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THE MEASUREMENT OF MECHANICAL POWER FLOW INTO A SIMPLE PANEL

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Acoustics and Vibration Laboratory
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Cambridge, Massachusetts

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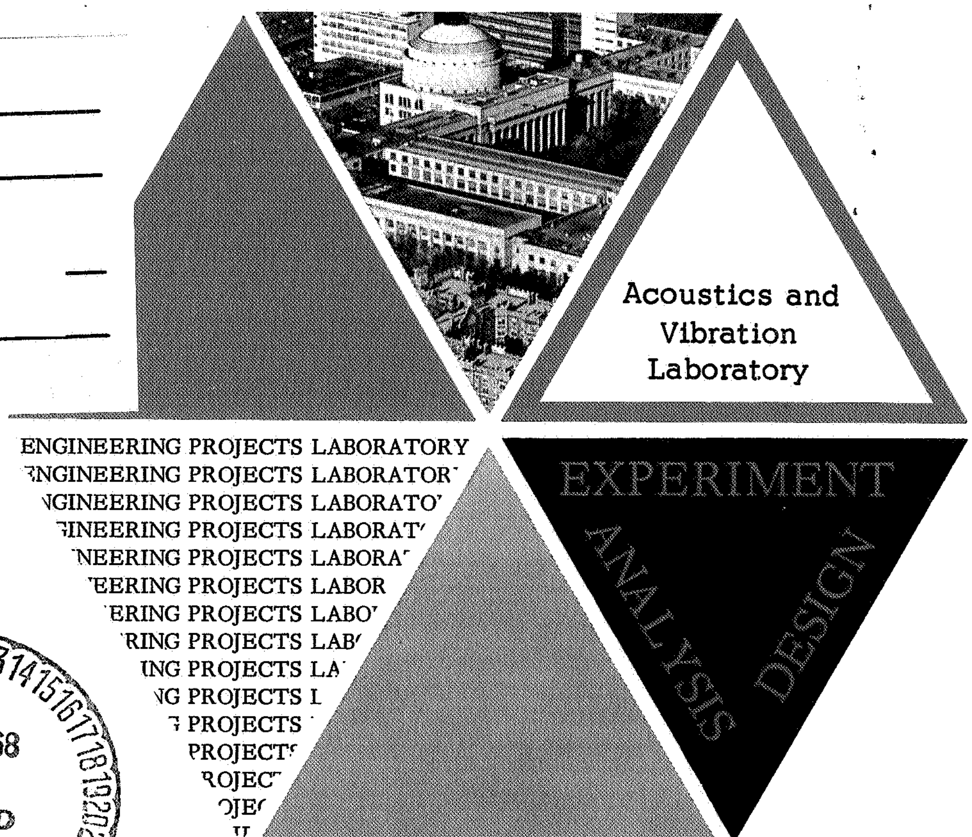

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ABSTRACT

The purpose of this investigation was to see how well we could measure mechanical power flow into a vibrating structure. To do this we excited a simple panel through a mechanical impedance head. The acceleration signal of the impedance head was integrated to give velocity, and then the velocity signal and the force signal from the impedance head were multiplied together and averaged to give the average power flow into the panel. The power dissipated in the panel, which is a function of the mean square acceleration and loss factor of the panel, was determined and compared to the measured power input. The agreement between the curves that we obtained show that power can be measured with a reasonable degree of accuracy.

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Introduction

The material presented in this report represents the first phase of a more extensive investigation. Overall we are interested in the correlation between the power flow into a complex structure and the vibratory response of the structure. By a complex structure we mean a structure for which the modal density is so high that normal mode analysis is no longer useful. This could easily come about for example when variations of dimensional tolerances and fabrication make it impossible to calculate the frequencies of the modes. We hope to arrive at a concrete basis for stating a dynamic principle, similar to the Saint Venant principle of statics, which would relate the space-time averaged acceleration over a portion of a structure to the total power flowing into the structure.

As a first step we have experimented with a simple panel excited by a single source. We measured the power flow into the panel with an impedance head, and compared this value with a calculated value of the power dissipated in the panel. The purpose of this experiment was two-fold: (1) We wanted to present an example where the results could be easily understood, and (2) we wanted to satisfy ourselves that our equipment was working properly. A similar test has been performed by other experimenters, nevertheless we felt that it was worth our while to repeat it before we tackled more complicated structures [1].

