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INDUSTRIAL APPLICATION

Lightweight flatbed trailer design by using topology and thickness optimization

Gang-Won Jang · Min-Su Yoon · Jae Ha Park

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Abstract A new design for a lightweight flatbed trailer with high bending stiffness and torsional frequency is presented. The design procedure consists of two main steps: topology optimization and thickness optimization. During topology optimization, a creative frame layout different from existing ladder-type frames can be obtained by searching the best layout out of all possible layouts of a simplified design domain model. After approximating the result of topology optimization as a thin-walled structure, the approximated thicknesses of the plates are optimized to minimize the mass of a trailer. The bending stiffness and torsional frequency obtained by topology optimization are set as design constraints for thickness optimization. Due to the closed cross-section, the optimized trailer can efficiently increase the stiffness-to-mass ratio to a large extent. Discrete thicknesses are employed as design variables for thickness optimization so that the thicknesses of the plates of a trailer can be included in those of commercially available high-strength steel products. The final model has a 29% reduction in total mass, a 21% decrease in mean compliance with a uniform bending load, and a 169% increase in torsional frequency.

G.-W. Jang (✉) · J. H. Park
School of Mechanical and Automotive Engineering,
Kunsan National University, Kunsan,
Jeonbuk 573-701, Korea
e-mail: gangwon@kunsan.ac.kr

M.-S. Yoon
Research Institute of Industrial Science & Technology,
79-5 Youngcheon, Dongtan, Hwaseong,
Gyeonggi 445-813, Korea

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1 Introduction

High stiffness, high strength, and light weight are important issues when designing vehicle structures. To achieve such goals, the recent applications of CAE-based structural optimizations to the design of lightweight vehicle parts with high static and dynamic performances are regarded as efficient approaches. For example, Lan et al. (2004) reduced the body weight of a bus without losing its strength and rigidity by optimizing the cross-sectional parameters and wall thicknesses. Jung and Kwon (2006) improved the static stiffness and dynamic crash behavior of a rolling stock end frame of a train cabin by using topology optimization. Yoshimura et al. (2005) optimized automobile frame cross sections, types of material, and plate thicknesses in the form of a multi-objective problem by using genetic algorithms.

In this investigation, a flatbed trailer is optimized to have a high stiffness-to-mass ratio by applying CAE-based optimization tools within consistent problem formulations. Since structural optimizations have not been widely applied to the design of heavy vehicle structures such as a flatbed trailer, substantial improvement in structural performances can be expected by using a systematic optimization procedure. Figure 1 illustrates the proposed procedure of the optimizations. The ultimate design goal is to increase the bending stiffness and torsional frequency of a flatbed trailer with less mass than that of the original trailer model.

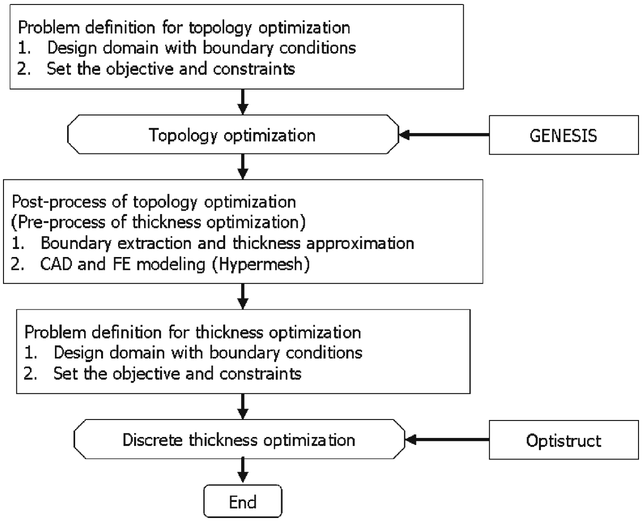


Fig. 1 A design procedure for a flatbed trailer

The proposed design procedure consists of two main optimization steps: topology optimization and thickness optimization. The focusing performances are set differently during the two optimization steps. In topology optimization, a new layout design to maximize bending stiffness and torsional frequency is to be found with the same amount of mass as that of the original trailer model. The most attractive feature of topology optimization is that a new design free from the existing structural layouts can be expected. This is because topology optimization does not modify the existing models but starts from a simplified initial model (or a design domain model) and searches for the best layout among all possible solutions of the design domain model. In this study, a commercial software, GENESIS (Vanderplaats R&D 2005), is used for topology optimization.

In many cases, especially in the case of three-dimensional problems, the optimized results of topology optimization do not have smooth boundaries, so they are not favorable for manufacturing. Also, because initial topology optimization models are discretized with solid elements, it is not easy to obtain results consisting of plate-like thin-walled members. If a very large number of solid elements are employed for

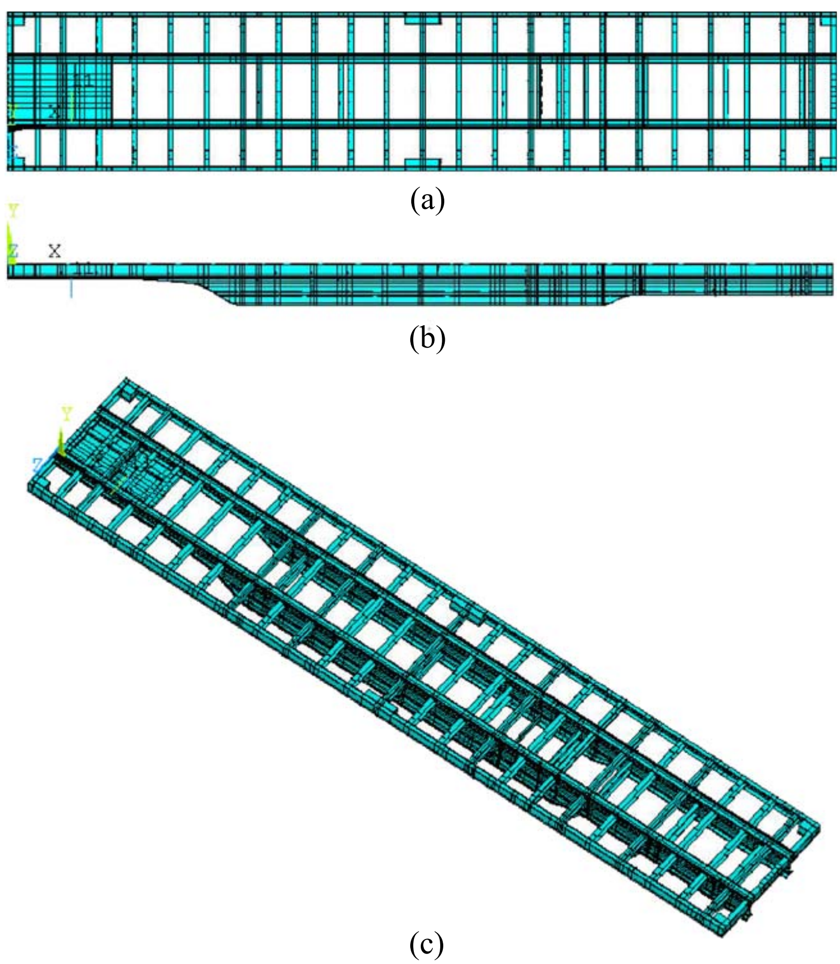


Fig. 2 The original model of the flatbed trailer: **a** top view, **b** side view, and **c** isotropic view

discretization of an initial topology optimization model, thin-walled members with relatively smooth boundaries can be expected. However, this is accompanied by a tremendous increase in numerical cost. Thus, instead of increasing the mesh density of an initial topology optimization model, it is desirable to post-process the result of topology optimization as a thin-walled structure with approximated thicknesses and further optimize the thicknesses by using consistent problem formulation with topology optimization.

