

Research Article

Lightweight Design of an Outer Tie Rod for an Electrical Vehicle

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The lightweight design of the outer tie rod installed on an electrical vehicle was achieved through material selection and optimization technique. The aluminum alloy Al6082M was selected as a steel-substitute, and its structural shape was optimized by applying metamodel-based optimization. In this process, finite element analysis was performed to predict the structural responses, such as buckling resistance and fatigue life. First, for an arbitrary base design made of steel, the structural responses were calculated. Then, the design variables were defined to find a lightweight design made of Al6082M. Secondly, metamodel-based optimization based on the kriging interpolation method was applied, leading to determination of an optimum design. The suggested optimum design has the minimum weight satisfying the critical design requirement. Finally, the numerical results of the buckling resistance and fatigue life were validated, through buckling and fatigue tests.

1. Introduction

The current trend of the structural design of automobile parts is towards lightweight design. Lightweight automobile parts can be developed by selecting steel substitutes, such as aluminum, magnesium, plastic, or composite material, applying manufacturing technology and/or adopting optimization techniques [1–3]. Lower weight requires lower energy consumption. In particular, the realization of lightweight design in electrical vehicles gives rise to an ecofriendly vehicle with high efficiency. The importance of lightweight design in the automotive industry has increased, due to the consistently tightening regulations for protecting the environment [3, 4].

There have been many cases of lightweight structural design in body and chassis parts during the past 20~30 years. Among the parts of suspension and steering systems, there have been many cases of lightweight design application for the control arm or knuckle, but there are very few cases for the outer tie rod [5]. This is because the OTR is relatively much lighter than the other parts. However, along with recent trends, car makers and parts manufacturers are interested in lightweight design of the OTR. That is why some parts are developed with their target weight predetermined in

units of gram-force during the proto-design stage [3, 5]. Furthermore, consistent lightweight design is an essential means of extending the driving range of electric vehicles [4].

In this study, a lightweight design of the OTR for an electrical vehicle is suggested, by substituting SM45C with Al6082M and applying optimization techniques, considering the buckling and durability performance. The structural responses of buckling and durability are considered in the structural design process of the outer tie rod. Usually, since the buckling is the critical performance, only the buckling is included in the optimization process. Then, after an optimum design is determined, the durability analysis is performed, to investigate whether the suggested design satisfies its criterion.

The most exact method to determine the optimum design is to utilize the sensitivity information of responses and a gradient-based algorithm, since it can provide one design satisfying the K-T necessary condition. In contrast, a metamodel-based optimization method is suitable for problems of structural design requiring much computation time, difficult computational sensitivity problems, or problems having a wavy response function, even though it cannot give the K-T point [3, 5]. The optimization problem of the OTR belongs to this category. Thus, this research

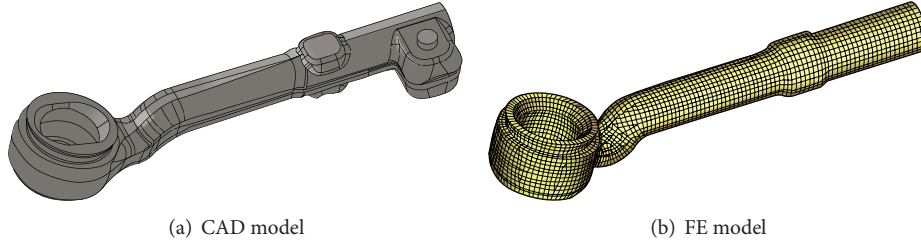


FIGURE 1: The base model of steel OTR.

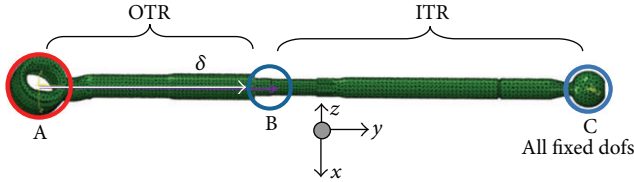


FIGURE 2: Loading condition for buckling analysis.

adopts metamodel-based optimization, to find an optimum design of the OTR. It is known that the kriging metamodel provides accurate predictions of highly nonlinear responses. The design process using the kriging metamodel is as follows [6, 7]. First, the DOE using Latin hypercube design is carried out, to define the design samples. Second, buckling analysis is performed for as many as the number of sample points. Third, surrogate models of the related responses are built. Thus, the responses of weight and buckling resistance force are replaced by kriging metamodels. It is easy to solve the approximated formulation, since all the responses are replaced by simple mathematical equations. Thus, it is not important to select an algorithm to solve the formulation in this step. As a final step, an optimum design is calculated, by considering formulation. Then, the fatigue life of the suggested optimum design is calculated.

