Oral Presentation

The 7th TSME International Conference on Mechanical Engineering
13-16 December 2016

AME0011

Preliminary Design of Lightweight Body of Electric Bus for Thailand

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Abstract

This study aims to design, analyze and optimize a lightweight structure suitable for long-range electric bus which can be fabricated in Thailand. The design must meet with the structural requirements. Aluminium has been selected instead of conventional steel to reduce structural weight and enhance energy consumption efficiency. A finite element model of electric bus is created and loading conditions including gravitational, longitudinal and lateral loads are defined. Topology optimization is performed to analyze the positions of roof and side members using Optistruct software. The topology results of bus body show the novel configuration of roof members. The preliminary bus body structure is modeled according to the optimization results and reanalyzed to evaluate bending stiffness, torsional stiffness and natural frequency. Finally, the analysis results show that the new configuration of roof members can effectively reduce the weight of bus body by 183 kg (18.9%) compared to that of conventional structure without compromising structural stiffness.

Keywords: Lightweight bus body, Electric Bus, Finite Element Method

1. Introduction

Over the past decade, global warming crisis and air pollution are the major concerns of the world community. Several developed countries have paid high attention to the protection of global temperature increase and also the reduction of greenhouse gas emission to reduce environmental impact. According to the report of the International Energy Agency (IEA) [1], global temperature would increase about 6°C by 2050. The 2 Degree Celsius Scenario (2DS) plan is published to propose global emission pathway for each sector which aims to limit the average temperature increase to 2°C. The transport sector has a great potential to reduce 21% of CO_2 emission by increasing the number of plug-in electric vehicle to 75% of all vehicle sales by 2050. In addition, the Electric Vehicles Initiative (EVI) is multi-government policy forum that encourages the use of electric vehicles and projects to employ at least 20 million electric vehicles by 2020. In Thailand, electric vehicle currently is the topic of interest after Thai government has approved "Electric Vehicle Promotion Roadmap" [2] and projected Thailand as the hub of electric vehicle industry in ASEAN by 2019. Electric bus is planned to service in public transport sector and support Thai makers to manufacture 1,000 electric buses annually.

For electric vehicles, the cost of batteries is considerably more expensive than other components. Hence, the lightweight structure is necessary to enhance energy efficiency of electric vehicle or extend more running range per charge [3]. For electric bus, aluminum alloys is preferable due to its advantages of its low density one third that of steel, high strength to weight ratio and excellent corrosion resistance [4]. In addition, the benefit of 100 kg weight reduction of city bus can save 0.2l/100km (diesel) which is equivalent to 1.13 kWh/100km in electrical energy [5].

To design a bus body, many literature investigated the effects of real driving conditions on the structural strength of bus body [6, 7]. Linear static analyses were performed by means of finite element analysis according to each situation. Lightweight design for bus body yields an influential solution to the efficiency of energy consumption. Methodologies in optimization were employed to reduce the bus weight including gauge optimization to find the proper thickness of bus structure based on given structure [8-10], and topology optimization to optimize the material distribution over the design space. With topology optimization, the bus structure is obtained directly from the stiffness requirements. The analysis may identify the optimum position of structure under given circumstances. Only few papers have studied on the topology optimization to design lightweight bus body [11, 12].

The primary objectives of this study are to use topology optimization via Optistruct FEM software to identify the positions of structural members of the electric bus body and to design the aluminium bus body according to design conditions and structural stiffnesses. The bus body is analyzed to ensure the strength requirements. The optimized design proposes a new feature on roof structure which can enhance its structural performance and reduce weight of bus body compared to conventional structure.

2. Design Methodology

This section is dedicated to the methodology of design processes of electric bus body. The design procedures could be parted into 2 major steps as presented in Fig. 1.

Firstly, the structural optimization step is proposed as the sequence of design procedures based on topology analysis with pre-defined non-design and design spaces. An initial finite element model of bus body is created. Topology optimization is carried out by Optistruct version 13.0 to attain an optimized layout of structural members of the bus body under the design conditions. Design conditions for bending and torsional stiffness are included in topology analysis.

After the structural optimization, the bus body model is created following topology optimization results. Structural members are selected and assigned to bus body. Design conditions are then reanalyzed to assure that the performance of bus body should meet with requirements.

