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Title: Free Displacer and Ringbom Displacer for a Malone Refrigerator

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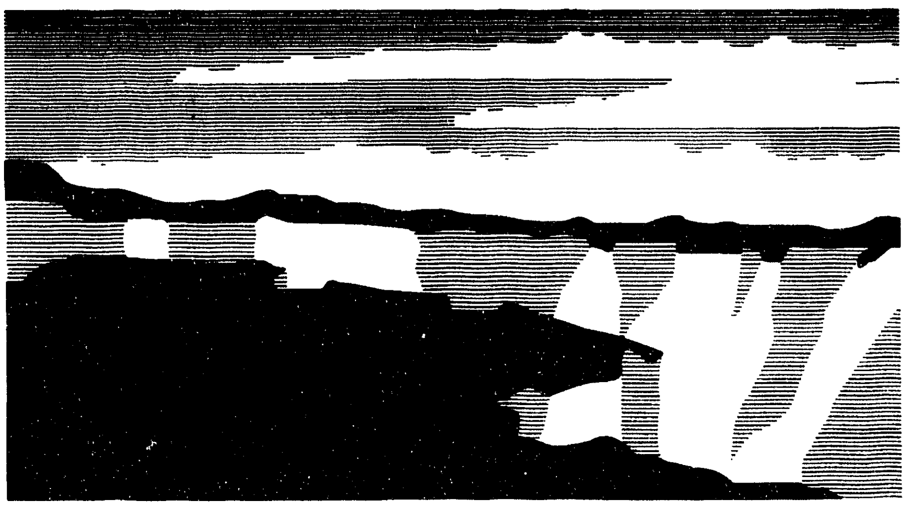

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Free displacer and Ringbom displacer
for a Malone refrigerator

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ABSTRACT

First measurements with a free displacer used in a liquid working fluid are presented. The displacer was operated both in harmonic mode and in Ringbom mode, in liquid carbon dioxide. The results are in reasonable agreement with expectations.

INTRODUCTION

Malone refrigeration uses a liquid near its critical point (instead of the customary gas) as the working fluid in a Stirling, Brayton, or similar regenerative or recuperative cycle.[1] Thus far, we have focused on the Stirling cycle, to avoid the difficult construction of the high-pressure-difference counterflow recuperator required for a Brayton machine. Our first Malone refrigerator[2] used liquid propylene (C_3H_6) in a double-acting 4-cylinder Stirling configuration.

To begin development of a simpler, less expensive, and more environmentally acceptable Malone refrigerator, we have begun construction of a linear free-piston apparatus using liquid carbon dioxide. In this paper we report the first measurements with part of that apparatus: its free displacer. We have operated the displacer both in its "free" mode,[3] with essentially harmonic motion, and in Ringbom mode,[4] in which the displacer spends part of the time at rest against stops at the extremes of its travel. As with gas Stirling machines, the harmonic operation is easier to model, but Ringbom operation is easier to implement. In both cases, operation is consistent with expectations. The most significant differences from gas Stirling machines are: the large contribution of the dense working fluid to the effective moving mass of the displacer; the large bounce-space spring constant due to the low compressibility of the working fluid; and the small kinetic energy loss in Ringbom operation.

APPARATUS

Scale drawings of two perpendicular central cross sections of the displacer and related parts are shown in Fig. 1. The displacer itself is a hollow stainless steel cylinder. Brass rings on both ends (not shown in the figure) provide bearing surfaces, allowing the displacer to slide in its stainless steel cylinder without galling, with about 50 μm clearance. The maximum peak-to-peak stroke allowed by the cylinder length was 2.0 cm, although there was no intent of operating beyond 1.5 cm stroke because in the last few mm of travel the displacer blocked the tubes to the regenerator. Because of construction difficulties, we had no heat exchangers or regenerator in the system for the present measurements; instead, thirty 1-cm-i.d. tubes in parallel served as a fake regenerator, with about the same fluid volume as the real regenerator would have had, but of course with much less flow impedance.

The displacer hollow or bounce space, penetrated by the stationary rod, caused the displacer to move in response to oscillating pressure around it, in the usual[3] free-piston fashion. The rod was made of case-hardened steel, ground to fit the reamed hole into the displacer. A hole through the length of the rod, connected to a tube outside the apparatus at the bottom, allowed external control of mean pressure in the hollow to balance the force of gravity on the displacer. To minimize leakage between the rod and the reamed hole in the displacer, the lengths of the rod and its clearance hole in the displacer were made almost as long as the displacer.

