PRESSURE AND FREQUENCY EFFECTS ON THE DAMPING OF A CANTILEVER BEAM IN A MAGNETIC FIELD

by

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NOMENCLATURE

S	stress, pounds per square inch
b	distance from neutral axis to outer fiber, inches
I	section moment of inertia, inches to the fourth power
P	bending load, pounds
L	length, inches
m	mass, slugs
c	coefficient of viscous damping, pound-seconds per inch
k	spring constant, pounds per inch
x	amplitude of vibration, inches
t	time, seconds
ω	damped natural frequency, radians per second
n	number of cycles
δ	logarithmic decrement, dimensionless
Ln	natural logarithm
A_o	initial amplitude, millimeters
A_n	amplitude after n cycles, millimeters
A	amplification factor, dimensionless
x_{res}	amplitude of vibration at resonance, inches
x_{st}	amplitude at zero frequency, inches
ζ	damping factor, dimensionless
C_c	critical dashpot constant, pounds-seconds per inch
C_{eff}	equivalent dashpot constant, pounds-seconds per inch

F	force, pounds
Q	quality factor, dimensionless
W	stored energy, inch-pounds
D	specific damping energy, inch-pounds per cubic inch per cycle
E'	elastic modulus, pounds per square inch
E''	dissipative elastic modulus, pounds per square inch
f_o	undamped natural frequency, radians per second

INTRODUCTION

The amplitude of a free vibrating system will diminish in time and the rate at which this motion is depleted is the damping. This damping is a result of the energy dissipated by the material and other effects while under cyclic stress. The many parameters affecting damping have not been described analytically because of no known relationships among them. All work done in this area has been in isolating a few of the parameters involved and obtaining trends in the damping capacity of materials.

The engineering significance of the damping properties of uniform solid materials has been studied for almost two hundred years. In 1784 Coulomb [1]* speculated on the microstructural mechanisms of damping and performed experiments showing that damping under torsional oscillations was caused by internal losses in the material.

The study of damping in solids expanded rapidly because of its engineering applications. It was soon realized that the optimization of total damping with a structural system provides a useful concept for controlling resonant frequencies. Lazan [2] reports that the basic approach used thus far is to determine the unit properties of materials by testing simple specimens under simplified conditions and the idealization of the properties or the formulation of equations. Then an analysis can be made of local stress, strain, temperature, and so on, in a nonhomogeneously loaded part or configuration to determine both localized and gross behavior.

* Numbers in brackets designate references at the end of this report.

Lazan [3] also reports that material damping displays a variety of behavioral patterns. In general, damping is highly nonlinear and not representable by a viscous damping model. However, if generalizations are made, it is important to define clearly the scope of the testing conditions.

The successful engineering utilization of damping as an important design property is dependent on the success of the damping research being conducted. Before this can be accomplished, the proper interpretation and effective utilization of damping data in engineering situations requires more general theories and computational procedures than are presently available.

DISCUSSION OF PROBLEM CONSIDERED

The purpose of this report is to isolate the effects of pressure and frequency on the damping of a cantilever beam. This was done by holding constant all but one of the variables that change the damping capacity of a vibrating member.

The large number of parameters that change the damping of a vibrating system are apparently unrelated. It is generally known how these parameters individually affect the damping, but a correlation among them remains unknown. Later in this chapter, each parameter is discussed separately.

It is necessary to define damping in an analytical manner. The usual approach is based on the decay rate of a vibrating system. The method used in this experiment for defining damping is the logarithmic decrement. A description of this method and others are presented later in this chapter.

Factors Affecting Damping

The major parameters that affect damping are listed below with their known trends.

1. Stress Amplitude: The most important variable affecting internal friction in solids is the stress or strain amplitude of the cyclic action. Rowett [4] as well as others have noted in their experiments that the damping capacity varied as the third power of the stress amplitude.

2. Stress Distribution: Damping change due to different stress

distributions in a vibrating system has also been noted. For this reason, most experiments have been conducted on thin walled tubes to give a uniform stress throughout the specimen when it is subjected to a torsional stress.

3. Temperature: The internal damping increases as the temperature is increased in most materials. Hatfield, Stanfield, and Rotherham [5] showed that the specific damping capacity increased rapidly with increase in temperature.

4. Stress History: The dependence of damping on prior stress history has been observed. Most materials show an increase in damping under cyclic stress, but exceptions have shown the opposite effect. In practically all structural materials, however, the stress history effect is very small until the stress approaches or exceeds the fatigue strength of the material and then the change in damping with number of fatigue cycles may become pronounced.

5. Magnetostrictive Damping: Anderson [6] showed that if a ferromagnetic material is placed in a magnetic field, the internal damping will decrease until the specimen has been saturated. Some reports indicate that this decrease in damping can be as large as half of the total internal damping in certain materials. Specimens have also been placed in an alternating magnetic field and this also decreased the internal damping.

6. Pressure and Frequency: The effects of pressure and frequency on the damping of a vibrating system are the parameters discussed in this experiment. The development and discussion of their effects are presented in later chapters.

Methods of Defining Damping

Almost all descriptions of damping are derived from the linear single degree of freedom system with a viscous damper in parallel with the spring. Plunkett [3] gives the six major descriptions used and the relationships among them.

1. Logarithmic Decrement: This method of defining damping is the one most commonly used. It is based on the concept of energy dissipated per cycle of vibration. For a system that is vibrating in a single mode shape, the frequency is a determined quantity and the energy loss per cycle is a function only of amplitude or stress level. The equation used is

$$\delta = \frac{1}{n} \ln \frac{A_o}{A_n} \quad \text{(See Appendix A).} \tag{1}$$

2. Amplification Factor: If a constant sinusoidal excitation force is applied with gradually increasing frequency, it is found that the amplitude of vibration steadily increases to a maximum and then decreases as the frequency is further increases. There is one value of frequency, near where the amplitude is a maximum, at which the applied force is exactly in phase with the vibrating velocity. The applied force is then completely dissipated in damping at the resulting amplitude and this amplitude is a measure of damping. The ratio between the vibrational amplitude at resonance and that at zero frequency is the amplification factor. The equation used is

$$\frac{x_{res}}{x_{st}} = A\,. \tag{2}$$

3. Equivalent Dashpot Constant: The damping factor is expressed in terms of critical damping. The equation used is $\zeta = \dfrac{C_{eff}}{C_c}$, (3)

where $C_{eff} = \frac{F}{\omega x_{res}}$ and $C_c = 2\sqrt{km}$.