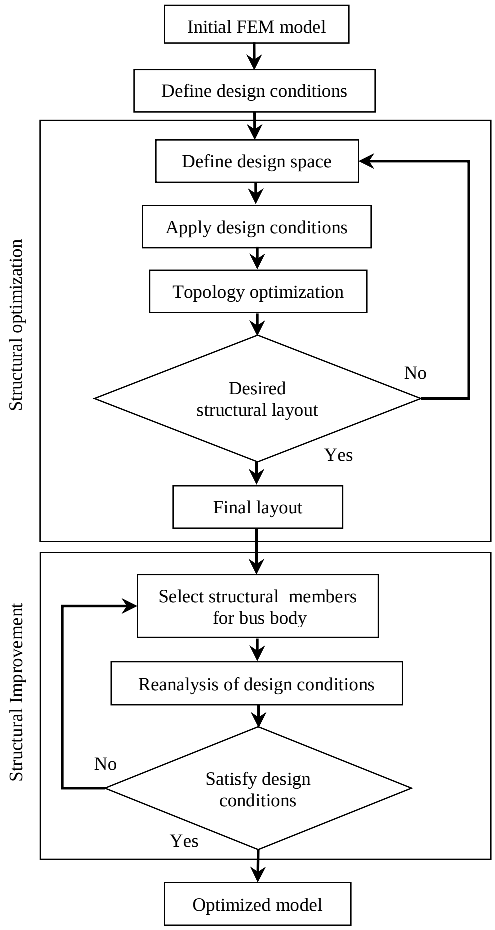

Fig. 1 Flowchart of design methodology

2.1 Finite Element Model for Topology

An initial low-floor electric bus body was modeled as a box-like model placed on chassis with the dimensions of 2.55 m width, 11.8 m length and 3.36 m height. Material of bus body has been selected as aluminium alloy 6061 grade with a T6 temper. Material density is 2,700 kg/m^3. Young’s modulus is 70 GPa. Poisson’s ratio is 0.3 and yield strength is 275 MPa. Chassis model was fabricated from steel with density of 7,850 kg/m^3, Young’s modulus of 210 GPa, Poisson’s ratio of 0.3 and yield strength of 351 MPa.

The weights corresponding to subsystems are assigned to the bus body and chassis. The weights of 3 battery packs are located. The first pack is 834 kg on the roof frame above front axle. The second pack of 1,668 kg is on front wheel wells and the third pack of 834 kg is on the rear compartment of chassis. The 250-kg air conditioning system is placed on the roof. The weights of 480-kg window frames and 120-kg doors are distributed on bus body. The passenger seats of 400 kg and floor of 350 kg are assigned on chassis.

The bus model is meshed with 25-mm 4-node quadrilateral shell elements (QUAD4) with a 6 degree-of-freedom per node. The finite element model of bus body is divided into 2 spaces as shown in Fig. 2; design spaces (shown as yellow areas) and non-design spaces (shown as red frames). Initially, the design spaces comprise of roof and side structures, and the non-design spaces include the front and rear pillars, cantrails and chassis structure.

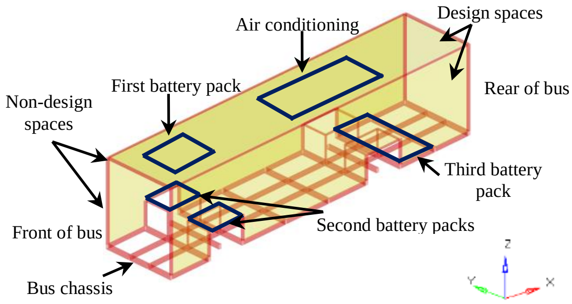

Fig. 2 Initial FEM model for topology analysis

2.2 Design Conditions

In order to design the bus body, the loads imposed on the bus body due to normal running conditions and structural stiffnesses are taken into considerations in the topology optimization. Static analysis are performed to evaluate and ensure that the bus body having both structural requirements and sufficient structural strength for driving conditions [13]. The design conditions consist of:

1. Bending stiffness (K_B) is determined to evaluate ability of bus structure for carrying the weight of components (e.g., air conditioning system, seats, battery packs) and also its structural weight. The bus structure is subjected to gravitational acceleration in vertical direction (Z-axis) and the wheel supports are constrained as pinned support as shown in Fig. 3(a). K_B can be calculated by

$$K_B = W\left(\frac{1}{d_{R,\max}} + \frac{1}{d_{L,\max}}\right) \tag{1}$$

where W is the total weight. The displacements $d_{R,max}$ and $d_{L,max}$ are measured from maximum vertical displacements on the right and left side of the bus structure, respectively, as shown in Fig. 3(a).