Thus, during the next optimization step, the optimal plate thicknesses are to be found to have minimum trailer mass without spoiling the performances obtained as a result of topology optimization. In thickness optimization, the optimization proceeds by focusing on reducing the total mass while the optimized structural stiffness by topology optimization is maintained; the mass of a trailer is set as the design objective, and the bending stiffness and torsional frequency are constrained above the optimized values of topology optimization. As in topology optimization, commercial software is employed for thickness optimization. Because the result of topology optimization needs a post-process based on a designer's engineering experience, which, in the end, should be modeled by using a CAD software, the use of the same software for topology and thickness optimization does not make the design process fast. Thus Hypermesh (Altair 2007a) is used to build a finite element model from the post-processed result of topology optimization, and Optistruct (Altair 2007b) is used for thickness optimization.

From the viewpoint of manufacturability, the optimized thicknesses of the plates should be available from existing mass-produced plates with discrete thicknesses. The final model of a trailer is expected to be manufactured by using high-strength steel, ATOS 80 (POSCO 2008, http://www.posco.co.kr). So, discrete design variables are used for thickness optimization

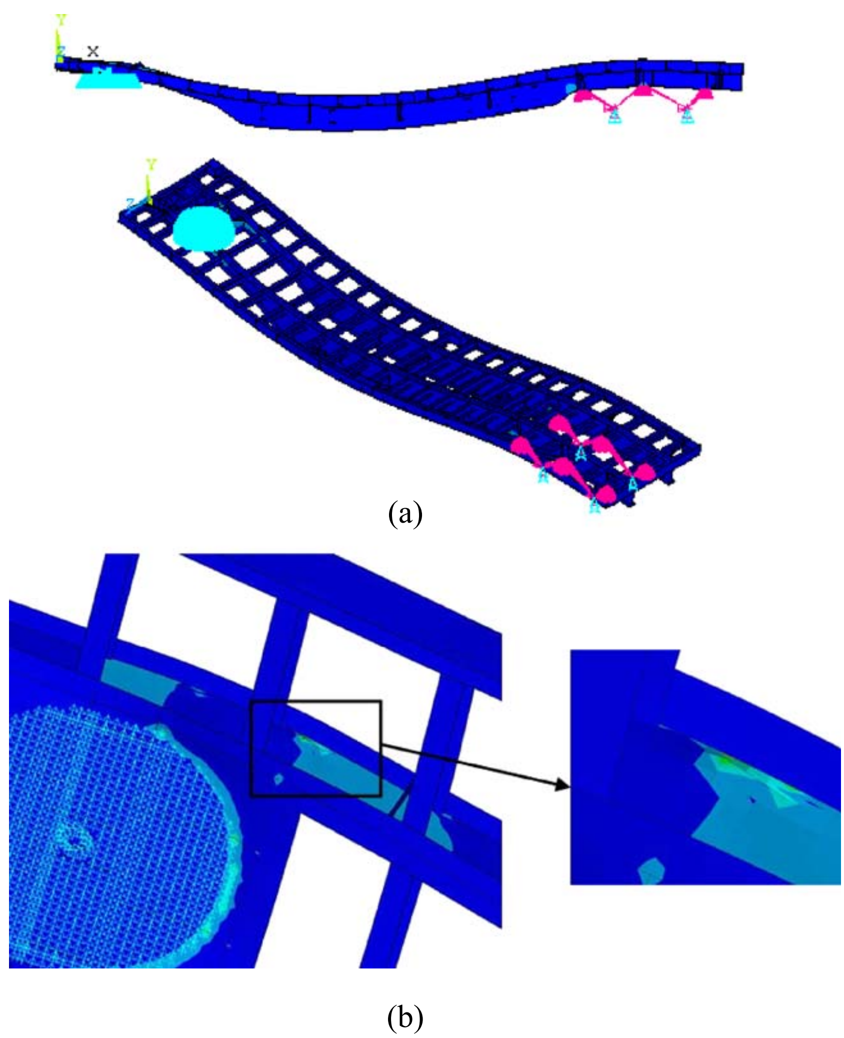


Fig. 3 Analysis results of the original model under uniform bending pressure along the master beams: **a** deformation and **b** maximum stress point around the connection part with the trailer

so that plate thicknesses can be included in the mass-produced dimensions of ATOS 80.

2 New layout design by topology optimization

2.1 Original model

Figure 2 shows an original model of a flatbed trailer that consists of two master beams, 23 cross beams for the upper layer, and nine cross beams for the lower layer. The master beams placed along the length of the trailer mainly support bending loads while the cross beams perpendicular to the master beams act as torsional stiffness members. To increase the torsional stiffness of a trailer, cross beams are connected between the top or bottom flanges of the master beams. All the beam members have open cross sections with different thicknesses depending on their locations. The length and width of the trailer are 12,226 mm and 2,500 mm, respectively, and the total mass is 2,468 kg. A tractor is connected with the trailer through a king pin, which is shown in the rectangular panel on the left of Fig. 2a. In the figure, two wheel axles are positioned below the right side of the trailer.

In this study, the bending stiffness and torsional frequency of the trailer are set to be the target optimization performances while the mass is minimized.

