SYSTEM IDENTIFICATION OF A SMALL SIZE MACHINE

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ABSTRACT

A bottom-up system identification methodology was adopted to specify the performance of a small size machine – the µ4. Analyses of the modal properties of a motion axes module were undertaken. Mass line and auto modal assurance criterion were used to assure proper setup and sufficient measurement points were used. Based on animation movies the main characteristics of each mode were identified. Comparison and correlation between measured and simulated modes was carried out. The measured and simulated modal properties of the motion axes module were shown to have a good correlation with less than 15% discrepancy. The results of the modal measurements and simulation will be used to improve the mechanical and control design for a high-dynamic motion control goal.

Keywords: system identification, FEM, modal measurements.

1 INTRODUCTION

The increasing demand within electro-optics and information technology industries for ultra-precise micro-mechanical components sets new demands for improved small sized production machines. Numerous research efforts to develop small so-called "desk top" size machines have been undertaken in the last two decades. However, most of these machines are still at the development stage and not been widely commercialised, moreover their application to high accuracy and fine surface quality is constrained by their low dynamic stiffness. The Integ-µ4 machine was conceived in 2008 (Shore et al., 2013) by Cranfield University Precision Engineering Institute. The machine specification was aimed to be around the leading capability of diamond turning and micro-milling machines and with the ability to automatically shift from turning to milling operations seamlessly. The overall size of the machine was set as a European scale washing machine in the range of 0.6× 0.6×1m volume. In order to reduce assembly costs the machine design was based on common modules with simple interfaces. The µ4 motion axes were split into two near identical modules. Thus, each one of the modules (Figure 1) can be used as a test rig to identify and specify the current design and to validate the improvements to the design. The research goal is to achieve high-dynamic motional control of “desktop” sized machine tool required for free-form manufacturing (Fang et al., 2013).

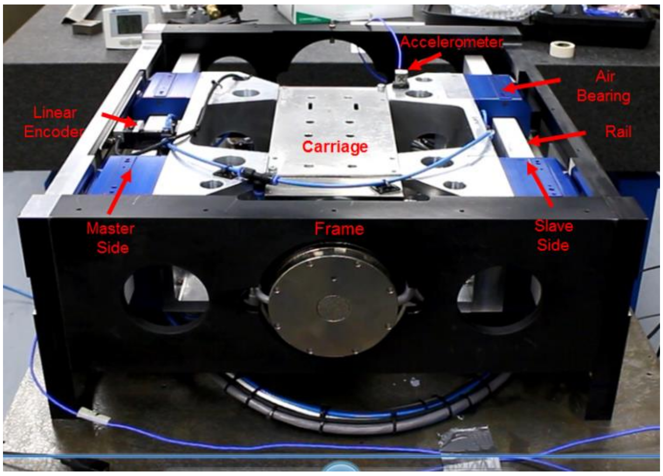

Figure 1: Motion module

2 SYSTEM IDENTIFICATION METHODOLOGY

The system identification methodology is a bottom-up process in which the lowest level components are tested and simulated first, then used to facilitate the testing of higher level components. The motion module bottom-up identification can be divided into four stages (Figure 2): stage A the frame components, stage B the frame assembly, stage C the frame assembly with linear rails and stage D the full assembly of the module. The modal properties of one of the motion axes module were simulated using Finite Element Method (FEM), measured and analysed using modal measurement equipment specified and procured for this research. Each component and assembly was supported in free-free conditions and frequency response functions measured using hammer excitation. In the free-free supported boundary condition rigid body modes and flexible modes are sufficiently separated enabling identification of the rigid body modes for debugging the measurement setup. The rigid and flexible body modes were synthesised using stabilisation diagrams choosing the frequency and damping values. The results were validated by observing the animation of the modes and by comparing the properties to those of a FEM.

