Design and Simulation of a Porous Rectangular Board-Foldable Trolley

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Abstract. To reduce the structural weight and volume and satisfy an optimal balance of performance constraints, a study using a porous rectangular foldable board and structurally improved trolley model was conducted, and motion simulations were performed to derive the static and transient structural properties. The simulations revealed that the fatigue lifetimes of both the main structure and the porous boards were in a high-standard-value range, whereas the transient analysis indicated that both the directional deformation and equivalent stress exhibited minimal local values. For porous boards simply supported only on two sides, the trolley had a much smaller collapsed volume than other types of trolley. The static analysis indicated that this trolley prototype had a good safety coefficient and long fatigue lifetime. The transient analysis showed that the porous area had a higher strength and bending load than the non-porous area. The overall transient response was close to that of a steady-state, showing a negligible effect on the structure.

1. Introduction
Lightweight design is a major requirement in the development of many products. Foldable trolleys, for example, have been widely used at various sites by a range of users. Their main structures are primarily made of steel and aluminium, whereas the load-boards are usually non-porous and made of polyethylene or metals. Generally, after the trolley is folded, the volume range of the width of the board still occupies the storage space. Therefore, to satisfactorily optimise designs, in terms of the weight and performance constraints, the volume and lightness of the product’s minimum compliance system are gradually becoming a concern [1]. When a laminated composite plate was subjected to a dynamic load-induced over vibration, high translocation, and acceleration, its structural properties were severely degraded [2]. Therefore, a simulation was conducted of the static, modal, and transient structures, based on the finite-element method (FEM), to determine the loading status. With some improvements to the trolley structure, a side-suspended trolley, suspended on two girders, was developed. Based on the FEM, it was found that the improved trolley satisfied the strength and rigidity standards [3]. Finite element analysis (FEA) was used to determine the static and dynamic response of the components under different load conditions [4]. According to the structure and characteristics of the material, the stress and deformation were analyzed and the maximum deformation was obtained [5]. Through the modeling and simulation of a bicycle frame, it is known from the maximum stress and displacement that, after the structural optimization, the rigidity is improved and the total deformation is reduced [6]. Motion simulation was used to study the structural properties of a trolley by using the software ANSYS to investigate the stress and fatigue of a trolley subjected to a given loading. Application-related parameters, such as the total deformation, stress ratio, and strain, were obtained to analyze the fatigue lifetime, which was shown by the simulations to be far longer than standard, thereby indicating an optimal design [7]. Significant positive correlations can, generally, be observed between material and structure. To reduce the weight of a structure, designers usually fabricate holes at specific sites on the plates in the structure, as long as the structural support capability is not affected. Under a strength constraint on the plates, most of the plates that were produced had a high porosity, specific surface area, and an open cell rate that allowed the products to exhibit the best performance [8]. The perforated board withstood changes in load and support conditions, greatly reducing local buckling and strengthening the fatigue resistance of the material [9]. Supported by a
lower plate with a porous surface, the pressure distribution and load carrying capacity were also increased when the surface roughness of the upper plate was increased [10].

Based on the consideration of a lighter body structure, and with sufficient load-carrying capacity, changing the solid gear to a lattice structure observed not only a reduction in the weight but also reduced the vibrations significantly [11]. A loading analysis of a honeycomb-like, bidirectionally hollow thin plate using ANSYS was performed, and it was found that such a plate had a higher loading capability and better plastic deformation characteristics than a solid plate [12]. The structural optimisations that were performed on this basis significantly increased the stiffness of the bicycle frame, thus reducing its deformations. The buckling response of square composite plates with elliptical holes in the centre was studied, and it was found that large elliptical holes generated a pressure under the thinnest plate, indicating that, to prevent buckling that may be generated under a relatively low stress, large elliptical holes should be avoided on composite laminated plates [13].

This study is based on a reported improvement of the trolley structure, in which the trolleys are foldable on two sides to reduce the collapsed volume [14–18]. Briefly, porous foldable boards were built, and a structurally improved trolley was designed. Then, an FEM-based engineering simulation approach was employed to simulate the static structural properties and transient structural motions of the trolley.

2. Materials and methods

In this study, a static structure simulation was performed to analyze the structural performance and fatigue strength of the entire trolley structure. In order to understand the reaction of the load (object) when it is applied to the structure instantaneously, a transient structure analysis was conducted to analyze the bending behaviour of the load boards.

