Chapter 36
The Effects of Boundary Conditions, Measurement Techniques, and Excitation Type on Measurements of the Properties of Mechanical Joints

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Abstract This paper investigates how the responses of mechanical joints are influenced by using different experimental setups. The experiments are conducted on both a monolithic beam and a bolted beam, and the beams are excited by hammer tests and a shaker. Multiple boundary conditions are also studied. It is found that the hammer tests performed on the "free" boundary condition monolithic beam (for multiple bungee lengths and positions) had a negligible influence on the system in terms of damping ratio and frequency variation. Multiple sensors attached to the monolithic beam are studied; the effect of multiple accelerometers manifests as a significant shift of frequency and damping due to the additional mass. In the case of the jointed beam, both mirror-like and rough interfaces are used. Several sets of different interface pairs, bolt torques, bolt preloads, excitation frequency sweep rates and bolt tightening orders are considered in this study. The time varying changes in stiffening and damping are measured by testing multiple combinations of the experimental setup at different levels of excitation. The results showed that the mirror-like surface finish for the interface has higher damping values compared to the rough surface across multiple bolt torque scenarios (such as preload and tightening order) and modes of vibration. Guidelines for a more reliable measurement of the properties of a mechanical joint are made based on the results of this research.

Keywords Bolted joints • Nonlinear vibration • Experimental setup • Measurement effects • Testing guidelines

36.1 Introduction

The study and measurement of nonlinear systems is, and will continue to be, a challenging endeavor. Inherent in the measurement of nonlinear systems is the challenge of measuring multiple stable and unstable equilibria, which is paramount for identifying the system level features of a nonlinearity. Much of the difficulty associated with nonlinear systems originates in a poor understanding of the physics governing the nonlinearity. As the system is nonlinear, even small differences in the constitutive models of a nonlinearity can lead to dramatic differences in the system's responses [1, 2].

Often, experimental measurements of a nonlinear system convolute the effects due to the experimental setup and the effects due to the nonlinearity of interest [3, 4]. For instance, a clamped boundary condition is often prohibitively difficult

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G. Kerschen (ed.), *Nonlinear Dynamics, Volume 1*, Conference Proceedings of the Society for Experimental Mechanics Series, DOI 10.1007/978-3-319-15221-9_36

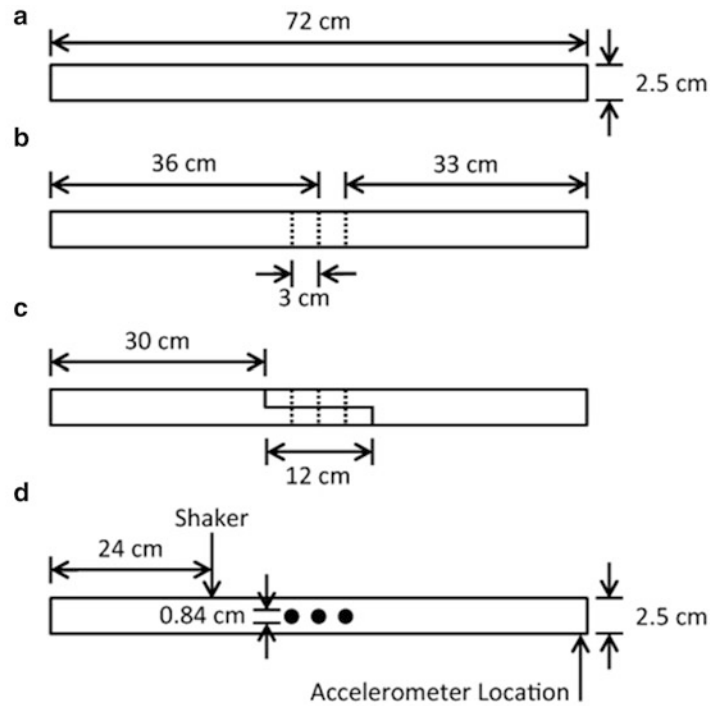

Fig. 36.1 The geometry of the (**a**) monolithic system, (**b**) monolithic system with bolt holes, (**c**) jointed system, and (**d**) top view of the monolithic system with bolt holes. For both the monolithic system with bolt holes and the jointed system, the size and position of the bolt holes and shaker/stinger attachment point are the same

to achieve; in most cases, some amount of torsional stiffness, friction, and even intermittent contact can be introduced by the boundary condition. At the opposite extreme, the free boundary condition is often replicated via bungee cords that act as weak springs. What effect does this added stiffness have on the system though?

In the study of mechanical joints, many questions persist because a complete understanding of the interfacial physics does not, and will not for the foreseeable future, exist [4]. At present, the salient questions of interest in measuring the response of a mechanical joint are:

- How does the mechanical joint affect the system's stiffness?
- How does the mechanical joint affect the system's damping?
- How do these quantities change with excitation parameters (amplitude, frequency, etc.)?

To address these questions, it is paramount that the effects of the mechanical joint are separated from other sources of uncertainty and/or nonlinearity, including the boundary conditions, manner of excitation, and the system itself. In order to approach this problem, the Brake-Reuss beam [5] is proposed as a candidate system for studying lap joints in a dynamic framework. The Brake-Reuss beam (Fig. 36.1) is a set of three separate systems that are designed to determine the effects of a lap joint to a system's dynamic response. The first system (Fig. 36.1a) is a monolithic beam with no mechanical joints, which serves as a reference system that is used to deduce the contribution of the lap joint to the system's damping and stiffness. The second system (Fig. 36.1b) is a monolithic beam with three through-holes, through which bolts are passed and tightened in order to determine the contribution of the bolts (in terms of added mass in addition to stiffness and damping) to the dynamic response. Figure 36.1c shows the third system, in which the lap joint is present. In all three systems, a shaker can be attached via a stinger approximately 24 cm from one end of the beam. The transfer function for the acceleration is then measured at the opposite tip of the beam. Preliminary analysis of this system [5] indicates that there is a strong effect due to the lap joint on the system's frequency response functions. This is an important property as this strong effect is not always discernible in systems with mechanical joints [6].

One significant difference between the present research and the system studied in [5] is that the shaker's attachment to the beams is located along the centerline of the beam. In [5], the shaker's attachment was offset from the centerline, which led to the excitation of the torsional modes in addition to the bending modes of the beam. A result from the preliminary analysis of [5] found that the torsional modes are so sensitive to the interfacial conditions (such as alignment) that the effects of the residual stresses due to the bolt preloads is not distinguishable from other effects. The bending modes, in contrast, are observed in [5] to be sensitive to the residual stresses due to the bolt preloads (and the order in which the bolts are tightened).

In what follows, a more thorough analysis of the system of [5] is presented. This analysis studies the effects of different loading/excitation conditions and boundary conditions. The interfacial properties are also directly measured for different surface finishes, alignments, and bolt torques, and these results are correlated with the properties observed in the frequency response functions. Finally, recommendations for the measurement of nonlinear systems with mechanical joints are made.

36.2 Effects of Boundary Conditions

One method to isolate the effects of a mechanical joint on a system's dynamic response, is to design the test setup to be as linear as possible. To test the different permutations of experimental setups, the monolithic beam (Fig. 36.1a) is used. All test specimen are fabricated from stainless steel 304. The basic experimental setup is shown in Fig. 36.2, which includes the beam, two bungee cords, two PCB 356A03 Triaxial ICP Accelerometers (Accel), two 356A02 Triaxial ICP Accelerometers used in a second experimental setup, a PCB 086C03 ICP Impact Hammer (Hammer), a Bruel & Kjaer PM Vibration Exciter Type 4809 (Shaker), LMS 16 Channel Spectral Analyzer, and a modular support rig. The excitation effects are tested in two groups: first with the impact hammer, and second with the shaker. In the first test, the mode shapes and natural frequencies of the beam are measured using a roving hammer technique with 58 impact lines with two impacts on each line. Table 36.1 contains the modal frequencies and damping, and Fig. 36.3 displays the mode shapes. The numbering in Fig. 36.3 is for the modes calculated using the Finite Element software ABAQUS 6.10. The first longitudinal mode is the seventh mode of the system, which is not excited during the present experiments.

The first group tested the effects the impact hammer tip type (metal or plastic), the impact force amplitude (high or low), the bungee cord length (0.318 m or 0.1651 m), the location of the bungee cords (inside, 10.2 cm separated centered, or outside, 5.72 cm from edges of the beam), the accelerometers' attachment condition (glued or waxed), the position of the accelerometer cables (hanging, supported out the side, or straight down as shown in Fig. 36.4), and the number of sensors

