Proceedings

of the

SEM Annual Conference on Experimental and Applied Mechanics

Portland, Oregon

June 4-6, 2001

SEM

Society for Experimental Mechanics, Inc.
7 School Street
Bethel, Connecticut 06801 USA
(203) 790-6373; FAX (203) 790-4472
E-Mail: sem@sem1.com
Website: www.sem.org

Modal Dynamic Test-Analysis Validation of Trolleybuses

Agustin Tristan-Lopez. Ph.D.Director. Miguel-Angel Hernandez-Perez
Ingenieria y Estadistica Especializada, Mariano Jimenez 1830, 78280 San Luis Potosi, Mexico
Email: ici1@prodigy.net.mx
Ing. Roberto Ibarra-Serna. Subdirector de Licencias y Uso de Suelo. Calle 5 de febrero y Vicente Villada
Villa Gustavo A. Madero, 07050, Mexico,D.F.

ABSTRACT
Test-analysis validation for a trolleybus structure is done using structural modification applied to a finite element model developed for modal analysis. The structure has been tested using experimental modal analysis, calculating modal frequencies and shapes with excitations in three orthogonal directions. Structural mass modification brought a very close agreement between the model and the test results. The validated model was used in FE simulations of dynamic impacts on the vehicle.

Introduction

The Mexico City Electric Transportation Service (STEDF – Servicio de Transportes Electricos del Distrito Federal), has recently purchased an important group of new trolleybuses, according to very strict quality norms, including structural dynamic specifications. Among these, it was requested that the trolleybus has to resist a load condition of impact, similar to the pass of the vehicle over bumpers or other objects that could occur in the Mexico City's trolleybus alleys.

Finite Element simulations have been developed to analyze some static loading conditions, and theoretical modal analysis was realized at the beginning of the project. Both the static and the dynamic models were validated through specific testing procedures: For static analysis it was performed a series of load tests, using plastic bottles filled with water, deformations were measured with strain gages and a sonic digitizer [1]. Modal dynamic testing using an impact hammer and a set of accelerometers was performed.

The purpose of these tests was to provide the test-analysis validation, not only to be certain that the finite element model was close to reality, but also to identify specific points needing further analysis or reinforcement [2,3]
While test-analysis validation was not primarily considered, we have suggested the STEDF to proceed to this kind of studies in order to have precise simulations, more reasonable models and more accurate results.

The structure has been modeled using different types of elements, including 3D-beams, 3D-springs, plates and masses (Fig. 1). Static and dynamic finite element analysis have been performed using the NISA/DISPLAY FEA software [4,5].

Test-analysis validation indicates that the model needed to be modified in order to be close to reality. We used an interative procedure of structural mass modification, after 31 iterations the final FE model was in very close agreement to tests, even below the expected error values. In this paper we present exclusively the results of test-analysis validation and the convergency curves. Once the FE has been validated, we performed the simulations of the dynamic impact, these results are out of the purpose of this paper and therefore are not presented here.

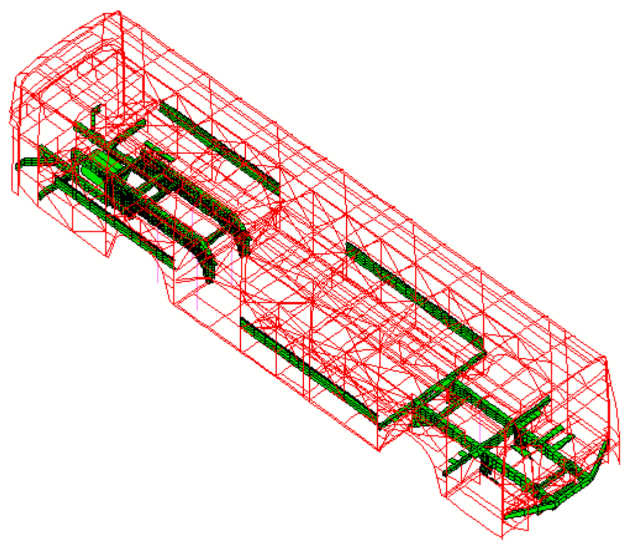
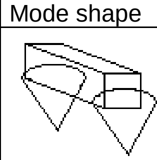


Fig. 1-Finite Element model of the trolleybus

FEA modal analysis

A modal decomposition of the first 25 modes has permitted to identify the rigid and elastic modes. This analysis and the determination of the modal shapes not only gives an insight of the behavior of the structure, but also provides a good idea regarding the position of the nodal points (points in the structure with null deflection) in every mode. With this information it is possible to avoid those nodal points and define the best positions to locate the accelerometers or the points to excite the structure.