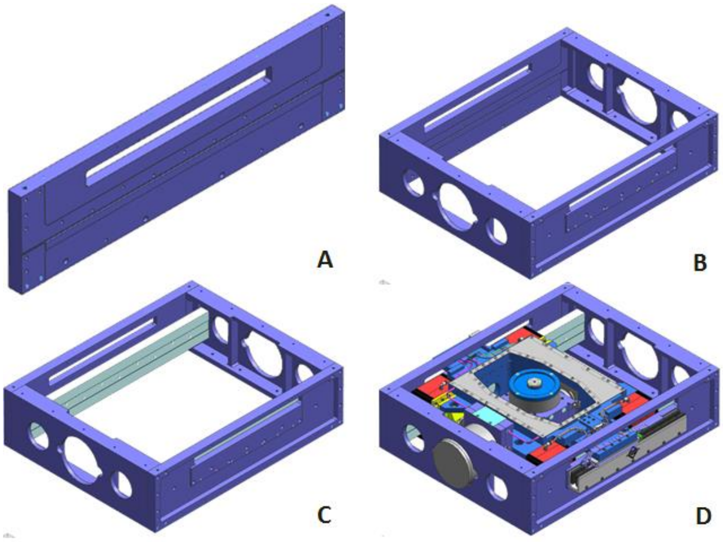

Figure 2: System identification methodology

3 FEM ANALYSIS

The FEM analysis was based on the system identification bottom-up methodology in which lowest level components were simulated first and then a higher lever components and assembly. The simulation followed the free-free boundary condition thus; no boundary conditions were applied to the simulation which caused six eigenvalues with values close to zero. These eigenvalues corresponds to the six rigid body Degree Of Freedom (DOF). Modelling an assembly requires that the surface to surface interactions between the parts need to be specified. Using the fact that the contact surfaces were diamond machined, the contact surfaces were simulated as glued surface to surface.

Table 1 shows the list of the first 12 modes for two levels of assembly – frame only and frame with rails which correspond to Figure 2 B and C respectively. The effect of adding the linear rails to the frame assembly can be analysed by comparing the mode shapes and frequency. The red arrows in Table 1 represent the change of identified mode shapes between each level. As expected, adding the rails increases bending stiffness which affects dramatically up to the 4th mode. The added rails cause extra mode shapes, i.e. 606Hz, due to the added extra mass and DOF.

Mode number	Frame only [Hz]	Frame with rails [Hz]
1	109	116
2	117	158
3	215	387
4	355	486
5	436	488
6	479	543
7	533	606
8	638	650
9	647	659
10	712	739
11	748	749
12	837	763

Table 1: List of modes

4 MODAL MEASUREMENTS

The components and assemblies were supported on a floating table and a bubble wrap mattress for a free-free boundary condition (Fu and He, 2001). This method will allow operating one of the motion modules while measuring, which is more practically than using bungee cables suspension. In order to measure free body modes 2 excitations and 6 responses are needed, however practical tests show (Leurs et al., 1997; Madjlesi et al., 2005) that best results are obtained with at least 6 excitation (e.g., 2 nodes in 3 directions) which have been used. An impact hammer was used for the excitation and three axis accelerometers were used in various locations for measuring the response. The accelerometers locations were chosen based on the mode shapes results from the FEM. The measurement product is the accellerance Frequency Response Function (FRF) which is then post processed by parameter estimation techniques (curve fitting) to identify the modal parameters: frequency, damping and mode shape.

The measurements were assessed based on the mass line value and Modal Assurance Criterion (MAC) to assure the measurements quality (Allemang, 2003).

4.1 Mass line

For a single DOF model, i.e. rigid body mode, the structure vibrates as a rigid body at the resonance frequency. If the rigid body modes and flexible modes are sufficiently spaced, the accellerance is approximately equal to the asymptote called the mass line which equals to m^{-1}(Fu and He, 2001). In case of error in measurement data or in adequate suspension setup the rigid and flexible modes wouldn't be sufficiently spaced and the mass line wouldn't equal the mass of the measured components. An example of the FRF (Figure 3 A) of frame with rails measurements and the evaluated mass property (Figure 3 B). As can be seen the evaluated mass property is within 10% of the measurements error.