4. Quality Factor: If the amplitude is constant, the sum of the kinetic and potential energies is almost constant, and the energy stored may be measured by the maximum value of either one. The quality factor is defined in terms of the ratio of the energy dissipated to the energy stored. The equation is $Q = \frac{2\pi W}{\Delta W}$, (4)

where ΔW is the energy supplied to the system per cycle by the external force.

5. Complex Modulus: The complex spring constant is defined in terms of the steady state response to forced vibration. The real part is that portion of the spring force in phase with the displacement divided by the resulting displacement, and the imaginary part is that part in quadrature divided by the displacement. The specific damping energy is

$$D = \frac{2\pi E'' W}{E'} \quad . \tag{5}$$

6. Bandwith: This method is based on the difference in the two frequencies at which the amplitude is the same if the exciting force is the same. The equation is $2\zeta = \frac{\Delta f}{f_o}$, (6)

where Δf is the difference between the two frequencies at which the amplitude is x_o .

The relationships among the various definitions of damping may be established only for a linear single degree of freedom system. The comparison is $\zeta = \frac{1}{2Q} = \frac{\Delta f}{2f_o} = \frac{\delta}{2\pi} = \frac{E''}{2E'} = \frac{1}{2A}$. (7)

Other methods of expressing damping which are less used are:

1. A frequency phase method [7]
2. A loss factor [8]
3. Bluntness of resonance curve [8]
4. Specific damping capacity [8]

PARAMETERS STUDIED

Effects of Pressure on Damping

The effect of air pressure on the damping of a vibrating system has been known for some time. The vibrating cantilever beam displaces air as it is cycled. Some air flow around the beam is produced and both a viscous drag and pressure drag is produced by the vibrating system.

In a report published by Baker and Allen [9] it was found that the pressure drag air damping is proportional to amplitude and that viscous drag air damping is independent of amplitude. Both the analytical solution and experimental results show that damping is a linear function of air pressure.

As the air pressure is decreased, the damping also decreased until a level is reached where air has no more effect on the damping. McWithey and Hayduk [10] showed that the damping caused by air pressure drag and viscous air drag may become a significant portion of the damping present at atmospheric pressure. These contributions to damping become negligible below a pressure of 0.3 inches of mercury.

The two reports listed above are in agreement on the effect of pressure on damping of cantilever beams. However, a disagreement arises on the pressure at which damping has no effect on a vibrating system. Since both reports used different sized specimen for their experiments, they indicate that the damping due to pressure is dependent on the size or effective flat plate area of the specimen. Both reports do agree that a pressure was reached where no effect on damping is observed. Baker and Allen [9] derived a relationship where the effect of air pressure can be predicted from the size of the specimen. McWithey and Hayduk [10] also noted that once a pressure is reached so the effect of air on damping is negligible, the vacuum environment has no significant effect on the damping characteristics of a vibrating beam.

Effects of Frequency on Damping

The effect of frequency on the damping of a vibrating beam is not entirely established. A disagreement of the frequency effect is apparent in the literature surveyed.

Kimball and Lovell [11] state that over a considerable frequency range and stress amplitude for a number of solids of different physical properties, the frictional loss per cycle of stress at a point in the solid is independent of the frequency. Föppl [12] also found that there is no frequency effect except at very low frequencies for materials subject to creep when the frequency is so low that the strain velocity is of the same order of magnitude as the creep velocity. A report by Robertson and Yorgiadis [13] also shows that damping is independent of frequency.

The above experimenters performed their tests using tubes in torsion. This provided a constant stress level throughout the specimen. Ockleston [14] and other investigators found a slight frequency effect on damping when vibrating a beam. The vibrating beam does not have a constant stress level throughout the specimen and therefore the measurements recorded at different frequencies were not necessarily at the same stress amplitude.

EXPERIMENTAL PROCEDURE

The two test specimens chosen for this experiment were a 3/8 inch by 2 inch mild steel bar, hereafter referred to as the "large beam", and a 3/16 inch by 3/4 inch bar hereafter referred to as the "small beam". Both the large and small beams were used for the pressure effects tests. Only the small beam was used for the frequency tests.

Both pressure and frequency tests specimens were mounted securely to a support inside a vacuum tank. The beams used during the pressure tests were mounted in a 8 inch by 4 inch support hereafter referred to as the "small support". The small beam used for frequency tests was mounted in a 23 inch by 4 inch support hereafter referred to as the "large support". An electric coil was also mounted inside the vacuum tank to provide a magnetic field.

A strain gage was mounted on the beam next to the support before each test. The gage output was recorded directly on a Sanborn Recorder. See Appendix B for the complete list of equipment used.

The tests conducted to find the effect of pressure on damping were done on both the small beam and the large beam. The experimental procedure that follows is for both beams unless otherwise noted.

Both the small beam and large beam were tested at a constant stress level. The large beam tests were conducted at a stress level of 3000 pounds per square inch. Due to the limited amount of deflection possible in the vacuum tank, the small beam was tested at a stress level of 1000 pounds per square inch.

The recorder and strain gage amplifier were first balanced. The strain gage was then calibrated with the recorder to read stress as a function of recorder stylus displacement (See Appendix C). Preliminary tests were then conducted to determine the saturation magnetization current needed to insure that sufficient field strength existed to align the domains of the two specimens (See Appendix D). Before each test, the recorder and amplifier were rebalanced and the calibration procedure was repeated.