The theoretical modal analysis for this type of structures has shown in the past that a first set of natural frequencies associated to rigid modes is expected in the range from 1 to 6 Hz. A second group regards elastic modes between 8 and 14 Hz, the remaining modes appear in frequencies higher than 30 Hz (table 1).

Mode	Frequency Hz	Mode shape
1 Z Rigid roll	1.33	

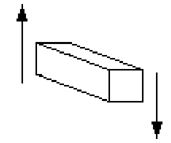
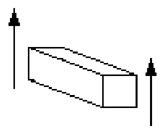
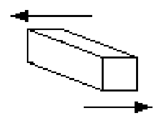
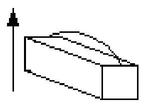
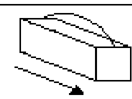
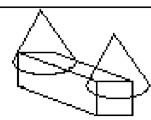
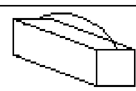
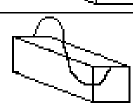
2 Z Rigid pitch	3.48	
3 Z Rigid bounce	4.45	
4 Y Rigid, rotation about Z	9.92	
5 Z Mixed Bounce and flexural mode	12.31	
6 X Mixted Rock and flexural model	14.04	
7 Z Rigid inverse roll	14.81	
8 Z Flexural mode	14.91	
9 Z Flexural mode	19.44	

Table 1 – Mode frequencies and shapes from the theoretical Finite Element model

Modal testing

The model to be used for modal testing was obtained from the FEA model, using an envelope of the structure, with main interest on the platform (Fig. 2).
The 16 points on the platform and 3 additional points on the top of this envelope were divided in two types according to its purpose:

- Response points for the accelerometers (points 1, 11, 16, 17, 18, 19)
- Excitation points for the impact hammer (other points)

The equipment used for this test, shown in fig.3, included three piezoelectric accelerometers, impact hammer, a four channel data acquisition system Zonic Medallion[6], notebook computer and MEScope modal analysis software [7].

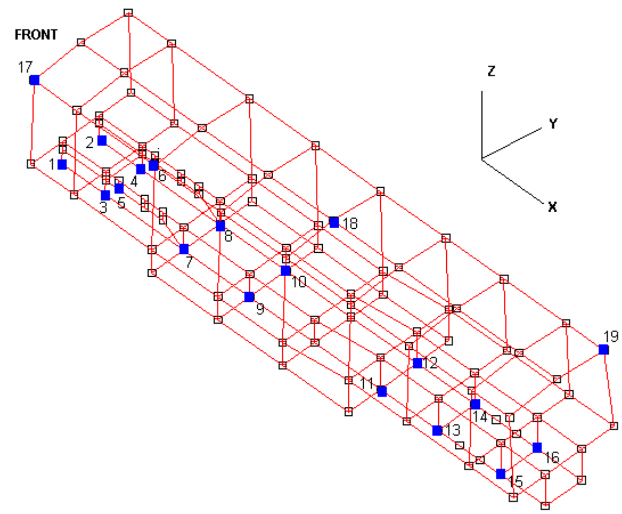


Fig. 2 - Location of points and model for modal testing

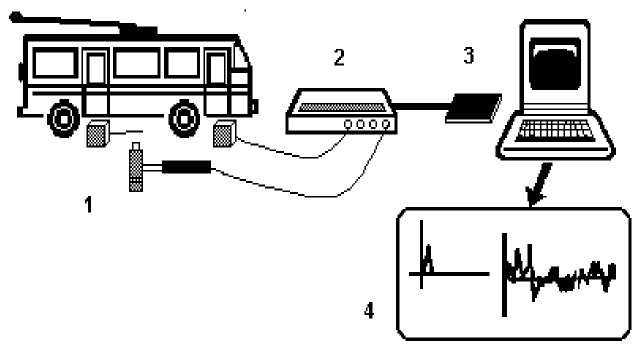


Fig. 3 – Equipment for modal testing: (1) Three acelerometers and impact hammer, (2) Four channel data acquisition system, (3) PCMCIA and notebook computer, (4) software for data analysis and post-processing for modal analysis

