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Damping strategies for steel lattice sandwich constructions

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Abstract. A square steel sandwich plate with lattice corrugated core is explored for damping improvement. A range of damping materials are filled inside the openings provided by the corrugated core, or are applied on the surfaces of the facesheets. The dynamic properties such as natural frequency and damping factor are experimentally measured for the sandwich plate with each filling solution. The relative performance of each insertion is compared in terms of damping capacity and added mass.

Keywords: steel sandwich constructions; lattice structures; damping materials; vibration test; damping capacity

1. Introduction

Sandwich panels have been used for many years as light-weight structural components to resist static loading such as roofs of houses or industrial buildings (Zenkert 1995). Recently, industrial interest exists in using hollow tubes with steel sandwich walls for the ram (or the moving head) of milling machines, see for example Meo *et al*. (2005), Mai *et al*. (2007) and Srikantha Phani *et al*. (2006). It is imperative that the ram is stiff and light to ensure a high accuracy of the finished product at a high speed of operation, thus reducing the machining time. It is also required that the damping capacity of the ram is high to minimise vibration response levels. Thus, the requirements for an ideal ram structure are: high stiffness, high damping and low mass. As yet untried, the usage of sandwich walls for hollow tubes can be attractive from two considerations: a sandwich construction eliminates the need for stiffeners, giving lower mass compared to a monolithic hollow tube for comparable stiffness (Budiansky 1999, Mai *et al*. 2007), and the opened architecture provided by a lattice sandwich core can be used for inserting additional damping materials (Mai 2014, Srikantha Phani *et al*. 2006).

As the first step, the current study will explore a simpler structure of sandwich construction: a square sandwich plate with lattice corrugated core for damping enhancement, see Fig. 1. The design of this plate is identical to the proposed design of the sandwich walls of a typical ram machine (Srikantha Phani *et al*. 2006), with mild steel chosen as the parent material for both facesheets and corrugated core. Although many studies have been conducted on the vibration characteristics of honeycomb sandwich beams and panels made from aluminium and composite

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materials (see for example Amith Kumar *et al*. 2013, Boucher *et al*. 2013, Huang *et al*. 2005, Maheri *et al*. 2008, Renji and Shankar Narayan 2002, Wang and Yang 2000), little is known about the damping properties of steel corrugated-core sandwich constructions.

The usage of materials inserted inside the cavities of structures for damping improvement has been a research topic for many years. The filling materials are usually classified into two categories: *active damping* such as smart fluids (Srikantha Phani and Venkatraman 2005, Yeh 2013), piezoelectric layers (Baillargeon and Vel 2005), and *passive damping* such as rubbers (Huang *et al*. 2005), energy absorbing foams (Woody and Smith 2006), sand (Sun *et al*. 1986), polymeric or elastomeric beads (McDaniel and Dupont 2000, Park and Palumbo 2009), and syntactic foams (Amith Kumar *et al*. 2013). In our current study, it is more suitable and practical to apply passive damping materials (sand, beads and foams) into the ram structure. Sun *et al*. (1986) investigated the damping loss factor of structures containing sand, and found that the thickness of the sand layer determined the frequencies at which the maximum damping occurred. McDaniel and Dupont (2000) measured the damping of a steel box beam inserted with elastomeric beads, and showed that the damping would initially be high and decrease until the granular fill (the beads) reached “steady state” motion in response to forcing by the beam. Park and Palumbo (2009) conducted a study on the damping of sandwich beams filled with light-weight particles made of polyimide. They found that the frequency of maximum damping and the amount of damping obtained depended on the properties of the particles such as type, size and mass of the polymer. Recently, Amith Kumar *et al*. (2013) have explored honeycomb sandwich composite panels filled with syntactic foam, and showed that the frequency and damping ratio of these structures are significantly influenced by the cell size of the honeycomb infused in the syntactic foam core.

In this study, a range of damping materials are alternately filled inside the openings provided by the corrugated core of the sandwich panel in Fig. 1, or are applied on the surfaces of the facesheets. The dynamic properties such as natural frequency and damping factor are experimentally measured for the sandwich plate with each filling solution. The relative performance of each insertion is compared in terms of damping capacity and added mass. Consequently, the results of the current study can provide the database for the potential development of hollow tubes with steel sandwich walls for milling machine applications.

