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BOTTOM STRUCTURE FOR DutchEVO CAR: FORMULATION OF THE PROBLEM AND THE ADJUSTMENT OF THE OPTIMIZATION SYSTEM

Natalia S. Ermolaeva, Kirill G. Kaveline and Jan L. Spoormaker

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1. Introduction

The DutchEVO project has been initiated as a part of Delft Interfaculty Research Cooperation (DIOC) program "Smart Product Systems" in order to stimulate innovative, multidisciplinary research. The purpose of the project is to develop a knowledge base for sustainable product design. A sustainable car, in accordance to DutchEVO project task, means environmentally friendly, affordable, lightweight car satisfying all current and/or future legislations on safety, emissions, recyclability etc. The methodology of lightweight design is expected to be the most valuable approach to reach the goals of the project. The application of a structural optimization technique with respect to the minimum mass directly supports this methodology.

One of the design possibilities for the DutchEVO car, which is considered as advantageous for the car structure, helping to solve the "safety – lightweight design" conflict, is the raised floor concept [Kanter de, Vlot, Kandachar and Kaveline 2001]. This concept was developed later into the type of a platform structure for the car body, when the bottom part of the car carries the main loads. According to automotive standards [Fenton 1998], a car body design is based, initially, on bending and torsional stiffness as the main requirements concerning statical loads (crashworthiness is outside the scope of the current sub-project).

The optimization problem is formulated here for sizing optimization of the DutchEVO car bottom structure with respect for minimum mass. The structural optimization system developed earlier [Ermolaeva and Spoormaker 2001], based on the Multipoint Approximation method with Response Surface fitting (MARS) [Toropov 1989] and MSC.Marc FEA code, is adjusted here to be applied to the design of the considered structure. Problem dependent interfaces need to be programmed in order to connect structural optimization and FE analysis programs. In current paper, bending stiffness, strength and structural stability (non-buckling behaviour) constraints were considered as the first step of system application. Torsional stiffness requirement will be included later, after the proper adjustment of the system. The possibility to use different materials for the bottom structure is evaluated. Here we consider conventional structural materials and two composites with synthetic and natural fibres.

2. Structural and finite element model

The concept design of the DutchEVO car suggests the bottom structure to be constructed of panels and stiffeners, latter forming some kind of a frame on which panels are fixed (Fig. 1). The frame consists of cross-members between two reinforced side rocker beams and two backbone profiles. It is supposed to be made of a wrought aluminium alloy. The panels can be of an aluminium alloy too, but also there is a possibility to apply sandwich or composite panels instead. The composites reinforced by natural or synthetic fibres can reduce weight essentially. We also consider steel as a conventional material for automotive body structures for comparison and for the adjustment of the optimization system.

