

DESIGN AND CONSTRUCTION
OF A
LINEAR MAGNETIC PARTICLE BRAKE

by

MATTHEW VALENTINE THOMPSON

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Signature of Author

Department of Mechanical Engineering
May 8, 1981

Certified by

Woodie C. Flowers
Thesis Advisor

Accepted by

Chairman, Department Thesis Committee

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Submitted to the Department of Mechanical Engineering
on May 8, 1981 in partial fulfillment of the
requirements for the Degree of Bachelor of Science in
Mechanical Engineering

ABSTRACT

The basis for the design and construction of a linear magnetic particle brake was an evaluation of the possible use of a linear magnetic particle brake to control the damping in an above-the-knee prosthesis.

The prototype functioned successfully, with a maximum output to weight ratio of 8:1 and a maximum output to zero current drag ratio that averaged 3.45:1. The brake exhibited smooth, repeatable operation. The brake also exhibited a nearly linear response for the output forces versus the input currents until it reached magnetic saturation.

Linear magnetic particle brakes require careful design to achieve performance levels equivalent to those achieved by commercially available rotary magnetic particle brakes.

A next generation design is proposed that could be used in practical applications and would have a maximum output to weight ratio of 48:1.

Thesis Supervisor: Dr. Woodie C. Flowers
Title: Associate Professor of
Mechanical Engineering

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INTRODUCTION

A magnetic particle brake is a device that generates braking forces or torques through a shearing action in a magnetically solidified powder in a gap between a stationary and moving member. Rotary magnetic particle brakes are commercially available. These brakes offer smooth, repeatable, and energy efficient operation, with the braking torques produced exhibiting linear proportionality to the current supplied, independent of the speed of rotation. At present, linear magnetic particle brakes are not commercially available.

Currently, a rotary magnetic particle brake is being used in a microprocessor-controlled prosthesis for above-the-knee (A/K) amputees. The microprocessor controls the damping at the knee joint of the prosthesis by controlling the current supplied to a rotary magnetic particle brake. Rotations about the knee joint become linear extensions and compressions through a torque-arm link at the joint. A ballnut assembly is used to convert the linear extensions and compressions into shaft rotations that can be braked by the rotary brake. The rotary brake and the ballnut assembly are assembled into a device that looks and acts like a shock absorber. Problems with this arrangement include: mechanical inefficiencies in the linear-rotary conversion, backlash, and noise.

The primary concern of this thesis is the investigation of the feasibility of the design, construction, and use of a linear version of the rotary magnetic particle brake in the A/K prosthesis. Performance specifications to be considered in the evaluation of a linear brake include: 0 - 400 pounds (0 - 1612 Newtons) force output on a maximum of 150 - 200 milliamperes input current, flat frequency response with minimal lag to signals as high as 5 Hertz, and a device weight of no more than 10 ounces (0.283 Kilograms). These criteria apply to an operating travel length of 3 inches (76.2 millimeters).

Another application of a linear magnetic particle brake might be in long-leg braces. The rotary brake assembly used on the A/K prosthesis is installed in the center of the lower leg of the prosthesis. Because a long-leg brace has a human leg inside it, this type of installation is not possible. The bulkiness of the rotary brake damper is the limiting factor prohibiting its use on the exterior of the brace. A compact linear design would permit the development of an active long-leg brace whose damping characteristics could be controlled in a manner similar to the microprocessor-controlled A/K prosthesis. Another application of a linear brake would involve the combination of a linear brake, a position transducer, and a control circuit. These components could be packaged as an electric damper whose performance could be programmed to achieve a wide variety of operating characteristics, such as viscous

or coulomb damping, from a single piece of hardware.

The thesis begins with a discussion of the operating mechanism of rotary magnetic particle brakes and the presentation of linearized rotary brake formulae. Then the design and construction of the prototype is discussed. Next, the methods and results of the performance tests are presented. The thesis concludes with a discussion of a second-generation design, the criteria established for the design of linear brakes, and the feasibility of a linear brake for use in prosthetic applications.

The prototype linear brake constructed for this thesis functioned successfully. The signal to noise ratio, defined as the maximum brake output obtained versus the zero current drag, ran from 3:1 to 4:1. The brake exhibited a region of linear output with the smooth, repeatable operation, independent of translation velocity, that is characteristic of commercially available rotary magnetic particle brakes.

I - THEORY OF OPERATION

A. BACKGROUND

A rotary magnetic particle brake is essentially a magnetic particle clutch with one member held stationary. The development of the magnetic particle clutch is credited to Jacob Rabinow of the National Bureau of Standards in 1948.(1) Rabinow's clutch was called a magnetic fluid clutch because he used an oil-particle mixture as the medium to fill the gap between the rotating members. The clutch was developed for use in servo systems because it exhibited linear proportionality to the current supplied and smooth operation; useful attributes for control purposes.

A problem with the wet clutch was that the rotating seals used in the clutches deteriorated from contact with the oil-particle mixture, which was effectively an abrasive slurry. During the 1950's, Lear Incorporated (Grand Rapids, Michigan) produced a commercial clutch that used a dry powder to circumvent the seal problem.(2)

The consistency of operation of dry-powder clutches is dependent upon the physical properties of the powder.

(1) - Rabinow, Jacob - National Bureau of Standards Technical News bulletin, Vol 34, No 5, "Magnetic Fluid Clutch", May 1948

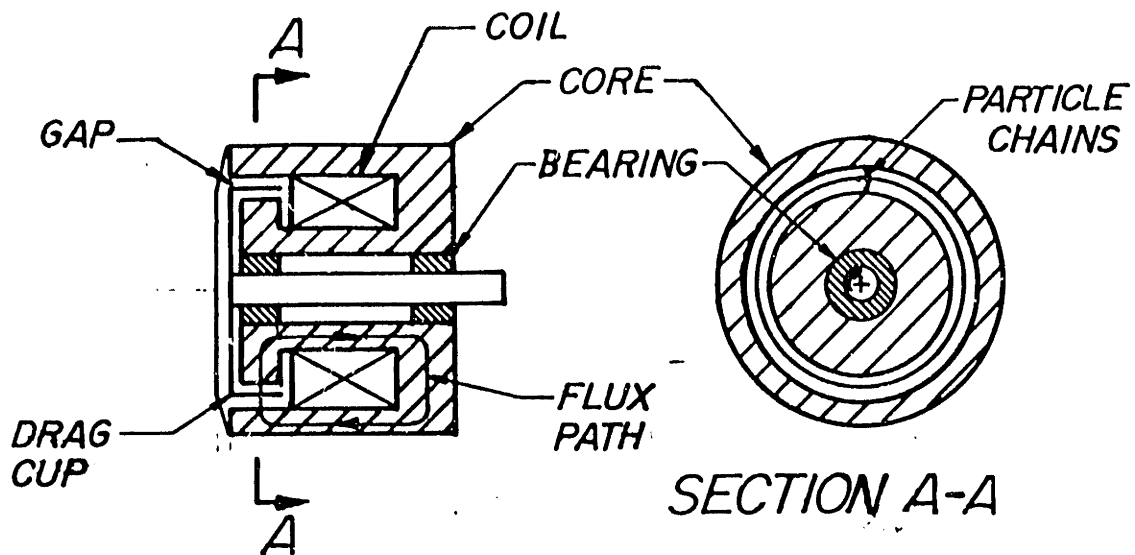
(2) - Blackburn, Elisha Philips - "A mathematical Model for a Dry Powder Magnetic Clutch", Master's Thesis, University of California, June 1961.

