International Conference on Advances in Material Science and Mechanical Engineering (ICAMSME 2020)
N.B.K.R.I.S.T., Vidyanagar, Andhra Pradesh, India – 524 413

Experimental Characterization of Stiffness and Damping of Assemblies with Bolted Lap Joints

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Abstract

Mechanical joints influence the dynamics of assembled structures by contributing flexibility and damping to the assembled system. There is hence a great deal of interest in understanding how to design interfaces to obtain preferential damping in the overall assembled system while not compromising on the stiffness of assemblies. Despite the many strides made in modelling interfaces, due to the difficulties in modelling damping, the experimental route to characterize the influence of joints remains the preferred method. This paper offers one such experimental characterization of bolted lap joints common to most mechanical assemblies. We investigate the stiffness and damping characteristics of an assembly with a simple lap joint under changing preloads and excitations. By conceptualising damping as energy dissipation in the system, the area enclosed by hysteresis loops is used to estimate damping and stiffness. We find that damping increases with increasing loading and decreases with increasing preloads. We also observe that the stiffness is weakly influenced by loading and preloads. Results are instructive to design interfaces with bolted connections.

Keywords: Bolted Joints, Dynamic behaviour, Contact Damping, Lap joints, Hysteresis loop.

1. Introduction

Most mechanical systems are assemblies of individual subcomponents that are manufactured separately and then assembled by welding, gluing, riveting, bolting etc. For any such an assembled structure, the location, number, and nature of its joints influence the stiffness and damping in the system. In metal structures, 90% of the damping in the structure is attributed to joints, as stated by Oldfield *et al*. (2005). Consequently, more attention must be paid to the mechanism causing this joint damping. Based on the direction of the force transmitted to the interface, the joints are divided into normally loaded and tangentially loaded joints. In normally loaded joints, the amount of damping is very small compared to tangentially loaded joints provided that there is no uplift or slapping in the tangentially loaded joint.

For a bolted lap joint, which forms the focus of this paper, adequate tightening torque constraints the joint from having uplift and slapping, thus reducing the damping due to normal loads. Ahmadian *et al.* (2007) stated that in the case of a tangential load applied to such a joint, the phenomena associated with joint damping is microslip and macroslip. Microslip occurs when the interface partially slips locally while a large part of the contact is still sticking. When the tangential excitation force increases or when the normal load decreases, the components start to move at the interface and microslip occurs. Identifying the joint characteristics of bolted lap joints subjected to tangential and/or normal loading remains non-trivial, and has hence received much research attention.

Though there are several ways to estimate characteristics of joints, which include methods using harmonic balance schemes as suggested by Menq *et al*. (1991), or methods that use the frequency-based inverse receptance coupling scheme as was done by Sanati *et al*. (2017), or yet other time-domain logarithmic decrement based schemes as was suggested in Dion *et al*. (2018), the preferred methods remain those which use load-displacement hysteresis loops to estimate joint stiffness and damping (Rivin (1999)). Hysteresis loops have been successfully used to illustrate microslip and macroslip in a bolted joint (Ouyang *et al*. (2006)), and also to investigate the influence of increasing loading scenarios on the interface damping of a bolted joint (Sanati *et al*. (2018)). The main idea in the reported studies using hysteresis loops have been to harmonically excite the system at the natural frequencies of interest and evaluate the amount damping by equating the area enclosed by the hysteresis loop to the energy dissipated due to interfacial damping.

In the study of bolted joints, experiments play a very important role. The experimental setup needs to be designed to isolate the effects of joints from that of the structure. Kuether *et al*. (2018) studied the variability of the dynamic response of a Brake-Reuss beam for experiments conducted under different conditions. The beam was subjected to different levels of amplitudes of harmonic excitations, and, in separate experiments, to changing preloads. Receptances were reported to change with excitations and preloads – indicative of nonlinearities. Smith *et al*. (2018) showed that the boundary

conditions, loading and measurement techniques affect the joint dynamics of a free hanging Brake-Reuss beam. The variability of joint dynamics with changes in bungee length and position, sensor size and location, cable orientation of sensors, length of stinger, excitation amplitudes and types of signals result in slight variations in the joint dynamics while changes in tightening torque values, order of tightening the bolts, and retorquing showed considerable changes in the joint dynamic response.