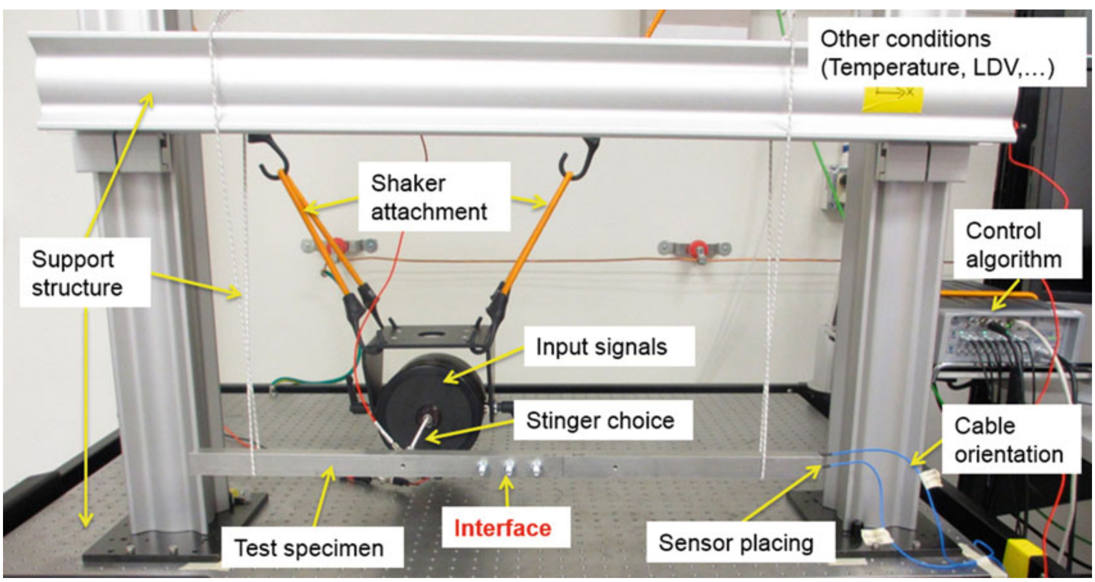

Fig. 36.2 Basic test setup for testing the effects of boundary conditions, excitation techniques, and sensor setup

Table 36.1 Modal testing results on the monolithic beam

Mode	Frequency [Hz]	Modal damping [%]
1st Bending	246.3	0.03
2nd Bending	674.5	0.04
3rd Bending	1, 308.5	0.03
1st Torsional	1, 965.0	0.03
4th Bending	2, 130.7	0.04
5th Bending	3, 126.8	0.04
2nd Torsional	3, 931.8	0.05

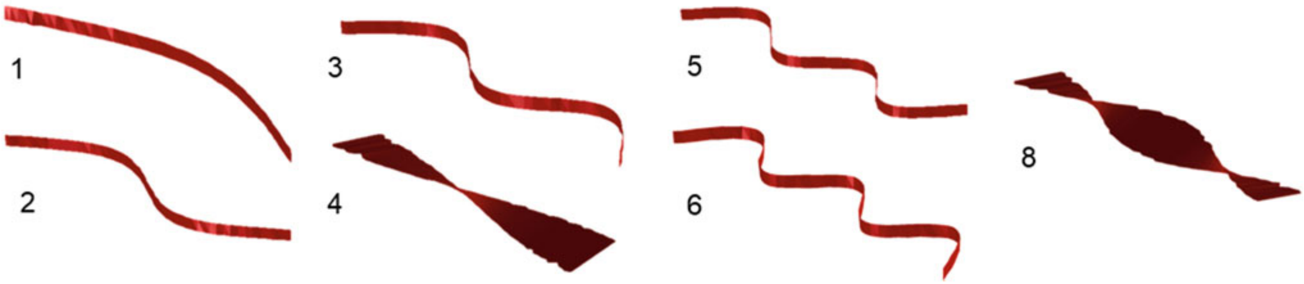

Fig. 36.3 Mode shapes measured in modal analysis

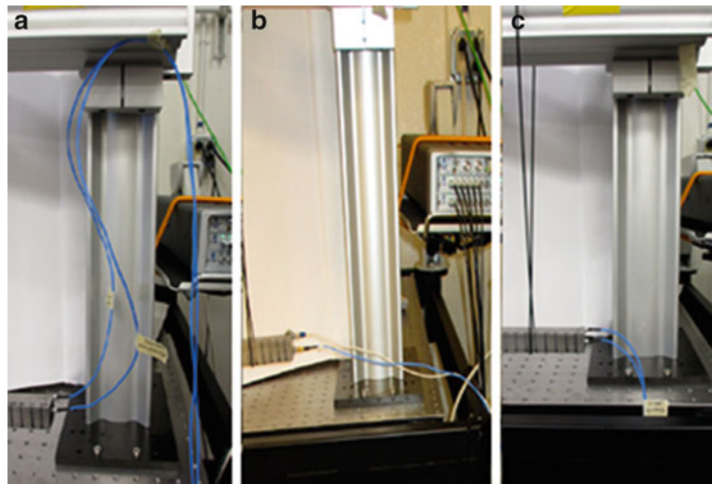

Fig. 36.4 Sensor cable positions: (**a**) supported above, (**b**) supported across, and (**c**) unsupported

Table 36.2 Experimental setups tested for nonlinear influences

Test number	Bungee cord type	Bungee cord location	Bungee cord length (m)	Accelerometer mass (grams)	Accelerometer attachment technique	Hammer mass (grams)	Accelerometer cable orientation
1	White	Inside	0.318	1	Glue	160	Above
2	White	Inside	0.318	1	Glue	235	Above
3	White	Outside	0.318	1	Glue	160	Above
4	White	Outside	0.1651	1	Glue	160	Above
5	Black	Outside	0.318	1	Glue	160	Above
6	Black	Outside	0.318	1	Wax	160	Above
7	Black	Outside	0.318	10.5	Wax	160	Above
8	Black	Outside	0.318	1	Glue	160	Across
9	Black	Outside	0.318	1	Glue	160	Unsupported

on the specimen. The various tests, not including multiple sensors, are listed in Table 36.2. The tests are run with a single impact in order to not average any measured nonlinear response out of the results. The resulting frequency response functions (FRFs) are shown in Fig. 36.5. In order to verify that the test beam is isolated from all sources of external excitation aside from the impact hammer, the support structure is impacted after each change in the bungee cord location or length, at three different locations: at the center of the cross bar, at a location adjacent to the bungee cord, and at the attachment location to the table. The results of these tests show that there is not any significant nonlinearities in any of the test setups, which is desirable. The only significant nonlinearities appear in the torsional modes, which are constrained by the bungee cords; the severities of the nonlinearities are listed in Table 36.3. These nonlinearities could potentially be avoided by an alternative attachment method of the bungee cord to the end of the beam. In the absence of being able to manufacture the beam to attach to the bungee cords in a specific manner, the principle option for an alternative attachment method would be to glue to the bungee cords to the ends of the beam; however, this is dependent on the strength of the glue and whether the bungee is stretched before being glued.

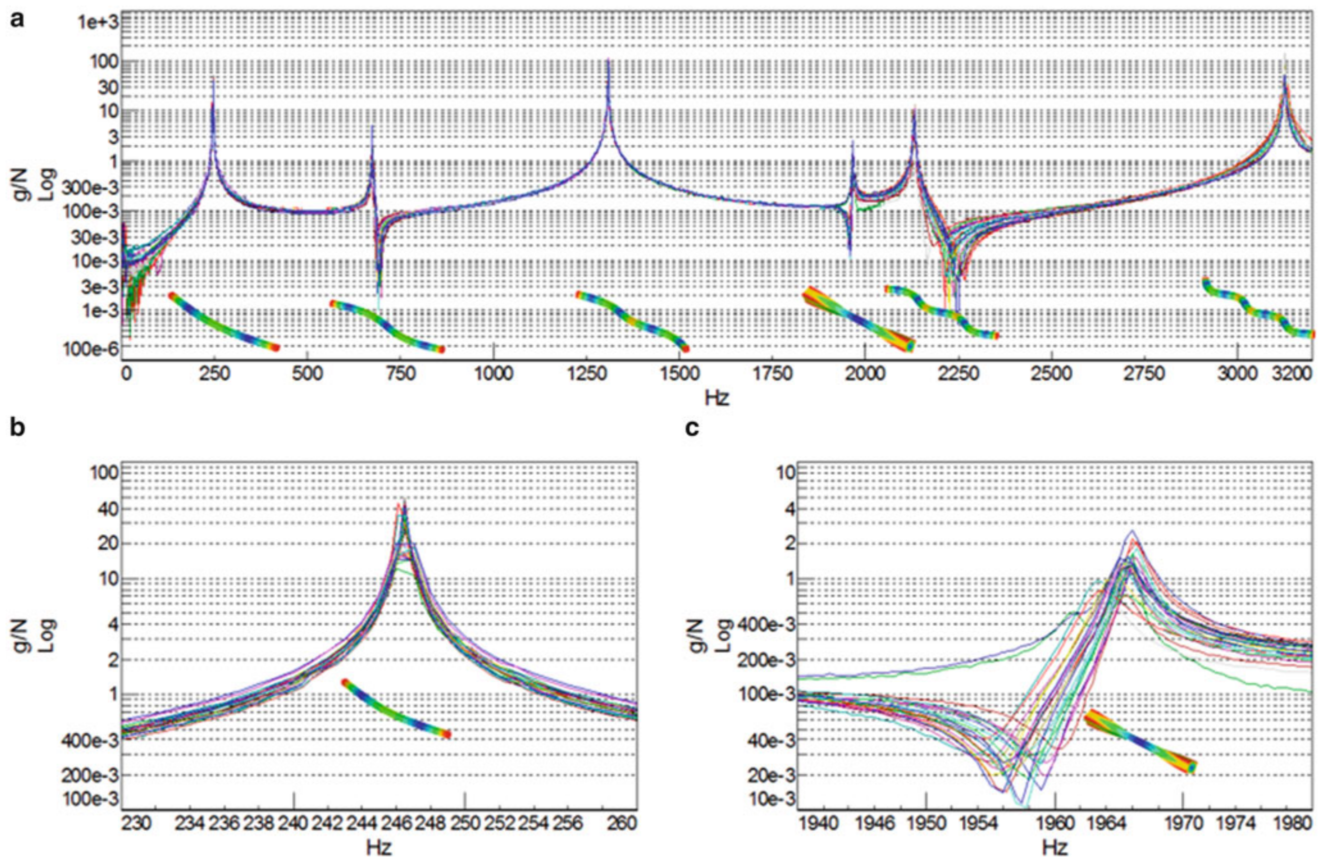

Fig. 36.5 The FRFs for the experimental setups listed in Table 36.2, for (**a**) the entire frequency range studied, (**b**) the first natural frequency, and (**c**) the fourth natural frequency

The final impact hammer test of the monolithic beam is multiple sensors placed on the beam; since the mass of the accelerometers are much smaller than that of the beam (approximately 0.02 % of the beam's mass), it is predicted that effects of more sensors would not be noticeable. Sensors that have a mass of 5 g are placed onto the beam as shown in Fig. 36.6 in four different testing scenarios. The results (Fig. 36.7) show that the torsional modes are most effected by the added masses, resulting in a 17 Hz shift in the first torsional mode and increased damping in the second torsional mode. The severities of the nonlinearities are listed in Table 36.3. As can also be seen in Figs. 36.7b and c, both the reference response and the response of loading scenario three are grouped together in the fourth mode while the reference response and the response of loading scenario one are grouped together in the seventh mode. This is a result of the additional sensors' placement on a node of those modes. The shift in frequency and damping could be associated with the sensors changing the moment of inertia of the beam at the anti-nodes of vibration.