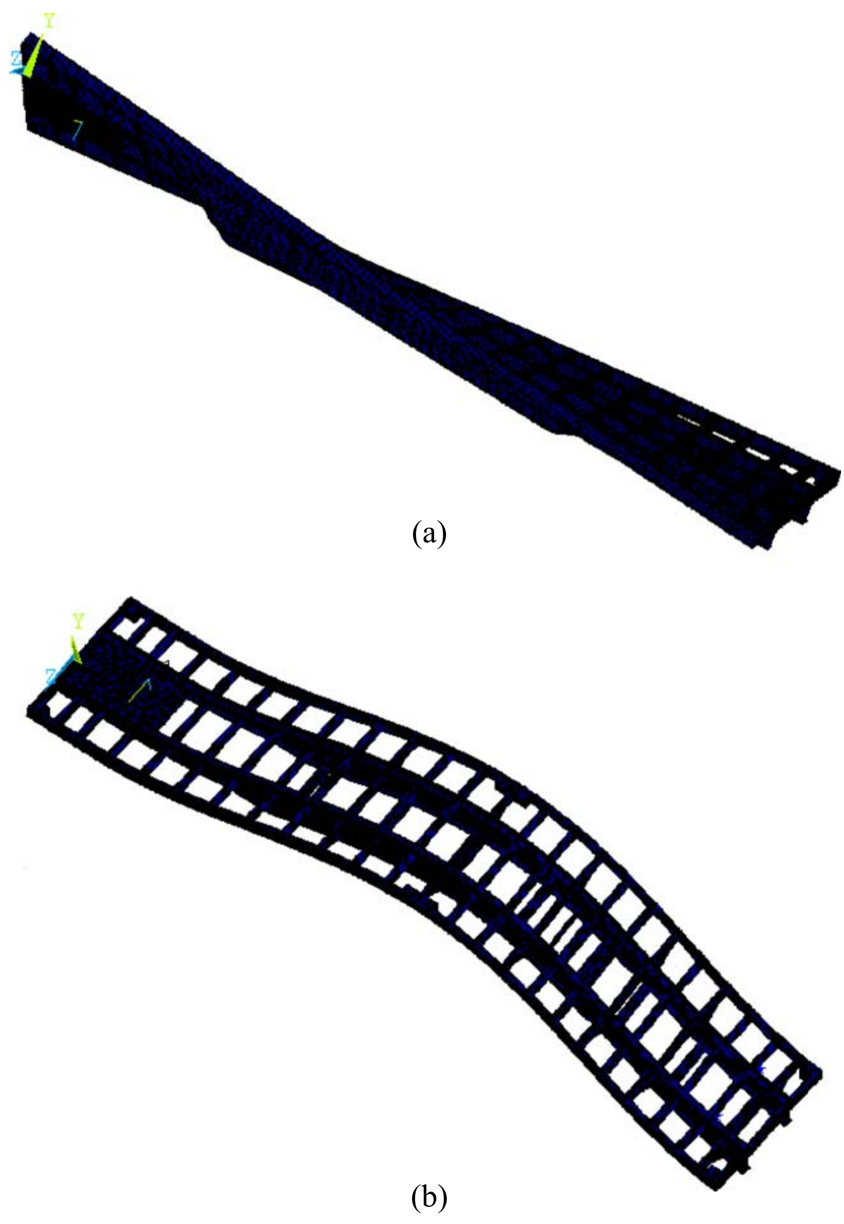


Fig. 4 Modal analysis results of the original model: **a** the first torsional mode at 3.78 Hz and **b** the second bending mode at 10.74 Hz

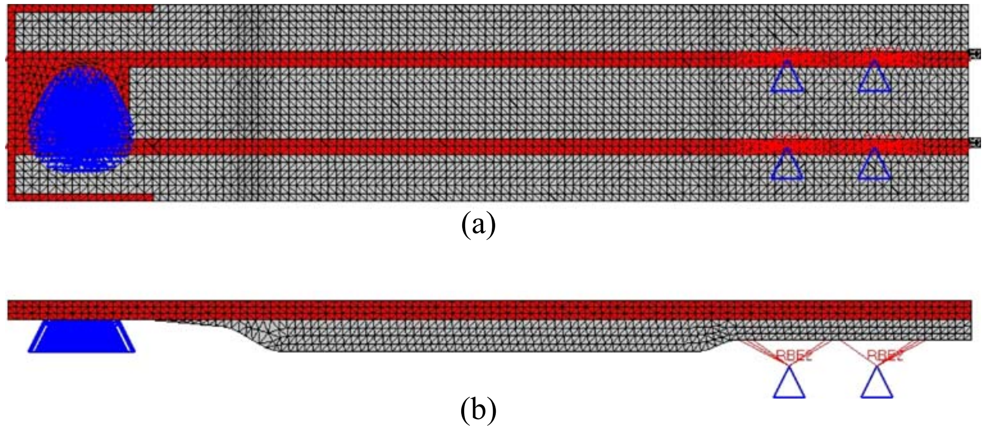


Fig. 5 Design domain of a trailer: **a** top view and **b** side view (*dark elements*: non-design domain, *bright elements*: design domain)

Figure 3 shows the static analysis results of the original flatbed trailer under uniform bending pressure along the master beams. The connection part to the tractor and the wheel axles are considered fixed in the analysis. The bending stiffness of the trailer is maximized such that the mean compliance (or twice the strain energy) of the trailer in the results of Fig. 3 is to be minimized during the optimization process. For the original model, the mean compliance by the bending load is calculated as 2.468e6 Nm, and the corresponding maximum deflection of the master beams is 12.69 mm. The maximum stress appears near the connection region with a tractor where the von Mises stress is 1,419 MPa.

In order to increase the torsional stiffness of a trailer, the natural frequency of the torsional mode should be increased. Figure 4 shows the modal analysis results for the original trailer model. The boundary condition for the modal analysis is the same as that for the bending problem in Fig. 2. While the shape and thickness distributions of the master beams have a major effect

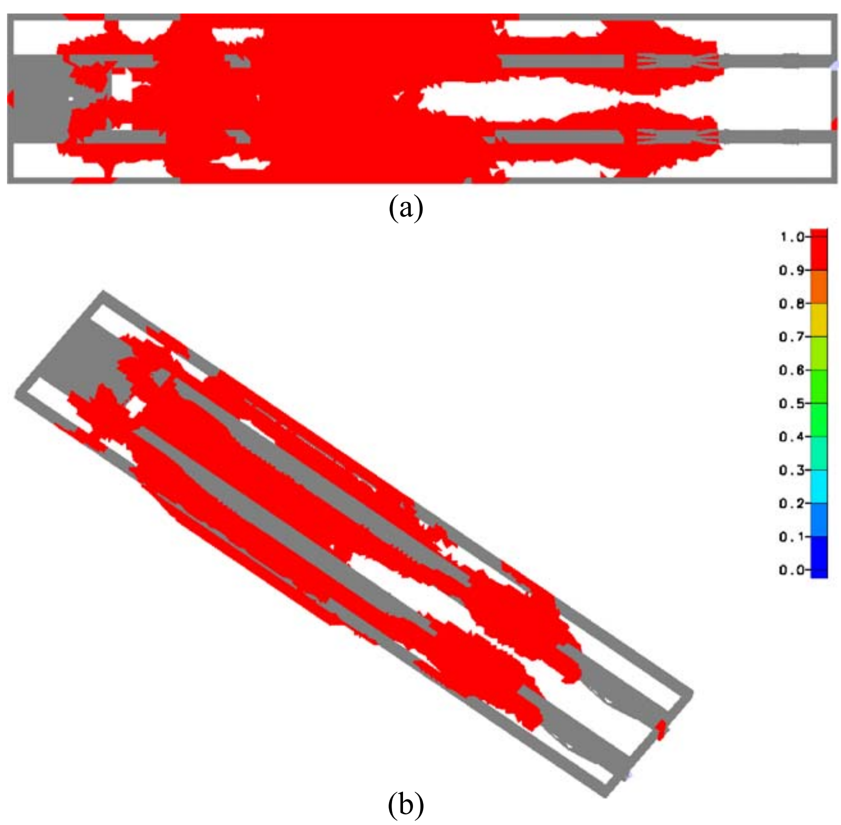


Fig. 6 Optimized topologies of a trailer by considering only the bending stiffness ($w_1 = 1$ and $w_2 = 0$ and in (2a)): **a** top view and **b** isotropic view

