技術論文

Stiffness optimization process using topology optimization techniques and lattice structures

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要旨

モーターサイクルに限らず、あらゆる製品において軽量化は大変重要な課題となっている。一般的に軽量化は製品の要求機能を考慮しつつ、材料や構造、形状などを最適なものにすることで達成され、形状については「トポロジー最適化」という技術の活用が盛んに行われている。一方、製造工法の観点では積層造形技術（3D プリント）の発展によって、ラティスと呼ばれる軽量構造の造形が近年可能になりつつある。本論文では両技術を組み合わせることで、より剛性の高い構造を設計することが可能であることを示した上で、最適化技術を活用し、剛性と軽量化を両立させた二輪車のエンジンカバー形状を検討した内容を紹介する。

ABSTRACT

In recent years, topology optimization technology has been widely applied as a means of achieving weight reduction. On the other hand, the development of additive manufacturing technology (3D printing) is making it possible to manufacture structures called lattices. In this paper, we show that it is possible to create even stiffer structures by combining both technologies, and we develop a design process that includes optimization calculations.

1 INTRODUCTION

Weight reduction is a particularly important issue not only for motorcycles, but for any product. In general, weight reduction is achieved by optimizing the materials, structure, shape, etc., while considering the required functions of the product, and in recent years, the technology of “topology optimization”[1] has been actively utilized for structure and shape.

Topology optimization is a method for calculating the optimal shape under given load boundary conditions and constraints. Without geometrical constraints, the solution is characterized by complex and organic shapes, and additive manufacturing is often chosen as a method to form such complex shapes. One of the features of additive manufacturing is the ability to create microstructures called lattice structures. Lattice structures are expected to have various functional benefits such as strength, stiffness, heat, and vibration, and are also effective in terms of weight reduction. For structure-related topology optimization, a mathematical method called the SIMP (Solid Isotropic Material with Penalization method)[2] is widely used. In this method, the material density of the elements is a design variable, of which most commercial software implementations employ.

In this report, the process, and the effect of integrating the above topology optimization and lattice structure design techniques are shown with specific case studies. At first, assuming the problem of a simple double-ended fixed beam subjected to an equally distributed load on its top surface, the stiffness of the beam under normal topology optimization is compared with the stiffness of the beam when the lattice structure is applied. Then, we applied some of the process to an engine cover to investigate a geometry that would provide both greater stiffness and weight reduction.

2 OPTIMIZATION OF A BEAM

To start things simply, let us assume the problem where we apply evenly distributed loads to a double-ended iron beam (Figure 1). Physical properties and load boundary conditions were as shown in Table 1.

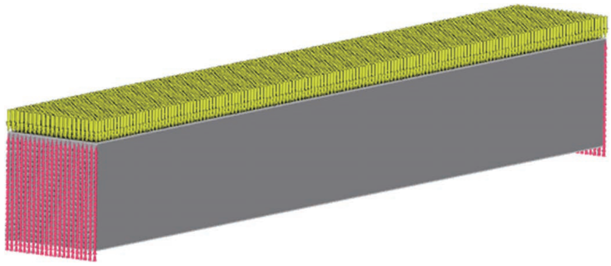

Fig. 1 Double-ended fixed beam

Table 1 Physical properties and load boundary conditions

Dimensions	10×10×80 mm
Young Modulus	205 GPa
Poisson Ratio	0.3
Boundary Condition	Both ends are completely fixed
Load Condition	Evenly distributed load on the upper surface 1600 N (2×10^6 Pa)

2-1. Topology optimization of a double-ended beam

First, the topology is optimized based on the SIMP method. The shape is output with the relative density threshold set to ρ = 0.5 and then smoothed using the smoothing function of the design tool. The resulting volume of the part is 3990 mm^3. The optimization-related conditions and results are shown in Figure 2 and Table 2.

Figure 3 shows a conceptual diagram of the shear force and bending moment of a beam under equally distributed loading with both ends fixed. The bending moment diagram of the beam has a parabolic shape, with larger values near the fixed ends and in the center of the beam. To minimize strain energy, the second moment of the cross section must be increased in areas of high bending moment, so members are placed to the edge of the design domain near the fixed end and in the central upper and lower face areas, while the center of the beam space can be thinned out. The shape shown in Figure 2 obtained by topology optimization and the second moment of area shown in Figure 4 generally reflect the above and are structurally appropriate from a mechanical point of view.