The vacuum tank was then evacuated to a pressure of 700 microns. The beams were then vibrated and the output of the strain gages was recorded. The pressure in the vacuum tank was increased and the above procedure repeated. A minimum of two recordings of the vibration was taken at each pressure. At pressures where the data points did not conform to the trend established, the tests were repeated and better results were obtained. All tests were conducted at room temperature, approximately 70 degrees F.

The increments of pressure increase in the vacuum tank varied from 100 microns in the lower pressure range to 1 inch of mercury in the higher pressure range. All tests were conducted with the specimen in a saturated magnetic field.

The tests conducted to find the effect of frequency on damping were performed on the small beam in the large support. The apparatus was balanced, calibrated, and tests were run to determine required saturation magnetization current.

The beam was then placed in the large support, vibrated, and the results recorded. The length of the large support was then shortened to change the frequency of the beam. The above procedure was then repeated. All tests were conducted at room temperature and at a stress level of 1000 pounds per square inch. The pressure in the tank during these tests was no greater than 1000 microns. Preliminary tests were also run to insure that damping due to air was not present. A minimum of 8 tests were conducted at each frequency.

RESULTS AND DISCUSSION

The test results show conclusively that at a stress level of 3000 or 1000 pounds per square inch, pressure does have an effect on the damping of the two test specimen. Both the large beam and the small beam show a decrease in damping as the pressure is reduced. The damping, measured by the logarithmic decrement, decreases in a linear manner as the pressure is reduced.

Several differences between the plot of pressure versus logarithmic decrement for the large beam (Figure 1), and the plot of pressure versus logarithmic decrement for the small beam (Figure 2) are as follows:

1. The straight line portion of the graph for the large beam ends around 10 pounds per square inch pressure while the straight line ends around 4 pounds per square inch for the small beam. These results indicate that the damping due to pressure is dependent on the shape and size of the beam.

2. The decrease in damping of the small beam is greater than that of the large beam over the pressure range. This was observed because the small beam was excited at a larger amplitude to obtain the 1000 pounds per square inch stress level.

Below a pressure of 10 pounds per square inch for the large beam, and 4 pounds per square inch for the small beam, the damping approaches a constant value. This result is anticipated since the density of the air media is becoming very small. The effect of damping due to air pressure is not present in either test specimen below a pressure of 0.1 inch of mercury.

Figure 3 shows a plot of pressure in the low range versus logarithmic decrement for the small beam. For the large beam, a similar curve can be drawn.

In all cases in this phase of the experiment, the logarithmic decrement was used to measure damping, with n being equal to 50.

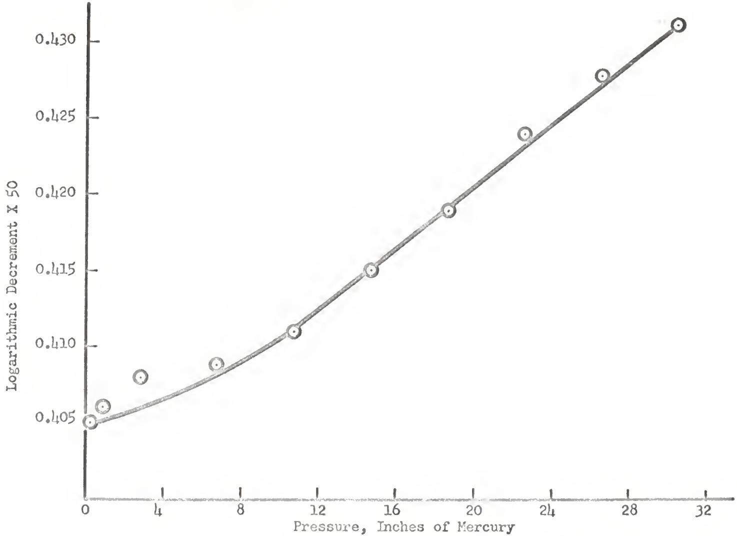

Figure 1 Pressure versus Logarithmic Decrement, Large Beam in Small Support.

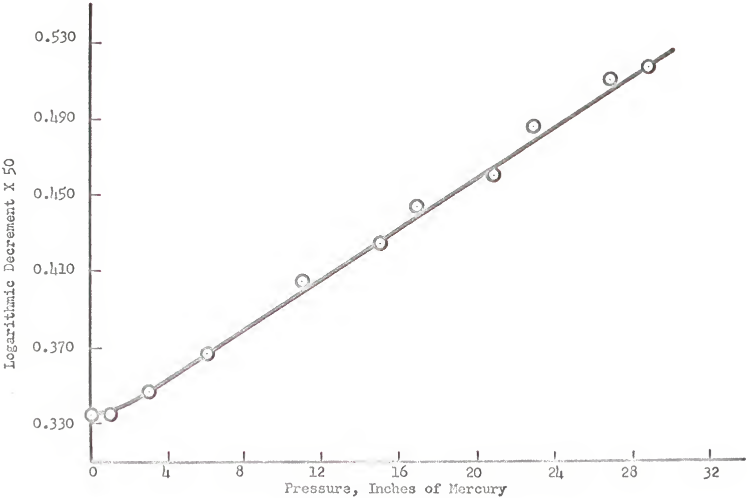

Figure 2 Pressure versus Logarithmic Decrement, Small Beam in Small Support.