36.3 Effects of Excitation and Measurement Conditions

The second series of experiments utilizes a shaker to excite the beam. The type of stinger, amplitude of excitation, type of signal, sweep direction, and the use of a Polytec OFV-55x Fiber-Optic Interferometer (LDV) compared to an accelerometer are studied. The shaker is attached to the beam with a PCB 208A03 force transducer at the location indicated in Fig. 36.8. The type of stinger is tested first to determine which stinger causes the smallest deviations from the FRFs measured using the impact hammers. The stingers tested are shown in Fig. 36.9 and the results generated by supplying a sweep signal are shown in Fig. 36.10. These results show that the wire and M2 stingers have a significant influence on the response of the system; the two stingers listed also excited the torsional mode which should not have been excited with the position of the stinger on the beam. The torsional modes being excited indicates that the stingers are bending and the restoring force is causing energy to be inputted into the mode.

Table 36.3 Severity of the nonlinear influence from different experimental setups on the natural frequency and modal damping of the beam

Category		Effect on frequency	Effect on damping	Notes
Hammer	Metal	REF	REF	Frequency range with good coherence: 0–8 kHz
	White plastic tip	Low	Low	Frequency range with good coherence: 0–3.2 kHz
	Metal tip with added mass (75 g)	Low	Low	More energy input into structure at low frequencies
Bungees	Full length (13″)	REF	REF	
	Half length	Low	Low	Marginal frequency shifts (1 Hz)
	Inside position	Low	Low	Marginal frequency shifts (1 Hz)
	Outside position	Low	Low	Marginal frequency shifts (1 Hz)
Accelerometers	2 accelerometers glued	REF	REF	Accelorometers glued at one end of the beam, each 4 g
	2 accelerometers attached via wax	Low	Low	Marginal frequency shifts (1 Hz)
	Number of sensors: Scenario 1 (Fig. 36.6)	Mid	Mid	Moderate change in frequency and damping
	Number of sensors: Scenario 2 (Fig. 36.6)	High	High	Frequency shifts from 1 Hz to 30 Hz down (mode dependent); torsional & higher bending modes highly damped
	Number of sensors: Scenario 3 (Fig. 36.6)	High	Low	The frequency shifted up for 2nd torsional but down for higher modes, damping has minor changes
	Cable orientation: up, down, on table	None	Low	Cable down causes slightly higher damping for some modes (+0.01 %)
Test rig		None	None	impacts on different spots of the test rig & table

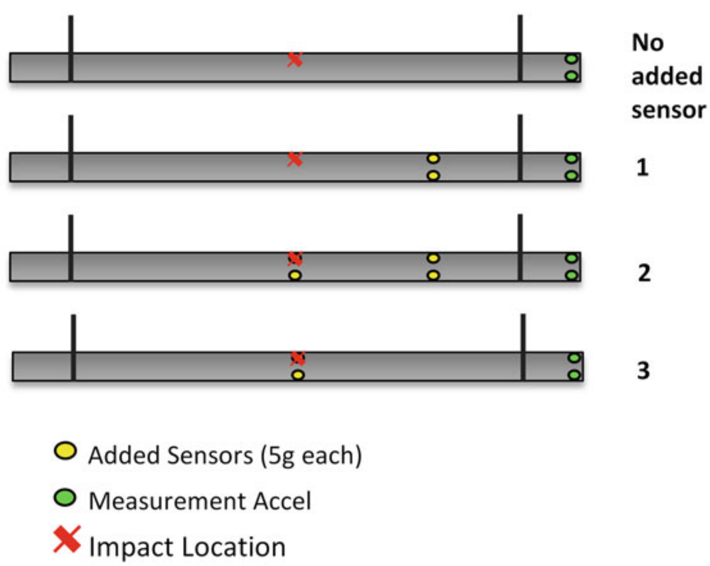

Fig. 36.6 Added sensor testing scenarios

The subsequent shaker excitation experiments use the 10–32 UNF stinger, which has the smallest influence on the response of the beam, and the corresponding FRFs are shown in Fig. 36.11. The severities of the nonlinearities for all shaker tests are listed in Table 36.4. The FRFs show that the use of the LDV only has the largest frequency shift, resulting from the removal of the accelerometers, and the white noise has extra peaks between the third and fourth peak, which could be from stinger modes being excited more than when sweep signals are generated.

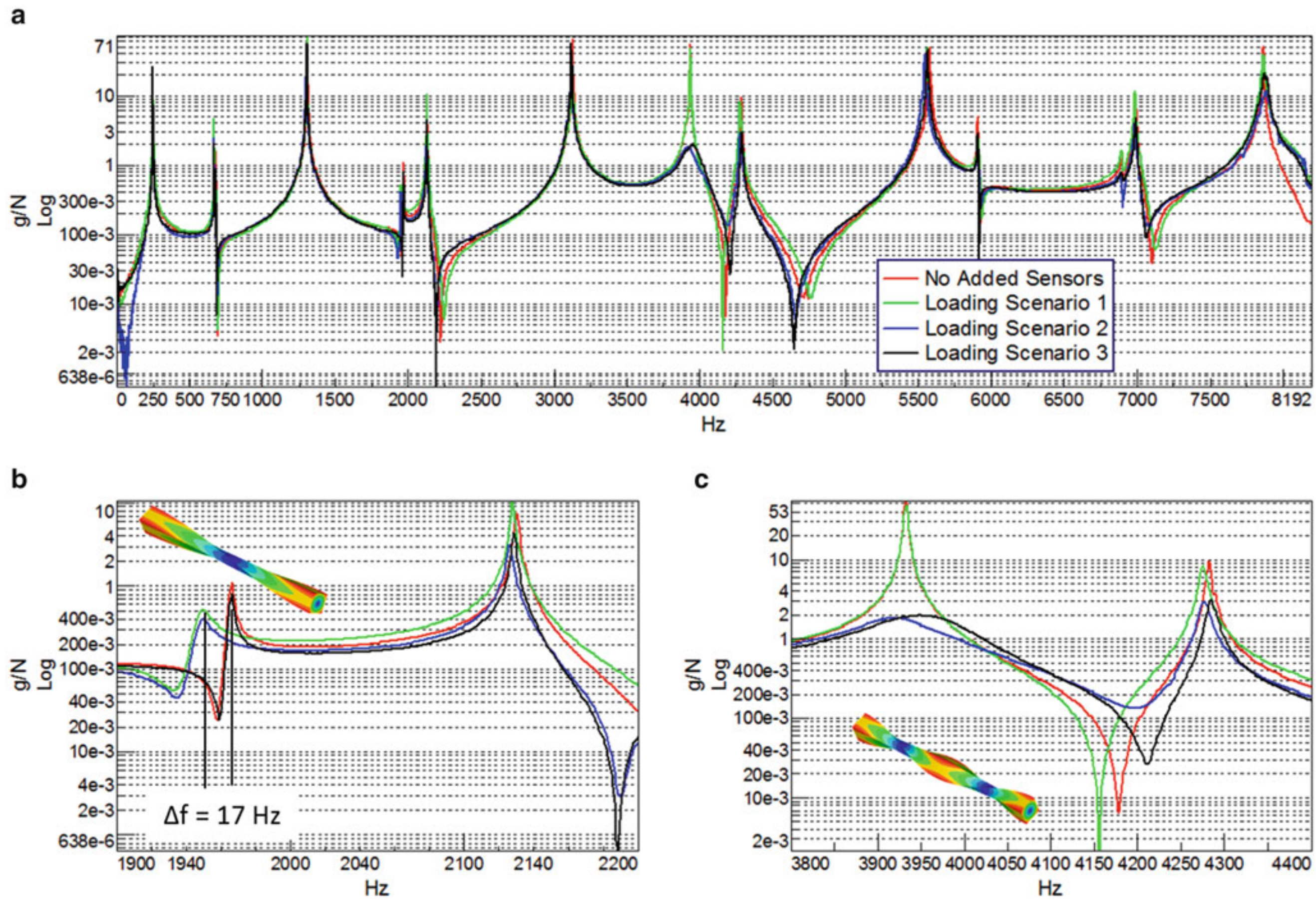

Fig. 36.7 The FRFs of the added mass test scenarios for (**a**) the entire frequency range, (**b**) the fourth and fifth natural frequencies, and (**c**) the seventh and eighth natural frequencies