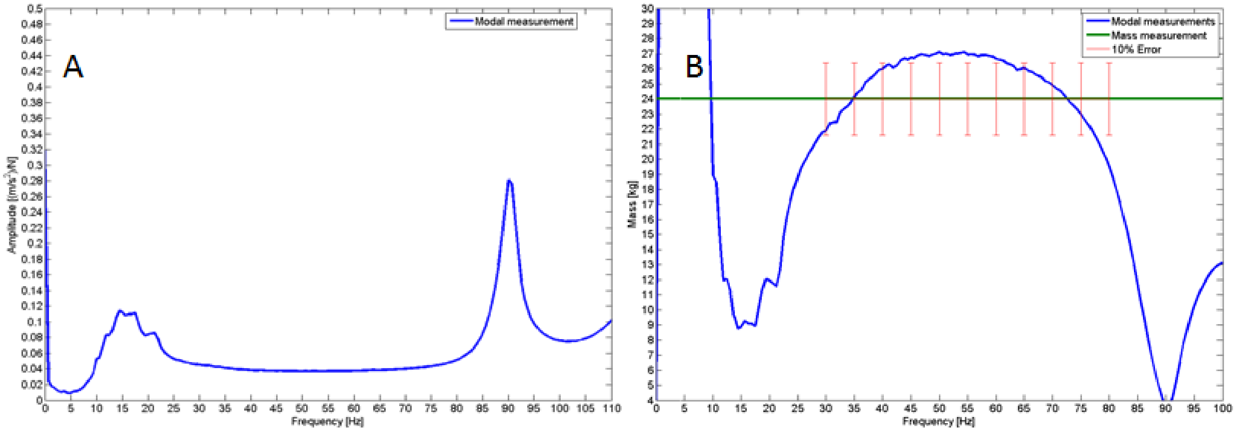

Figure 3: Mass line of frame with rails assembly

4.2 Stabilisation diagram

The stabilisation diagram is used as part of the modal parameter estimation, i.e. frequency and damping properties. The stabilisation diagram is often plotted based on the Least Squares Complex Exponential (LSCE) and Least Squares Complex Frequency (LSCF) curve fitting algorithms (Cauberghe et al., 2004; Peeters et al., 2004). The user selects the model order and the bandwidth of the fitting. Hence, the user selects only the real modes based on the quality of the fitting from the diagram (Figure 4).