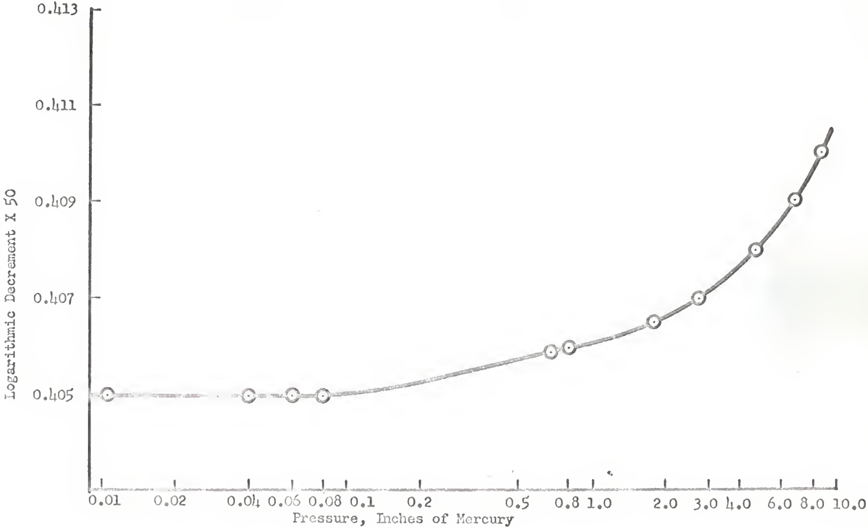

Figure 3 Pressure versus Logarithmic Decrement,
Small Beam in Small Support.

The results of the damping effect due to frequency tests show that for the vibrating small beam at a stress level of 1000 pounds per square inch, a change in beam frequency does change the internal damping of the beam. This result is shown in Figure 4.

In this phase of the experiment, the logarithmic decrement was used to measure internal damping. The value of n was adjusted so that the stress level was 1000 pounds per square inch at A_o and 700 pounds per square inch at A_n. The gain was adjusted on the Sanborn Recorder so that the same indicated amplitude corresponded to the stress level desired in each case.

The logarithmic decrement was calculated using the formula

$$\delta = \frac{1}{n} \ln \frac{A_o}{A_n} \qquad \text{(See Appendix A)} \qquad (1)$$

in all tests.

The highest fundamental frequency obtained during these tests was 15.65 cycles per second, which was well within the frequency response range of the recording instrument.

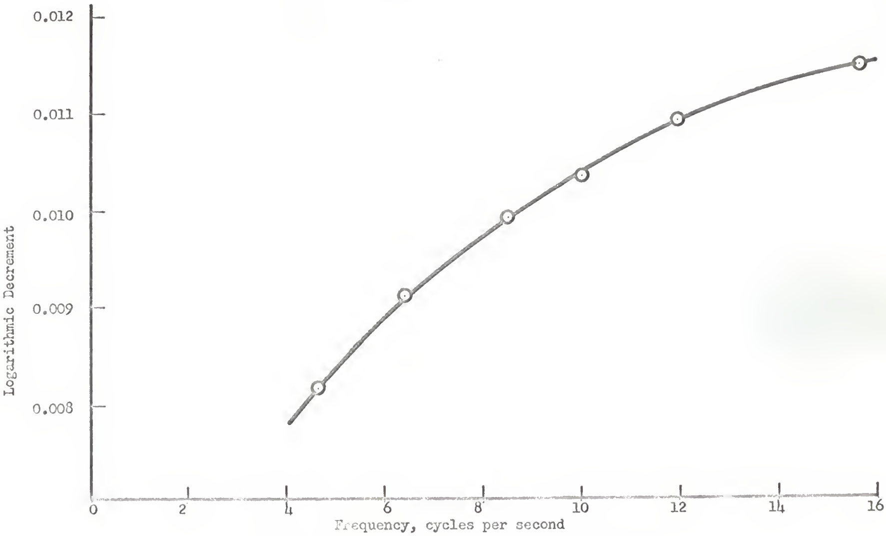

Figure 4 Frequency versus Logarithmic Decrement, Small Beam in Large Support.

CONCLUSION

This experiment demonstrates that air pressure has an effect on the damping of a cantilever beam. The damping decreases in a linear manner as the pressure is decreased. In both specimen tested, the air pressure ceases to have an effect on damping below 0.1 inch of mercury.

It is also apparent from this experiment that frequency has an effect on the internal damping of a cantilever beam. The damping of the material tested shows a decrease as the frequency is decreased. These tests are not conclusive because the vibrating cantilever beam does not have a constant stress distribution. A damping effect due to the nature of the stress may be present.

Much work needs to be performed on finding the true damping characteristics of different materials. This is a definite prerequisite before a correlation can be made among all the causes of damping in materials.

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APPENDIX

APPENDIX A

DERIVATION OF LOGARITHMIC DECREMENT

Friction may be presented in various forms in mechanical systems. The quantitative description of friction in a system is much more difficult than that of the elastic or the inertial properties of the system. Therefore the treatment of friction or damping in vibration is approached from the point of view of convenience. Introduced largely for this reason, the term "viscous damping" specifies that the damping force is proportional to the velocity of the motion. This type of damping occurs when the resisting force is due to viscous resistance in a fluid medium such as the fluid friction in an ideal dashpot.

The general equation for a free damped vibration of a simple spring mass system is

$$m\ddot{x} + c\dot{x} + kx = 0 \quad . \quad [15] \tag{8}$$

The standard solution of a differential equation of this type is to assume a solution of the form

$$x = C_1 \, e^{r_1 t} + C_2 \, e^{r_2 t} \quad , \tag{9}$$

where r_1 and r_2 are the two roots of the auxiliary equation

$$mr^2 + cr + k = 0 \quad , \quad [16] \tag{10}$$

and C_1 and C_2 are arbitrary constants. The solution of equation (10) gives the two roots as

$$r_{1,2} = \frac{-c}{2m} \pm \frac{1}{2}\sqrt{\left(\frac{c}{m}\right)^2 - \frac{4k}{m}} \quad . \tag{11}$$

Equation (11) simplifies to

$$r_{1,2} = \frac{-c}{2m} \pm \frac{1}{2m}\sqrt{c^2 - 4km} \quad . \tag{12}$$

If the algebraic sign of $\sqrt{c^2 - 4km}$ is positive, then the motion given by equation (9) represents a gradual subsidence since the exponents are both negative and decreasing with increasing time. Therefore, the displacement x(t) approaches the equilibrium position asymptotically.

If $\sqrt{c^2 - 4km} = 0$, then the system is critically damped and the displacement again approaches the equilibrium position asymptotically.