2. Torsional stiffness (K_T) is evaluated as the structural rigidity of bus body to withstand twisting which may occur due to uneven road conditions. To calculate K_T, the coupling forces (F) are applied to both left and right of front axle whereas the translation in Z direction is fixed at the middle of the front axle.

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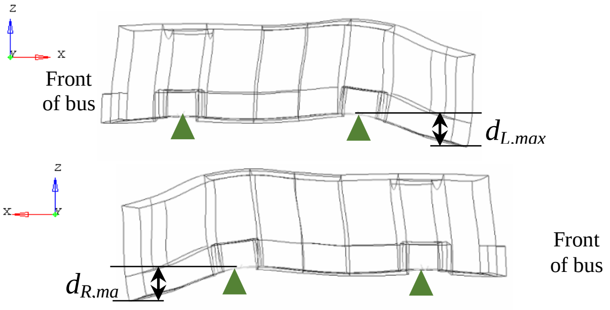
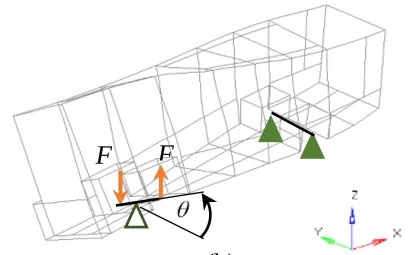
The rear wheels are constrained as pinned support as shown in Fig. 3 (b). The angle of twist (θ) of front axle is measured to calculate K_T as following Eq. (2)

$$K_T = \frac{Ft}{\theta} \qquad (2)$$

where t is track width of axle.



(a)



(b)

Fig. 3 Boundary conditions for (a) bending and (b) torsional stiffness

3. Longitudinal loading occurs when bus is accelerated or decelerated. The whole structure of bus is subjected to acceleration with magnitude of 0.75g in longitudinal direction (X-axis). The constraints are defined in the same manner as in the bending case, while loads from gravity ramain.

4. Lateral loading happens when the vehicle is driven around the corner or turning. This loading is applied to the whole bus structure with an acceleration of 0.75g in lateral direction (Y-axis). The constraints of wheels are applied as in the bending case whereas gravity load is valid.

5. Natural frequency of the bus structure should be higher than the frequency induced by road conditions to avoid the occurrence of resonance.

3. Structural optimization

3.1 Topology Optimization

Topology optimization is a mathematical method used to optimize a structure in mechanics problem [14]. The topology optimization is used to find the optimum distribution of material under designated conditions [15]. In this optimization problem, the purpose is to acquire the appropriate preliminary layout of the bus body that can reduce the mass of bus structure. Therefore, the mass fraction of bus body is set as a response parameter of the optimization problem. In order to constrain the weight in an appropriate range for an electric bus body (approximately 1.5 to 3.1 tonnes), the mass fraction is assigned to be between 0.1 and 0.2 of the total weight of non-design spaces. The objective of optimization problem is to minimize the compliance of bus body. In other words, the stiffness of bus body is to be maximized. The parameters of topology optimization problem can be performed as:

Objective: Minimize $\frac{1}{2}u^T Ku$,

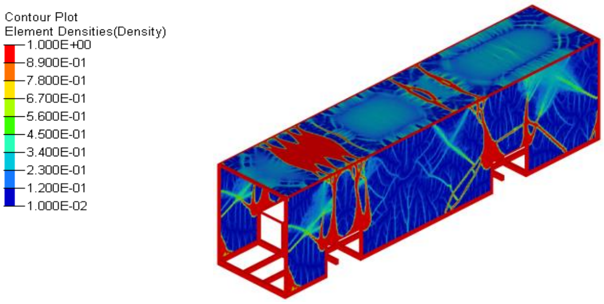
where u are the displacement vectors and K is the global stiffness matrix.

