Development of Design Process for Auxiliary Table of Vibration Testing Machine

Bor-Tsuen Wang [1], Feng-Rong Zhuang [1] and David Lee [2]
[1] Department of Mechanical Engineering,
National Pingtung University of Science and Technology
[2] King Design Company
E-mail: wangbt@mail.npust.edu.tw

Abstract

This paper discusses the design process for the development of auxiliary table of vibration testing machine. The design process is first laid out, and a typical auxiliary table is studied to show the design evaluation. There are three stages in terms of design analysis. First, the table in free boundary is performed by both finite element analysis and experimental modal analysis to validate the analytical finite element model. Second, the fixed boundary conditions corresponding to the real mounting is considered and validated as well. Third, the flatness index to evaluate the table performance is adopted and shown to justify the table design. Finally, the design criteria for a new design of the auxiliary table are presented. This work establishes the design methodology of auxiliary table that can be suitable and coped with the requirement of vibration testing machine.

Keywords: auxiliary table, vibration testing, finite element analysis, experimental modal analysis

1. Introduction

There are many kinds of environmental tests, and vibration test is one of them. A product generally requires environmental vibration tests by using vibration testing machine to characterize the vibration properties, reliability and the ability to fit the environmental test specification.

The coil structure of a vibration testing machine has a small diameter and limits the size of test-device. The vertical auxiliary table or so called head expander is designed to mount on the coil structure, in order to increase the test surface for accommodating large test objects. The shaker dynamic characteristics for a vibration testing machine is not able to be changed, so different size of auxiliary tables should be well designed to suit the shaker's characteristic and related vibration test specifications.

Wang and Chen [1, 2] performed model verification of an auxiliary table (450×450) for both free and fixed boundary conditions. Via experimental modal analysis (EMA) and finite element analysis (FEA) techniques, the equivalent finite element (FE) model can be validated by the comparable agreement of modal parameters. Wang *et al.* [3] had also done a similar study for the 600×600 type of auxiliary table. Chen [4] developed a complete procedure for the design verification and evaluation procedure for the auxiliary table and also established the flatness evaluation model for the table to define the table quality. Wang *et al.* [5] followed Chen's model [4] to compare the performance of different auxiliary tables in terms of flatness index. Wang *et al.* [6] applied the similar process to perform the model verification for the carriage of free-fall shock testing machine. Wang *et al.* [7] studied the coil structure by FEA and EMA to obtain the validated FE model.

The purpose of model verification is to validate the correctness of mathematical model, and so forth the validated model can be applied to model modification, force prediction and response simulation as well as other on-purpose applications. This paper will address the concept and develop the design process for the auxiliary table of vibration testing machine.

Feldmaier *et al.* [8] developed the FE model for the car suspension system and validated the model via EMA, and then the suspension response due to particular loadings can be predicted. Pavic *et al.* [9] studied the vibration transmission between floors of a building and updated the FE model by experimental verification. Wang and Li [10] built a small scale model of boat to perform EMA so as to verify the corresponding FE model, and so forth the dynamic response of double stage vibration isolation system can be well characterized.

This paper not only develops a design verification process for the auxiliary table design, but a 750×750 type of table is also adopted to follow the process to detail the design validation. The design principle is also discussed to provide the design engineer with a practicing guideline. On following the developed process, the new type of table can be fabricated and largely reduce the cost and time for developing the effective and competitive auxiliary table.

2. Development of Design Process for Auxiliary Table

Fig. 1(a) shows the flow chart of design analysis and verification for the initial design of auxiliary table. The steps for the initial design evaluation are discussed as follows:

1. Free Boundary Model Verification: The initial

design of the auxiliary table is performed by both FEA and EMA, respectively. Base on the modal parameters comparison, the FE model in free boundary can be validated. The material properties can be properly justified according to experiments.

2. Fixed Boundary Model Verification: The auxiliary table is attached to the coil structure on the vibration testing machine as in practical test condition. Both FEA and EMA are also performed, respectively, for the table in fixed boundary condition. The fixed boundary parameters that are spring constants can be well calibrated for the vibration testing machine. For different machines, the fixed boundary parameters must be redefined.
3. Auxiliary Table Performance Evaluation: From the validated fixed boundary table model, the flatness performance index (PI) can be defined and evaluated by both FEA and experiments. Upon the comparison of PIs between analysis and experiments. The PI of initial design table can be obtained and used as the reference specification.

