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REVIEW ARTICLE

Experimental studies on the energy dissipation of bolted structures with frictional interfaces: A review

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Abstract: Bolted joints play a more and more important role in the structure with lighter weight and heavier load. This paper aims to provide an overview of different experimental approaches for the dynamic behavior of structures in the presence of bolted joints, especially the energy dissipation or damping at frictional interfaces. The comprehension of energy dissipation mechanisms due to friction is provided first, while the key parameters and the measurement techniques, such as the excitation force, the preload of the bolt, or the pressure at the interfaces, are briefly introduced. Secondly, the round-robin systems aim to measure the hysteresis parameters of the frictional joints under tangential loads are reviewed, summarizing the basic theory and the strategies to apply the excitation force or acquire the response in different testing systems. Followed by parameter identification strategies for bolted structures, the test rigs with one or more simplified bolted joints are summarized to give an insight into the understanding of typical characteristics of bolted structures, which are affected by the presence of friction. More complex test rigs hosting real-like or actual engineering structures with bolted lap or flange joints are also introduced to show the identification process of the dynamic characteristics of bolted connections employed in specific applications. Based on the review paper, researchers can get the basic knowledge about the experimental systems of the bolted structures, especially several classical round robin systems, such as the Gaul resonator and widely used Brake-Reuß beam system. Readers can take advantage of this background for more creative and effective future studies, make more progress on the study of assembled structures and understand the influence of bolting frictional connections on the dynamic response better.

Keywords: review study; bolted joints; frictional damping; experimental rigs; measurements

1 Introduction

Joining techniques, like bolting, riveting, clamping, etc., are widely used in structural engineering for the assembly of mechanical components. The contact interfaces between the connected parts have a significant influence on the dynamic response of an assembled structure when it is loaded by time-varying forces. The influence of the bolted interfaces draws more and more attentions since it becomes more and more important for the structure with lighter weight and heavier load. A search result at the Scopus platform is given in Fig. 1, it shows that the number of published papers in the term "bolted joints" is continuously increasing, in fact, the publications in the last year are over 10 times as much as that in 30 years ago.

The dynamic response is closely related to the capability of the structure to dissipate vibration energy and most of the total damping usually comes from the joints instead of the material damping, especially

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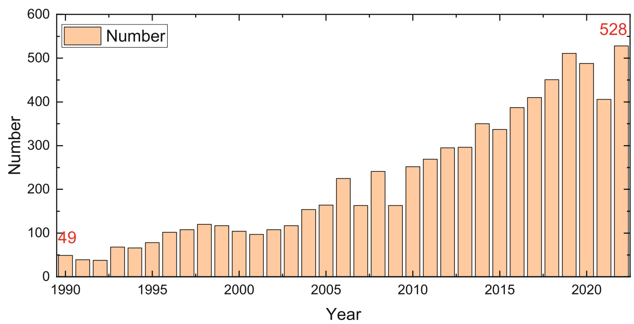

Fig. 1 Number of published papers in the term "bolted joints" at the Scopus platform.

for structures made of metal [1]. The uncertainties on the nonlinear dynamic behavior of the systems where they are involved, and therefore the amount of damping produced by vibrations, are due to the complex geometries and frictional contact of bolted joints [2]. This is especially true for lightweight structures like in aerospace applications where thin components are prone to vibrations and consequently to High Cycle Fatigue damage.

The commonest type of removable joint used in mechanical applications is the lap joint or the flange joint [3, 4], as shown in Fig. 2, which is suitable to transfer loads and globally ensure displacement compatibility of the parts through the friction force at the contact interface produced by the pressure distribution on the fastening.

The dynamic behavior of the built-up structures drew the attention of researchers since the early stage of the 20th century since there appears to be much higher damping than similar one-piece structures [5]. When build-up structures are subjected to dynamic excitations, energy dissipation (or vibration damping) may occur if relative kinematics or deformation takes place at the contact interface due to partial looseness of the joint. Within this framework, the development of mathematical models, which can be used to predict the interfacial dissipation and stiffness of single-bolted joints, is a common goal of research in the scientific community [3]. The validation of such predictive models which account for the influence of the contact parameters on the dynamic response of a jointed structure requires an extensive test campaign devoted to the measurement and control of the static loads' distribution determined by the preliminary assembly (bolt tightening) and to the vibratory response during operation (transient and forced response).

Various test rigs and experimental measurement techniques have been developed focusing on different aspects: the measurement of the preload due to the tightening procedure, the measurement of the contact parameters to model the non-linear compliance of the bolted joints, the identification of the hysteretic damping, the definition of the uncertainties and repeatability of the test, etc.

In the early stages, most of the test rigs are used in specific laboratories, and the variabilities in the used test articles, test setups, excitation systems, and acquisition procedures due to lab-to-lab or experimentalist-to-experimentalist procedures are rarely considered. Due to the huge amounts of documents found in the literature, it is not easy to divide the main topics of this investigation into a few categories, and a feeling of loss might be more rather than understanding for a researcher who is approaching the dynamics of bolted structures for the first time. To avoid trials and unignorable errors affecting the test rig's design or the accuracy of the measurement techniques, a researcher must currently update his knowledge on the current state of the art, to make further investigation.

Fortunately, workshops on joint mechanics have been held in the 21st century to overcome the

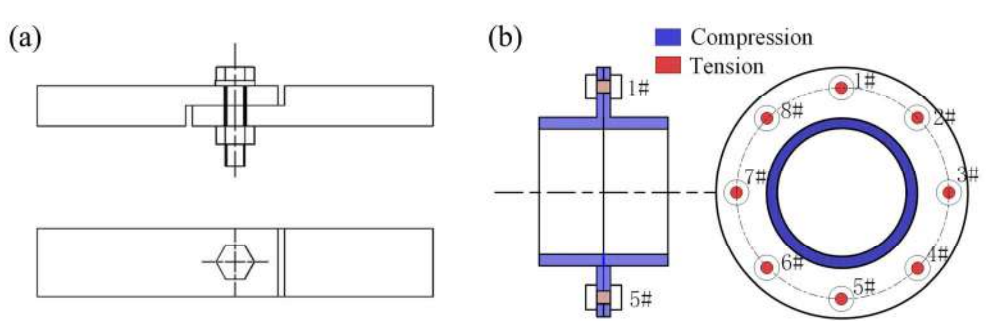

Fig. 2 Illustration of a common (a) lap joint and (b) flange joint.

difficulties and attempts to build the round-robin/benchmark for measurements of hysteresis or dissipation in standard joints have been encouraged [6]. Several experimental apparatuses, such as the dumbbell joints experiments and Brake-Reuß Beams, have been introduced in the book by Sandia National Laboratories [7] and by Brake [3]. More round-robin systems to study jointed structures and the systems designed to refine the testing techniques have been the objectives of summer schools like the NOMAD (Nonlinear Mechanics and Dynamics Summer Research Institute, Sandia National Laboratories), the Tribomechadynamics research camps, the Workshops of the Research Committee on Mechanics of Jointed Structures, specific sessions dedicated to this topic at conferences like the session "Dynamics of joints" at ISMA (International Conference on Noise and Vibration Engineering), Leuven and the Sessions included in the "Nonlinear Structures & Systems" at IMAC (A Conference and Exposition on Structural Dynamics).

Thanks to the authors of these studies, more and more experimental investigations have been carried out systematically in recent years using similar test apparatuses, and more developments on the experimental techniques for the nonlinear joints have been acquired. However, an overview of the experimental investigation on the bolted structures is still needed, through which the development history and orientation, as well as the advantages and limitations of experimental setups and techniques, can be easily acquired to start a further investigation.

The main outcome of this paper is to provide an overview of the experimental investigation of the bolted structures with frictional interfaces. Section 2 discusses the investigation of energy dissipation mechanisms for typical bolted structures, mainly focused on the achievements acquired in recent years. Section 3, introduces well-known test rigs for the measurement of the hysteresis parameters on mechanical systems exhibiting bolted frictional interfaces, and more deep comprehension of the energy dissipation characteristics can be obtained. In Section 4, typical test rigs for the investigation of the nonlinear dynamic behavior of the structure in the presence of bolted joints are described. The widely used measurement techniques, including the excitation and testing strategies, and the methods to measure the preload, will be concluded generally. Meaningful conclusions and recommendations can be acquired, which we hope can guide future experimental research on bolted structures.

2 The energy dissipation mechanisms of bolted structures

2.1 The frictional damping induced in contact interfaces

The first systematic approaches to describe the mechanism of frictional damping for practical applications can be mainly tracked in the past century: without any intention of completeness, authors think it is worth mentioning the analytical formulation by Hartog [8] (1931) for the calculation and experimental validation of a friction damper modeled according to a lumped parameter system, and the formulation of the tangential contact loads on spherical and flat punch by Mindlin (1949) [3] according to the continuum mechanics theory as an extension of the Hertz theory applied to spherical contacts. In both cases, it is assumed that relative slip would occur when the tangential stress at the interface exceeded the product of normal pressure times a constant coefficient of friction (usually known as Coulomb's friction model although Guillaume Amontons formulated the same equation circa 80 years before Charles-Augustin de Coulomb's studies).

The experimental proof of the mentioned behavior is reported in the works of Johnson [9] and Goodman [10], while the following studies showed that the damping is due to energy dissipation using dry friction in joints under the action of shearing forces [11]. Indeed, the investigation made by Andrew [12] suggests that the vibration normal to the joint interfaces is generally low undamped, and the partial slip is generally caused by the tangential relative motion of the contact surfaces. The experiments carried out by Beards [13] showed that structures can be damped by using correctly fastened joints to allow controlled interfacial slip during vibration. The work of Earles and Philpot [14] is concerned with the energy loss resulting from the relative slip at a flat, metallic contact interface because of the mutual elastic elongation of the joint members, but seemingly disregards the energy loss due to microscopic deformation of the surface asperities.

Nowadays, it is widely accepted that macro-slip, partial slip, and micro-slip at the contact interface between two vibrating bodies are responsible for damping [15]. Bolted joints consist of a mechanical system assembling two or more parts by the insertion of a screw and the screw thread with the nut. Constraint is determined by the tightening torque that produces a geometrical interference along the direction orthogonal to the contact surfaces, which prevents global relative motion by friction. This leads to a special pressure distribution at the interface between the parts that are tightened together, which is maximum in the vicinity of the bolt hole, and will rapidly decrease with the distance away from it [16, 17] (Fig. 3) with the formation of the classical receding contact between parts.

The macro-slip is the gross slipping that takes place on a macroscale level between two bodies having relative motion, which is assumed to be held by Coulomb's law of friction. In the case of joints with a large clamping pressure, sliding on a macroscale level is normally prevented, and only local, and partial slip (or micro-slip if the region is a small fraction of the whole) is initiated at the contact, which involves very small relative displacements of the asperities at the contact interfaces. A further increase in the joint clamping pressure will cause greater penetration of the asperities, which further limits the relative motion causing the sticking status [18].

Considering the distribution of the normal pressure at the bolted interfaces, it is believed that an in-plane relative motion at the contact interface will cause micro-slip in the regions where the normal pressure is not so large (Fig. 4) [19]. In the case of cyclic excitation, the peripheral micro-slip region in a bolted joint causes energy dissipation and hence damping. The roughness of the actual contact (Fig. 5) area makes the frictional contact more complex, which may decrease the nominal contact surface in a considerable number of relevant cases [20]. This fact is of tremendous practical importance since the size and shape of the real contact area affect a large number of physical properties. The contact between two surfaces with roughness can be modeled as an equivalent rugged surface contacting a smooth rigid plane, and with a large number of separate asperities of different sizes spreading over the irregular surface contacting the rigid plane [21]. And a variety of measurement technologies, such as the ultrasonic method and the digital image correlation, have been developed for this purpose during the last decades.

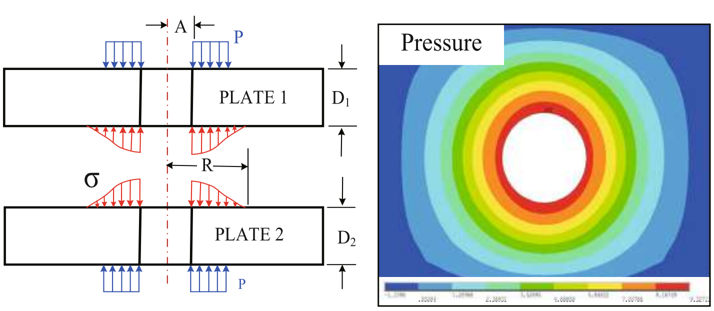

Fig. 3 Qualitative pressure distribution of bolted joint.

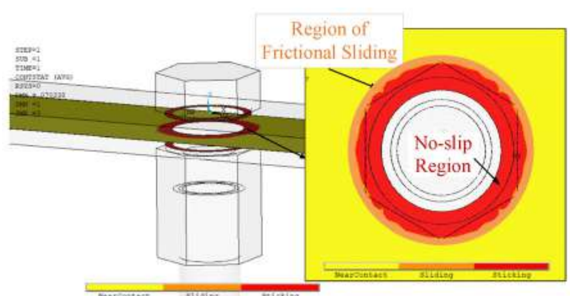

Fig. 4 Slip in the contact patch.

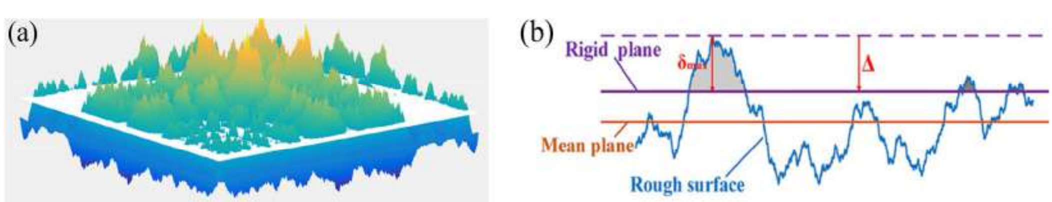

Fig. 5 (a) Rough surface (in color) with a smooth rigid plane (white), and (b) the 2D section of the surface. Reproduced with permission from Ref. [21], © Springer Nature 2017.