The FRF (Frequency response functions) were fitted using the modal software MEScope with a set of 228 experimental curves to get the modal parameters.
Spectral curve fitting is normally done over a set of very “clean” or “perfect” laboratory spectrum curves, nevertheless the STEDF facilities were not optimal for testing, due to the presence of machines in the vicinity, vehicles passing around, several sources of noise and dust. Even that some of the response spectrum curves showed defects (Fig. 4), it was necessary to perform the analysis and provide the best results as possible. The data acquisition system and the software is very flexible, giving several calibration options and setup capabilities, this facilitates the test and has permitted to get satisfactory results.

Despite the problems during the test, it was possible to get reasonable results that have been used later in the test-analysis validation (table 2). In fact the modal frequencies obtained in the test correspond very well to the expected range of values [1, 8].

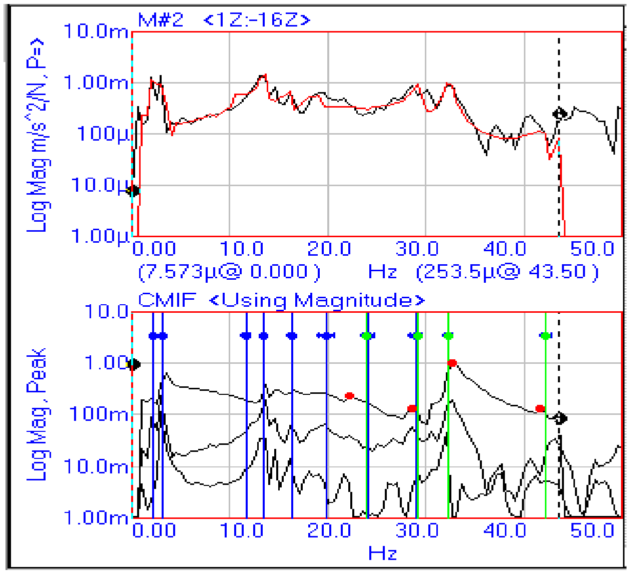


Fig. 4 – Spectral curve fitting

Mode in Z	Frequency Hz	Damping %
1 Rigid Pitch	1.9 - 2.1	4.0 – 4.7 %
2 Rigid bounce	2.7 – 3.1	1.6 – 1.9%
3 Flexural mode	11.3	3.2%
4 Flexural mode	13.2 – 13.4	1.7 – 2.0%
5 Flexural mode	15.9 – 16.2	2.0 – 2.3%
6 Flexural mode	29.1- 31.9	1.6 – 1.8%
7 Torsional mode	34.6 – 35.3	1.7 – 2.1%
Mode in Y		
1 Rigid rotation about Z	2.6 – 2.7	2.2 –2.4%
2 Flexion in plane XY	18.3	4.5
3 Flexion in plane XY	34.7	5.6
Mode in X		
1 Rigid Rock	– 3.4	2.5 – 2.6
2 Flexion in plane XY	15.8	4.4
3 Flexion in plane XY	18.5	4.6

Table 2. Results from modal analysis test

Test-analysis validation

The FE model had to be modified in order to represent the reality as close as possible. We decided to do the modifications for the first 6 modes shown in table 3.

Structural mass modification was performed iteratively using the FEA software. The NISA/DISPLAY software can do structural optimization and sensitivity analysis, nevertheless it does not allow to modify exclusively the mass elements, therefore we developed the following method:

a) A set of mass elements is introduced in the model, each mass is connected to the structure using rigid links; for this analysis we used 8 mass elements b) The mass values are defined at one step and a modal decomposition is calculated. c) Convergency is verified at the step, calculating the difference among the frequencies of the model *fmi* and the reference *fri* given by the modal test. e) New mass values are introduced in the model and steps (b) and (c) are repeated until the desired convergency is attained.