Constraints: 0.1< mass fraction < 0.2 and

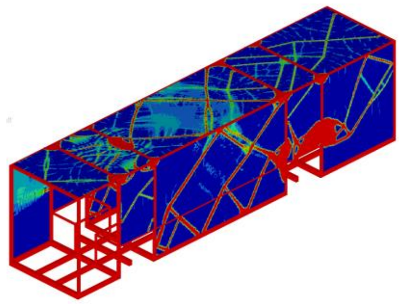
Design variables: Side and roof spaces

3.2 Topology Optimization Results

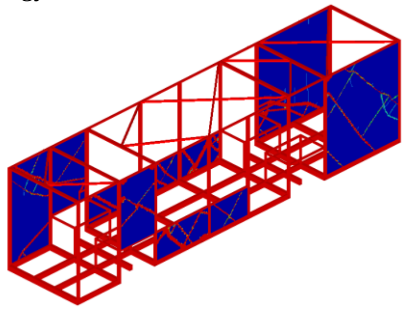
During topology analysis, the design variables of the initial bus model in Fig. 2 are gradually reduced by structural members obtained by each optimization loop as depicted in Fig. 4. The contour plot shows the distribution of element densities varied from 0 to 1. The topology result in Fig. 4(a) shows that the pillars and roof members should present at front and rear around wheel hubs. Fig. 4(b) portrays configurations of roof members like a cross-structure which significantly different from the roof structure of conventional bus. After iterative topology optimization process, the final structural layout of bus structure is obtained as illustrated in Fig. 5.



(a) Topology result for side pillars and transverse roof beams

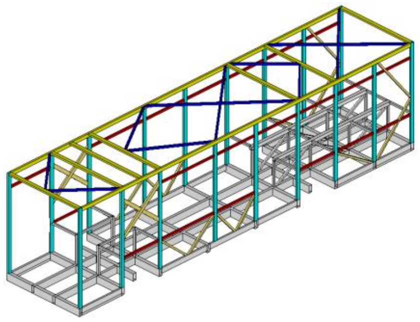


(b) Topology result for cross-roof members

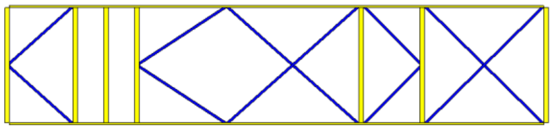


(c) Topology result for side cross bars

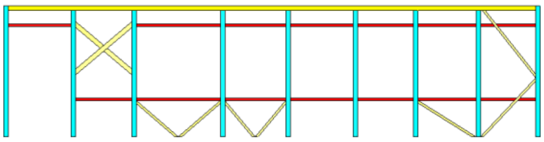
Fig. 4 Topology results



(a) Whole structure



(b) Roof structure



(c) Side structure

Fig. 5 Structural layout from optimization

4. Structural Improvement

After the optimized structural layout is obtained as described in the previous section, the structural profiles are selected and assigned to the final bus model. The structure is then analyzed according to the design conditions stated in section 2.2.

The structural improvement is focused on the roof structure. Due to the weights of battery pack and air conditioners placed on the roof structure, vertical deflections of roof structure are extensively large which may lead to inadequate bending stiffness. Thus, transverse beams are added to enhance the roof stiffness as depicted in Fig. 6. The section of roof members is changed by consideration of area moment of inertia of the member. The size optimization is implemented to find the optimum thicknesses for selected roof members. The sections and thicknesses of roof members are listed in Table. 1.