Analysis

In general what we want to do is relate the mechanical power flowing into a vibratory system to the amount of energy stored in the system in the form of kinetic and potential energy. Such a relation is almost the same as the definition of loss factor

$$\eta = \frac{\left\langle -\frac{dE}{dt} \right\rangle}{\omega_o \langle E \rangle} \tag{1}$$

which relates the rate at which energy is dissipated within a system to the total energy that is stored in it [2]. The pointed brackets in equation (1) are used to denote a time average and ω_o is the center frequency of a narrow band excitation. Since for a stationary process the rate of energy dissipation is equal to the power flow in, equation (1) is the relation that we desire.

We can write $\left\langle -\frac{dE}{dt} \right\rangle$ as P_{diss} for power dissipated, and interpret $\langle E \rangle$ as twice the total kinetic energy

$$\frac{M \overline{\langle a_r^2 \rangle}}{\omega_o^2} .$$

The bar above a_r^2 was introduced because the response acceleration a_r should be averaged not only with respect to time, but also with respect to the two-dimentional surface of the structure. The loss factor is related to the reverberation time T_r by $\eta = \frac{13.8}{T_r \omega_o}$. Using these relations equation (1) becomes

$$P_{diss} = \frac{13.8 \ M_s \overline{\langle a_r^2 \rangle}}{T_r \ \omega_o^2} . \tag{2}$$

This is a convenient form because it contains quantities that can be measured directly. Equation (2) is the basis of our comparison -- we measured P_{in} directly and calculated P_{diss} from the mean-square acceleration.

Experimental Apparatus and Procedures

Figure 1 shows the apparatus that we used for power measurements in a block diagram, and Figure 2 is a picture of the experimental set-up. We excited the panel with third octave band noise for center frequencies ranging from 200 Hz to 10 kHz. The panel was one-eigth inch thick aluminum with an irregular shape having a total area of about 7.5 square feet. The impedance head has outputs that are proportional to the force and

and the acceleration at the point where it is attached to the panel. We added two 20 Megohm resistors at the preamp inputs to raise the RC cut-off frequency of the force and acceleration signals to about 100 Hz. After amplification the acceleration signal was integrated and then multiplied by the force signal. The integration circuit was designed so that signals below 25 Hz would be attenuated. The multiplier output was averaged using a simple lag circuit with a 50 second time constant and the resulting D.C. level was read on the oscilloscope.

We used a lightweight accelerometer (two grams) to measure the reverberant field acceleration of the panel at usually about 20 positions.

Reverberation times for third octave band excitation were measured with the instrumentation shown in Figure 3. For frequencies less than 4000 Hz we excited the panel with a loudspeaker and recorded the decaying signal on the graphic level recorder. For frequencies above 4000 Hz we excited the panel with a small crystal disk and recorded the decaying signal using an oscilloscope and a camera.

Experimental Results

Our first tests with the panel were not successful essentially because there was not enough damping in the system. Although the force and velocity signals were quite large, the correlation between them was negligible. We resolved this difficulty by changing our method of supporting the panel so that more energy could be absorbed by the supports. At first we suspended the panel vertically by two cords that behaved more or less like pendulums. Later we suspended the panel horizontally with four cords such that the panel vibrations caused the cords to stretch and, in some cases, even flex the wooden beams to which the cords were attached. This change in the method of suspension increased the loss factor of the system by roughly a factor of ten.

Contrary to other experiments [1], we found that the RMS acceleration of the panel varied by as much as 10 db at different positions. Therefore, to obtain the total kinetic energy of the panel, we found it necessary to take about twenty measurements of the panel acceleration and then average these readings. For center frequencies above 1000 Hz the maximum variation of acceleration was only about 2 db so we did not do as much space averaging in this frequency range.

The results that we obtained are shown in Figure 4. Overall the agreement is quite satisfactory. For this test we held the acceleration of the panel constant at 1/2 g_{rms} so that non-linearities at the supports due to an actual loss of physical contact would be minimized.

The sudden rise in the curve at 4000 Hz comes about because the critical frequency is reached; this also shows up in the loss factor curve (Figure 5) [3]. The critical frequency is that frequency at which the bending wavelength in the panel becomes equal to the wavelength of the sound radiated. We found that for frequencies less than critical frequency that the suspension was the major source of damping, whereas for frequencies above the critical frequencies sound radiation was the major source of damping.