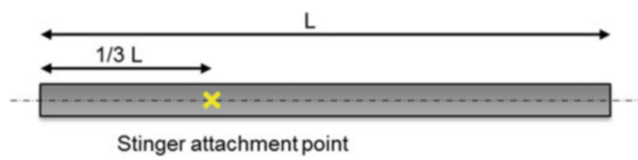

Fig. 36.8 Location of the shaker attached to the beam

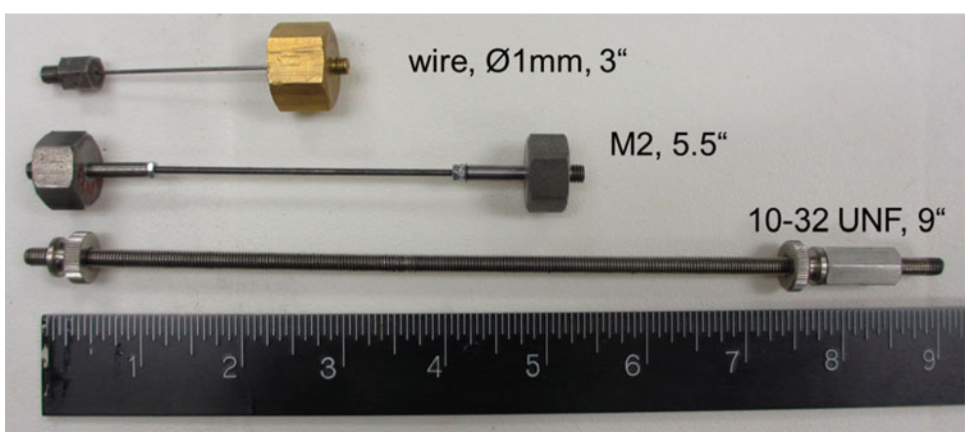

Fig. 36.9 The three stingers tested for nonlinear effects

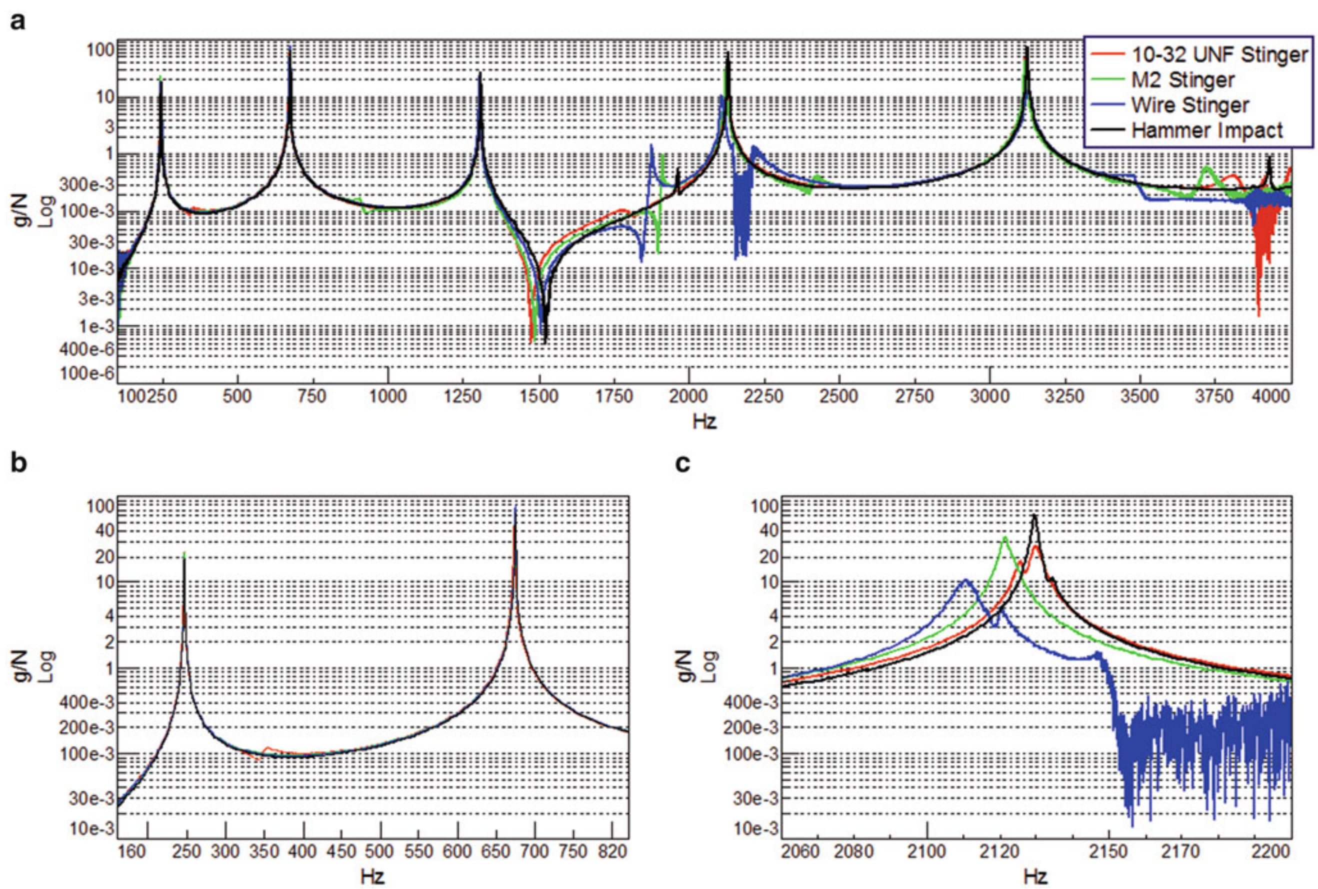

Fig. 36.10 The FRFs of the stinger tests and one impact test for (**a**) the entire frequency range, (**b**) the first and second natural frequencies, and (**c**) the fourth natural frequency (as measured with the impact hammer)

36.4 The Jointed Beam

The nonlinearities in a jointed beam currently are difficult to identify comprehensively due to the variability in the interfacial conditions (surface finish, alignment, asperity distribution, residual stresses, machining variations and curvature, etc.). The profile influences the pressure distribution as well as the friction in the joint. Two different surface finishes are used in this study: a rough finish and a mirror-like finish. The rough finish specimens are manufactured using wire electric discharge machining (EDM) and the mirror finish by laser cutting and polishing. The rough finish specimens developed a warped interface, such as the one shown in Fig. 36.12. The pressure distribution of the interfaces is mapped using FujiFilm Prescale Light Weight and Mid-Sensitivity films, with 5, 10, and 20 Nm torques on the bolts holding the specimens together, shown in Fig. 36.13. The bolts are torqued to half the listed torque then to the full torque value in order to achieve a more uniform pressure distribution. The results of the pressure test shows that the rough interface specimens' pressure distribution is dependent on the bolt tightening order; whereas the mirror-like interface is independent of the order as the pressure distribution approximately is constant.

The jointed beams are then tested using the shaker with a constant sweep up or a stepped sine force control signal. The first test is of the bolt tightening orders, shown in Fig. 36.14, using the constant sweep signal, with results shown in Fig. 36.15 for the rough interface specimens and Fig. 36.16 for the mirror-like interface specimens. The results for the rough interface specimens shows that the response of the beam is more dependent on which outside bolt is tightened first; while the mirror interface response shows that it is independent of the tightening order. The beams are then tested using a sine step force control signal with different levels of bolt torque magnitude (hand tightened, 3, 5, 8, 10, and 20 Nm), shown in Figs. 36.17 and 36.18 for the rough interface specimens and mirror-like interface specimens, respectively. The response for the rough interface specimens appears to converge to a saturation point at approximately 10 Nm; however when the torque is increased further the response changes indicating that a saturation point was not reached. The mirror-like interface specimens' response shows that saturation is reached around 8 Nm.

