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Christopher Niezrecki · Javad Baqersad
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Rotating Machinery, Optical Methods & Scanning LDV Methods, Volume 6

Proceedings of the 37th IMAC, A Conference and Exposition on Structural Dynamics 2019

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Editors
Christopher Niezrecki
Department Mechanical Engineering
University of Massachusetts Lowell
Lowell, MA, USA

Javad Baqersad
Kettering University
Flint, MI, USA

Dario Di Maio
Department of Mechanical Engineering
University of Bristol, Queens Building
Bristol, UK

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Preface

Rotating Machinery, Optical Methods & Scanning LDV Methods represents one of the eight volumes of technical papers presented at the 37th IMAC, A Conference and Exposition on Structural Dynamics, organized by the Society for Experimental Mechanics, and held in Orlando, Florida, January 28–31, 2019. The full proceedings also include volumes on Nonlinear Structures and Systems; Dynamics of Civil Structures; Model Validation and Uncertainty Quantification; Dynamics of Coupled Structures; Special Topics in Structural Dynamics & Experimental Techniques; Sensors and Instrumentation, Aircraft/Aerospace, Energy Harvesting & Dynamic Environments Testing; and Topics in Modal Analysis & Testing.

Each collection presents early findings from experimental and computational investigations on an important area within structural dynamics.

The organizers would like to thank the authors, presenters, session organizers, and session chairs for their participation in this track.

Lowell, MA, USA — Christopher Niezrecki
Flint, MI, USA — Javad Baqersad
Bristol, UK — Dario Di Maio

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Chapter 1
Detection of Sources of Nonlinearity in Multiple Bolted Joints by Use of Laser Vibrometer

Arnaldo delli Carri, Sante Campanelli, and Dario Di Maio

Abstract The use of non-contact measurement methods for detecting and locating sources of nonlinearities can be potentially a break-through in the nowadays experimental modal analysis. The primary goal is to define more effective test strategies, whereby contact sensors will measure the nonlinear vibration responses at the best location possible. Jointed structures are a typical example where a large number of the joint can pose the question of where what and how to measure the nonlinear response. Upon the identification of one, or more, nonlinear response mode the objective is to determine where is the source of such nonlinear vibration. Nonlinearity can be characterised when its source is well defined and can be adequately tested. This paper will attempt to detect and locate the source of nonlinearity from a multi-beam jointed assembly. The approach will be carried out by using both contact and non-contact measurement methods, the results of which will be compared and evaluated. The operator to detect the source of nonlinearity will be the coherence function applied to random response data.

Keywords SLDV · Bolted joints · Nonlinear vibration testing

1.1 Introduction

This paper attempts to exploit the potential of the scanning LDV system to measure vibration nonlinear response at a much greater number of locations, than it can be done by using setup based on contact sensors. The objective is to use such denser measurement grid to identify one, or more, sources of nonlinearity. It is not intended to replace the accelerometer for the characterization and quantification of a source of nonlinearity. The localization of nonlinearities is fundamental when such sources are discrete as for bolted structures. It is notorious that structures with a high number of interfaces will exhibit nonlinear responses when subjected to high amplitude of excitation forces. From a model validation viewpoint, the accurate localization, characterization and quantification of the nonlinearity can make the modelling work more effective and time efficient. Resources can be dedicated to improving the model where it is needed by inclusion of the nonlinear physics.

The localization of nonlinearities was usefully carried out in [1]. It was demonstrated that by setting up a good number of accelerometers on a structure the localization can be done with a good level of accuracy. However, it looked clear that such a method of localization depends on the setup of the contact sensors, and when an engineering judgment is not based on the identification of nonlinearity it might be possible that nonlinearities stay hidden to the sensors. Following the methodology applied to locate the source of nonlinearity (explained in the following sections), it become interesting to use a different technology which would enable much denser measurement grind than offered by contact sensors. Three research works addressed the topic of localization by using the scanning laser vibrometer. One research was focussed on bolted flanges and the use of continuous scanning methods to identify the source of nonlinearity. That approach showed that by mapping the response phase of the deflection shape measured at constant frequency and several level of amplitudes one could determine the level of nonlinearity exhibited by a vibration mode [2]. The scanning laser vibrometer was used in its step-scanning mode in two attempts, where high resolution mode shapes were measured to determine which mode would exert more nonlinear

A. delli Carri
School of Mechanical, Aerospace and Automotive Engineering, Coventry University, Coventry, UK

S. Campanelli
Department of Mechanical Engineering, Universita' Politecnica delle Marche, Ancona, Italy

D. Di Maio (✉)
Department of Mechanical Engineering, University of Bristol, Queens Building, Bristol, UK
e-mail: aeddm@bristol.ac.uk

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response [3]. The outcome of that research showed that some mode shapes create high strain distribution at the flanges' region and thus enabling interface nonlinear conditions. A more recently publication was focussed on use the stepped scanning method to measure the response phase of the deflection shapes of an aero-engine casing assembly onto which accessories were mounted to generate sources of nonlinearities [4]. The research showed that response phase of the deflection shape was a good indicator of the source of nonlinearity.