There is some disagreement between P_{in} and P_{diss} for center frequencies less than 2000 Hz. One probable reason for this may be that, besides the panel, the beam supports were also vibrating somewhat. However, the mass of the beams was not included as part of the mass of the structure, M_s, used in our calculations. Above the critical frequency the mass of the beams is not important.

Conclusions

The good agreement between the direct measurement of the power flow into the panel and the indirect measurement of the power dissipated, as indicated in Figure 4, gives us strong reason for believing that our instrumentation is working correctly and that we can measure power flows in the frequency range from 200 to 10,000 Hz. By using the results of Figure 4 we could predict space average acceleration levels in the plate by simply measuring the power flow into the plate.

As mentioned earlier these satisfactory results were obtained when the loss factor ranged from .0003 to 0.01 as indicated in Figure 5. Similar, or better, results would be obtained for systems with greater damping. For systems with lighter damping however the measurement of power flow becomes very difficult. We were unable to obtain meaningful results when the loss factor was ten times smaller than that shown in Figure 5.

REFERENCES

1. Noiseux, D. U., Dietrich, C. W., Eichler, E., and Lyon, R. H., "Random Vibration Studies of Coupled Structures is Electronic Equipments - Volume II," Bolt, Beranek and Newman, Incorporated, Report No. 1061, 15 October 1963.
2. Smith, P. W. and Lyon, R. H., "Sound and Structural Vibration," NASA Contractor Report, NASA CR-160, March, 1965.
3. Beranek, L.L., Noise Reduction, McGraw-Hill Book Company, New York, 1960.