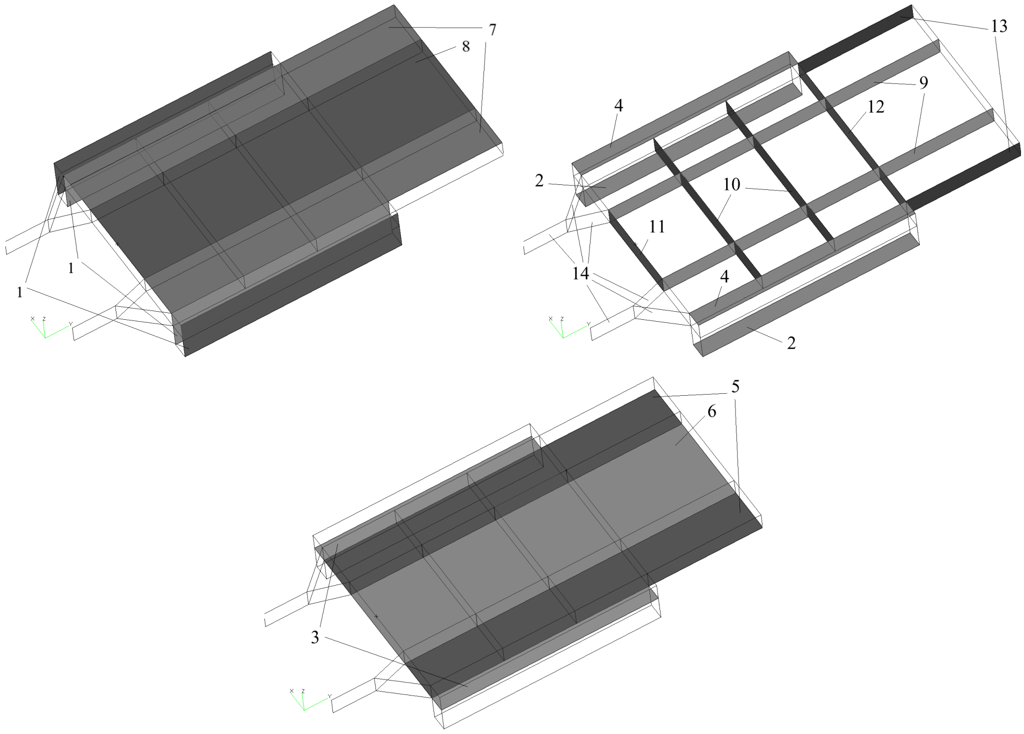

Figure 1. Structure of the bottom part of DutchEVO car: 1 – vertical and 2, 3, 4 – horizontal components of the side rocker beams; 5, 6 – components of the lower and 7, 8 – upper panels of the floor; 9, 13 – longitudinal and 10, 11, 12 – transverse stiffeners; 14 – backbone profiles

The FE model is presented in Fig. 2. It was tested for the applied design load in the middle of the span between hinges used as the supports. In this model, we use the four-node bilinear thick-shell element from MSC.Marc standard element library [MSC.Marc Volume B: Element Library, Version 2000]. A linear mechanical analysis is performed. Stresses and strains are calculated in each of the five layers of the element and in four integration points, within the plane of each layer. We also use the buckling option (via eigenvalue analysis) to obtain the first five buckling modes and the corresponding critical load.

3. Optimization problem

3.1 Formulation

A structural optimization problem for the given bottom structure of the DutchEVO car is formulated here as follows:

- minimize the mass of the structure under the constraints of bending stiffness, strength and buckling; the structural model along with the supports and the design load is presented in Fig. 1 and Fig. 2; (the requirement for the torsional stiffness will be included later, first the optimization system should be adjusted to the given model)
- design variables are the thicknesses of all plain structural elements, i.e. the components of the lower and upper panels and flats forming the cross-members (see Fig. 1);
- side constraints are limited from below based on sensitivity analysis, and from above – based on manufacturability of polymers.

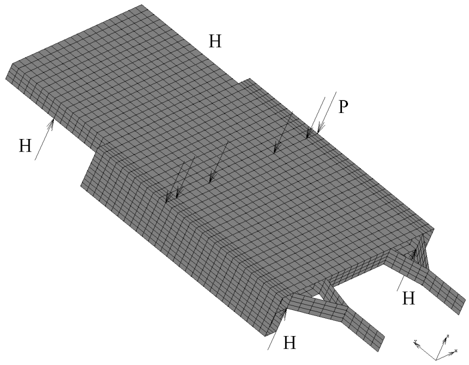

Figure 2. FE model of the bottom structure of DutchEVO car. P - design load; H – hinges