For convergency we stated the following criteria in a maximum of 50 iterations:

Absolute error for mode i :

$$AEi = \left| fmi - fri \right| \leq \delta 1 = 0.1$$

Mean absolute error:

$$MAE = \frac{\sum \left| fmi - fri \right|}{n} \leq \delta 2 = 0.05$$

Mean square error:

$$MSE = \frac{\sqrt{\sum (fmi - fri)^2}}{n} \leq \delta 3 = 0.05$$

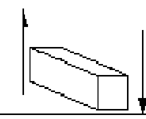
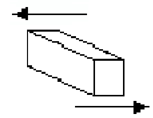
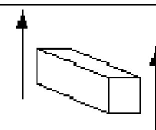
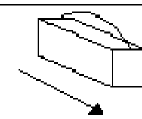
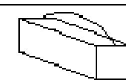
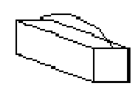
Mode	Frequency Hz	Descriptión
1 Z Rigid	1.9 – 2.1	
2 Y Rigid	2.6 – 2.7	
3 Z Rigid	2.7 – 3.1	
4 X Rigid	3.2 – 3.4	
5 Z Flexible	11.3	
6 Z Flexible	13.2 – 13.4	

Table 3. Modes from the modal analysis test

Convergency was attained in 20 iterations for MSE and in 31 iterations for MAE (fig.5), while the absolute error for each mode, AEi , needed 28 iterations (fig.6).

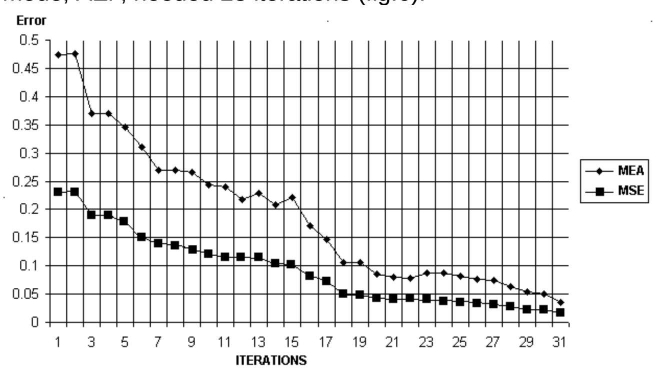


Fig. 5 – Convergency of MEA and MSE

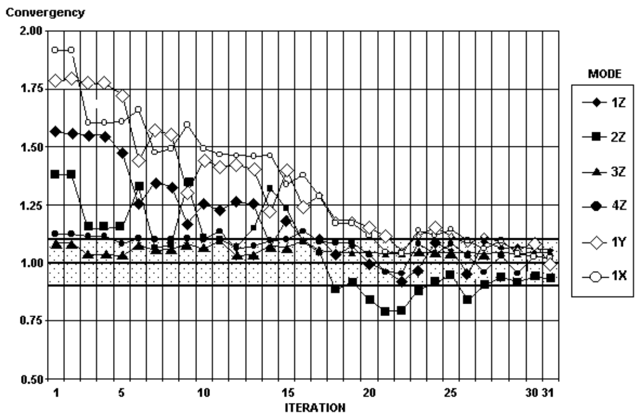


Fig. 6. Convergency of structural modification vs. Modal frequencies and shapes

Flexible modes converge rapidly with an error below 10% from the first iterations, while rigid modes converge in more iterations. Final values are given in table 4 and figure 7 shows the modal shapes of the FE model that are comparable to the experimental modal shapes.

As soon as the test-analsys validation has been satisfactory, a program of simulation of dynamic loads was initiated using the validated finite element model. The analysis indicated some design modifications for the manufacturer, in order to resist the load conditions, specially regarding welding and a few structural dimensions of spars.
With these improvements the STEDF has approved the design of the trolleybuses, that are actually in use in Mexico City as part of the transit solution with electrical vehicles.

