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EDDY CURRENT DAMPER

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ABSTRACT

A high torque capacity eddy current damper has been successfully developed as a rate limiting device for a large solar array deployment mechanism. The eddy current damper eliminates the problems associated with the outgassing or leaking of damping fluids. It also provides other performance advantages, such as damping torque rates, which are truly linear with respect to input speed, continuous 360-deg operation in both directions of rotation, wide operating temperature range, and the capability of convenient adjustment of unit damping rates by the user without disassembly or special tooling.

INTRODUCTION

The eddy current damper shown in Figure 1 consists of a copper alloy disk which rotates between opposed samarium cobalt magnets. Rotation of the disk in the magnetic field generates circulating eddy currents within the disk which create a damping torque proportional to rotation speed. The damping output can be dramatically increased by coupling the eddy current disk to a gearhead speed increaser (Fig. 2). The overall damping rate is magnified by the square of the gear ratio since the gearhead acts to simultaneously increase disk speed while reducing the transmitted torque from the mechanical input of the unit. The damper design presented in this paper uses a four-stage planetary gearhead to boost the unit damping rate to 2260 N-m-sec/rad (20,000 in.-lb/rad/sec). Damping rates can be easily adjusted in the field by rotating the unit end bell, thereby misaligning magnets on either side of the eddy current disk.

The damper design also incorporates other special design features intended to minimize the size and weight of the unit and improve reliability.

EDDY CURRENT CONCEPT

Eddy current dampers have been utilized for many years in aircraft applications, but have not been fully developed for the special requirements of spacecraft application. The operating concept is relatively simple and can be compared to a generator with a shorted output. Referring to Figure 3, eight samarium cobalt permanent magnets are equally spaced on both sides of a copper alloy disk to provide a constant and uniform magnetic field. The rotation of the disk in the field produces a generated voltage in the disk. This voltage develops circulating currents within the disk which results in a

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restraining torque proportional to velocity. The drag torque created is a very linear function of the rotational speed of the copper alloy disk. As the shaft rotational speed increases, the damping torque increases.

DISK MATERIAL INVESTIGATION

An important design tradeoff in eddy current dampers is encountered in the selection of materials for the eddy current disk. Minimum damper size and weight can be achieved by using materials with low electrical resistivity. Unfortunately, those alloys which have the lowest resistivity are highly susceptible to the influences of temperature changes on damping rates.

A study of available alloys and their properties was conducted. Figure 4 summarizes the temperature versus resistivity characteristics of relevant materials. Figure 5 lists selected materials for comparison.

The initial baseline magnetic design utilized a pure copper disk. A disk thickness of 0.2 cm was necessary to meet performance requirements. This thickness dictated the minimum magnetic air gap and consequently determined the magnet size along the direction of magnetization. Within the overall envelope constraints, the use of samarium cobalt magnets allowed a larger gap and thicker disk. Total unit length and weight increased, but not in direct proportion.

Figure 4 could also be considered as a plot of temperature coefficient versus disk thickness and/or magnet length.

The material chosen was Drive Harris 30 Alloy. It offered the best combination of performance versus temperature within the size and weight constraints. Other applications may result in the selection of a different alloy depending on system requirements and mechanical size limitations. From Figure 5, the disk thickness must be 2.9 times thicker than copper (ratio of resistivity). The thickness selected for the final design was increased to allow for magnetic fringing and leakage. Magnet length was selected to operate at the maximum energy product.

The temperature sensitivity using 30 alloy was lower than copper by a factor of 2.6. This resulted in a damping rate increase of 11 percent at $-51^{\circ}C$ and a decrease of 12 percent at $104^{\circ}C$.

A less obvious advantage with using the 30 alloy was the greater thermal capacity of the thicker alloy disk. During the required duty cycle, energy absorption in the disk resulted in a temperature rise of only $5^{\circ}C$. This temperature rise was reflected in a performance change due to the temperature coefficient. With a thinner copper disk, the thermal capacity was proportionally less, with a correspondingly higher temperature rise and adverse effect on performance.