We instrumented the apparatus to measure pressure P and displacer position x_d as functions of time. Pressure was measured with a piezoresitive sensor connected to one of the regenerator tubes, and x_d was measured by a linear transformer sensing the position of a magnetic rod extending from the top of the displacer. Signals from both transducers were smoothed by 6-pole Butterworth filters with 50 Hz cutoffs, to reduce noise and, in the case of the position sensor, to filter out the 100 Hz transformer sensing frequency. The time average of x_d was used to feedback control the mean pressure in the hollow, to balance the force of gravity on the displacer.

As working fluid for these measurements, we used carbon dioxide at T = 290 K and P = 13 MPa ± 1 MPa. The pressure at this operating point is about double the critical pressure of 7.4 MPa, and the temperature is only slightly below the critical temperature of 304 K. The fluid properties at this operating point are different from those of the gases usually used in Stirling machines, with resulting differences in some of the behavior of the apparatus. First, at this operating point, the compressibility is lower than that of an ideal gas: $P\kappa_s = 0.06$, where κ_s is the adiabatic compressibility, compared to the monatomic ideal gas value $P\kappa_s = 1/\gamma = 0.60$. Thus, with the effective spring constant k of the hollow given by

$$k = A_r^2/V_h\kappa_s = 1400 \text{ N/m} \quad , \tag{1}$$

(where A_r is rod area and V_h is hollow volume) the spring constant is ten times stiffer than it would be with an ideal gas working fluid. Second, the density ρ

$= 0.92$ gm/cm^3, nearly that of water. Because of this high density, the effective moving mass m_{eff} of the displacer has significant contributions from moving fluid, particularly in the narrow tubes connecting the displacer cylinder to the fake regenerator. We find $m_{eff} = 9.6$ kg, with 27% of that mass due to moving metal, 1.4% due to the fluid in the hollow, 64% due to the fluid in the connecting tubes, and 7% due to that in the fake regenerator. With these values of k and m_{eff}, the resonance frequency for the displacer is $\sqrt{k/m_{eff}}/2\pi = 2$ Hz, a little below any frequency at which we operated the system.

Third, with viscosity $\mu = 0.00015$ kg/m-s about a seventh that of water, we must consider the viscous effects in the connecting tubes on the displacer dynamics. The Reynolds number of the flow in the tubes is about 30,000 at the highest amplitude observed in the next section. Using the steady-flow friction factor to estimate the damping, and using the oscillating flow for inertial effects as in the paragraph above, we estimate that viscosity could reduce the displacer amplitude by roughly 10% and cause a phase shift between motion and driving pressure of roughly 30°. At smaller amplitudes, viscosity should have an even smaller effect.

Hence, for qualitative understanding of the results below, one may regard this system as a harmonically driven mass, with small spring-like restoring force and small damping. (Note, however, that we expect ultimately to operate this system with a dilute solution of alcohol in carbon dioxide, in order to raise the critical temperature; this will increase κ_s at this operating point by a factor of 2 or 3, and hence make k even more important. We also expect to add much more damping with real heat exchangers and regenerator.)

HARMONIC OPERATION

For small enough displacement amplitudes that the displacer doesn't hit the ends of the cylinder, the motion of the displacer should be that of a simple harmonic oscillator. Neglecting damping,

$$-\omega^2 m_{eff} x_d = -kx_d - A_r P \qquad (2)$$

is the equation of motion, where P and x_d here are the amplitudes of the sinusoidal oscillations of pressure and displacer position, and $\omega = 2\pi f$ is the angular frequency of the oscillations. The solid line in Fig. 2 shows x_d/P as a function of frequency, calculated using Eq. (2) and the values of m_{eff}, A_r, and k given above. For comparison, the dashed line shows the same result but with k neglected, to give an indication of the small importance of the hollow's spring constant at these high frequencies. The symbols show experimental measurements of the same quantity, taken with pressure amplitudes from 0.2 to 0.7 MPa. They are in good agreement with the lines, though the quality of the data and calculations is not good enough to provide evidence for the size of the spring-constant term. We were prevented from making measurements at lower frequencies, where k would have been more important, by irregular frictional effects in the <u>power</u> piston, which caused nonsinusoidal pressure oscillations.