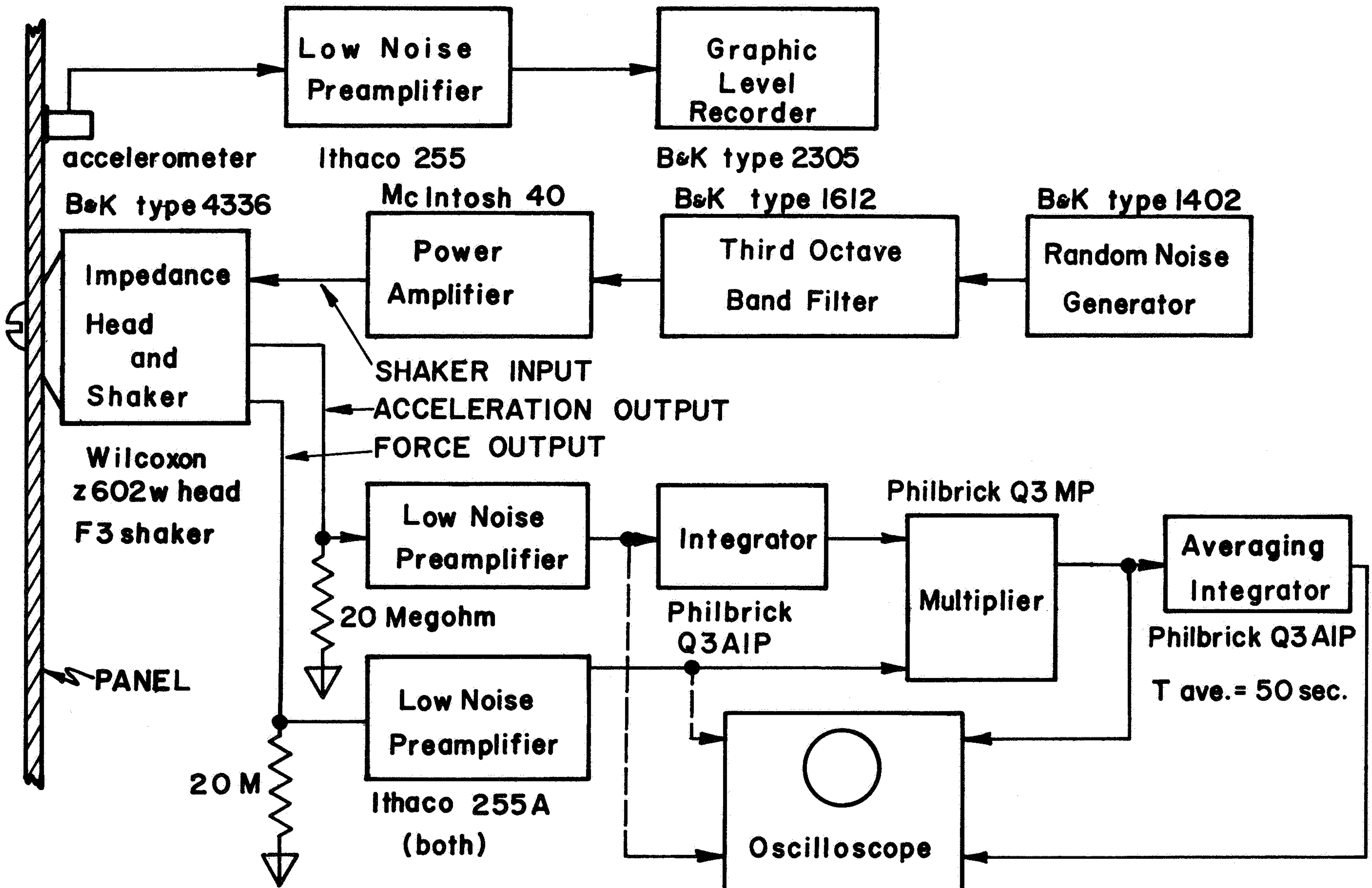

Fig.1 POWER MEASURING SYSTEM

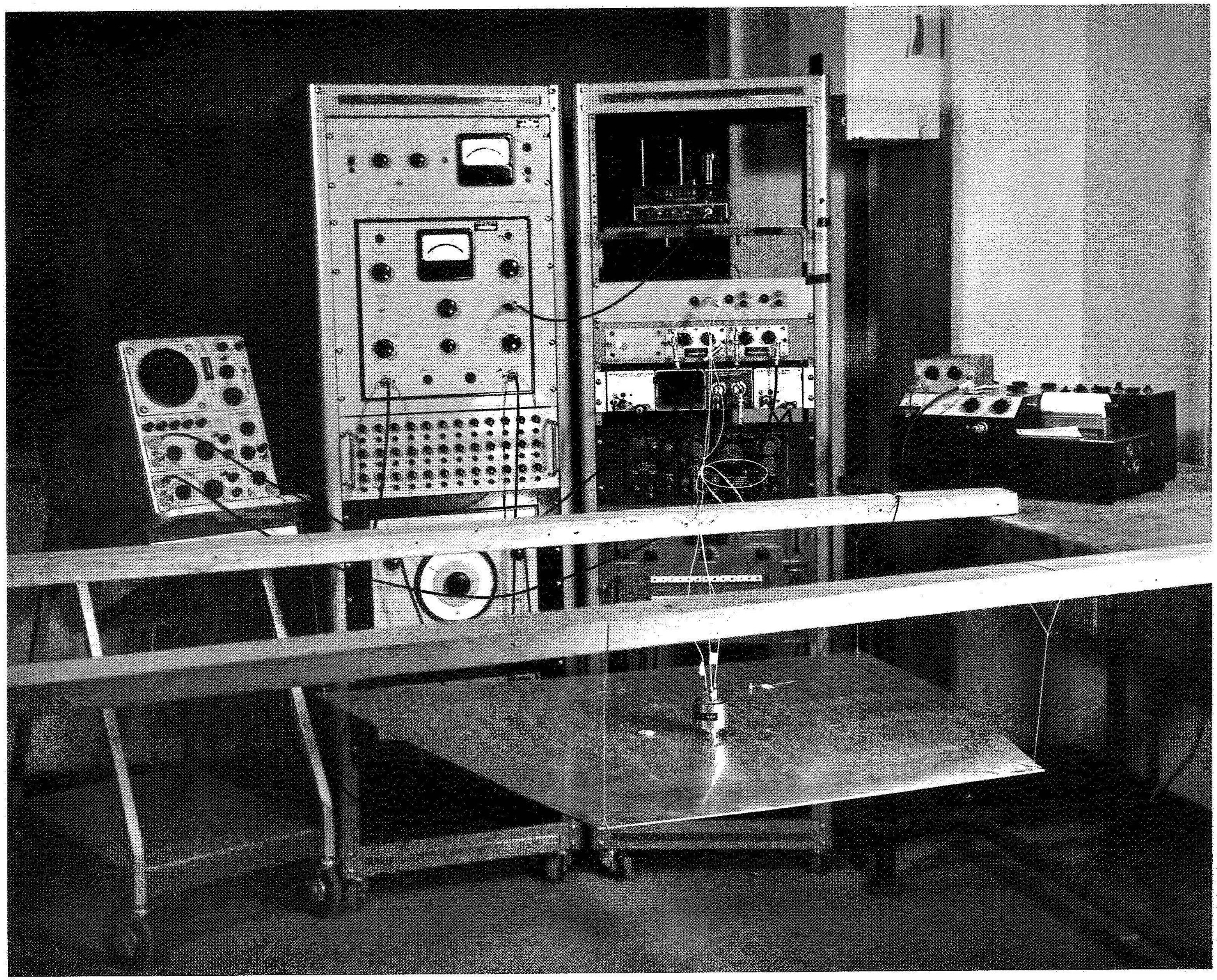