For the validation of the suggested optimum design, buckling and fatigue tests were performed. The design of the OTR was done using CATIA [8], MSC. Nastran [9], and MSC. Fatigue [10] for the durability analysis and Abaqus [11] for the buckling analysis.

2. Base Design of the Steel OTR

The OTR is a part that belongs to the steering system changing directions, according to the motion of the steering wheel. Going from steering wheel to steering gear, pitman arm, relay rod, and ITR (inner tie rod), steering power is finally delivered to the wheel through the knuckle after passing through the OTR [5]. The ITR is attached to the OTR and is served as pivot point for the steering gear. The base design of the OTR and its structural analysis were executed for SM45C early on during this car program. However, its material was changed to Al6082M to reduce the weight more. Thus, the structural analysis of the steel OTR was investigated in the early proto-design stage.

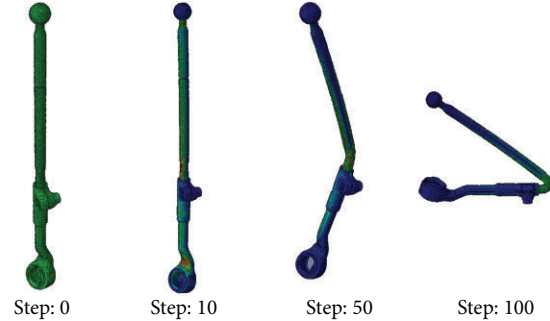


FIGURE 3: Buckling analysis result of steel OTR.

The CAD model and FE model of the initial design are shown in Figure 1. The weight of the steel OTR without bolt and nut was about 378 g. Usually, buckling is the most critical performance in the structural design of an OTR. That is, if the constraint related to buckling is satisfied, the rest of the structural constraints lie in their allowable values. Thus, in this study, only the buckling is included in the optimization process.

The loading condition for buckling analysis is shown as Figure 2. The steel OTR is connected with the ITR at point B. Thus, the buckling analysis was performed in the combined state. The ITR was considered as a predetermined product, since the part manufacturer involved is just responsible for the design of the OTR. A load δ is applied sequentially on point A in the y direction, by fixing all degree of freedoms of point C. At that time, the buckling resistance force can be determined. The buckling resistance force is the reaction force of point C, at the moment when the buckling is generated. The result of the buckling analysis is represented in Figure 3. The buckling resistance force of the steel OTR was obtained as 32.6 kN, which satisfies the design criterion specified by car maker A.

3. Structural Optimization of the Aluminum OTR

3.1. Definition of Design Variable and Optimization Formulation. In general, aluminum has been selected as a steel-substitute material to reduce the weight, since aluminum has almost one-third of the density of steel. In this research, aluminum called Al6082M [3, 5] developed by the part manufacturer was used to replace the steel. However,

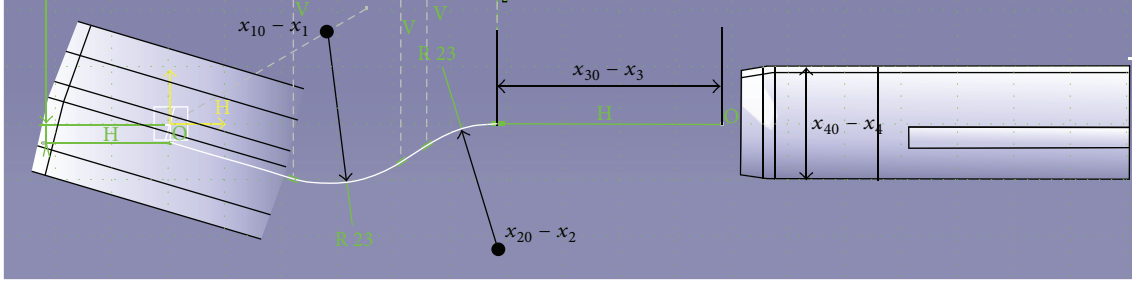


FIGURE 4: Shape design variables.

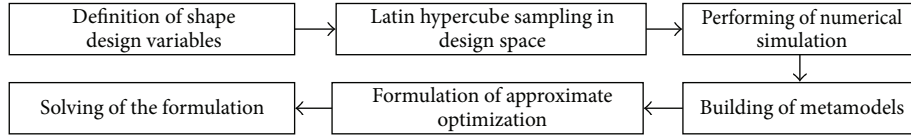


FIGURE 5: Flow chart for the structural design of Al6082M OTR.

TABLE 1: Material properties of SM45C and Al6082M.