RINGBOM OPERATION

Malone machines tend to have short strokes, low operating frequencies, and high pressure amplitudes. These characteristics lend themselves to Ringbom operation[4]. For example, for a peak-to-peak stroke of 1 cm at 5 Hz, the kinetic energy loss of our displacer in Ringbom mode is only about 1 Watt, negligible compared to the refrigerator's expected cooling power of 1 kW. The many practical advantages of Ringbom operation over harmonic free piston operation are discussed at length by Senft.[4]

To achieve Ringbom operation in our system (which was designed for harmonic operation) without requiring excessively large pressure oscillations or low frequencies, we put 5-cm-diam teflon stops in the ends of the cylinder to limit the displacer's peak-to-peak travel to 0.39 cm. Two pairs of stops were used: the "ungrooved stops" presented full-area smooth flats to the displacer; the "grooved stops" had 20 radial and 1 circumferential grooves of a few mm^2 cross section facing the displacer so that the contact area between displacer and stops was about half the original stop area and no contact area was more than a few mm from the nearest groove.

Figure 3 shows displacer motion in the Ringbom regime, obtained with a storage oscilloscope, typical of what we observed with the ungrooved stops. The motion is qualitatively what one would expect for a Ringbom displacer. With such a dense, incompressible working fluid, the fluid cushion seems very effective in stopping the displacer quickly but with no observable bouncing. But note that the displacer begins accelerating away from rest near the peak in pressure, rather than near the zero crossing in pressure as one would expect for a mass-dominated Ringbom system.

Suspecting that the ungrooved stops were preventing easy acceleration of the displacer from rest, we installed the grooved stops. Figure 4 is typical of observations with these stops, and shows acceleration of the displacer from rest slightly before the zero crossings in the pressure, as one would expect for a mass-dominated Ringbom system with a small k. But some bouncing of the displacer as it comes to rest against the stops is apparent.

At an operating frequency of 4.25 Hz, we recorded displacer position as a function of time at several pressure amplitudes, and measured the fraction of time the displacer spent at rest against its stops. There is significant uncertainty in these measurements because of the bouncing phenomenon, especially at low stopped-time fractions. The results are shown in Fig. 5. Below about 0.25 MPa, the displacer exhibits harmonic motion, with zero stopped time; above this pressure amplitude, it exhibits Ringbom motion.

To check these results for reasonableness, we calculated the stopped time using a simplified version of Senft's analysis.[4] We model the system as a mass m_{eff} on a spring k as discussed above, driven by a force $A_r P\sin(\omega t)$, and constrained by rigid non-bouncy stops at $x_d = \pm L$. At rest against one stop, the displacer begins to move away from the upper stop at time t_1 when the pressure force falls below the spring force, so t_1 is given by

$$A_r P\sin(\omega t_1) = -kL \quad . \tag{3}$$

The displacer then moves according to

$$m_{eff} \frac{d^2 x_d}{dt^2} = -\ kx_d - A_r P\sin(\omega t) \quad , \tag{4}$$

subject to the initial conditions $x_d = L$ and $dx_d/dt = 0$ at $t = t_1$. Obtaining the analytical solution for $x_d(t)$ is straightforward but tedious. The time t_2 at which the displacer hits the lower stop is then given by $x_d(t_2) = -\ L$. Finally, the fraction of time spent stopped is given simply by $1 - (t_2 - t_1)\omega/\pi$. This function is plotted as the solid line in Fig. 5. The agreement with the measurements is reasonable, considering the uncertainty in the measurements due to displacer bouncing. We also display with dashed line the calculated result for k = 0, to indicate the magnitude of the effect of the bounce space spring constant.

ACKNOWLEDGMENTS

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4. James R. Senft, Ringbom Stirling Engines (Oxford, New York, 1993).