FIG. 2 Photograph of experimental setup

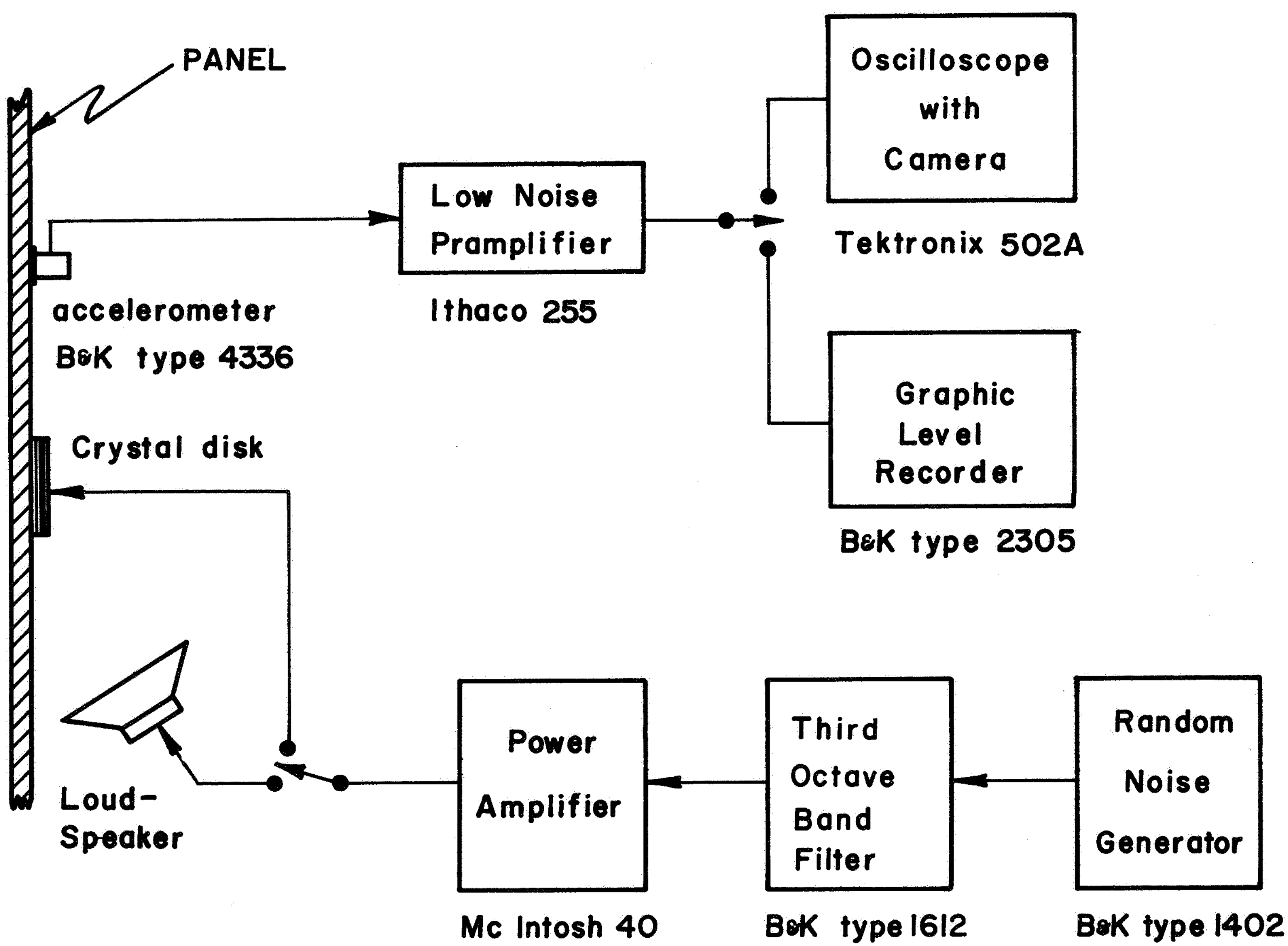

Fig.3 EQUIPMENT FOR REVERBERATION