The mathematical formulation of this optimization problem in general form can be written as follows:
Minimize

$$F_0(\boldsymbol{x}) \rightarrow \min, \ \boldsymbol{x} \in \boldsymbol{R}^N \tag{1}$$

subject to

$$F_j(\boldsymbol{x}) \leq 1, \ \ j = 1,...,M \ \text{ and } \ A_i \leq x_i \leq B_i, \ i = 1,...,N \tag{2}$$

where $\boldsymbol{x}$ is a vector of design variables, F_0 is an objective function, F_j (j = 1,…,M) are constraint functions and A_i, B_i (I = 1,…,N) are side limits. In given formulation M = 3 and N = 14. Solution of the optimization problem is an iterative process that involves repetitive evaluations of the objective and constraint functions.
The objective function is expressed via the volume of the finite elements. This volume is calculated by means of a user subroutine based on nodes coordinates.
Constraints are specified as follows:

$$F_1(\boldsymbol{x}) = \frac{d_{\max}(\boldsymbol{x}) f_d}{d_0} \tag{3}$$

where $d_{\max}(\boldsymbol{x})$ is the maximum node deflection in the direction of the design load P; f_d is the factor of safety for bending stiffness; d_0 is the allowed deflection;

$$F_2(\boldsymbol{x}) = \frac{s_{eq,\max}(\boldsymbol{x}) f_s}{s_f} \tag{4}$$

where $s_{\mathrm{eq,max}}(\boldsymbol{x})$ is the maximum equivalent stress under the design load chosen from all integration points and layers of all finite elements in the model; f_s is the factor of safety for bending strength; s_{f} is the stress at failure (meaning of stress at failure depends on the type of material [Ashby 1999]);

$$F_3(\boldsymbol{x}) = \frac{P f_{buck}}{P_{cr}(\boldsymbol{x})} \tag{5}$$

where f_{buck} is the factor of safety for buckling mode; $P_{\mathrm{cr}}(\boldsymbol{x})$ is the lowest value of critical load.
The values of $d_{\max}(\boldsymbol{x})$, $s_{\mathrm{eq,max}}(\boldsymbol{x})$ and $P_{\mathrm{cr}}(\boldsymbol{x})$ result from the mechanical behaviour analysis in each design point.

3.2 Method

The Multipoint Approximation method based on Response Surface analysis (MARS) [Toropov 1989] was used as the structural optimization tool. This optimization technique is based on approximations of objective and constraint functions in specially or randomly distributed points of a design space. The random plan of numerical experiments gives an advantage to perform a continuous procedure even if there is no solution in one or several points of the design space. This situation often occurs during the FEA when structure buckles and FE program terminates with wrong exit number. The MARS technique skips this point and searches for the nearest point where the solution exists.

According to the approximation concept of the method the original functions (1) and (2) are replaced by approximate ones that considerably reduce computational time. Instead of the original optimization problem a succession of simpler approximated sub-problems, similar to the original one, is formulated:

minimize

$$\tilde{F}_0^k\left(\boldsymbol{x}, \boldsymbol{a}^k\right) \rightarrow \min, \ \boldsymbol{x} \in \boldsymbol{R}^N \tag{6}$$

subject to

$$\tilde{F}_j^k\left(\boldsymbol{x}, \boldsymbol{a}^k\right) \leq 1, \ \ j = 1,..., M \ \text{ and } \ A_i^k \leq x_i \leq B_i^k, \ A_i^k \geq A_i, \ B_i^k \leq B_i, \ i = 1,..., N \tag{7}$$

The current sub-optimal point $\boldsymbol{x}_*^k$ is considered as a starting point of the next (k + 1) iteration. Each function $\tilde{F}^k$ in (6) and (7) is an explicit approximation; $\boldsymbol{a}^k$ is a vector of tuning parameters; A_i^k and B_i^k are move limits defining the range of applicability of the approximations.

The weight least-squares minimization problem is to be solved to determine the components of vector $\boldsymbol{a}$. The iteration process stops when one of the termination conditions occurs, for example, if the approximations are sufficiently good, none of the move limits is active, and the search sub-region is small enough.