Though there is a lot of reported work on estimating bolted joint characteristics using hysteresis loops, the combined influence of changing tightening torques and changing loading scenarios on the response (stiffness and damping) characteristics of the assembled system does not appear to have been addressed yet. This is indeed the main contribution of this paper – to report on how response of the assembled system changes with preloads and loading conditions. To do so, we perform systematic experimental characterization of a bolted lap joint. While designing our experimental setup we were mindful of the challenges in experimentation reported by Kuether *et al*. (2018) and Smith *et al*. (2018). The aim is to understand how to design interfaces to obtain preferential damping in the overall assembled system while not compromising on the stiffness of assemblies. The remainder of the paper is organized as follows. The experimental setup and procedures, and methods to estimate joint characteristics from experiments are detailed in Section 2. This is followed in Section 3 with discussions on the variation of joint stiffness and damping with the changes in joint preload and excitation levels. The paper in then concluded in Section 4.

2. Experimental Setup and Methodology

2.1. Experimental Setup

The experimental setup is as shown in Fig. 1. It comprises of a beam and base made of En-8 material. The base was intentionally made large to have a very high stiffness, and it was secured to the table using four M12 bolts. The beam is mounted on the base and bolted together by two M8 bolts. The beam has a diameter of 50 mm, and an overhang of 360 mm. The setup was designed to have the first bending mode in the range of 150 to 250 Hz. The M8 bolts used are of a grade of 12.9 with a recommended tightening torque of 42 Nm. Since this study is interested in understanding the influence of tightening torque of the interface characteristics, experiments were conducted with two levels of tightening torque. One level was below the recommendation, i.e., 30 Nm, one level is at the recommended value of 40 Nm. Considering the inputs from Kuether *et al*. (2018) and Smith *et al*. (2018), the location of the sensors and the excitation, cable orientation, type of excitation signal and amplitudes, and length of stinger were fixed to reduce variability.

LabVIEW software was to generate the input signal to NI-9263 output module, which sends an analog signal to the power amplifier (B&K Power Amplifier Type 2732) which amplifies the voltage of the signal. This was connected to a modal exciter (B&K Modal Exciter Type 4824). The excitation was transmitted to the free end of the beam with a 100 mm stinger with a load cell (Dytran - model 1051V4) at the end to measure the applied force. The load cell was fastened to the free end using 10-32 UNF threaded fastener. The response was measured in the direction of applied force by an accelerometer (Meggit- model 44A16-1032) which was magnetically mounted close to the interface. The signals from the accelerometer and the load cell were given to a NI- 9234 input module which send to signals to a computer system. The stinger length was adjusted to 100 mm to have the minimum variation in the measured receptance and to prevent bending of the stinger during excitation.