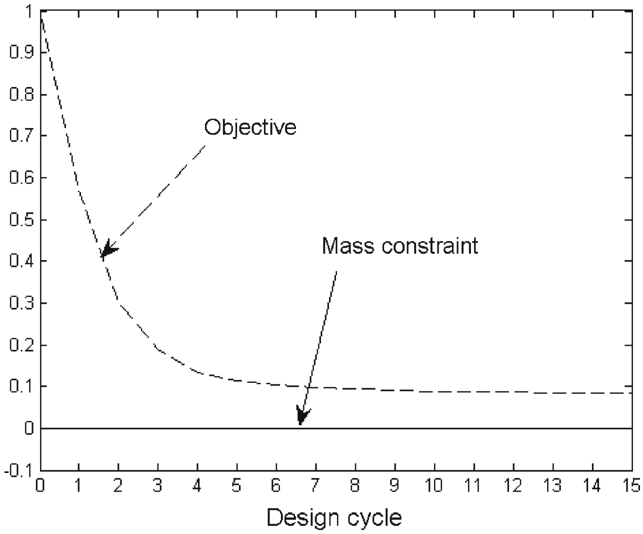


Fig. 7 Optimization history of the objective and the mass constraint for the result in Fig. 6

on bending stiffness, the layout as well as the shapes and thicknesses of the cross beams are very important factors for the torsional frequency.

2.2 Problem formulation and results

Figure 5 illustrates the design domain of the trailer for topology optimization. The design domain is large enough to completely embrace the original model. For finite element discretization of the design domain, 68,067 tetrahedron solid elements are employed. The dark elements in the model are not included in the design domain, so these elements do not change during the optimization. The master beams and the connection part to the tractor are treated as non-design members. The top flanges of the master beams and the upper plate of the connection part are denoted as dark elements in Fig. 5a. Also, in Fig. 5b, members of the rectangular frame wrapping around the trailer are set as non-design members.

The stiffness and densities of the elements in the design domain are parameterized with design variables and will be updated during optimization while minimizing the objective. In GENESIS, the parameterization is based on SIMP (Solid Isotropic Material with Penalization) (Vanderplaats R&D 2005):

$$E_i = E_{\min} + (E_0 - E_{\min})\, x_i^p, \tag{1a}$$

$$\rho_i = \rho_0 x_i, \tag{1b}$$

$$x_{\min} \le x_i \le 1, \tag{1c}$$

where E_i and ρ_i are the Young's modulus and the density of the i-th element in the design domain, respectively, and E_0 and ρ_0 denote those of the solid

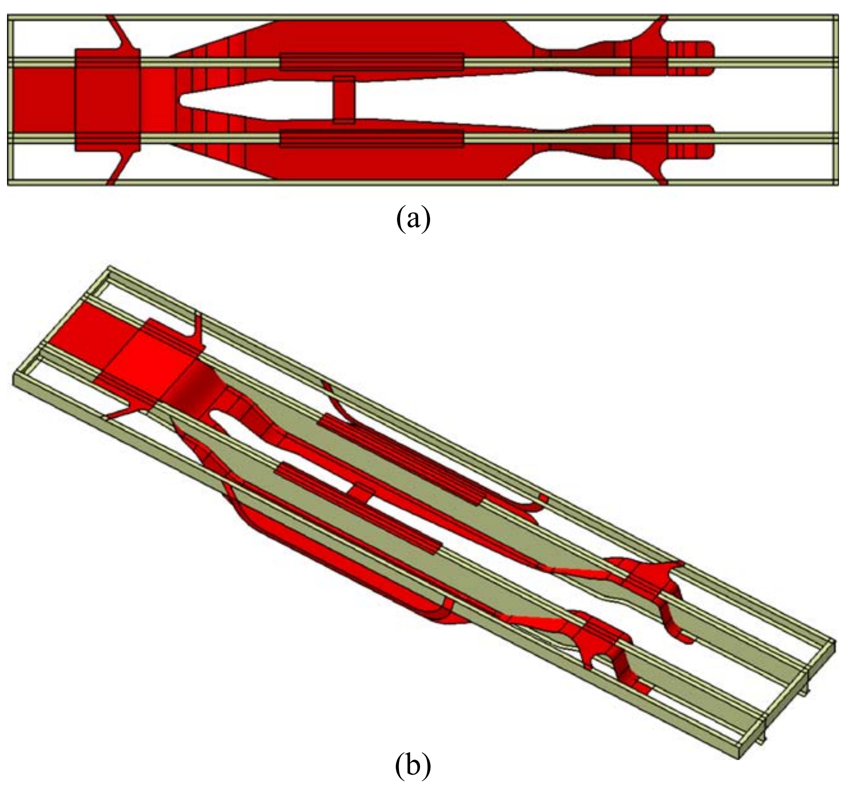


Fig. 8 A post-processed trailer from the result in Fig. 6: **a** top view and **b** isotropic view

Table 1 A comparison of the performances of the original model and the post-processed model after topology optimization (bending stiffness maximization problem with $w_1 = 1$ and $w_2 = 0$ in (2a))

	Mass [kg]	Mean compliance [Nmm]	Torsional frequency [Hz]
Original model	2,674	2.468e6	3.70
Optimized model	2,596	1.620e6	4.73

material, i.e., structural steel in this problem. In (1a) and (1b), x_i is the design variable linked to the i-th element, which varies continuously between $x_{\min}$ and 1. If $x_i = 1$, its corresponding element is regarded as solid, whereas if $x_i = x_{\min}$, the element gives little structural influence and can be regarded as void. By using the penalization parameter p in (1a) as larger than 1, x_i can converge close to either $x_{\min}$ or 1 (Bendsøe and Sigmund 2003). In this work, $p = 5$ is used. At the end of the optimization, the optimized structure of the trailer can be obtained by considering elements with high design variables. To prevent numerical singularity of a system equation, GENESIS uses $E_{\min} = 10^{-6} E_0$ as a default.