FIGURE CAPTIONS

1. Scale drawing of the apparatus, showing two central cross-sections. For clarity, bolts are not shown.

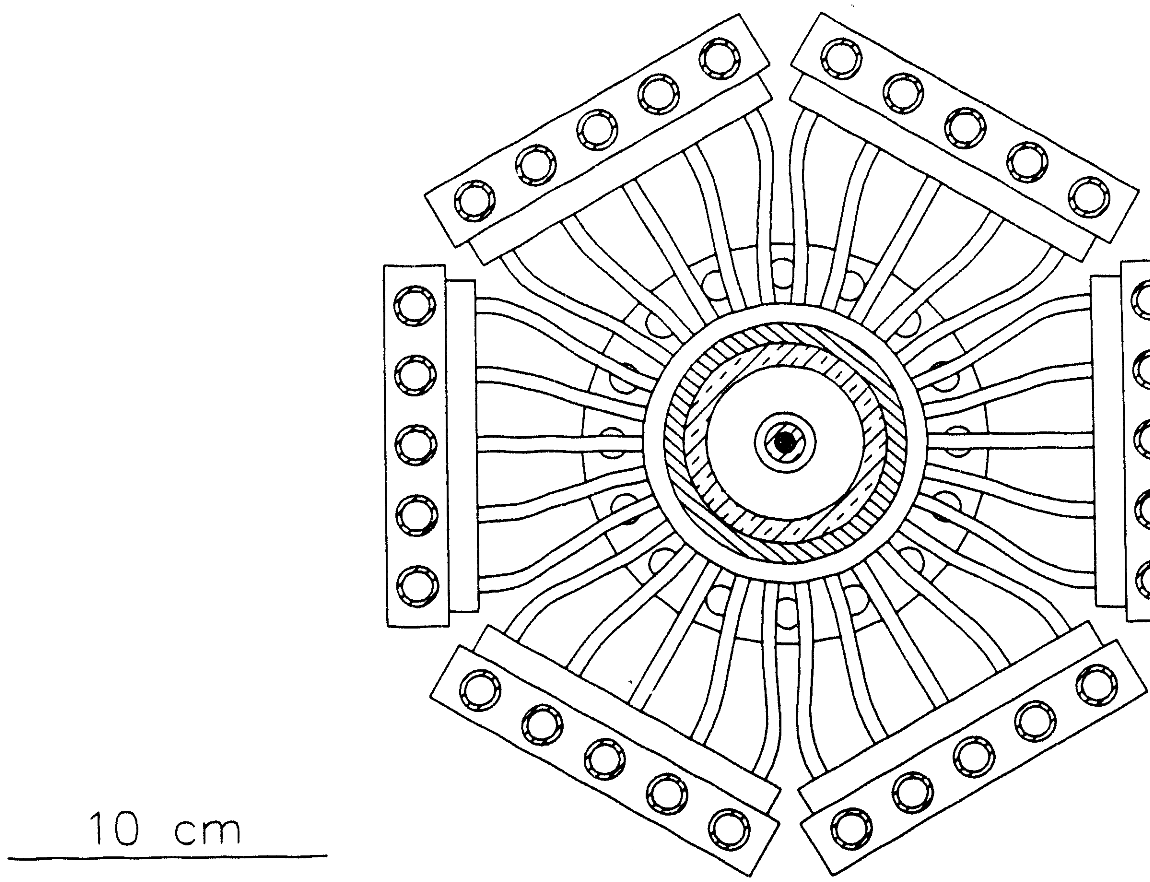
2. Displacer amplitude, normalized by pressure amplitude, as a function of the square of the period of the oscillation. The circles show experimental results; the solid line is our estimate using Eq. (2), neglecting viscous effects. The dashed line further neglects the spring constant k of the hollow, so it represents just a harmonically driven mass.

3. Typical displacer motion as a function of time in Ringbom mode with ungrooved stops. The truncated waveform is the displacer position; the nearly sinusoidal waveform is the pressure.

4. Typical displacer motion as a function of time in Ringbom mode with grooved stops. The truncated waveform is the displacer position; the nearly sinusoidal waveform is the pressure.