MECHANICAL

Figure 2 depicts the overall damper configuration. Easily recognized in the figure are the input shaft, four planet stages, disk assembly between opposed magnets, and structural housing parts.

Disk Region

The damping torque is effected upon the solid copper alloy eddy-current disk. The disk (the final stage of the rotary damper drive train) is mounted to a stainless steel shaft which rotates in a pair of stainless steel bearings. The inboard end of the disk shaft is machined as a pinion which serves as the sun gear to the fourth planetary stage.

One member of each magnet pair is mounted to the housing and the other member is mounted to the end bell. These samarium cobalt magnets are bonded into indentations in their respective mounting surfaces and further secured by a lightweight stainless steel cover. This method provides positive positioning and ensures that the magnets are protected from any potential sources of abrasion or chipping.

Adjustment of the unit's damping rate is achieved by purposely misaligning opposing magnets through rotation of the end bell with respect to the disk housing. Locking screws and lock wiring positively position the end bell at any of nine damping rate settings from 1130 to 2260 N-m-sec/rad. (10,000 to 20,000 in.-lb sec/rad). Lower damping rates could be easily incorporated into the design by changing the gear ratio, the disk size, or the number of magnets.

Gear Train

The design has four planetary stages which combine for a 1600:1 gear ratio. Since the damping rate of the disk is increased by the square of the gear ratio, it constitutes a damping rate multiplier of 2.56×10^6. All gears are made of 15-5 PH stainless steel heat treated to H1025 and are machined to AGMA Class 10.

The planet gears rotate on sintered bronze bearings impregnated with Bray Oil Company Type 815Z oil. To provide bearing redundancy, each planet gear rotates on the outer diameter of a sleeve bearing, and each sleeve bearing rotates on the outer diameter of a planet carrier post. In addition to the 815Z oil impregnation, the gears and bearings are lubricated with Bray 601 Micronic grease.

Ring gear teeth are machined into the stainless steel housing bore. The ring gears for stages one and two are identical in pitch diameter and diametral pitch and are machined in a single operation. Similarly, the ring gears for stages three and four are identical in pitch diameter and diametral pitch and are machined in a single operation. Machining the gear teeth directly into the housing bore precludes the potential problems of shrink fitting a ring gear into the bore of a thermally dissimilar housing.

All sun gears have at least 18 teeth so that gear tooth stresses are kept within a reasonable range. The sun gears of stages one through three (starting at the mounting end) are integral to the planet carrier of the subsequent stage. The sun gear of stage four is a pinion machined onto the end of the eddy-current disk shaft.

The input shaft transmits applied rotational motion to the first stage of planetary gearing. It is positioned and supported by a pair of stainless steel ball bearings which are located within the input housing. The input shaft is machined from 15-5 PH stainless steel.

Structural Housings

The three structural housing parts are evident in Figure 2. From left to right, the disk housing, the gear housing, and the input housing.

Since the disk housing consitutes a part of the magnetic flux path, it is machined from a 416 stainless steel. Structurally it supports one set of samarium cobalt magnets and the inboard end of the disk/shaft assembly.

The gear housing serves both as a primary structural part and as two integral ring gears. The input housing supports the input shaft ball bearings (ABEC Class 7P) and is made from 6061 aluminum for weight reduction. By controlling bearing fits and machining tolerances, the differential expansion of the input housing and the input shaft bearings is held to an acceptable level. Locking inserts in the input housing are provided for mounting.

TESTING PROBLEM

Initial tests on the first unit were erratic and produced misleading results. Variations in unit speed were noted when a constant input torque was applied. These variations had a once-per-revolution component as well as multiple cycles per revolution. Each gear stage was producing cyclic load behavior at a frequency consistent with the gear ratio. Breakaway torque was also above specification limits.