Next, we confirm the stiffness performance of the topology optimized shape using static analysis. We set up the simulation using the same loading and boundary conditions as in the topology optimization step. The maximum displacement in the loading direction is 2.204×10^{-2} mm. Figure 5 shows a displacement contour plot in the load direction. The stiffness performance of a functionally graded lattice structure will be examined and compared in the next section with respect to the above results.

Table 2 Optimization conditions and results

Objective	Minimize strain energy (Maximize stiffness)
Constraint	Volume ratio 50% or less
Extraction Density Threshold	0.5
Volume	3990 mm^3

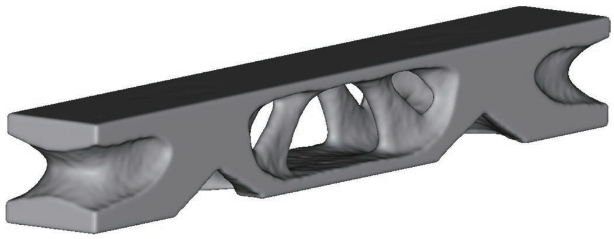

Fig. 2 Topology optimized beam

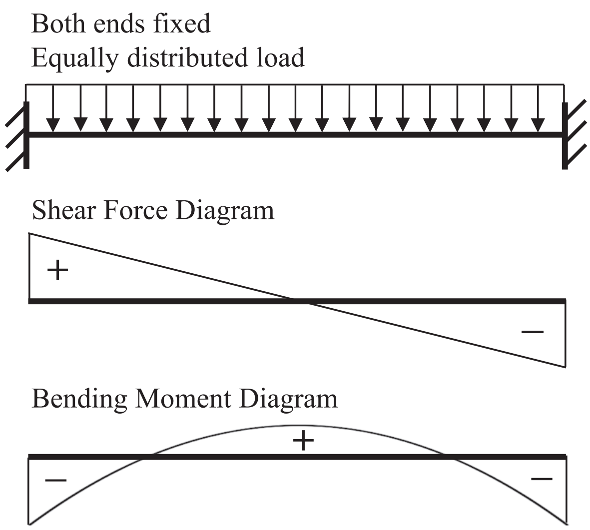

Fig. 3 Conceptual diagram of the beam

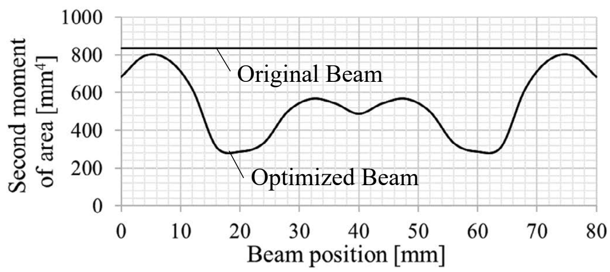

Fig. 4 Second moment of area around neutral axis

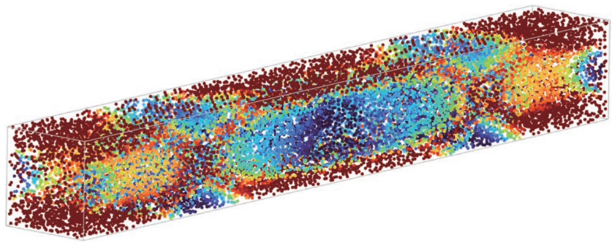

Fig. 6 Point cloud map of intermediate density

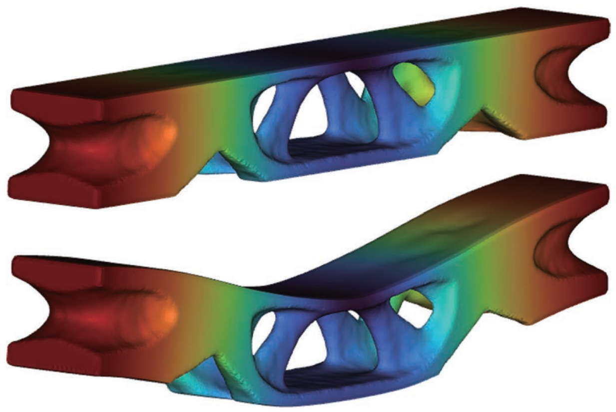

Fig. 5 Displacement contour plot of topology-optimized beam in load direction (deformation magnification factor 0 × and 300 ×)

