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Identification of Dynamic Damping Properties of a Flexible Structural Adhesive

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K E Y W O R D S

Identification

Dynamic damping properties

Young's and shear moduli

Experimental modal analysis

A B S T R A C T

In this paper dynamic damping properties of a nominated flexible structural adhesive have been identified using an extended-direct modal based joint identification method. It has been revealed that damping characteristics of adhesive are correlated to both frequency and mode shape. Young's and shear moduli increase with frequency but damping on the other hand, decrease. The results showed that mode shape has an important role on the dynamic mechanical properties of adhesive. Modes that dominantly create normal stress on the adhesion surface represent higher stiffness and lower loss factor compared to shear modes. The different level of loss factors between bending modes and shear modes are notable, but damping property of bending modes and shear modes on the other hand, is in the same order and decreases with frequency. It has been shown that the effective, viscoelastic, mechanical properties of the adhesive can be identified successfully, using the suggested method of identification.

1. Introduction

Due to the increasing demand for more efficient vehicles with improved performance and also in order to decrease air pollution and for environmental considerations, there is a need to manufacture lighter structures. Adhesives have many advantages over other traditional joining methods. Better stress field over the bonding surface, decreasing noises and improving handling quality for automobiles and trains and also working as a sealant simultaneously, have attracted manufactures and designers to utilize this type of connection [1-4].

Because of joint's roll where it is encountered to high level of strains and also local effects, they can significantly affect dynamical properties of structures [5-6]. Therefore in order to better understand mechanical properties of adhesive, many investigations should be performed on this type of bonding medium.

Nobari et al. [7] has provided a direct modal based joint identification method that uses experimental modal parameters (natural frequencies and mode shapes) as inputs and applies model updating to identify mechanical properties (Young's and shear moduli) of the joint. One of the benefits of this method is that it is not restricted to shear tests as many other standard adhesive tests are. The other advantage of the method is that one can implement it to any models regardless of relative impedance of the joint with respect to adherent's. In this method it is assumed that Finite Element (FE) model of all adherents are coincident with their experimental modal parameters. Therefore any discrepancies between FE model of the structure and experiments are justified with miss-modeling of the joint (here adhesive). Meshki et al. [8] used the method developed by Nobari to investigate the effects of initial bonding imperfection of adhesive joints on equivalent identified Young's and shear moduli.

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The research demonstrated to be robust enough to detect imperfections of 1% of bonding area. Jahani and Nobari [9] have developed the identification method to identify effective, dynamic Young's and shear moduli and also damping characteristics of the adhesive joint, at the same time. This method uses the same philosophy as used by Nobari. In this method both Young's and shear moduli can be identified simultaneously. Jahani and Nobari demonstrated their Identification method by implementing it on a simple joint where in the regarding mode shapes (bending and shear modes) were fully separated. The investigation done by Meshki et al. showed many specific characteristics that belong to modes which have a mixed-shape of bending and shear and it should be noticed that in real-life structures, mode shapes could be complex and imply a mix of normal and shear stress in the adhesion zone.

In this research, effective, dynamic damping characteristics of a nominated flexible structural adhesive have been investigated. Three lap-joint specimens have been built and identification process has been performed to determine, effective, frequency and mode shape-dependent damping, Young's and shear moduli of adhesive. The specimens are designed such that they have complex mode shapes, in order to investigate dynamic damping characteristics of adhesively bonded joints in mixed mode shapes.

2. Theory

As the method used in this research is comprehensively explained by Jahani and Nobari [8], here for sake of convenience, a summary of it is presented. First some definitions are made:

X True model derived from experiment

A Analytical model or FE model of the structure

Δ Miss-modelled parameter which by adding it to A, discrepancies between FE model of structure and experimental data is emitted.

Therefore the correcting structural matrices Δ can be calculated as:

$$[M_{\Delta}] = [\Delta M] = ([M_x] - [M_a]) \tag{1}$$

By assuming that coordinates of the structures A and Δ are fully compatible, the equation of motion of coordinates of A is:

$$[\mathrm{M}]_{\mathrm{n_a.n_a}}\{\ddot{\mathrm{x}}_{\mathrm{a}}\}_{\mathrm{n_a.1}} + [\mathrm{C}]_{\mathrm{n_a.n_a}}\{\ddot{\mathrm{x}}_{\mathrm{a}}\} + [\mathrm{K}]_{n_a.n_a}\{\mathrm{x}_a\}_{n_a.1} = \{\mathrm{f}\}_{n_a.1} \tag{2}$$