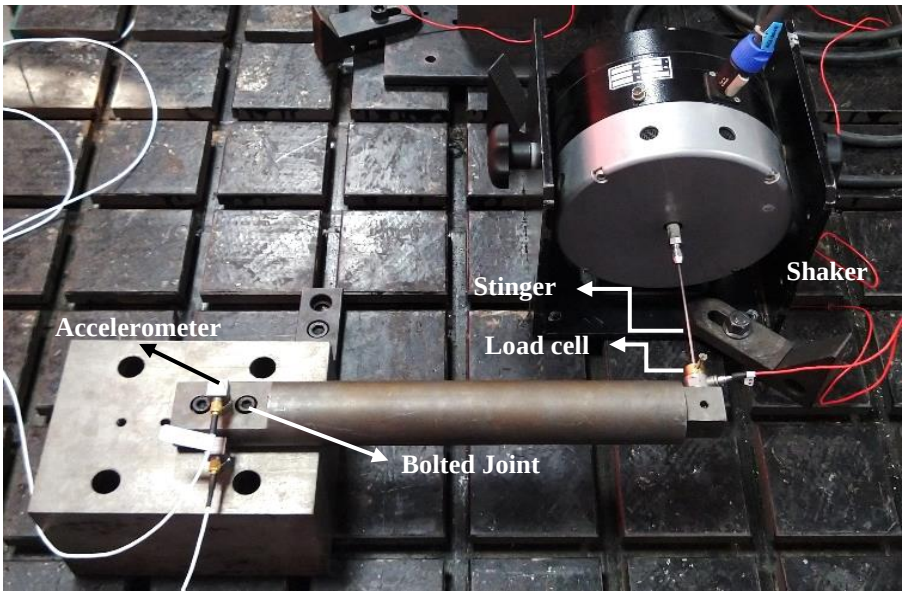
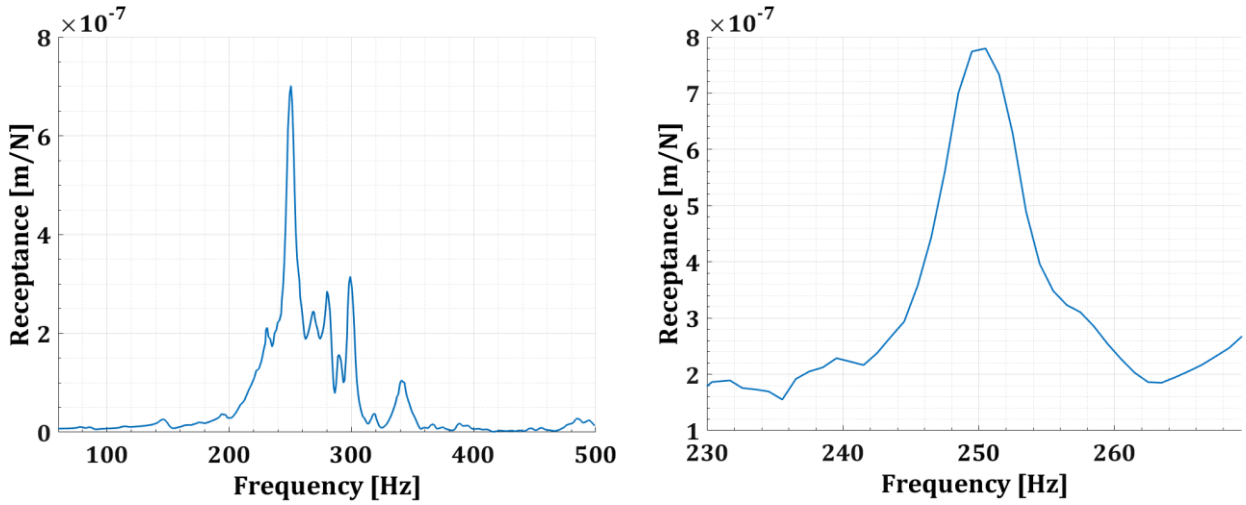


Fig. 1 Experimental setup used to identify the bolted joint damping characteristics

2.2. Experimental Procedure

The structure was first excited using an impact hammer to analyse where the receptance peaks were and it was observed to have two peaks in the range 60 to 500 Hz. A slow linear sine sweep excitation signal from 60-500 Hz, with excitation time of 1 s per frequency, was then given as the input to shaker. In this region of the receptance (magnitude) in Fig. 2 (a), two peaks were observed. But the actual peak corresponding to the first bending mode of the beam was identified from its mode shapes calculated at each of those peaks. The structure was then excited at a frequency range around the first natural mode with an excitation time of 5 s per frequency, providing more time to stabilize, thus also helping in identifying the exact natural frequency corresponding to the first mode. Results for this are shown in Fig. 2 (b). After identifying the exact natural frequency corresponding to the first bending mode, the setup was excited at that frequency using a sine excitation for a longer duration till the output response measured at the jointed end was stable. The output acceleration picked up by the accelerometer was integrated twice in the frequency domain and then transformed into the time domain to obtain the displacement x at the joint. Since this study seeks to understand the influence of changing preloads (tightening torques) and excitation levels on the response of the assembled system, experiments were repeated for two levels of preloads, i.e., for 30 Nm and for 40 Nm, and with excitation voltages of 1.5 V and 3 V for each level of preload.