The formulation of the present topology optimization of the trailer is given as

$$\underset{\mathbf{x}}{Minimize}\ F_T(\mathbf{x}) = w_1 \left(\frac{U}{U_0}\right) + w_2 \left(\frac{f_0^t}{f^t}\right) \tag{2a}$$

$$subject\ to\ G_T(\mathbf{x}) = \sum_{i=1}^{N_e} \int_{\Omega_i} \rho_0 x_i d\Omega - M_0 \leq 0, \tag{2b}$$

$$\mathbf{x} = \{x_i\}^T \quad (i = 1, 2, \ldots, N_e)\,. \tag{2c}$$

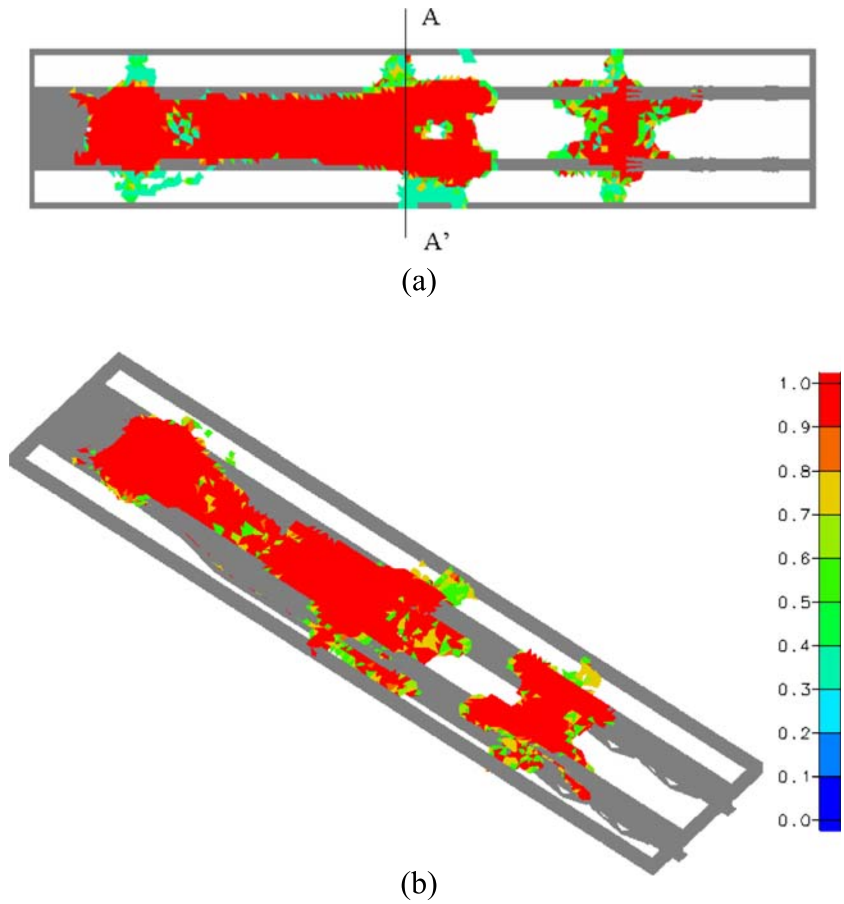
In (2a), the objective $F_T(\mathbf{x})$ is set as a multi-objective that combines the total mean compliance U and the torsional natural frequency f^t of the trailer. To calculate the mean compliance, the load case with uniform bending pressure of 0.241 kg/mm^2 on the top flanges of the master beams (total load 90 tons) is solved. In (2a), note that because maximizing the torsional frequency is the goal of optimization, the inverse form of the torsional frequency is used for the minimization problem. In order to fairly appreciate the contributions by the mean compliance and torsional frequency, they are normalized by their initial values, U_0 and f_0^t. In (2b), the total mass of the trailer is constrained below M_0, which is set as the total mass of the original model in Fig. 2. Thus, the resulting structure of topology optimization will have maximum bending stiffness and torsional frequency with the same amount of mass as in the original model.

Figure 6 shows the optimized topology when only the bending stiffness is considered for the objective in (2a), i.e., $w_1 = 1$ and $w_2 = 0$. In the figure, the elements whose design variables are less than 0.3 are not illustrated. Because the master beams are the main load-carrying members for the bending load case, the elements around the flanges of the master beams are optimized to have high stiffness to increase the bending moment of inertia of the master beams. As can be noted in Fig. 2a, due to the fixed boundary condition on the connection part with the tractor and around the wheel axles, high-bending curvatures arise, and thus high-strain energy densities. Accordingly, the optimized result in Fig. 6 has high design variables around the connection part with the tractor and around the connection part with the wheel axles to reinforce the structural stiffness. On the other hand, the connection members between two master beams such as the cross beams of the original trailer are regarded as inefficient for the bending stiffness and do not appear in the result. Figure 7 shows the iteration history of the objective and the constraint. A total of 15 iterations are required until convergence with a computation time of 5,340 s on a PC platform having a dual-core 64-bit $\times 86$ processor with 3 GHz and 4-gigabyte memory. Typical numerical problems of topology optimization such as checkerboard patterns are automatically suppressed by GENESIS.

Figure 8 shows the post-processed trailer for which the plate members are approximated from the elements with high design variables. As discussed in the introduction, because the topology optimization model in Fig. 5 is discretized with solid elements, it is not easy to obtain results consisting of plate-like thin-walled members. Thus, while a result from topology optimization is post-processed to the trailer with the plate members through thickness approximation, it is possible that the performances of the post-processed structure can deteriorate. Fortunately, the loss of performance after the post-process is not severe; Table 1 shows that the performance of the post-processed trailer has a 34.4% decrease in the mean compliance and 2.9% decrease in the mass compared to the original model. Torsional frequency is 4.73 Hz. The performances of the post-processed trailer shown in Table 1 are calculated by rediscretizing the structure with shell elements.

If a very large number of elements were used for discretizing the design domain, the result could have more

Fig. 9 Optimized topologies of a trailer by considering both the bending stiffness and torsional frequency ($w_1 = w_2 = 1$ in (2a)): **a** top view and **b** isotropic view



thin members and accordingly reduce the difference between the performances of the result of topology optimization and those after thickness approximation. However, the computation time until the convergence of the optimization will increase tremendously considering that not only the number of elements but also the number of design variables increase. Thus, instead of using a large number of elements for the design domain, it is desirable to further optimize the thicknesses of the post-processed trailer.