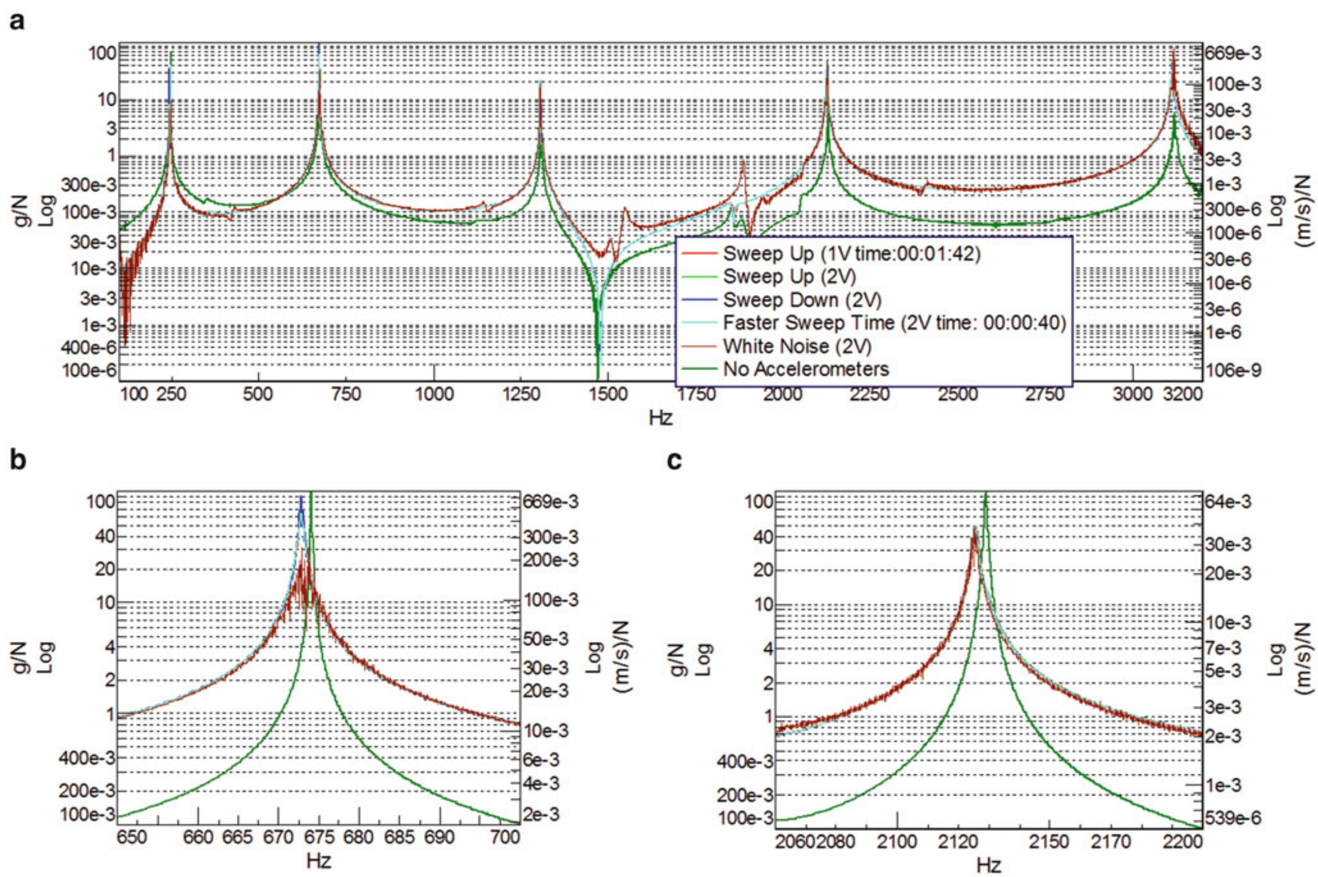

Fig. 36.11 The FRFs of the shaker signal tests over (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the fourth natural frequency

Table 36.4 Severity of the nonlinear influence of shaker test setups on frequency and damping of the beam

Category		Effect on frequency	Effect on damping	Notes
Stingers	Impact test	REF	REF	
	10-32 UNF (9″ length)	Low	Low	No torsional modes excited, matches best to hammer tests
	M2 (5.5″ length)	Mid	None	Frequency-shift down
	Wire (3″ length)	Mid	None	Frequency-shift down, torsional modes excited, stinger bending occurs
Excitation Amplitude (V)	High (2 V, sweep up)	REF	REF	
	Low (1 V, sweep up)	None	Low	Marginal frequency shifts (1Hz)
Signals	Sweep up	REF	REF	
	Sweep down	None	Low	Linear structure tested
	Sweep rate low (10 Hz/s)	Low	None	Needs to be investigated in more detail
	Sweep rate high (80 Hz/s)	Low	None	Needs to be investigated in more detail
	White noise	Low	Low	Noisy FRF, no anti-peaks
Accelerometers	Two accelerometers glued	REF	REF	
	No accelerometers (LDV)	Mid	Low	Frequency-shift to slightly higher frequency

Fig. 36.12 The rough interface specimen's warped interface

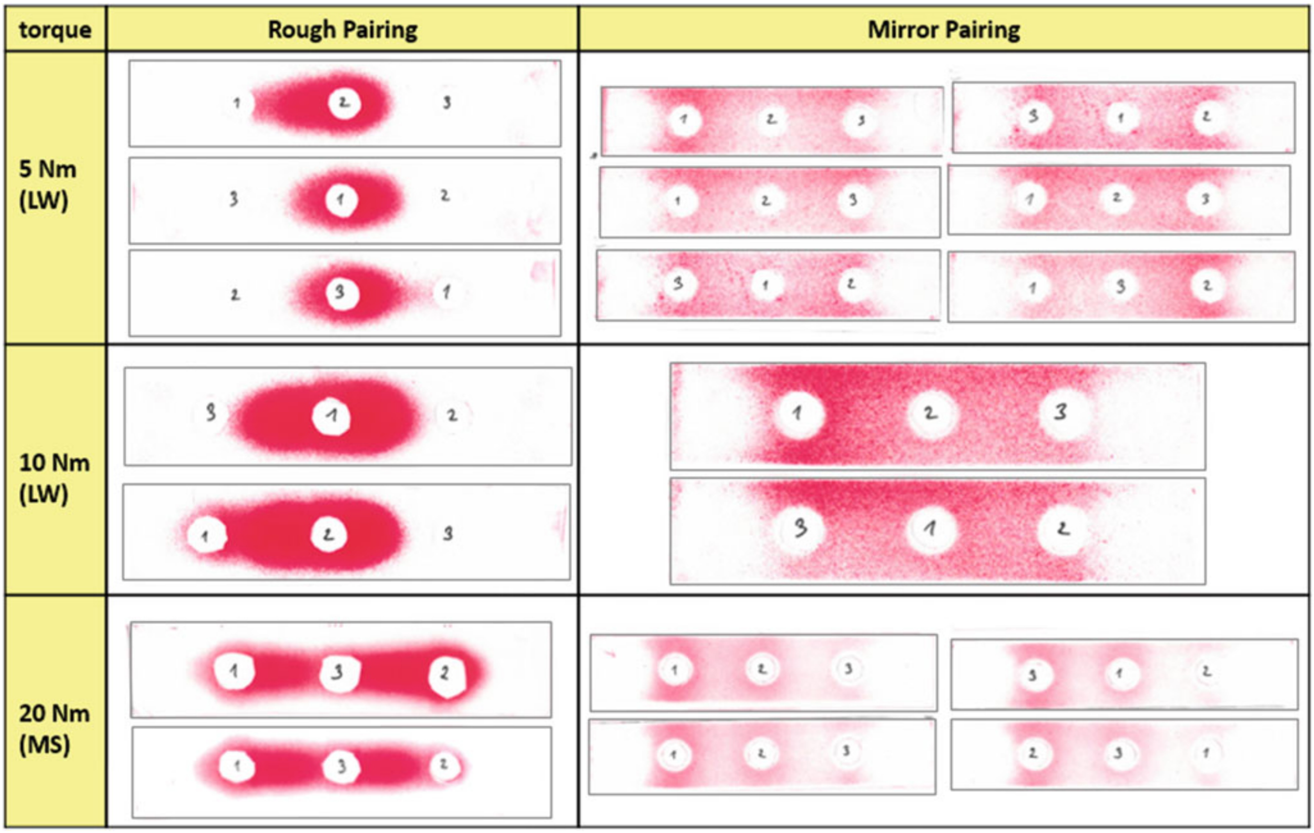

Fig. 36.13 Pressure distributions of the specimens with bolt tightening order indicated on each pressure distribution