(*a*) (*b*)

Fig. 2 Receptance plot for 40 Nm tightening torque and 1.5 V excitation voltage for: (*a*) a slow sine excitation from 60-500 Hz, and (*b*) frequency band focused around the first bending mode

2.3. Estimation of joint characteristics from experiments

Since the response of the system in the frequency range of interest suggests that there is only one dominant mode (natural frequency), excitation of the system at that mode and the resulting response can be used to construct a hysteresis curve – which is shown schematically in Fig. 3.

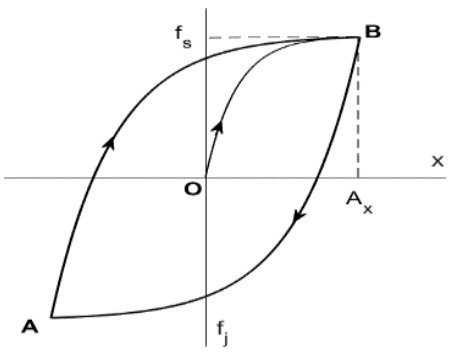


Fig. 3 Representation of a typical hysteresis curve

The area under this hysteresis curve E, is the energy dissipated, and, stiffness can be taken to be the ratio of maximum force (f_s) to the amplitude of displacement (A_x), i.e., $K = f_s/A_x$. To estimate damping, consider that the area enclosed by these hysteresis loops E, can be expressed as:

$$E = \oint F\,dx = \oint C\dot{x}\,dx = \oint C\dot{x}^2\,dt \tag{1}$$

wherein the damping force is assumed to be of a viscous nature, i.e., $F = C\dot{x}$, wherein C is the damping, and $\dot{x}$ is the velocity. Since the forcing was harmonic, the corresponding response too was harmonic. This response at a frequency ω can be assume to be of the form of: $x = A_x \sin(\omega t - \phi)$, which when substituted in Eq. (1), results in:

$$E = \int_0^{2\pi/\omega} C\omega^2 A_x^2 \cos^2(\omega t - \phi)\, dt = \pi\omega A_x^2 C. \tag{2}$$

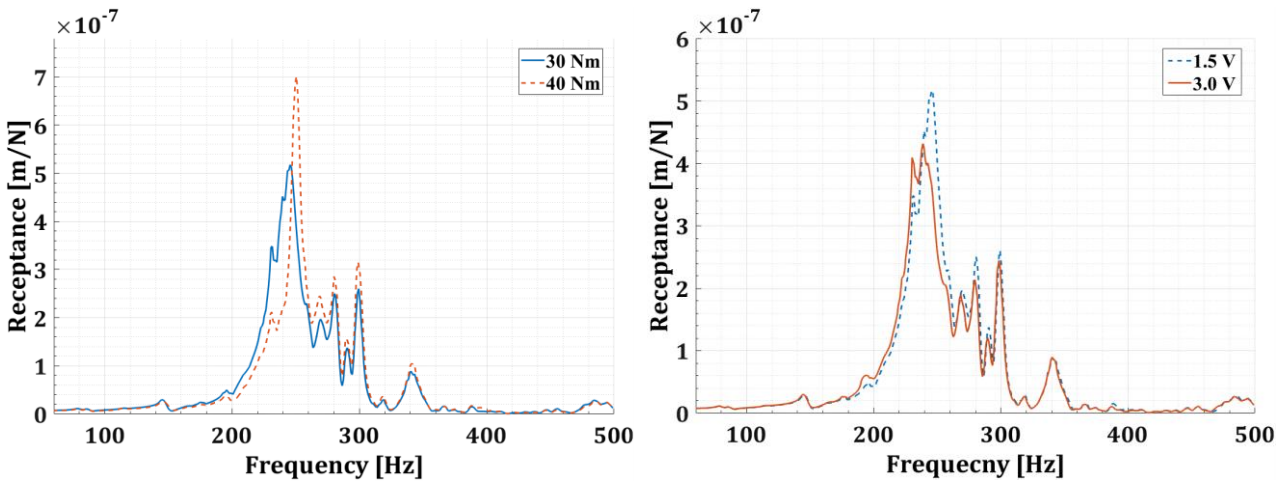
From Eq. (2), damping in the system can be expressed as:

$$C = \frac{E}{\pi\omega A_x^2}. \tag{3}$$

The above procedure and equations are used to estimate the stiffness and damping of the system under changing preloads and excitations. We would like to highlight here that the procedure we adopt cannot separately describe the stiffness and damping characteristics of the interface and can only give us a sense of the overall change in the stiffness and damping of the system with changing preloads and excitations. This may indeed be a limitation of this study.