The outline of this study is as follows. First, in Section 2, the design and manufacture of a steel sandwich plate with corrugated core is described, and various damping materials with different filling methods are alternatively applied to the plate. In Section 3, vibration tests are performed in order to determine the natural frequencies and damping factors of the sandwich plate with each damping insertion. In Section 4, the measured natural frequencies of the bare plate are compared with finite element (FE) predictions in order to assess the stiffness of the fabricated structure, and the relative merit of the plate filled with competing damping materials is discussed. Section 5 concludes the paper with some concluding remarks on the suitable choices of damping improvement for steel sandwich constructions.

2. A steel sandwich plate and damping fillings

A square sandwich plate with corrugated core was designed with dimensions shown in Fig. 1. The manufacturing route of the plate is as follows. First, the facesheets were cut into designed dimensions from planar mild steel sheets, while the sandwich core was manufactured by folding a plane steel sheet into the designed corrugated configuration, see Fig. 1. Second, the facesheets and

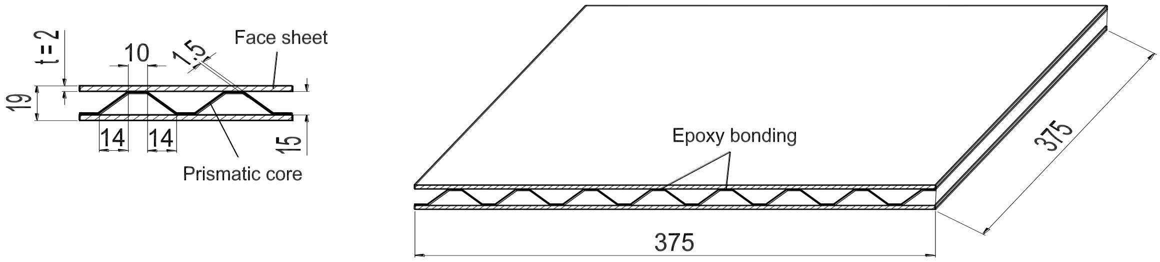
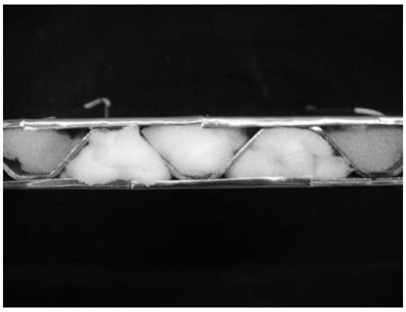
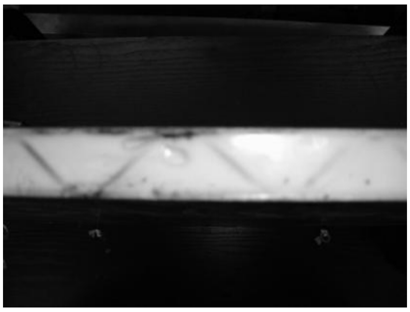


Fig. 1 A corrugated-cored sandwich panel made from mild steel. (Dimensions are in mm)

(a) Polystyrene beads (mass ratio = 1.003)



(b) Polymeric foam (mass ratio = 1.020)



(c) Syntactic foam (mass ratio = 1.190)

(d) Damping tape (mass ratio = 1.040)

(e) Sand (mass ratio = 1.450)

Fig. 2 The sandwich plate filled with different damping materials: (a) polystyrene beads; (b) polymeric foam; (c) syntactic foam; (d) damping tape; and (e) sand. The mass ratio is defined as the ratio of the filled plate to the bare plate the core were bonded together by adhesive epoxy DP490 from 3M Scotch-Weld company. A manual applicator gun was used to apply adhesive layers uniformly on the bonding areas between the facesheets and the core, and the adhesive was allowed to cure under pressure applied by deadweights. The total mass of the manufactured sandwich plate is m = 6.080 kg.