Where n_a is the number of DOFs of A.In Eq. (2) $\{\mathrm{f}\}_{n_a.1}$ is the interfacing forces between A and Δ. By using state-space technique, Eq. (1) can be restated as below:

$$\begin{bmatrix} [\mathrm{C}] & [\mathrm{M}] \\ [\mathrm{M}] & \varnothing \end{bmatrix}_{2n_a.2n_a} \begin{Bmatrix} \dot{q}_1 \\ \dot{q}_2 \end{Bmatrix}_{2n_a.1} + \begin{bmatrix} [\mathrm{K}] & \varnothing \\ \varnothing & -[M] \end{bmatrix}_{2n_a.2n_a} \begin{Bmatrix} q_1 \\ q_2 \end{Bmatrix}_{2n_a.1} = \begin{Bmatrix} \{f\} \\ \varnothing \end{Bmatrix}_{2n_a.1} \tag{3}$$

Where in Eq. (3):

$$q_1 = x, q_2 = \dot{x} \tag{4}$$

Now by changing our perspective from updating to joint identification and noticing that we already assumed that the only element that is needed to be updated is the joint, we can modify Eq. (3) to calculate correcting matrices of the joint part of the structure.

$$[A_\Delta]=\begin{bmatrix}[C_\Delta] & [M_\Delta]\\ [M_\Delta] & \varnothing\end{bmatrix},[B_\Delta]=\begin{bmatrix}[K_\Delta] & \varnothing\\ \varnothing & [M_\Delta]\end{bmatrix} \tag{5}$$

$$[A]\begin{Bmatrix}\dot{q}_1\\ \dot{q}_2\end{Bmatrix}+[B]\begin{Bmatrix}q_1\\ q_2\end{Bmatrix}=\{F\} \tag{6}$$

The solution of Eq. (6) is:

$$\begin{Bmatrix}q_1(\mathrm{t})\\ q_2(\mathrm{t})\end{Bmatrix}=\begin{Bmatrix}\{\overline{X}\}\\ s\{\overline{X}\}\end{Bmatrix}e^{st}=\{\overline{q}\}e^{st} \tag{7}$$

And therefore:

$$[\mathrm{s}[\mathrm{A}_\Delta]+[\mathrm{B}_\Delta]]\{\overline{q}\}=0 \tag{8}$$

Eq. (8) is a classic generalized eigenvalue problem. Because the structure is assumed to be damped, all eigenvalues and eigenvectors have real and imaginary parts. Now first by separating modes into kept ones which belong to the frequency range of interest and emitted ones which are out of the range of investigation and then by categorizing coordinates into corrected ones related to the adherents and coordinates related to the joint part, Eigenvector of the i-th mode and its respective complex conjugate can be expressed as:

$$[\Psi_a]=\begin{bmatrix}[\Psi_{akc}] & [\Psi_{aec}]\\ [\Psi_{akj}] & [\Psi_{aej}]\end{bmatrix},\{\psi\}_i=\begin{Bmatrix}\{\psi'\}_i\\ s_i\{\psi'\}_i\end{Bmatrix},\{\psi^*\}_i=\begin{Bmatrix}\{\psi^*\}_i\\ s_i^*\{\psi^*\}_i\end{Bmatrix} \tag{9}$$

Now by transforming real coordinates of A to principal coordinates $\{P_a\}$, one can reach to:

$$\begin{Bmatrix}q_1\\ q_2\end{Bmatrix}=\{q_a\}_{2n_a.1}=[\Psi_a]\{P_a\} \tag{10}$$

For structure Δ:

$$[\mathrm{A}_\Delta][\Psi_a]\{\dot{\mathrm{P}}_a\}+[\mathrm{B}_\Delta][\Psi_a]\{\mathrm{P}_a\}=-\{\mathrm{F}\} \tag{11}$$