2.1. Development of the geometric model

Numerous porous foldable boards and a structurally-improved trolley prototype, in which the fixed framework structure was improved to an adjustable one to reduce the collapsed volume of the trolley, were built. The folding operation is as shown in Figure 1.

![Figure 1. Trolley folding operation.](image)

Three rectangular load boards were designed to have multiple symmetrically-placed holes, each with a diameter of 30 mm. Each board was 450 (length) × 180 (width) × 11 (height) mm when folded, to decrease the weight and reduce the space occupied. When expanded, the trolley had dimensions of 791.49 (length) × 455.2 (width) × 802.64 (height) mm, whereas the collapsed trolley had dimensions of 700 (length) × 180 (width) × 280 (height) mm.

Most of the components are connected by a set of screws, respectively, included the board B is also connected by two groups of hinges between the boards of A and C, as shown in Figure 2. After modelling, the computer aided engineering (CAE) simulation can be simplified as needed.
2.2. Setting simulation conditions

For the framework and load boards of the foldable trolley, the main material was an aluminum alloy; a type of material widely used in engineered structures due to its high strength and low weight. We used commercially-available metal parts for several of the hardware fittings of the trolley. Knowledge about the material coefficients of the metal parts was beyond the scope of this study and, therefore, we simply used the aluminum alloy coefficients to represent them. The relevant parameters of geometry and materials for the trolley are listed in Table 1. The element Control was program-controlled, the solver target used Mechanical APDL (ANSYS Parametric Design Language).

Table 1. Model analysis.

| Item         | Model Geometry |
|--------------|----------------|
| Bounding Box |                |
| Length X     | 791.49 mm      |
| Length Y     | 802.64 mm      |
| Length Z     | 455.20 mm      |
| Properties   |                |
| Volume       | 4,136,700 mm³  |
| Mass         | 11.459 kg      |
| Scale Factor | 1              |
| Statistics   |                |
| Nodes        | 3,383,425      |
| Elements     | 2,040,337      |
| Mesh Metric  | Element Quality|

When performing a mesh calculation, we set the following parameters: solid, physical preference was set to be mechanical, relevance was 100, size function was adaptive, relevance center was chosen to be coarse, mesh transition was selected to be fast, span angle center was coarse, smoothing was set as medium, mesh metric was set as element quality, nodes were 3,383,425 and elements were 2,040,337, to ensure accurate analysis and simulation (see Figure 3). As shown in Figure 3, the closer the bar of the statistical mesh number is to the right, indicates better quality.
The constraints for the static analysis were as follows: For the load boards, four board lockers (D in Figure 4) and several hinge screws (J in Figure 4) were used as a fixed support to anchor the three foldable boards (A, B, and C in Figure 4) and the two groups of hinges (E, F, G, and H in Figure 4) that connected two adjacent foldable boards. This was aimed at placing a constraint on the freedom of Ux, Uy, and Uz. The load was set to be 800 N in magnitude and was exerted in a downward direction. Four wheels were used as displacement support and for constraining the freedom of Uy.

The constraints for the transient analysis were approximately the same as those for the static analysis, except the displacement support, which was moved to the two sides (D, E, and G-J in Figure 5) of the three load boards. Due to the four board lockers that are connected and fixed between the load board and the fixed support, the freedom of Ux and Uy was constrained. There was no support in the Uz direction and, so, Uz was free.
3. Results

After setting the parameters for the materials, meshes, loads, and constraints, we found that, when an 800 N force was exerted on the static structure, the structure underwent a deformation and the force and load board formed a mutual vertical stress. The static and fatigue simulation results are shown in Table 2.

Table 2. Structural statics and fatigue analysis.

| Object                  | Force: 800 N | Occurs on                          |
|-------------------------|--------------|------------------------------------|
| Total Deformation       | Min.         | Max.                              |
|                         | 0 mm         | 0.12581 mm                         |
|                         |              | Board A (Max)                      |
| Equivalent Elastic Strain| 0 mm/mm      | 0.0015746 mm/mm                  |
|                         |              | Screw / Board A (Max)             |
| Equivalent Stress       | 0 MPa        | 110.96 MPa                         |
|                         |              | Screw / Board A (Max)             |
| Safety Factor           | 2.5235       | 15                                 |
|                         |              | Screw / Board A (Min)             |
| Life                    | 4,878,500 cycles | 1e+32 cycles                     |
|                         |              | Screw / Board A (Min)             |

3.1. Static analysis.

In this study, an 800 N simulation force was exerted on the boards and hinges of the entire trolley structure (with a total weight of 11.459 kg). After the total deformation analysis, we found that the bending part showed a maximum stress in structural rigidity displacement of 0.12581 mm. It can be seen, in Figure 6, that the maximum displacement occurred at the outer free end of load board A, whereas the main loading regions on the boards did not show any obvious deformation.
When the 800 N force was exerted on the load boards, the maximum equivalent elastic strain was 0.0015746 mm/mm, and was located in a small region around the head edge of a screw on the hinge connecting the load board A, as shown in Figure 7.