2.2 Typical damping model for structure with joints

Plenty of research has been done in the past decades to develop mathematical models, for the prediction of the amount of energy dissipation due to the friction of joints. A thorough review of these can be found in the work of Mathis [22], S. Bograd [23], and Wentzel [24], et al. In this review paper, typical damping models are listed in Table 1, while the features are concluded.

2.3 The key parameters and the measurement techniques

As already mentioned, the presence of slip phenomena at the frictional interfaces of assembled structures is

Table 1 Typical damping model for structure with mechanical joints.

No.	Model type	Name	Feature
1	Linear damping models	1.1 Linear phenomenological model	• One of the oldest and perhaps most popular methods of modeling dissipation [25]. $F = c\dot{u} + ku,\quad c = (2\zeta\omega_n)m$
		1.2 Viscoelastic constitutive models	• Combinations of viscous and elastic elements are used to represent realistic viscoelastic properties of continua [26].
		1.3 Integral damping models	• Formulated by describing the damping force as a function of the cumulative history of the system [27]. $F = \int_0^t G(t-\tau)\dot{u}(\tau)\mathrm{d}\tau$
2	Rate-independent models	2.1 Duhem models	• A general model for the evolution of rate-independent hysteretic internal variables of the form [28]. $\dot{z} = \begin{cases} g_1(u,z), & \text{if } \dot{u} > 0 \\ g_2(u,z), & \text{if } \dot{u} < 0 \\ 0 & \text{if } \dot{u} = 0 \end{cases}$
		2.2 Bouc model	• Bouc investigated the mathematical theory of hysteresis in the late 1960s and early 1970s, which led to the study of the class of functional operators [29]. $F = [\mathrm{B}(u)](t) \equiv ku(t) + z(t)$ where $z \equiv \int_0^t G(v(t,s))\phi(u(s))\dot{u}(s)\mathrm{d}s$
		2.3 Bouc–Wen model	• A class of semi-physical phenomenological models that captures both the linear-elastic and elastoplastic restoring forces in systems that exhibit hysteretic phenomena [30]. $F = k_f u + f(z),\quad f(z) = (k_j - k_f)z$
		2.4 Prandtl–Ishlinskiĭ model	• Constructed from combinations of individual, discrete elastoplastic elements [31]. $F = ku + \mathrm{sgn}(\dot{u})\phi = \begin{cases} ku + \phi, \text{if } \dot{u} > 0 \\ ku - \phi, \text{if } \dot{u} < 0 \end{cases}$
		2.5 Massing model	• Studied material plasticity using a parallel arrangement of elastoplastic elements [32].
		2.6 Iwan model	• Iwan conceptually expanded on the Masing model to include several other possible arrangements [33]. • Masing model is classified as a parallel-series Iwan model.
		2.7 Dahl model	• An electromechanical circuit that successfully modeled the hysteretic phenomena due to material plasticity [34].

(Continued)

No.	Model type	Name	Feature
3	Rate-dependent models	3.1 Tribological models	• The most popular of these, referred to as Amontons and/or Coulomb's laws of friction [35]. $$F=\mu N \operatorname{sgn}(\dot{u})$$
		3.2 LuGre	• An extension of the model that incorporates the Stribeck effect, velocity dependence on friction in the transition between static and Coulomb (dynamic) friction [36]. $$F=\sigma_0 z+\sigma_1 \dot{z}+f(\dot{u}),\quad \dot{z}=\left(1-\operatorname{sgn}(\dot{u})\left(\frac{z}{g(\dot{u})}\right)\right)\dot{u}$$
		3.3 Elastoplastic friction model	• Developed by Dupont et al. [37], which attempts to further reflect the original hypothesis of Dahl. $$F=\sigma z+\sigma_1 \dot{z}+\sigma_2 \dot{u},\quad \dot{z}=\left(1-\alpha(z,\dot{u})\frac{z}{z_{ss}(\dot{u})}\right)\dot{u}$$
		3.4 Leuven model	• Developed by Swevers et al. [38], considering the micro-damping term in the expression for F. $$F=F_h(z)+\sigma_1 \dot{z}+\sigma_2 \dot{u},$$ $$\dot{z}=\left(1-\operatorname{sgn}\left(\frac{F_d(z)}{s(\dot{u})-F_b}\right)\left|\frac{F_d(z)}{s(\dot{u})-F_b}\right|^n\right)\dot{u}$$
		3.5 Lampaert model	• To simplify the Leuven model, using memory stacks to store and update reversal points [39].
		3.6 Valanis model	• A model for end chronic plasticity, a stick–slip element with postslip stiffness [40].

responsible for damping. The pressure distribution caused by a bolted joint, the levels of excitation, and the friction coefficient are the most important factors that determine the slip forces and therefore the amount of friction damping at the contact interface. The experimental investigations by Nanda and Behera [41] and Masuko et al. [42] showed that the intensity of the normal load, surface roughness, bolt diameter, presence of washers, material type, and amplitude of excitation play important roles in damping capability.

It is of vital importance to measure the excitation force and the preload of the bolt (or the pressure at the interfaces) accurately since they play a crucial role in the property of the bolted joints. The force sensors are used by almost every researcher to measure the excitation force applied to the structures, as shown in Fig. 6. The force sensors and the instructions

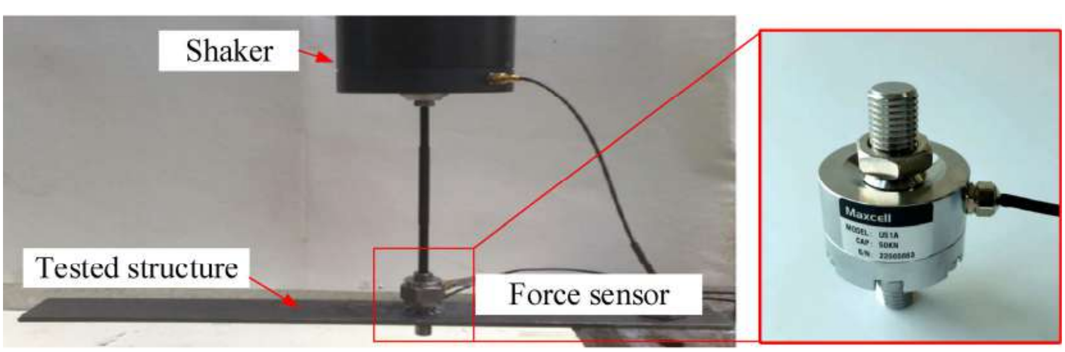

Fig. 6 The typical force sensor and the methods of application.

are easy to acquire from commercial companies, while the applied force can also be controlled by some commercial testing system (such as LMS Test. Lab), which won't be repeated here.

Bickford [43] concluded some basic methods to control the preload in the assembly process of the bolted joints, including the torque control, torque and turn control, and stretch control method. Using these traditional methods, the preload can be indirectly controlled in assembly, but the exact value can't be acquired in operation. Typical measurement techniques for the preload of the bolted joints are listed in Table 2, including the strain gauges, indicating washer, pressure films, and ultrasonic techniques, which are suitable for different structures and purposes.

The strain gauges are one widely used direct measurement method to monitor the preload of bolts, especially for the joints under dynamic loads. There are mainly two ways to use strain gauges, as shown in Fig. 7: one is bonding the strain gauge on the outside of the bolt, and two strain gauges are used on both sides of the bolt shaft in axially symmetric positions to cancel the influence of bending [44]; second, the strain gauges have been equipped with copper leads and can be used directly in pre-drilled bolt elements with exclusive adhesives. Both techniques have been promoted by commercial companies, but the method with two strain gauges outside the bolt was used in the labs more widely, for it is more convenient and economical to be applied by self.

Table 2 Measurement techniques for the preload of the bolted joints.

No.	Measurement techniques	Features
1	Strain gauges—outside	• Monitor the strain of the bolt • Used in the labs more widely • More convenient and economical to be applied by self
2	Strain gauges—into the bolt	• Directly in pre-drilled bolt elements • Promoted by commercial companies
3	Preload indicating washer	• Measure the applied compress load • Promoted by commercial companies
4	Pressure films	• Distribution of the pressure at the interface • Extremely thin, non-invasive, and highly economical • Promoted by commercial companies
5	Ultrasonic techniques	• Monitor the variation of the preload of bolt • Ultrasonic wave propagated depends on axial length

The preload or pressure-indicating washer [45], as well as the electronic pressure films [46], were used by more and more researchers in recent years, as shown in Fig. 8. They can be used under the head of a bolt or the nut to measure the applied compress load or contact pressure within a bolted interface during dynamic excitation [43]. The pressure sensor is extremely thin, non-invasive, and highly economical, and can reveal precisely not only the magnitude but also the distribution of the pressure (tensile load) at the interface of the bolt. But the washer or pressure films may also influence the damping characteristics of the interfaces, thus necessary contrast tests are needed sometimes in experimental studies using these measurement methods.

Ultrasonic techniques (Fig. 9) used for axial force monitoring in bolts are based on the fact that the velocity of the ultrasonic wave propagated along the bolt depends on the axial stress [43, 47]. As the bolt is tightened, the amount of time required for the ultrasound to make its round trip increases for the path length increases as the bolt is tightened, and the average velocity of sound within the bolt decreases

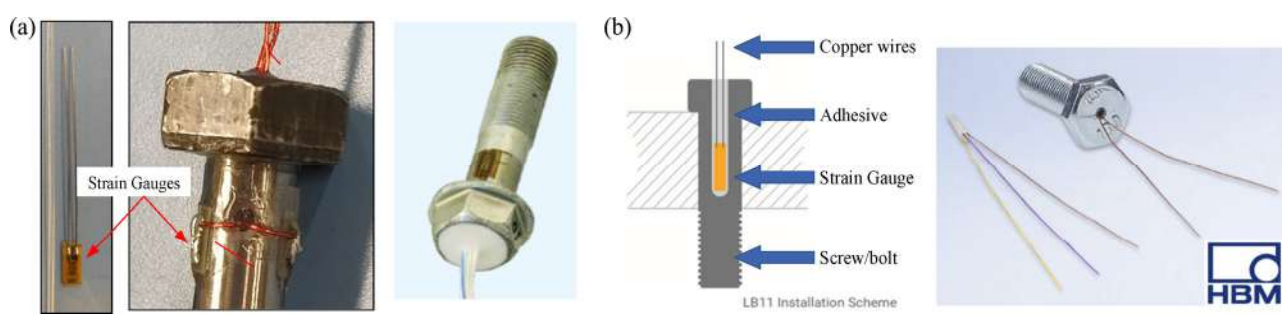

Fig. 7 Measurement techniques using the strain gauges, (a) outside the bolt, (b) inserted into the bolt.

Fig. 8 Measurement techniques, (a) preload indicating washer, (b) pressure indicating washer, and (c) pressure film.

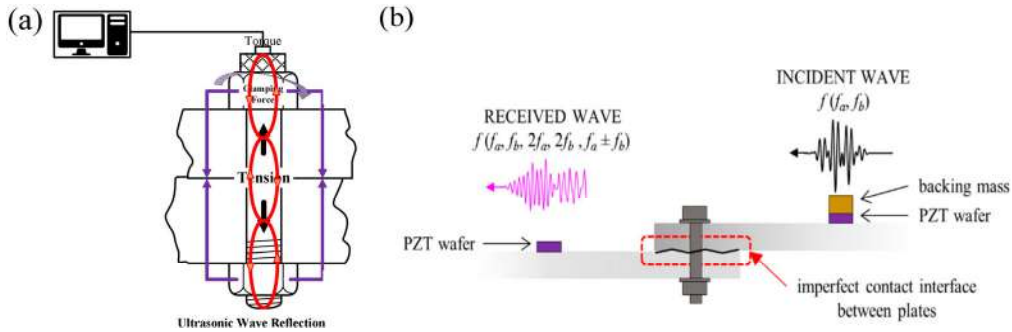

Fig. 9 Ultrasonic techniques, (a) Axial force monitoring, (b) Contact acoustic acoustic nonlinearity. Reproduced with permission from Ref. [49], © by the authors. Licensee MDPI, Basel, Switzerland 2021.

because the average stress level has increased. Methods based on guided waves (GWs) have further been developed to indicate the bolt's contact status, by demonstrating that the energy propagated across the bolt [48]. Moreover, some studies have investigated the use of mixed-frequency responses for monitoring bolted joints, using combined-frequency responses for assessing the condition of the bolted joint (Fig. 9(b)) [49, 50].

3 Measurements of the hysteresis damping

3.1 The basic theory for hysteresis damping testing

When subjected to a cyclic motion, a hysteretic behavior of the friction force concerning the relative displacement is produced (Fig. 10) [51]. Such curves are often used to extract the equivalent stiffness and damping properties of frictional interfaces to build the dynamic model (such as the Jenkins model, as shown in Fig. 11, the Iwan model, or the Valanis model) for the bolted structure [52]. Thus, test rigs that can measure the hysteresis characteristic of interfaces in different conditions are widely needed, which was also proposed by the ASME Research Committee in 2019 [53].

Hysteresis damping (Fig. 10) is one of the most widely used concepts to represent the frictional characteristics in the interfaces, which could be

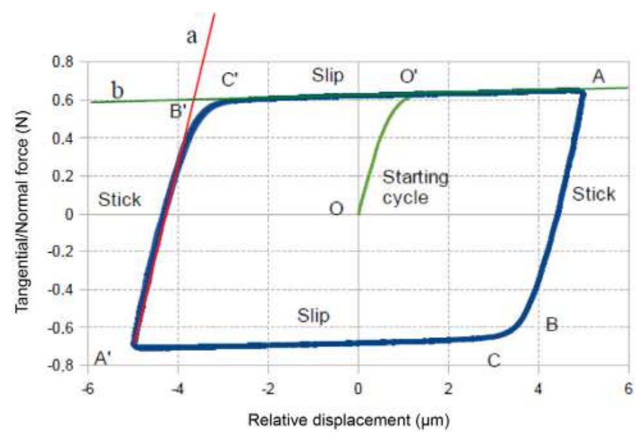

Fig. 10 Schematic of the hysteresis curve. Reproduced with permission from Ref. [51], © ASME 2012.

