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

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

where $\zeta$ is the effective composite viscous damping ratio for a particular mode of vibration; $\zeta_i$ 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:

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

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

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

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

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: [a]Pyrometer readings were erradic 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.

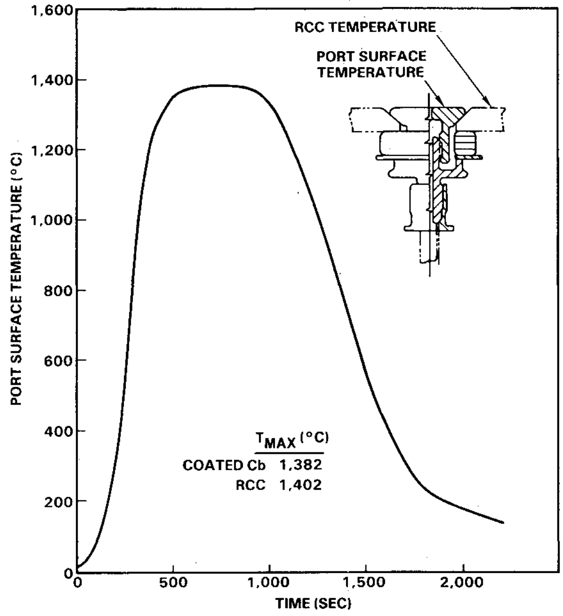


Fig. 18 Columbium port predicted temperature for STS-9.