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Light Weight Optimisation of Electric Bus Body Structure Using Finite Element Methods

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Abstract

A lightweight structure has been widely identified as one of the key components for a sustainable success of electric vehicle (EV) implementation. The current study focuses on investigating a possibility of reducing a structural weight of an existing EV bus prototype developed in Thailand. The Multi Objective Genetic Algorithm (MOGA) was applied to the structural stiffness analysis for minimising the bus weight by a determination of an appropriate thickness of the structure. The numerical analysis only considered the body structure of the EV bus prototype, which was modelled by beam elements. The structural stiffness analysis performed in this work consisted of loading behaviours commonly encountered during normal operation of the bus i.e. bending, braking, cornering, and torsional. For optimisation, constrain variable considered was corresponding torsional stiffness. The design parameters of cross member thicknesses were divided into six different groups depending on physical area of concern. The resulting stiffness of the structure with optimum cross-sectional thickness under different design constrains would be compared together to determine the appropriated thickness of structural members under a purpose of weight reduction for the EV bus. Furthermore, the corresponding stresses of the structure with chosen cross-sectional areas under normal driving situations were calculated to evaluate the overall structural strength of the proposed design.

***Keywords**:* Bus Structure, Optimisation, Weight Reduction, Finite Element Method

1. Introduction

Currently, Thailand is one of the top greenhouse gas (GHG) emitting countries in the world [1]. This has resulted in several measures being introduced into a national roadmap including an introduction and implementation of electric vehicles (EV) in Thai society. Coupled with a recent Paris global climate agreement pledge by Thai government to an unconditional 20% GHG emission reduction by 2030 [2], proper EV applications in Thailand is now very imminent. As a result, a significant interest has recently arisen from all relevant parties, i.e. private sectors, academics, research institutes, in order to contribute to the success and sustainability of the project.

Among the types of EV considered, an EV bus has been identified as a key component in a public transport sector and been given a priority for the first phase of the policy. This was mainly because this particular section of the industry contains mainly local Thai companies and is not strongly influenced by the oversea car manufacturers. Furthermore, three technical areas has been widely identified as a key challenge in future EV development i.e. lightweight structure, battery system, and drivetrain. The present work focused on the first area due to a limitation in design and engineering analysis of load-bearing structures at local bus assemblies. There are several approaches suggested in a literature to achieve a weight reduction of the bus structure. It could be achieved by either the choice of the lighter materials [3, 4], by eliminating the non-necessary members of the structure [5], or by remodelling the design of the structure [6]. For the current work, in order to keep the overall original design of the bus as well as for a convenience of retaining the existing figs and fixtures of the bus assemblers, a weight reduction was carried out by adapting the beam thickness of each bus member parts.

Additionally, several criteria has been employed by various researchers to come up with weight reduction solutions, mainly via the help of computational simulations. A passenger bus was simulated under torsion conditions and 8% reduced weight was proposed by removing or modifying the members which displayed stress under 100 MPa [7]. In another work, the weight of bus superstructure could be reduced by 2.65% from substituting lighter fibre-reinforced materials in pillars and sidewall beams [8]. The roof and sidewall thicknesses were varied to reduce the transit bus mass under rollover conditions [9].

The aim of this study was to investigate a possible weight reduction of an existing electric bus prototype developed by National Metal and Materials Technology center (MTEC, NSTDA) as shown in Fig. 1 by revising the member thicknesses without eliminating or reshaping the members and the design of the structure. The optimisation process under torsion condition was applied to find the appropriate thickness for different groups of structural member using a commercial CAE software, ANSYS [10]. The structural strength of the optimised body was then compared with that of the original body in terms of the

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Fig. 1 9-metre BEV bus prototype

resulting combined stress by carrying out the structural analysis under several normal driving conditions.