For the purpose of avoiding plastic deformation, which may cause permanent damage to the mechanical materials, the stress should be lower than the yield strength or tensile strength of the materials (i.e., 280 MPa), and a stress below this value does not generate plastic deformation or destructive deformation. The analysis in this study indicated that when the load boards were subjected to the external force, the maximum equivalent stress per unit area inside the boards was 110.96 MPa and was confined to a small region around the head edge of a screw on the hinge connecting load board A, as shown in Figure 8.

3.2. **Fatigue analysis.**

A fatigue test was conducted for tension-tension and tension-compression load with a stress ratio, R, of -1, -0.5, 0, and 0.5, and up to the endurance limit of 108 with frequency (Table 3).
The safety coefficient of a regular-livelihood product is defined as the ratio of the yield stress to the allowable stress, with a coefficient ≥1.5 indicating the safety of the product. Aluminum is a ductile material. This study found that the minimal safety coefficient of 2.5235 corresponded to a small region around the head edge of a screw on the hinge connecting load board A (Figure 9), whereas the safety coefficients of other parts on the trolley were all greater than that value, and hence were in a safety range.

![Safety factor](image)

**Figure 9. Safety factor.**

Based on the mesh distribution and stress intensity factors, we analyzed and predicted the fatigue lifetime, and determined that the minimum value of 4,878,500 cycles was for a small region around the head edge of a screw on the hinge connecting load board A. In comparison, both the structural framework and load boards had a fatigue lifetime greater than 1e+32 cycles, with the fatigue lifetime of the screws lying in between (Figure 10). These results indicated that repeated loading would not compromise the fatigue lifetimes of the structural framework and load boards.

### Table 3. Alternating stress R-Ratio.

| Alternating Stress MPa | Cycles   | R-Ratio | Alternating Stress MPa | Cycles   | R-Ratio |
|-----------------------|----------|---------|------------------------|----------|---------|
| 275.8                 | 1700     | -1      | 77.57                  | 50,000,000 | -0.5    |
| 241.3                 | 5000     | -1      | 72.39                  | 100,000,000 | -0.5    |
| 206.8                 | 34,000   | -1      | 144.8                  | 50,000    | 0       |
| 172.4                 | 140,000  | -1      | 120.7                  | 190,000   | 0       |
| 137.9                 | 800,000  | -1      | 103.4                  | 1,300,000 | 0       |
| 117.2                 | 2,400,000| -1      | 93.08                  | 4,400,000 | 0       |
| 89.63                 | 55,000,000| -1  | 86.18                  | 12,000,000 | 0       |
| 82.74                 | 100,000,000| -1 | 72.39                  | 100,000,000 | 0       |
| 170.6                 | 50,000   | -0.5    | 74.12                  | 300,000   | 0.5     |
| 139.6                 | 350,000  | -0.5    | 70.67                  | 1,500,000 | 0.5     |
| 108.6                 | 3,700,000| -0.5    | 66.36                  | 12,000,000 | 0.5     |
| 87.91                 | 14,000,000| -0.5 | 62.05                  | 100,000,000 | 0.5     |
Figure 10. Analysis of fatigue lifetime.

Figure 11 shows a plot of the available lifetime (in cycles) as the fatigue results change, as a function of loading at the critical location (rectangle area) on the trolley, with the loading history expressed as a ratio in the range of 0.5–1.5. With respect to the lifetime, the damage coefficient or safety coefficient varied significantly (with a trend of decline) when the loading history was close to a critical data range, where the sensitivity of the lifetime to the loading history could be estimated.

Figure 11. Fatigue sensitivity.

