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 (150 μ) 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 $\boldsymbol{\tau}=\mathbf{r}\times\mathbf{W}=-rW\hat{k}$ where $\hat{k}$ is a unit vector directed normal to and out of the plane of Fig. 6. For small displacements

$$s-y\approx r \tag{1}$$

and

$$\tan\theta\approx\theta=\frac{\epsilon}{h+L}=\frac{S}{x+h}. \tag{2}$$

The torque $\boldsymbol{\tau}$ on the beam is then

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

For stability

$$|\boldsymbol{\tau}|\leqslant|\boldsymbol{\tau}_{\max}|=|\mathbf{N}|, \tag{4}$$

where

$$\mathbf{N}=\frac{YI}{R}\hat{k} \tag{5}$$

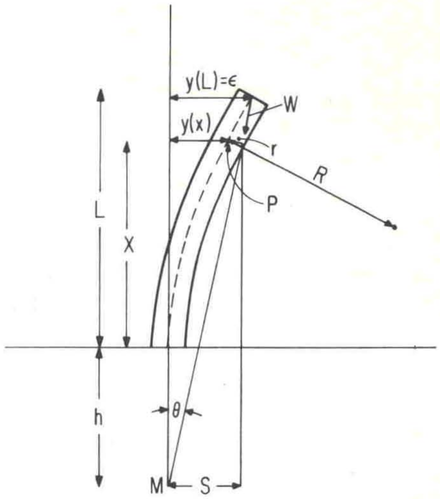

FIG. 6. A diagram indicating the relevant parameters of the buckling beam problem.

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

$$\frac{1}{R}=\frac{d^2y}{dx^2}. \tag{6}$$

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

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

the general solution of which is

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

Upon applying the appropriate boundary conditions

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

we obtain

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

which reduces to

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

The values of h and L appropriate to the stress apparatus described in this paper are $h=35.56$ cm and $L=91.44$ 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

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

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 $Y=2.0\times10^{12}$ dyne/cm^2,[23] we find

$$W_{\max}=9.72\times10^8 \text{ dynes}.$$

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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