Mode	Freq. Hz fmi	Mean experimental value fri	Relative fmi/fri	Error %
Mode 1z	2.1	2.0	1.050	5%
Mode 1y	2.7	2.65	1.001	0.1%
Mode 2z	2.7	2.9	0.931	6.9%
Mode 1x	3.37	3.3	1.021	2.1%
Mode 3z	11.8	11.3	1.044	4.4%
Mode 4z	13.6	13.3	1.023	2.3%

Table 4

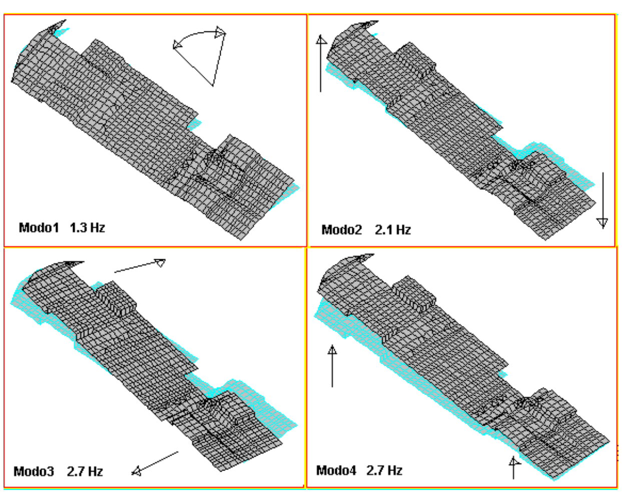


Fig.7 a – Modal shapes of the FE model after validation

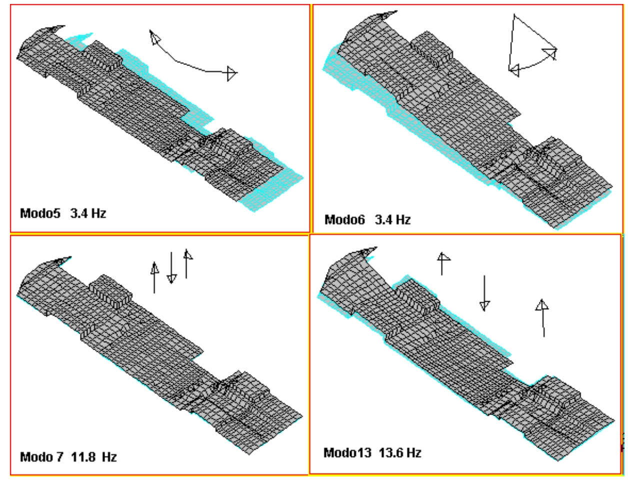


Fig.7b – Modal shapes of the FE model after validation

Conclusions

Test-analysis validation for dynamic behavior of the structure of a trolleybus has been successfully performed for the first 6 modes, results were in good agreement between the experimental results from modal testing and the finite element model. Experimental analysis was developed with an impact hammer and three accelerometers. Once the FE model has been validated, further dynamic analysis were carried out to simulate impact conditions on the structure and some improvements of the structure were done by the manufacturer, according to our results. The trolleybuses are actually in use in Mexico City.

References

[1] Tristán A. Hernández M. Ibarra R. *"Evaluación Estructural de trolebuses"*. Instituto Tecnologico de San Luis Potosi. IV International Meeting of Mechanical Engineering, pp 135-145. October 1998
[2] Brown D. Carbon G. y Ramsey K. *"Survey of excitation techniques applicable to the testing of automotive structures"* Society of Automotive Engineers, 770029, Intl. Automotive Engineers Congress and Exposition, Detroit, 15 pp, Feb-Mar 1977.
[3] Dovel G. *"Modal Analysis, a Dynamic tool for design and troubleshooting"*, Mechanical Engineering, pp 82-86, Mar 1989.
[4] Engineering Mechanics Research Corporation, *"Display III user´s manual"*, Michigan, 1995
[5] Engineering Mechanics Research Corporation, *"NISA II user´s manual"*, Michigan. 1995
[6] Zonic Corp. *"Medallion user´s manual"*, Milford Ohio, pp 1-71, 1998.
[7] Vibrant Technology Inc, *"ME´scope Visual Engineering Series"* Operating manual Ver. 1.0, Vibrant Technology, Jamestown California.343 pp. October 1998.
[8].Centre d´Evaluation et de recherche des nuisances et de l´energie. *"Confort dans l´autobus"*, Institut de recherche des transports. Note d´information No. 19, September 1981.