The MARS programming code contains the common type of built-in approximations: linear, multiplicative and reciprocal. In order to reduce the computational time, the mechanistic approach can be implemented to develop simplified approximations of the constraint functions. It was shown earlier [Ermolaeva and Spoormaker 2001] that if the mechanical behaviour of an optimized structure, or the parts of this structure, can be even approximately described by engineering equations then mechanistic approximations can be easily derived and applied to corresponding mechanical constraints. In this case, the convergence of the optimization procedure needs less number of iterations and, hence less structural analysis calculations. The mechanistic approximations derived for the side rocker of the DutchEVO car, represented by hollow rectangular beam, reduced the number of FEA runs by 3-4 times [Ermolaeva and Spoormaker 2001]. Mechanistic approximations for the considered structure will be shown in the following section.

4. Optimization system adjustment

4.1 MARS optimization system

The optimization system usually combines three following parts: optimization programs (optimizer), problem-dependent analyzer and interface programs that connect the optimizer with the analysis software. These interfaces depend on the structure of the analysis code and have to be created for each specific application. We showed earlier [Ermolaeva and Spoormaker 2001] how the MARS optimization code [Toropov 1989] could be linked to MSC.Marc FEA package. The MSC.Marc FEA package provides a user subroutines feature, which allows users to substitute their own subroutines for those existing in the program [MSC.Marc Volume D: Users Subroutines and Special Routines, Version 2000]. It sufficiently simplifies interface programming and makes the MSC.Marc FEA code attractive to user willing to interfere in the calculation process. In addition to subroutines used in previous version of the MARS system [Ermolaeva and Spoormaker 2001], that allow one to get necessary information for particular nodes and elements, for current problem we employ the subroutine 'ufconn.f', in order to obtain a connectivity matrix used further for the volume of elements estimation. All programs: optimizer, analyzer and interfaces, are combined in MARS optimization system via 'start_optimization.bat' file to start the optimization procedure. A scheme of the developed optimization system one can find in [Ermolaeva, Willemen and Spoormaker, 2001].

4.2 Model sensitivity analysis

In order to evaluate the sensitivity of the constraints and objective function to the change in variables, the following analysis was done. The FEA of the modelled structure was performed for one of the variables changed while the other variables being constant. It resulted in the following (see geometry components thickness of which correspond to the design variables in Fig. 1):

- geometry 1 (walls of the rocker beams) and geometry 5 (outer panels of the lower part of the floor) influence the critical load sign (negative values of a critical load cannot be processed by the optimization code);
- geometries 2, 3 and 4 (horizontal components of the rocker beams) affect mostly the bending stiffness constraint and stress-strain behaviour of the structure;
- geometries 6 (central panel of the lower part of the floor), 7, 8, (components of the floor upper panel), 9, 13 (longitudinal stiffeners) and 14 (a component for engine support at the front part) influence generally the critical load.
- the buckling constraint appeared to be the most sensitive function.

According to this analysis the lower limits has been adjusted to the current optimization problem.

4.3 Mechanistic approximation

Approximations chosen for the current optimization problem are the following: linear – for the objective function, reciprocal – for bending stiffness constraint, multiplicative – for strength constraint and, initially, for buckling constraint. Trial calculations showed that all approximations except for buckling constraint are sufficiently good. For the buckling constraint, we tried also the reciprocal approximation from the set of the built-in functions and proposed to use mechanistic approach to derive an approximation that is close to the buckling behaviour of the given structure. Within the mechanistic approach, the simplified model is employed to build the approximations (indices *k* and *j* are omitted here for simplicity):

$$\tilde{F}(\boldsymbol{x},\boldsymbol{a}) = \tilde{F}(f(\boldsymbol{x}),\boldsymbol{a}), \tag{8}$$

where $f(\boldsymbol{x})$ is the function presenting the structural response using the simplified model. The approximation based on the simplified model satisfies all basic requirements. It reflects the main properties of the original complex model; it is computationally less expensive and relatively noiseless [Toropov and Markine 1996].