Testing to characterize the problem was conducted. As these tests progressed, it was noted that the anomalies were becoming less pronounced. This indicated a need for unit run-in prior to substantive testing. Due to the slow operating speed (0.035 rad/sec), run-in time required to stabilize unit performance was established at 8 hr. The unit was driven with an input torque of 40 N-m for 4 hr in each direction. This resulted in 167 revolutions of the input shaft.

After completion of the run-in, speed variation was negligible. Equally significant was the breakaway torque which reduced to 1.1 N-m or less. Not only were the once-per-revolution variations decreasing, gear train smoothness and friction torque was also improving. It was concluded that run-in primarily benefited the sliding contact surfaces on the planet bearings. Experience with gear trains using ball bearings and high quality gears indicates that run-in has a minor effect. The sleeve bearings, being porous, cannot be lapped or honed since the debris generated could be trapped in the pores, but run-in polishes the machined surfaces and smooths unit operation.

PERFORMANCE

The damper has the following performance and mechanical characteristics:

Continuous operation	40 N-m (350 in.lb)
Maximum torque level	80 N-m (700 in.lb)
Damping rate range (adjustable)	1130 to 2260 N-m-sec/rad (10,000 to 20,000 in.-lb-sec/rad)
Breakaway torque	1.1 N-m (10 in.-lb)
Angular range	Continuous rotation
Backlash	$<$0.50 deg
Weight	0.35 N (1.54 lb)

The damper provides a smooth resisting torque proportional to input speed. It will perform for more than 1000 cycles over a continuous rotational range with a maximum failure rate of 0.042 x 10^{-6} failures/cycle (per classical reliability analysis) under 40 N-m of applied load. It will also perform for shorter periods of time at 80 N-m. In fact, an early test unit continued to operate after 50 cycles while loaded to 120 N-m.

Breakaway torque is kept low by using lightly loaded ball bearings at the damper disk shaft. The 1.1 N-m breakaway torque value is low in view of the unit's high damping rate and input torque range. Backlash is minimized by tightly controlling bearing fits and gear dimensions, especially at the stage nearest the input shaft. Machining tolerances for these dimensions are typically held to $\pm$0.0025 to 0.0050 mm.

The damper can be field adjusted, without special tools, to any of nine settings from 1130 to 2260 N-m-sec/rad.

QUALIFICATION

The unit successfully completed qualification testing consisting of pyrotechnic shock, random vibration, and thermal-vacuum environments. Functional tests at maximum rated torque were conducted after each environment. Test levels and conditions are listed below.

1. <u>Pyroshock</u> with flat plate shock simulator;
   4 shocks, 2 in each of 2 axes.
   Ramp from 170 G at 100 Hz to 4700 G at 850 Hz.
   Then constant 4700 G to 10,000 Hz.

2. Random Vibration

Frequency (Hz)	Power Spectral Density (G^2/Hz)	(dB/Oct.)
20		
20-50	0.1	+6 to 0.64 G^2/Hz
50-200	0.64	
200-400		-6 to 0.16 G^2/Hz
400-500	0.16	
500-1000	0.16	
1000-2000		-6 to 0.04 G^2/Hz
	Overall	18.6 GRMS

3. Thermal Vacuum Cycling

Pressure 1 x 10^{-5} Torr
Temperature Cycling -51°C to 104°C for 8 cycles.

CONCLUSION

The Rotary Damper is a desirable alternative to a fluid damper, avoiding problems of contamination and limited rotation characteristics. Thermal effects are also minimized by material optimization. The unit described has successfully completed qualification.