Now by pre-multiplying Eq. (11) with $[\Psi_a]^T$ and using orthogonality property of eigenvectors $[\Psi_a]$, one will have:

$$\{\dot{P}_k\}-[s_{ak}]\{P_k\}=[\Psi_k]^T\{F\} \tag{12}$$

$$\{\dot{P}_e\}-[s_{ae}]\{P_e\}=[\Psi_e]^T\{F\} \tag{13}$$

Where $[s_{ak}],[s_{ae}]$ are diagonal matrices which represent complex eigenvectors and their respective complex conjugates. Assuming that the updated structure $(X=A+\Delta)$ is vibrating at its r-th frequency of damped vibration $s_{x_r}=-\xi_{x_r}\omega_{x_r}\pm i\omega_{x_r}\sqrt{1-\xi_{x_r}^2}$, Eq. (11) can be rewritten as:

$$\left(s_{x_r}[A_\Delta]+[B_\Delta]\right)\left([\Psi_k]\{P_{x_k}^a\}_r+[\Psi_e]\{P_{x_e}^a\}_r\right)=-\{F\} \tag{14}$$

Assuming that the maximum frequency of interest is much lower than the minimum frequency out of the study range $\left\| s_{a_k} \right\| \ll \left\| s_{a_e} \right\|$, the inertial of the higher modes are negligible relative to the stiffness of the eliminated modes and the Eq. (13) can be rewritten as follow:

$$\left\{ P_{x_e}^a \right\}_r = -\left[s_{a_e} \right]^{-1} \left[\Psi_e \right]^T \left\{ F \right\} \tag{15}$$

Now by substituting Eq. (15) into Eq. (14) and doing some algebraic calculations one reaches to:

$$\left\{ F \right\} = -\left(\left(s_{x_r} \left[A_\Delta \right] + \left[B_\Delta \right] \right)^{-1} + \left[R_a \right] \right)^{-1} \left(\left[\Psi_k \right] \left\{ P_{xk}^a \right\}_r \right) \tag{16}$$

Where $\left[R_a \right]$ represents the flexibility matrix of the emitted modes. Now in order to get rid of vector $\left\{ F \right\}$, we can substitute it from Eq. (12) into Eq. (16) as here:

$$\left[\Psi_{akj} \right]^T \left(\left(s_{x_r} \left[A_{\Delta_j} \right] + \left[B_{\Delta_j} \right] \right)^{-1} + \left[R_{aj} \right] \right)^{-1} \left(\left[\Psi_{akj} \right] \left\{ P_x^a \right\}_r \right) = \left(\left[s_{a_k} \right] - s_{x_r} \right) \left\{ P_x^a \right\} \tag{17}$$

It should be noted here that vector $\left\{ P_x^a \right\}$ is calculated from experimental eigenvector $\left\{ \psi_x \right\}_r$ from Eq. (18)

$$\left\{ P_x^a \right\}_{r_k.1} = \left[\Psi_{ak} \right]_{n_x.k}^{+} \left\{ \psi_x \right\}_{r_{n_x}.1} \tag{18}$$

Now if we assume that $\left\| \left[R_{a_j} \right] \right\| \ll \left\| \left(s_{x_r} \left[A_{\Delta_j} \right] + \left[B_{\Delta_j} \right] \right)^{-1} \right\|$ or in other words if we consider much enough kept modes, Eq. (17)

$$\left[\Psi_{akj} \right]^T \left(\left(s_{x_r} \begin{bmatrix} \left[\Delta C \right]_j & \left[\Delta M \right]_j \\ \left[\Delta M \right]_j & \varnothing \end{bmatrix} + \begin{bmatrix} \left[\Delta K \right]_j & \varnothing \\ \varnothing & -\left[\Delta M \right]_j \end{bmatrix} \right)^{-1} \right)^{-1} \left(\left[\Psi_{akj} \right] \left\{ P_x^a \right\}_r \right) = \left(\left[s_{a_k} \right] - s_{x_r} \right) \left\{ P_x^a \right\} \tag{19}$$

With regard to state-space technique, the upper half and lower half of each column of $\left[\Psi_{akj} \right]$ and its respective complex conjugate are related to each other as follow:

$$\left[\Psi_{akj} \right] = \begin{bmatrix} \ldots, \begin{Bmatrix} \left\{ \psi' \right\} \\ s_i \left\{ \psi' \right\} \end{Bmatrix}, \begin{Bmatrix} \left\{ \psi'^* \right\} \\ s_i^* \left\{ \psi'^* \right\} \end{Bmatrix}, \ldots \end{bmatrix} = \begin{bmatrix} \left[\Psi^u \right] \\ \left[\Psi^l \right] \end{bmatrix} \tag{20}$$