Fig. 1(b) shows the structural evaluation flow chart of new design. The reference PI of initial design is used to evaluate the new design of auxiliary table. The iterative design process can be observed in Fig. 1(b) at different stages.

3. Case Study of the Design Evaluation

This section presents the design evaluation of a 750×750 type of auxiliary table following the flow chart as shown in Fig. 1(a). Fig. 2 shows the picture of the auxiliary table, and Table 1 shows the physical parameters of the auxiliary table. More detail evaluation procedure and results are discussed as follows.

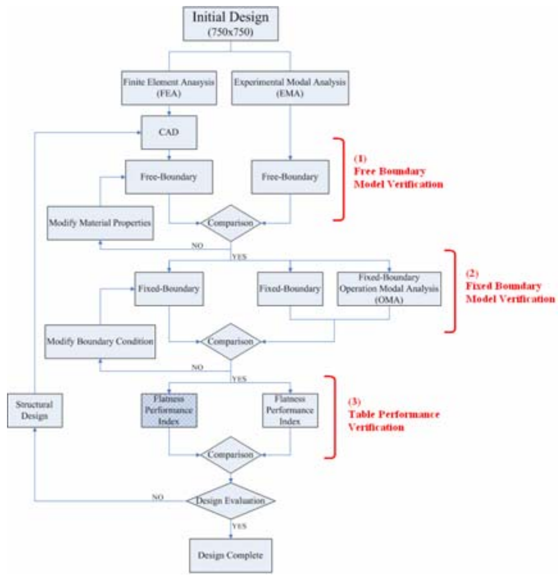
3.1 Free Boundary Model Verification:

The objective of this step is to verify the FE model of auxiliary table in free boundary. Both FEA and EMA are performed, respectively.

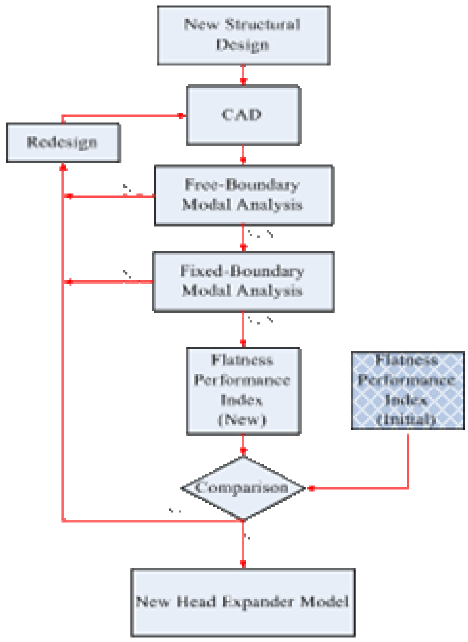
For FEA, the table geometry model is first established by CAD software, INVENTOR, and then transferred to FEA software, ANSYS, by SAT interface. Fig. 3 shows the details of FE model that is constructed by linear hexahedron elements (SOLID 45) without any displacement constraint for free boundary.

Fig. 4 shows the experimental setup, and Fig. 5 reveals the 85 measurement points of the auxiliary table. No. 85 is selected as the fixed point to apply impact force, and only those circled in Fig. 5 are tested with roving accelerometers. Table 2 displays only the first two modes to illustrate the successful verification of FE model in terms of modal parameters. One can see the mode shapes agree well, and the averaged error of natural frequency between analysis and experiments for all modes in 2000 Hz is 4.43%. The maximum is 12.46%, and minimum is -2.95%.

From the reasonable agreement of modal parameters, the FE model without constraints can be well verified, and the optimum material parameters are listed in Table 1.