5. Fraction of time the displacer is stopped in Ringbom mode, as a

function of pressure amplitude, at f = 4.25 Hz. Circles are experimental data with grooved stops, and solid line is simple calculation described in text. Dashed line is calculation omitting effect of spring constant of bounce space.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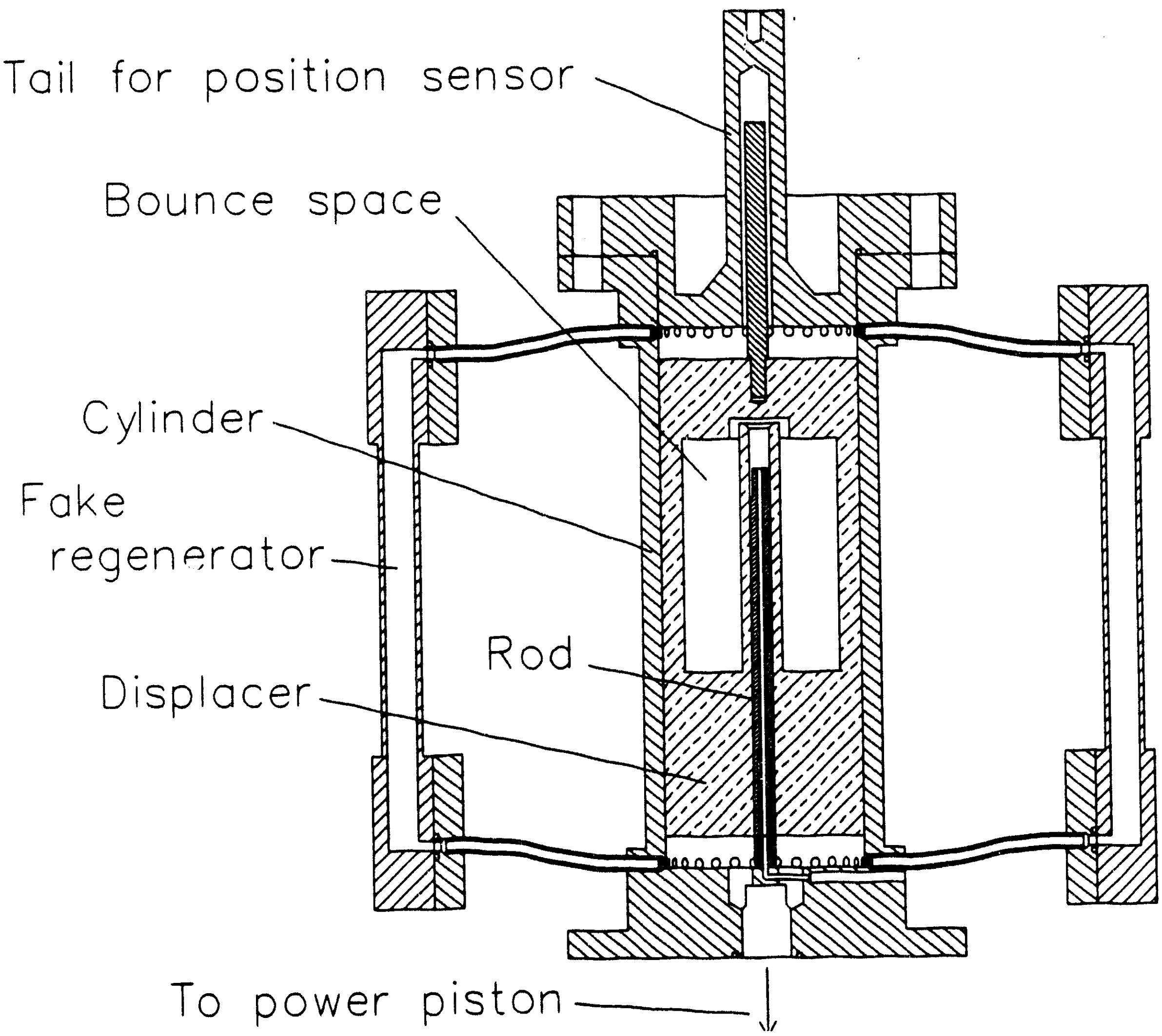
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Bounce space