If $\sqrt{c^2 - 4km}$ is negative, the roots r_1 and r_2 are a pair of complex conjugates with negative real parts. The roots are

$$r_{1,2} = \frac{-c}{2m} \pm \frac{i}{2m}\sqrt{4km - c^2} \quad , \tag{13}$$

where $i = \sqrt{-1}$.

If we now define the damped natural frequency ω to be $\sqrt{4km - c^2}$, then the following simplifications are

$$x = C_1 e^{-\frac{c}{2m}t} e^{i\frac{\omega}{2m}t} + C_2 e^{-\frac{c}{2m}t} e^{-i\frac{\omega}{2m}t} \quad , \tag{14}$$

$$x = e^{-\frac{c}{2m}t}\left[C_1(\cos\frac{\omega}{2m}t + i\sin\frac{\omega}{2m}t) + C_2(\cos\frac{\omega}{2m}t - i\sin\frac{\omega}{2m}t)\right] , \tag{15}$$

and

$$x = e^{-\frac{c}{2m}t}\left[(C_1 + C_2)\cos\frac{\omega}{2m}t + i(C_1 - C_2)\sin\frac{\omega}{2m}t\right] , \tag{16}$$

or

$$x = e^{-\frac{c}{2m}t}\left[A\cos \frac{\omega}{2m}t + Bi \sin \frac{\omega}{2m}t\right] \quad , \tag{17}$$

where $A = C_1 + C_2$ and $B = C_1 - C_2$. (18)

We now consider the amplitude at some time t and at a later time $t + T$, where T is the period of vibration and is

$$T = \frac{2\pi}{\omega} \quad . \tag{19}$$

From equation (17), the result of the ratio is

$$\frac{x(t)}{x(t + nT)} = \frac{e^{-\frac{c}{2m}t}}{e^{-\frac{c}{2m}(t + nT)}} \quad , \tag{20}$$

where n is the number of cycles. Equation (20) is true because

$$\cos \frac{\omega}{2m}t = \cos \frac{\omega}{2m}(t + nT) \quad , \tag{21}$$

and

$$\sin \frac{\omega}{2m}t = \sin \frac{\omega}{2m}(t + nT) \quad . \tag{22}$$

The ratio then becomes

$$\frac{x(t)}{x(t + nT)} = e^{\left[-\frac{c}{2m}t + \frac{c}{2m}t + \frac{c\pi n}{\omega m}\right]} \quad , \tag{23}$$

or

$$\frac{x(t)}{x(t + nT)} = e^{\frac{c\pi n}{\omega m}} \quad . \tag{24}$$

But

$$\frac{c\pi n}{\omega m} = \mathrm{Ln}\left(\frac{x(t)}{x(t + nT)}\right) \quad , \tag{25}$$

or

$$\frac{c\pi}{\omega m} = \frac{1}{n} \mathrm{Ln}\left(\frac{x(t)}{x(t + nT)}\right) \quad . \tag{26}$$

The logarithmic decrement, δ , is defined as

$$\delta = \frac{1}{n} \mathrm{Ln}\left(\frac{x(t)}{x(t + nT)}\right) \quad , \tag{1}$$

or

$$\delta = \mathrm{Ln}\ e^{\frac{c\pi}{\omega m}} = \frac{c\pi}{\omega m} \quad . \tag{27}$$

APPENDIX B

EQUIPMENT LIST

The following is a list of equipment and measuring instruments used for this experiment. An explanation of the experimental equipment is given to clarify the figures. The numbers refer to the notation on Figures 5, 6, and 7.

1. Vacuum Tank: A 12 inch inside diameter pipe 4 feet long. Both ends are sealed by an O-ring and a 15 inch square 3/4 inch thick glass plate.

2. DC Power Supply: Consolidated Electrodynamics Corporation, Type 3-131, variable range up to 4.5 amperes at 31 volts, M.E. no. 1664. This power supply was used in the frequency tests.

3. DC Power Supply: Electronic Instrument Company, model no. 1064, serial no. 6812, M.E. no. 3555. This power supply was used when testing with the small support only.

4. Sanborn Recorder: Model no. 127, serial no. 1352, with Sanborn Strain Gage Amplifier, model no. 140B, serial no. 74, M.E. no. 1067.

5. DC Ammeter: Weston model 901, no. 106, M.E. no. 1348, 0 amp. to 10 amp. full scale.

6. Actuator: A 3/16 inch diameter brass welding rod with an O-ring seal, hand operated. The actuator excites the beam by a displacement and release method.

7. Mercury Manometer: King Manometer, model BUS 36, King Engineering Corporation, M.E. no. 3663.

8. Vacuum Pump: Disto-Pump, model no. 1399, serial no. 3539, Welch Scientific Company.

9. Pirani Vacuum Gage: Type GP-110, Consolidated Electrodynamics Rochester Division, M.E. no. 1714. The Pirani Gage was calibrated against the Mechanical Engineering McLeod Gage.

10. Large Support: Two 3/4 inch by 4 inch steel plates 23 inches long. A 1/4 inch by 2 inch steel web was welded on the top of the support to promote rigidity.

11. Small Beam: A 3/16 inch by 3/4 inch hot rolled steel bar 43 inches long used for both the pressure and frequency tests.

12. Large Beam: A 3/8 inch by 2 inch hot rolled steel bar 43 inches long used for the pressure tests.

13. Coil: A 10 inch inside diameter cylinder, 36 inches long, wound with 1400 turns of wire.

14. Calibration Weights: 1/5 pound to 4 pounds.

15. Small Support: Two 3/4 inch by 4 inch steel plates 8 inches long.

The strain gages used were Budd Metalfilm Strain Gages, type C6-121, 1/8 inch gage length. The gages were bonded with Eastman 910 Adhesive.