As the mass of the joint part (here cured adhesive) can easily be measured with good enough accuracy, here we assume that $\left[\Delta M \right]_j = \varnothing$ so after some simplifications, Eq. (19) will reduce to:

$$\left[s_{x_r} \left[\Psi^u \right]^T \left[\Delta C \right]_j \left[\Psi\ \ \right] + \left[\Psi^u \right]^T \left[\Delta K \right]_j \left[\Psi^u \right] \right] \left\{ P_x^a \right\}_r = \left(\left[s_{a_k} \right] - s_{x_r} \right) \left\{ P_x^a \right\}_r \tag{21}$$

Now if we assume a proportional damping for the joint part (adhesive) and if the initial guess for the stiffness of the joint is $\left[K_o \right]_j$ then the correcting matrices would be like:

$$\left[\Delta C \right]_j = \Delta \beta_j \Delta E \left[K_o \right]_j \; and \left[\Delta K \right]_j = \Delta E \left[K_o \right]_j \tag{22}$$

3. Analytical model

As already explained in the theory section, the analytical model of the structure (FE) will be updated to its experimental modal data. The dimensions of the lap-joint are presented in Fig. 1. The adherents are solid bars made of commercial Aluminum and the adhesive is Sikaflex 255-FC. More details of the adhesive are presented in Ref. [9]. The beams can be modelled with line elements while the adhesive should be modelled with solid elements. Therefore in order to keep compatibility, a Multi-Point Constraint (MPC) technique needed to be implemented. The FE model of the lap-joint is presented in Fig. 2.

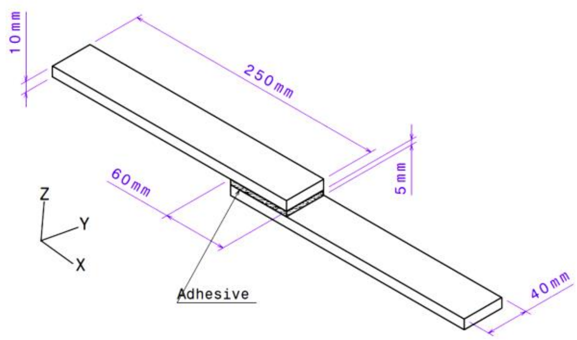

Fig. 1. Dimensions of the lap-joint specimen

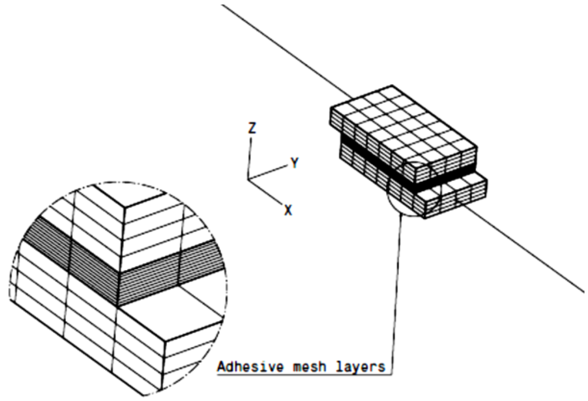

Fig. 2. Finite element model of the lap joint

In this research the frequency range of interest was chosen between 0 to 1600 Hz. This range covers from low speed wind turbines to high speed micro-jet engines. Because former studies have revealed that adhesives behave differently in their bending modes and shear modes [8], mode shapes are categorized into two groups. In this research a mode that creates mostly normal stress on the adhesion surface (regardless of imposing normal or shear stress in the adhesive layers) is called a bending mode, otherwise if it creates dominant shear stress on the adhesion surface is recognized as a shear mode. With the aim of visualizing the mode shapes, an initial guess of 5 MPa is made for the adhesive Young's modulus and Fig. 3 to Fig. 9 represent the mode shapes of the structure. In the frequency range of interest, there exist seven modes which consist of four bending and three shear modes.