Various damping materials were added to the sandwich plate (Figs. 2(a)-(e)); for each case, the mass ratio is defined as the ratio of mass of the filled plate to that of the bare plate. Either damping tapes (type 2552, from 3M Scotch-Weld company) was placed on the facesheets (Fig. 2(d)), or the cavity of the sandwich core was inserted with some damping materials. The core was filled with

- Polystyrene beads (diameter of 2 to 5 mm), Fig. 2(a).
- Polymeric foam (from Caligen LTD company), Fig. 2(b).
- Syntactic foam (a mixture of glass microbubbles and resin[1]), Fig. 2(c). The microbubbles and resin were uniformly mixed in the ratio of 50:50 by weight, giving the foam material properties as: Young's modulus E_f = 1.5 GPa, Poisson ratio ν_f = 0.26, density ρ_f = 706 kg/m^3.
- Sand, Fig. 2(e).

Polystyrene beads, polymeric foam, and sand were loosely filled inside the corrugated core with the plate edges closed by sellotape (or electrical tape). These materials could be removed easily from the plate due to the loose fillings. In contrast, the syntactic foam had a strong coupling with the sandwich plate due to the chemical reaction between microbubbles and resin during the synthesis process of the foam. Thus, this can give an increase in both the stiffness and the damping capacity of the plate.

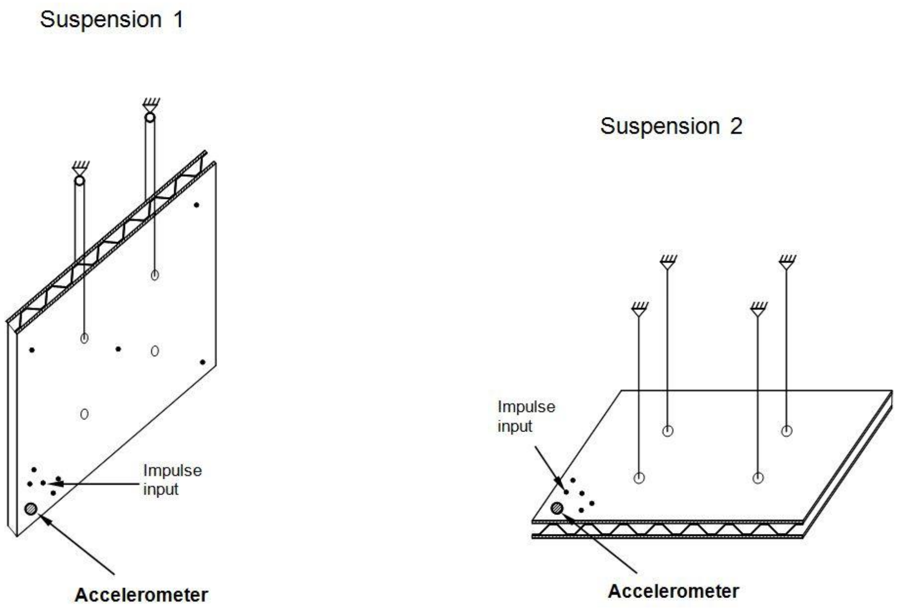
3. Vibration tests

Vibration tests were performed to determine the dynamic properties of the sandwich plate with each damping filling. The natural frequencies and damping (Q factor) were measured for the first 5 vibration modes of the plate within the frequency range 0-1.5 kHz. Great care was taken to support the sandwich plate at only nodal points to preserve the "free" boundary conditions. These points were identified at which nodal lines of different modes intersect, and small holes were drilled there. In order to explore the sensitivity of the measured damping to the support configurations, two different positions of the elastic suspension threads were used, see Fig. 3.

The testing procedure was as follows. An impulse excitation (input) as shown in Fig. 4 was applied to randomly chosen points at corners of the plate (positions of antinodes of vibration, see Fig. 6) using an instrumented hammer (with force transducer PCB-218C). The vibration response (acceleration) was measured using an accelerometer mounted by sticky glue at one corner of the plate, see Fig. 3. The signals from the impulse hammer and the accelerometers were passed through charge amplifiers (DJB CA 4270) and converted to digital form onto a computer using a data acquisition card (NI DAQ 6023 E). The data was acquired for a total duration of 5 seconds at 5 kHz sampling rate, with a provision made for pre-triggering. This time duration is adequate to capture the entire decaying response, see Fig. 4. This gives a frequency resolution of 0.2 Hz.