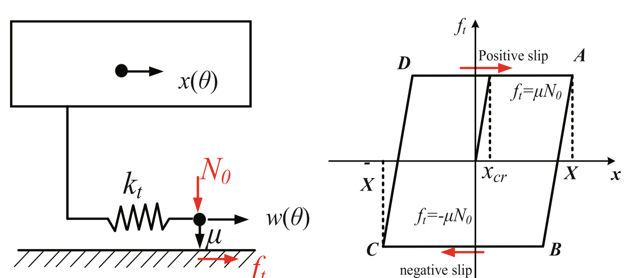

Fig. 11 Schematic of the Jenkins contact element.

represented by the loss factor:

$$\eta = \Delta E / 2\pi U_{\max} \quad (1)$$

Where, ΔE is the energy dissipated per period of vibration, which equals the area of the hysteresis curve; while $U_{\max}$ is the maximum potential energy. Then, the damping coefficient can be obtained,

$$\xi = \eta/2 \quad (2)$$

And, the slope of the hysteresis curve is the tangential stiffness k_T of the interface.

It is believed that the energy dissipated by the joints subjected to the oscillating load is independent of frequency in bolted structures, and hence the energy loss per cycle can be measured at zero or any other frequency. The tests performed at or near zero frequency are called static hysteresis tests, while the testing system excited by shakers at other frequencies are dynamic hysteresis tests [54, 55].

3.2 Round robin systems to acquire the hysteresis parameters

It is commonly believed that the energy dissipation mechanism in bolted joints is mainly caused by the slip in the tangential direction [3], thus test rigs with isolated bolted joints excited in the tangential direction may be an effective way to get the hysteresis characteristics of joints.

3.2.1 Test rigs with directly applied tangential force

The tangential force and the displacements along the force are two key parameters to form the hysteresis curves. The quasi-static tests based on the universal testing machine (Fig. 12) may be the simplest way to get the hysteresis curves of joints, carried out by Ferrero [56] and Abad [57] et al. The bolted plates are settled on the universal testing machine with dynamic control, as shown in Fig. 12(b); The preload of the bolt is applied by the dynamometric spanner and measured using a loading cell connected to a portable bridge of Wheatstone model P3 (Fig. 12(c)). Benefiting from the convenience of the quasi-static test, it has been widely used to identify the hysteresis parameters [58].

The sinusoidal excitations are normally applied by the shakers in the dynamic hysteresis test, while the excitation force is measured by the force transduces. The displacements in tangential direction can be measured directly by Laser Doppler Vibrometers (LDV) and capacitive displacement sensor or can be acquired through a double integration of the acceleration signals tested by the acceleration pickup, which will be explained in detail later.

Eriten et al. [59] obtained the hysteresis curves for a bolted joint by using the apparatus mainly designed for fretting experiments on mechanical lap joints. As shown in Fig. 13, a piezo actuator is used to impose a motion in the tangential direction, while a tri-axial load cell is applied to measure tangential force as well as possible misalignment forces.

Similar test apparatus were also developed and used to measure friction hysteresis by Li [60], which is shown in Fig. 14, and the friction force and the bolt preload were continuously measured during the test. The numerical model was developed to extract

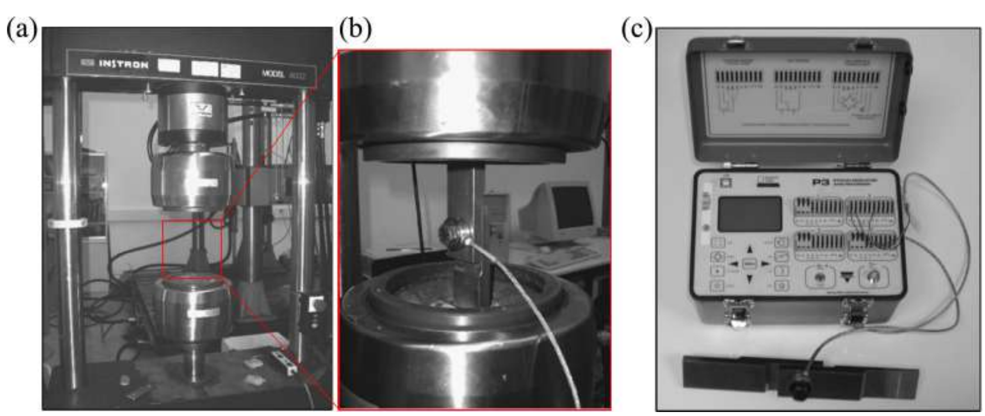

Fig. 12 Quasi-static tests, (a) equipment, (b) assembly in the machine, and (c) instrumentation. Reproduced with permission from Ref. [57], © Elsevier Ltd. 2011.

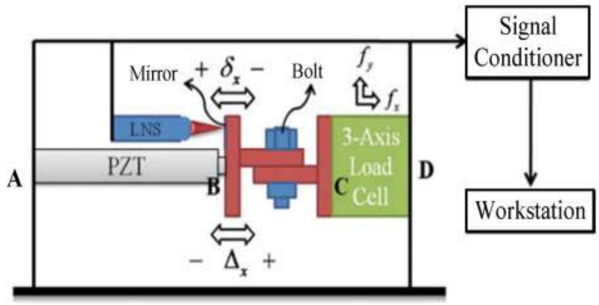

Fig. 13 The apparatus designed by Eriten. Reproduced with permission from Ref. [59], © Society for Experimental Mechanics 2011.

the tangential contact stiffness from the measured data, and the fretting wear behaviors have also been experimentally studied as the vibration loading is applied [61].

The relative motion between the joint (the slip distance) was measured by halves Laser Nano Sensor (LNS) to obtain more accurate data. The apparatus satisfies the following four design requirements specific to fretting of mechanical joints: 1) high reliability of the applied displacement or applied force, 2) high accuracy of interfacial displacement measurements, 3) high accuracy of interfacial force measurements, and 4) minimal misalignment and out-of-plane motions [62].

The hysteresis curves of the joints vary along with the excitation, and the bolt torque, as shown in Fig. 15. Results show that the hysteresis curves are much fuller under a higher excitation force, which means more damping can be induced [63]. But the influence of the preload seems to be more complex, and there is a certain preload that makes the damping maximum, as shown in Fig. 15(b).

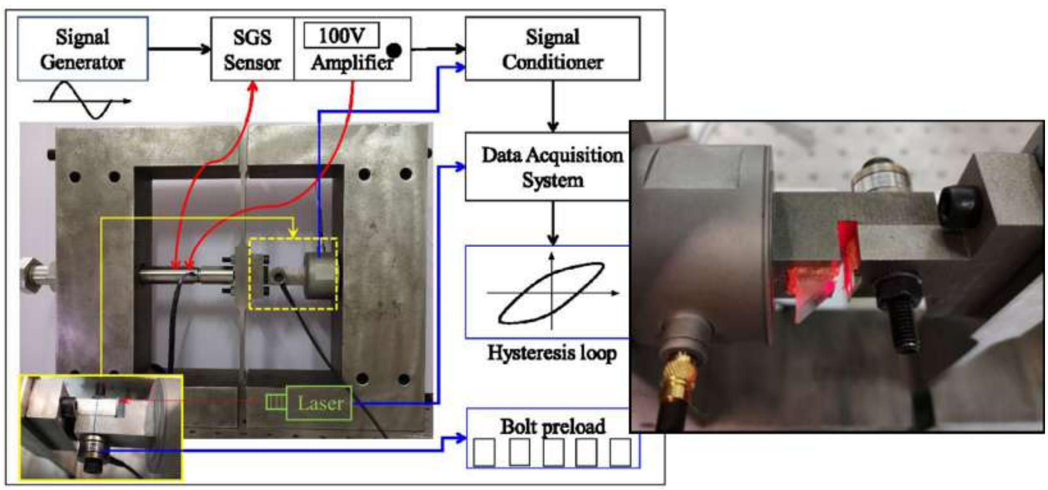

Fig. 14 The developed test apparatus in Ref. [60]. Reproduced with permission from Ref. [60], © Elsevier Ltd. 2020.

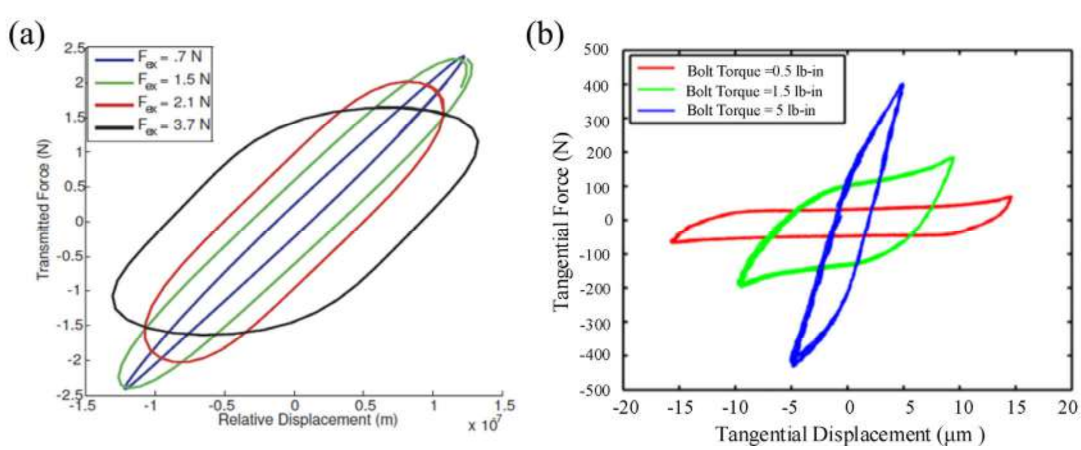

Fig. 15 Hysteresis loop with varied (a) excitation, and (b) bolt torque. Reproduced with permission from Ref. [62], © Elsevier Ltd. 2012.

3.2.2 Resonator with a larger excitation force

The tangential force applied by the piezo actuator or a shaker is limited, which brings the challenge to get the hysteresis curves in a larger excitation force. Sanati et al. [64] identified the stick-slip properties of the joint using a mechanical resonator, as shown in Fig. 16, where a lumped mass with a linear spring is attached to the lap joint structure. The sinusoidal excitations were applied by a sharker in a frequency band around the resonance frequency for different excitation levels, thus the applied excitation force will be enlarged due to the resonance, which will be measured by the force sensor between the sharker and the lumped mass. The displacements at the mechanical resonator and bolted beam were also measured to form the hysteresis loop, using a fiber-optic displacement sensor and a wide frequency bandwidth capacitive sensor, respectively.

The Gaul resonator (Fig. 17) is another widely used test rig to get the hysteresis parameters of bolted lap joint [65]. It consists of two masses connected by a means of flexural spring and a bolted joint, resulting in an oscillatory system, and the normal pressure at the lap joint contact interface is guaranteed by bolt tightening. A flexible connecting rod placed between the excitation point at the force pickup and the shaker assures the applicated tangential excitation, and makes the structure well isolated from other experiments hardware. The Gaul resonator was also used in the research work of Maren [66] and Ma [67], et al, and influences of normal contact force and frequency dependence were examined.

There are two widely used methods to set the Gaul resonator into vibration and obtain the damping parameters of the interfacial joints: one is using the shock excited by a hammer or the shaker, and the other is the sine sweep or stepped sine measurement [3].

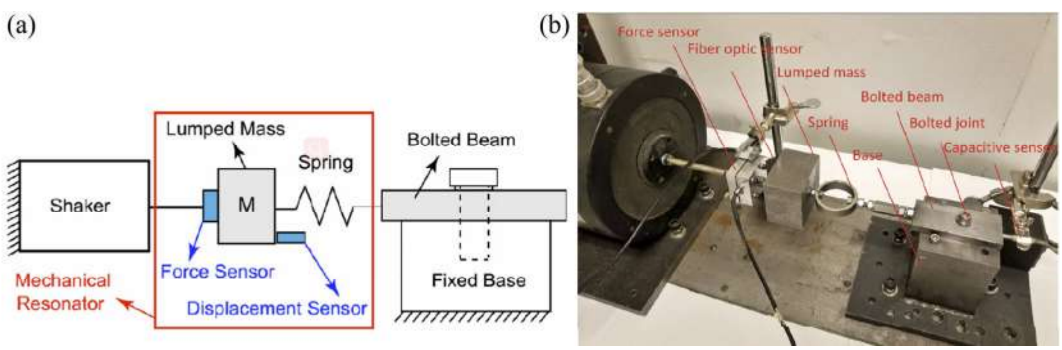

Fig. 16 The test rigs used by Sanati et al. [64]: (a) schematic of the developed approach and (b) experimental setup. Reproduced with permission from Ref. [64], © Springer Nature 2018.