The present paper aims (i) at starting from those earlier success to detect the nonlinearities of a structure, and (ii) at combining the method of localization based on accelerometers with the use of a scanning laser vibrometer. A novel structure will be used to explore these new developments. The structure is made of ten blocks each of which bolted by two bolts to form a single prismatic beam. The source of nonlinearity can be moved by opening one or more pairs of bolts. The next section will explain in more details the tests structure and setup. The paper will proceed by attempting the same localization method using a set of four reference accelerometers and the measurement made by a scanning laser vibrometer over a grid of 30 points.

1.2 Test Structure, Setup and Experimental Method

The test structure was designed with the idea to repeat the same basic unit ten times. Figure 1.1 shows the basic unit which can be bolted by two pairs of M8 bolts. Figure 1.2 shows both the solid mode and the real unit made of mild steel. The full assembly of the ten units is presented in Fig. 1.3. The 18 bolts were tightened up to 20 Nm to assure full clamping conditions.

The beam was suspended by strings from a frame and a shaker was installed at the bottom of the assembly, as showed in Fig. 1.4a. Thirty measurement points were marked on the beam where the laser beam would measure the vibrations. Four reference accelerometers were also included in the measurement setup, one of which at the drive point location. Figure 1.4b shows the measurement setup.

A custom-made control LabView panel was used to drive the laser beam onto the measurement points and to acquire the vibration response. It was decided to avoid the use the Polytec control panel to give full accessibility to the generation of the excitation signal. Four random noise signals, with 1M samples at 10 kHz sampling frequency, were generated at four different amplitudes. These four signals were stored and used for measuring the 30 LDV measurement points and the four accelerometers. This approach was decided to allow the excitation signal from the generator to be always the same, avoiding