(a) Initial design evaluation



(b) New design evaluation

Figure 1: Design process flow chart

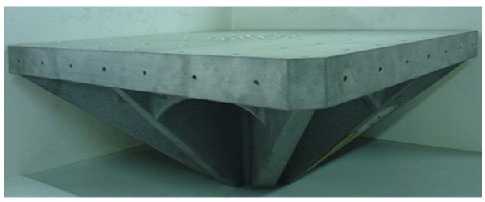


Figure 2: 750 type auxiliary table of vibration testing machine

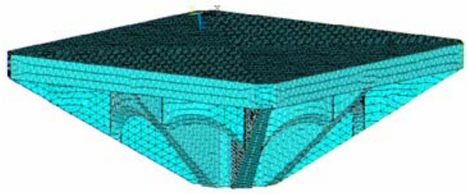


Figure 3: Finite element model for free boundary

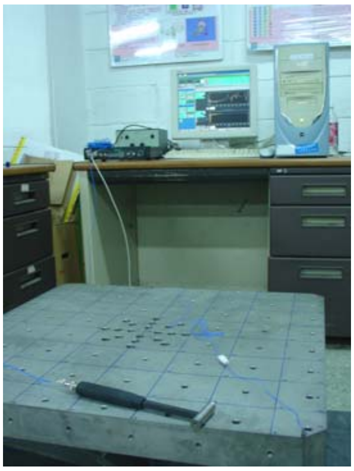


Figure 4: Experimental instrument setup for free boundary

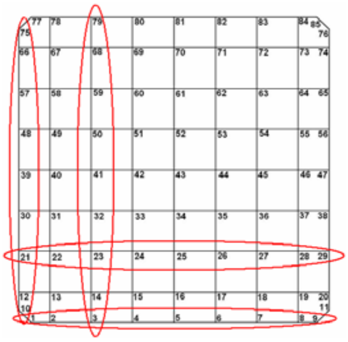
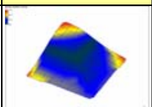
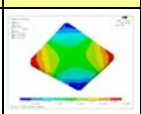
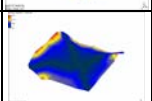
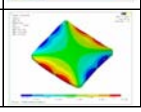


Figure 5: Grid of experimental measurement points

Table 1: Physical parameters of the auxiliary table

length	0.75(m)	Young's modulus	$5.6\times10^{10}(\mathrm{N/m^2})$
width	0.75(m)	density	$2.65(kg/m^3)$
height	0.3(m)	Poisson ratio	0.29

Table 2: Free boundary model verification

EMA			FEA			Diff., (%)
mode	Natural frequency (Hz)	mode shape	mode	Natural frequency (Hz)	mode shape	
E-01	979		F-07	1022		4.29
E-02	1180		F-08	1290		9.32

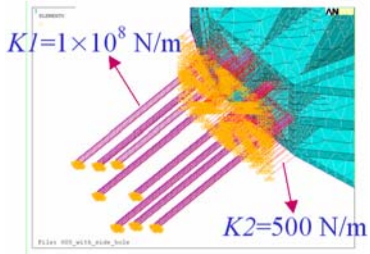
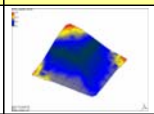
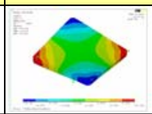
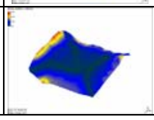
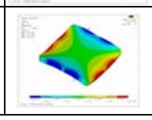


Figure 6: Finite element model for fixed boundary

Figure 7: Experimental setup for fixed boundary and flatness measurement

Table 3: Fixed boundary model verification

EMA			FEA			Diff., (%)
mode	Natural frequency (Hz)	mode shape	mode	Natural frequency (Hz)	mode shape	
E-01	981		F-07	1022		4.18
E-02	1180		F-08	1297		9.92

3.2 Fixed boundary model verification

Once the free boundary model was validated, the table mounted to the coil structure of vibration testing machine can be modeled by spring elements (COMBIN 14) to represent the contact surface between the table and the face of coil structures. These are two types of spring elements as shown in Fig. 6. The longer ones $(K1=10^8\ \mathrm{N/m})$ represent the bolted area contact, while the shorter ones $(K2=500\,\mathrm{N/m})$ are the face contact zone. Both the spring constants are optimized such that the errors of natural frequencies determine from FEA and EMA are minimums.