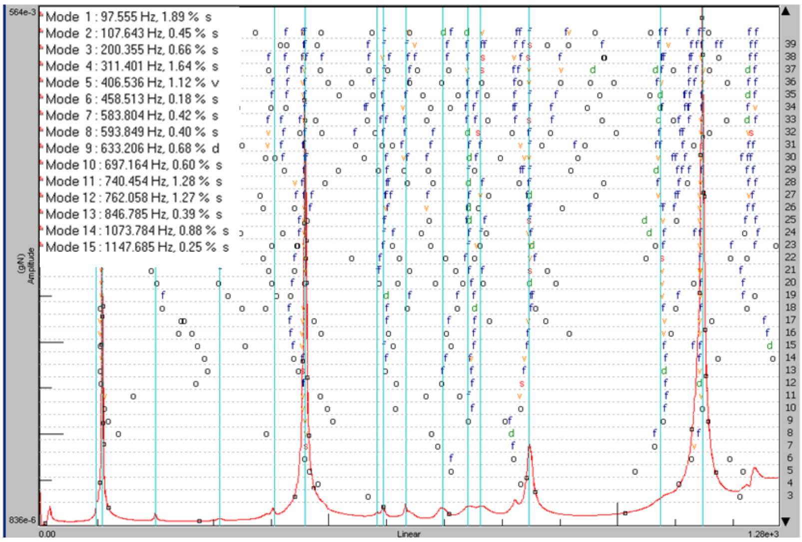

Figure 4: Stabilisation diagram of frame only measurement

4.3 Auto MAC

The MAC is a mathematical tool used to compare two vectors of equal length, i.e. mode shape. If a linear relationship exists between two modes the MAC value will be near one. The Auto MAC tool can be used to assess whether sufficient measurements points have been used by calculating the correlation between measured modes. Insufficient measurement point will cause a correlation between different modes. In example, Figure 5 shows the AMAC of the frame with rail measurements. In this measurement only three rigid body modes and twenty flexible modes have identified. The synthesised flexible modes are not correlated which assures the use of the measurement data.

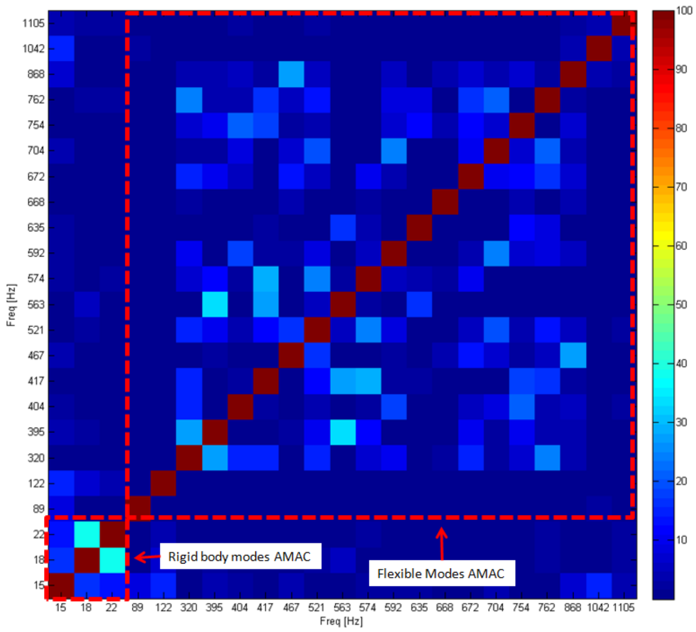

Figure 5: AMAC of frame with rails measurement

5 FEM AND MEASUREMENTS CORRELATION AND COMPARISON

The estimated modal properties – frequency and mode shapes based on the measurements were compared and correlated. An animation movie was made for each mode which shows the mode shape of the assembly. Each measured mode was identified by comparing the animation to the FEM animations. This process was made by focusing on the main characteristics of the mode shape i.e., bending, twisting, symmetric and anti symmetric of the assembly. As an example, the main characteristics of two modes are introduced, the first order bending of the upper beams in symmetric or anti-symmetric shapes, as seen in Figure **6** A and B respectively. These FEM modes at 639Hz and 647Hz were identified as measured modes at 584Hz and 594Hz respectively

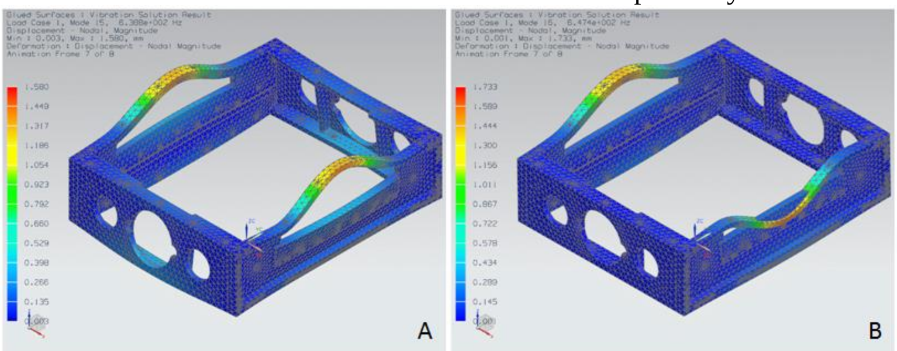

Figure 6: Two mode shape at 639Hz (A) and 647Hz (B) of frame only assembly

After finding the corresponding modes between FEM and measurements, a correlation plot was used to assess the discrepancy between the measurements and the simulation frequency values (Figure 7). The discrepancy increased as the complexity of the analysis raises, i.e. 6% for frame only and 15% for frame and rails. However, these values are low when taking into consideration the measurements and modelling errors. As expected the measured frequencies were always lower than those simulated since the simulation lacks the damping properties of the surface to surface contacts. For each measurement several modes were missing due to their low amplitude being below the sensor noise level.