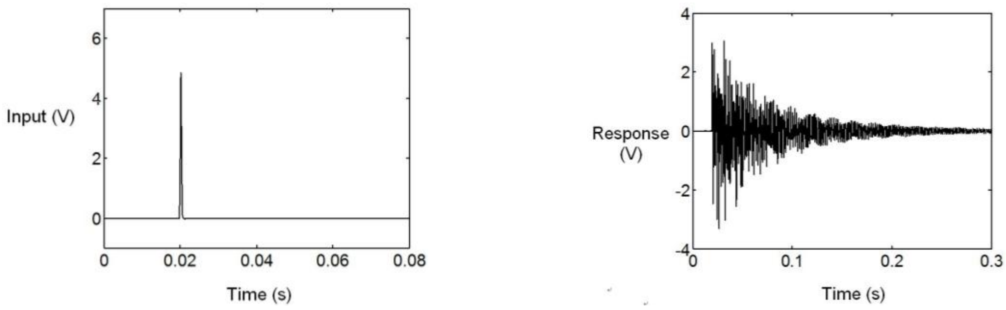
Typical time series data for the impulse and acceleration response are shown in Fig. 4. The frequency response function (FRF) $H(\omega)$ is calculated from these time series data using the definition

[1] 3M Glass bubbles K15, and UKR 137 epoxy resin from 3M Scotch-Weld company.



(a) (b)

Fig. 3 Experimental set-up for vibration tests. The plate is tested under 2 free-free support conditions: (a) 2-string support; and (b) 4-string support



(a) (b)

Fig. 4 Typical time history data for input (impulse) and output (acceleration) response

$$H(\omega) \equiv \frac{O(\omega)}{I(\omega)} \tag{1}$$

where $O(\omega)$ and $I(\omega)$ respectively denote the Fourier transforms of the output and input time signals in Fig. 4.

It is standard practice (Ewins 2000) to measure the average of the FRF by repeating the experiment several times (typically 10 times) so that the coherence function can be calculated, which gives a quality check on the measured FRF. The coherence function $\gamma^2(\omega)$ at each frequency point is defined as (McConnell 1995)

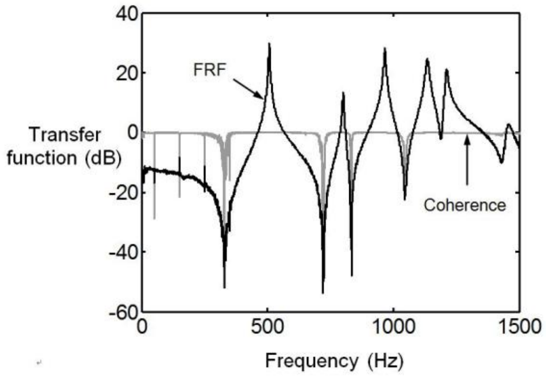


Fig. 5 Frequency response function (FRF) along with the coherence (shown in grey colour) of the plate with damping tape in 2-string support condition

$$\gamma^2(\omega) \equiv \frac{\left|S_{xy}(\omega)\right|^2}{S_{xx}(\omega)S_{yy}(\omega)} \tag{2}$$

where S_{xy} denotes the cross-correlation between the input and the output while S_{xx} and S_{yy} respectively denote the auto-correlation functions of the input and output signals. Coherence is a measure of linearity between the input and output: values close to unity (zero on log scale) indicate adequate linearity (McConnell 1995).

For a typical example of the measurement, the amplitude of the averaged FRF and the coherence are given in Fig. 5. Good coherence was seen for all relevant frequencies from the lowest resonance up to 1.5 kHz. Modal parameter identification algorithms using circle-fitting (Nyquist plot) were applied to the FRF data surrounding each peak in the amplitude spectrum in Fig. 5 in order to estimate the natural frequency and Q factor.