Cylinder
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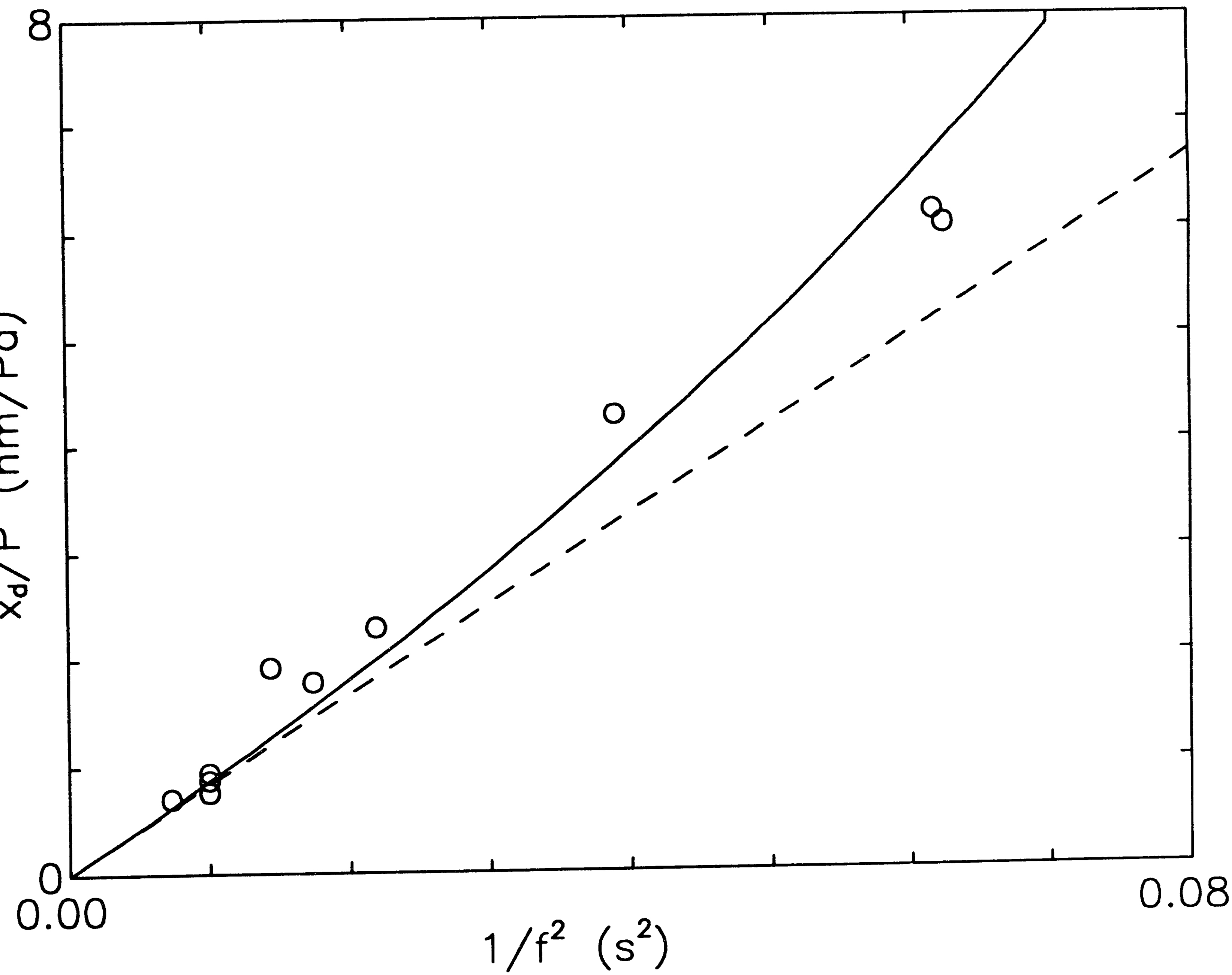
Rod
Displacer
To power piston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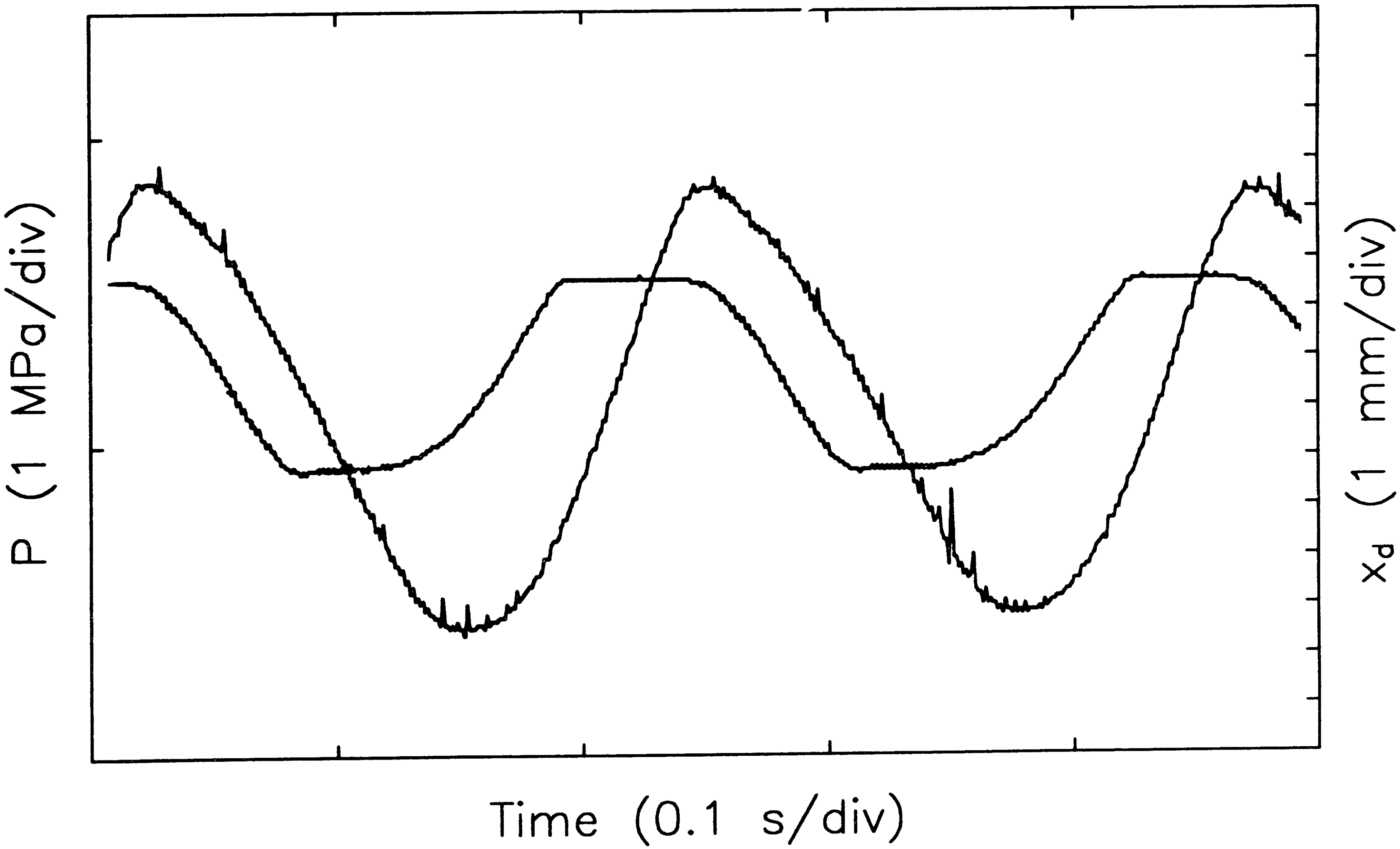