Fig. 7 shows the experimental setup for the auxiliary table mounted onto the vibration testing machine. The EMA procedure is the same as that for free boundary test. Table 3 shows the comparison results between FEA and EMA. Again, only the first two modes are shown. The averaged error for all of natural frequencies is 10.17%. The maximum is 18.23%, and minimum is -2.09%, within 2000 Hz for all modes. The mode shapes also reveal reasonable agreement. The FE model is considered sufficient and able to be used for response simulation. It is also noted that the spring constants can be valid for only the same testing machine. Otherwise, the spring constants need to be calibrated separately for other machines.

3.3 Auxiliary Table Performance Evaluation

Chen [4] developed a flatness evaluation model to define the performance index (PI) of the auxiliary for the judgment of table quality. Fig. 8 shows the auxiliary table subject to base excitation. When there is the base harmonic displacement input $Z(t)$, the transmissibility block diagram can be shown in Fig. 9. If $Y_i(t)$ is the time domain displacement response at location i, then the frequency domain response can be written:

$$Y_i(f)=Z(f)\cdot\overline{TR}_i(f) \tag{1}$$

where $Y_i(f)$ and $Z(f)$ are Fourier spectra of $y_i(t)$ and $z(t)$, and $\overline{TR}_i(f)$ is the transmissibility of the table .

In considering the comparison with the experimental data, i.e. acceleration response actually, the acceleration spectrum $A_i(f)$ and the acceleration transmissibility can be obtained as follows:

$$A_i(f)=Y_i(f)\cdot(2\pi f)^2 \tag{2}$$

$$TR_i(f)=\overline{TR}_i(f)\cdot(2\pi f)^2 \tag{3}$$

From Eq. (1), one can get:

$$A_i(f)=Z(f)\cdot TR_i(f) \tag{4}$$

Here, $A_i(f)$ represents the surface response of the auxiliary table without feedback control. In practice, the vibration testing machine is equipped with controller that can precisely control the surface response at some control sensor location, i.e. i_{cs}, exactly the same as the specified acceleration response. Let A_{input} be the acceleration level of white noise to be specified, and therefore, the acceleration level at control sensor location will be the same and expressed as follows:

$$\overline{A}_{i_{cs}}=A_{input} \tag{5}$$

Referred to the block diagram in Fig. 10, the relation holds as follows:

$$Z_{i_{cs}}(f)\cdot TR_{i_{cs}}(f)=A_{input} \tag{6}$$

Therefore,

$$Z_{i_{cs}}(f)=\frac{A_{input}}{TR_{i_{cs}}(f)} \tag{7}$$

Let $Z_{i_{cs}}(f)$ replace $Z(f)$ in Fig. 10, and then the real acceleration response at any location i of the surface can be obtained:

$$\overline{A}_i(f)=Z_{i_{cs}}(f)\cdot TR_i(f) \tag{8}$$

Let k be the index of frequency resolution, the flatness of the test surface $\varepsilon_i(f_k)$ can be defined:

$$\varepsilon_i(f_k)=\frac{\overline{A}_i(f_k)-\overline{A}_{i_{cs}}(f_k)}{\overline{A}_{i_{cs}}(f_k)}=\frac{\overline{A}_i(f_k)-A_{input}}{A_{input}} \tag{9}$$

The physical measuring of $\varepsilon_i(f_k)$ is the difference of acceleration level between location i and i_{cs} (the control sensor location) at frequency f_k. Assume that there are N_s measurement points on the test surface and N_f spectral lines, i.e. i= 1, 2, ..., N_s, and k= 1, 2, ..., N_f, respectively.

Several indices can be defined as follows:

$$\varepsilon_{i,avg}^f=\frac{\sum_{k=1}^{N_f}\varepsilon_i(f_k)}{N_f}=\frac{\varepsilon_i^f}{N_f} \tag{10}$$

where

$$\varepsilon_i^f=\sum_{k=1}^{N_f}\varepsilon_i(f_k) \tag{11}$$

$$\varepsilon_{\max}=\max[\varepsilon_i^f] \tag{12}$$

$$\varepsilon_{\min}=\min[\varepsilon_i^f] \tag{13}$$

$$\varepsilon_{avg}(f_k)=\frac{\sum_{k=1}^{N_s}\varepsilon_i(f_k)}{N_s}=\frac{\varepsilon(f_k)}{N_s} \tag{14}$$