4. Results and discussions

The predicted and measured vibration properties of the first 5 modes up to 1.5 kHz are shown in Tables 1A and 1B, corresponding to the two suspension configurations in Figs. 3(a) and (b), respectively. The values are reported for the bare sandwich plate with and without filling materials. Overall, the results are sensitive to the choice of suspension for some damping insertions and vibration modes (e.g., sand) but not for others (e.g., bare plate and damping tape).

4.1 Natural frequencies and modeshapes of the bare plate: *Predicted and measured results*

Consider first the predicted natural frequencies and mode shapes of the sandwich plate given by finite element (FE) analysis. The commercial finite element software ABAQUS (version 6.9, 2009) has been used to construct a model in order to assess the dynamic performance of the sandwich plate. Both the facesheets and the corrugated core are simulated using 4-noded shell elements with reduced integration (element type *S4R* in the ABAQUS notation). The facesheets

Table 1A Predicted and measured natural frequencies ω and damping Q factors in the 2-string suspension

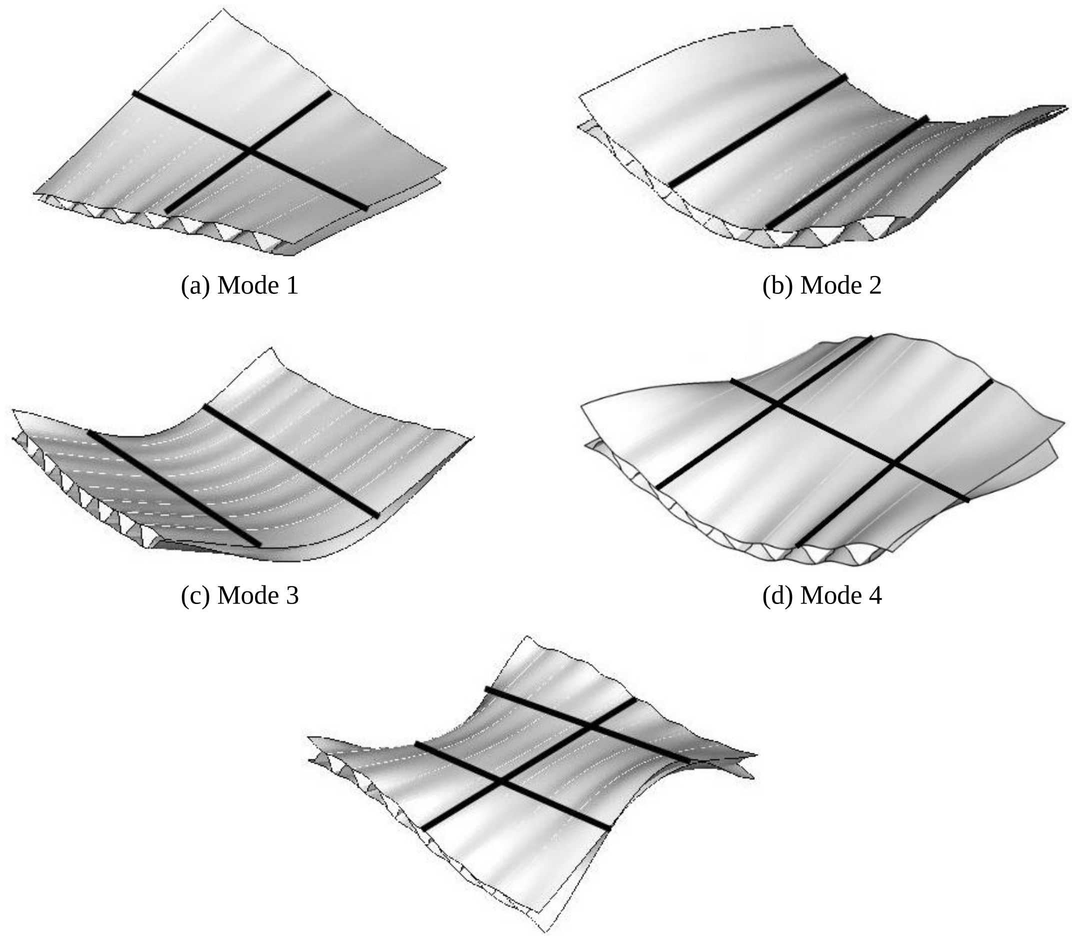
Modes	FEM predictions	Measured natural frequencies and damping Q factor – 2 string suspension											
		Bare plate (m = 6.080 kg)		Beads ($1.003 \times m$)		Foam ($1.020 \times m$)		Damping tape ($1.040 \times m$)		Syntactic ($1.190 \times m$)		Sand ($1.450 \times m$)	
	Frequency (Hz)	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor
1	519	509	769	509	667	505	714	507	147	496	345	438	49.5
2	798	807	714	806	526	801	526	801	156	798	417	682	8.9
3	989	967	345	966	233	958	244	966	159	941	250	833	9.7
4	1146	1144	345	1144	244	1136	135	1135	96	1206	185	1050	5.1
5	1244	1214	345	1214	233	1205	156	1211	114	1233	185	1149	3.8