By using the material coefficients of alum–Si cast and extruded aluminum [19] as the foundation for fatigue data calculation and by setting the multiplication factor as 1, our calculation showed that the equivalent stress dropped from its maximum value of 110.96 MPa to the minimum value of 0 MPa within one cycle, as displayed by the stress–strain curve—a plot of local stress versus corrected strain response—in Figure 11. Our calculations also predicted a stable periodicity over repeated stress cycles (Figure 12).
3.3. Transient analysis.
To determine the characteristics of the transient response, we conducted an analysis of the vibration displacement of the porous board after the force is applied. The transient integration parameters used were: an amplitude decay factor for the second-order transient parameters, transient integration parameters (\(\Gamma=0.1000, \alpha=0.3025, \Delta=0.6000\)), Hilbert-Huang Transform (HHT) time integration (\(\alpha_f=0.1000, \alpha_m=0.000\)), and center of mass \((X, Y, Z) = (56.899, 35.492, -0.19929)\). The curve is shown in Figure 13.

The load boards were placed on the two sides of the supporting framework. With such a simple physical structure, the entire load was perpendicular to the load surface. To further understand the strength of the load boards and their bending load, we performed a transient analysis and found that, for directional deformation, the transient response attained a stable status at 0.38 s after the maximum value and at 0.43 s after the minimum value as the transient response evolved gradually from the initial to the stable status, as shown in Figure 14.

![Figure 12 Local stress and strain response.](image1)

![Figure 13. Digitally fitted curve.](image2)
Figure 14. Transient response of directional deformation from the transient structural analysis.

We conducted directional deformation analysis along the Z-axis direction in the porous and non-porous areas of the load boards, and found that the non-porous area of board A exhibited the maximum deformation of -0.08436 mm, followed by the porous area of board A (-0.05746 mm), the porous area of board C (-0.03534), the non-porous area of board C (-0.02146 mm), the porous area of board B (-0.00594 mm), and the non-porous area of board B (-0.00474 mm), as shown in Figure 15.

Figure 15. Directional deformation from the transient structural analysis.

In regards to the equivalent stress, the transient response reached a stable status 0.41 s after the maximum value and 0.45 s after the minimum value during the gradual evolution of the response from the initial to the stable status, as shown in Figure 16.
By analyzing the equivalent stress along the Z-axis direction, we found that the non-porous area of board A had a maximum equivalent stress of 4.826 MPa, followed by the porous area of board A (3.2205 MPa), the porous area of board C (2.9598 MPa), the non-porous area of board C (2.1358 MPa), the porous area of board B (1.7672 MPa), and the non-porous area of board B (0.7127 MPa), as shown in Figure 17. These values were much lower than the yield strength of 280 MPa for the materials.

4. Discussion and conclusions
In this study, for the static analysis, we simulated an 800 N force being exerted on the load boards to determine the most probable site at which the trolley materials would show deformation and fatigue, and we found that the maximum displacement of 0.12581 mm—though a very small value—occurred at the free outer end of the load boards, whereas the regions subjected to most of the loading did not show deformation and, consequently, did not undergo the associated plastic deformation.

The maximum equivalent elastic strain of the load boards subjected to loading and the maximum equivalent stress of plastic deformation, as predicted by our simulations, were both present in small regions around the head edge of a screw on the hinge connecting load board A. However, the small values of these two parameters indicated that the plastic deformation characteristics of the structure were not affected.

Regarding the safety coefficient, our analysis revealed that it was always greater than 1.5 and, thus, the safety of the entire structure was only impacted to an extremely low extent. The fatigue lifetime was predicted to be at least 4,878,500 cycles, and, when loaded repeatedly, the trolley did not show a change in the fatigue lifetime, neither in the main framework nor in the load boards.
The screws, which had an almost-negligible effect on the entire performance of the trolley, may be replaced with other high-strength materials, such as low-carbon alloy steels or medium-carbon steels, so that the product quality may be further optimized.

Transient analysis showed that the maximum deformation (in board A) of the porous area had a higher strength and higher bending load than the non-porous area. The overall transient response was close to the steady-state, with a negligible effect on the structure, and the porous plate will not cause a large vibration displacement due to loading with a vertical force under 800N.

The results of the study indicate that replacing regular load boards (that have supports on all the four sides) with rectangular, porous foldable load boards (that have supports on only two sides), significantly reduces the collapsed volume, the weight of the trolley, and the storage space after folding, to achieve an optimal design by providing structural improvements.

Foldable porous board can be applied to tables, chairs, ladders, and so on, to meet the requirements of lightweight and storage-minimum specifications. In terms of materials, materials with high stability and impact strength, such as ABS, can be used.

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5. References

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