Oxidation destroys the magnetic properties of the powder. Some lubrication is necessary to prevent bunching and to minimize the zero-current drag. Magnetic hysteresis must be minimized in order to maintain repeatability and output precision. These problems were solved in a proprietary development by Force Limited (Santa Monica, California) for a magnetic particle brake powder.(3)

B. OPERATING MECHANISM

A sectioned drawing of a typical rotary magnetic particle brake is shown in Figure 1. Note that the flux lines cross the gap perpendicular to the surface of the driven member. When the brake is activated, the particles in the gap align themselves with the flux lines to form chains. The chains are attracted to each other along the flux lines and bear down upon the surface of the driven member. The kinetic coefficient of friction between the particles and the driven member determines the drag force on the driven member. Assuming that the kinetic coefficient of friction is constant, the drag force on the driven member will be independent of the velocity of that member. During operation the particle chains will incline at a friction angle that is a function of the direction of operation, the coefficient of friction, and the intensity of the magnetic field.

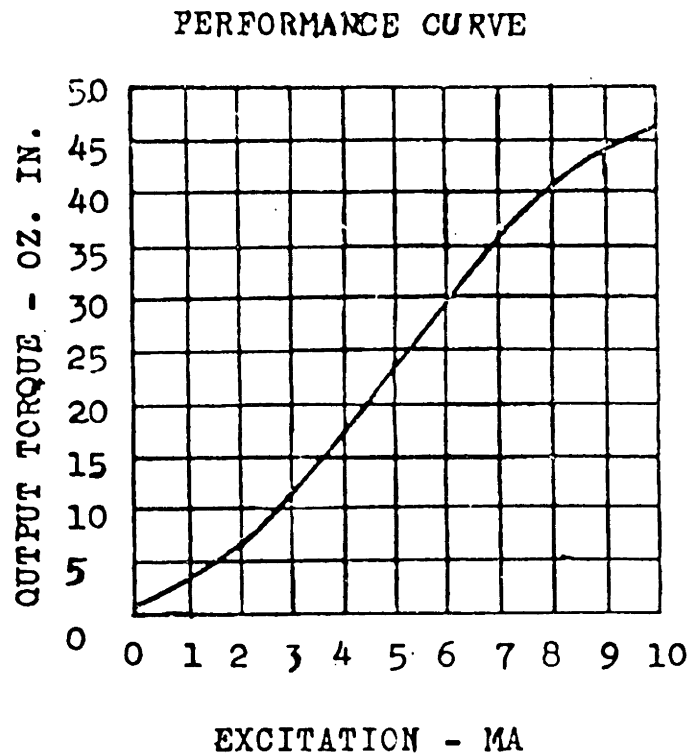
(3) - Force Limited - Catalog F - 69, "Principle of Operation of Force Magnetic Particle Clutches".



*FIGURE 1 - SECTION OF A TYPICAL ROTARY
MAGNETIC PARTICLE BRAKE*

The basic force equation for magnetic materials relates force as a function of a constant, the area normal to the flux path, and the square of the flux density. Flux density is related directly to the product of the magnetizing force and the permeability of the material. The magnetizing force is given by the product of the coil current, the number of turns in the coil, and the reciprocal of the path length. This would imply that the relation describing force output to input current should follow a square law. Magnetic particle brakes have fairly linear operating characteristics. The reason for the linear performance of magnetic particle brakes is that non-linearities in the powder and the steel in the magnetic circuit cancel the square law effect. The performance curves of rotary

magnetic particle brakes show regions of non-linearity at the null point, near zero current, and at the saturation point of the magnetic circuit, with a large linear region connecting the two. A typical performance curve for a commercially available magnetic particle clutch is shown in Figure 2.(4)



*FIGURE 2 - TYPICAL PERFORMANCE CURVE FOR
A COMMERCIALY AVAILABLE
MAGNETIC PARTICLE CLUTCH*

C. FORCE EQUATIONS

(4) - Ibid.

The equations used in determining the performance of magnetic particle brakes are typically empirical relations. An approximation for the maximum output of a rotary brake at saturation is given by:(5)

$$T = D \times A \times 20$$

T = torque in inch-pounds

D = drag cup diameter in inches

A = drag cup area in square inches

Linearizing this by dividing by the radius to find the force yields:

$$F = A \times 40 \quad (1)$$

F = force in pounds

A second equation that defines torque versus flux:(6)

$$T = 5.59 \text{ E8} \times (f \times Kq \times R / A) \times \text{phisq}$$

f = coefficient of kinetic friction between powder particles (nominal value = 0.26)

A = area of powder gap normal to the flux in square inches

Kq = ratio of actual area occupied by particles to available area normal to the flux path (nominal value = 0.80) (Kq approaches unity for a finely divided powder of sufficient quantity to fill the working portion of the gap)

(5) - Adkins, Paul - Chief Engineer, Force Limited - Personal Communication, April 2,1981.

(6) - Conductron Corporation, Grand Rapids, Michigan - "Application Criteria for Magnetic Particle Clutch Servos and Systems".

phisq = square of the total flux through the
gap in webers

R = radius to the shear surface in inches

Linearized, this becomes:

$$F = 5.59 \text{ E8} \times (f \times Kq / A) \times \text{phisq} \quad (2)$$

Another approximation was found using an analysis that considered the force as a result of the driven member being clamped between the poles of a magnet.

$$F = 2 \times F_n \times f$$

F_n = normal force

$$F_n = K \times A \times B_{sq}$$

B_{sq} = square of the flux density (lines per
square inch)

For force in pounds, flux density in lines per square inch, and area in square inches:

$$K = 1.3864 \text{ E-8}$$

so

$$F = 2.7728 \text{ E-8} \times (f \times A) \times B_{sq}$$

but

$$B_{sq} = \text{phisq} / (A \times A)$$

and

$$1 \times \text{E8} \text{ lines} = 1 \text{ weber}$$

so

$$F = 2.7728 \text{ E8} \times (f / A) \times \text{phisq} \quad (3)$$

Equations (1), (2), and (3) were used to calculate the performance of the linear brake.

II - DESIGN

A. APPROACH

Factors influencing the design of the linear magnetic particle brake included: the design should be for a true linear brake, rather than a redesign of the existing device using linear-rotary conversion; the flux lines through the operating gap must be perpendicular to the desired linear motion; and the geometry of the magnetic circuit should be such that the field can be easily calculated with minimal fringing losses.

The three-pole configuration, shown in Figure 3, was chosen because with a narrow gap, this geometry would minimize fringing and produce flux lines perpendicular to the plane of the gap. Another reason for choosing this geometry was that it could be easily constructed by modifying an existing three-pole transformer.

This configuration was tested using two permanent magnets glued together with Eastman 910, 100 mesh iron powder, powder removed from a rotary brake, and a brass slider. This is shown in Figure 4. A steel slider was not used because it tended to snap off center towards one of the poles while the slider was pushed or pulled. The brass slider did not have this problem and so it required minimal guidance to maintain its position relative to the poles. The brake had a gap of 0.015 inches (0.381 millimeters) on each

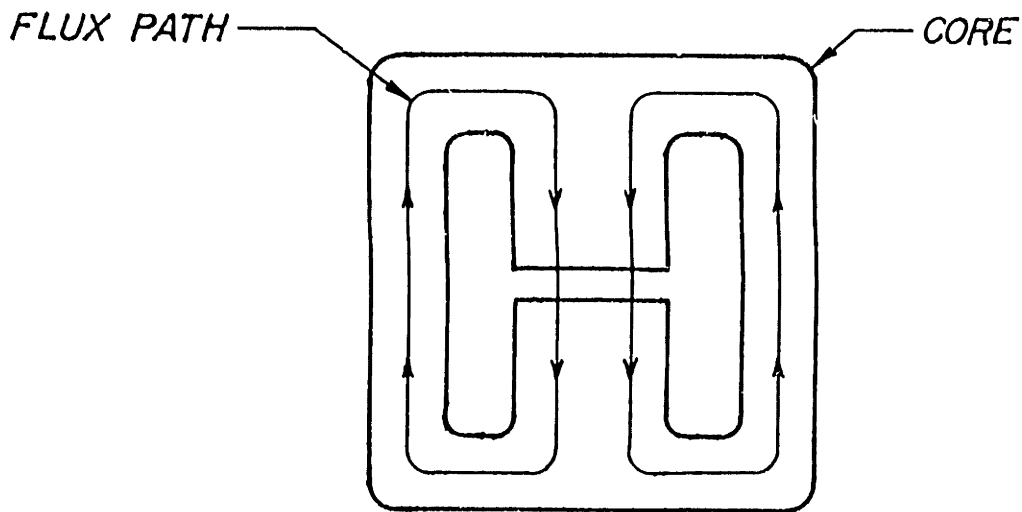


FIGURE 3-THREE POLE CONFIGURATION

side of of the slider. The brake functioned, developing approximately 0.5 pounds (2 Newtons) of force. The slider moved smoothly with no perceptible signs of stick-slip phenomena. The brake developed less force with the 100 mesh particles, which were visibly coarser than the particles from the rotary brake.