Table 1B Results in the 4-string suspension

Modes	FEM predictions	Measured natural frequencies and damping Q factor – 4 string suspension											
		Bare plate (m = 6.080 kg)		Beads ($1.003 \times m$)		Foam ($1.020 \times m$)		Damping tape ($1.040 \times m$)		Syntactic ($1.190 \times m$)		Sand ($1.450 \times m$)	
	Frequency (Hz)	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor	Frequency (Hz)	Q factor
1	519	509	714	509	455	505	625	507	147	496	286	431	85
2	798	806	625	806	400	800	333	800	152	796	400	529	3.8
3	989	967	417	967	294	967	161	967	175	940	217	827	6.7
4	1146	1144	333	1144	256	1135	118	1135	98	1204	159	988	6.4
5	1244	1213	345	1214	263	1211	106	1211	123	1231	167	1165	9.4

are tied to the core by tie constraints (rigid connections). The plate is meshed such that 75 elements exist along the width and length directions. Engineering values are applied such that the facesheets and corrugated core have the same material properties as those of the manufactured sandwich plate with Young's modulus E = 200 GPa, Poisson's ratio ν = 0.3, and density ρ = 7800 kg/m^3. The FE simulation is used to predict the first 5 natural frequencies (given in Table 1) and modeshapes (shown in Fig. 6) of the bare sandwich plate in free-free boundary conditions.

It can be seen from Tables 1A and 1B that good agreement exists between the predicted and measured natural frequencies of the bare plate in both suspensions. This indicates that the adhesively bonded sandwich plate meets the desired stiffness. Thus, the predicted modeshapes given by the FE simulation are taken to be reliable, and are shown in Fig. 6. The first mode with a natural frequency of 509 Hz is a pure twisting mode, see Fig. 6(a). The 2nd and 3rd modes are predominantly the first bending mode of a free-free beam, vibrating in two orthogonal planes (perpendicular and along the prismatic direction), Figs. 6(b) and (c). The more complex modeshapes of higher frequencies (modes 4 and 5) are shown in Figs. 6(d) and (e).



(a) Mode 1

(b) Mode 2

(c) Mode 3

(d) Mode 4

(e) Mode 5

Fig. 6 Modeshapes predicted by the FEM simulation

4.2 Natural frequencies of the filled plate: *Observed results*

Consider next the experimentally measured frequencies of the sandwich plate filled with additional materials. Except for the 2nd and 4th modes of the sand-filled plate, the other frequencies of all filling solutions are almost the same in both suspension configurations. For these two modes, the discrepancy of the frequencies can be traced to the loose filling of sand into the plate. In the vertical suspension (the 2-string configuration), the sand is more packed at the bottom of the plate where the accelerometer is mounted, recall Fig. 3. In contrast, the contact between sand and the upper facesheet of the sandwich plate is less tightly packed in the horizontal suspension (the 4-string configuration).