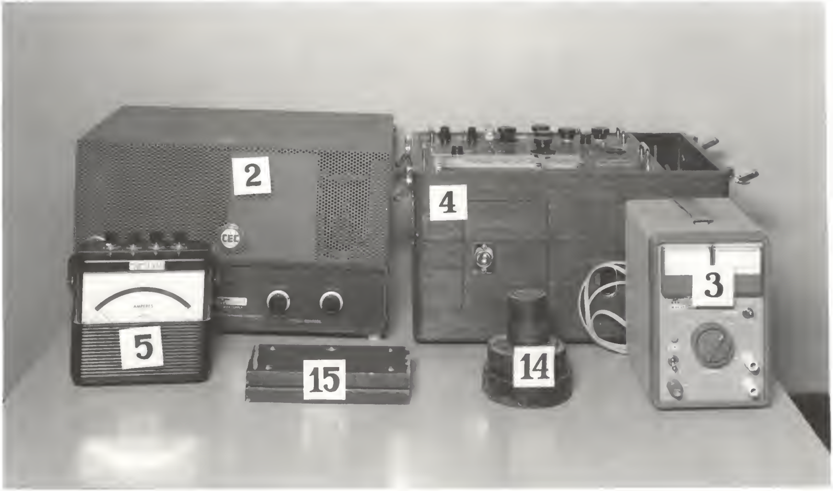

Figure 5 Experimental Equipment

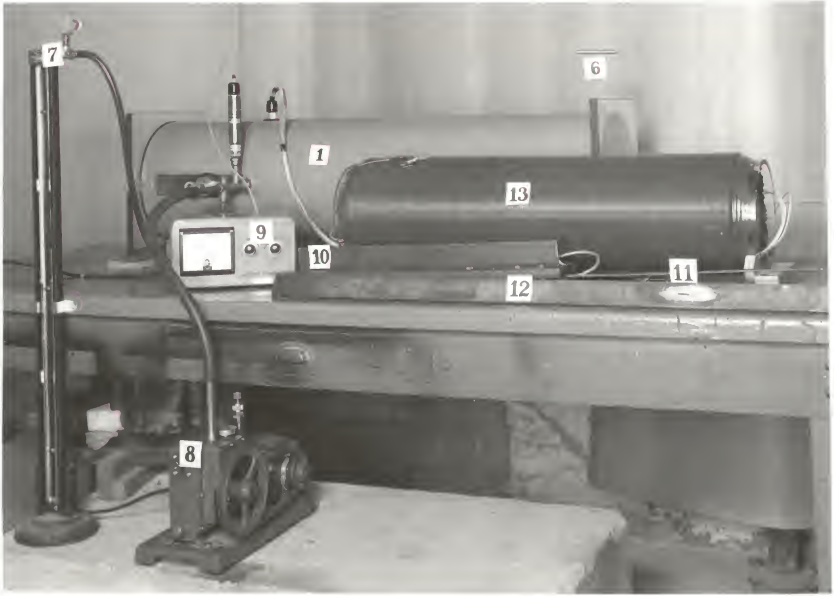

Figure 6 Experimental Equipment

Figure 7 Experimental Equipment

APPENDIX C

CALIBRATION PROCEDURE

The maximum stress in a cantilever beam is a function of the load and the beam characteristics. For the cantilever beam loaded at the end, the maximum stress is given by

$$S = \frac{Mb}{I} = \frac{PLb}{I} \quad . \tag{28}$$

For the large beam, the moment of inertia was 0.00879 in.^{4}, the beam length was 35.375 inches, and b was 3/16 inch. Using these values, the stress-load relationship was found to be

$$S = 754\ P \quad . \tag{29}$$

For the small beam, the moment of inertia was 0.000413 in.^{4}, the beam length was 35.313 inches, and b was 3/32 inch. The stress-load relationship was

$$S = 800\ P \quad . \tag{30}$$

The length of the small beam was changed during the frequency tests. The stress-load relationship found for these cases was

$$S = 22.7\ PL \quad . \tag{31}$$

Before each test, the beam was loaded progressively from 1/5 pound to 4 pounds. The resulting strain from this loading was measured by a strain gage mounted on the beam next to the support and recorded as a deflection on the Sanborn recorder. This gave a direct relationship between the maximum stress on the beam and a deflection on the recorder. After calibration, the recorder deflection is independent of the gage factor and temperature of the strain gage and all other strain gage or recorder characteristics.

Calibration curves were made for each case. Figure 8 is the calibration curve for the large beam in the small support. Figure 9 is for the small beam in the small support. Figure 10 is a typical calibration curve for the small beam in the large support.

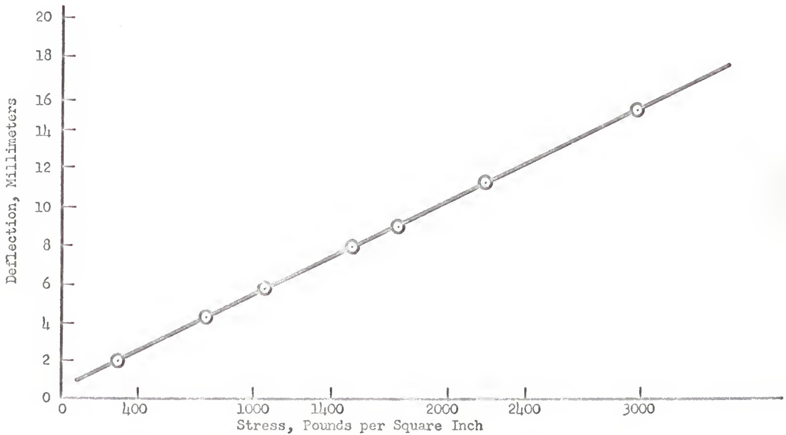

Figure 8 Calibration Curve, Maximum Stress versus Deflection, Large Beam in Small Support.