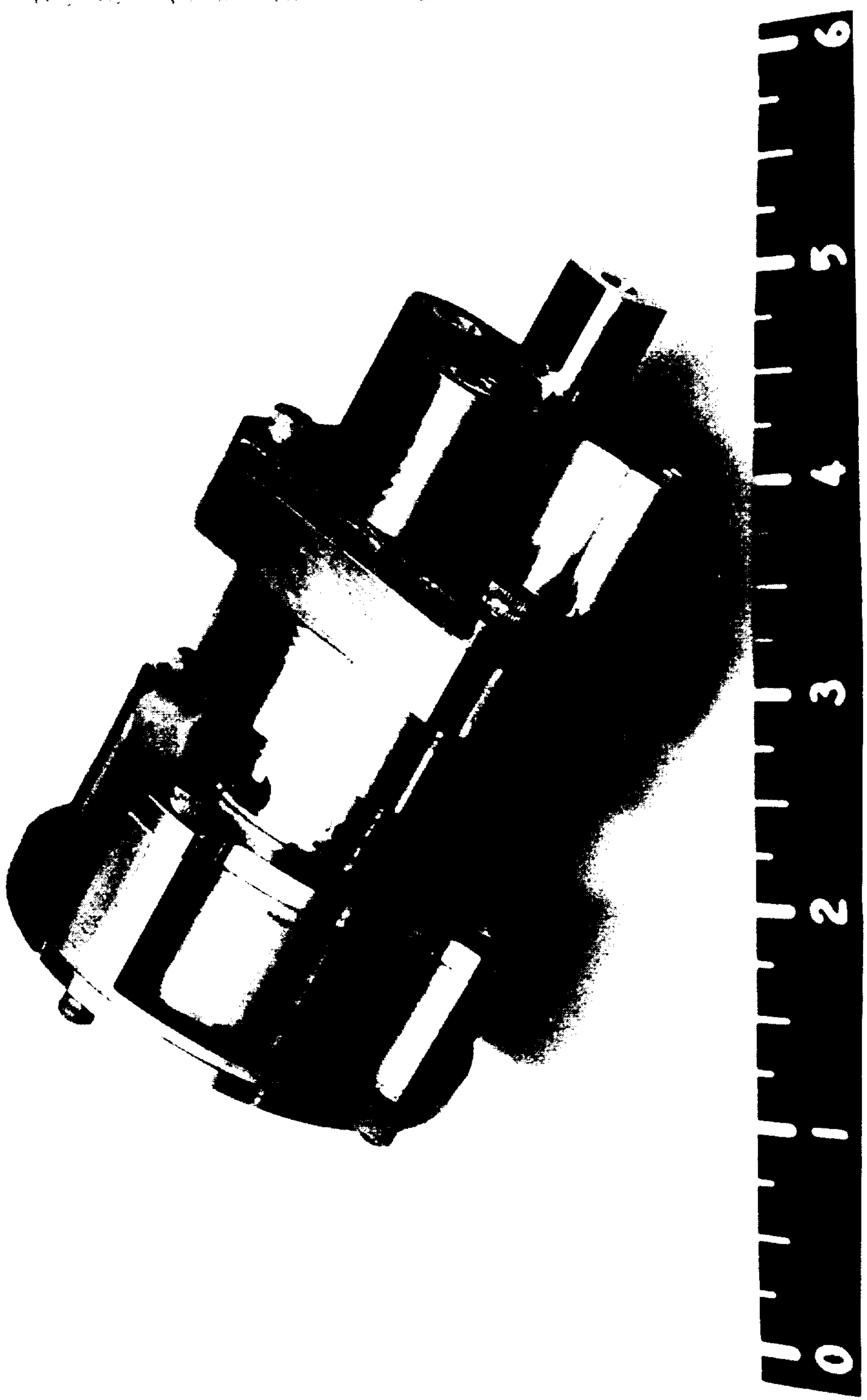