The success of the permanent magnet linear brake encouraged the pursuit of the three-pole configuration. Structurally stable guiding and sealing hardware were required of the prototype. Anything that would be near the operating gap should be made of non-magnetic materials

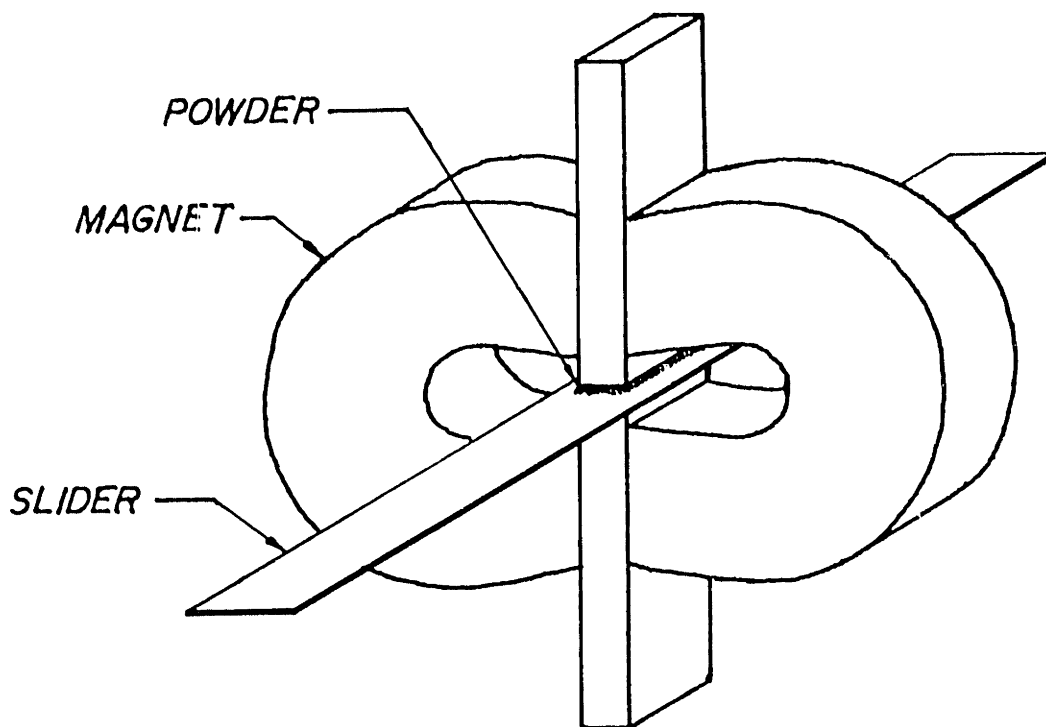


FIGURE 4 - PERMANENT MAGNET LINEAR BRAKE

so that the path of least reluctance would be through the gap, so the support hardware was made of aluminum. It was decided to use two different slider materials: brass and steel. The brass slider would be useful in determining if particle bunching is a problem since the particles could be easily scraped off the slider by the seals. It would also provide a shifted data curve because of its non-magnetic properties. The steel slider would reduce the reluctance of the gap and thus produce the same forces as the brass slider but at significantly lower currents. This assumes that the

coefficients of friction between the particles and the sliders are comparable. Since the kinetic coefficients of friction for iron on brass (0.30) and iron on mild steel (0.23) bracket the nominal value for Equation (2), this assumption should be valid.

Another consideration affecting the design of the brake was that the brake and its supporting jig should interface with standard INSTRON gripping fixtures to facilitate the performance testing.

B. ANALYSIS

Application of Equation (1) yielded a maximum output at saturation of 22.5 pounds (90.7 Newtons).

An input current of 150 milliamperes was used as the basis for the flux input to Equations (2) and (3). Details of the solution of the magnetic circuit are compiled in the Appendix. The number of turns in the coils was found by cutting open a transformer and counting the windings. The estimated error of the counting operation was 5 per cent. This error carries through the equations leaving an error of 10 per cent in the final values.

One difficulty in performing the flux calculation was that a permeability curve for the powder was not available. The initial calculation treated the permeability of the gap equal to the permeability of the air. Equation (2) yielded 12.55 pounds (50.58 Newtons) and Equation (3) yielded 7.8

pounds (31.43 Newtons). The field intensity at this operating point was calculated at 43 800 lines per square inch (0.665 webers per square meter). Correcting for the steel slider resulted in a field intensity of 61 700 lines per square inch (0.955 webers per square meter), the result of Equation (2) equal to 24.90 pounds (100.35 Newtons), and the result of Equation (3) equal to 15.43 pounds (62.18 Newtons). Rabinow measured the permeability of his oil-powder mixture as eight times that of air. This would saturate the brake with the steel slider installed at field intensity of 90 300 lines per square inch (1.40 webers per square meter) with Equation (2) yielding 53.33 pounds (214.92 Newtons) and Equation (3) yielding 33.04 pounds (133.15).

The large disagreement of the saturated outputs of Equation (1) and Equations (2) and (3) suggest that the permeability of the powder at that operating point is less than eight times that of air.

C. CONSTRUCTION

An assembly drawing of the prototype is shown in Figure 5.

The magnetic core of the brake was made by machining two identical transformer cores such that the cores interlocked. The 0.036 inch (0.914 millimeter) gap was produced by milling off more material from the center pole