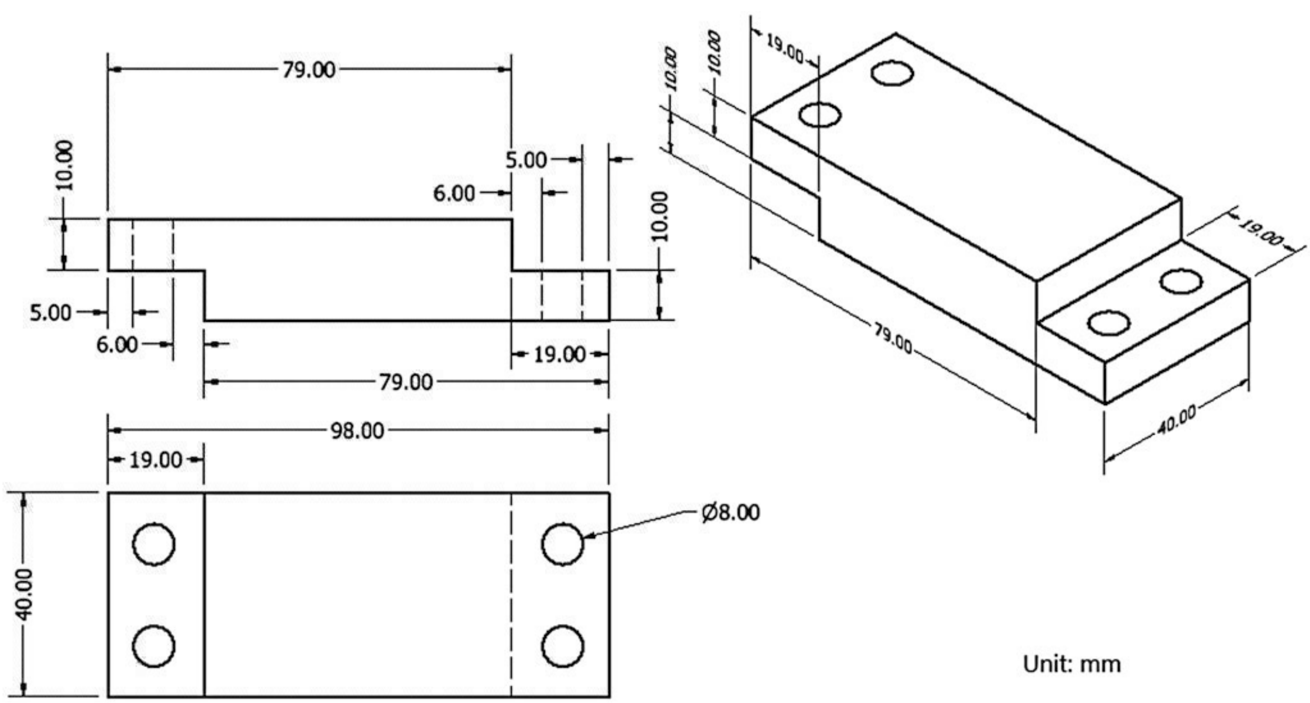

Fig. 1.1 CAD drawing of the basic unit

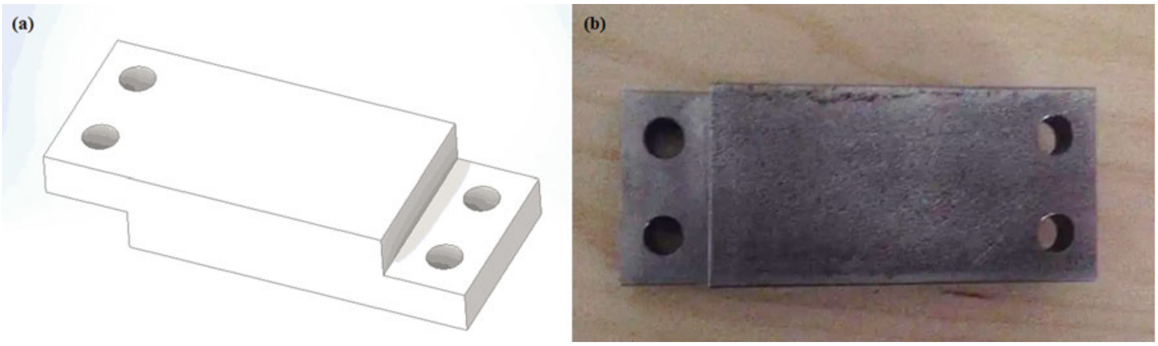

Fig. 1.2 Solid model in (**a**) and real unit in (**b**)

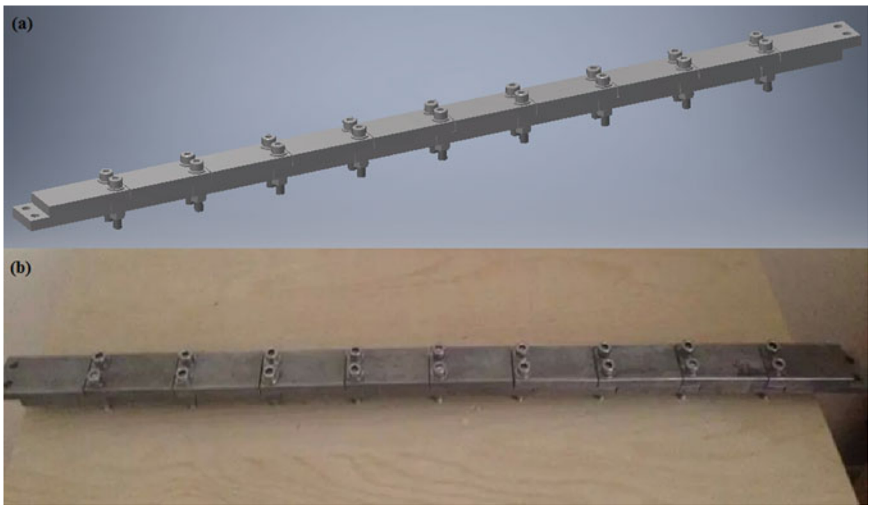

Fig. 1.3 Solid model in (**a**) and real assembly in (**b**)

variations if it were to be generated every time. The four amplitudes were identified as follows, 18 mV, 180 mV, 1080 mV and 1800 mV, respectively. The gain of the amplifier was fixed at one level and never changed.