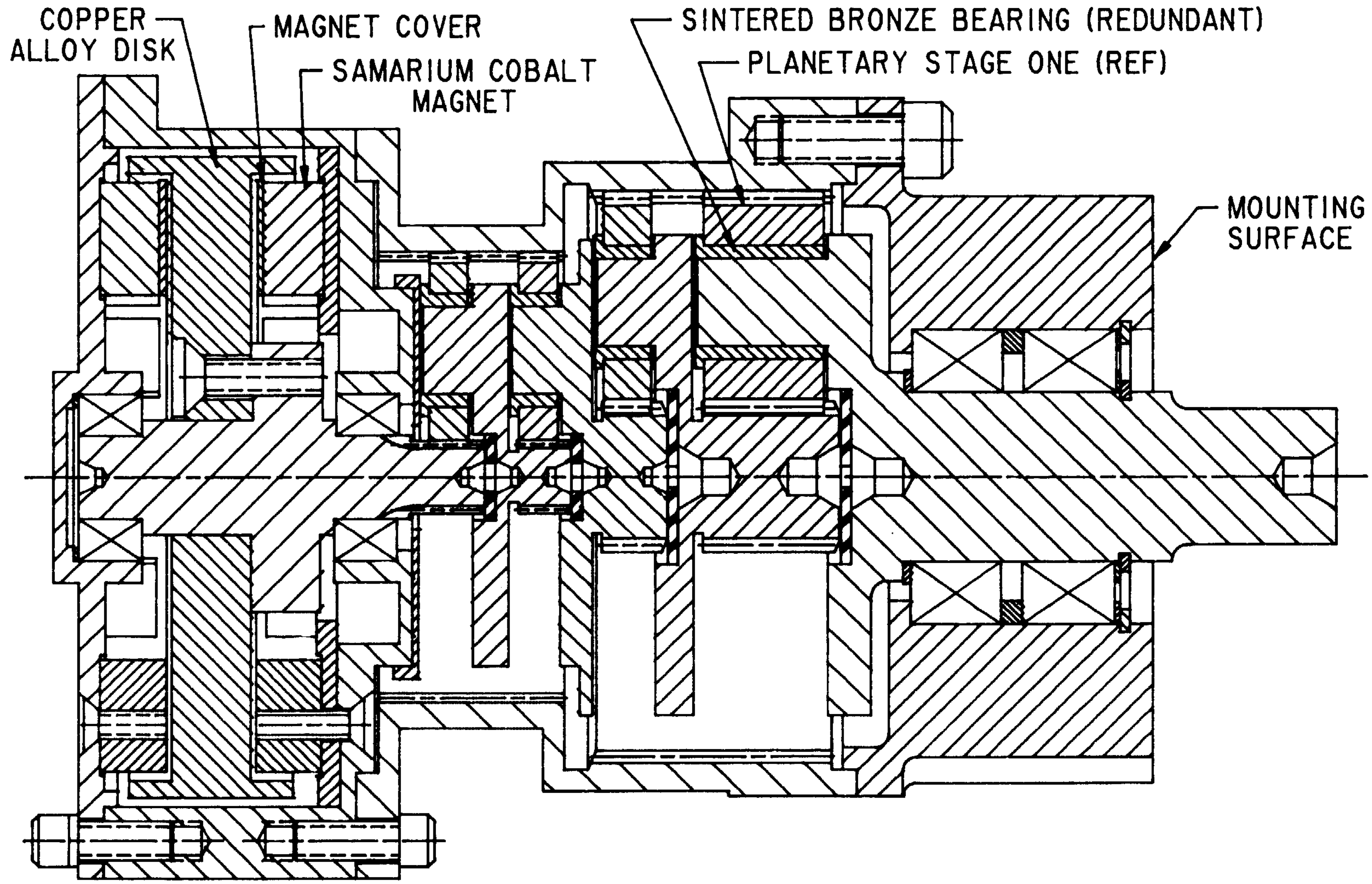