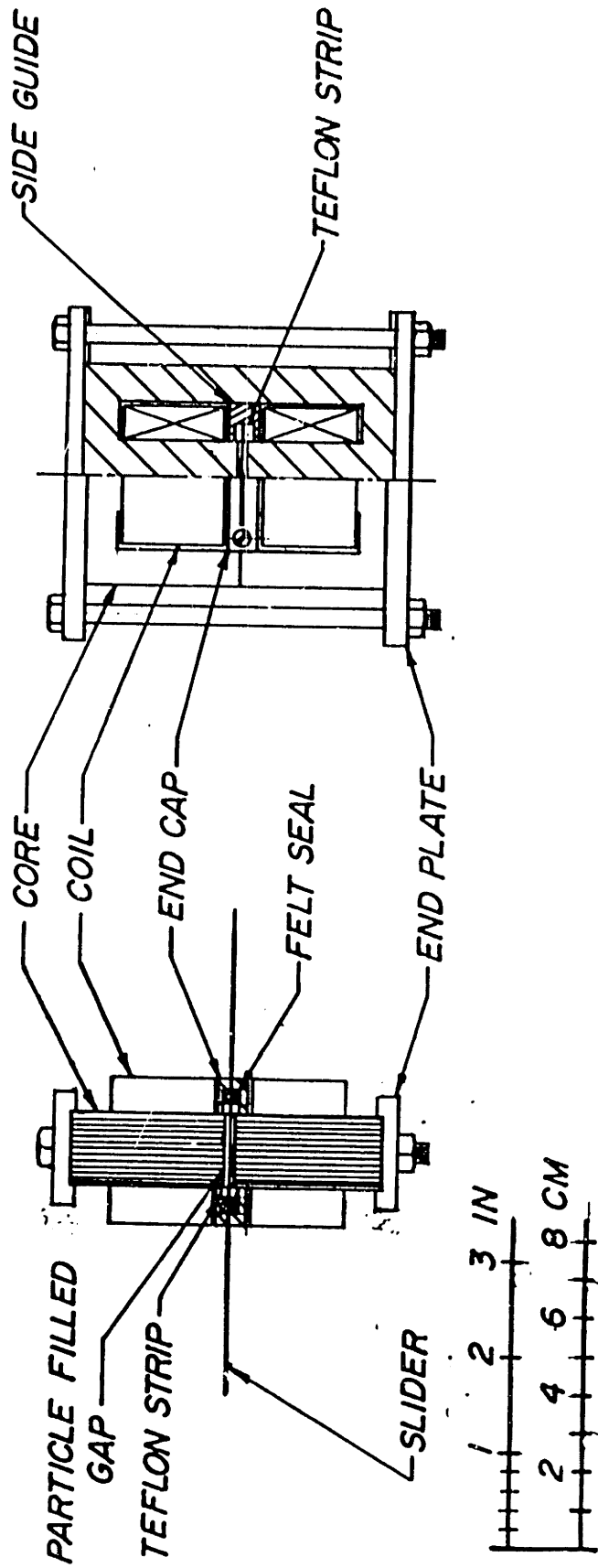


FIGURE 5 - SECTIONED ASSEMBLY OF PROTOTYPE LINEAR BRAKE

of each section. Delamination of the cores during machining was a problem. Clamping and cutting such that the cutting force tended to push the laminations together solved the problem. A third transformer was cut apart to count the core windings. The primary and secondary of each transformer was wired in series so that the current flowing into each set of windings would be defined so that the number of ampere-turns could be calculated. The two cores were clamped together using aluminum endplates and two 4 inch long (101.6 millimeter), 5/16 inch bolts.

The side guides for the slider were machined from 6061 aluminum. The guides were lined with teflon strips to allow smooth operation of the slider. The end caps were also made of aluminum. The end caps sealed in the powder by holding a felt seal against the slider and the pole pieces. Two teflon strips, one on each side of the slider, supported the felt on the pole side to prevent the felt from extruding into the gap. The endcaps were attached to the guides using 3-48 brass screws, clamping the entire assembly around the poles.

The INSTRON support jig was made from a 6.5 inch long (165.1 millimeter) piece of 4 inch (101.6 millimeter) I-beam. Aluminum end plates clamped the brake to the I-beam with .5 inch (12.7 millimeter), 1/4-28 inch bolts.

The steel slider measured 0.0118 inches thick (0.2997 millimeters) after polishing. The brass slider was 0.016 inches thick (0.4064 millimeters). Both sliders were 1 inch wide (25.4 millimeters) and 6.5 inches long (165.1 millimeters). The first time the brake was assembled, 100 mesh powder was used in the gap. A finer magnetic particle brake powder was obtained from Force Limited (Santa Monica, California) and subsequently replaced the 100 mesh powder.

The assembled brake without the supporting jig weighed 2 pounds (0.907 Kilograms). Photographs of the assembled brake and jig are shown in Figure 6.

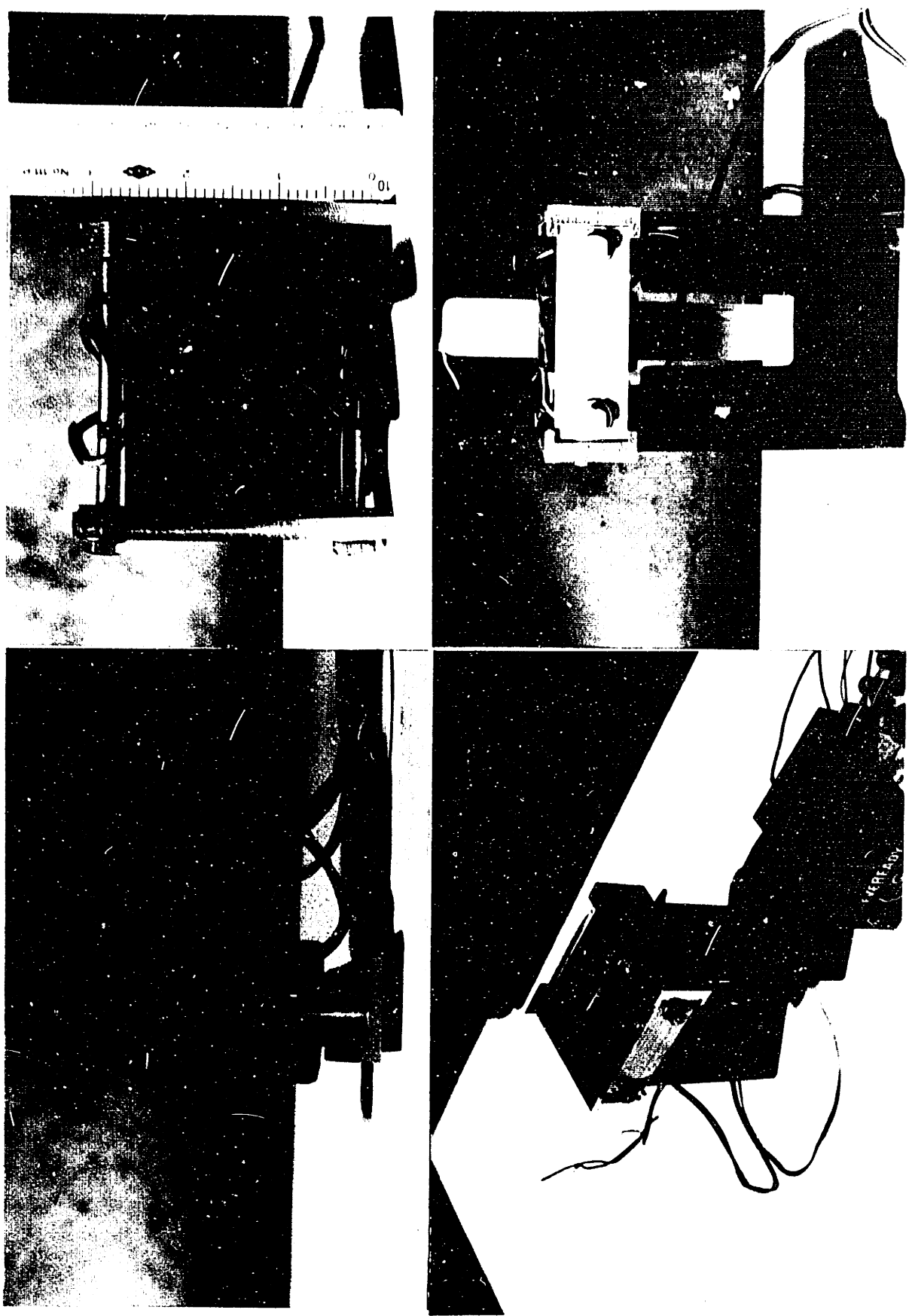
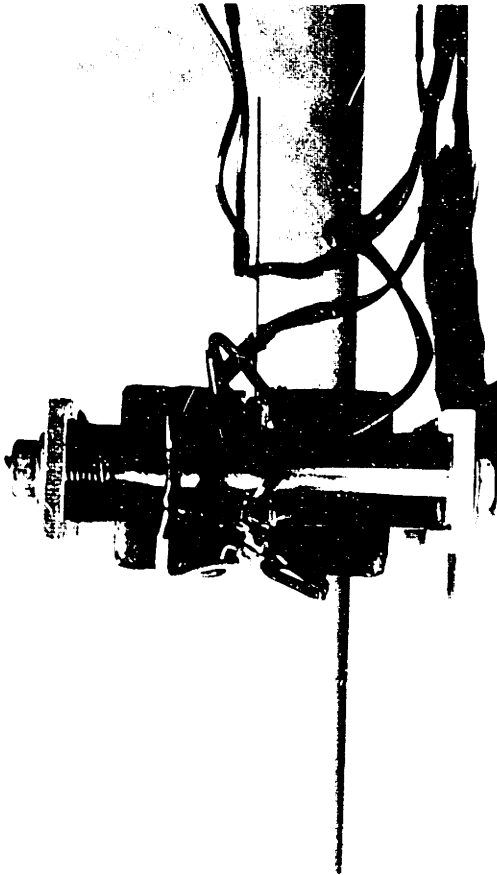
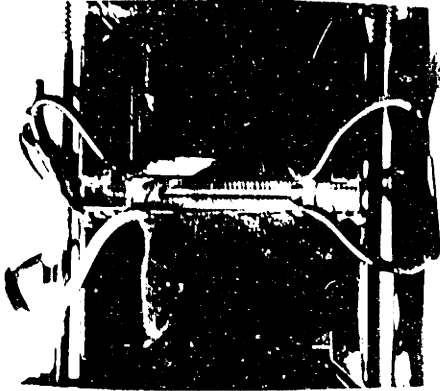
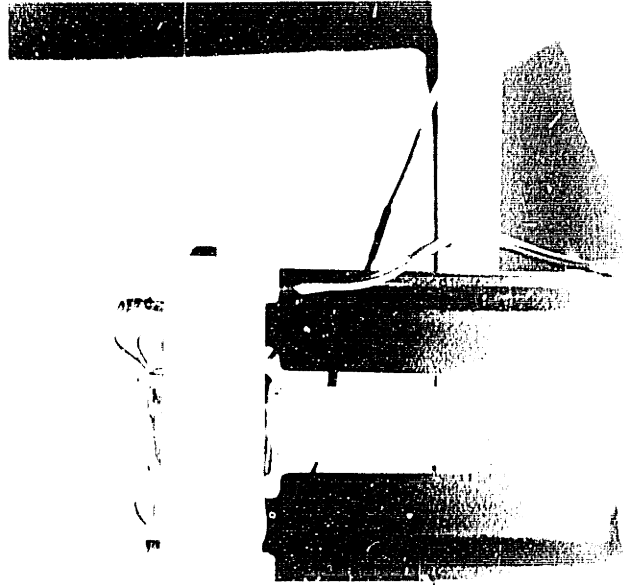