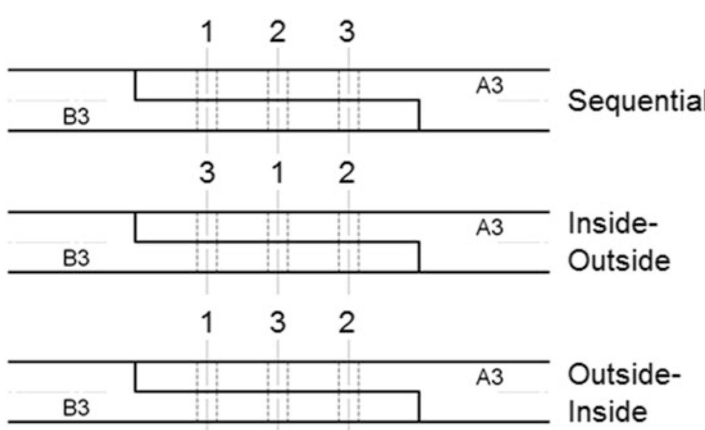

Fig. 36.14 Bolt tightening order shown on a mock joint

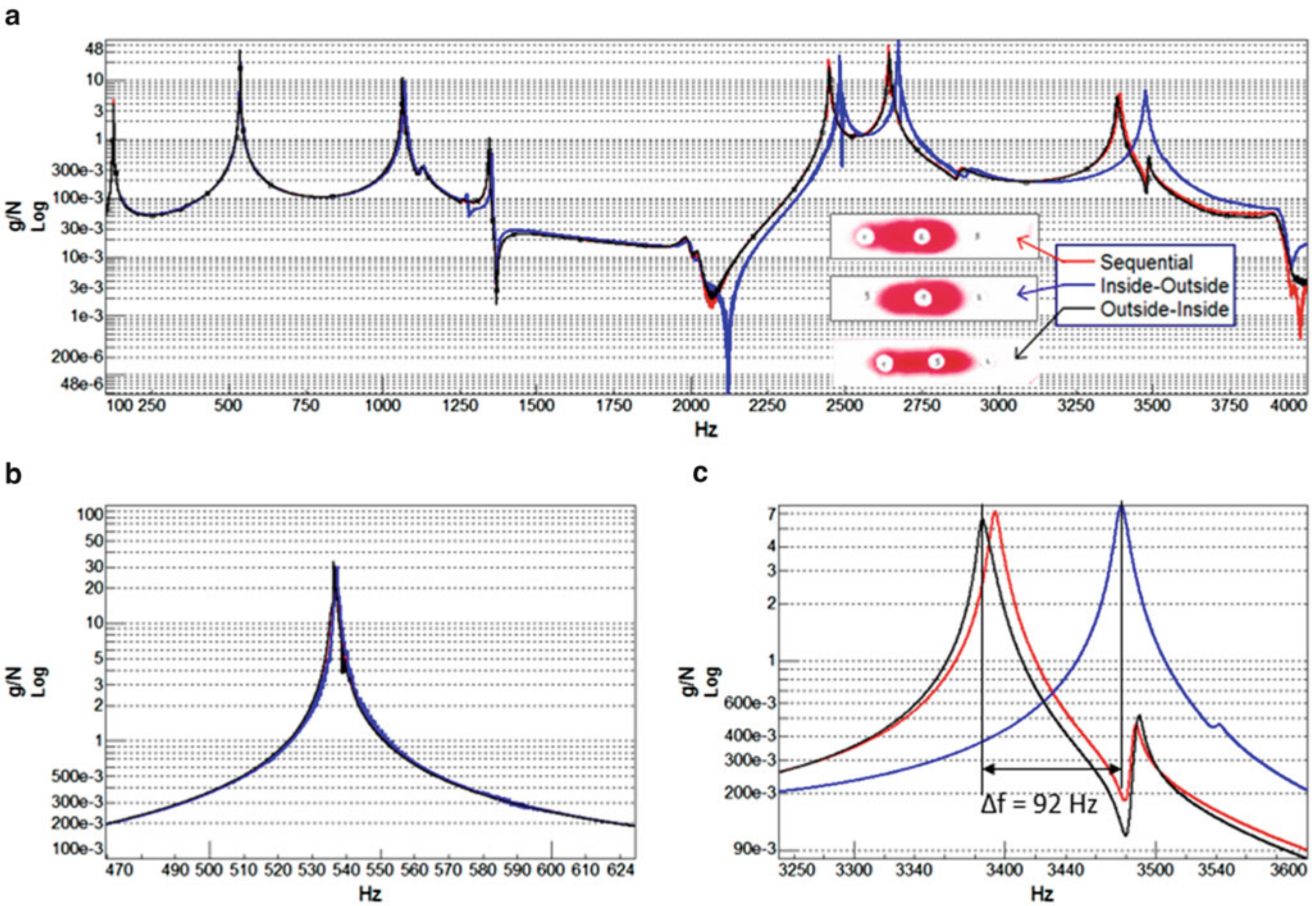

Fig. 36.15 The FRFs of the bolt tightening order of the rough interface for (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the seventh natural frequency. The tightening orders are: sequential from one side to the other (*red*), the middle then the outer bolts (*blue*), and the outer then the middle bolt (*black*)

There is little repeatability in the responses of jointed structures, due to the roughness of the interfaces being different on the micro-scale from specimen to specimen and from experiment to experiment [5]. To assess experiment-to experiment variability, the following procedure is used:

- The reference test is conducted after the assembly is first bolted together with a specified bolt torque.
- The beams are disassembled then immediately reassembled with the same specified bolt torque, and immediately tested.
- The beams are disassembled and reassembled with the same specified bolt torque, then allowed to rest for approximately 600 s before being tested.
- Lastly, the bolts are then retorqued back to the specified bolt torque level without disassembling the beams, and then tested.

The responses of the rough interface specimens and mirror-like interface specimens are shown in Figs. 36.19 and 36.20, respectively, for this series of tests. Unlike previous measurements of this system [5], the experiment to experiment variability is found to be low. The discrepancy between this result and [5] is potentially due to the excitation of the torsional modes in [5] but not in the present work, and warrants further investigation. The final experiment is to study the effects of the sine step force control step direction, shown in Fig. 36.21. The results show that the response is direction independent, which is in stark contrast to nonlinear systems like the Duffing oscillator.

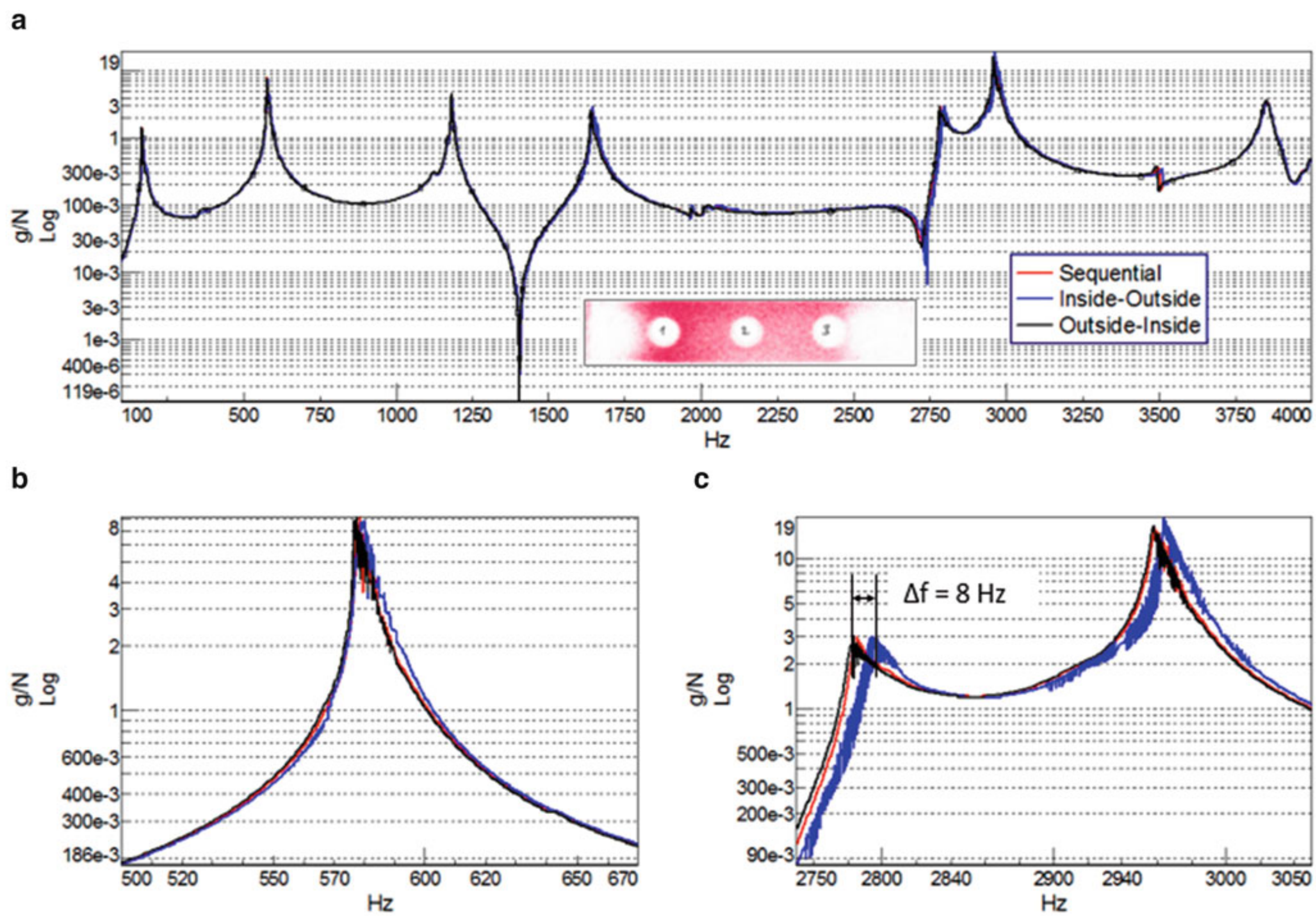

Fig. 36.16 The FRFs of the bolt tightening order of the mirror interface for (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the seventh natural frequency. The tightening orders are: sequential from one side to the other (*red*), the middle then the outer bolts (*blue*), and the outer then the middle bolt (*black*)

36.5 Conclusions

This research sought to identify a set of best-practices for conducting experiments of nonlinear systems, specifically those with jointed interfaces. The effects of the boundary conditions, excitation techniques, and measurement techniques on the measurements of the system's stiffness and damping properties are studied. Specific recommendations from this preliminary work are:

- For impact hammer measurements, the tip and hammer mass do not significantly affect the measurements of damping and stiffness. Thus, the impact hammer configuration should be chosen based on the frequency range of interest with harder tips without added mass being more appropriate to study higher frequencies.
- The responses of systems with free boundary conditions created via the use of bungee cords are insensitive to the bungee length, stiffness and position.
- The use of accelerometers, even when they are a small fraction of the mass of the system (0.02 % in the present work), can significantly affect the measurements of damping and stiffness. If used, accelerometers should be located away from node points, and the number should be kept to a minimum.
- For shaker excitation, care should be used in selecting an appropriate stinger. Of those tested, the thinner stingers are found to exhibit bending, which results in the appearance of nonlinearities in the system's frequency responses.
- The excitation sweep rate, magnitude, and direction are not found to have significant effects for the present system.
- The conditions of the interface have significant effects on the responses of the system, and future work should aim to thoroughly quantify the interface conditions.