The test programme was designed as such. The first trial was focussed on assuring the complete linearity of the structure for the 20 Nm torque applied to the 18 bolts. Hence, levels 18 mV, 180 mV and 1800 mV were attempted. The second trial, labelled as configuration A, was carried out by reducing the torque from 20 Nm, to 13 Nm and 8 Nm of the pair of bolts, the fifth from the top. The third trial, labelled configuration B, was carried out using the same torque levels reducing the pair of bolts, the third from the top, and by resetting the fifth pair to 20 Nm. A total of three configuration were tried.

1.3 Theoretical Method

The analysis was performed using conditioned spectral techniques from [5–8] and already applied in an earlier form in [1, 9]. The general equation of motion for a n-DOFs nonlinear system is

$$\boldsymbol{M}\ddot{x}(t) + \boldsymbol{C}\dot{x}(t) + \boldsymbol{K}x(t) + \sum_{i=1}^{M}\left[q_i \cdot g_i\Big(w_i x(t),\, w_i \dot{x}(t)\Big)\right] = f(t)$$

Where $\boldsymbol{M}$, $\boldsymbol{K}$, $\boldsymbol{C}$ are the $n \times n$ mass, stiffness and damping matrices, $x(t)$ and its derivatives are the $nx1$ displacements, velocities and accelerations vectors, $f(t)$ is the $nx1$ forcing vector.

In addition to the usual linear terms, there are M nonlinear (vectorial) terms that contribute to the system: under the summation operator one can discriminate the scaling factor q_i that quantifies the strength of the nonlinearity with respect

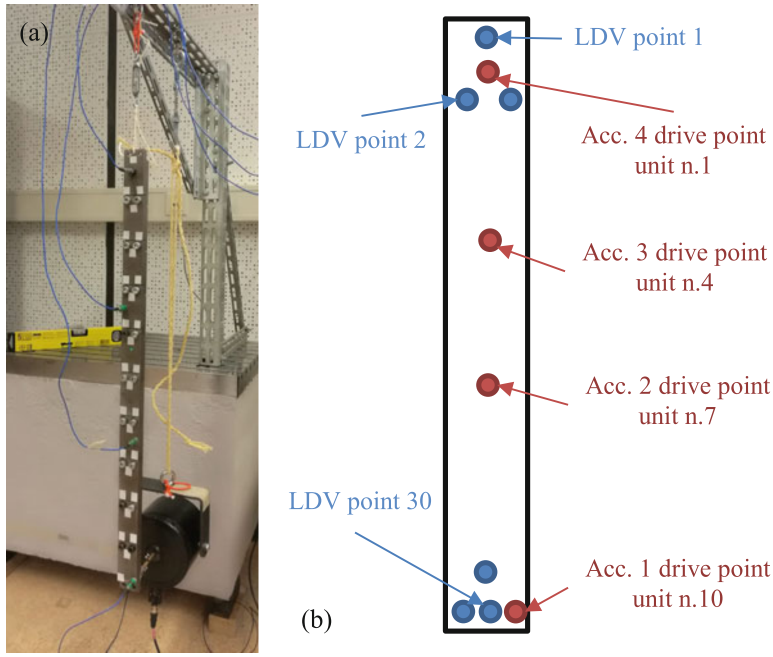

Fig. 1.4 Test setup in (**a**) and measurement points in (**b**)

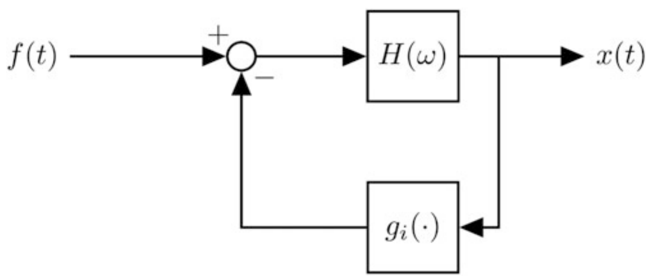

Fig. 1.5 Block diagram

to the other linear terms, the nonlinear function $g_i(\cdot, \cdot)$ that characterises the shape of the nonlinearity and the boolean-like vector w_i used to describe the location of the nonlinear term.

Using the Fourier Transform $\mathcal{F}[\cdot]$ to pass from the time domain to the frequency domain:

$$\left[\boldsymbol{K} - \omega^2 \boldsymbol{M} + j\omega \boldsymbol{C}\right] X(\omega) = F(\omega) - \sum_{i=1}^{M} \mathcal{F}\left[q_i \cdot g_i\left(w_i x(t), w_i \dot{x}(t)\right)\right]$$

$$\boldsymbol{A}(\omega) X(\omega) = F(\omega) - G(\omega)$$

$$X(\omega) = \boldsymbol{H}(\omega) \cdot (F(\omega) - G(\omega))$$

This can be viewed as a set of nonlinear feedback forces acting on an underlying linear system and can be represented as a block diagram in Fig. 1.5.