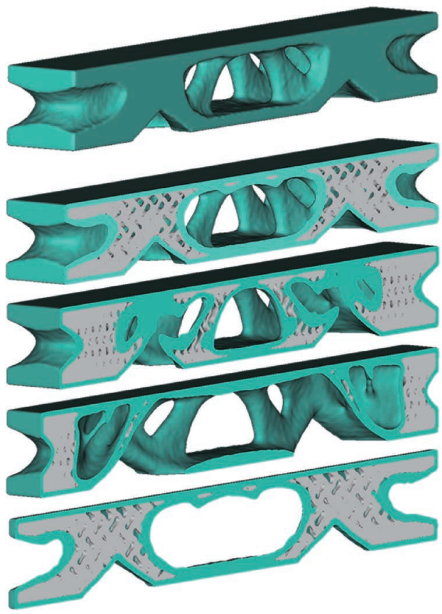

Fig. 7 Cross-sectional structure of shell and infill

2-2. Functionally graded lattice structure

In examining the lattice structure, we adopted a "shell and infill" approach. The structure consists of an outer shell and an inner lattice structure. This configuration is bio-inspired and resembles the bone structure to improve the stiffness.

An example structure is shown in the figure below. Figure 6 shows a point cloud map of intermediate density using the SIMP method, and Figure 7 shows a cross-sectional structure of shell and infill. Here, the topology optimization threshold of intermediate density was set to 0.4, and the outer shell thickness to t = 0.6 mm. We used a gyroid lattice with a cell size equal to 3 mm for the internal lattice structure. The intermediate density distribution of the topology optimization controls the thickness of the gyroids and is set to gradually change from 0.25 to 1.57 mm — thicker at sections with higher loading. The final volume was almost identical to the volume of the shape obtained from pure topology optimization.

Then we perform an analysis to confirm the part's stiffness performance. The maximum displacement in the load direction was 2.100×10^{-2} mm (Figure 8). If we compare the displacement of the functionally graded lattice structure with the shape obtained by the pure topology optimization example of the previous section (2.204×10^{-2} mm), we see that the stiffness is improved by about 5%.

Note that the external shape of the shell and infill structure shown in Figure 7 is not the same as the external shape of the beam in Figure 2, but is about 6% larger in volume and about 5% larger in average second moment of area (however, both shapes satisfy the constraint of 10 × 10 × 80 mm for the overall beam dimensions, which is the design domain). This is because the intermediate density threshold for extracting the external shape from the topology optimization results is intentionally set smaller.

The reason the external shape can be made larger keeping the same total mass is that the optimal intermediate density distribution can be faithfully generated in the actual shape by using a lattice structure for the interior. Due to the superior reproducibility of the intermediate density distribution obtained by optimization into the actual shape, the specific stiffness can be expected to be higher than that of a solid structure.

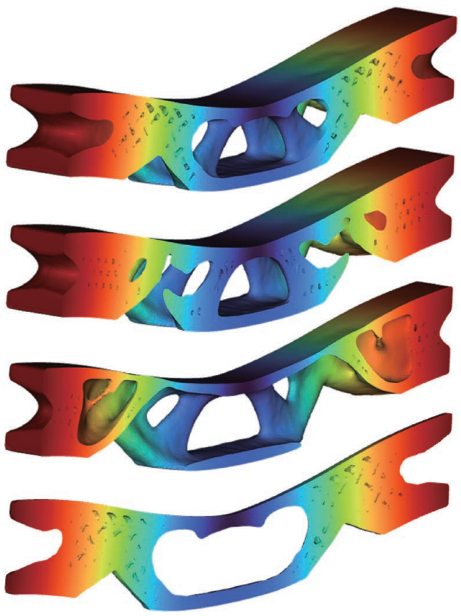

Fig. 8 Displacement contour plot of functionally graded lattice structure cross-section