The given structure is a complex built-up construction consisting of many plane elements. The structural instability of this structure is defined mainly by the local buckling modes of its components. It is reasonable to take into account the local modes of each structural element, the critical load for which is proportional to the thickness of the component in power three. Then the approximation function for buckling constraint could be written in the following form:

$$\tilde{F}(\boldsymbol{x},\boldsymbol{a})_3 = a_0 + \sum_{i=1}^{N} \frac{a_i}{x_i^3}\,. \tag{9}$$

Analysis of different types of approximations for buckling constraint (see Fig. 3) gives the following result. Application of multiplicative approximation shows the tendency to non-convergence of the optimization problem. Comparison of reciprocal and mechanistic approximations results in less variations of analysed function for the latter approximation type.

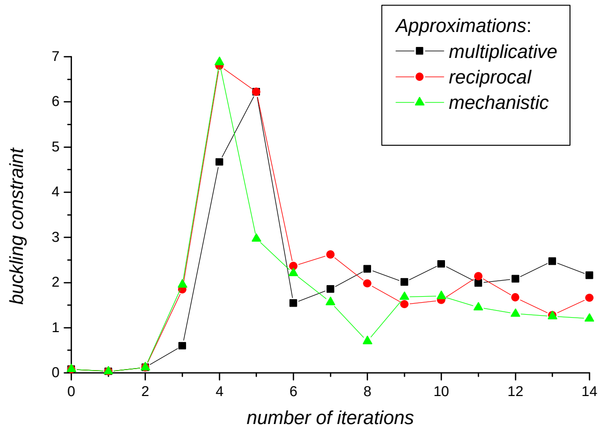

Figure 3. Behaviour of the buckling constraint of different approximation types

4.4 Sensitivity to the initial design

In order to be sure that the optimal solution is global, the sensitivity of the constraints and objective function to the initial design change was analysed. The starting point in the space of design variables was taken in three variants:

- model Bend_09_k1 – x_i for i = 5, …, 9, 13, 14 were taken at their lower side limits; all other variables were taken equal to 1 mm;
- model Bend_09_k2 – 4 mm for all variables;
- model Bend_09_k3 – 8 mm for all variables.

The values of the constraints and the objective function for the optimal solutions under different initial designs are presented in Table 1.

Table 1. Sensitivity to the initial design

Model	Number of response analyses	Bending stiffness constraint	Strength constraint	Buckling constraint	Optimal mass, kg	Error, %
Bend_09_k1	211	0.994	0.236	0.951	96.9	-
Bend_09_k2	136	1.000	0.236	0.953	96.6	0.3
Bend_09_k3	243	0.961	0.234	0.951	98.1	1.6

As one can see the difference in optimal mass is within the acceptable range of engineering error. Buckling mode for these optimal solutions is the same – of local type: upper panel between longitudinal and transverse stiffeners (Fig. 4).

5. Results of the optimization

Results of the structural optimization for the bottom structure made of steel are presented in the previous Section. According to the results in Table 1, the initial design for model Bend_09_k2 can be considered as preferable, because of the minimal value of the objective function and lowest number of FEA runs (consequently, the lowest number of iteration steps and computational time). The bending stiffness constraint appears to be active in this solution.

Optimisation of the given structure made of other materials needs additional adjustment through the analyses of the optimisation process and achieved solutions. Approximation functions and initial design were taken as in the previous model for steel structure. Lower bound limits were changed. They were set up during the system adjustment. This makes difficult to compare the optimisation results, because due to these changes the initial formulation of the optimisation problem cannot be considered completely the same for the structures made of different materials. Nevertheless, optimisation solutions for a wrought aluminium alloy structure (model Bend_10_b) and the structure composed of an aluminium alloy frame covered with composite plates of two different compositions are given in Table 2. Applied composite materials were polypropylene (PP) matrix reinforced with randomly distributed glass (model Bend_11_b) or hemp fibres (model Bend_12_a); the latter is the representative of natural fibre composites.