where

$$\varepsilon(f_k)=\sum_{i=1}^{N_s}\varepsilon_i(f_k) \tag{15}$$

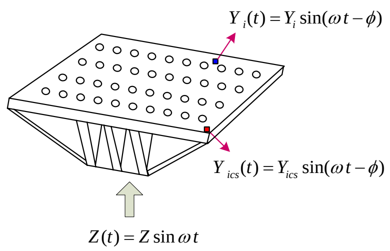


Figure 8: Auxiliary table subject to base excitation

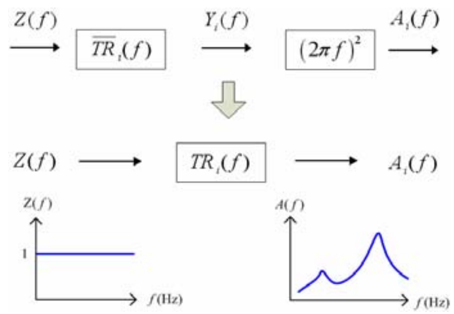


Figure 9: Transmissibility block diagram

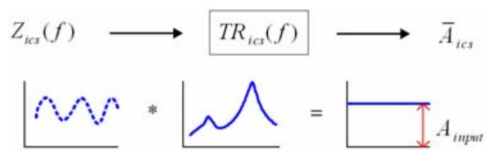


Figure 10: Determination of excitation spectrum $Z_{i_{cs}}(f)$

Finally, more compact PIs for the auxiliary table can be defined as follows:

$$\mathrm{PI}_{avg} = \varepsilon_{avg,i_{cs}} = \frac{\sum_{k=1}^{N_f}\sum_{i=1}^{N_s}[\varepsilon_i(f_k)]}{N_s \cdot N_f} \tag{16}$$

$$\mathrm{PI}_{diff} = \left|\varepsilon_{\max} - \varepsilon_{\min}\right| \tag{17}$$

For a control sensor location i_{cs}, PI_{avg} as shown in Eq. (16) represents the flatness of the test surface of the auxiliary table as well as PI_{diff} in Eq. (17).

Fig. 7 is the experimental setup for flatness measurement, while Fig. 11 shows the sensor connection to the controller (Dactron) and FFT analyzer (SigLab). The controller can perform feed back control to ensure the specified white noise response at the control sensor location. The FFT analyzer is used to record the transmissibility to further determine the PIs. Table 4 lists the experimental instrument.

Table 5 shows the PIs over the test surface obtained from experiments and FEA for the control sensor at location $i_{cs} = 85$, i.e. the corner of the test surface . One can observe the flatness surfaces and PI value's from experimental and FEA are comparable.

Table 6 reveals the overall flatness distribution over the test surface according to different control sensor locations. For those contour lines with $\mathrm{PI}_{avg} = 0$ can be the best choice of control sensor locations as indicated by arrows in Table 6.

Up to now, the complete design evaluation and experimental verification are shown. The next stage issue will be how to design a new auxiliary table out–performance over the current one.

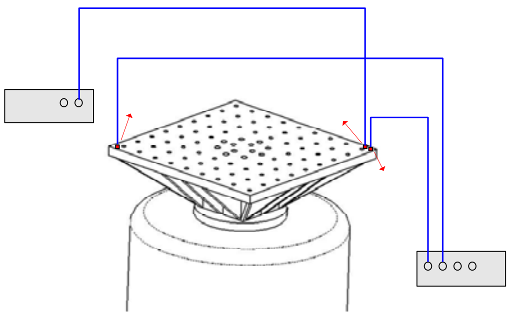


Figure 11: Sensor connection for flatness measurement

Table 4: Experimental instrument

Instrument	Type
Electromagnetic vibration testing machine	KD-9363EM-600F2K-50N120
Control instrument	Dactron
Spectrum analyzer	SigLab model 20-42
Accelerometer	Kistler Type:8732A500
Modal parameter extraction software	ME\`scopeVES

Table 5: PIs from experiment and FEA for the control sensor location $i_{cs} = 85$