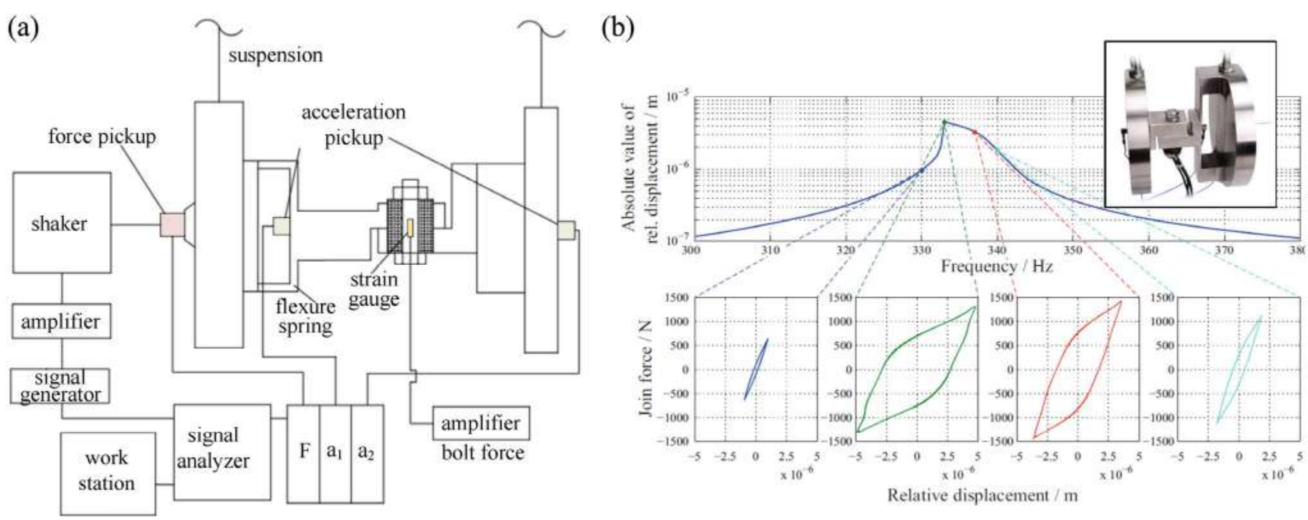

Fig. 17 Longitudinal resonator developed by Gaul and Lenz [65]: (a) Sketch of the setup and (b) relative joint displacement spectrum and corresponding friction hysteresis. Reproduced with permission from Ref. [65], © Springer-Verlag 1969.

The accelerations $a_1(t)$ and $a_2(t)$ at different points of the system (Fig. 17) and the driving force are measured by piezoelectric pickups. This can also be obtained using laser doppler vibrometer techniques. The tangential force F_t is equivalent to the product of the acceleration $a_2(t)$ and the mass of the right part of the resonator [68].

Figure 17(b) shows the friction hysteresis obtained by a stepped sine measurement, under different excitation frequencies [3]. It can be seen that the hysteresis increases (opens up) for excitations around the resonance frequency, where the excitation force is larger, and sliding of the contact interfaces may occur. The enclosed area of a hysteresis corresponds to the energy dissipated in the bolted joint, which limits the response amplitude and leads to the typical flat top peaks of the FRFs.

A similar setup called the dumbbell oscillator was developed in Sandia National Laboratories [7], which also features a single bolt lap joint that connects two mass dumbbells (Fig. 18). The dumbbell oscillator is a basic design, with a quite high-frequency range because of the absence of a spring structure, which can be modified by a relatively soft spring, leading to the same structure as Gaul.

Goyder et al. [69] gave another type of resonator for the experimental identification of damping. In this arrangement, a rectangular spring connects two swinging masses in resonance conditions through bolts (Fig. 19). The spring element is made from a welded rectangular frame, which is simple to construct and straightforward to analyze. Different test rig configurations can be used to show a correlation between the number of bolts and the damping [70].

4 Parameters identification of bolted structures

Excitation forces are not always applied in the tangential directions for realistic engineering structures. It is important to identify the dynamic parameters of bolted structures under different excitation forces and with different modes of vibration, which will be introduced in this section.

4.1 Strategies to identify the dynamical parameters

4.1.1 Oscillation decay curve method

Decaying oscillation appears when the exciting force is suddenly switched off or otherwise disengaged from the system. The decay rate of the transient response is related to the damping (or the energy dissipation) in the structure, which can be post-processed to derive a curve describing the energy dissipation dependence of damping on force amplitude [7], as shown in Fig. 20. In nonlinear frictional interfaces, the rate of decay, or the damping per cycle, appears to vary with time since they are governed at least partly by the force in the joint at any particular time.

The detailed derivation processes of the equivalent damping factor and the energy dissipation based on the free decay curve have been given out in the handbook of Sandia National Laboratories [7]. The equivalent damping factor for a linear SDOF system, ζ, is the slope of the envelope ($\log(x)$ as a function of t) divided by the resonance frequency ω_r.

$$\zeta = -\frac{1}{\omega_r}\frac{\mathrm{d}(\log(x))}{\mathrm{d}t} \tag{3}$$

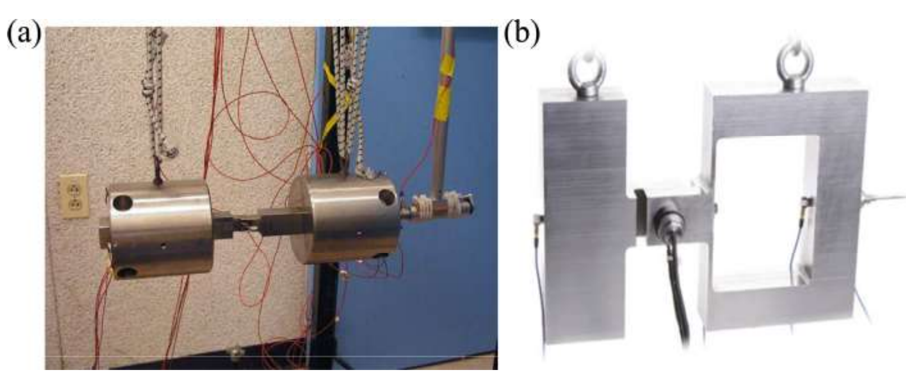

Fig. 18 Experiment setup, (a) Dumbbell and (b) improved version. Reproduced with permission from Ref. [3], © Springer International Publishing AG 2018.

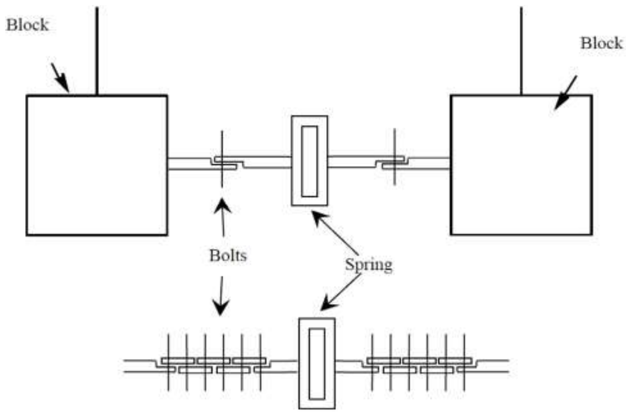

Fig. 19 Configuration used by Goyder et al. [69]. Reproduced with permission from Ref. [69], © ASME 2012.

The envelope of the transient decay can be established by taking the Hilbert transform of the time response, H = Hilbert(x), and then computing the envelope as $E=\sqrt{H^2+x^2}$. And to study the modal damping of a specific mode, the structural response was filtered close to that mode's frequency using a 10th order Butterworth filter in Matlab in Hartwigsen's studies [71]. Using the filtered data, the envelope decay computations for each of the modes could be carried out, and post-processing for studying the modal damping factors could be performed.

It is convenient to excite the structure using a harmer, and only the dynamic responses need to be measured for the analysis by accelerometers, LDV, or any other suitable methods, while the input forces were usually measured by the integral force transducer mounted on the impact harmer to study the non-linear effects caused by different excitation forces.

4.1.2 Stopped-sine analysis

Impulsive force can be easily applied by a hammer, but the dynamic response excited is composed of several natural frequencies, which will complicate the signal processing, and it is difficult to control the value of the force. A shaker allowing for a consistent application of force is suggested, which can be used at a certain natural frequency to simplify post-processing techniques. The free-decay response of the system can be recorded after the shaker is switched off, as shown in Fig. 21, which is also called the stopped-sine analysis [72].

Techniques used to analyze the nonlinear damping in the oscillation decay curve method, such as the Hilbert transform, can also be used for the stopped-sine analysis. The method was introduced in detail in Ref. [72] by Dion et al., which was used to measure the damping induced by micro-sliding through a cut-beam. The continuous wavelet transform (CWT) of the recorded free-decay signal is calculated in the experimental studies of the nonlinear bolted beam by Peeters et al. [73], which is used to identify the instantaneous frequency and damping, as shown in Fig. 22.

4.1.3 Frequency response functions method

It may be the most common method to study the dynamic properties of the structure through the frequency response functions (FRFs), based on which the damping of the structure can be identified using some strategies such as the half-power bandwidth

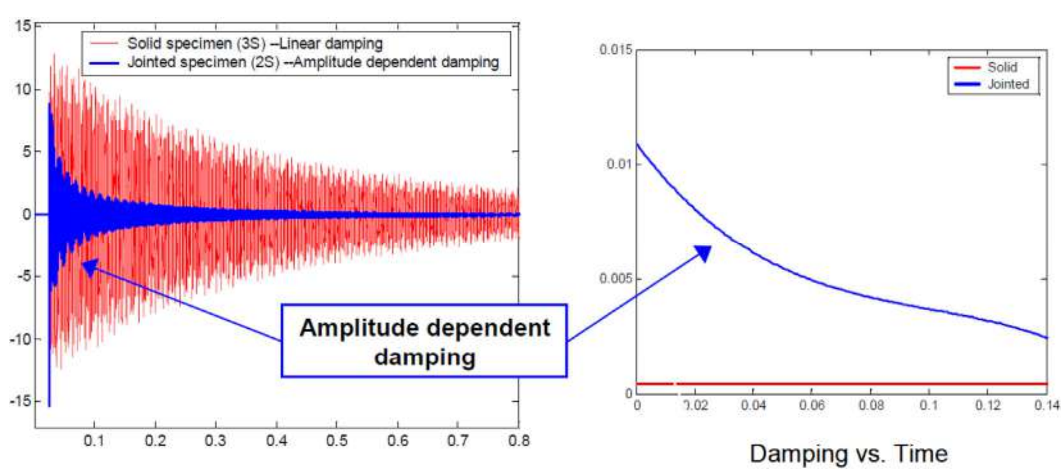

Fig. 20 Jointed interface damping effects on decay rate. Reproduced with permission from Ref. [7], © Sandia National Laboratories.

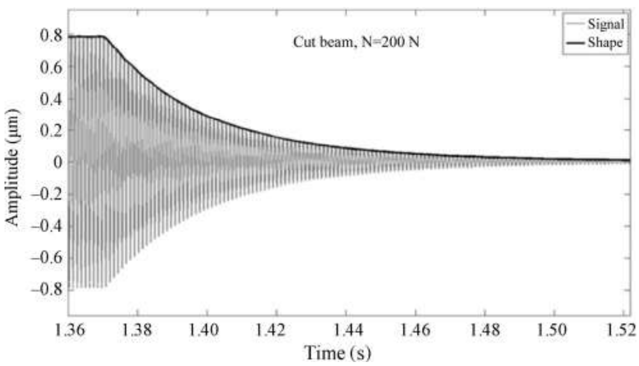

Fig. 21 Free-decay response of the cut-beam system. Reproduced with permission from Ref. [72], © Elsevier Ltd. 2012.

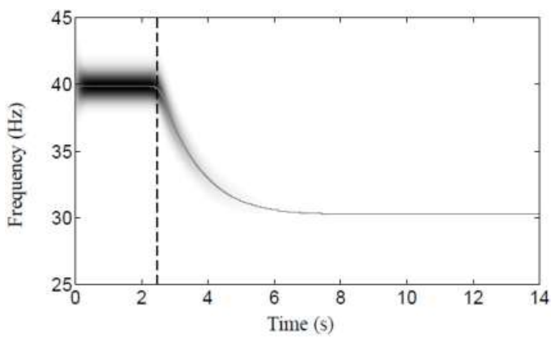

Fig. 22 Evolution of the instantaneous frequency [73].

method. As shown in Fig. 23, the damping ratio can be derived as

$$\zeta = \frac{\omega_2 - \omega_1}{2\omega_r} \tag{4}$$

Where, ω_r is the resonance frequency, Q is the dynamic response peak at ω_r, while the ω_1 and ω_2 are the frequencies at the half-power value $Q/\sqrt{2}$.

Different experimental techniques have been developed in last decades to acquire the FRFs, including

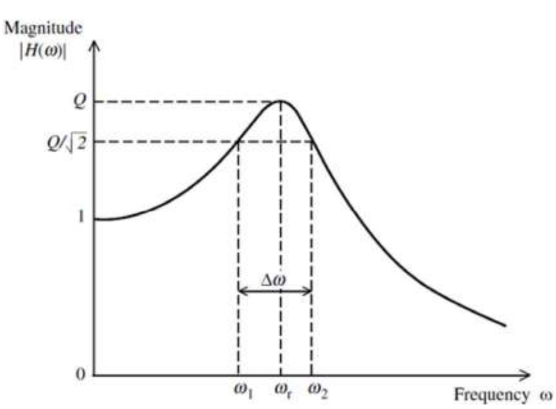

Fig. 23 Determination of damping from FRFs. Reproduced with permission from Ref. [74], © Elsevier Ltd. 2003.

the impact hammer test, random noise excitation tests, sine sweep, stepped sine measurement, and so on, as shown in Fig. 24.

Basic excitation and testing strategies for the impact hammer test are just similar to that for the oscillation decay curve, and the measured exciting forces and responses are transformed into the frequency domain by using the algorithms like fast discrete fourier transform (FFT) [75]. For linear systems, the impact hammer test and the random noise excitation tests can obtain the FRFs with limited experimental efforts, but it is not suitable for the nonlinear bolted structure for the response typically depends on the input force. Impact hammer testing can lead to a more complex behavior at the interface due to the nonlinear coupling of multiple modes excited by the impact hammer, and a repeatable excitation force is harder to achieve (in terms of magnitude, precise location, and angle of impact).

Sweep sine is the excitation with the instantaneous frequency increase versus time, and the dynamic response of the system is assumed to be a steady-state solution for each frequency [72]. With this method, the excitation level can be precisely controlled during the tests, and experimental results showed that the resonance peak of a bolted structure tends to bend to the left hand as the excitation increase, as shown in Fig. 25, which means a stronger activation of the nonlinearity caused by the dry friction due to the higher energy level [3]. It is believed that slower sweep rates will be helpful to obtain a quasi-steady-state behavior, but it will make the measurements more time-consuming.

Though the sweep sine method is widely used, it is believed that only the time-harmonic excitation is useful for the nonlinear system. The excitation frequency is kept constant until the dynamic response of the structure gets to be steady, and the amplitude at each frequency point change with a frequency step will be recorded (Fig. 26), which is called step sine testing. More and more attention has been paid to this method to study the nonlinear characteristics caused by bolted interfaces [80].