The insertion of additional materials to the sandwich plate adds mass to the structure, recall Fig. 2. For polystyrene beads, polymeric foam and sand, the contribution of these materials to the overall stiffness of the structure is infinitesimal: the natural frequencies are decreased compared to those of the bare plate, see Tables 1A and 1B. For example, sand insertion increases the total mass of the plate by 45%, and all the frequencies of the sand-filled plate are reduced by 5 to 15% (in the 2-string suspension). This can be expected from the theoretical point of view: the frequencies scale with the square root of the ratio of stiffness/mass. For the case of syntactic foam, the foam couples with materials of the sandwich plate, leading to an increase in the total stiffness of the structure. As a result, at high frequencies (modes 4 and 5), the syntactic foam-filled plate has higher natural frequencies (1206 and 1233 Hz) than those of the bare plate (1144 and 1214 Hz), Table 1A.

4.3 Damping Q factor: *Measured results*

Q factor is chosen as a measure of damping: it denotes the dynamic magnification of the static response at resonance. Lower *Q* factors (or high damping) would mean less amplification of the static response and hence low vibration levels. In a free decay, $2\pi/Q$ is the fractional energy loss per period of vibration. The *Q* factor is related to the critical damping factor ζ by $Q = 1/(2\zeta)$ and to the modal loss factor η by $Q = 1/\eta$.

The measured values of the *Q* factors for each mode of vibration are included in Table 1. In any measurement of damping, it is important to be aware that the support method may add damping to the inherent damping of the structure. In order to explore the sensitivity of the observed damping to the support configurations, two different positions of the elastic suspension cords were used, as shown in Fig. 3. For consistency, the lower measured value of damping (higher *Q*) is always taken to be the more reliable - the true modal damping may be lower than either.

Consider first the bare plate. The 2-string suspension gives higher *Q* factor in the range of 345 to 769. This observed damping level is as low as that of a mild steel structure (compared to data from Harris and Piersol (2002)), again confirming that the adhesively bonded sandwich plate meets the stiffness of the designed requirements.

Consider next the method of filling the core cavity with different damping materials. The results are compared with those of the bare plate in terms of mass adding and *Q* factor reduction. First, filling the sandwich plate with polystyrene beads adds very little mass, and increases slightly damping capacity as the *Q* factor is reduced by about 15 to 35% for all modes, see Table 1. Second, polymeric foam insertion increases the total mass by only 2% but decreases *Q* by 10 to 50% for lower modes (the 1st, 2nd, and 3rd modes) and up to 70% for higher modes (e.g., mode 5 in the 4-string suspension). Third, syntactic foam reduces *Q* factor by an average factor of 2, but increases the total mass by 20%. Finally, sand significantly reduces *Q* to the designed range of damping improvement: *Q* is about one order of magnitude. However, sand filling adds a lot of additional mass to the plate by about 50%.

Last, consider the method of surface treatment by 2552 damping tape. The damping tape adds only 4% additional mass to the sandwich plate. The measured *Q* factors are approximately the same for both suspensions, and are in the range of 100 to 180. These values are reduced by a factor of 3 (for higher modes) to 5 (for lower modes), compared to those of the bare plate, Tables 1A and 1B.

5. Conclusions

In this paper, a steel sandwich panel with open-corrugated core has been explored for damping enhancement. The structure was designed, and manufactured by bonding the facesheets to the core using an adhesive epoxy. The measured natural frequencies of the bare plate are in good agreement with the FE predictions. This indicates that the fabricated sandwich plate meets the desired stiffness, and the modeshapes predicted by the FE simulation are reliable.

A range of damping materials were alternately added to the structure by two methods: (i) filling the open-cored cavity with polystyrene beads, polymeric foam, sand, or syntactic foam; and (ii) applying a damping tape on the surfaces of the plate. For the former (i), sand insertion has the highest damping improvement with *Q* factor reduced to be in the desired range (about one order of magnitude). However, the total mass of the sand-filled plate is increased significantly by 45%. For the later (ii), damping tape with only 4% added mass decreases *Q* factor by 3 to 5 times: the reduced *Q* factor is in the range of 100 to 180. Overall, the method of applying damping tape on the surfaces of steel sandwich construction is proposed for damping enhancement due to the small added mass and high efficiency.

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References

3M Scotch-Weld company; URL: www.3m.com

ABAQUS Standard User’s Mannual (2009), Version 6.9.