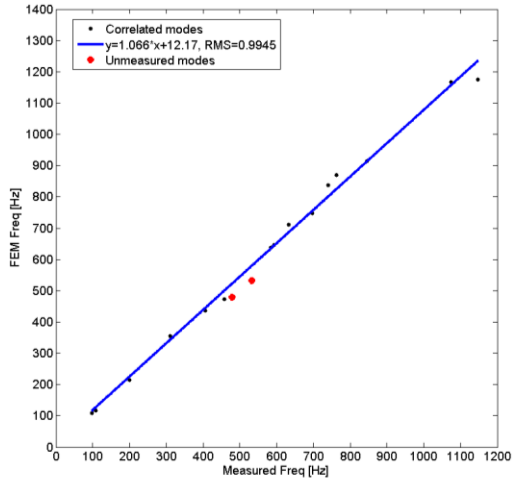

Figure 7: FEM and measurements correlation

6 CONCLUSIONS

A bottom-up system identification methodology was adopted to investigate a novel small size machine – µ4. The system identification was based on FEM and measurements analysis and showed a good correlation. Based on this methodology, the modal results will be used as an input to improve the mechanical design and help implement control technologies to achieve high dynamic motional control goals. Furthermore, the FEM-measurements correlation can now be used to simulate the performance of the improved motion axes module. Thus, it can reduce the cost and time of design iterations.

The next phase of the research will be focused on investigating which technology and techniques should be used in order to achieve high-dynamic motional control. Several solutions are considered – innovative mechanical design, neural network based on low-cost measurement components, or a feed forward control design.

REFERENCES

Allemang, R.J. (2003) 'The modal assurance criterion–twenty years of use and abuse', Sound and Vibration, 37(8), pp. 14-23.

Cauberghe, B., Guillaume, P., Verboven, P., Parloo, E. and Vanlanduit, S. (2004) 'The secret behind clear stabilization diagrams: The influence of the parameter constraint on the stability of the poles', Proceedings of the 10th SEM international congress & exposition on experimental and applied mechanics. pp. 7-10.

Fang, F., Zhang, X., Weckenmann, A., Zhang, G. and Evans, C. (2013) 'Manufacturing and measurement of freeform optics', CIRP Annals-Manufacturing Technology, 62(2), pp. 823-846.

Fu, Z. andHe, J. (2001) Modal analysis. Butterworth-Heinemann.

Leurs, W., Gielen, L., Brughmans, M. and Dierckx, B. (1997) 'Calculation of rigid body properties from FRF data: Practical implementation and test case', Proceedings of the IMAC XV, Tokyo, Japan, pp. 1-7.

Madjlesi, R., Khajepour, A., Ismail, F., Wybenga, M., Rice, B. and Mihalic, J. (2005) 'Optimisation of engine mounting systems using experimental FRF vehicle model', International Journal of Vehicle Noise and Vibration, 1(3), pp. 232-250.

Peeters, B., Van der Auweraer, H., Guillaume, P. and Leuridan, J. (2004) 'The PolyMAX frequency-domain method: A new standard for modal parameter estimation?', Shock and Vibration, 11(3-4), pp. 395-409.

Shore, P., Morantz, P., Nicholls, J., Walker, D., Stephenson, D., Allen, D. and O'Neill, W. (2013) Integrate knowledge centre in ultra precision and structured surfaces . EPSRC.