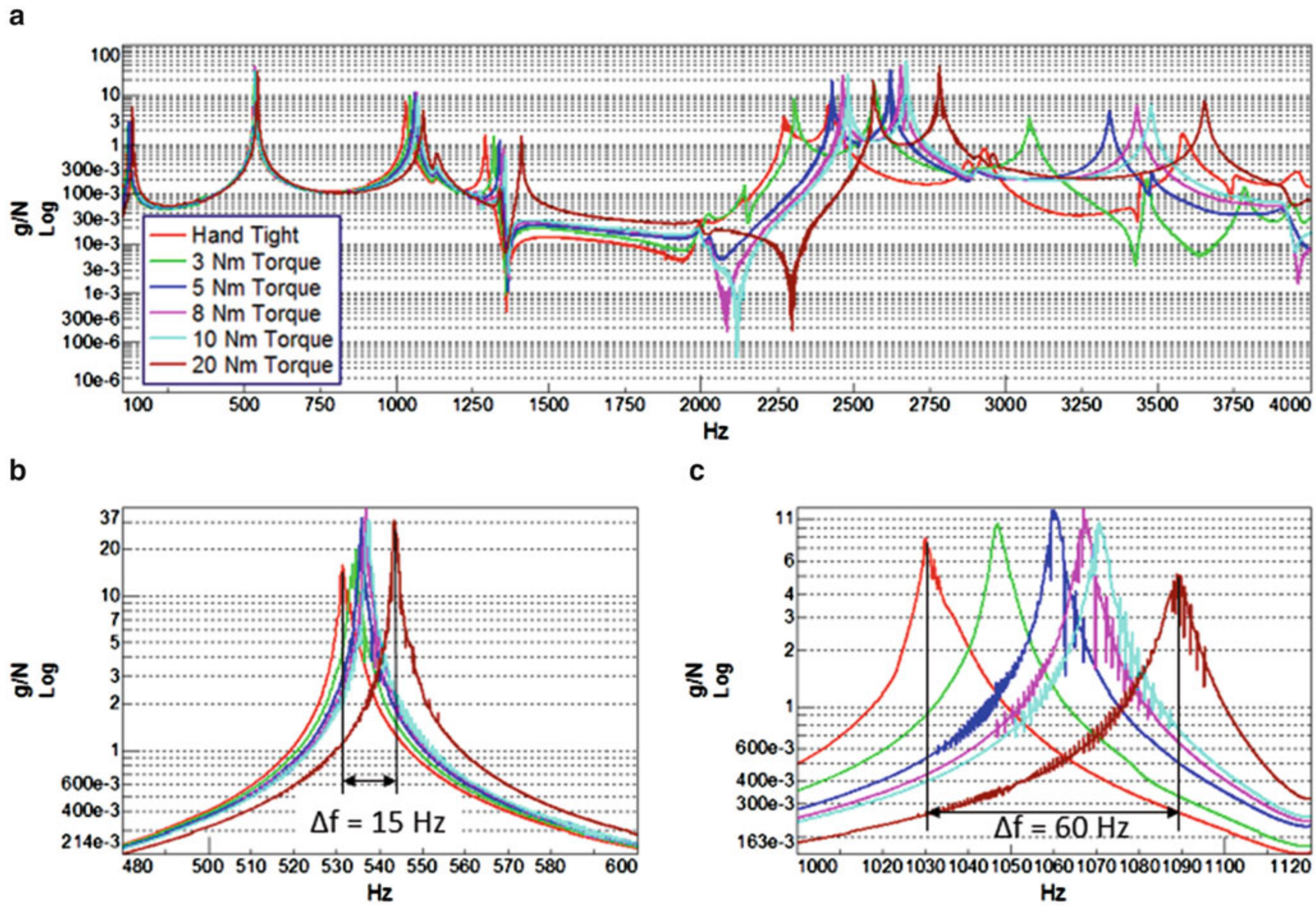

Fig. 36.17 The FRFs of the bolt torque of the rough interface for (**a**) the entire frequency range, (**b**) the second natural frequency, and (**c**) the fourth natural frequency

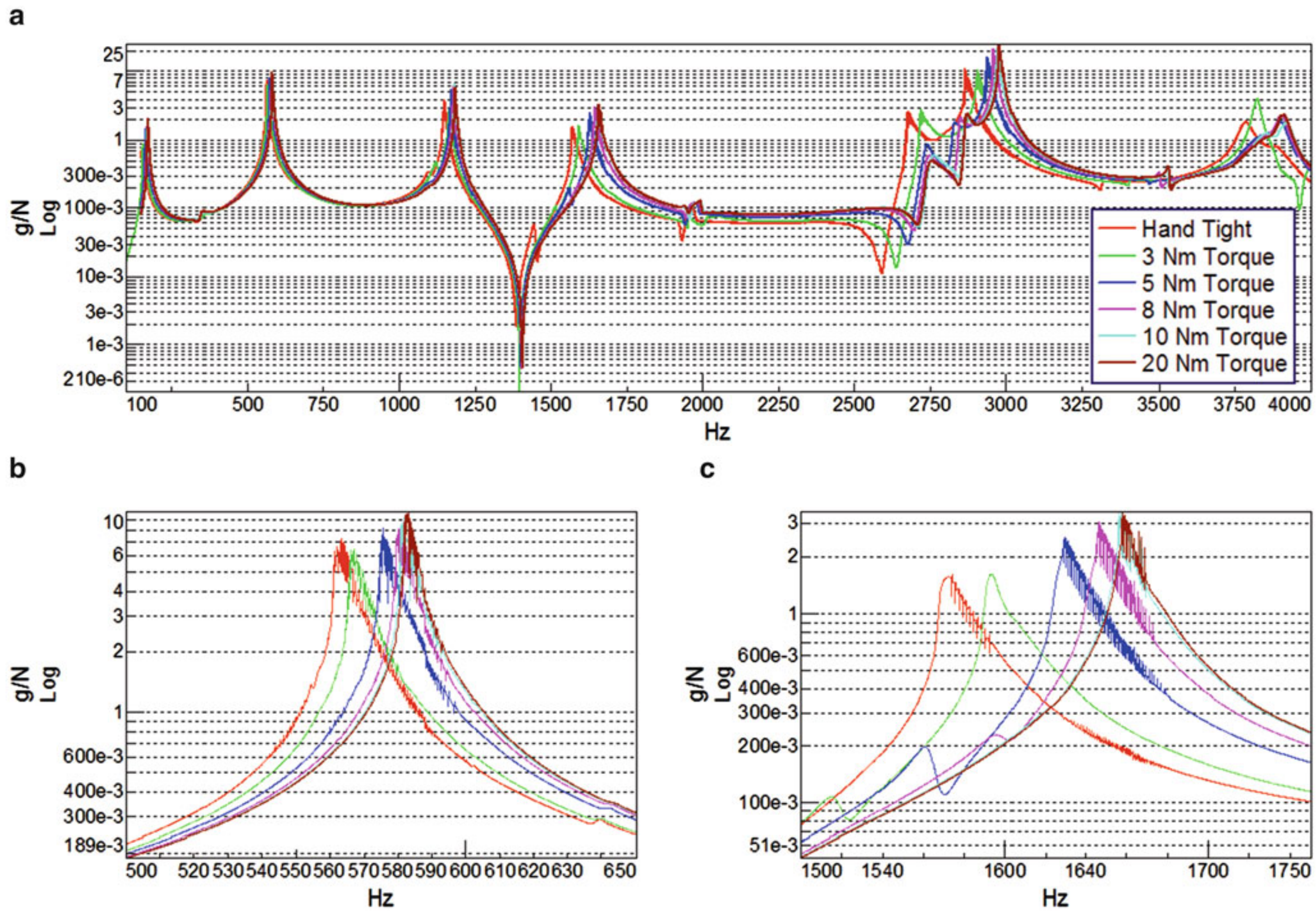

Fig. 36.18 The FRFs of the bolt torque of the mirror interface for (**a**) the entire frequency range, (**b**) the second natural frequency, and (**c**) the fourth natural frequency