4.2 Testing work for the beam system with lap joints

The beam with shear lap joints may be the simplest

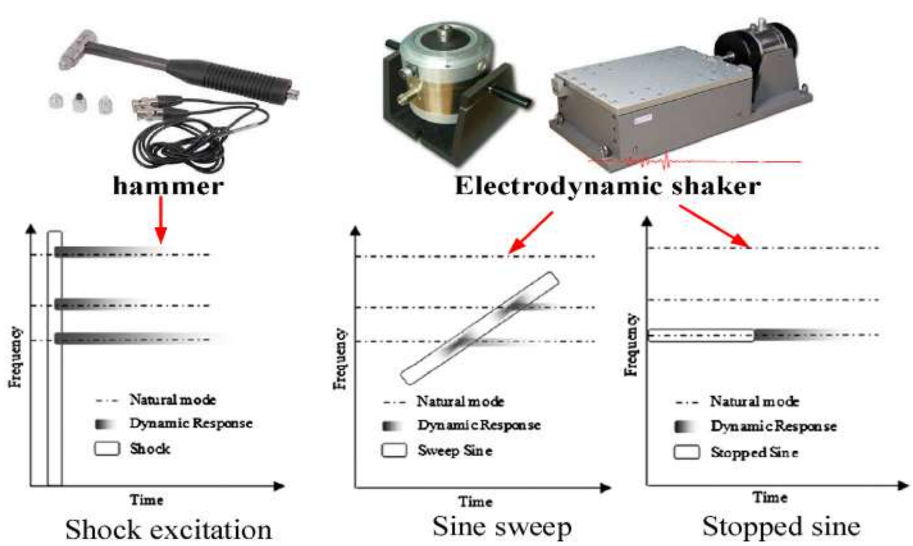

Fig. 24 Excitation rigs and features for different tests.

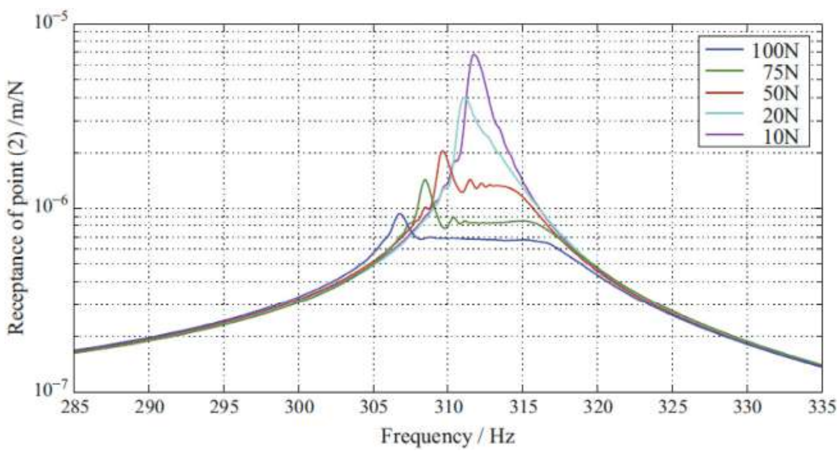

Fig. 25 FRFs of the bolted structure under varying excitation. Reproduced with permission from Ref. [3], © Springer International Publishing AG 2018.

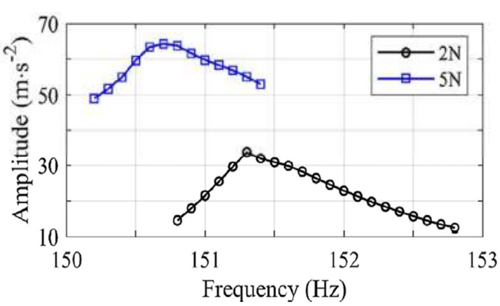

Fig. 26 Responses during the step sine testing. Reproduced with permission from Ref. [92], © Elsevier Ltd. 2019.

and most widely used structure to study the influence of the presence of a bolted connection on the system's dynamics. The effects of friction damping on the dynamic response of the bolted beam (or plate) system were tested by Pian [76] and Ungar and Carbonell [77] around the 1960s. Comparison tests between one-piece beams and built-up beams were often carried out to discern the source of damping due to joints and the material, which has been overviewed by Ferri [78] and Gaul and Nitsche [1] in different periods.

4.2.1 *Nonlinearity and the uncertainties of friction phenomena*

To identify the damping induced by the friction at the contact interface and distinguish it from the typical material hysteretic damping, comparative dynamic tests between a beam with a lap joint and its monolithic counterpart were performed by Maloney and Peairs [79] (Fig. 27). The difference in the identified dynamics of two structures is assessed by measuring the response either with accelerometers

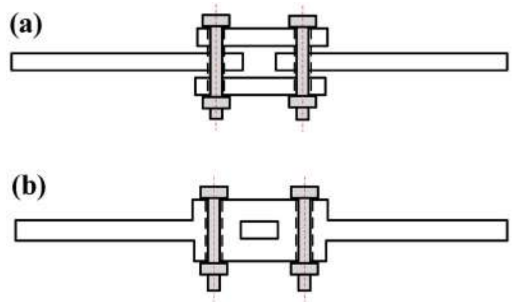

Fig. 27 Beam system: (a) jointed beam, and (b) monolithic beam.

or laser vibrometers [80], while the excitation force is provided either by a hammer or an electromagnetic shaker. The damping can be examined in an approximate local linear framework by using the log decrement and half-power bandwidth approaches, or a nonlinear framework using a hybrid surface irregularities model [79]. Testing work using a similar beam system were also done by Zaman et al. [81] and Esteban and Rogers [82], through different testing methodologies such as modal analyses, time-domain response, or energy dissipation analyses.

Eriten et al. [83] performed the nonlinear system identification (NSI) of the effects of frictional connections in the dynamics of a bolted beam assembly, as shown in Fig. 28. They brought an interesting discussion about the global effects that friction phenomena can have on structures in a vibration regime, by isolating and identifying the frictional effects of the lap joint on the structural dynamics.

In the experimental study of Eriten et al [83], three different impulsive excitations are presented, and the velocity time series have been measured at each sensing position of the bolted beam, as shown in

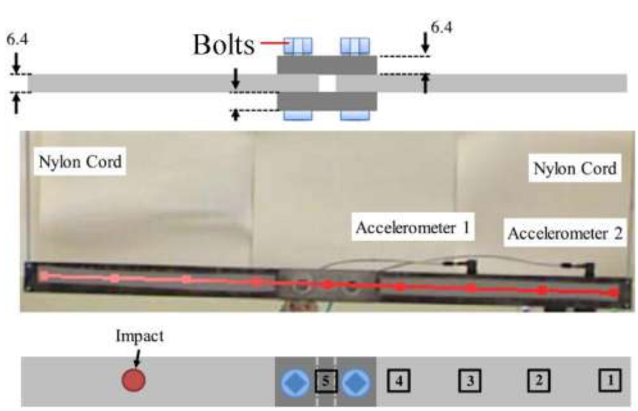

Fig. 28 Bolted beam used in Ref. [83]. Reproduced with permission from Ref. [83], © Elsevier Ltd. 2013.

Fig. 29(a). The analysis was performed by decomposing measured time series using empirical mode decomposition (EMD) and modeling the resulting dominant intrinsic modal functions (IMFs) in terms of sets of intrinsic modal oscillators (IMOs). The energy applied to each mode (IMO) is estimated at the time instant immediately after the application of the hammer excitation, which scales with the square of the measured initial velocity of the mode at the point of impact. Results show a significant increase in the friction-induced damping nonlinearity for the odd-order modes when the larger impulse is applied since the damping ratios increase with increasing energy. But on the contrary, even-order modes are not influenced much by the varied impulse, confirming the small influence of frictional effects on the dynamics of these modes. Based on the developed methodology, the nonlinear damping effects of the frictional interface and distribution along the span of the beam were estimated, while the dependencies of these nonlinear effects on the level of energy can be analyzed.

Uncertainty is another major effect of contact interfaces besides nonlinearity, which is currently a widely discussed topic, such as that investigated by Ewins [84]. There are two quite distinct types of uncertainty: aleatoric and epistemic. Aleatoric uncertainty relates to imprecision or a lack of knowledge of the precise values, thus it is a matter of seeking more precise estimates of the parameters, by making repeated measurements or specifying tighter manufacturing tolerances. Epistemic uncertainty refers to the inadequacy or incompleteness of a set of parameters that are used to describe behavior, which can rarely be addressed by making repeat estimates of the initial parameter set, since it may not be at all obvious which parameters are missing. Varied stochastic linear/nonlinear models have been developed for such uncertain structures based on the dynamic response analysis or system parameter identification of bolted structures.

The double-bolted joint benchmark was proposed by Jalali et al. [85] and Teloli et al. [86], as shown in Figs. 30 and 31, to experimentally illustrate the uncertainties in bolted joint dynamics. The modal parameters of bolted beams were identified by the peak-picking modal parameter estimation approach [87], and the effects of variability in the surface

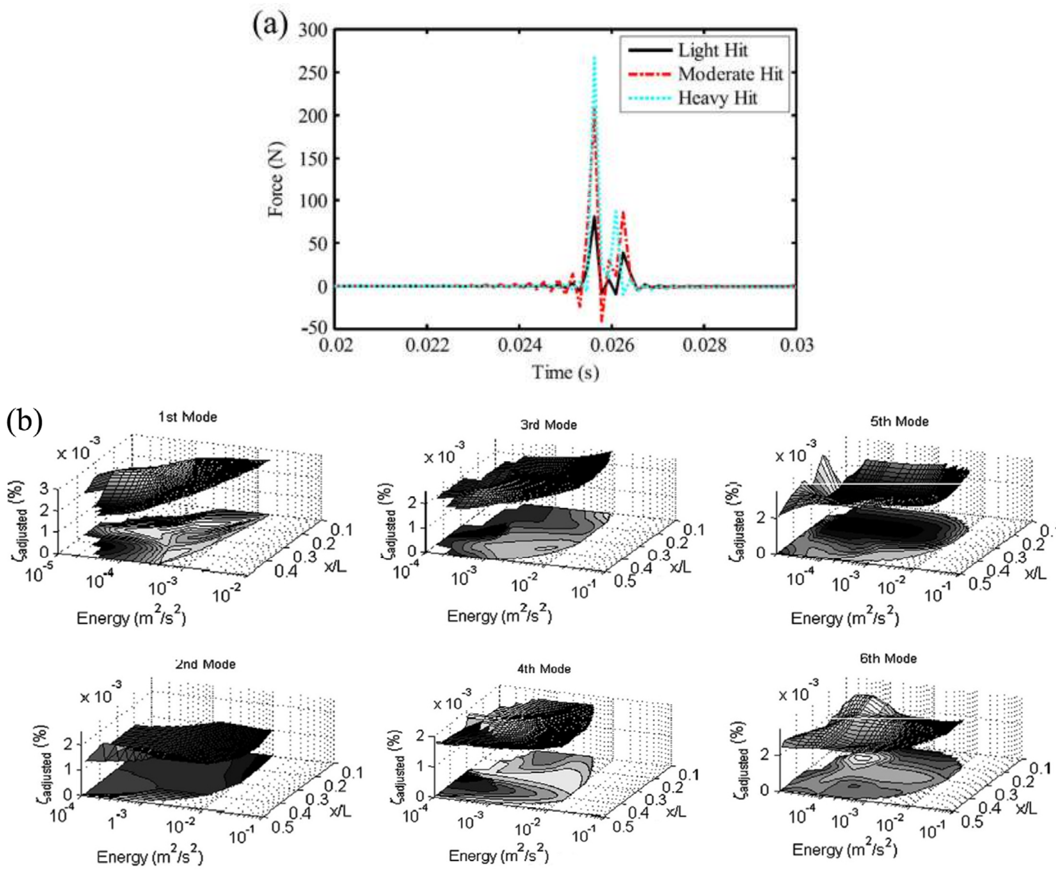

Fig. 29 Testing results by the time instant studies, (a) different impulsive excitations, and (b) equivalent modal damping ratios extracted by the NSI methodology. Reproduced with permission from Ref. [83], © Elsevier Ltd. 2013.

roughness quality and bolt preload are investigated, as in Fig. 30(b). The variability reduces at higher preload, and the lower modes depend mainly on one of the equivalent joint parameters that are likely to be more correlated. Variability in the damping ratios is much more remarkable compared with natural frequencies, and there is higher variability in the damping ratios for lower preload though no clear relationship between variability in the damping ratios and preload has been obtained.

The sweeping sine test was carried out in Ref. [86], considering the white-box modeling of a Bouc-Wen model for capturing the dynamical behavior of the jointed structure. The measured and the predicted hysteresis loops for different amplitudes are shown in Fig. 31(b), which verify the stochastic Bouc-Wen model can produce an adequate prediction for the experimental tests. the confidence bands tend to decrease near the plastic regime, i.e., as the nonlinear restoring force goes asymptotically to a horizontal line opposite to what occurs on the sloping lines of the elastic regime, which is a notable feature for the stochastic model's loops. The test bench was also investigated under the excitation of the white noise, swept-sine, and stepped-sine, to improve the Bouc–Wen model for predicting the dynamics of bolted structures, even taking into account data variability [88].