FIGURE 6 - PHOTOGRAPHS OF PROTOTYPE



III - EXPERIMENTAL PROCEDURE

A. SET-UP DESCRIPTION

The brake and jig assembly was mounted in an INSTRON tensile test machine using the flat specimen jaws. A photo of the test apparatus in action is shown in Figure 7.

The brake coil resistance was measured at 82 Ohms. The coils were wired in parallel because this would double the current through each coil in a series wiring.

The power supply initially consisted of four 9 volt batteries wired to provide an 18 volt output. The current to the brake was varied using a 1000 Ohm potentiometer. Problems with the resolution of the potentiometer led to its replacement with eight fixed-value resistors ranging from 5.1 to 390 Ohms. The 9 volt batteries were replaced with two 22.5 volt batteries because the power demand resulted in a substantial voltage drop of the power supply.

The coil current was measured by measuring the voltage across a 1 Ohm (measured at 1.062 Ohms) resistor that was wired in series with the power supply. These voltages were measured on a recording trace oscilloscope to observe the dynamic response of the coil. A digital voltmeter was used to measure the steady state voltages.

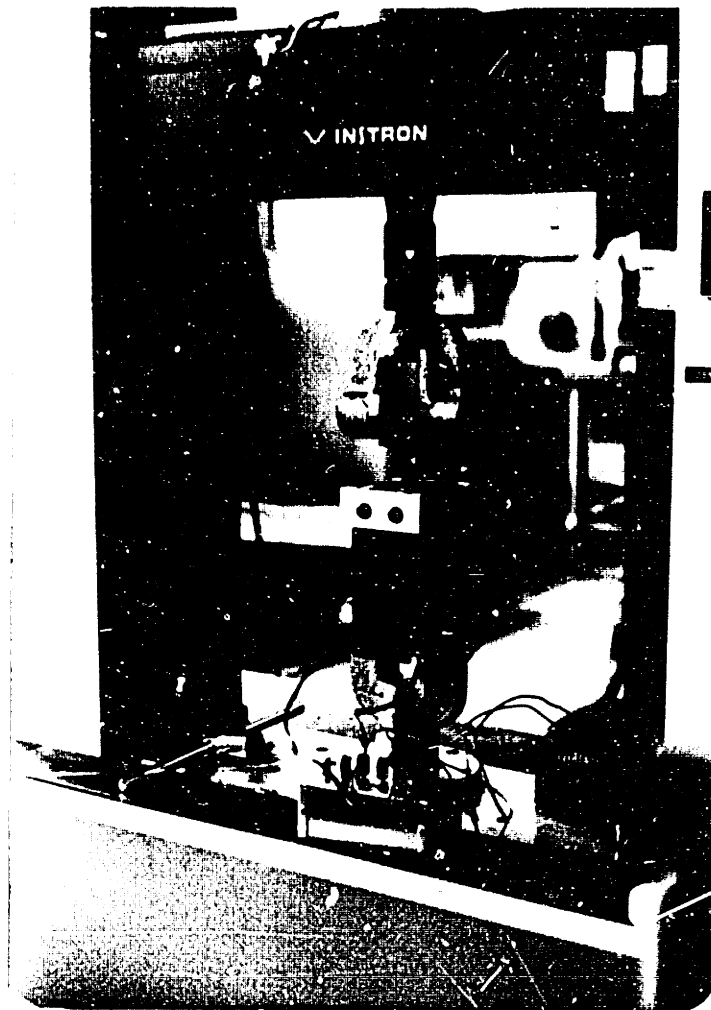


FIGURE 7 - BRAKE DURING PERFORMANCE TEST

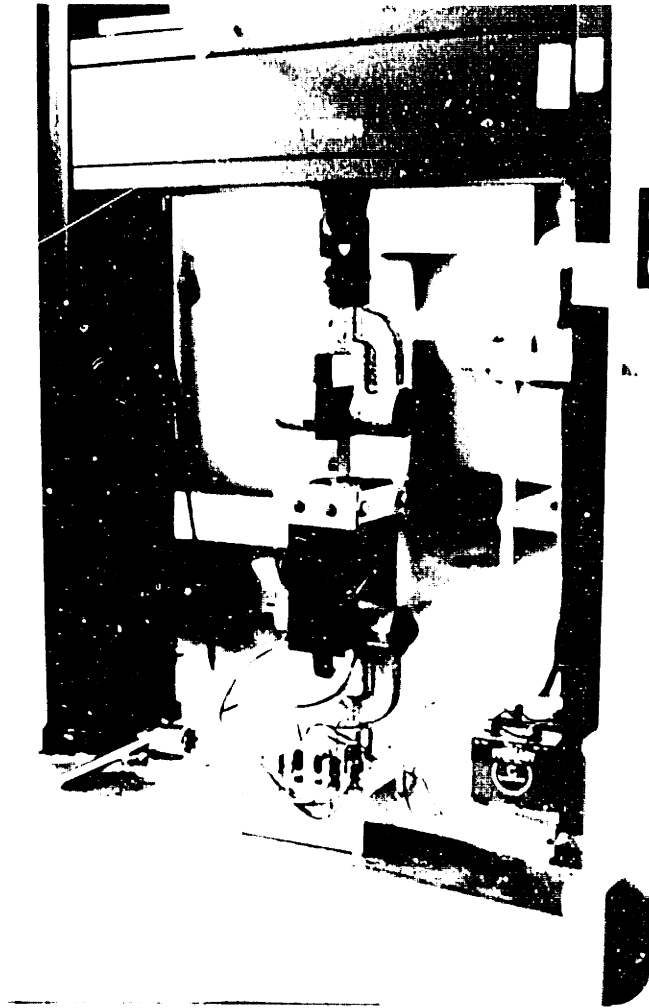


FIGURE 7 - BRAKE DURING PERFORMANCE TEST

B. OPERATING PROCEDURES

Data was taken by starting the INSTRON, triggering the oscilloscope, and then switching on the power supply. The resistors used to adjust the supply voltages were only rated for half watt operation. During continuous operation a resistor would rapidly heat up, changing its resistance and the current supplied to the brake. Because of this problem, data was taken as a series of blips, typically 2 to 5 seconds in duration. A temperature rise detected by finger contact was used as the criterion to determine the cycling time.

