VIBRATION CHARACTERISTICS OF SANDWICH STEEL

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1. MOTIVATION

Noise pollution, as a consequence of heavy vibration, is today's main issue which has impeded the further modernization of railroad projects. Sandwich systems [1,2] may provide a satisfactory solution. Recent advances in controlled production of structural steel plates bonded with high strength adhesive, providing much better structural integrity, have made it possible to think of this material as a full grade structural substitute for steel bridges. This part of the study covers the beam element and the results so obtained are found to be closed to the experiments.

2. ASSUMPTION OF THE ANALYSIS

Following assumptions are made in addition to Euler-Bernoulli beam assumptions.

1. The core primarily undergoes shear deformation which provide the major part of the damping.
2. The face-core bond is assumed perfect i.e. no slip at the interface.
3. Transverse direct strain in both faces and the core is neglected and the system has single transverse displacement coordinate on a cross section.
4. The core functions as linear viscoelastic material and follows Hooke's law in frequency domain.

3. THEORETICAL ANALYSIS

The gist of the mechanism lies in the shear strain- displacement relation of the core as shown in Fig. 1 and Fig. 2. The core shear strain is given by

$$\gamma_c = (\gamma_{cs} + \gamma_{cb}) = \frac{1}{t_c}(\Delta u + d\gamma) \tag{1}$$

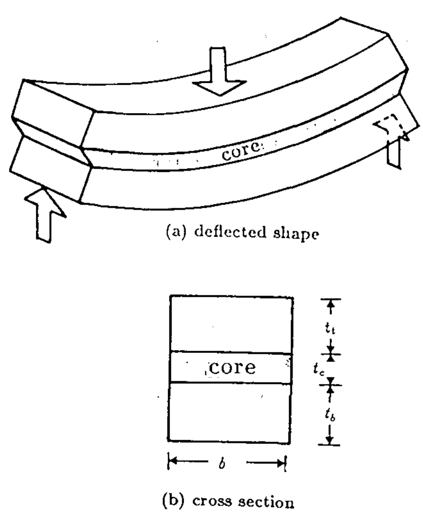

Fig. 1 Sandwich beam system.

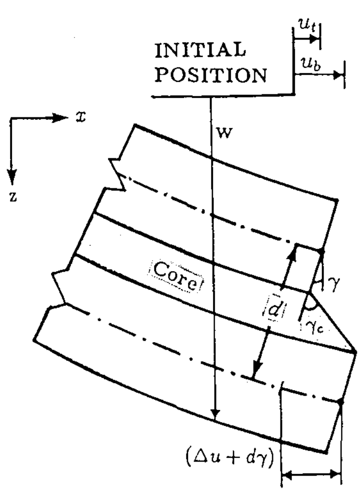

Fig. 2 Core shear deformation.

By applying compatibility condition at the face-core interface and eliminating the longitudinal displacement coordinates in terms of single transverse coordinate, the equation of motion for the sandwich system is obtained as

$$w^{VI} - S_1^*(F+1)w^{IV} + \frac{m}{K}(w^{II}\ddot{w} - S_1^*\ddot{w}) = \frac{1}{K}(q^{II} - S_1^* q) \quad (2)$$

As stated earlier, for better evaluation of system damping mainly from core shear deformation mechanism, it is better to decompose the system as spring-mass system (steel part) and damper (core part).

The solution of Eq. 2 for system modal loss factor and modal frequency yields

$$\omega_n = \left[\frac{K\alpha_n^4[1+S_2+(F+1)[S_2+S_2^2(1+\eta_c^2)]]}{m[1+2S_2+S_2^2(1+\eta_c^2)]}\right]^{1/2} \quad (3)$$

and

$$\eta_n = \frac{S_2\eta_c F}{1+S_2+(F+1)[S_2+S_2^2(1+\eta_c^2)]} \quad (4)$$

Table-1 Beam data

length	0.600	m
width	0.030	m
E steel	2.06×10^{11}	N/m^2
ρ steel	7.80×10^3	kg/m^3
G core	1.6×10^6	N/m^2
ρ core	1.50×10^3	kg/m^3
η core	0.248	
t steel	0.0045, 0.010	m
t core	75, 150, 300	μm

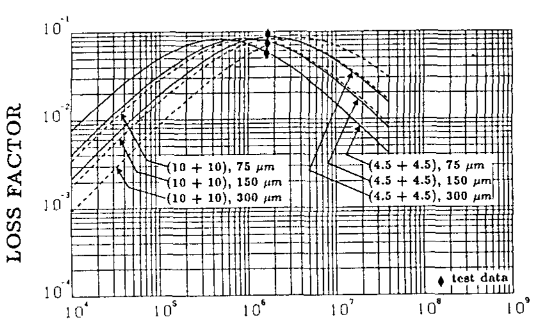

Fig. 3 G (core) VS. LOSS FACTOR

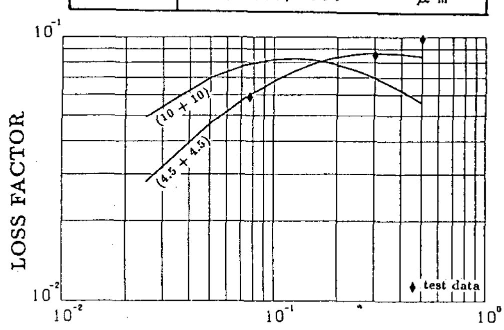

Fig. 4 T (core) VS. LOSS FACTOR

4. NUMERICAL EXAMPLE

For the purpose of parametric study a variation in parameters is considered as shown in Table.1. Results are presented for system energy loss factor with respect to changes in various parameters.

5. RESULTS AND THE DISCUSSION

It is evident that energy loss offered by sandwich system largely depends on core parameters (Fig. 3 and Fig. 4) No single parameter can be considered as decisive. Analytical work is still continued and more results shall be discussed at the time of presentation.

REFERENCES

1. Y.P. Lu, B. E. Douglas 1974 Journal of Sound and Vibration 32(4),517-521.
2. J. A. Moore, R.H. Lyon J.Acoust.Soc.Am.89(2),Feb.1991,Sound transmission loss characteristics of sandwich panel constructions.