Figure 1



MAGNETIC DAMPER

Figure 2

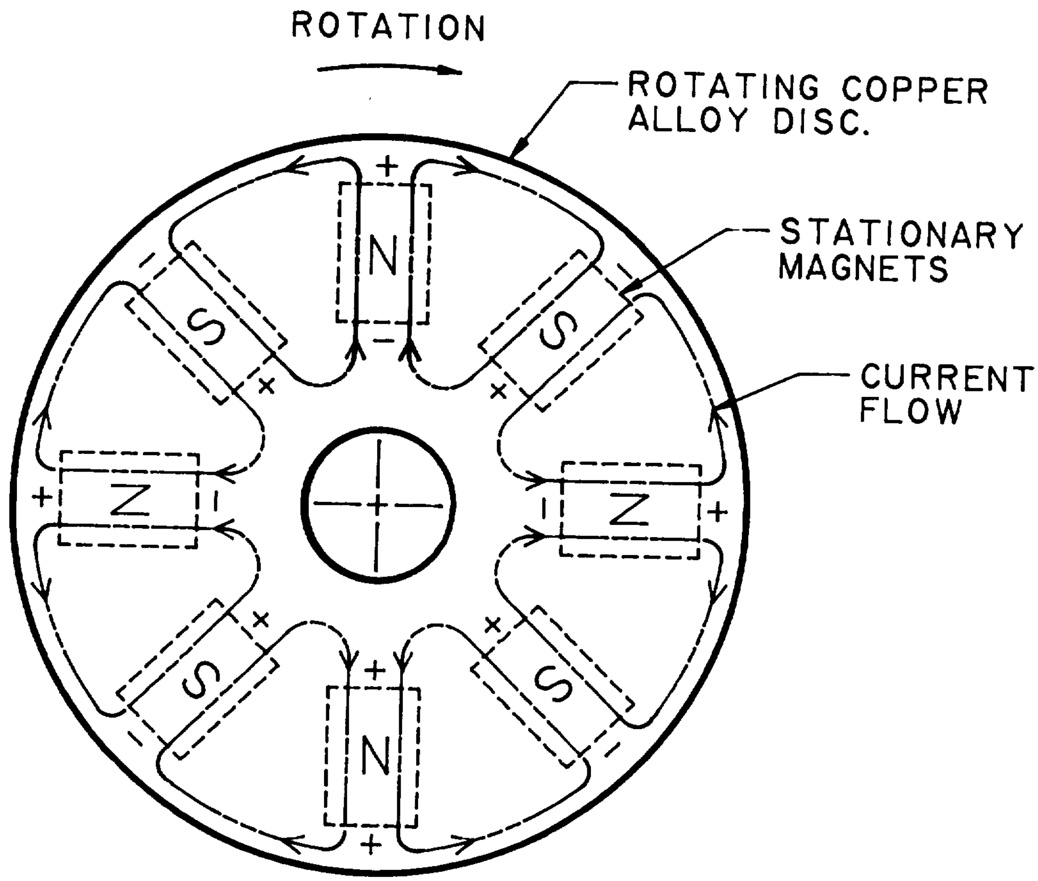
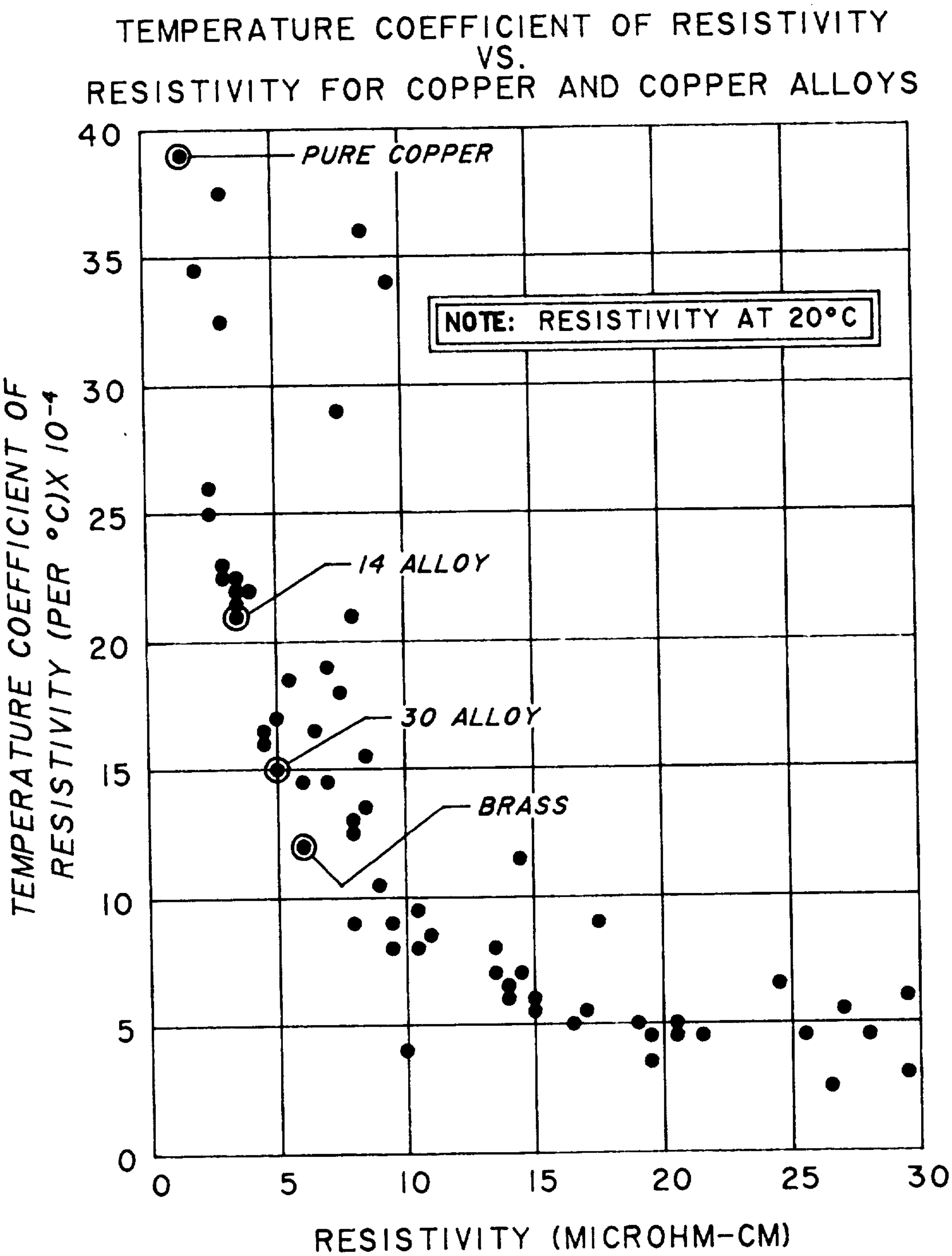