The performance tests were run using a crosshead speed of 5 inches per minute (2.18 millimeters per second). The crosshead speed was also varied from 2 to 20 inches per minute (0.847 to 8.47 millimeters per second) on several runs to observe the effects of slider velocity on the output force. The force output data was plotted on the INSTRON chart recorder. Transient response data had to be measured on the 20 and 50 pound scales (80.6 and 201.5 Newtons) because the slew rate of the plotter could only keep up with half-scale deflections.

IV - RESULTS AND DISCUSSION

A. TEST RESULTS

Two sets of performance tests were run using the steel slider; one with the 100 mesh powder and one with the particle brake powder. One test series was run on the brass slider using the particle brake powder.

The data consisted of a series of plots of force versus chart speed from the INSTRON chart recorder and a corresponding voltage measurement. A typical raw data plot is shown in Figure 8.

In the first test with the steel slider and the 100 mesh particles, the zero current drag varied as a ramp from 2.5 to 4.8 pounds (10.1 to 19.3 Newtons) over the 3 inch travel (76.2 millimeter) of the slider. For the second test with the magnetic particle powder, the steel slider was polished before installation. The felt seal was also packed tighter because some powder leakage had been observed during the first test. The zero current drag during the second test remained constant at 2.7 pounds (10.9 Newtons). The zero current drag for the brass slider installation was also constant at 2.2 pounds (8.9 Newtons).

The steady state force output data is plotted in Figure 9. The seal drag is subtracted from each point. The current is the equivalent current for the two coils wired in

CHART SCALE - 5 POUNDS PER INCH

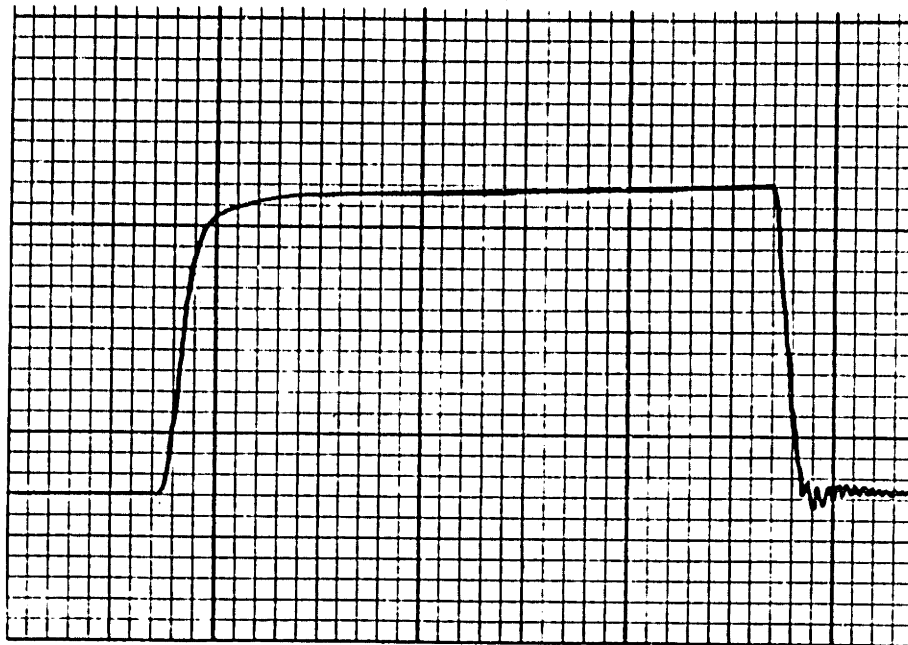


CHART SPEED - 100 INCHES PER MINUTE

FIGURE 8-TYPICAL RAW DATA (FULL SIZE)

series. The performance curves are fitted using a least squares linear regression. The data for the steel slider with the 100 mesh powder has a correlation coefficient of .9764. The first portion of the data for the steel slider with the magnetic particle powder has a correlation coefficient of .9974. For the second portion the correlation coefficient is .9667. The data for the brass slider has a correlation coefficient of .9898.

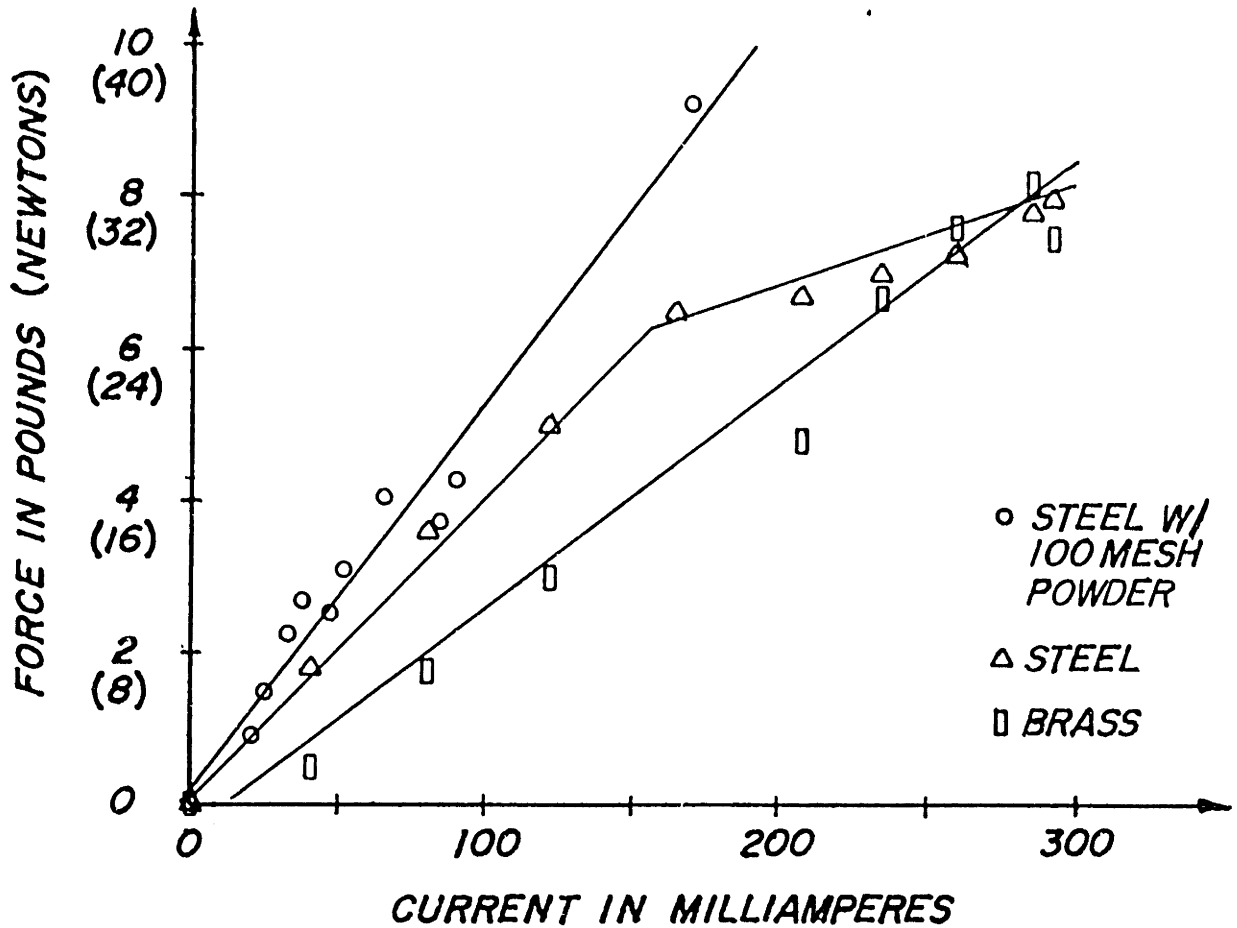


FIGURE 9 - FORCE VERSUS CURRENT DATA WITH FITTED CURVES

Initially, the steel slider was not polished. The data for the steel slider showed both the surface roughness of the steel and the large particle size as roughness in the raw data. Polishing the steel and using the finer magnetic particle powder alleviated this problem, with the brake exhibiting smooth operation. The brass slider had a good surface finish and did not exhibit this problem.