3. Results and Discussion

We first discuss how the receptances change with preloads and excitations in Fig. 4, and then discuss the hysteresis curves in Fig. 5. As is evident from Fig. 4 (a) - as the tightening torque was increased from 30 Nm to 40 Nm, the natural frequency of the system shifted to the right. This is congruent with a stiffening behaviour with increase in the joint preload. Moreover, the peak value of receptance increases with increase in tightening torque. This change implies that the product of stiffness and damping ratio decreases. Fig. 4 (b) depicts a decrease in the frequency from 246 to 239 Hz with an increase in the level of excitation voltage from 1.5 V to 3 V, and a decrease in the peak receptance values. This is indicative of a softening-like behaviour.



(*a*) (*b*)

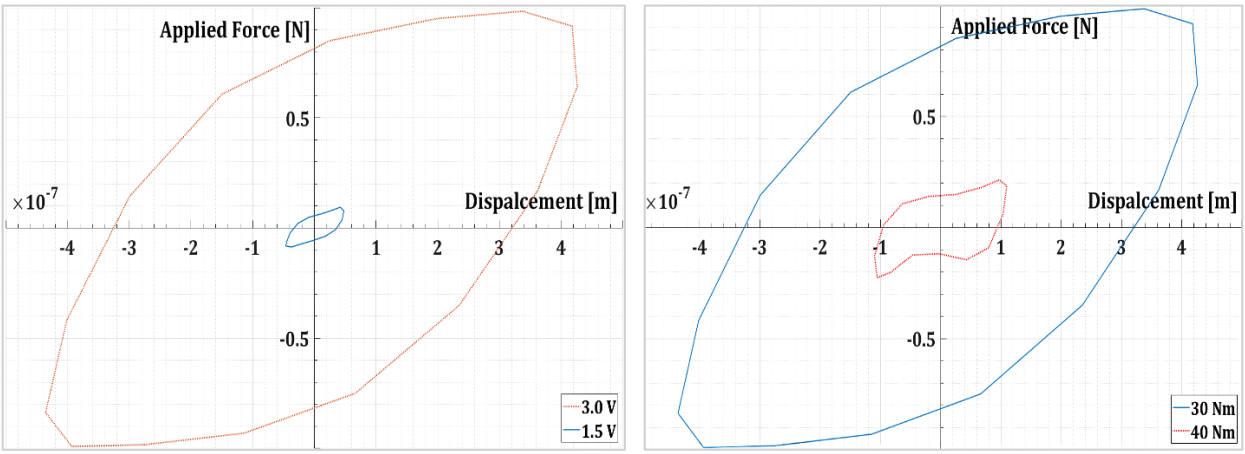
Fig. 4 Change in receptances for: (*a*) changing tightening torques for excitation voltage of 1.5 V, and (*b*) changing excitation voltages for a tightening torque of 30 Nm.

Changing hysteresis behavior with changing preloads and excitations are shown in Fig. 5. As is evident from Fig. 5(a), the area enclosed by the hysteresis loops increases with an increase in the applied excitation voltage. The area enclosed by the hysteresis loop is equal to the energy dissipated by the system and increasing the voltage increases the excitation force and thus increases the energy input to the system. This implies that with more energy provided to the system by increasing the excitation voltage, more energy is dissipated in the system at the joint. This also suggests the presence of a nonlinearity in the system. In Fig. 5 (b), it is observed that with increasing the tightening torque at the bolts, the area enclosed by the hysteresis loop decreases. This decrease in the area implies that with increasing tightening torque, the normal load on the joint is increased to such an extent that it reduces the slipping happening at the interface thus decreasing the dissipation due to friction. Damping that is estimated from these hysteresis curves using Eq. (3) is listed in Table 1, which also list the stiffness evaluated from the results in Fig. 3.