TIME MEASURMENTS.

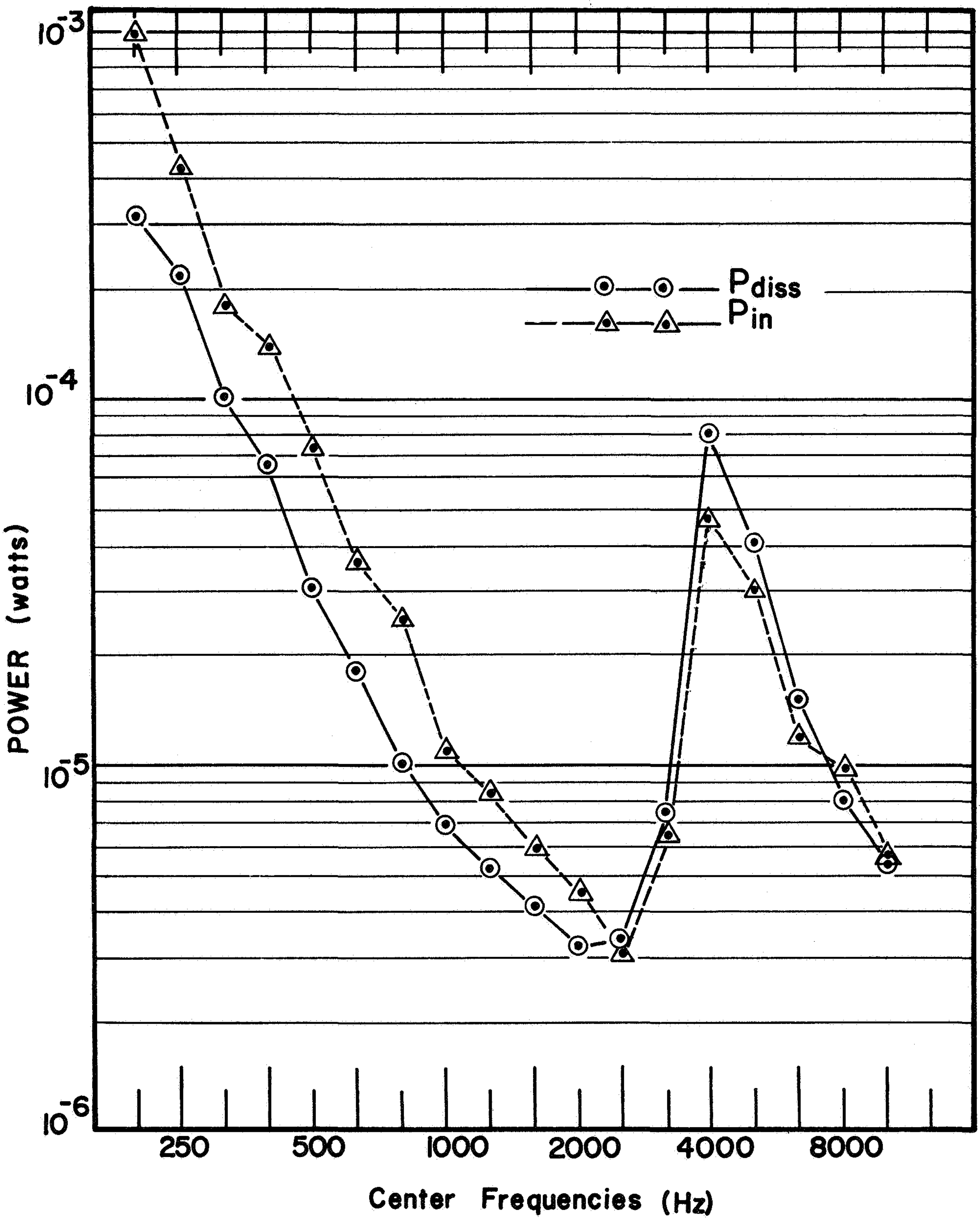

Fig.4 Comparison of P(in) and P(diss) for panel.