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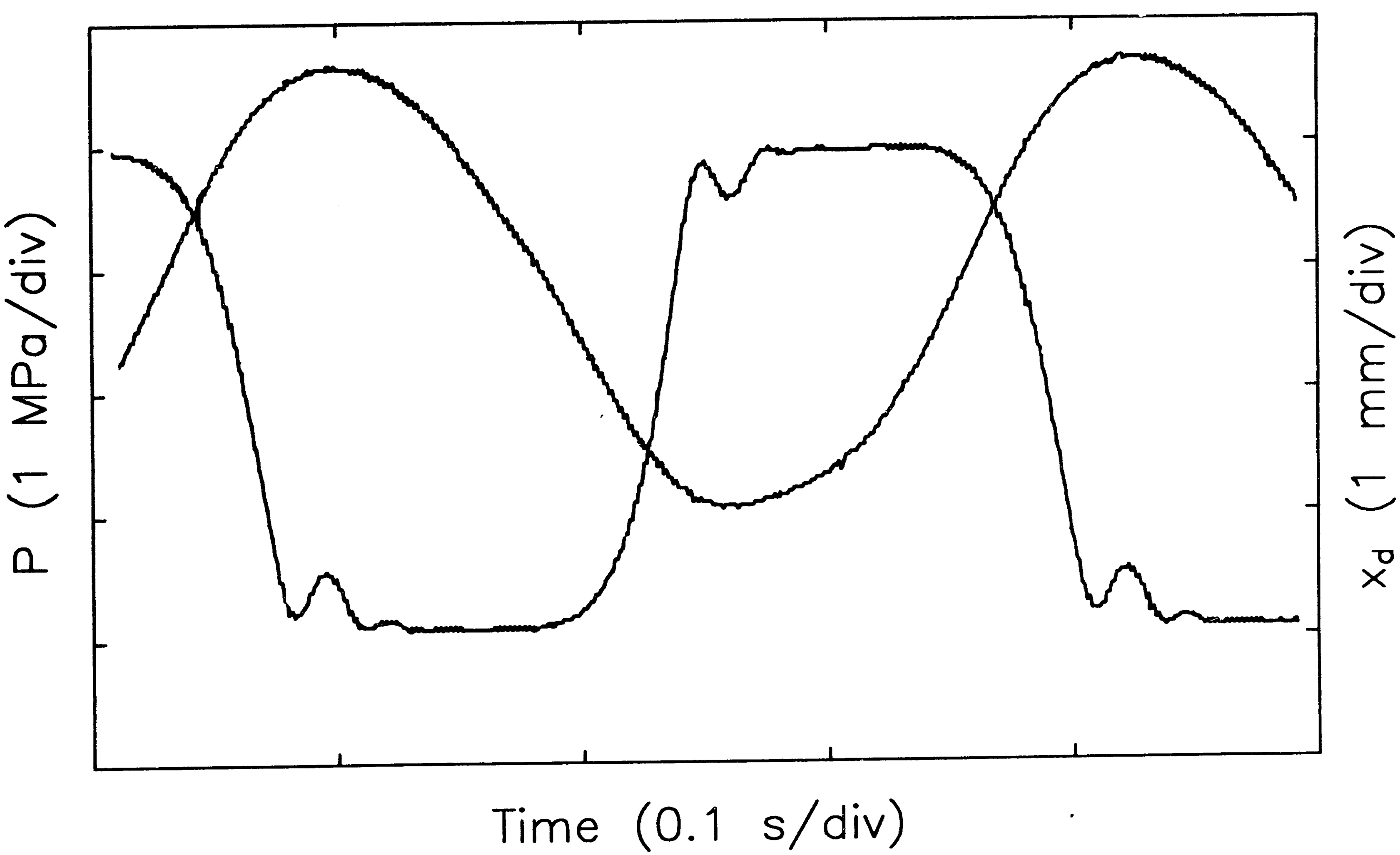
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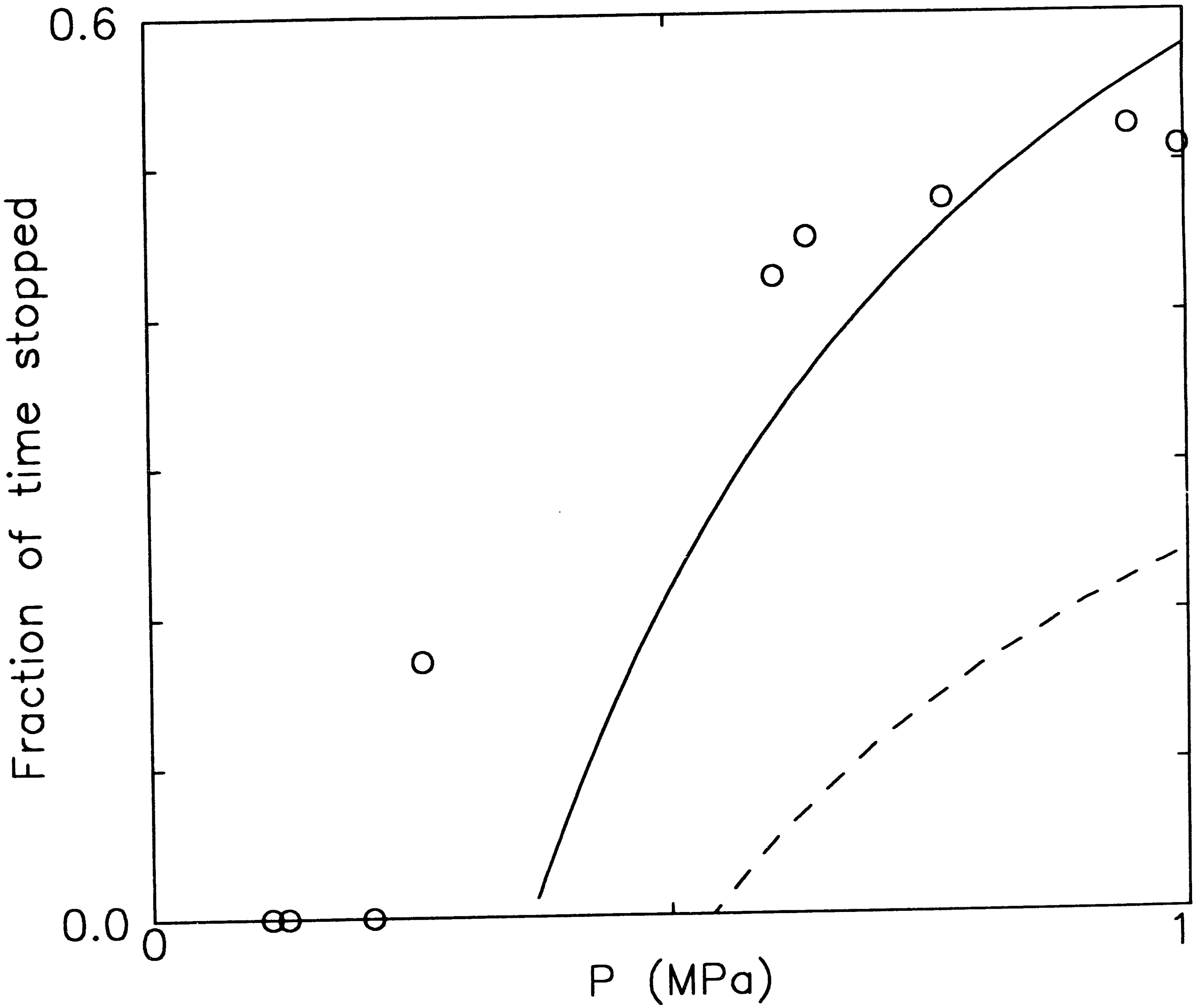
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