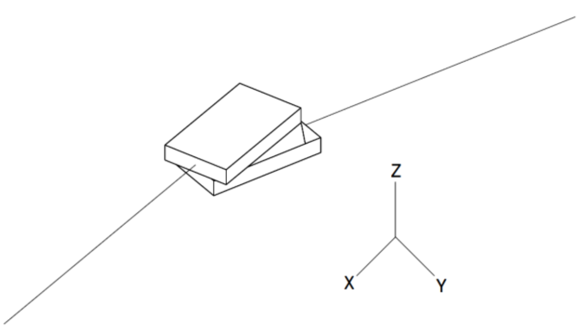

Fig. 3. First shear mode

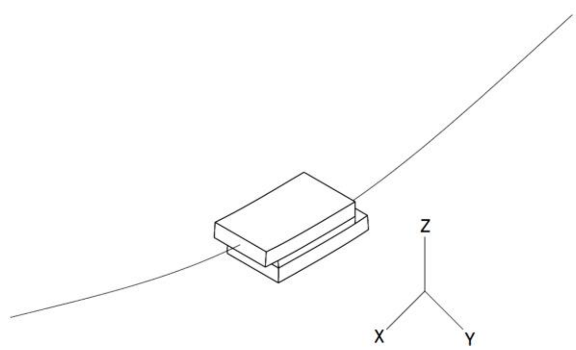

Fig. 4. First bending mode

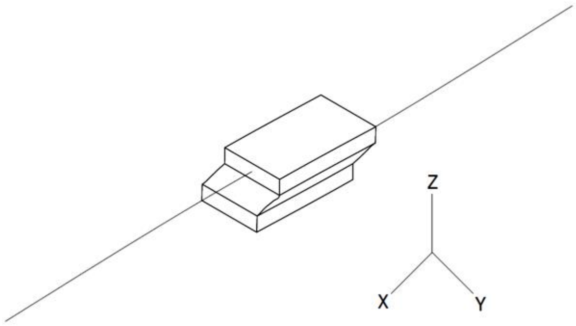

Fig. 5. Second shear mode

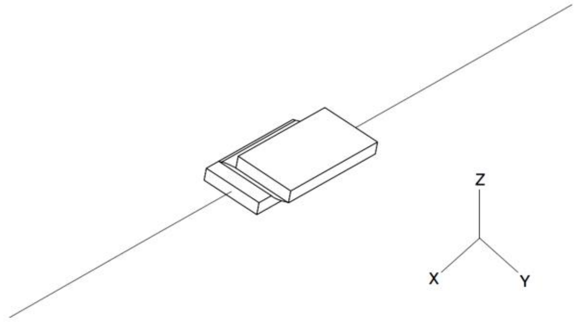

Fig. 6. Third shear mode

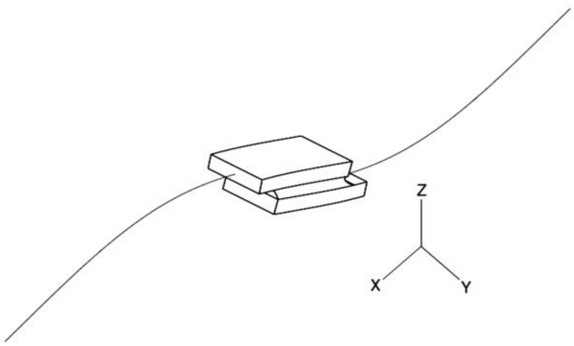

Fig. 7. Second bending mode

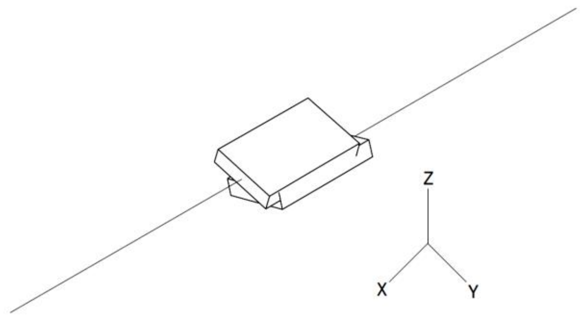

Fig. 8. Third bending mode

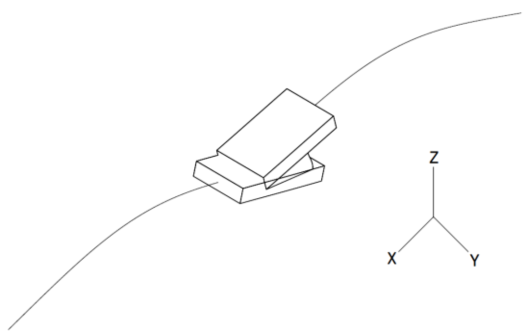

Fig. 9. Fourth bending mode

4. Experimental analysis

As already expressed in section 2 of this paper, in order to isolate the discrepancies of modal data of experimental model and the analytical model of the adhesive joint, mechanical properties of adherent substructures should be updated using the relevant experimental modal data. This can guarantee that the differences of modal data between analytical model and experimental one is just due to mis-modeling of mechanical properties of the adhesive part of the joint. Therefore before preparation of specimens each single Aluminum bar was updated using Inverse Eigen Sensitivity method. Then the bonding area of each bar was prepared by sandpaper to have a smooth area and then washed by the chemical solution recommended by the producer of the adhesive and then dried with a clean cotton cloth. After that a film of primer which is also recommended by the producer was applied on the bonding surface and after 30 minutes the adhesive was applied on the bonding surface by a gun. A sheet of 5-mm Balsa wood was used to control the adhesive thickness. The specimens where kept in a clean area with room temperature for at least 10 days to be fully cured.