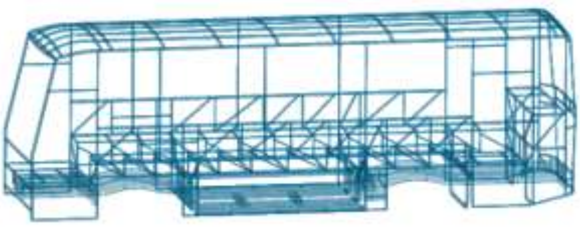
2. Methodology

The 9-metre EV bus investigated in this work was made from the structural steel with the material properties: density of 7850 kg/m^3, Young's modulus of 200 GPa, Poisson's ratio of 0.3, and static yield stress of 250 MPa.

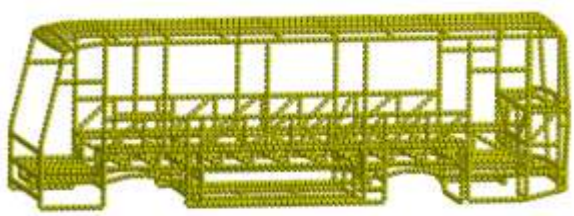
Only structural members and chassis were considered in the analysis excluding windows, axles and wheels, and minor equipment. Furthermore, the main components of the EV bus were taken into account by means of point masses acting on each group of corresponding structural members. The components considered in this study were an air conditioning system, electrical system, electric drive motor, battery packs with the assigned mass of 50, 150, 190, and 2,000 kg respectively. In addition, a total passenger weight of 2,040 kg, i.e. 30 passengers assumed at 68 kg each, was also added into the model.

The overall structural body of the bus is shown in Fig. 2 (a). The FE model contained 74,800 beam elements and 147,571 nodes by 5 millimetres size as depicted in Fig. 2 (b).

The Finite Element Method (FEM) was applied to calculate how the bus structure would deform under various loading cases related to normal driving



(a) CAD model



(b) FE mesh model

Fig. 2 CAD and FE model for the bus structure

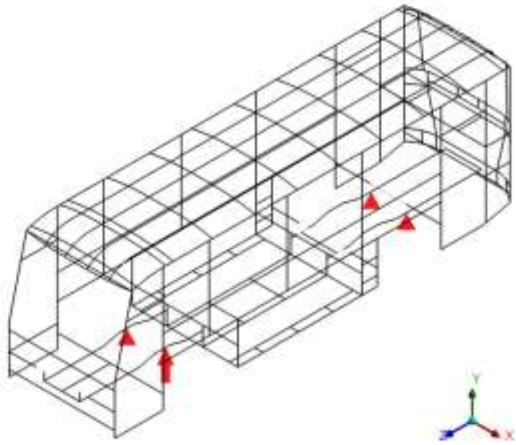


Fig. 3 Constraints boundary condition for torsion case

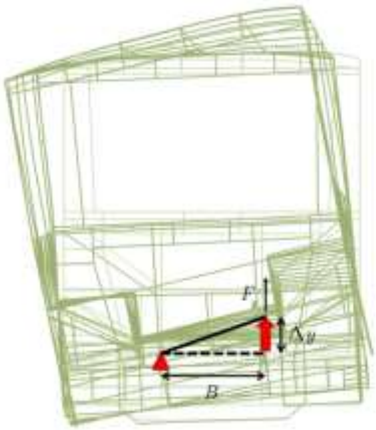


Fig. 4 Calculation of the torsion stiffness

behaviours of the public transport buses. A linear-static structural analysis was used in this study. In the pre-process step, the geometry of the bus structure was imported as a group of line bodies onto which each part was meshed into small elements and the relevant material properties were applied as well as the point masses representing the main components on the bus included into the model. Then two types of boundary conditions were applied. The first type was a constraint to either fix or allow translational and rotational displacements of nodes. The other type was a loading condition i.e. forces, displacements, accelerations to represent the actual load that would be acting on the whole structure in the driving situation of interest. Finally, the computed results of numerical solutions such as the deformations, stresses and stiffness were displayed in the post-processing step.