If we use $w_1 = 1$ and $w_2 = 1$ in (2a), both the bending stiffness and the torsional frequency are considered for the design objective. The result is shown in Fig. 9. As in the result of the bending stiffness maximization in Fig. 6, the bending stiffness is increased by mainly reinforcing the elements around the top and bottom flanges of the master beams. In addition, the elements around the connection part with the tractor and around

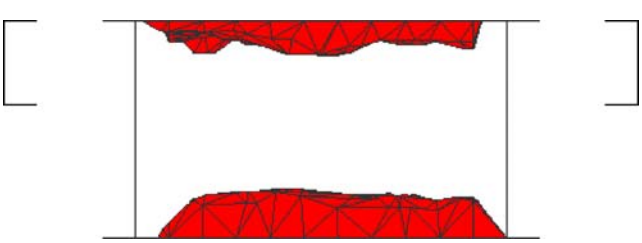


Fig. 10 Cross-sectional view of the result in Fig. 9 (A–A' view)

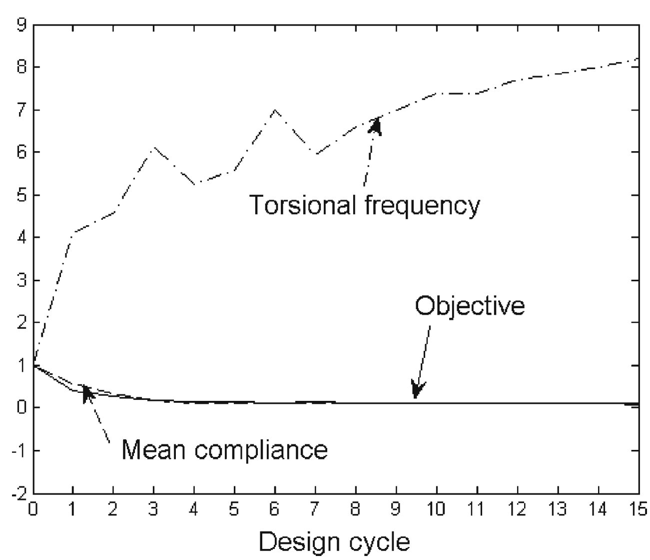


Fig. 11 Optimization history of the objective

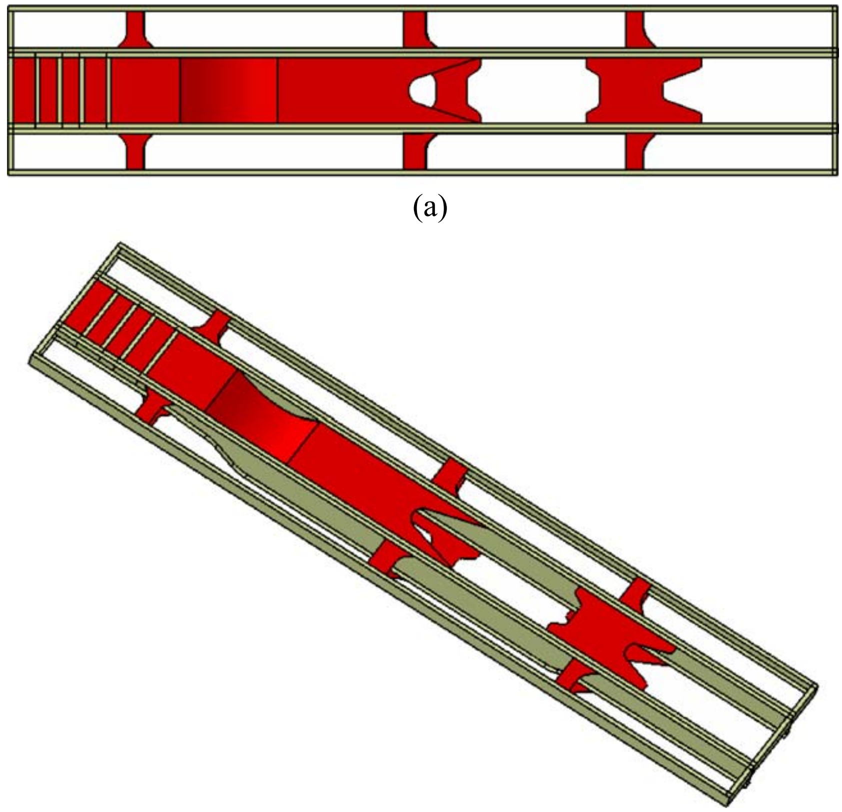


Fig. 12 A post-processed trailer from the result in Fig. 9: **a** top view and **b** isotropic view

the connection part with the wheel axles where high-bending curvatures arise have large design variables. In comparison with the bending-stiffness-maximized result in Fig. 6, the difference is that connection elements exist between the master beams. In Fig. 9, large plate members cover the top and bottom flanges of the master beams from the front connection part with the tractor to the central region of the trailer. Figure 10 illustrates the view of the cross section AA' shown in Fig. 9. By distributing material around the outer region of the cross section, the torsional rigidity can be efficiently increased. This layout is also effective for increasing the bending rigidity. Thus, the overall shape of the optimized trailer can be regarded as a box beam. In GENESIS, a multi-objective other than (2a) can be employed such as the negative form for the torsional frequency (its result is almost the same as that by (2a)) Fig. 11 shows the iteration history of multi-objective optimization. The increase in the torsional frequency and monotonic decrease in the mean compliance as the optimization proceeds are shown in the figure. Although the torsional mode was observed as the first mode in the original model, while the optimization proceeds, the sequence of modes might change, and the optimizer can lose the target frequency. To deal with this numerical difficulty, the mode-tracking option should be enabled in GENESIS (Vanderplaats R&D 2005).

Figure 12 illustrates the post-processed trailer from the topology optimization result in Fig. 9. Table 2 lists the increase in the performances of the post-processed trailer; compared to the original model, the mean compliance decreases 33.3%, and the torsional frequency increases 188.4%. Thus, the box-beam-shaped thin-walled closed cross section layout is quite effective for increasing the torsional stiffness and bending stiffness. In Table 2, although the total mass of the trailer was set equal to that of the original model during topol-

Table 2 A comparison of the performances of the original model and the post-processed model after topology optimization (bending stiffness and torsional frequency maximization problem with $w_1 = 1$ and $w_2 = 1$ and in (2a))

	Mass [kg]	Mean compliance [Nmm]	Torsional frequency [Hz]
Original model	2,674	2.468e6	3.70
Optimized model	2,607	1.645e6	10.67