Experiment				FEA			
PI_{avg}	PI_{diff}	$\varepsilon_{\max}$	$\varepsilon_{\min}$	PI_{avg}	PI_{diff}	$\varepsilon_{\max}$	$\varepsilon_{\min}$
-23.42	80.39	26.09	-54.29	-44.83	48.53	0.13	-48.4

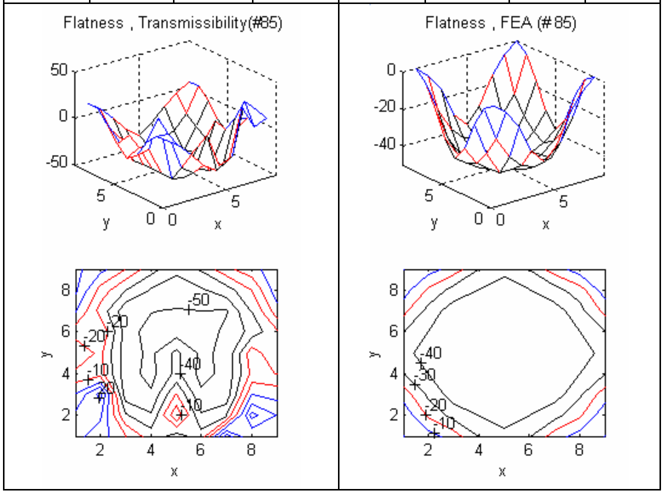
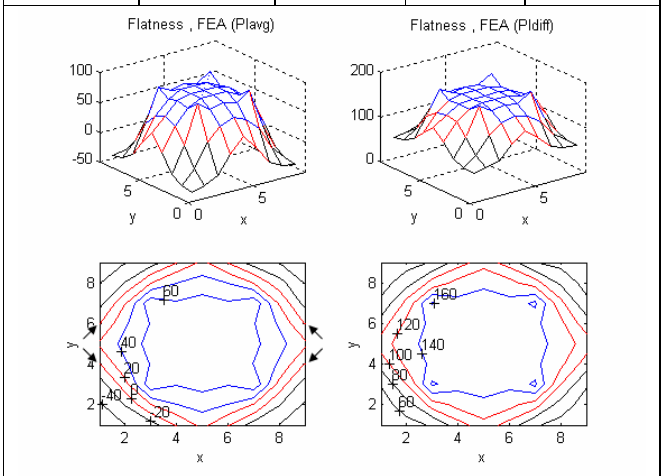


Table 6: Overall flatness distribution over the test surface according to different control sensor locations

PI_{avg}				
avg	max	min	std	rms
20.16	76.25	-44.91	41.71	46.09

PI_{diff}				
avg	max	min	std	rms
107.89	164.16	48.45	38.22	114.38



4. Discussions on New Design of Auxiliary Table

The auxiliary table can be required for different sizes of test surface and so forth the height, thickness, rib shape and etc. should be properly designed to ensure the proper performance in vibration testing. The optimization problem can be formulated and verbly stated as follows:

1. Objective Function: This work suggests choosing the distribution of PI_{avg} as shown in

Eq. (16) and revealed in Table 6 as the objective function to be as flat as possible. Consequently, the PI_{avg} and PI_{diff} surfaces shown in Table 6 are with the smallest values and the most flatness.

2. Design Variables: There are two phases of new design consideration. Phase I: the geometry design is focused on new shape or different layout of ribs for example. Phase II: the dimension optimization for the selected geometry, such as the height or thickness.
3. Constraints: For reducing the weight of auxiliary table, the new design should be as light as possible. The new design must be able to fabricated as well as suitable to fit the coil structures.

5. Conclusions

This work addresses the design approach for the auxiliary table. Both FEA and EMA techniques are employed to conduct model verification of the table in free and fixed boundary conditions, respectively and therefore the mathematical model or FE model can be validated and used for response prediction. The flatness evaluation of the auxiliary table is established and characterized by several PIs that can be referred as the design criteria to develop new types of auxiliary tables. A 750×750 table case study is presented to illustrate the design process. A general requirement of new design is also briefly discussed. The developed methodology can not only provide a systematic approach for auxiliary table design, but also largely cut down the development effort and time as well as the cost.