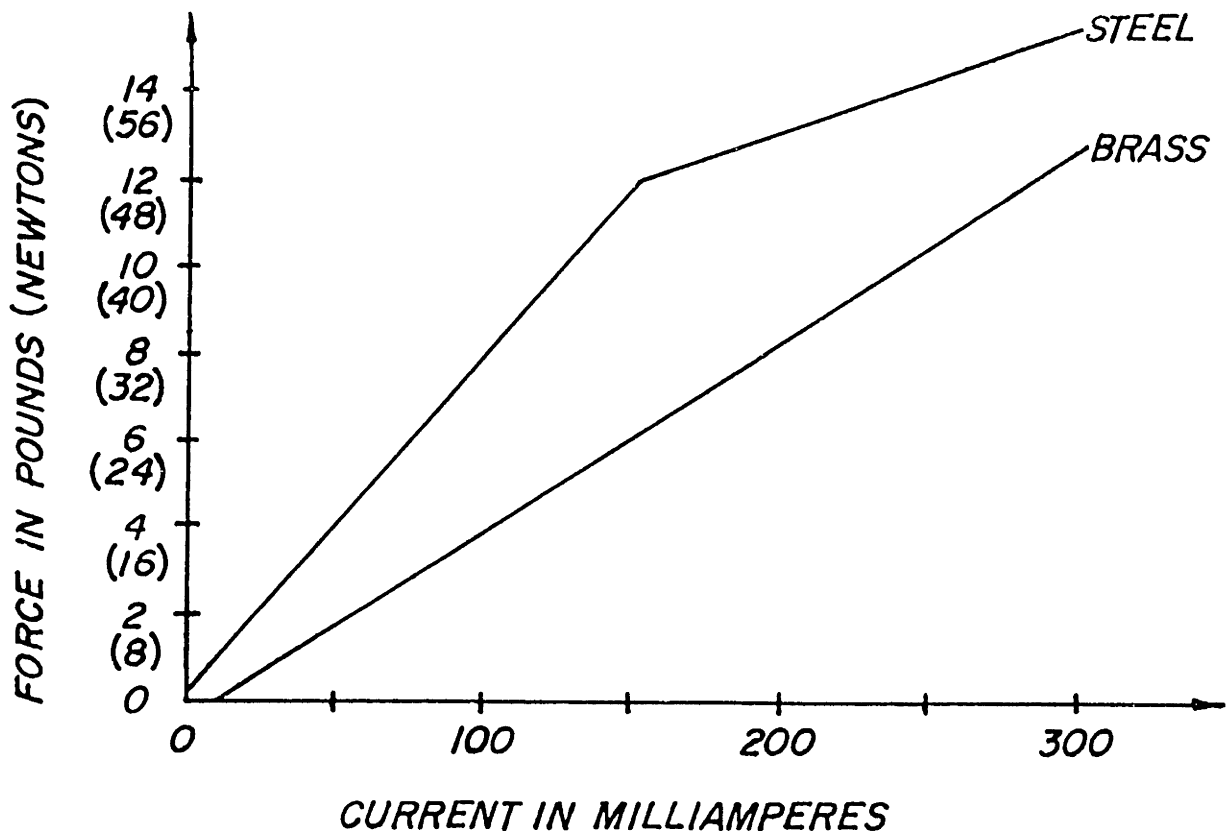
The force output showed no perceptable changes for crosshead velocities varying from 2 to 20 inches per minute (0.847 to 8.47 millimeters per second). At 20 inches per minute (8.47 millimeters per second) some vibration appeared on the output, with an estimated magnitude of 0.2 pounds (0.806 Newtons). Several tests were run with the INSTRON in cycling mode. The force showed a momentary increase of 15 per cent of the total output at the turnaround point before returning to the value of the previous direction.

The transient response data over all the tests gave a time constant averaging 121.6 milliseconds with a standard deviation of 27.5 milliseconds. The transient response for the coil currents gave time constants varying from 5 to 48.5 milliseconds, depending upon the load resistance in the circuit.

B. ANALYSIS AND DISCUSSION

An implication of Equation (2) is that the degree of packing of the powder in the gap plays a role in the net output of a magnetic particle brake. By weighing the powder removed from the gap and dividing by the volume of the gap, the particle density was calculated for the tests with the magnetic particle powder. The optimal powder density by measured by filling a 5 milliliter container (5.06 ml, tare 10.76 gms), tapping it to settle the powder, refilling it, and then weighing the container. This gave a density of 3.6561 grams per cubic centimeter (0.132 pounds per cubic

inch). The amount of powder recovered from the gap yielded a density of 2.2763 grams per cubic centimeter (0.082 pounds per cubic inch) for the brass slider installation and 1.8943 grams per cubic centimeter (0.068 pounds per cubic inch) for the steel slider installation. A force correction factor was calculated by dividing the optimal powder density by the actual powder density. The correction factor for the steel slider data was 1.93 and the correction factor for the brass slider was 1.60. The shifted performance curves are shown in Figure 10.



**FIGURE 10-BRAKE PERFORMANCE WITH
POWDER CORRECTION FACTOR**

The permeability of the powder was backcalculated by examining the current differential between the steel slider and brass slider performance curves. The calculation assumed that the "knee" in the steel slider data corresponded to the knee in the permeability curve of the transformer core steel. Equating the flux densities gave the permeability of the powder to be 1.67 times that of air at this operating point. Details of the calculation are in the Appendix.

A calculation of the force output based on this permeability yields 33.86 pounds (136.46 Newtons) from Equation (2) and 16.78 pounds (67.62 Newtons) from Equation (3). The corrected output of the prototype at this operating point was 11.90 pounds (47.96 Newtons).

The inductance of the coils was backcalculated from the time constants of the transient data for the coil current. The inductance was calculated at 4 Henries per coil. This yields a time constant of 48.5 milliseconds for the brake current.

V - CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The response of the brake was substantially less than the output calculated by Equation (2). Equation (3) gave a better correlation. However, without a permeability curve for the powder, it is impossible to conclusively state the validity of Equation (2) or (3) for this application. The trend of the data suggests a good correlation with Equation (1) for the maximum force. The brake response did exhibit linear performance over a very definite operating range.

The brake was never totally saturated during operation; at saturation, the data should show a nearly constant force output for increasing currents. It is possible that inaccuracies in the coil counting and added reluctances for the core joints gave lower fluxes than those calculated for the equations.

It is unlikely that this brake could meet the performance specifications for the prosthetic application presented in the introduction. The necessary increase in gap area to bring the force up to the desired levels would require so much extra material in the magnetic circuit that it would be impossible to come close to the desired weight. Using a stack of sliders within the gap would increase the area but the increase in the input current would be prohibitive. The number of turns in the coil also needs to

be reduced to improve the transient response. This too would require an increase in current to generate the same magnetizing forces.

A permeability curve for the powder is necessary before any further analytical work can be undertaken. A regulated power supply would be useful in studying the brake output over periods longer than a few seconds. This would also permit an investigation of the energy dissipation rate.