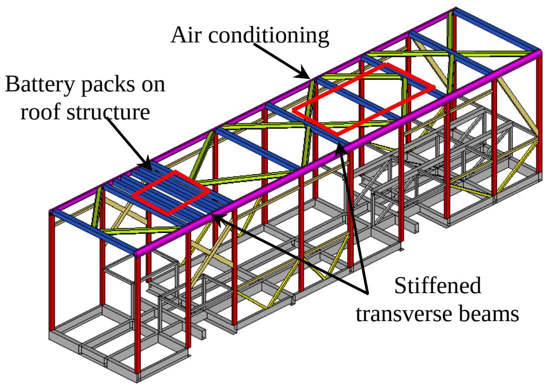
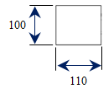
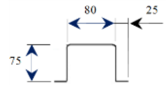
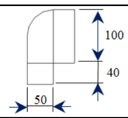


Fig.6 Optimized model

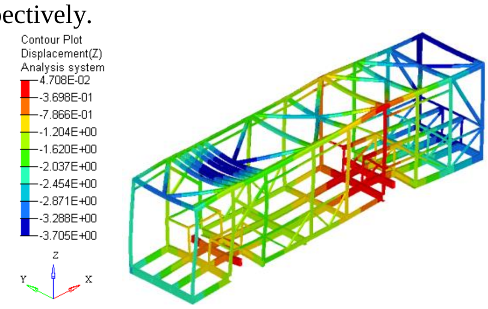
Table. 1 Profiles of roof structure

Part	Thickness (mm)	Dimensions (mm)
Transverse beam (square)	5	
Cross beam (hat shape)	4	
Cantrail	4	

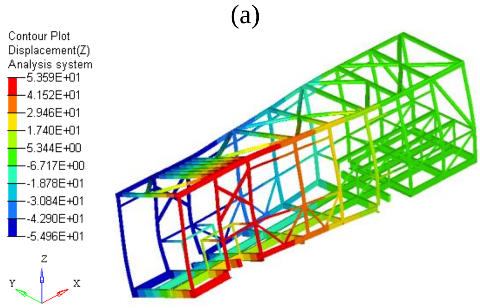
5. Results and Discussions

After the optimized bus structure is obtained, the loading conditions based on design criteria are applied in order to evaluate structural integrity. The analysis results from each case are reported and the performance comparisons of the optimized and the conventional models are discussed.

The maximum vertical deformations according to bending load case and the angle of twist of torsion case are measured. The deflections of the bus structures from bending and torsion case are presented in Fig. 7 with a proper scale factor. The maximum deformations on the left and right side of the structure in bending case are 3.47 and 3.16 mm, respsctively. The twisting angle from torsion load is 46.7 degrees. Therefore, the bending and torsional stiffnesses of the optimized model are 71,576 N/mm and 35,812 N.m/deg, respectively.



(a)



(b)

Fig.7 Deflections of (a) bending and (b) torsion cases

For longitudinal loading case, the maximum equivalent stress in magnitude of 62 MPa is presented around the front axle. In the case of lateral loading, the maximum equivalent stress is noticed adjacent to the rear axle of the chassis with value of 105.2 MPa. The results of natural frequency of the bus structure are observed as following; the first mode is 4.6 Hz, the first torsion mode is 7.1 Hz, and the first bending mode is 19.3 Hz. The frequency of first bending mode is slightly lower than a range of 22-25 Hz as referred in literature [16].

When the structure is examined according to von Mises yield criterion, there is no yielding presents on any parts of bus structure for all design conditions. The most severe case with the maximum equivalent stress is when the bus body is subjected to lateral loading from cornering as mentioned earlier. Fig. 8 shows distribution of equivalent stress under lateral load case.

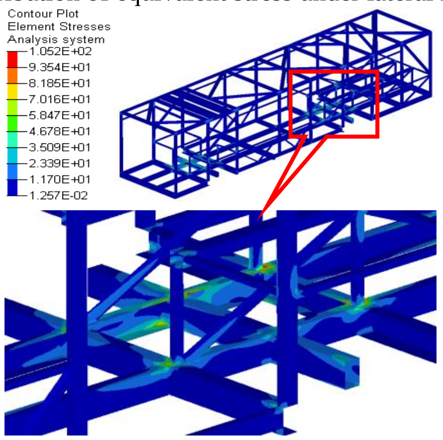
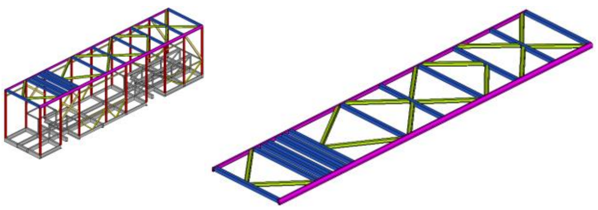
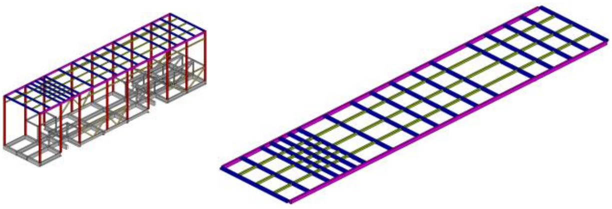