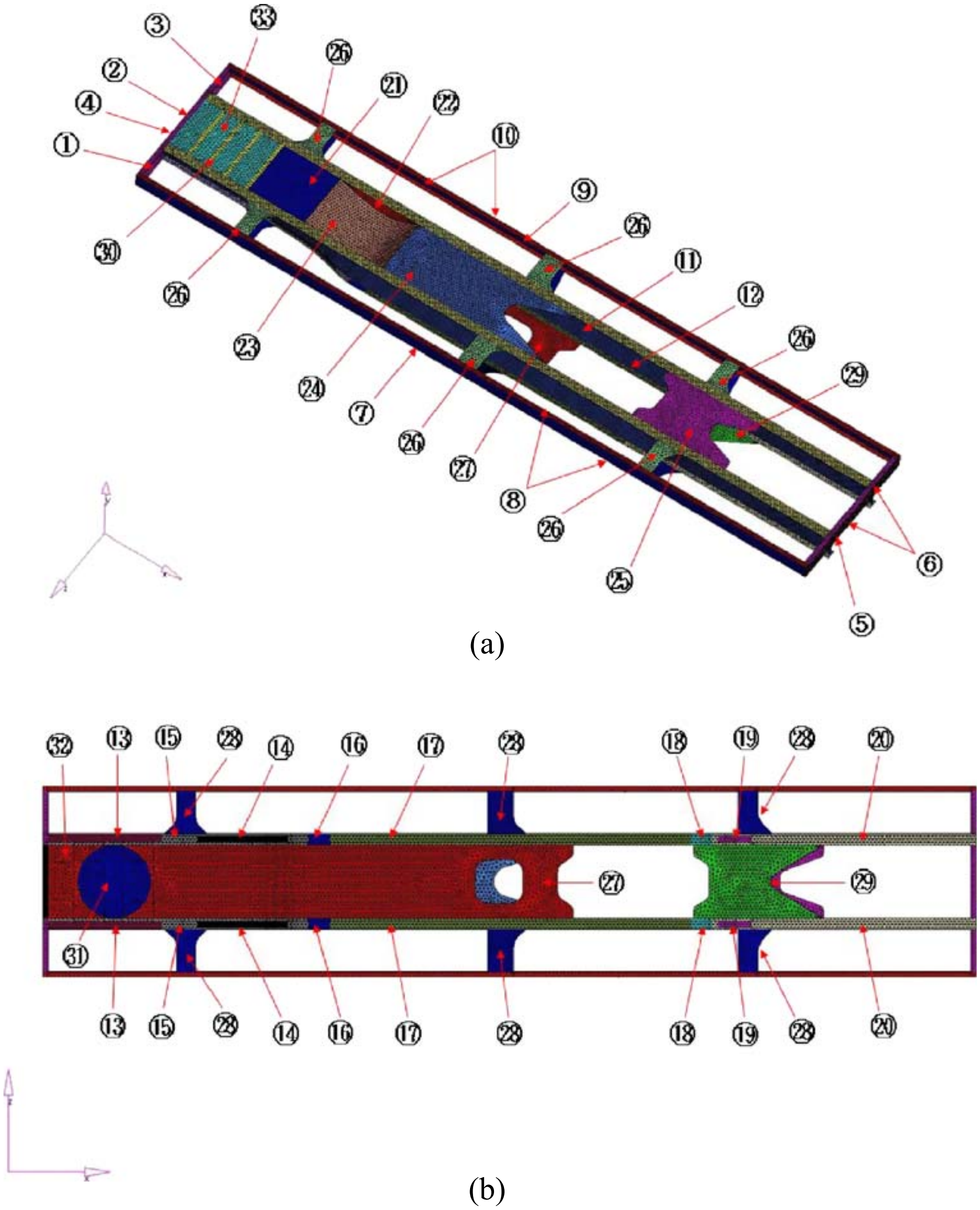


Fig. 13 Design variables for thickness optimization: **a** isotropic view and **b** bottom view

ogy optimization, the post-processed mass decreased slightly since the post-process was based purely on the engineering experience of the designer.

3 Thickness optimization

Figure 13 denotes design variables for thickness optimization. The post-processed trailer in Fig. 12 is reconstructed by using 33 plate members, each of which has its corresponding thickness design variable. As in the original model, the bottom flanges of the master beams are partitioned into many plates and are thus allowed to have different thicknesses along the length direction.

The formulation for the present thickness optimization problem is given as

$$\underset{\mathbf{t}}{Minimize}\; F_S(\mathbf{t}) = \sum_i \rho_0 A_i t_i \tag{3a}$$

$$subject\ to\; G_{S1}(\mathbf{t}) = \frac{U}{U_T} - 1 \leq 0, \tag{3b}$$

$$G_{S2}(\mathbf{t}) = \frac{f_T^t}{f^t} - 1 \leq 0, \tag{3c}$$

$$\mathbf{t} = \{t_i\}^T, (i = 1, 2, \ldots, 33), \tag{3d}$$

where t_i is the thickness of the i-th plate of the trailer, a design variable for optimization. In (3a), A_i denotes the area of the i-th plate, and thus the objective $F_S(\mathbf{t})$ is

Table 3 Optimized thicknesses of the plates of the trailer in Fig. 13 (t_i^0: initial thickness, t_i^{opt}: optimized thickness) (unit: [mm])

i	t_i^0	t_i^{opt}
1	4	3.1
2	4	3.1
3	4	3.1
4	4	3.1
5	4	3.1
6	4	3.1
7	4	3.1
8	4	3.1
9	4	3.1
10	4	3.2
11	5	5.7
12	14	10.0
13	14	5.9
14	14	14.0
15	14	12.0
16	14	14.0
17	14	8.9
18	14	6.0
19	14	14.0
20	14	4.8
21	6	3.1
22	4.5	3.1
23	4.5	3.1
24	4.5	3.1
25	4.5	3.1
26	4.5	3.1
27	9	3.2
28	4.5	3.1
29	4.5	3.1
30	6	3.1
31	9	3.1
32	4.5	9.8
33	6	6.0