Amith Kumar, S.J., Sabeel Ahmed, K. and Santosh, M.B. (2013), “Parametric analysis on the vibration characteristics of sandwich composites structure”, *Int. J. Mech. Eng. Rob.*, **1**(1), 80-84.

Baillargeon, B.P. and Vel, S.S. (2005), “Active vibration suppression of sandwich beams using piezoelectric shear actuators: experiments and numerical simulations”, *Intel. Mater. Sys. Struct.*, **16**(6), 517-530.

Boucher, M.A., Smith, C.W., Scarpa, F., Rajasekaran, R. and Evans, K.E. (2013), “Effective topologies for vibration damping inserts in honeycomb structures”, *Com. Struct.*, **106**, 1-14.

Budiansky, B. (1999), “On the minimum weights of compression structures”, *Int. J. Solids. Struct.*, **36**(24),

3677-3708.
Caligen LTD company; URL: www.caligen.co.uk
Ewins, D.J. (2000), *Modal Testing*, (2nd Edition), Research Studies Press Ltd., England.
Harris, C.M. and Piersol, A.G. (2002), *Harris' Shock and Vibration Handbook*, McGraw-Hill, New York, NY, USA.
Huang, H., Joe, C.R. and Kim, D.U. (2005), "Mechanical behavior of ruber-filled multifunctional honeycomb sandwich composite", *Sandwich Structures 7*: *Advancing with Sandwich Structures and Materials*, 671-680.
Maheri, M.R., Adams, R.D. and Hugon, J. (2008), "Vibration damping in sandwich panels", *J. Mater. Sci.*, **43**(20), 6604-6618.
Mai, S.P. (2014), "Experiment on box-section sandwich beams in three-point bending", *J. Sand. Struct. Mater.*, **16**(5), 534-550.
Mai, S.P., Fleck, N.A. and Lu, T.J. (2007), "Optimal design of box-section sandwich beams in three-point bending", *Int. J. Solids. Struct.*, **44**(14-15), 4742-4769.
McConnell, K.G. (1995), *Vibration Testing*: *Theory and Practice*, (1st Edition), Wiley Interscience, New York, NY, USA.
McDaniel, J.G. and Dupont, P. (2000), "A wave approach to estimating frequency-dependent damping under transient loading", *J. Sound. Vib.*, **231**(2), 433-449.
Meo, F., Merlo, A., De la O Rodriguez, M., Brunner, B. and Ippolito, M. (2005), "Advanced hybrid mechatronic materials for ultra precise and high performance machining systems design", *Proceedings of IPROMS* (*Virtual International Conference*), Oxford, UK, July.
Park, J. and Palumbo, D.L. (2009), "Damping of structural vibration using lightweight granular materials", *J. Exp. Mech.*, **49**(5), 697-705.
Renji, K. and Shankar Narayan, S. (2002), "Loss factors of composite honeycomb sandwich panels", *J. Sound Vib.*, **250**(4), 745-761.
Srikantha Phani, A. and Venkatraman, K. (2005), "Damping characteristics of electro-rheological fluid sandwich beams", *J. Acta Mech.*, **180**(1), 195-201.
Srikantha Phani, A., Mai, S.P., Fleck, N.A. and Woodhouse, J. (2006), "A prismatic-core design for a milling machine component", *Technical report*, Cambridge University Engineering Department, UK.
Sun, J.C., Sun, H.B., Chow, L.C. and Richard, E.J. (1986), "Predictions of total loss factors of structures, Part II: Loss factors of sand-filled structure", *J. Sound Vib.*, **104**(2), 243-257.
Wang, B. and Yang, M. (2000), "Damping of honeycomb sandwich beams", *Mater. Proc. Tech.*, **105**(1-2), 67-72.
Woody, S.C. and Smith, S.T. (2006), "Damping of a thin-walled honeycomb structure using energy absorbing foam", *J. Sound Vib.*, **291**(1-2), 491-502.
Yeh, J.Y. (2013), "Vibration analysis of sandwich rectangular plates with magnetorheological elastomer damping treatment", *Smart Mater. Struct.*, **22**(3), 035010.
Zenkert, D. (1995), *An Introduction to Sandwich Construction*, Engineering Materials Advisory Services Ltd., UK.

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