B. NEXT GENERATION DESIGN

A second generation design is shown in Figure 11. This design should be able to develop a saturation output of 94 pounds (379 Newtons), based on Equation (1). The projected weight of the brake is 31 ounces (0.879 Kilograms). The operating travel length is 2.5 inches (63.5 millimeters). By using eight separate windings, the transient response should improve because each coil will require fewer turns to produce the necessary magnetizing force, which is summed around the circuit.

Advantages of this design are: the sliding member has a thick wall and can withstand tensile and compressive loads; this configuration minimizes the amount of connecting material in the magnetic circuit; the circular sliding member should be easy to seal with a bearing seal; the circular geometry is easier to make because it is easier to meet radial tolerances than linear tolerances; the

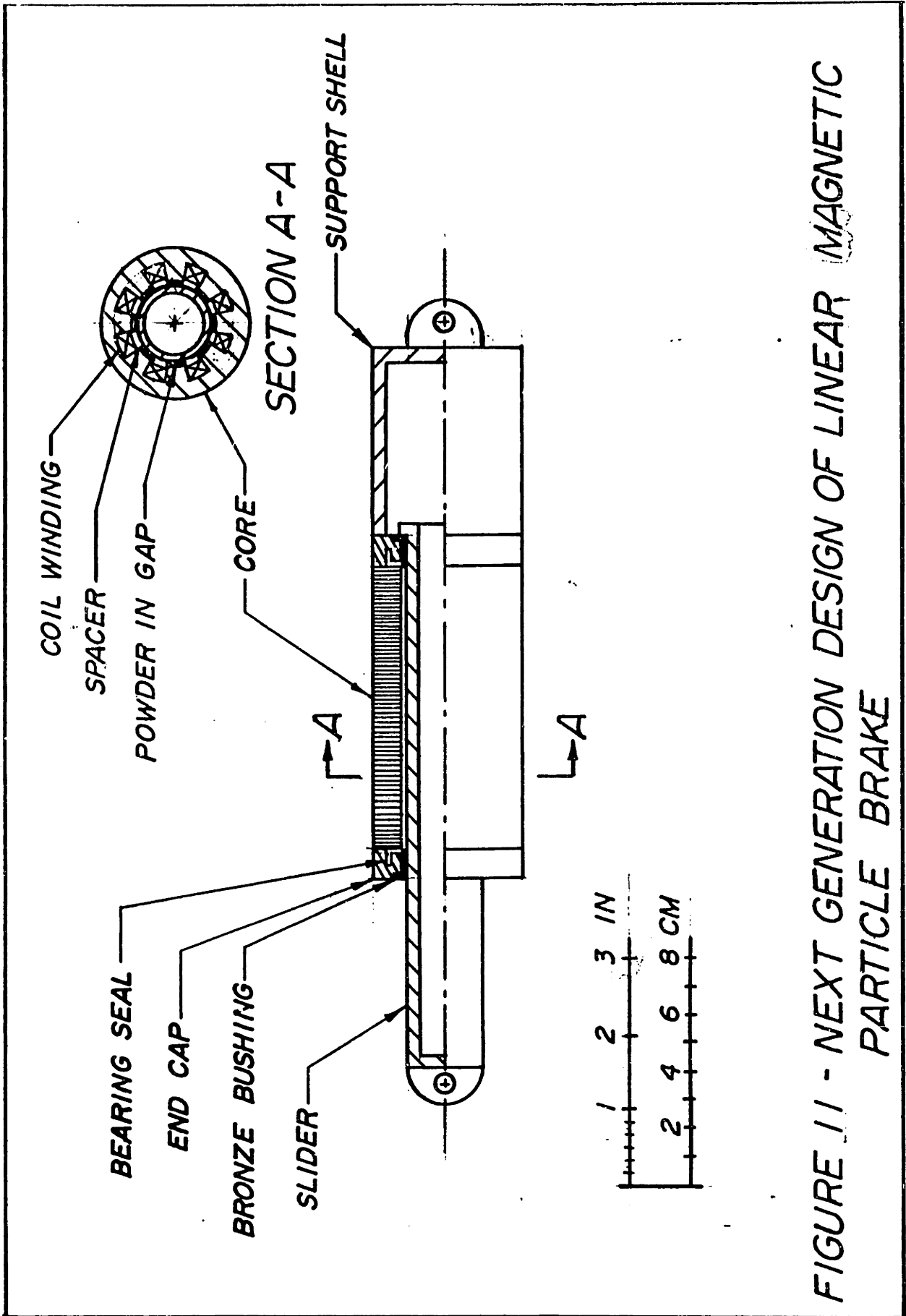


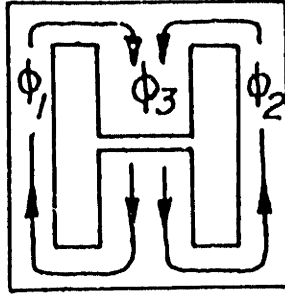
FIGURE 11 - NEXT GENERATION DESIGN OF LINEAR MAGNETIC PARTICLE BRAKE

circular geometry should facilitate assembly and powder insertion; and the device will look and interact in a manner similar to a conventional shock absorber.

This design operates by using the slider as the return path for the magnetic circuit. Aluminum spacers keep the powder in front of the poles and assure that the path of least reluctance is through the gap. The direction of current flow through the coils is alternated so that the flux lines weave back and forth across the gap.

The construction and evaluation of this next generation design is the next step in developing a useful linear brake. This design has the advantage that it can be tested in practical use situations whereas the prototype was primarily designed to test the principles of operation.

A. FLUX CALCULATION



The sum of the reluctances is:

$$R = R_g + R_3 + \frac{R_1 R_2}{R_1 + R_2}$$

Where the subscripts refer to each segment of the magnetic circuit as shown in the above diagram and g is the subscript for the gap.

The total magnetomotive force is:

$$F = 2NI$$

Where N is the number of turns in each coil and I is the current flowing through them.

The flux is simply:

$$\Phi = \frac{F}{R}$$

So for this circuit:

$$\Phi = \frac{2NI}{R_g + R_3 + \frac{R_1 R_2}{R_1 + R_2}}$$

Reluctance is given by:

$$R = \frac{l}{\mu A}$$

Where A is the area, l is the path length, and μ is the permeability.

Paths 1 and 2 have half the area of path 3 and equal reluctances. This makes the denominators of the R_1R_2 and the R_3 terms equal, so:

$$\phi = \frac{2NI}{\frac{l_g}{\mu_g A} + \frac{l_s}{\mu_s A}}$$

Where l_s is the length of the steel in one loop of the circuit.

Since $B = \frac{\phi}{A}$

$$B = \frac{2NI}{\frac{l_g}{\mu_g} + \frac{l_s}{\mu_s}}$$

For the brass slider:

$$B = \frac{2NI}{\frac{l_g}{\mu_g} + \frac{t_b}{\mu_b} + \frac{l_s}{\mu_s}}$$

And for the steel slider:

$$B = \frac{2NI}{\frac{l_g}{\mu_g} + \frac{(t_s + l_s)}{\mu_s}}$$

Where t is the slider thickness and the subscripts s and b refer to the steel and brass sliders.

B. BACKCALCULATION OF GAP PERMEABILITY

By equating the two flux equations for the brass and steel sliders and rearranging:

$$\mu_g = \frac{\mu_s \mu_b (I_b l_{gs} - I_s l_{gb})}{\mu_b (I_s l_s - I_b (t_s + l_s)) + \mu_s t_b I_s}$$

In performing this calculation it is assumed that equal braking forces are the result of equal flux densities.

C. REFERENCES

N.H. Bryant, W.H. Erickson, Electrical Engineering; Theory and Practice. (John Wiley & Sons, New York 1963)

Chapter 4 - Magnetics and Magnetic Circuits.

The permeability curve for the transformer sheet steel was taken from page 62 of the above reference.