In order to perform experimental modal analysis, a free-free suspension setup has been chosen to avoid difficulties associated with a suitable and reliable clamp setup. Specimens were suspended from a pair of 30-cm strings. After performing modal analysis, the Modal Assurance Criteria (MAC) matrix was used to pair experimental and FE mode shapes. Table 1. shows experimental parameters of the fourth bending mode.

Table 1. Experimental modal data of the 4th bending mode

	Freq. (Hz)	Deviation (%)	Loss factor	Deviation (%)
Spec.1	1452.05		0.0464	
Spec.2	1413.37	0.73	0.0418	13.61
Spec.3	1449.34		0.0545	

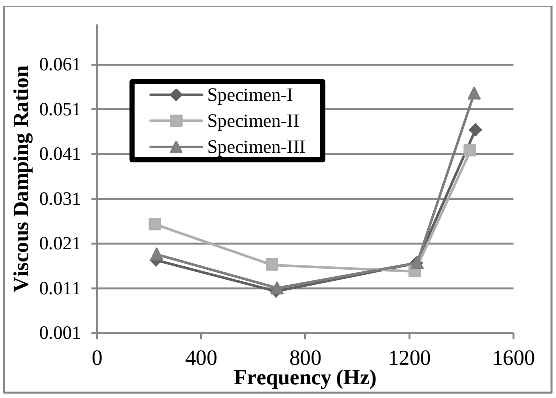

Fig. 10. Loss factor of bending modes

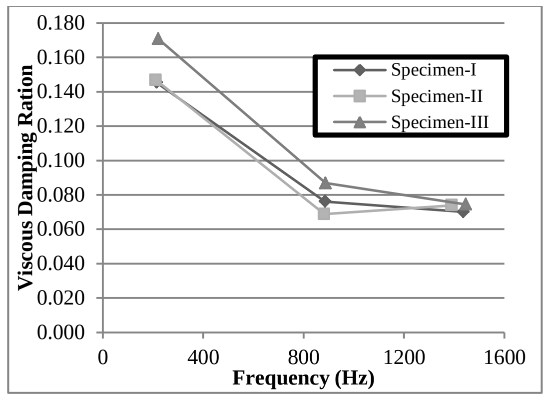

Fig. 11. Loss factor of shear modes

5. Joint identification results

In order to calculate correcting matrices $\left[\Delta K\right]$ and $\left[\Delta C\right]$, one can solve eq.(21) for all modes simultaneously, using least square technique. However, because of viscoelastic nature of adhesive, its mechanical properties are functions of frequency [8-9], in this investigation each natural frequency and mode shape was updated to its respective experimental data, one by one. Bending modes were used to identify Young's modulus and shear modes for shear modulus. Fig. 12 and 13 shows the identified, effective, dynamic Young's and shear moduli of the adhesive. It can be seen that the Young's and shear moduli of adhesive increase by frequency and then slow down to some magnitude. This is coincident with [8-9]. The stiffness multiplier β_j of the proportional damping also identified and presented in Fig. 14 and 15. β_j can't express damping characteristic of the adhesive alone. By multiplying it by identified Young's and shear moduli, damping property of the adhesive can be calculated as are presented in Fig. 16 and 17. Like Young's and shear moduli, dynamic damping represents significant dependence on frequency. For both bending and shear modes, damping reduces with frequency. It is important to notice that although the loss factor and β_j has different order of magnitude for bending and shear modes, the damping property of bending and shear mode is in the same order of magnitude. This important phenomenon reveals the importance of mode shape in structural dynamics of adhesively bonded joints and dictates the importance of identification of adhesive joints. It is demonstrated that mode shape has as much important information about the adhesive joint as frequency and in order to optimize the design of an adhesively bonded structure, one should take both frequency and mode shape effect on the adhesion zone into account**.**