6. Acknowledgment

This work is financially supported by National Science Council, Taiwan and King-Design Company under the contract No.: NSC-95-2622-E-020-004-CC3.

7. Reference

[1]Wang, B. T., and Chen, C. C., Model Verification of Vertical Auxiliary Table for Vibration Testing Machine, Proceeding of the 21st National Conference on Mechanical Engineering The Chinese Society of Mechanical Engineers, Kaohsiung, Taiwan, pp.2431-2436, 2004. (in Chinese)

[2]Wang, B. T., and Chen, C. C., Model Verification of Vertical Auxiliary Table in Mounted Condition, 2004 AASRC/CCAS Joint Conference, Taichung, Taiwan, No.A6-1, 2004. (in Chinese)

[3] Wang, B. T., Lee, P. W., and Chen, Y. L., Model Updating for Vertical Auxiliary Table of Vibration Testing Machine, 2005 Taiwan ANSYS Users Conference, Hualien, Taiwan, pp. 4-27~4-34, 2005. (in Chinese)

[4]Chen, C. C., Design Verification and Validation of Head Expander for Vibration Testing Machine, Master Thesis, Department of Mechanical Engineering, National Pingtung University of Science and Technology,2004. (in Chinese)

[5]Wang, B. T., Zhuang, F. R., and Lee, D., Analysis and Verification of Flatness Performance Index for Rib-Reinforced Head Expander, Paper Summary of the 23rd National Conference on Mechanical Engineering The Chinese Society of Mechanical Engineers, Tainan, Taiwan, No.C3-028, 2006. (in Chinese)

[6]Wang, B. T., Chen, K. C., and Lee, D., Model Verification of the Carriage for Free-Fall Shock Testing Machine, Proceeding of the 22nd National Conference on Mechanical Engineering The Chinese Society of Mechanical Engineers, Taoyuan, Taiwan, No.C3-003, 2005. (in Chinese)

[7]Wang, B. T., Lin, C. H., and Lee, D., Experimental Modal Analysis and Model Verification of Voice-Coil Structure of Shaker, Proceeding of the 20th National Conference on Mechanical Engineering The Chinese Society of Mechanical Engineers, Taipei, Taiwan, pp.529-536, 2004. (in Chinese)

[8]Feldmaier, D. A., Sung, S. H., Nefske, D. J., and Doggett, S. J., Modal Analysis Tests for Correlating an Automobile Rear Suspension Model, Proceeding of the 22th International Modal Analysis Conference, Dearborn, Michigan, Paper No. s08p03, 2004.

[9]Pavic, A., Widjaja, T., and Reynolds, P. , The Use of Modal Testing and FE model Updating to Investigate Vibration Transmission Between Two Nominally Identical Building Floors, International Conference on Structure Dynamics Modeling – Test, Analysis, Correlation and Validation , Nadeira Island, Portugal, pp.347-355, 2002.

[10]Wang, G., and Li, L., Finite Element Analysis and Experimental Research on the Reduction of Vibration and Structure Noise in Ship, Proceeding of the 8th International Congress on Sound and Vibration, Hong Kong, China, pp.1373-1380, 2001.

振動試驗機輔助平台設計流程之發展

王栢村[1]、莊豐榮[2]、李昆達[3]
[1]國立屏東科技大學機械工程學系教授
[2]國立屏東科技大學機械工程學系研究生
[3]金頓科技股份有限公司 研發處處長
國科會計畫編號：
NSC-95-2622-E-020-004-CC3

摘要

本文旨在探討振動試驗機之輔助平台設計流程，舉一實際平台範例說明設計方法及其驗證流程，並探討新平台設計之概念構想。透過實驗與理論模態分析驗證有限元素模型為等效之分析模型。將實驗及理論分析之頻率響應函數代入響應評估程式，獲得實際及預測之響應，以比較驗證理論分析所預測平坦度之正確性。藉此，設計新型式平台並套入此平坦度預測程式，以獲得更優的平台設計。本文建立平台之設計分析與響應預測方法，可做為未來新型式平台之設計分析之依據。

關鍵詞：輔助平台，振動測試，有限元素分析，實驗模態分析