Several work from literature [11, 12] mentioned that torsional condition, was an extreme loading situation compared to other driving cases. Hence, it was chosen as the main condition for the optimisation process in this work. In order to evaluate the torsional stiffness, a vertical displacement was applied at one wheel arch location and the other three were attached to the ground as shown in Fig. 3. The torsional stiffness (TSF), K_t, was expressed by [13]

$$K_t = \frac{F_y \cdot B}{\tan^{-1}\left(\frac{\Delta y}{B}\right)} \qquad (1)$$

Where F_y is the reaction force in vertical direction, B is the distance between the left- and right-wheel in the same axle, Δy is the vertical displacement of the lifted wheel arch. This could be shown graphically in Fig. 4. To calculate the torsional stiffness, the left front wheel was lifted by 5 mm vertical displacement [14, 15] at 720 mm away from the location of the right front wheel along the same axle direction. The magnitude of the reaction force was obtained from the analysis results.

3. Optimisation Analysis

The Genetic Algorithm (GA) is widely used in the engineering problems with the aim to find an optimal value that satisfy all given constraints. However, majority of structural design problems nowadays come with a requirement to satisfy numerous design goals simultaneously. To deal efficiently with such optimisation problems, the Multi Objective Genetic Algorithm (MOGA) is one of the methods employed by many researchers. [16] MOGA is a hybrid variant of the popular Non-dominated Sorted Genetic Algorithm-II (NSGA-II) [17]. It is one of integrated modules available in a commercial FE software ANSYS (2016 ANSYS inc., USA).

In this work, a chosen design variable was a thickness of structural members. The bus body members were divided into six different groups according to different cross sectional thickness values as shown in Table. 1. In the optimisation process, the thicknesses of the members from each group were changed along the iteration, i.e. design points, and the torsional stiffness could be calculated at each design point using equation (1).

Table. 1 Cross-sectional dimensions of the structural member groups from different area of the bus (all in mm.)

Area	Width	Height	Thickness
Column	38	38	3.2
Battery carrier	100	100	6.0
Carrier side	50	100	4.5
Roof	25	50	3.2
Waist	50	50	4.0
Central	38	38	3.2

All the variables involved in the optimisation process are summarised in Table. 2. The main objective was to minimise the overall weight of the bus structure from an initial value of 1,954.1 kg. Two design constraints were considered i.e. the corresponding torsional stiffness and the resulting combined stress. For the stiffness, a lower and upper limit of acceptable values obtained during the optimisation had to be defined. An important part was how to select an appropriate range since such a wide range from 1,798 to 48,644 Nm/deg have been reported in the optimisation study of 7 to 11-meter bus in the literature [6, 13, 14]. However, after considering the corresponding stiffness of the original design via the simulation, the torsional stiffness range of 18,000-40,000 Nm/deg as suggested by Lan et al. [18] was chosen for this study. Moreover, the maximum combined stress occurred in any member was not allowed to exceed 90% of the yield strength.

Table. 2 Optimisation variables for weight reduction

Variables	Initial	Lower	Upper
Objective			
Weight (kg)	1,954.1	-	-
Constrains			
TSF (Nm/deg)	21,227	18,000	40,000
Combined stress (MPa)	61.8	-	225
Design variables			
Column (mm)	3.2	0.8	3.2
Battery carrier (mm)	6.0	0.8	6.0
Carrier side (mm)	4.5	0.8	4.5
Roof (mm)	3.2	0.8	3.2
Waist (mm)	4.0	0.8	4.0
Central (mm)	3.2	0.8	3.2

4. Results

The structure mass and the torsional stiffness of the bus body obtained from the simulation of all design points are displayed in Fig. 5. The minimum resultant weight was obtained at the point which showed the minimum torsional stiffness.

