

materials is clear if they are considered to be assemblages of constituent substructures

$$\zeta = \sum_i \zeta_i U_i^*$$

where ζ is the effective composite viscous damping ratio for a particular mode of vibration; ζ_i the damping ratio for constituent material i , generally a function of frequency and temperature; and U_i^* the fraction of the strain energy of deformation (in the particular mode shape) found in material i .

For illustrative purposes, consider an undamped composite rod with isotropic constituents executing extensional vibration in its fundamental mode (in vacuum and free fall, if desired). At the instant of maximum amplitude, the energy of vibration is found entirely as strain energy. If shear and end effects are neglected, we find uniform extensional strain and nonuniform stress at any cross section. The strain energy density can then be approximated as follows:

$$U = \frac{1}{2} E \epsilon^2 = \frac{1}{2} (E_f v_f + E_m v_m) \epsilon^2$$

Now, applying the "modal strain energy rule of mixtures" damping theory, we find that

$$U_f^* = \frac{E_f v_f}{E}, \quad U_m^* = \frac{E_m v_m}{E}$$

$$\zeta = \frac{\zeta_f E_f v_f + \zeta_m E_m v_m}{E}$$

Notice that this result is identical to Eq. (13). A similar derivation for a composite beam in flexure yields Eq. (9). A more general form of this theory would account for the fact that the total strain energy is the sum of that associated with different types of deformation, as follows:

$$\zeta = \sum_i \sum_j \zeta_{ij} U_{ij}^*$$

where the subscript j indicates different types of deformation (e.g., dilation or distortion of an isotropic material).

The description given in Ref. 1 indicates some similarity of this approach to that of Ni and Adams,¹⁰ but is applicable to composites of arbitrary composition and geometry.

Note that the preceding theories are only appropriate for use when applied to damping mechanisms with characteristic geometric scales much smaller than the constituent or specimen dimensions. (Thermoelastic damping due to transverse thermal currents, for example, does not meet this restriction.) In addition, material damping values should generally be treated as functions of temperature and frequency.

This author trusts that research activity in the general area of damping in advanced space structures will flourish and that the line of research related in Refs. 1 and 11 will continue to lead to new insights.

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Errata

Shuttle Entry Data System Preflight Test and Analysis

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TABLES 1 and 2 were inadvertently omitted and are printed below.

Table 1 SEADS columbium port test summary

Test temperature (°C)			
Target	Actual	Cycles	Remarks
1260	1259	6	No observable port/coating damage
1430	1410	6	No observable port/coating damage
1540	1543	3	Mid-coated surface polishing; coating not breached

Table 2 Shuttle nose cap flight heating environments

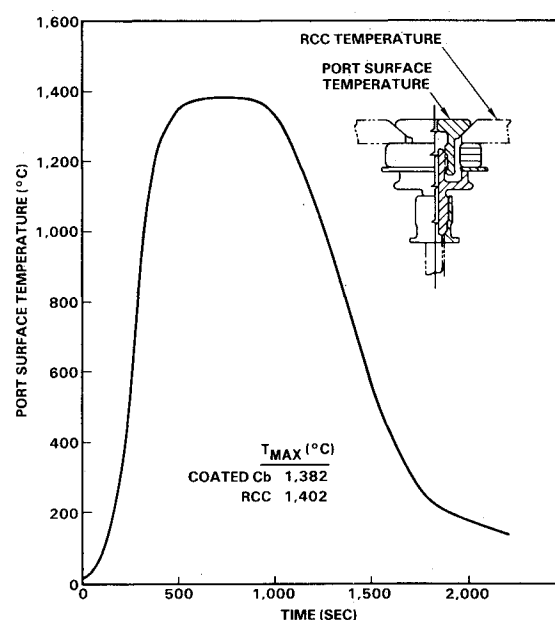
Mission	Vehicle	Max \dot{q} (W/cm ²)	Heat load (Q), (cal/cm ²)	RCC T_{max} (°C)
STS-2	Columbia	44.3	8990	1410
STS-3	Columbia	43.8	8195	1400
STS-5	Columbia	43.1	8135	1390
STS-6	Challenger	38.7	7355	1360
STS-7	Challenger	40.9	7080	1380
STS-9	Columbia	49.6	9560	1440
Design (14414.1C)	—	46.1	9000	1430

$Q = \int \dot{q} d\theta$

The footnote in Table 3 should have read: ^aPyrometer readings were erratic and were discarded. ^b Surface thermocouple 1.

In Fig. 18 the TMAX table was not included. The complete Figure is shown at right.

In Reference 13 the Volume number for the *Journal of Aerospace Sciences* is 29.

**Fig. 18 Columbia port predicted temperature for STS-9.**

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