

should be observable, all three cannot be observed simultaneously in any convenient scattering geometry. In the Z(XZ)Y geometry of Fig. 4, a spectral line is observed which weakens and moves to higher energy with increasing stress. It corresponds to the $1s(A_1) \rightarrow 1s(B_2)$ transition of Fig. 5. In the Z(YZ)Y geometry of Fig. 4, a spectral line which moves to lower fixed energy with increasing stress is observed and corresponds to the $1s(A_1) \rightarrow 1s(B_1)$ transition of Fig. 5.

Finally, an important characteristic of any uniaxial stress apparatus is the stress homogeneity. In this regard, it can be seen from Fig. 4 that the valley-orbit Raman line, though shifted, is not significantly broadened under the application of the applied stress. This lack of broadening is conclusive evidence of a high degree of stress homogeneity over the focal region (150μ) of the laser beam. For samples with a 1.25 cm length between the cups, the same high degree of homogeneity is observed over regions at least 0.25 mm^2 .

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APPENDIX

The pushtube (L) (Fig. 1) must be stable against buckling under the maximum applied compressive force. To determine the threshold for buckling, we treat the pushtube as a hollow cylindrical beam of length L fixed at one end as shown in Fig. 6. At the other end, a force W is directed through a point M a distance h below the surface to which the beam is fixed. With reference to the stress apparatus described in this paper, L is the equilibrium length of the pushtube, W is the compressive force and h is the distance from the bottom of the stem (O) (Fig. 1) to the top plate of the bellows section.

Let $y(x)$ be the displacement at height x as shown in Fig. 6. The torque of W about point P is $\tau = \mathbf{r} \times \mathbf{W} = -rW\hat{k}$ where \hat{k} is a unit vector directed normal to and out of the plane of Fig. 6. For small displacements

$$s - y \approx r \quad (1)$$

and

$$\tan\theta \approx \theta = \frac{e}{h+L} = \frac{S}{x+h}. \quad (2)$$

The torque τ on the beam is then

$$\tau = -\left[\frac{e(x+h)}{L+h} - y \right] W\hat{k}. \quad (3)$$

For stability

$$|\tau| \leq |\tau_{\max}| = |N|, \quad (4)$$

where

$$N = \frac{YI}{R}\hat{k} \quad (5)$$

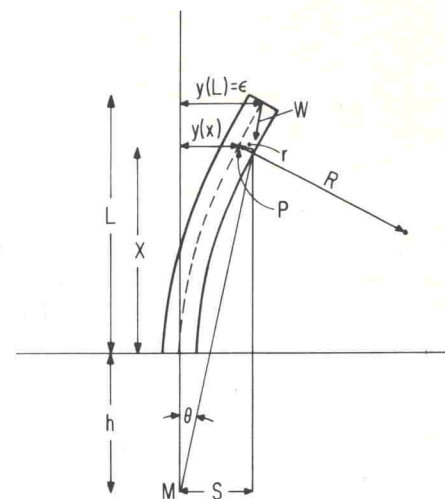


FIG. 6. A diagram indicating the relevant parameters of the buckling beam problem.

is the restoring torque supplied by the beam.^{21,22} Here Y is the Young's modulus for the material from which the beam is constructed, I is the moment of inertia per unit mass of the beam about a diameter of its circular cross section and R is the radius of curvature of the beam at point P . For small displacements, the radius of curvature R is given by

$$\frac{1}{R} = \frac{d^2y}{dx^2}. \quad (6)$$

Combining Eqs. (3)–(5) with Eq. (6), we obtain the following differential equation for y :

$$\left[\frac{YI}{W_{\max}} \right] \frac{d^2y}{dx^2} + y - \frac{e(x+h)}{L+h} = 0, \quad (7)$$

the general solution of which is

$$y = A \sin \left[x \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} \right] + B \cos \left[x \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} \right] + \frac{e(x+h)}{L+h}. \quad (8)$$

Upon applying the appropriate boundary conditions

$$y(0) = y'(0) = 0, \quad y(L) = \epsilon,$$

we obtain

$$1 = \frac{-1}{L+h} \left\{ \frac{YI}{W_{\max}} \sin L \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} + h \left[\cos L \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} - 1 \right] - L \right\}, \quad (9)$$

which reduces to

$$\tan \left[L \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} \right] = -\frac{h}{L} \left[L \left(\frac{W_{\max}}{YI} \right)^{\frac{1}{2}} \right]. \quad (10)$$

The values of h and L appropriate to the stress apparatus described in this paper are $h = 35.56 \text{ cm}$ and $L = 91.44 \text{ cm}$. The first solution to the above transcendental equation other than $W_{\max} = 0$ is

$$W_{\max} = (2.391)^2 \frac{YI}{L^2}.$$

It is to be noted that, as $h \rightarrow \infty$, the solution of Eq. (10) tends towards the Euler formula for the buckling of a column with one end fixed and the other end free.^{21,22} Now for a hollow tube

$$I = \frac{\pi}{4}(R_2^4 - R_1^4), \quad (11)$$

where R_2 and R_1 are, respectively, the inner and outer radii. Using $R_1 = 0.635$ cm, $R_2 = 1.016$ cm and $Y = 2.0 \times 10^{12}$ dyne/cm²,²³ we find

$$W_{\max} = 9.72 \times 10^8 \text{ dynes.}$$

The pushtube used in the stress apparatus described herein can therefore withstand an 8.92×10^8 dynes compressive force without buckling.

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