The sensitivities of the cross-sectional thickness for each structural group related to the structure mass and torsional stiffness are shown in Fig. 6. Generally, a change in member thicknesses would result in a positive sensitivity on mass and stiffness of the structure i.e. a lighter and less stiff structure could be achieved by a reduction in cross-sectional thickness. A

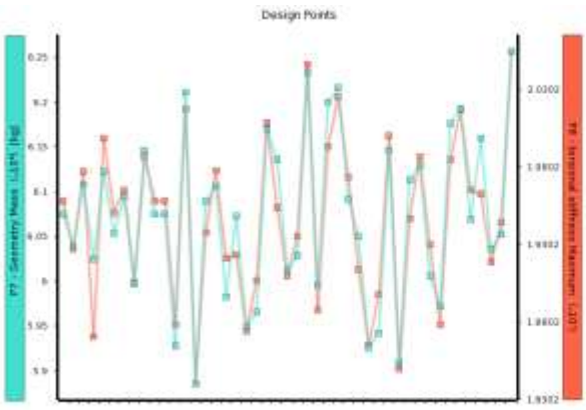


Fig. 5 Calculated structural weight and torsional stiffness of the design points

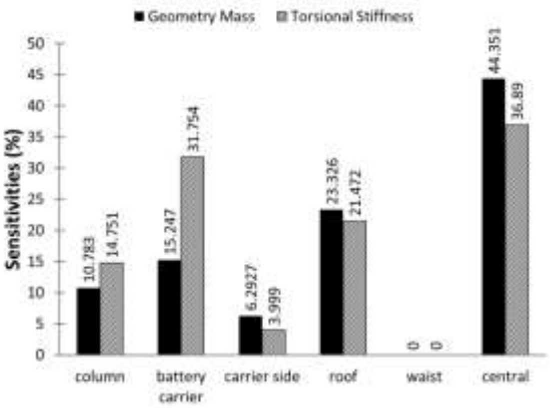


Fig. 6 Sensitivity of thickness of each structural member group on the structural mass and the torsional stiffness

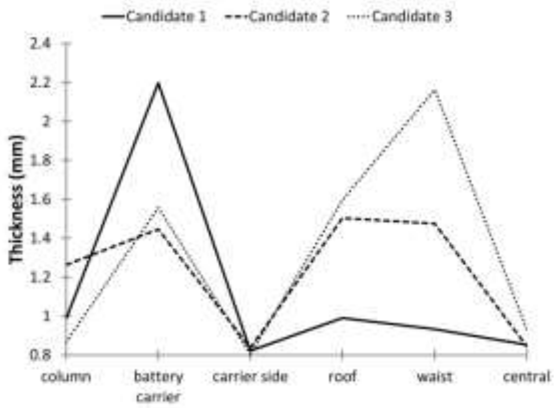


Fig. 7 Resulting design variables for three chosen candidate points

relatively higher sensitivity was observed in the group with a higher number of structural members i.e. central and roof members. Nonetheless, amongst six different member groups considered, only the thickness change of the member in waist area had no effect on the structure mass as well as on the torsional stiffness with a value of 0% sensitivities.

Following the calculation of the design points, three candidate points obtained from the optimisation process were proposed by the software according to, arranged by significance, the main optimisation objective of minimising the weight followed by the constraint of lowest possible stiffness. The corresponding results are listed in Table 3. The resultant structure weight and corresponding torsional stiffness of all three candidate structures are displayed in Table. 4 and Fig. 7, including the percentage variation relative to those of the original baseline structure.

Table. 3 Calculated thickness of the candidate points from the optimisation process (all in mm.)

Area	Candidate		
	1	2	3
Column	0.9874	1.2625	0.8673
Battery carrier	2.1975	1.4467	1.5573
Battery side	0.8219	0.8354	0.8031
Roof	0.9909	1.5032	1.5949
Waist	0.9328	1.4744	2.1631
Central	0.8520	0.8450	0.9303

Table. 4 Corresponding torsional stiffness and resulting structural weight of the candidate points

Calculated Parameter	Candidate		
	1	2	3
Weight (kg)	1366.2	1392.5	1399.0
Variation (%)	-30.09	-28.74	-28.41
TSF (Nm/deg)	18,029	18,006	18,034
Variation (%)	-15.07	-15.17	-15.04
Stress (MPa)	86.54	87.38	87.83
Variation (%)	40.04	41.40	42.12

5. Discussion

As the aim of this research was to minimise the bus body weight, the candidate 1 was then chosen as the suitable choice with the estimated 1366.2 kg of structure weight (5796.2 kg of the whole bus body weight).