4.2.2 Brake–Reuß beam systems and the development

In the last few years, the system called the Brake–Reuß beam was improved and settled down at the Sandia National Laboratory [89], which has been developed in several different configurations [90], sharing the same topology as shown in Figs. 32 and 33. The dynamics identification requires two more Brake–Reuß beams with the same geometry, but different characteristics: First, a monolithic beam with neither frictional interfaces nor bolts, which is used as a benchmark dynamical system (Fig. 32(a)); Second, a monolithic beam with the same dimensions

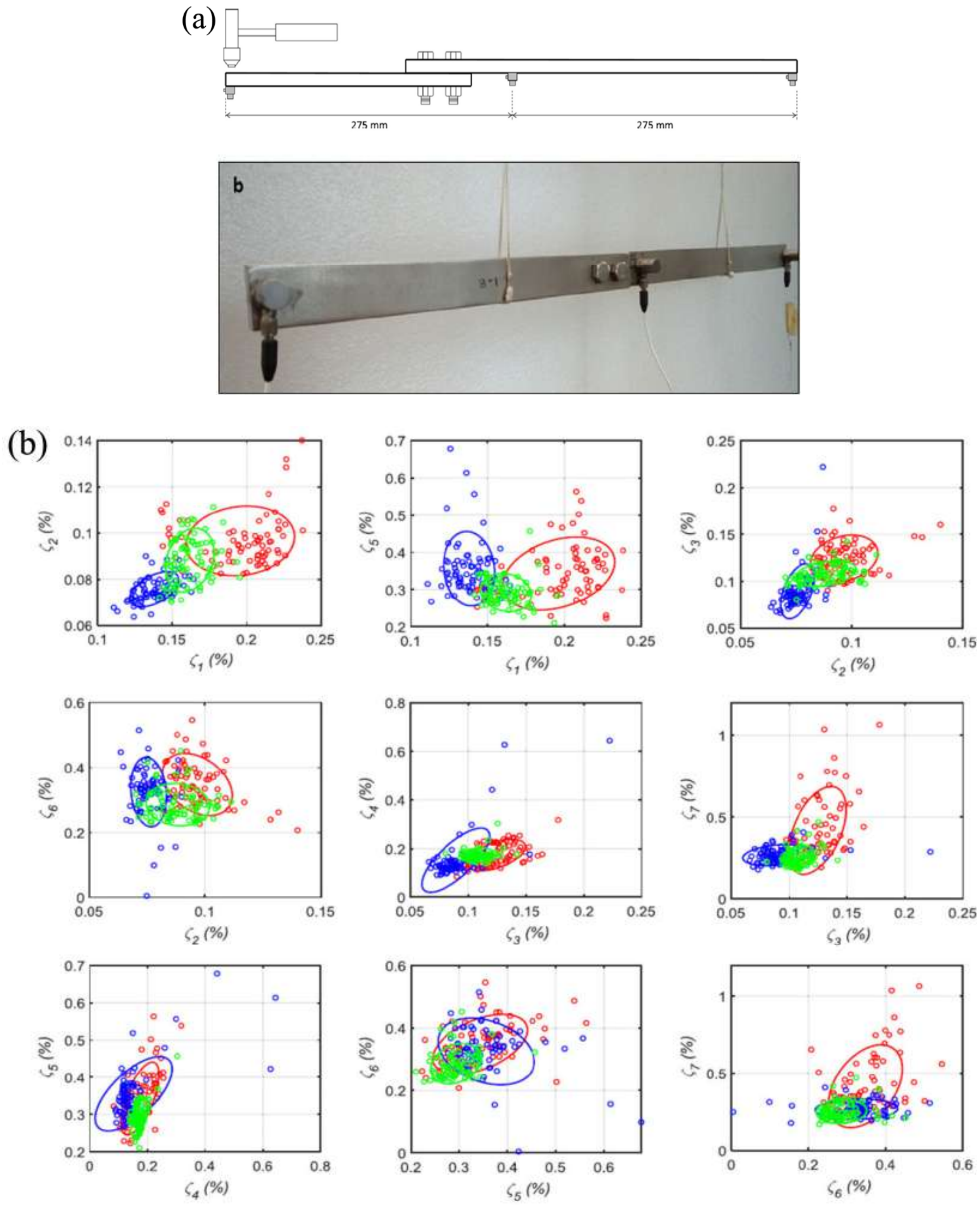

Fig. 30 Hammer testing of the double-bolted join beams with free boundaries, (a) test set-up, (b) Variability in the damping ratios: 7 N·m (red), 15 N·m (blue), and 23 N·m (green) based on modal parameters identification. Reproduced with permission from Ref. [85], © Elsevier Ltd. 2019.

(Fig. 32(b)), with the exception that has three through-holes that are used to allocate the bolts to quantify their effects of the system's dynamics; Third, two beams with a lap joint whose sides are held together by three bolts (Figs. 32(c) and 33).

One common test methodology for the Brake–Reuß beam is shown in Fig. 34 [92]. The hammer tests before and after the fixed sine test are used to obtain and compare the amplitude-dependent natural frequency and damping ratio, while the step sine test is used to search amplitude-dependent natural frequencies. Once the natural frequencies are identified, the structure is usually excited with a fixed sine wave at a prescribed frequency and force level. A high-speed camera is employed to capture the motion at the interface, and the Digital Image Correlation

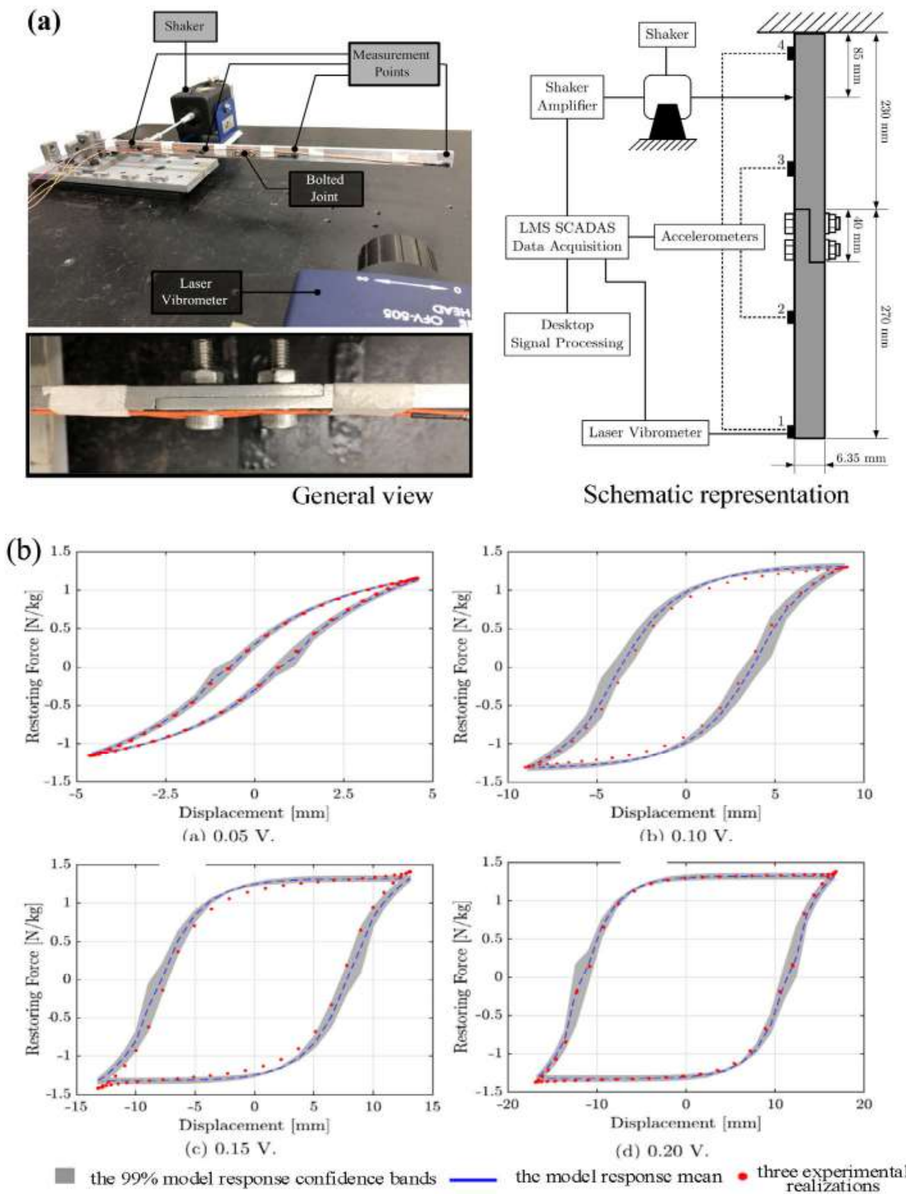

Fig. 31 FRFs tests of the double-bolted joint benchmark with cantilever beam: (a) test set-up and (b) the identified hysteresis loop through the Bouc–Wen versus experimental data for several excitation amplitudes. Reproduced with permission from Ref. [86], © Elsevier Ltd. 2020.

(DIC) method was used to get the displacements at the lap joint interface.

Based on the Brake–Reuß beam system, the effects of the applied boundary conditions, excitation, and measurement techniques [91] have been investigated, and different control strategies [93] on the measurements of the system's stiffness and damping properties have been developed. The Iwan model for the lap joint was widely used for the simulation of the bolted assemblies as shown in Fig. 35, the parameters of which can be acquired based on the experimental results. Lacayo and Allen [94] proposed a highly-efficient quasi-static algorithm to update the Iwan joint parameters, and the amplitude-dependent frequency and damping by loading the finite element model in the shape of one of its modes.

Electronic pressure films were used to measure the contact pressure within an interface instantaneously [96, 97], which can record all contact and not just the final contact. Dreher et al. [98] studied the pressure distribution and pressure change of the bolted Brake-Reuß Beam during dynamic loads, using an

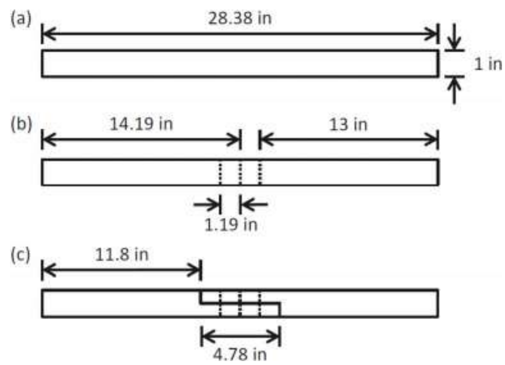

Fig. 32 Geometry of Brake–Reuß beam Reproduced with permission from Ref. [89], © The Society for Experimental Mechanics, Inc. 2014.

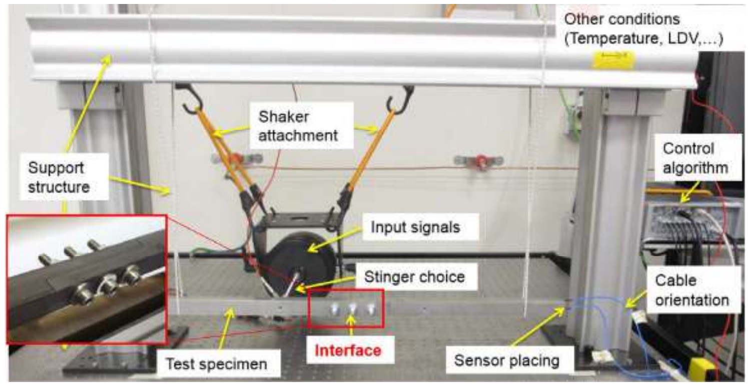

Fig. 33 Basic test setup of Brake–Reuß beam Reproduced with permission from Ref. [91], © The Society for Experimental Mechanics, Inc. 2016.

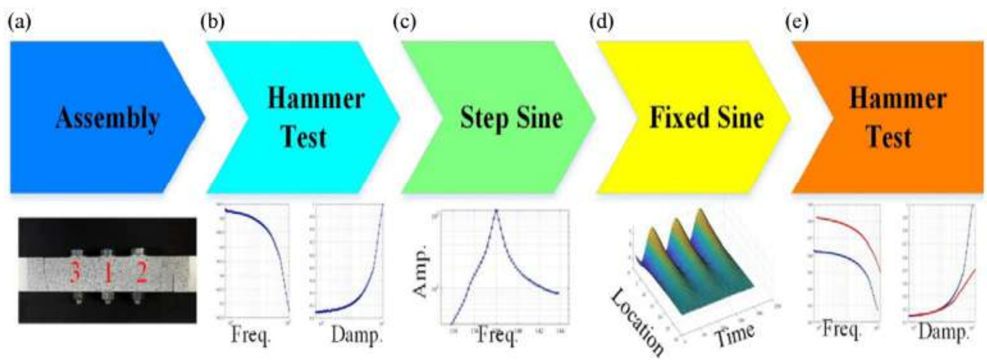

Fig. 34 Procedures for the experimental study. Reproduced with permission from Ref. [92], © Elsevier Ltd. 2019.

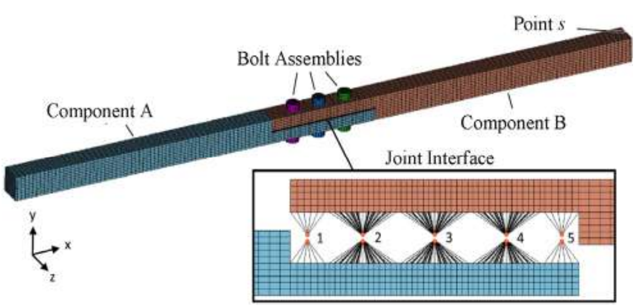

Fig. 35 Nonlinear Iwan joints model. Reproduced with permission from Ref. [94], © Elsevier Ltd. 2018.

electronic pressure sensor placed in the interface plane, as shown in Fig. 36(a). Results show that the contact area of the lap joint is lower for high bolt torques than for low bolt torques, as in Fig. 36(b) caused by the receding contact [99] (a Poisson's effect), and the contact pressure change significantly in both the edges of the interface and between the bolts used to hold the assembly together. As the response amplitude increases, the portion of the interface that is slipping increases, resulting in a softening behavior (i.e., the natural frequency decreasing). While the damping ratio significantly increases due to the increase in frictional energy dissipation.

In recent years the Brake–Reuß beam was also developed using the modified interfaces [90] (Fig. 37), to analyze the contribution of the part-to-part variability of the interface, which may introduce uncertainties in responses of bolted structures [92]. Meanwhile, the influence of the boundary conditions and the far-field structure itself (i.e., the distribution of the stiffness and mass) have been experimentally investigated [95], by employing the beams configurations shown in Fig. 38. A surrogate structure, which is easy to model and machine, containing the same joint as the system of interest can be used to deduce the properties of the joint. These properties can then be substituted directly into the system of interest as a spatially discrete joint model.