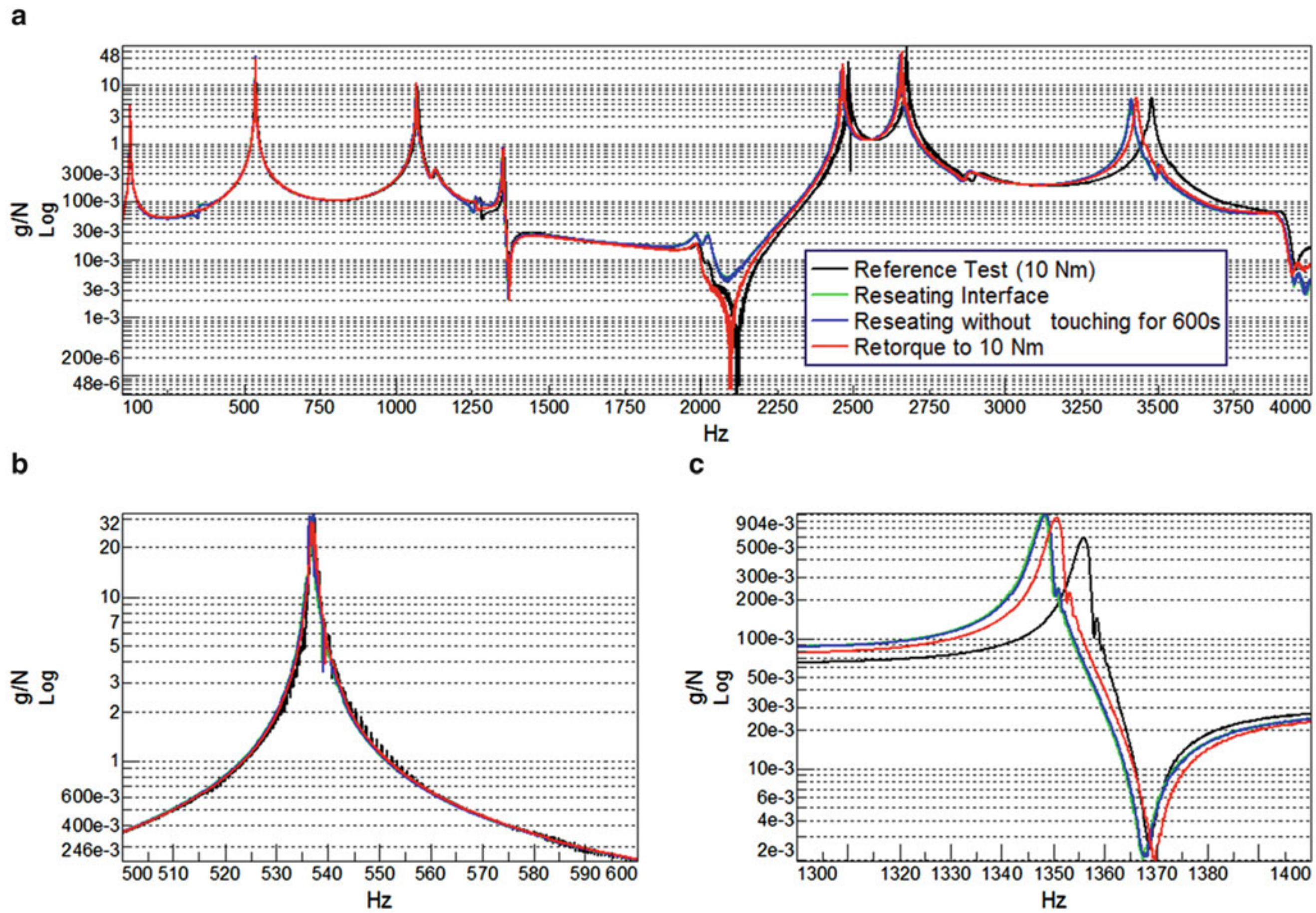

Fig. 36.19 The FRFs of the rough interface repeatability test for (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the fourth natural frequency

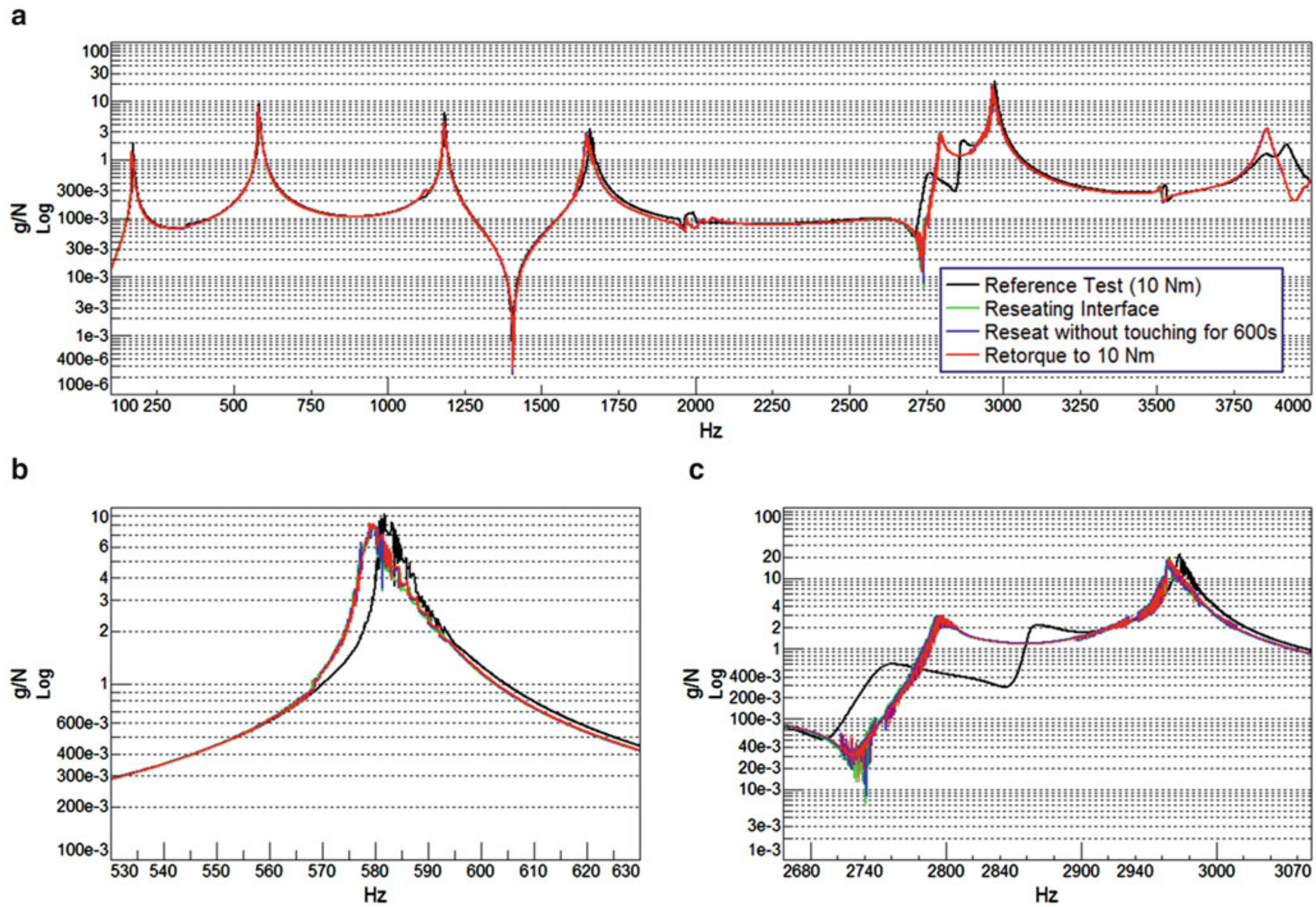

Fig. 36.20 The FRFs of the mirror interface repeatability test for (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the fourth natural frequency

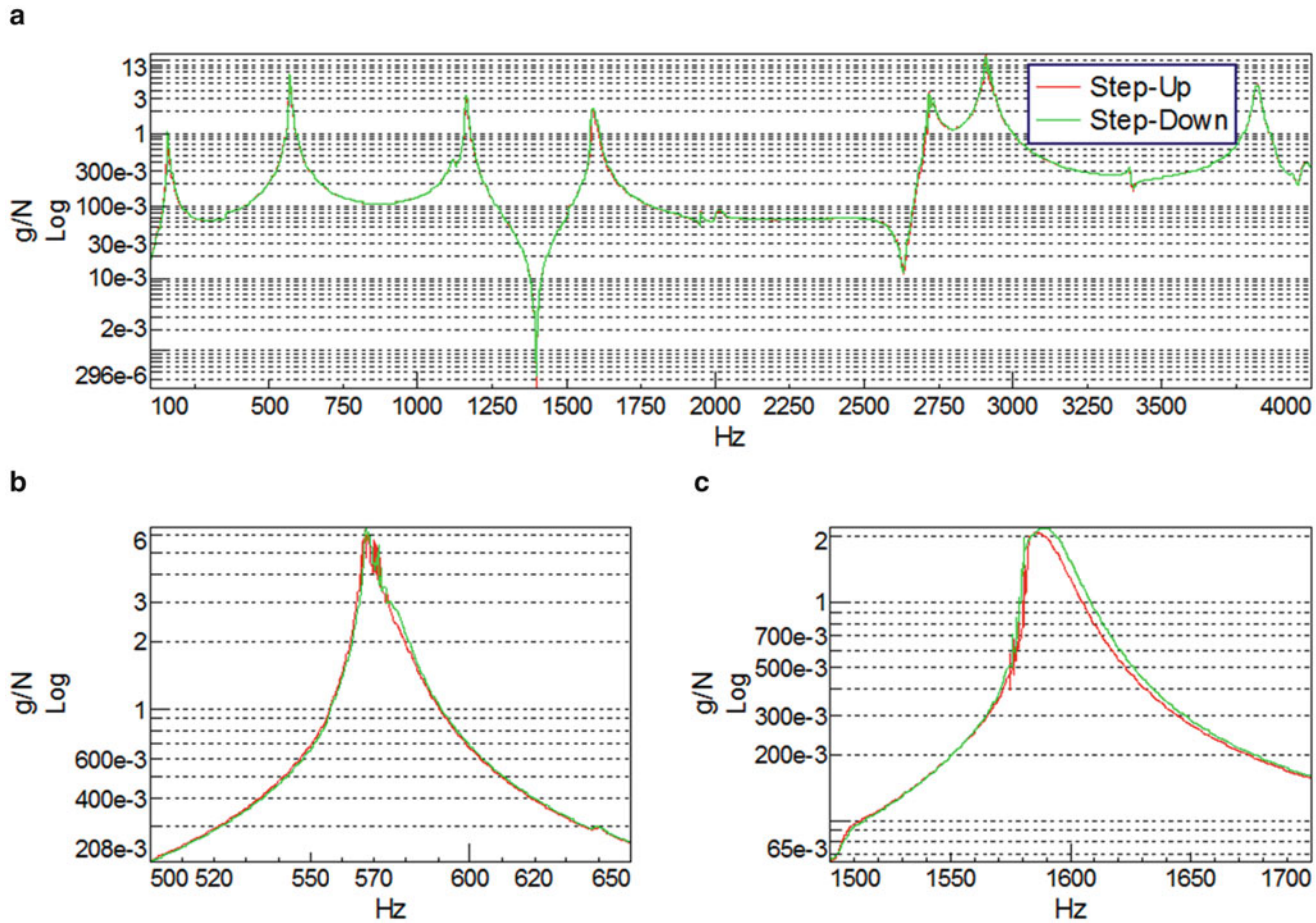

Fig. 36.21 The FRFs of the step direction for (**a**) the entire frequency range, (**b**) near the second natural frequency, and (**c**) near the fourth natural frequency

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