In order to investigate the appropriateness of the selected design point, the structural thickness of the original bus was modified according to the thicknesses obtained from the optimisation process. As explained in an earlier section, the point mass representing the main equipment installed on the bus was assembled to the optimised structure model. Under a bending condition, the weight of passengers including personal luggages, was represented by 2.5 times of the passengers mass considered in a normal case. Moreover, following the ISO8855 [19], 0.75g of the longitudinal and lateral accelerations were applied to simulate the braking and cornering cases, respectively. For the torsion analysis, the front left wheel arch was lifted 5 millimetres while the three other were fixed.

The combined stresses of the optimised bus structure was analysed and the contours of the stress distribution are depicted in Fig. 8. Under the bending case as shown in Fig. 8 (a), the maximum tensile stress of the optimised model occurred on the junction of chassis cross member and main member behind the front wheel arc similar to that predicted for the original model. However, the location of maximum compressive stress was changed after optimisation. In the optimised model, the location of the maximum compressive stress would occur on the body member behind the left front wheel as opposed to, the left window panel in front of rear wheel estimated in original model. For the braking condition shown in Fig. 8 (b), the location of maximum tensile stress in optimised model changed to the member below the battery compartment in front of left rear wheel as opposed to the right angle of the bus floor member behind the front wheel arc in the original model. On the other hand, the location of the maximum compressive stress was roughly the same. In cornering

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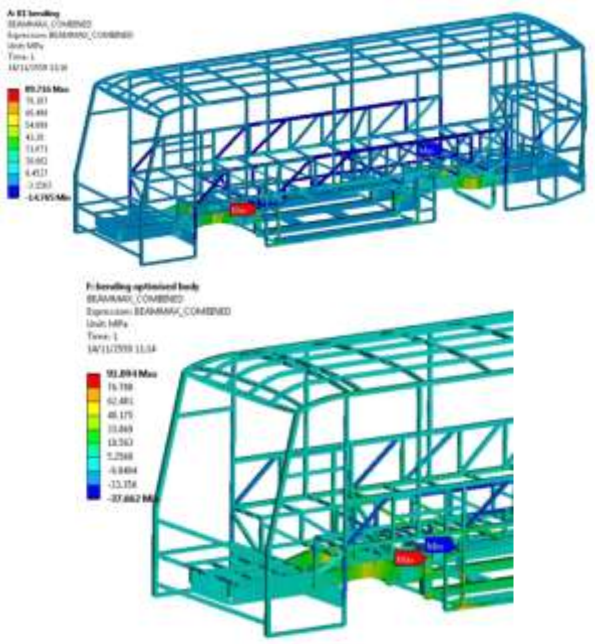
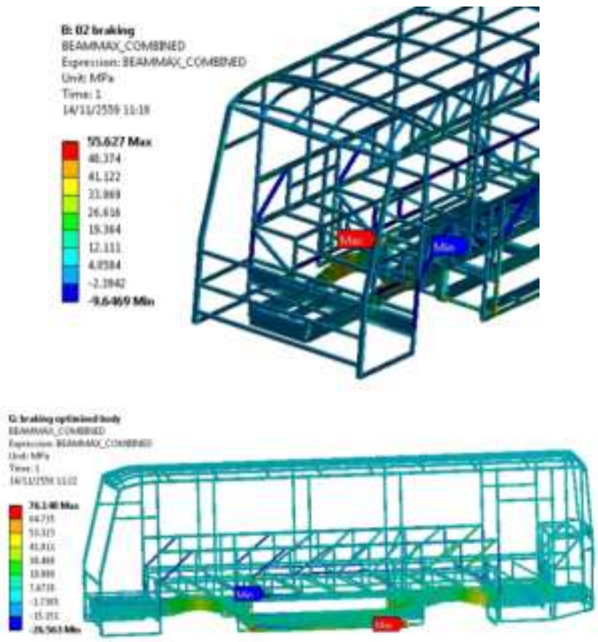
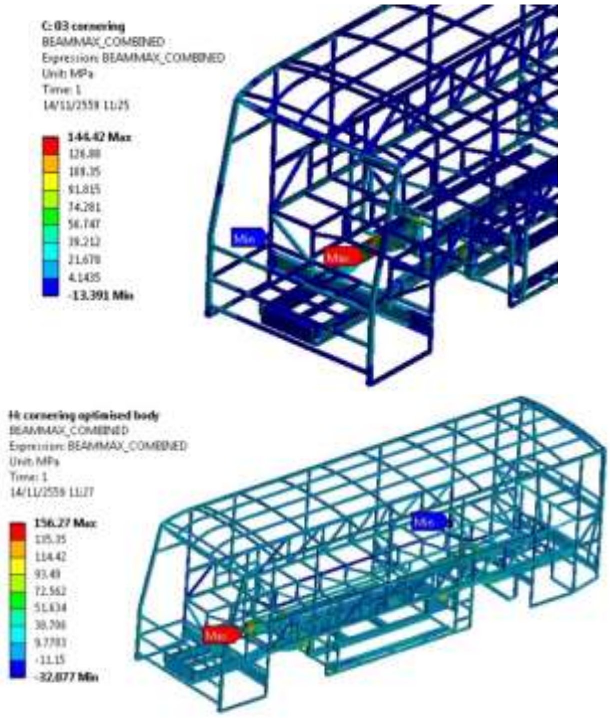
case, the maximum tensile stresses of both original and optimised models were on the right main member of front axle wheel arc. The maximum compressive stress of the optimised model changed to the member of bus floor behind the right rear wheel, as shown in Fig. 8 (c). Finally, the stress contours for the torsion condition are displayed in Fig. 8 (d). The maximum tensile and compressive stresses of the optimised model occurred on the same locations as calculated in the original model i.e. at the corner of the body structure in front of the rear axle and on the cross member of the body on the top of the rear axle for the maximum tensile and compressive stress respectively.