Property	SM45C	Al6082M
Yield strength (MPa)	343	340
Tensile strength (MPa)	569	380
Young's modulus (GPa)	210	72
Density ($\times 10^{-6}$ kg/mm ³)	7.85	2.71

the strength and stiffness of aluminum have lower values than those of steel for the same structure. Thus, it is necessary to find an optimized shape of the OTR made of Al6082M. The material properties of SM45C and Al6082M are summarized in Table 1.

Based on the modified design of the steel OTR, the design variables defining its shape are represented in Figure 4. The design variables, x_1 and x_2 , are related to the shape of its neck, defining the radius curvatures, while the design variables, x_3 and x_4 , are related to the length and sectional size of the straight part. In Figure 4, the values of $[x_{10} \ x_{20} \ x_{30} \ x_{40}]^T$ are $[23^\circ \ 23^\circ \ 58 \text{ mm} \ 18.5 \text{ mm}]^T$.

The optimization problem determining its optimum shape can be formulated as

$$\text{Minimize} \quad W(\mathbf{x}) \quad (1)$$

$$\text{subject to} \quad F_R(\mathbf{x}) \geq F_0 \quad (2)$$

$$\mathbf{x}_L \leq \mathbf{x} \leq \mathbf{x}_U, \quad (3)$$

where $\mathbf{x} = [x_1 \ x_2 \ x_3 \ x_4]^T$, $\mathbf{x}_L = [0^\circ \ 0^\circ \ 0 \text{ mm} \ 0 \text{ mm}]^T$, $\mathbf{x}_U = [9.0^\circ \ 3.0^\circ \ 15.0 \text{ mm} \ 2.0 \text{ mm}]^T$, W is the weight of the aluminum OTR, F_R is its buckling load, and F_0 is the allowable value.

Shape optimization based on the sensitivity information makes the setting of a shape variable difficult and distorts the finite elements in the optimization process [2, 3, 5, 6]. In addition, a number of nonlinear analyses are performed in an optimization process to solve (1)~(3), and the iterations

may not be terminated. Accordingly, it is more realistic that optimization is carried out by replacing the buckling load, F_R appearing in (2), with the metamodel. There are methods that generate a surrogate model, such as a response surface model, kriging, a neural network, and others. The present research utilizes the kriging metamodel, suitable for prediction of a highly nonlinear function [5, 6]. When an approximate model is used, the objective and constraint functions in (1)~(2) can be replaced as follows:

$$\text{Minimize} \quad \hat{w}(\mathbf{x}) \quad (4)$$

$$\text{subject to} \quad \hat{F}_R \geq F_0, \quad (5)$$

where $\hat{\cdot}$ means the metamodel of a response.

The buckling load should be larger than the limit, 25.0 kN. However, the allowable value of F_0 in (5) was set to 27.0 kN to compensate the discrepancy between numerical result and true response.

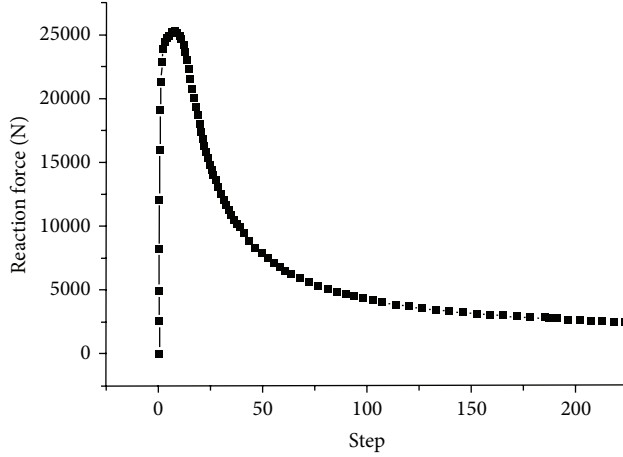
3.2. Application of Kriging Method to the Aluminum OTR. The flow chart for the structural design of the aluminum OTR is shown as in Figure 5 [7]. First, the DOE (design of experiments) is carried out to define the design samples. As a sampling method, the Latin hypercube design built into Matlab was utilized. In this step, the number of sample points was 50, which was determined by experience [2, 3, 5, 6]. Second, FE analysis using the Abaqus [6] was performed, with a number of sample points. The FE analysis on each row was carried out. The sample points and their responses are summarized in Table 2. Third, the kriging models of the weight and the buckling load were built. Thus, (4) and (5) could be solved by applying the GRG (generalized reduced gradient) algorithm.