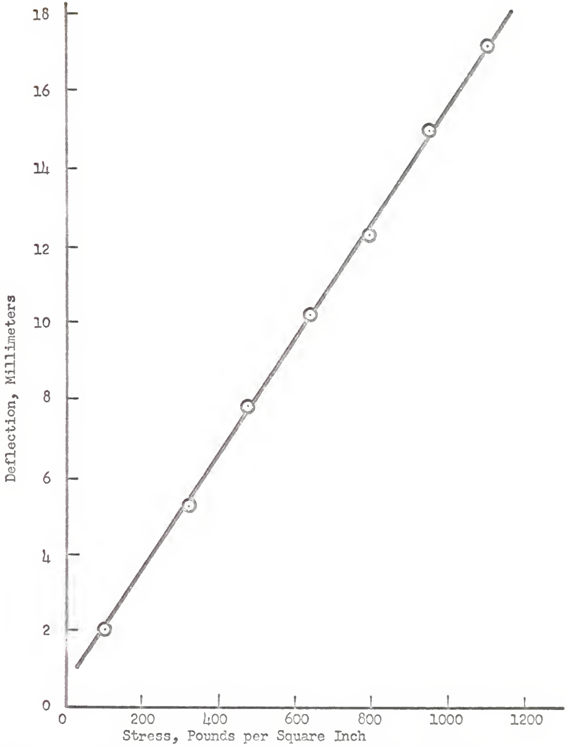

Figure 9 Calibration Curve, Maximum Stress versus Deflection, Small Beam in Small Support.

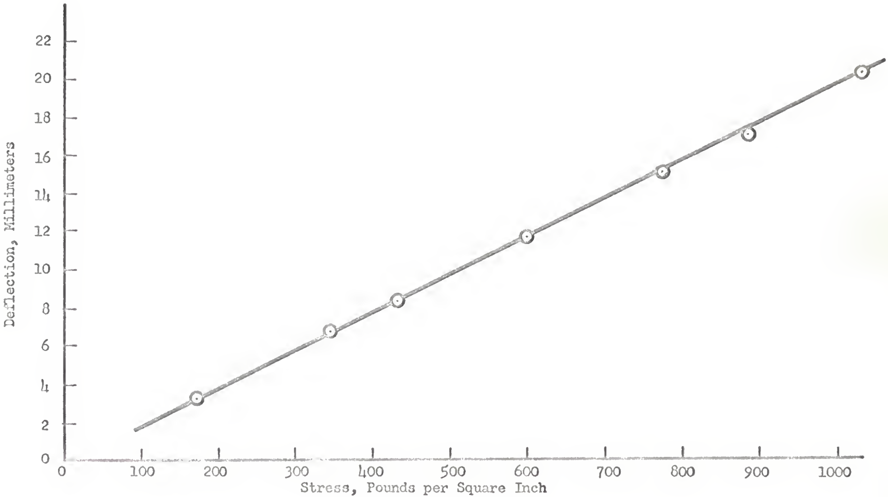

Figure 10 Typical Calibration Curve, Maximum Stress versus Deflection, Small Beam in Large Support.

APPENDIX D

DETERMINATION OF MAGNETIC SATURATION

Before the tests could be conducted, it was necessary to show that the magnetic field was strong enough to completely align the domains in the beams to insure magnetic saturation. This was done by increasing the current flowing through the coil and ploting magnetization current versus logarithmic decrement.

In the cases when the small support was used, saturation current was approximately 2.5 amperes. All tests were then conducted with 3.3 amperes flowing through the coil.

When the large support was used, the saturation level was approximately 3.5 amperes. All tests using the large support had 4.5 amperes flowing through the coil.

Figure 11 shows the magnetization curve of the large beam in the small support. Figure 12 shows the magnetization curve of the small beam in the small support. Figure 13 shows the magnetization curve of the small beam in the large support.

Saturation tests were not conducted for the cases when some of the metal on the large support was removed to change the frequency of the beam. However, the current was maintained at 4.5 amperes to insure saturation.

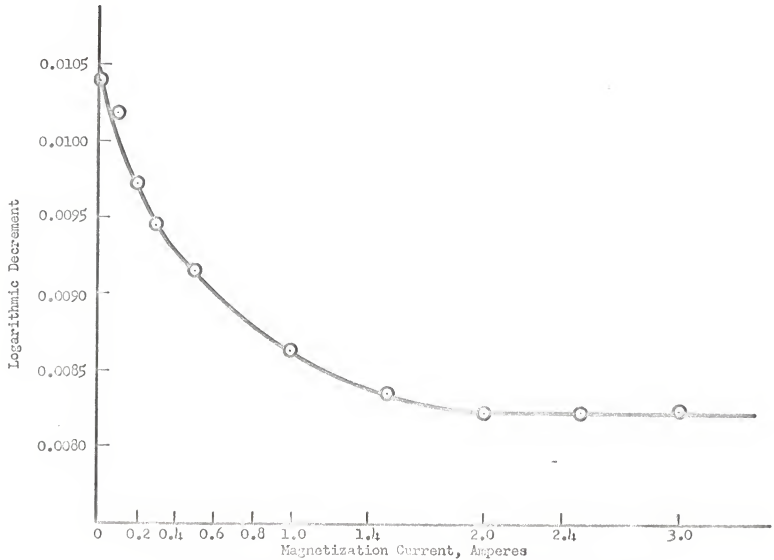

Figure 11 Saturation Current versus Logarithmic Decrement, Large Beam in Small Support.

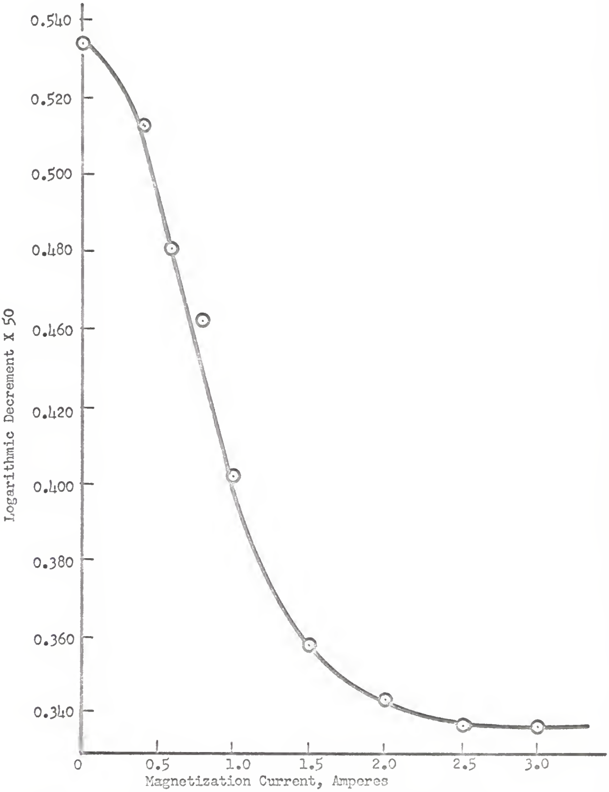

Figure 12 Saturation Current versus Logarithmic Decrement, Small Beam in Small Support.