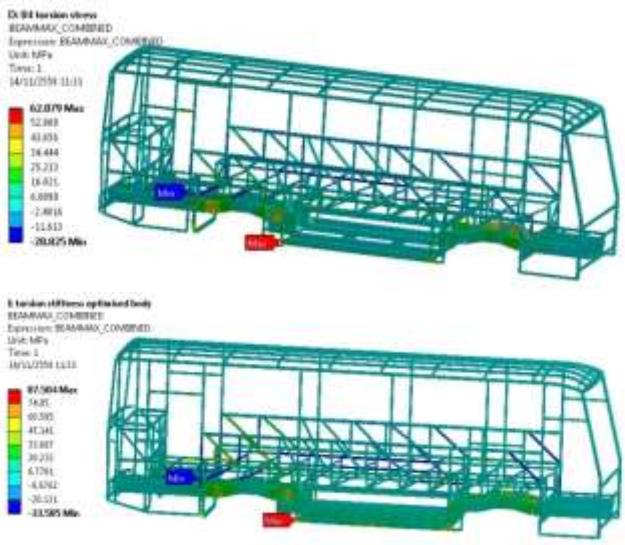
(a) Bending case



(b) Braking case



(c) Cornering case



(d) Torsion case

Fig. 8 Stress contours results showing the locations of maximum tensile and compressive stress between the original (top) and optimised structure (bottom) under different driving/load cases

Table 5 shows the minimum and maximum stresses of the modified bus structure against the original. In all cases of structural analyses, the magnitude of minimum stresses of the optimised body structure were slightly higher than those of the original body. Additionally, the maximum stresses from bending, braking, cornering, and torsion cases showed only slight difference between the two structures.

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Table. 5 Combined stresses (MPa) compared with baseline structure

Case	Baseline		Optimised	
	Min	Max	Min	Max
Bending	-14.77	89.72	-37.66	91.09
Braking	-9.65	55.63	-26.56	76.15
Cornering	-13.39	144.42	-32.08	156.27
Torsion	-20.83	62.08	-33.59	87.50

6. Conclusions

The body structure of the 9-meter BEV electric bus prototype was optimised using Multi Objective Genetic Algorithm or MOGA. The goal of this study was to reduce the structure mass by means of lowering the cross sectional thickness of different groups of beam members. The torsion loading condition was used in the analyses as a problem statement in the optimisation process. The corresponding torsion stiffness of the structure was then used as the design constraint. The weight and the torsional stiffness of the optimised body was 30.09% and 15.07% lower than that of the original body while the combined stress was increased by 40.04%. After the modification, the bus structure was supposedly weak at the connection joints around the battery carrier area and at both front wheel arcs of the bus structure.