In the kriging model, the global approximation model $\hat{f}(\mathbf{x})$ for a response $f(\mathbf{x})$ is represented as

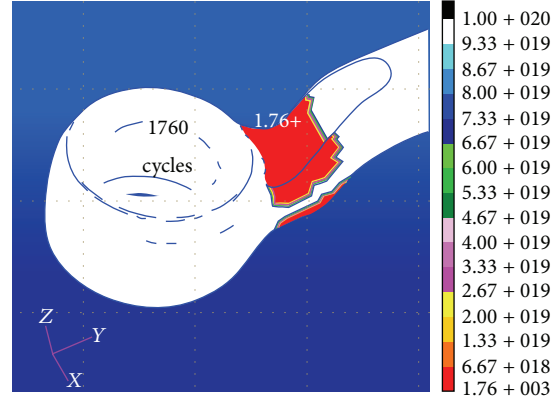
$$\hat{f}(\mathbf{x}) = \hat{\beta} + \mathbf{r}^T(\mathbf{x}) \mathbf{R}^{-1}(\mathbf{f} - \hat{\beta}\mathbf{i}), \quad (6)$$

TABLE 2: Design of experiments using LHD.

Exp. number	x_1 (°)	x_2 (°)	x_3 (mm)	x_4 (mm)	F_R (N)	W (g)
1	0.2	1.8	11.1	1.5	26,083	135.1
2	5.4	2.0	1.9	0.1	26,821	143.5
3	7.1	1.0	5.8	0.2	28,825	143.1
4				0.8	26,380	139.0
⋮	⋮	⋮	⋮	⋮	⋮	⋮
49	3.5	2.3	14.0	0.2	26,429	136.2
50	1.4	1.8	5.3	0.9	26,961	140.0



(a) Buckling analysis result



(b) Durability analysis result

FIGURE 6: Buckling and durability analysis result of optimum Al6082M OTR.

where \mathbf{R}^{-1} is the inverse of the correlation matrix \mathbf{R} , \mathbf{r} is the correlation vector, \mathbf{f} is the n_s observed data vector, and \mathbf{i} is the unit vector. The correlation matrix and correlation vector are defined as

$$R(\mathbf{x}^j, \mathbf{x}^k) = \text{Exp} \left[- \sum_{i=1}^n \theta_i |x_i^j - x_i^k|^2 \right], \quad (7)$$

$$(j = 1, \dots, n_s, k = 1, \dots, n_s),$$

$$\mathbf{r}(\mathbf{x}) = [R(\mathbf{x}, \mathbf{x}^{(1)}), R(\mathbf{x}, \mathbf{x}^{(2)}), \dots, R(\mathbf{x}, \mathbf{x}^{(n_s)})]^T,$$

where the number of observed data is 50, and the number of design variables is 4 in this study.

The unknown parameters $\theta_1, \theta_2, \dots, \theta_n$ are obtained from the following equation:

$$\text{maximize} - \frac{[n_s \ln(\hat{\sigma}^2) + \ln |\mathbf{R}|]}{2}, \quad (8)$$

where θ_i ($i = 1, 2, \dots, n$) > 0 . To solve (8), the GRG algorithm built into EXCEL was utilized. The Kriging interpolation method is well explained in [2, 3, 12–14].

3.3. Optimum Design Result and Durability Analysis. This research utilized an in-house program [3] to solve the

optimization problem defined in (3)~(5). The optimizer of the in-house program is the GRG method, which is an optimization algorithm built into the EXCEL program. The optimum parameters were investigated by changing more than 10 initial values. Then, the metamodels were built, by setting the weight and reaction force as response values.

For the kriging models of weight and buckling load, their optimum parameters are summarized in Table 3. These optimal parameters were used to find the optimum design of (3)~(5), and as a result, $[x_1^* \ x_2^* \ x_3^* \ x_4^*]^T = [4.24^\circ \ 2.39^\circ \ 11.29 \text{ mm} \ 1.86 \text{ mm}]^T$ was found, and the approximate weight and the approximate buckling load at the optimal solution were, respectively, 130.7 g and 27,205 N. The true weight of the optimum design was 131.0 g. The buckling analysis result represented as a force-displacement curve is shown in Figure 6(a). The true buckling load of the optimum design was calculated as 28,024 N.

For the suggested optimum design, durability analysis for the unit of OTR was performed, to investigate the design requirement imposed on the OTR. The loading condition applying an equivalent load is represented as a sine curve. The loading condition and design criterion were supplied by car maker A. The fatigue life calculated from MSC Fatigue is shown in Figure 6(b). The fatigue life of the suggested optimum design was determined as 1,760 cycles, which satisfies the design requirement specified by car maker A.