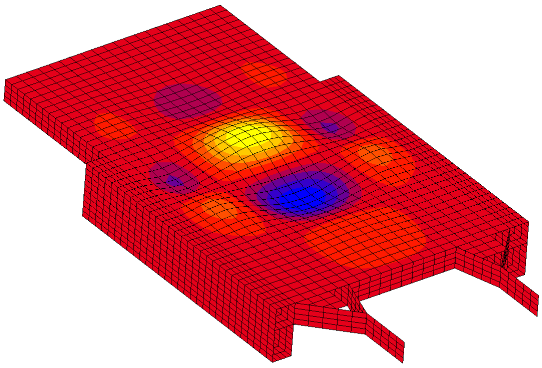

Figure 4. Buckling of the bottom structure

As one can see in Table 2, under the given formulation of optimization problem the bottom structure made of aluminium alloy has the lowest weight – 51.8 kg, which is about 46% lighter than the same steel structure. The vector of design variables (thicknesses of structural components in mm) corresponding to this optimal solution is the following – $\boldsymbol{x}$ = (1.0, 1.1, 3.0, 3.6, 1.3, 2.3, 1.5, 2.2, 1.8, 1.7, 4.0, 1.0, 3.4, 1.8). The bending stiffness and buckling are the active constraints. The structures made of aluminium alloy frame covered with the PP composite floor panels are 13% heavier than the same structure from only aluminium alloy.

The current result cannot be considered as direct recommendation for final design. It is true only for given formulation of the considered optimization problem, and can be far from the real optimum. Let us call that no technological aspects of assembly, as joint material (weld or adhesive), or production, except of the limitation of thicknesses due to manufacturability of polymer plates, are applied here. On the other hand, it is possible that the weight of the bottom structure can be further decreased. Considered here composite floor panels are comparatively weak due to the random distribution of short fibres. If the mechanical properties of the applied composites can be improved, due for example to oriented fibres, or special treatment of fibres and/or matrix, then the aluminium frame structure with the PP/hemp fibres composite floor panels might be lighter and can compete with the aluminium alloy structure. The torsion stiffness should be included into consideration to make the optimal problem formulation complete for static load resistance of the DutchEVO car bottom structure.

Table 2. Optimization results

model	Bending stiffness constraint	Strength constraint	Buckling constraint	Optimal mass, kg	Optimal mass relative to steel
Bend_10_b	0.992	0.167	1.0	51.8	0.537
Bend_11_b	0.981	0.396	1.0	62.2	0.644
Bend_12_a	0.996	0.406	0.949	59.2	0.613

6. Conclusion

In current paper, the sizing optimization problem of the DutchEVO car bottom structure with respect to minimum mass under bending load case requirements was formulated. FE model of the given structure was built and tested. Problem dependent interfaces were programmed in order to connect structural optimization and FE analysis programs. The MARS optimization system based on the Multipoint Approximation method with Response Surface fitting and MSC.Marc FEA code was adjusted here to be applied to the design of the considered structure. The mechanistic approach was implemented to define the buckling constraint approximation function.

The possibility to use different materials for the bottom structure was evaluated. In current paper, we considered conventional structural materials, steel and wrought aluminium alloy, and two composites with synthetic and natural fibres. It was shown that under given formulation of the optimization problem the bottom structure made of a wrought aluminium alloy has a lowest weight, which is about 46% less than for steel structure. The other considered materials also showed the potential to be used in given structural application.

In current paper, bending stiffness, strength and structural stability (non-buckling behaviour) constraints were considered as the first step of system application. In order to make an optimal design complete with respect to statical loads resistance, the torsional stiffness requirement should be considered in addition. This will be done as the next step of the development and application of the optimization system.

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Ermolaeva N.S., Dr.,
Delft University of Technology, Faculty of Design Engineering and Production, Industrial Design Engineering,
Landbergstraat 15, 2628 CE, Delft, the Netherlands
Tel: +31 15 278 5726
Fax: +31 15 278 1839
Email: N.Ermolaeva@io.tudelft.nl