7. References

[1] Top 25 greenhouse gas emitting countries. (2015). Statistic Data, URL: http://climatescorecard.org/, accessed on 03/10/2016.
[2] 6th annual Sustainable Innovation Forum (SIF15). (2015). Forum, URL: http://www.cop21paris.org/, accessed on 03/10/2016.
[3] Feng G. (1999). Finite element analysis for bus body structure. *Chinese Journal of Mechanical Engineering*, vol.35(1), pp. 91-95.
[4] Saito M., Iwatsuki S., Yasunaga K., and Andoh K. (2000). Development of aluminium body for the most fuel efficient vehicle, *JSAE*, vol. 21, pp. 511-516.
[5] Lee M.M.K., Pine T., and Jones T.B. (1999). Automotive box section design under torsion. Part 2: Behaviour and implications on weight reduction. *Proc. Instn Mech. Engrs*, vol.214(Part D), pp. 473-485.
[6] Jain R., Tandon P., and Vasantha Kumar M. (2014). Optimization methodology for beam gauges of the bus body for weight reduction, *Applied and Computational Mechanics*, vol.8, pp. 47-62.
[7] Osvaldo R., Edgar I.R., Victor H.J., Rafael S., and Armando O. (2012). Efficient optimisation of the structure of a passenger bus by iterative finite element models with increasing degrees of complexity, pater presented in *EngOpt 2012 – 3rd International Conference on Engineering Optimization*, Rio de Janeiro, Brazil.
[8] Jakub K. and Tadeusz U. (2013) Multi-material design optimization of a bus body structure, Journal of KONES Powertrain and Transport, vol.20(1).
[9] Liang C.C. and Le G.N. (2012). Lightweight optimization of bus frame structure considering rollover safety, *WIT Transactions on Ecology and The Environment*, vol.155, pp. 1185-1196.
[10] ANSYS (2016). *Structural Analysis Guide*. ANSYS Inc.
[11] Zhang X., and Tao. Z. (2009). A study on shape optimization of bus body structure based on stiffness sensitivity analysis, *IEEE*, pp. 1225-1229.
[12] Shinabuth D., Benyajati C., Lapapong S. Pimsarn M., and Okuma M. (2012). Finite element analysis of an electric bus body structure in real driving conditions, *3rd TSME-ICoME*, October.
[13] Boada B.L., Gauchia A., Boada M.J.L., and Diaz V. (2007). A genetic-based optimization of a bus structure as a design methodoly, *12th IFToMM World Congress*, Besancon.
[14] Guachia A., Diaz V., Boada M.J.L. and Boada B.L. (2010). Torsional stiffness and weight optimization of a real bus structure, *International Journal of Automotive Technology*, vol.11(1), pp. 41-47.
[15] Ministry of Urban Development. *Urban Bus Specifications-II*, Government of India.
[16] Abdullah K., David W.C., and Alice E.S. (2006). Multi-objective optimization using genetic algorithms: A tutorial, Reliability engineering & System Safety 91, pp. 992-1007.
[17] Kalyanmoy D. (2002). A Fast and Elitist Multiobjective Genetic Algorithm: NSGA-II, *IEEE Transactions on Evolutionary Computation*, vol. 6, no. 2, pp. 182-197.
[18] Lan F., Chen J., and Lin J. (2004). Comparative analysis for bus side structures and lightweight optimization, Proc. Instn Mech. Engrs, vol.218 Part D: *J. Automobile Engineering*, pp. 1067-1075.
[19] ISO8855 (1991). Road Vehicles-Vehicle Dynamics and Road-holding Ability. *International Organisation for Standardisation*. Geneva.