Fig. 8 Maximum equivalent stress in the case of lateral load

The optimized model from topology optimization offers new configurations of roof structure (Fig. 9a) which differs from the conventional bus body counterpart (Fig. 9b). The performance of both models are compared focusing on the bending and torsional stiffnesses, and mass of the bus body by using an identical side structure. The requirements for bending stiffness and torsional stiffness for the bus body are 69,000 N/mm [16] and 18,000 N.m/deg [7], respectively.



(a) Optimized roof structure



(b) Conventional roof structure

Fig. 9 Roof structures of optimized and conventional bus body

The comparisons of normalized structural stiffnesses of both bus models are plotted in Fig. 10. The optimized bus model yields higher stiffness than that of the conventional model and also passes the structural stiffness requirements. The bending and torsional stiffnesses of the optimized model are increased by 82.8% and 20.1%, respectively, when compared with the conventional model. Fig. 11 illustrates mass comparisons of structural parts in conventional and optimized mode. The mass of the roof structure of the optimized model is lighter than the conventional model by 31.3% while the mass of side structures are at the same level which leads to a reduction in the overall mass of bus structure by 183 kg (18.9%).

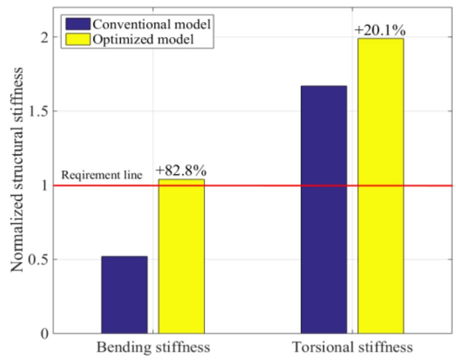


Fig. 10 Normalized structural stiffness comparisons of conventional and optimized bus models

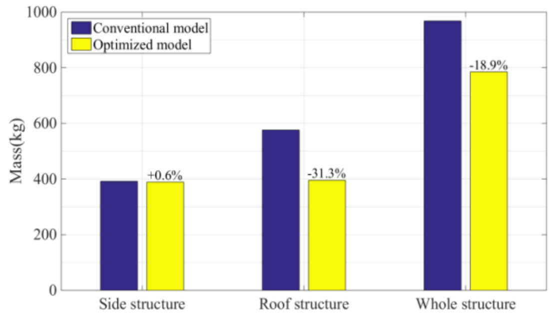


Fig. 11 Mass comparisons of conventional and optimized bus models

With the 183 kg weight reduction of optimized bus model, battery power required for 250 km driving range can be calculated. The optimized model can save battery capacity by 5.2 kWh compared to conventional model which reduces battery cost by US$ 1,560 (US$300 per kWh) [17] and decreases the raw material cost by US$ 390 [18].

6. Conclusions

The topology optimization was carried out to generate the optimum configurations of the electric bus body structure with an objective of compliance minimization. The structural members are then selected and assigned to the model in order to meet the design conditions based on structural stiffness of the bus body and driving conditions. With the new configurations of roof structure, the bending and torsion stiffness of bus body increase by 82.8% and 20.1%, respectively whereas the lightweight bus structure is achieved. The mass of the optimized bus body is decreased by 183 kg (18.9%) comparing to the conventional design without compromising structural requirements. In the current work, safety of bus structure in case of accident such as rollover is not considered and should be further studied.

7. Acknowledgement

The authors gratefully acknowledge financial supports provided by the Research and Researcher for Industry (RRi) under the Thailand Research Fund (TRF) no. MSD 59I0020.

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