As is evident from Table 1, an increase in the excitation voltage from 1.5 V to 3.0 V does not cause a considerable change in stiffness. The increase in excitation voltage increases the force applied at the joint causing more displacement at the joint. It is clear from Fig. 5 (a) that the peak values of joint force and amplitude of displacement increases with increase in the excitation voltage, but this is proportional enough to keep the variations in the stiffness low. It is observed that increase in the tightening torque from 30 Nm to 40 Nm evidently decreases the stiffness. This might be because the rate

increase of the joint force is lesser compared to that of the joint displacement amplitudes. Hence, any change in the excitation voltage or tightening torque around the recommended value does not appear to change the stiffness by a substantial amount.

It is further observed from Fig. 5 and Table 1 that damping increases with increase in the excitation voltage. Increase in the excitation voltage increases the tangential force applied at the joint thereby increasing the slipping at the joint. When slipping increases, more energy is dissipated due to friction and this is reflected as an increase in damping. Increase in tightening torque decreases damping. The decrease in damping is more for the higher value of excitation voltage than the lower value of excitation voltage. As tightening torque is increased, the normal load applied at the joint increases. As the normal load increases, the friction increases to a higher level thereby reducing the slip and reducing the damping. This is evident with the data of amplitude of displacement in Table 1.



(*a*) (*b*)

Fig. 5 Hysteresis curves for: (*a*) varying excitation levels for same resonant frequency for 30 Nm tightening torque (*b*) varying tightening torque values for the bolted joint for excitation of 3.0 V

Table 1 Experimental results of stiffness and damping and natural frequencies of the system for varying tightening torque and excitation voltages

30 (Nm)	**Voltage**	**1.5 (V)**	**3.0 (V)**
	f_n (Hz)	246	239
	Area (Nm)	6.24E-09	9.75E-07
	f_s (N)	0.094	0.98
	A_x (m)	4.64E-08	4.36E-07
	***K* (N/m)**	2044430.6	2257739.3
	***C* (Ns/m)**	594.5	1074.8
40 (Nm)	**Voltage**	**1.5 (V)**	**3.0 (V)**
	f_n (Hz)	250	246
	Area (Nm)	2.76E-09	4.57E-08
	f_s (N)	0.04	0.21
	A_x (m)	3.71E-08	1.10E-07
	***K* (N/m)**	1169869	1943865.8
	***C* (Ns/m)**	403.9	778.6

4. Conclusions and outlook

This paper was concerned with investigating the stiffness and damping characteristics of an assembly with a bolted lap joint under changing preload conditions of the joint and changing excitations of the system. System experimental characterization was carried out to construct hysteresis curves under changing test conditions. Damping was estimated by energy dissipated in a cycle given by the area enclosed by the hysteresis loops. Stiffness was evaluated from the ratio of the amplitude of tangential force at the joint to the amplitude of interfacial displacement.

In general, damping was found to decrease with an increase in the tightening torque of the bolts, and damping was also found to increase with increasing energy input into the system. Stiffness, on the other hand was observed to be very weakly influenced by an increase in tightening torque and/or increasing energy input into the system. This suggests that tightening bolts to just below the recommended tightening torques may not be an adverse idea – since there can be a potential gain in damping at the interface while not compromising the stiffness of the system.

Experiments and analysis presented herein are for the system, and results cannot be used to discern the behaviour of the interface alone. Further analysis is necessary to understand how the interface stiffness and damping characteristics are influenced by any change in interface preloads and the interface can dissipate energy with an increase in the energy provided to the system. This forms part of the planned future work.

Acknowledgements

The Authors gratefully acknowledge Mr. Subhash N and Mr. Karthik R – engineers in the Machine Tool Dynamics Laboratory at the IIT Kanpur for their help in conducting some of the experiments reported herein. The third author of this work also acknowledges the support of the Government of India's Impacting Research Innovation and Technology initiative through project number IMPRINT 5509 – that made possible the experiments at the IIT Kanpur.

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