ROTATION
ROTATING COPPER ALLOY DISC.
STATIONARY MAGNETS
CURRENT FLOW
N
S
S
N
N
S
S
N
+
-

Figure 3



TEMPERATURE COEFFICIENT OF RESISTIVITY
VS.
RESISTIVITY FOR COPPER AND COPPER ALLOYS
PURE COPPER
NOTE: RESISTIVITY AT 20°C
14 ALLOY
30 ALLOY
BRASS
TEMPERATURE COEFFICIENT OF RESISTIVITY (PER °C) X 10⁻⁴
RESISTIVITY (MICROHM-CM)
0
5
10
15
20
25
30
35
40

Figure 4

RESISTIVITY VS. TEMPERATURE COEFFICIENT

FOR SELECTED ALLOYS

MATERIAL	RESISTIVITY (MICROHM-CM)	TEMP. COEFF. OF RESISTANCE (PER °C)
SILVER	1.628	0.0038
COPPER (DRAWN)	1.724	0.0039
GOLD	2.44	0.0034
111 ALLOY (DRIVER-HARRIS)	2.9	0.0025
CD 1.07, SN 0.59, FE 0.02, SI 0.02	3.05	0.00224
MN 0.43, FE 0.01, MG 0.01	3.38	0.0019
BRONZE (COMMERCIAL)	4.2	0.0020
30 ALLOY (DRIVER-HARRIS)	5	0.0015
ZINC	5.97	0.0037
PD 10	6.05	0.00091
BRASS	6.21	0.0015

Figure 5