3 CASE STUDY OF OPTIMIZATION PROCESS APPLICATION

The following is a case study of weight reduction of a motorcycle engine cover using some of the optimization processes described above. The function of the cover is to minimize damage to the engine itself when the motorcycle falls over. Conventionally, it is difficult to reduce the weight of these parts as they tend to be heavy. In this case study, weight reduction was the first priority, and we examined how much weight could be reduced while allowing for a reduction in rigidity of up to 5%. The original cover shape is shown in Figure 9. In order to maximize the benefits of the shell and infill structure, the shape of the design area that did not interfere with other members was reexamined, and the optimal arrangement of members to maximize stiffness was then determined by topology optimization. The design domain shape for optimization is shown in Figure 10, and the density distribution obtained by the calculation is shown in Figure 11.

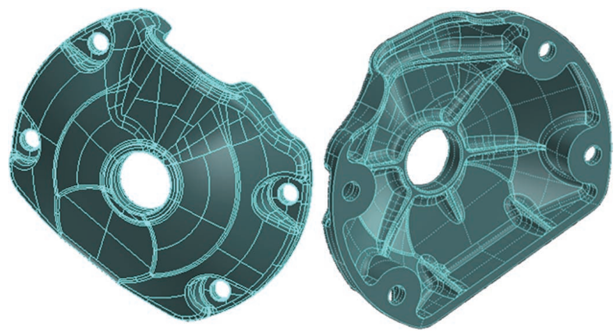

Fig. 9 Original engine cover shape

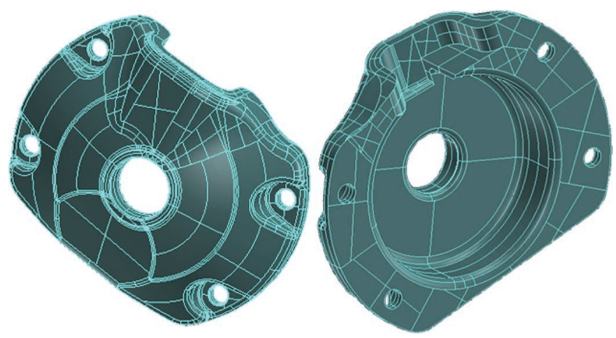

Fig. 10 Modified design domain for optimization

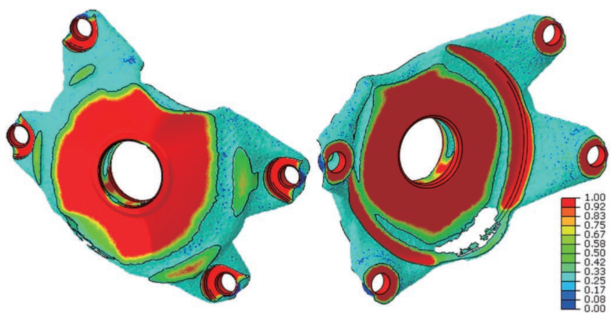

Fig. 11 Density distribution contours by topology optimization

Here, the static analysis conditions and optimization conditions are shown in Table 3. In this model, the inner surface of the bolt holes of the cover was completely fixed and the center of the cover was subjected to the forces that would be applied by the road surface during a fall(Figure 12).

Table 3 Static Analysis Conditions

Young Modulus	75 GPa
Poisson Ratio	0.33
Boundary Condition	Inner surfaces of the bolt holes are completely fixed
Load Condition	4880 N at the reference point outside the center of the cover
Constraint Condition	Coupling restraint between the reference point and part of the outer surface of the cover

Table 4 Optimization conditions

Objective	Minimize strain energy (Maximize stiffness)
Constraint	Volume ratio 70% or less
Extraction Density Threshold	0.3