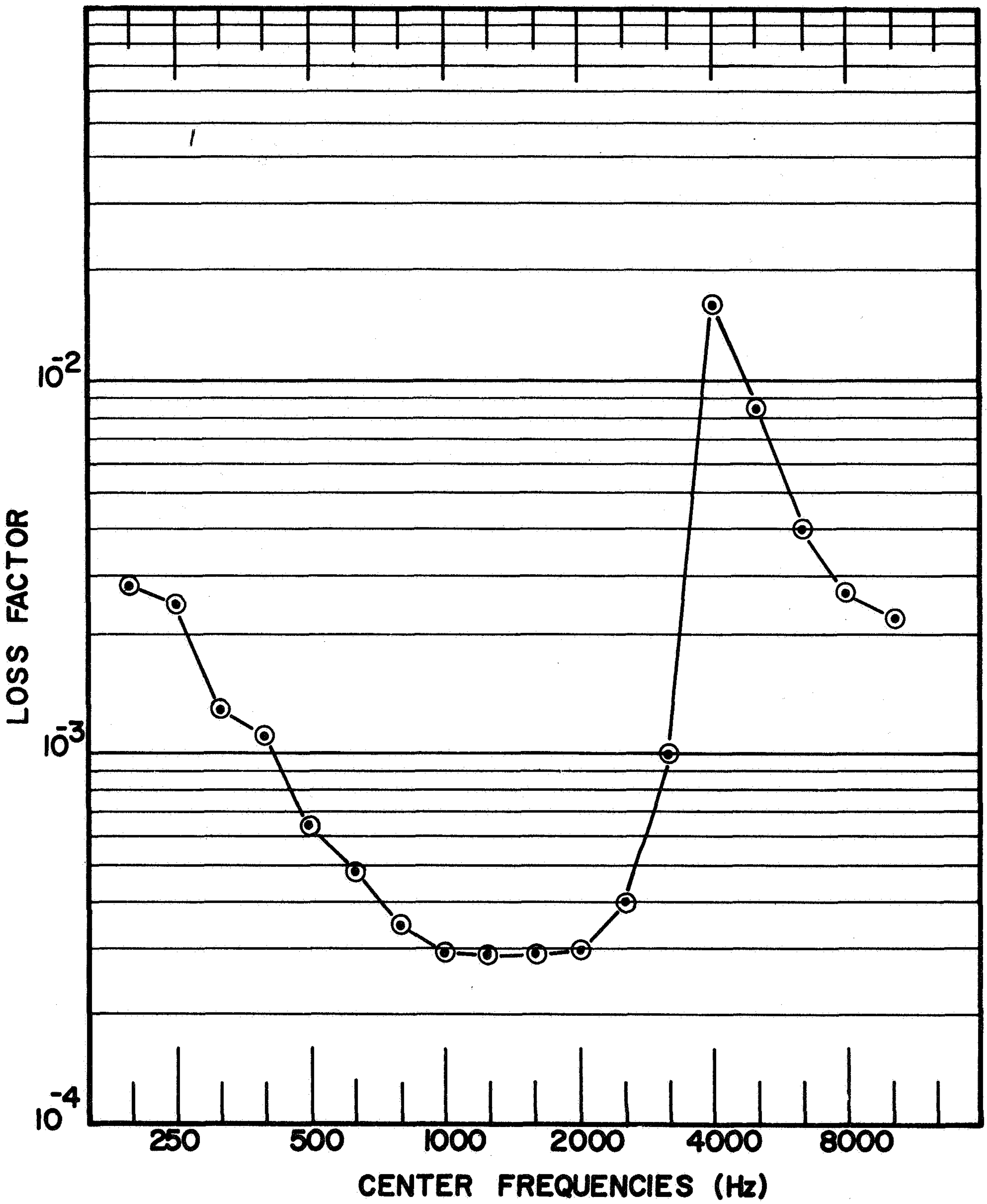

Fig 5. Loss factor of panel.