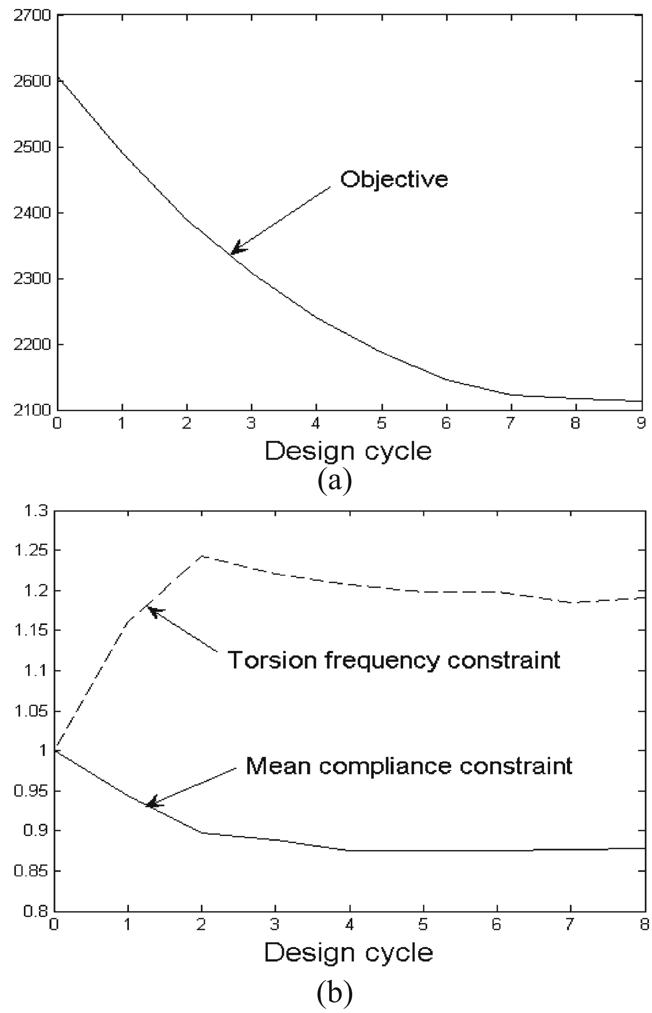


Fig. 14 Optimization history of the thickness optimization problem

the total mass of the trailer. In (3b) and (3c), U_T and f_T^t represent the mean compliance and torsional frequency of the trailer in Fig. 12, respectively, so $G_{S1}(\mathbf{t})$ and $G_{S2}(\mathbf{t})$ constrain the stiffness of the thickness-optimized trailer to be at least equal to those of the post-processed trailer from topology optimization in Fig. 12. Thus, the goal of thickness optimization is to reduce the mass of the trailer as much as possible without deteriorating the performances of the topology optimization result. Since the thickness approximation after the topology optimization is not carried out based on a rigorous analysis, there is a chance for thicknesses to vary a lot from their initial values during thickness optimization.

Because available plate thicknesses are mostly restricted as specific discrete values due to the manufacturing cost, t_i in (3d) is given as a discrete design variable having one of the following mass-produced plate thicknesses of commercial high-strength steel, ATOS 80 (POSCO 2008) (the discrete variable option in Optistruct (Altair 2007b) is enabled for thickness optimization.):

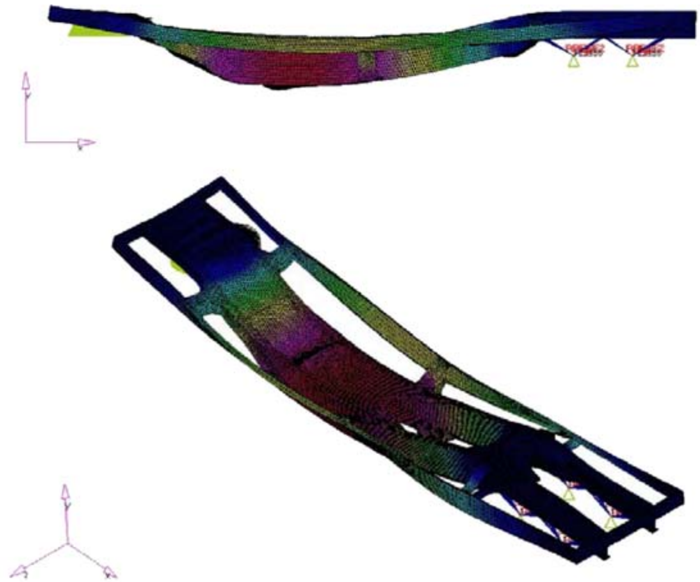
$$t_i \in \{3.1, 3.2, 3.6, 4, 4.3, 4.35, 4.4, 4.5, 4.8, 5, 5.7, 5.75, 5.8, 5.9, 6, 7, 7.7, 7.75, 7.8, 7.9, 8, 8.8, 8.9, 9, 9.8, 10, 12, 14\} \quad [\mathrm{mm}]. \tag{4}$$

The optimized thicknesses are listed in the third and sixth columns of Table 3, and optimization history is

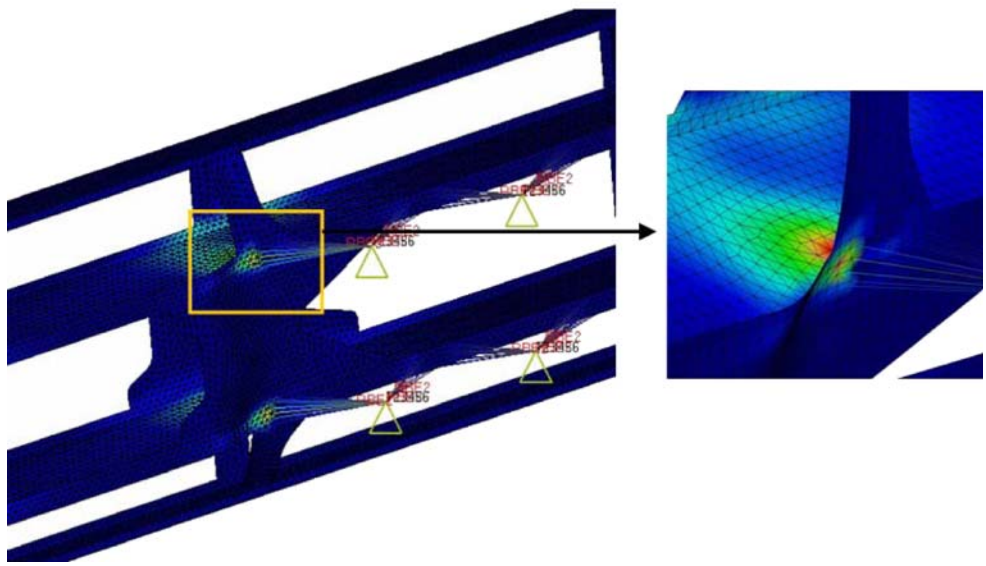
Table 4 A comparison of the performances of the original model and the thickness-optimized model

	Mass [kg]	Mean compliance [Nmm]	Torsional frequency [Hz]
Original model	2,674	2.468e6	3.70
Thickness-optimized model	1,902	1.946e6	9.94

Fig. 15 Analysis results of the final model of the flatbed trailer under uniform bending pressure along the master beams: **a** deformation and **b** the maximum stress point



(a)



(b)

shown in Fig. 14. The performances of the thickness-optimized trailer are listed in Table 4. In the table, compared to the original trailer model, the mass decreased 29% with 21% decrease in the bending mean compliance and 169% increase in the torsional frequency. Figure 15 illustrates the analysis results of the final model under the bending load condition where the maximum displacement of the master beams is 13.6 mm (7.2% increase from the original model). Although maximum stress constraints are not employed during optimization, the maximum von Mises stress of the final model is calculated as 350 MPa, 75.3% less than that of the original model. If this is not the case, an additional optimization procedure that changes the shapes or thicknesses of the local regions of the trailer can be used to lower the stress level.

4 Conclusions

A new trailer with increased bending stiffness and torsional frequency was designed by applying topology and thickness optimization. Through topology optimization, a creative layout with a closed cross-section was obtained as an optimum design to have maximized bending stiffness and torsional frequency. By applying discrete thickness optimization, the approximated plate thicknesses from the result of topology optimization were adjusted to minimize the total mass of the trailer without losing increased stiffness from the topology optimization. Compared to the original model, the final design of the optimized trailer has 21% decrease in the mean compliance by the bending pressure imposed along master beams, 169% increase in the torsional frequency, and 29% reduced mass.

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