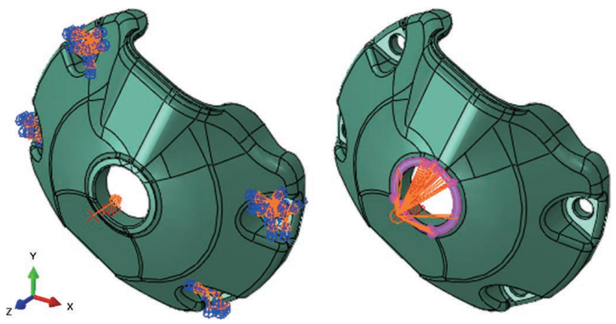

Fig. 12 Static Analysis Model

The next step is to divide the design domain into an outer shell domain and an infill domain. In this step, we modified the shape of the top of the cover, which is known empirically to have less frequent contact with the road surface. Although we intentionally left the lower cover shape, which is in frequent contact with the road surface, there was concern about weight increase if the infill area was also made to follow the shape of the lower cover. Therefore, for the infill area, we considered a shape with a density between 0.4 and 0.9 in the optimization calculation results as a guide, and for the outer shell, we created a shape with the minimum wall thickness that can be stably formed by additive manufacturing. Figure 13 shows the outer shell and infill areas.

Then, a beam-based lattice was placed in the infill area, and the spacing of the beams was adjusted according to the intermediate density distribution obtained from the topology optimization. The final cover shape appearance created is shown in Figure 14, and the lattice structure specifications are shown in Table 5. The load point displacements and volume of the original and final cover shapes are also shown in Table 6. The lattice pattern selected is called Voronoi[3], which is a method of space division in which the beams are randomly oriented by randomly placing the control points (site points). Since isotropic mechanical properties are expected, Voronoi is easy to handle when multiple loading directions are assumed.

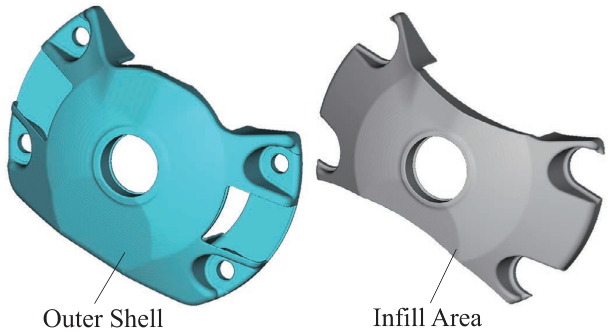

Fig. 13 Outer shell and infill shape

Fig. 14 Final shape including lattice structure

Table 5 Specifications of the lattice structure

Outer shell thickness	0.8 mm
Lattice Pattern	Voronoi
Lattice Cell Size	3.25～5.35 mm (Graded)
Strut Diameter	1.2 mm

Table 6 Weight and load point displacement

Specification	Load Point Displacement [mm]	Volume [mm^3]
Original Shape	0.147	29938
Final Shape	0.153	24977

Based on the displacement of the load point shown in Table 6, 16% weight reduction was achieved despite 4% reduction in stiffness. In terms of specific stiffness, it was 15% better than the original geometry.

As mentioned above, it was necessary to consider requirements other than stiffness with respect to the external shape (leaving areas of high contact frequency), so performance comparisons with simple topology optimization results were not performed. However, if the external shape is the same, it is expected that incorporating a lattice structure will provide higher specific stiffness, as in the beam example.

In this case study, the stiffness maximization calculation was performed during the topology optimization phase, but it may be possible to realize a structure with high tensile stiffness and low bending stiffness, for example, by setting displacement constraints. In addition to stiffness, it may be possible to achieve the desired properties in various physical fields, such as thermo-fluid, vibration, and acoustics, by devising a macroscopic structure in addition to the properties of the material itself. This concept is also called metamaterials, and further technological development is expected in the future.

SUMMARY

In this paper, we first show that the topology optimization results for a simple double-ended fixed beam are appropriate from the structural mechanics point of view. Then, by adopting a shell-and-infill structure and reflecting the optimization results (intermediate density distribution) in the lattice of the infill region, it is shown that the stiffness can be increased compared to a simple topology optimization. In addition, it was found that the shell and infill regions need to be set appropriately based on the optimization results, since the application of this method to a specific part may involve multiple requirements other than stiffness. In the proposed shell and infill structure, various parameters are assumed, including outer shell thickness, lattice pattern, lattice beam thickness, plate thickness, cell size, and its inclination range. The selection and optimization of these parameters, as well as the search for the optimal structure design process for other physical domains, is the next challenge.

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