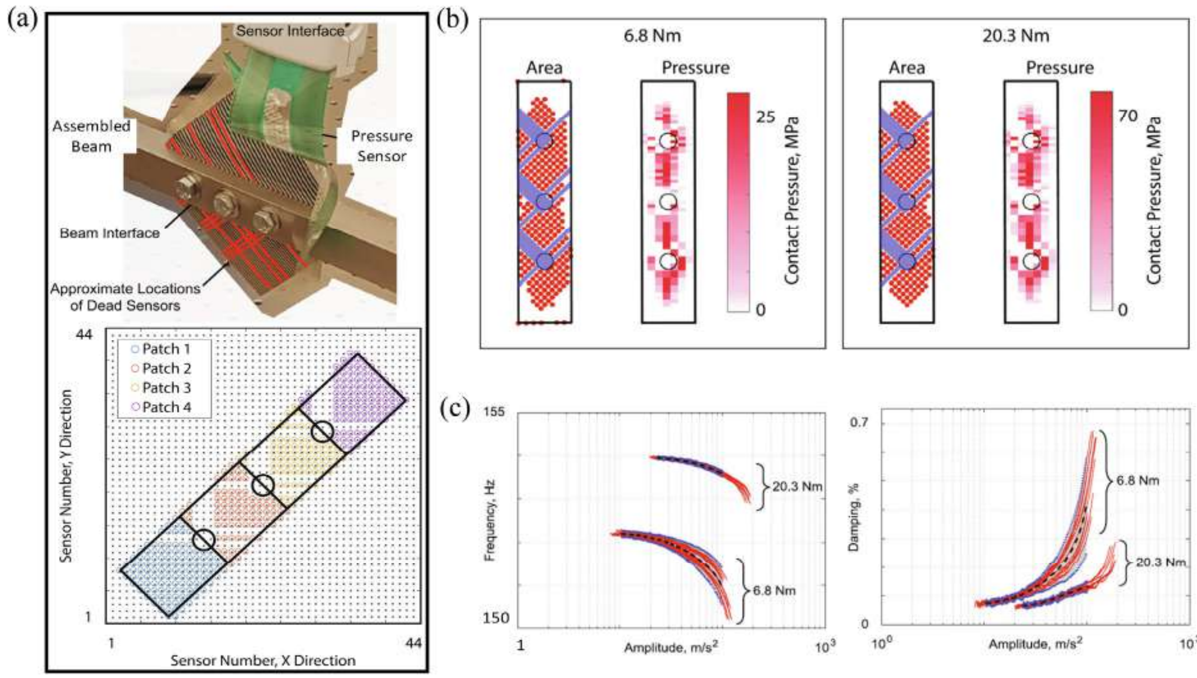

Fig. 36 Effects of contact pressure: (a) illustration of functioning sensor cells, (b) contact areas (left) and pressure distributions (right), and (c) the natural frequency (left) and damping ratio (right). Reproduced with permission from Ref. [98], © Elsevier Ltd. 2021.

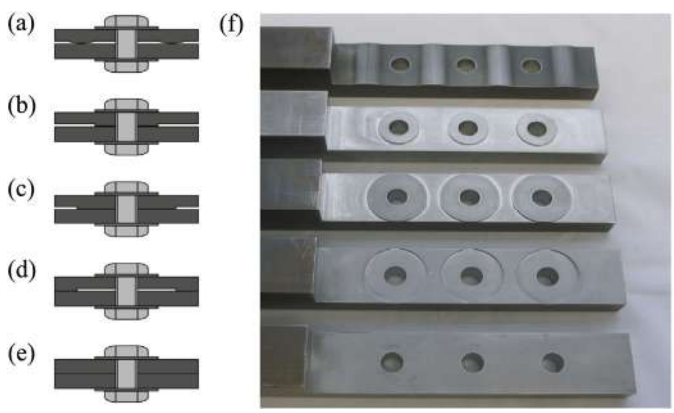

Fig. 37 Interface Modifications in Ref. [90]. Reproduced with permission from Ref. [90], © Elsevier Ltd. 2019.

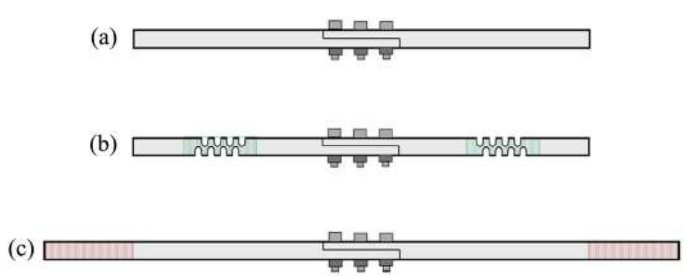

Fig. 38 Different configurations of the Brake-Reuß Beam, (a) nominal, (b) stiffness modified, and (c) mass/length modified. Reproduced with permission from Ref. [99], © The Society for Experimental Mechanics, Inc. 2017.

It seems that the developed Brake–Reuß Beam system can be used as a perfect round-robin in the study of bolted joints, and lots of tests have been done. But the vibration modes of the Brake–Reuß Beam are still limited, and the dynamical parameters identified can only be used for molding the specific structure with similar vibration characteristics. For the more complex realistic engineering structure, such as the shell with flange joints, more suitable testing systems are still needed to be developed.

A new lap-joint configuration, the so-called Orion beam, has been developed by Teloli [100], to highlight the damping that results from friction effects without compromising structural integrity. The Orion beam consists of two assembly beams and is connected by three M4 bolts spaced, as shown in Fig. 39, which are similar to the Brake–Reuß Beam. But the Orion beam suggests an assembly configuration that associates bolts dedicated to "static" functions and those performing "damping" functions, which ensures larger structural damping without degrading the structural stiffness. The clamped-free boundary condition is applied, and step-sine tests have been carried out by a shaker attached close to the clamped end. The central bolt is in tightened condition by setting large torque, to maintain the nominal frequency, whereas the external bolts experience several tightening torques to assess how the friction effects affect the system's damping.

4.2.3 *Beam with several bolts or special geometric features*

The contact beams jointed by the strong magnet were tested by the AERMEC Laboratory of Politecnico di Torino [101], as shown in Fig. 40, to investigate the nonlinear mechanical behavior caused by the frictional interfaces in complex traveling-wave vibration modes. The clamping force is provided by the magnet instead of a screw bolt, thus the effect of the interface between two beams can be studied separately. Furthermore, a novel, simplified, single-bolt system joining a two-beam structure was designed and tested [102], as shown in Fig. 41, to perform a comprehensive experimental study on the peculiar properties of a bolted joint, revealing that the frictional damping induced at the interfaces between two flanges is the main source of the structural damping.

Other beam systems showing different bolts'

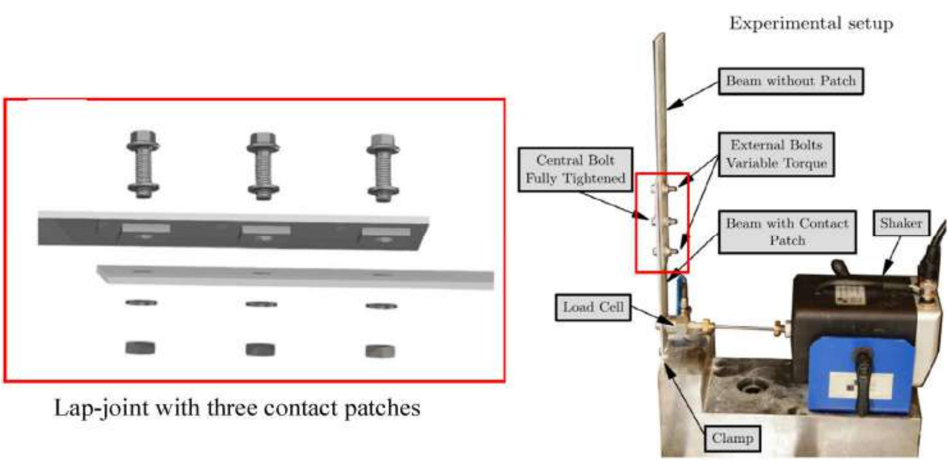

Fig. 39 Experimental setup of the Orion beam system. Reproduced with permission from Ref. [100], © Published by Elsevier Ltd. 2021.

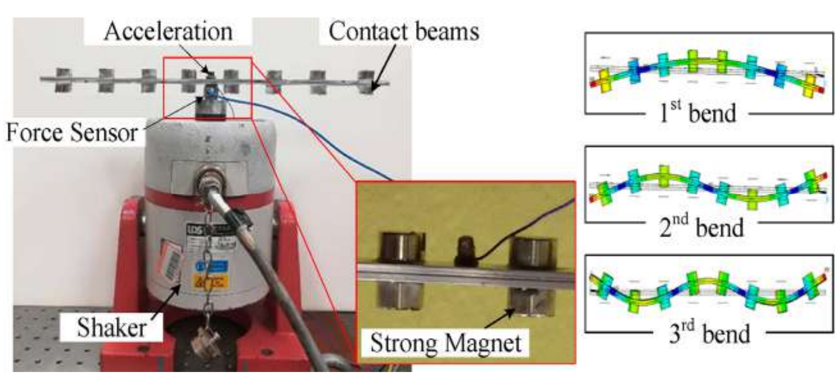

Fig. 40 Contact beams jointed by strong magnet.

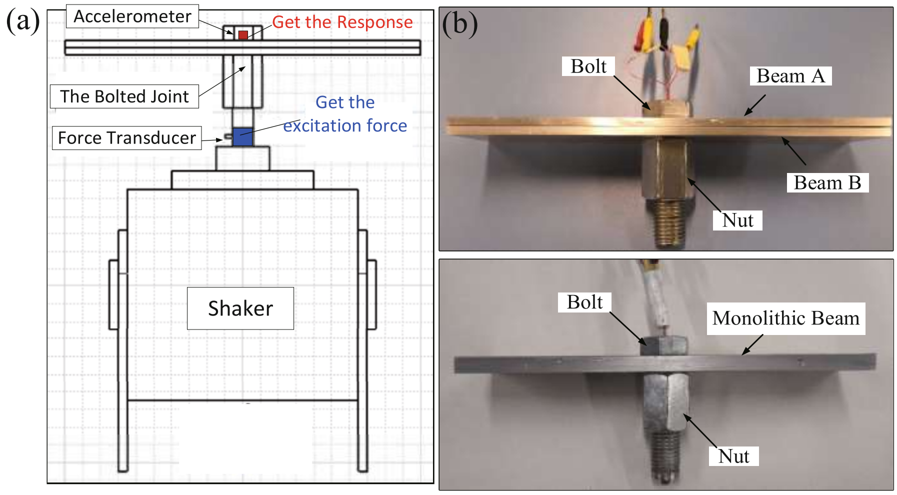

Fig. 41 Dynamical tests of the bolted joint.

layouts can also be found, and typical examples are the beam assembly with bolts regularly spaced along the beams' midline [103] (Fig. 42) and the "S4 Beam" (Fig. 43) [46]. The last one consists of two identical C-shaped beams brought into contact through joints at the ends, on which a hammer test was used to extract the nonlinear behavior of typical mode shapes. The S4 beam characterizes a new benchmark, which provides a rich data set that can be used to validate numerical predictions of damping [104]. Results showed that none of the modes showed large stiffness nonlinearities, but the damping of some modes changed by a factor of five or more [46].

4.3 Identification of dynamic properties of the assembled structure

4.3.1 Structures connected by flange joints

Aero-engine casings with bolted flange joints have been tested at the Imperial College of London and the University of Bristol [105], to identify the nonlinear parameters for a dynamic model. Two test structures of increasing complexity have been used: a simply bolted flange test casing and a sector of a Rolls-Royce aero-engine casing, both shown in Fig. 44.

Modal testing was performed with an LMS Test. Lab system, using an amplitude control measurement method on the accelerometer. While a technique based on two Laser Doppler Vibrometers (LDV) has been developed to measure the FRFs at a grid of measurement points, which can provide data on the relative out-of-plane displacement of the bolted flange joint during a vibration cycle [106]. The influence of different excitation levels and bolt torque on the global and local dynamic responses were fully tested to understand the nonlinear behavior of the frictional flange, based on which the bolted flange joints can be modeled [107]. The full engine casing assembly was also tested using the modal analysis with a Scanning LDV system, and the non-linear behavior of the aero-engine casing was identified and characterized [108].

A slip table (Fig. 45) was designed by Beaudoin and Behdinan for the experiments on the bolted flange of interest, such as the bolted flange in aero-engine casing [109]. The slip table is grounded on a heavy granite table and connected to an electromechanical shaker, and an extra mass was added at the extremity of the test structure to amplify the displacement. The excitation is prescribed at the shaker head in the form of harmonic displacement, and a control module records and adjusts the output to obtain the desired acceleration level. The non-linear frequency responses of aero-engine casings with bolted flanges were also experimentally studied by Boeswald et al. [110] in Germany, showing the influence of joint non-linearity.

Dynamic performances of pipe systems assembled by flange joints caught the attention of researchers. The behavior of the bolted flanges under bending loads was tested by Bouzid [111], aiming at verifying an analytical model used to treat flanges subjected to

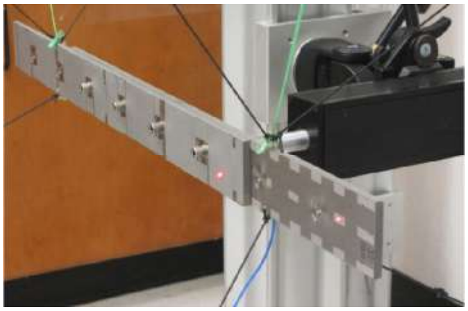

Fig. 42 Beam with several bolts. Reproduced with permission from Ref. [103], © ASME 2015.