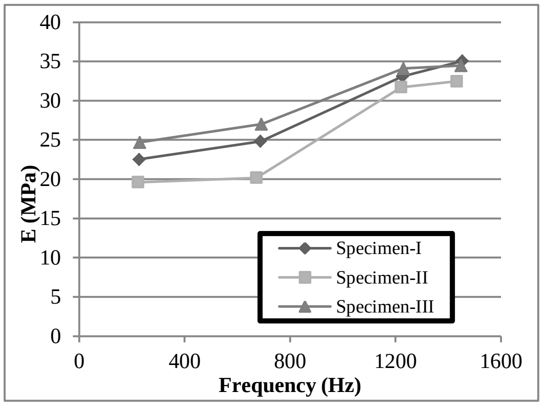

Fig. 12. Identified Young's modulus

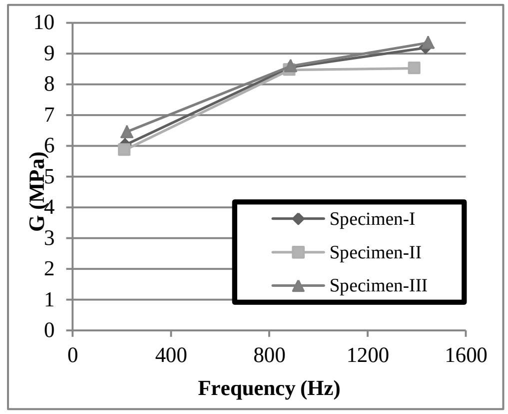

Fig. 13. Identified shear modulus

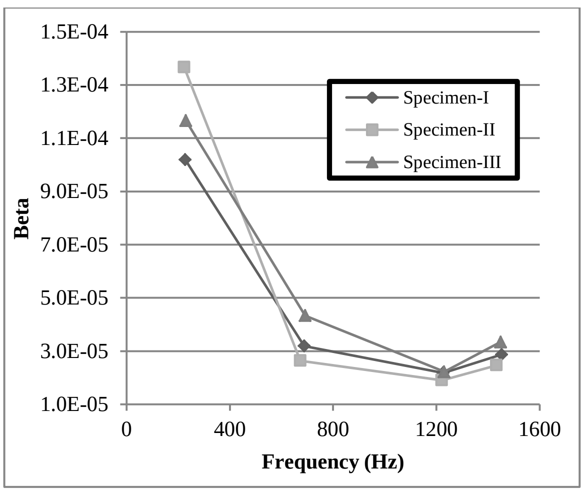

Fig. 14. Identified β_j of bending modes

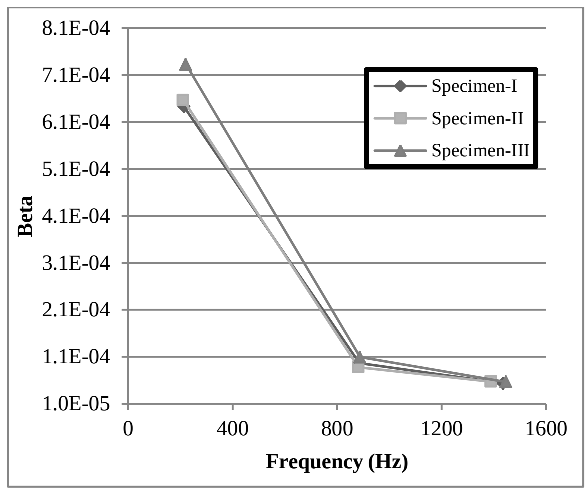

Fig. 15. Identified β_j of shear modes

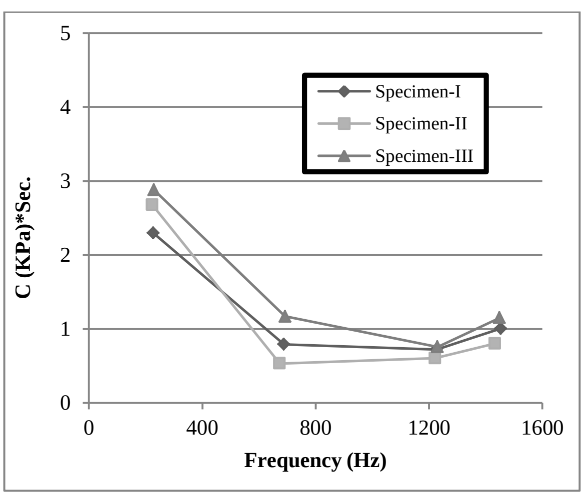

Fig. 16. Identified damping of bending modes

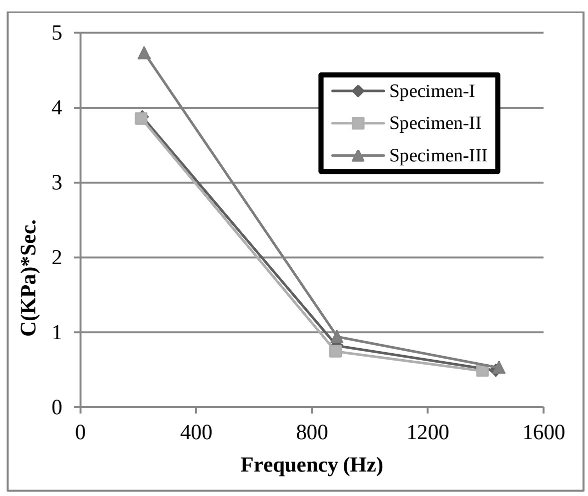

Fig. 17. Identified damping of shear modes

6. Conclusion

In order to investigate dynamic damping characteristics of a nominated adhesive, three identical lap-joint specimen where produced. Experimental modal data (natural frequencies, mode shapes and loss factors) of the joints were extracted and a direct modal based joint identification method was implemented to identify dynamic mechanical properties of the adhesive.

It has been revealed that as already expected, adhesive shows frequency dependency. Both identified Young's and shear moduli will increase with frequency and then slow down to some magnitude. Modal damping of the adhesive is more evident in shear modes with respect to bending modes. For shear modes, experimental viscously damping loss factor increases with frequency but for first three bending modes, damping reduces but there is a rise in the fourth bending mode. This is because the fourth bending mode can hardly be distinguished between shear and normal mode. Although the fourth mode is dominantly creating normal stress, but because the share of shear stress is relatively significant, the experimental damping ration of it is in the order of shear modes. This demonstrates the importance of joint identification for an adhesive. Identified damping of the adhesive shows a dominant reduction with respect to frequency (except for the fourth bending mode which increases a bit) and the interesting phenomenon is that for both bending and shear mode, damping characteristics is in the same