For the case of a 4-DOFs system with a grounded nonlinearity at DOF#1 and a non-grounded nonlinearity between DOF#3 and DOF#4 and excited at DOF#2

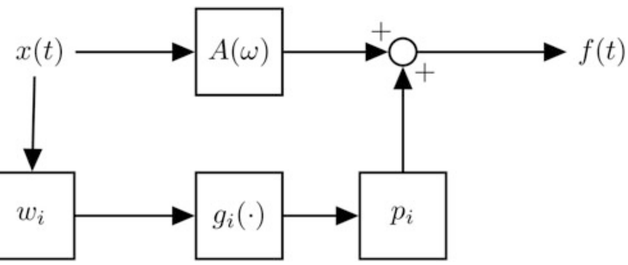

Fig. 1.6 MISO system

$$\begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ X_4 \end{bmatrix} = \begin{bmatrix} H_{11} & \cdots & H_{14} \\ \vdots & \ddots & \vdots \\ H_{41} & \cdots & H_{44} \end{bmatrix} \cdot \begin{bmatrix} -\mathcal{F}\left[q_1 \cdot g_1\left(x_1\right)\right] \\ F_2 \\ -\mathcal{F}\left[q_2 \cdot g_2\left(x_3 - x_4\right)\right] \\ \mathcal{F}\left[q_2 \cdot g_2\left(x_3 - x_4\right)\right] \end{bmatrix}$$

$$\begin{cases} X_1 = -H_{11}G_1 + H_{12}F_2 - H_{13}G_2 + H_{14}G_2 \\ X_2 = -H_{21}G_1 + H_{22}F_2 - H_{23}G_2 + H_{24}G_2 \\ X_3 = -H_{31}G_1 + H_{32}F_2 - H_{33}G_2 + H_{34}G_2 \\ X_4 = -H_{41}G_1 + H_{42}F_2 - H_{43}G_2 + H_{44}G_2 \end{cases}$$

Rewriting the equations in terms of the input force, one decomposes a single SIMO system with nonlinear feedback into a set of reverse MISO systems like in Fig. 1.6.

$$\begin{cases} F_2 = H_{12}^{-1}X_1 + H_{12}^{-1}H_{11}G_1 + H_{12}^{-1}\left(H_{13} - H_{14}\right)G_2 \\ F_2 = H_{22}^{-1}X_2 + H_{22}^{-1}H_{21}G_1 + H_{22}^{-1}\left(H_{13} - H_{14}\right)G_2 \\ F_2 = H_{32}^{-1}X_3 + H_{32}^{-1}H_{31}G_1 + H_{32}^{-1}\left(H_{13} - H_{14}\right)G_2 \\ F_2 = H_{42}^{-1}X_4 + H_{42}^{-1}H_{41}G_1 + H_{42}^{-1}\left(H_{13} - H_{14}\right)G_2 \end{cases}$$

By feeding in any combination of location vectors w_i, nonlinear operators $g_i(\cdot,\cdot)$ and scaling factors q_i, it is possible to assess the quality of the system by making use of standard spectral techniques. One of the best suitable metrics of causality is the multiple coherence between the one output and all the inputs, defined as

$$\gamma^2\left(\omega\right) = \frac{S_{FX}\left(\omega\right) \cdot S_{XX}^{-1}\left(\omega\right) \cdot S_{FX}^{H}\left(\omega\right)}{S_{FF}\left(\omega\right)}$$

Where S are the frequency-dependent averaged auto and cross-spectral density matrices. Finally, this can be turned into a *coherence index* by normalising its integral over the considered bandwidth:

$$\kappa = \frac{1}{\omega_2 - \omega_1} \cdot \int_{\omega_1}^{\omega_2} \gamma^2\left(\omega\right)\, d\omega$$

1.4 Preliminary Analysis

As stated in the previous section, accelerometers and laser data have different structure: the former consists of 30 different tests of 4 acceleration response channels acquired at the same time while the latter is a single collection of 30 velocity response channels acquired at different times. It is thus necessary to perform some preliminary data analysis to check the sanity of the data, in form of linearity and stationarity checks.

Linearity checks are performed by super-imposing FRFs and coherences of the systems at different force levels. The plots in Figure 1.7 show severe curve degradations in the tests at 8 Nm torque.