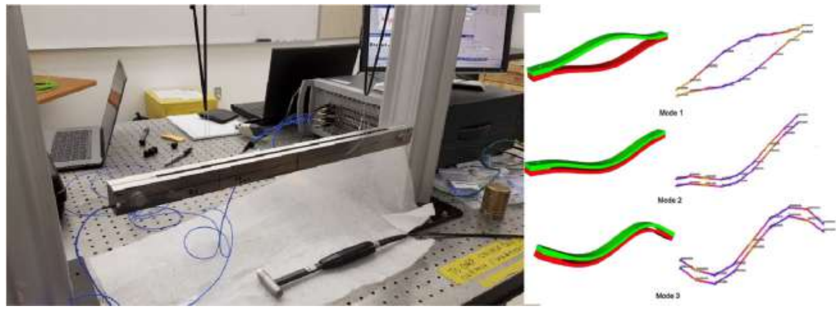

Fig. 43 S4 Beam and its typical mode shapes. Reproduced with permission from Ref. [46], © The Society for Experimental Mechanics, Inc. 2019.

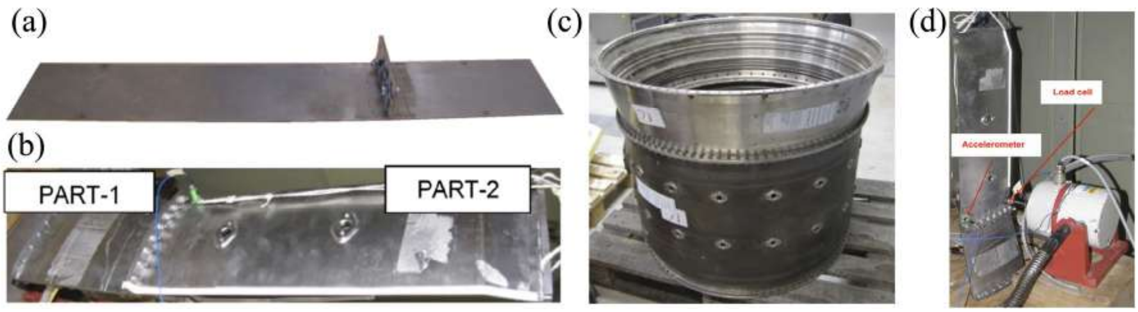

Fig. 44 Test rigs: (a) simplified bolted flange casing, (b) a sector of bolted flange casing, (c) full aero-engine casing, and (d) measurement system. Reproduced with permission from Ref. [105], © Springer Science Business Media Dordrecht 2015.

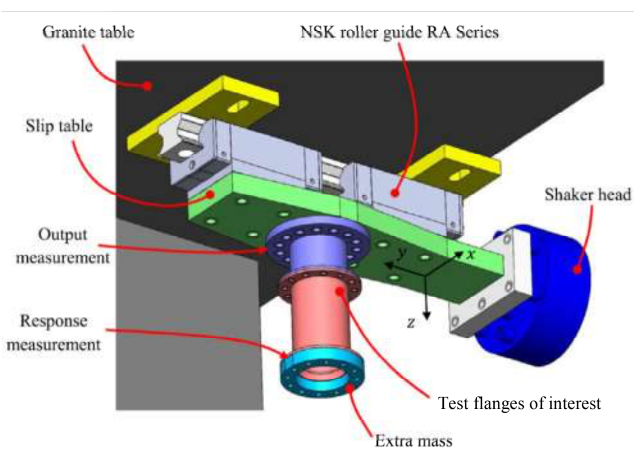

Fig. 45 A slip table for forced vibration tests. Reproduced with permission from Ref. [109], © Elsevier Ltd. 2018.

bending loads such as those produced by external moments and misalignments and capable of integrating leakage around the gasket circumference. Further dynamic experiments were carried out by Ahmadian et al. [112] and Luan et al. [113], by exciting the assembled structure at different points under free-free boundary conditions using a hammer (Fig. 46). Ahmadian et al. developed a linear dynamic model for bolted joints using the thin layer interface theory [112], and the parameters of the model, such as the stiffness and the damping, were tuned to achieve a great agreement with the measured mode frequency data.

Test results given by Luan et al. [113] showed that the initial transverse impulse excites both transverse and longitudinal vibrations with the same order of magnitude but with different frequencies, which might be induced by collision behavior between the junction interfaces: the interfaces collide twice during one transverse vibration cycle, which in return, explains the coupling mechanism between rotational and axial vibration.

4.3.2 *Engineering structures with several joints*

The Ampair 600 wind turbine [114] (Fig. 47(a)) is an important benchmark system in the real application for dissipation measurements of the assembled structure, which has been studied widely and shared by several research groups. The wind turbine consists of three blades each connected to the central hub with three bolted joints near the root of the blade, while the hub and generator housing are also

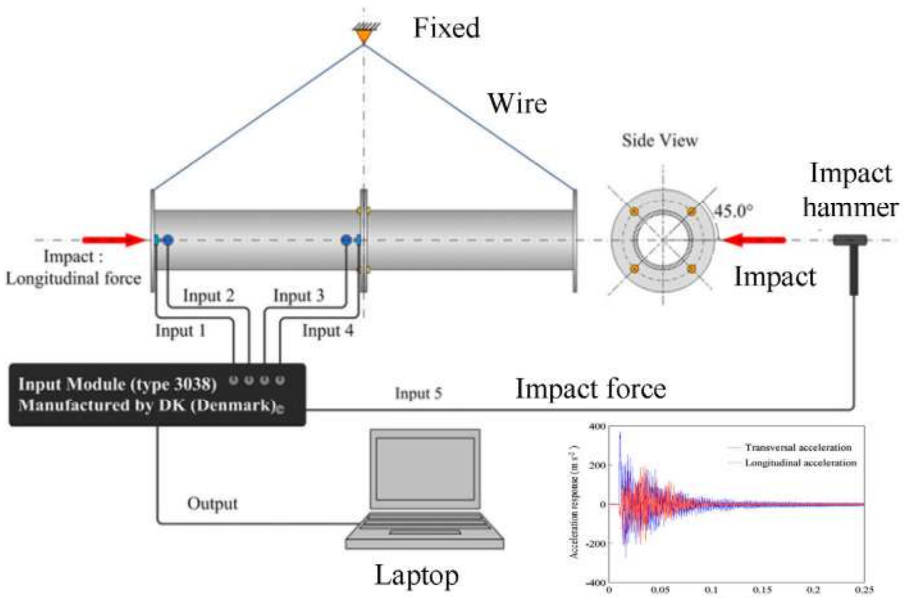

Fig. 46 Dynamic experiment setup of bolted pips. Reproduced with permission from Ref. [113], © Elsevier Ltd. 2011.

Fig. 47 Schematics of (a) the entire wind turbine [114], (b) bolted blade and hub [115], and (c) the tower structure. Reproduced with permission from Ref. [116], © The Society for Experimental Mechanics, Inc. 2014.

connected by a central bolt. Besides the entire wind turbine assembly, the subcomponents, such as a single- or three-blade and hub assembly (Fig. 47(b)) [115], or the tower structure with hubs (Fig. 47(c)) [116] have also been extensively tested and studied. The modal characteristics of the assembly, as well as the subcomponents, were tested using the impact hammer in free-free boundaries. The nonlinearity of the substructure was identified by some new tools such as the Hilbert transform algorithm or the Zeroed Early-Time FFT (ZEFFT) algorithm, while the influence of the load or the boundary conditions in tests was also investigated [117].

Some other complex industrial structures were also tested to characterize their behavior in the presence of bolted joints. Among them can be mentioned the catalytic converter system on motor vehicles [118], the cylindrical shells and four vanes connected by groups of bolts [119], and the mock aerospace hardware [7], as shown in Fig. 48. The nonlinear identification methods were developed to acquire the nonlinear mechanical properties caused by the bolted interfaces. In the test work of the mock aerospace hardware, the electrodynamic shaker was used, which can apply types of excitations, such as the specific shock, or a harmonic transient excitation.

A full-scale F-16 aircraft (Fig. 48(d)) was tested on the occasion of the Siemens LMS Ground Vibration Testing Master Class, and the dominant source of nonlinearity was expected to originate from the mounting interfaces of the two payloads [120]. A special experimental approach to identify the damping contributions has been developed as well, for types of joints used in the space industry [121]. As given in

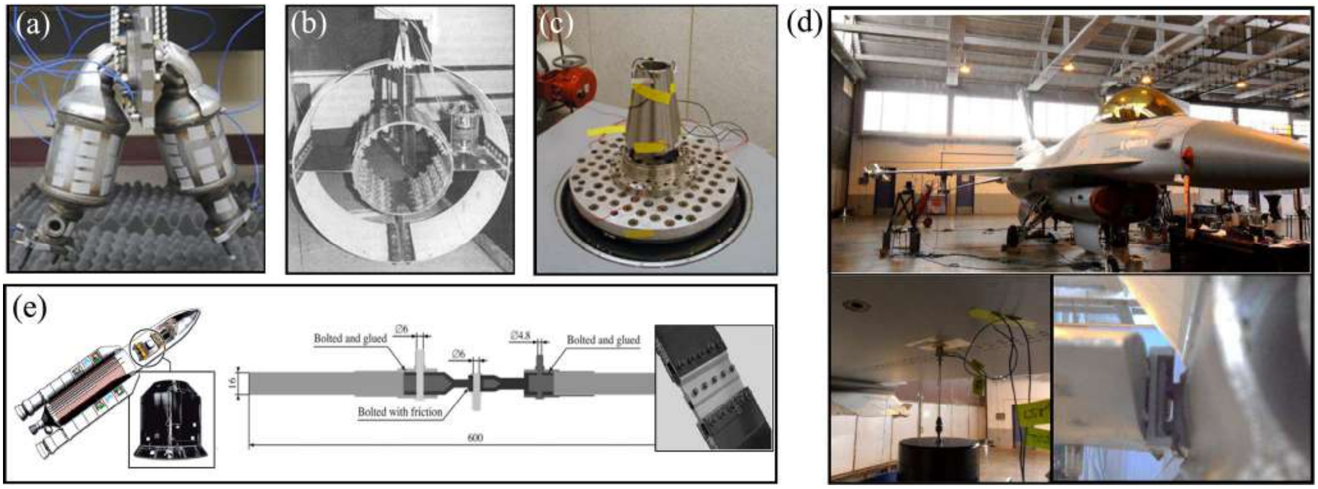

Fig. 48 Specimens of (a) the catalytic converter system, (b) bolted shells and vanes, and (c) the mock aerospace hardware; (d) the connected aircraft structure; (e) ARIANE 5 launcher component. Reproduced with permission from Ref. [122], © Emerald Group Publishing Limited 2010.

Fig. 48(e), the uniformly spread bolted joint is used to attach the SYLDA 5 to the rest of the structure, and experimental devices were used to measure the damping properties of various structural elements. The device was able to expose the specimens of joints to the same stress and strain levels, at the same frequency, as those the final structure will experience during the flight [122].

There are no certain round-robin systems for complex real engineering structures, but the strategy to carry out the tests and identify the dynamical parameters are valuable for reference. In studies, the geometric features, vibration shapes and the type of excitation force must be considered first. Based on this, a suitable strategy can be chosen, and the parameters we are concerned about can be identified.

5 Concluding remarks and recommendations

Bolted joints play a more and more important role in the structure with lighter weight and heavier load. Experimental studies used to be, and will still be one basic method to study the influence of the bolted interfaces, and acquire the frictional parameters for the predictive model. The continuous publications of new studies and the organization of conferences devoted to the bolted confirm the interest in this topic, and in fact, the publications in the term 'bolted joints' in last year are 10 times more than that 30 years ago. It is believed that more work and new contributions will be needed for the foreseeable future, in the industry of mechanical, aerospace, and civil engineering.

This review introduces different experimental approaches for the dynamic behavior of structures in the presence of bolted joints, especially the energy dissipation or damping at frictional interfaces. Features of different benchmarks system and advantages or disadvantages of different strategies have been concluded, and more future research can be recommended as follows.

1) The macro- or micro-slip in the tangential direction at the frictional interfaces is believed to be the main sources of the damping. Plenty of tests have been carried out to acquire the features of sliding and measure the damping caused by the frictional interfaces. But most of them consider only the simple tangential force and constant normal pressure, it may be valuable if the energy dissipation mechanisms under more complex loads, such as the bending, can be taken into consideration.

2) Variations of the contact interfaces during a long cycle operation need to be concerned, the damage to the interfaces will certainly change the stiffness or the damping induced. It is still a challenge to characterize the evolution of the interface and study it in both experimental and simulation methods.

3) The pressure distribution caused by a bolted

joint, the levels of excitation, and the friction coefficient are the most important factors that determine the slip forces and therefore the amount of friction damping at the contact interface. The force sensors are used by almost every researcher to measure the excitation force applied to the structures, while the strain gauges, indicating washer and pressure films are used to measure the preload or pressure of the bolted joints, which are suitable for different structures and purposes. In some cases, ultrasonic techniques are also used to monitor the changes in the bolted condition.

4) Types of test rigs have been developed in the last decades, including the equipment with one or several simplified bolts used to acquire the damping parameters of joints, or the round-robin system to measure the hysteresis characteristics in the isolated interface. In the early stages, most of the test rigs are used in specific laboratories, but it will be the current trend to build the round-robin/benchmark for measurements of hysteresis in standard joints.

5) For the bolted structures with more complex mechanical characteristics, abundant strategies to identify the dynamic properties have been developed, and common methods such as the oscillation decay curve method, sweep-sine or stopped-sine analysis, and FRFs were widely used both in academia and industry. However, the data obtained from different tests are difficult to be used, considering the uncertainties induced between lab-to-lab or experimentalist-to-experimentalist. A widely accepted round-robin on the measurement and prediction of dissipation or hysteresis measurements is still expected.

6) Most of the techniques were developed for the linear system, while the amount of energy dissipated has a nonlinear relationship with the amplitude of the system for a nonlinear bolted structure with frictional interfaces. Thus, a targeted methodology for the cohesive measure of energy dissipation in the nonlinear bolted structures is necessary, which may be one of the most important directions in future research.

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Declaration of competing interest

The authors have no competing interests to declare that are relevant to the content of this article.

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