order. It can be interpreted that because of the different behavior in bending and shear, although implementation of standard adhesive tests is mandatory, in order to evaluate FE model of an adhesively bonded structure operating in a vibratory environment, a joint identification can provide much more important information about the joint and consequently, the dynamic of the structure.

References

1. R.D. Adams, J. Comyn, W.C. Wake, Structural Adhesive Joints in Engineering, Elsevier Applied Science Publishers, London, 1997

2. R.W. Messler, Joining of Advanced Materials, Butterworth-Heinemann, Boston, 1993

3. A. Dellard, Advances in Structural Adhesive Bonding, Wood head Publishing Limited, Cambridge, London, 2010

4 .H. Xiaocong, A review of finite element analysis of adhesively bonded joints, Int. J. Adhes. Adhes., 31(2011) 248-264.

5. L. Gaul, S. Bohlen, Identification of nonlinear structural joint models and implementation in discretized structure models, Proceeding of the 11th ASME Conference on mechanical vibration and noise, Boston, USA, (1987) 213-219.

6. S.M.S. Sadati, A.S. Nobari, T. Naraghi, Identification of a nonlinear joint in an elastic structure using optimum equivalent linear frequency response function. Acta Mech. 223 (2012)1507-1516.

7. A.S. Nobari, D.A. Robb, D.J. Ewins, A new approach to modal-based structural dynamic model updating and joint identification. Mechanical systems and signal processing 9(1)(1995) 85-100.

8. M.M. Meshki, A.S. Nobari, K. Nikbin, Study of surface bonding imperfection effects on equivalent identified dynamic Young's and shear moduli using a modal based joint identification method", MSSP. (2014)

9. A.S. Nobari, K. Jahani, Identification of Damping Characteristic of a Structural Adhesive by Extended Modal Based Direct Model, Exp. Mech. 49(2009)785-798.

10. Sikaflex-255FC, Technical data sheet, http://usa.sika.com/dms/getdocument.get/5d88ecdb-803a-368a-9a8a-3a43bb2ec065/ipd-pds-sikaflex255FC-us.pdf