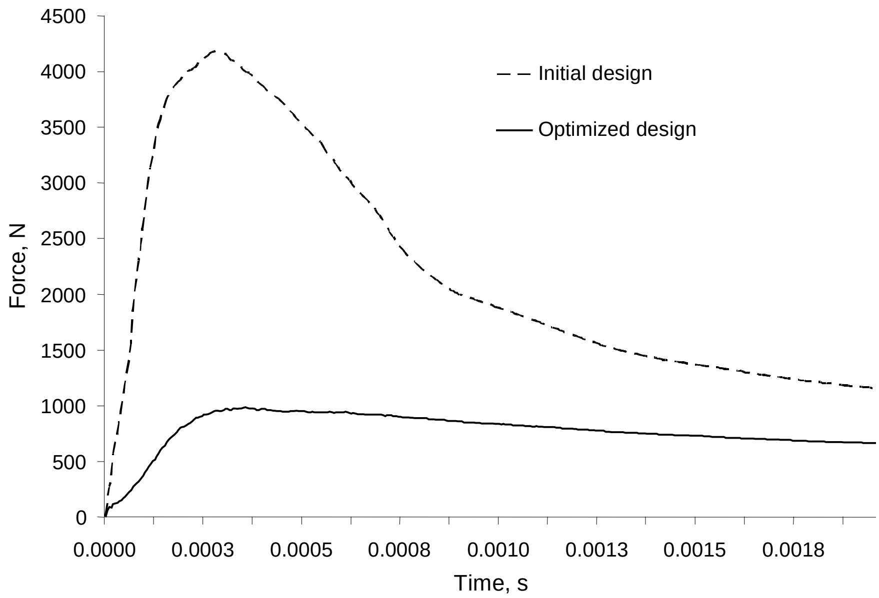

Fig. 5. Force–time diagram: initial design and the optimized design.

Table 1. Frontal force components (N), obtained with different optimization algorithms

Optimization algorithm			
GA	Two-stage GA	Hybrid GA	LS-OPT
1157	1134	1125	1067

It can be seen from Table 1 that the results obtained by use of GA, two-stage GA and hybrid GA are close to each other, but differ to a certain extent (less than 10%) from the result obtained by use of LS-OPT software. The latter fact can be explained by different approaches used for the response surface (RS) modelling. In the first case (GA, two-stage GA and hybrid GA) the RS was modelled as "static" – it was composed once and used for global and local search. In LS-OPT based solution the RS is updated in each global iteration step ("dynamic" RS). The data set is modified by performing FE computing with LS-DYNA explicit and implicit solvers. Such an approach is not complicated due to the compatibility of the FE software LS-DYNA and LS-OPT. Dynamic data exchange between MATLAB and LS-OPT, also handling the FE solvers and MATLAB processes together are much more complicated tasks.

It follows from the numerical simulations and experimental tests that the optimal design is most sensitive with respect to the thickness of the fastening component.

10. CONCLUSIONS

General conclusion of the study is that the car frontal protection system, satisfying the requirements of directive 2005/66/EC can be manufactured by use of existing equipment and materials. At the moment nine products have been passed through the type test.

Design optimization of the car frontal protection system was performed. Main attention was paid to the optimal design of the fastening components. The design procedure proposed contains an analysis with dynamic loading for car–pedestrian collision situation by use of LS-DYNA explicit solver, stiffness analysis with LS-DYNA implicit solver, response surface modelling (NN) and search for optimal design (MATLAB Optimization Toolbox, Genetic Algorithm and Direct Search Toolbox). The results of simulations were found to be close to corresponding experimental results.

A design improvement was proposed. Two-stage and hybrid GA algorithms were developed for the posed optimization problem. The obtained results were compared with the results of an alternative solution, realized by use of LS-OPT software.

The algorithm developed has been used by the authors without significant modifications for solving different optimization problems (material parameters identification, modelling of a new composite from recycled GFP, etc).

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Auto esikaitsesüsteemi komponendi optimeerimine

Meelis Pohlak, Jüri Majak ja Martin Eerme

On kavandatud uus auto esikaitsesüsteemi konstruktsioon, kasutades lõplike elementide meetodit, tehisnärvivõrke ja geneetilisi algoritme. Uurimuse fookuses on kinnitusklambrite optimeerimine, viimaks süsteem vastavusse uue eurodirektiivi nõuetega. Ülesanne lahendati mitmekriteriaalse optimeerimise abil. Leiti kinnitusklambrite sobiv kuju ja mõõtmed. Kasutatud numbriliste mudelite ja eksperimentide tulemused olid heas kooskõlas.