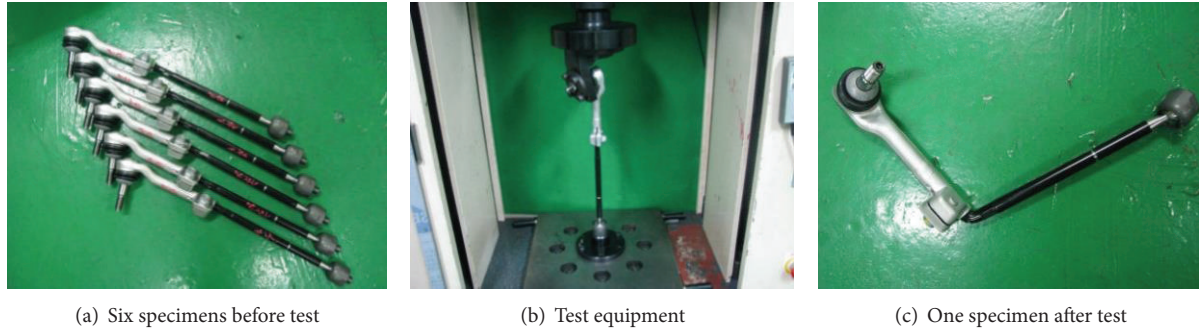


FIGURE 7: Buckling test result.

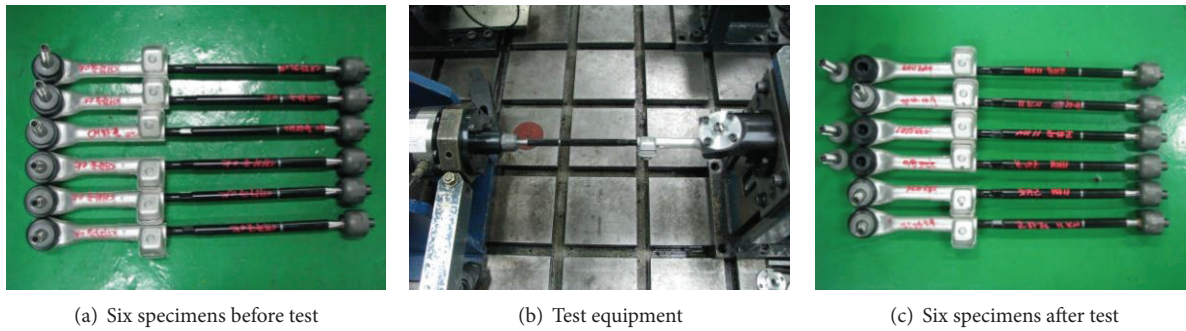


FIGURE 8: Fatigue test result.

TABLE 3: Optimal parameters of kriging models.

Response	β	θ_1	θ_2	θ_3	θ_4
W	138.6 kg _r	0.004	20.54	4.74	10.33
F_R	27,690 N	0.001	5.07	18.90	50.00

3.4. Buckling and Fatigue Tests. Based on the suggested optimum design, twelve specimens were made to validate the buckling and durability performance. For the buckling test, six specimens of the OTR connected with the ITR are shown in Figure 7(a). The test equipment is shown in Figure 7(b), in which the loading and boundary conditions are the same as those of the buckling analysis. As a result, the buckling loads were between 25,900~27,550 N, and all of them satisfy the criterion. Comparing the average value obtained from six buckling loads, the simulation result has a relatively small error, of about 5%. One specimen after the buckling test is shown in Figure 7(c).

For the durability test, six other specimens were prepared, as shown in Figure 8(a). The test equipment for the durability was set up as shown in Figure 8(b). Through the test, their fatigue lives were between 249,049 cycles and 312,659 cycles. Those results satisfy the requirement set by car maker A. It is found that the test results of the buckling and durability are consistent with those of the simulations.

4. Conclusions

The present research suggested an optimum design of Al6082M OTR, which is one of the parts of the steering

system of an electrical vehicle. The following conclusions can be made from this study.

- (1) The present research has succeeded in the development of a lightweight OTR, by replacing the current OTR, whose initial model was made of steel, with an Al6082M OTR, and applying the optimization technique. The metamodel-based optimization technique provides a realistic optimum, even though it cannot be considered as an exact solution. The weight of the suggested optimum design is 131.0 g, which is 65.3% lighter weight than the initial steel model.
- (2) It is proven that the design requirements related to the buckling and durability performance were satisfied through the tests. For the buckling test, six specimens were used, and their buckling loads were between 25,900~27,550 N. In the case of the durability test, all the fatigue lives of the six specimens exceeded the allowable cycle.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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