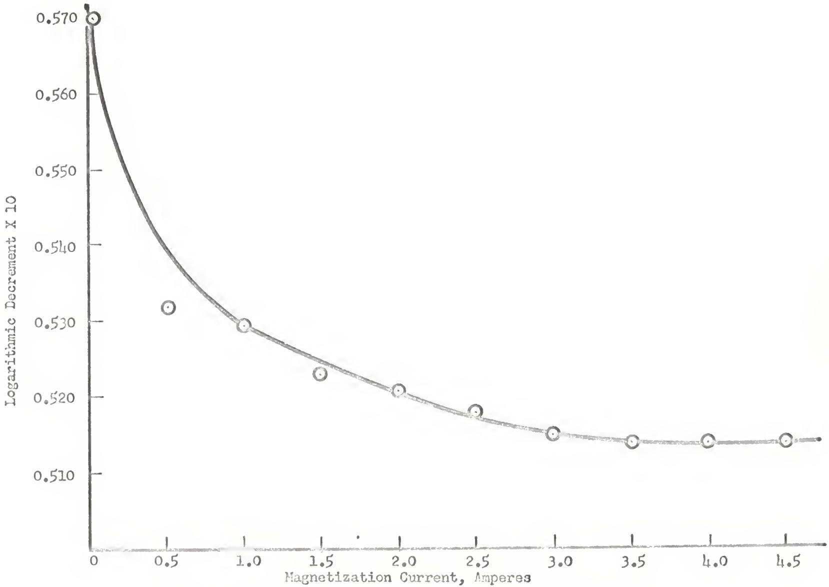

Figure 13 Saturation Current versus Logarithmic Decrement, Small Beam in Large Support.

APPENDIX E

TYPICAL DATA

Calibration Data: $S = 754\ P$, where P is the load in pounds.

Load, lbs.	Deflection, mm.	Stress, psi
0.4	2.0	301
1.0	4.3	754
1.4	5.7	1055
2.0	8.8	1508
2.4	9.1	1660
3.0	11.1	2262
4.0	15.0	3016

Experimental Data: For small beam in small support, 3.3 amperes magnetization current.

Pressure, in.	Stress, psi	A_o	A_n	n	$50\ \delta$
29.00	1000	33.2	19.8	50	0.519
27.00	1000	33.5	20.0	50	0.516
25.00	1000	32.9	20.2	50	0.488
23.00	1000	33.5	20.6	50	0.486
21.00	1000	33.2	21.0	50	0.457
19.00	1000	33.3	21.1	50	0.456
17.00	1000	33.0	21.2	50	0.443
15.00	1000	33.2	21.7	50	0.425
13.00	1000	33.4	22.0	50	0.417
11.00	1000	33.4	22.3	50	0.404
9.00	1000	33.3	22.3	50	0.400
7.00	1000	33.4	22.7	50	0.385
5.00	1000	33.0	23.1	50	0.356
3.00	1000	32.9	23.2	50	0.348
1.00	1000	33.5	23.9	50	0.336
0.06	1000	33.0	23.7	50	0.335
0.04	1000	33.1	23.7	50	0.335

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VITA

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Candidate for the Degree of

Master of Science

Thesis: PRESSURE AND FREQUENCY EFFECTS ON THE DAMPING OF A CANTILEVER BEAM IN A MAGNETIC FIELD

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PRESSURE AND FREQUENCY EFFECTS ON THE DAMPING
OF A CANTILEVER BEAM IN A MAGNETIC FIELD

by

JEROME PAUL DOW

B. S., University of Tulsa, 1965

AN ABSTRACT OF A MASTER'S REPORT

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1967

Name: Jerome Paul Dow Date of Degree: June, 1967

Institution: Kansas State University Location: Manhattan, Kansas

Title of Study: PRESSURE AND FREQUENCY EFFECTS ON THE DAMPING OF A CANTILEVER BEAM IN A MAGNETIC FIELD

Pages in Study: 42 Candidate for Degree of Master of Science

Major Field: Mechanical Engineering

Scope and Method of Study: Tests are conducted to isolate the effects of pressure and frequency on the damping of a vibrating cantilever beam. Two rectangular steel bars of different cross section are placed in a vacuum tank surrounded by a strong magnetic field. The pressure was lowered and the logarithmic decrement is plotted against pressure. A rectangular steel bar was also vibrated to measure frequency effects on internal damping. Tests were conducted in a vacuum and magnetic field environment. The frequency was changed by lengthening the vibrating beam. The pressure effects tests were conducted at a stress level of 3000 and 1000 pounds per square inch. The frequency effects tests were conducted at a stress level of 1000 pounds per square inch.

Findings and Conclusions: The experiment demonstrates that air pressure has an effect on the damping of a cantilever beam. The damping decreases in a linear manner as the pressure is decreased. A pressure can be reached where air has no effect on the damping.

The internal damping is shown to change slightly as the frequency changes. As the frequency is lowered, the internal damping is slightly lowered. This phase of the experiment is not conclusive because the vibrating cantilever beam does not have a constant stress distribution. Part of the results obtained may be caused by the nature of the stress.

MAJOR PROFESSOR'S APPROVAL ______________________