# APPLICATION OF TORQUE MARGIN FOR EDDY CURRENT DAMPERS

**Scott Starin, Tony Rodriguez**

CDA InterCorp
450 Goolsby Boulevard, Deerfield, Florida 33442-3019, USA
Telephone: (+1) 954.698.6000 / Fax: (+1) 954.698.6011
e-mail: scott@cda-intercorp.com
tony@cda-intercorp.com

## ABSTRACT

Eddy Current Dampers (ECDs) offer higher robustness, torque capacity and linearity than Fluid Dampers. One of the perceived disadvantages of ECDs when compared to Fluid Dampers is the magnitude of zero speed coulomb torque. However, the magnitude of total coulomb torque must be analyzed and considered when applying torque margin ratios, depending on the construction of the ECD and method of reaction torque generation.

## 1. INTRODUCTION

There are several basic constructions for ECDs. Originally, the market consisted mainly of "rotating disk" ECDs where the magnets are stationary and one or more rotating electrically conductive disks generate the eddy currents. Alternatively, "shorted alternator" designs typically use rotating magnets, with stationary, shorted coil windings, in a permeable core, to generate the eddy currents and reaction torque. While both of these designs operate on the same electromagnetic principle, the fundamental differences in the rotating elements result in dramatically different components of static friction of the ECD assembly. These differences must be considered when applying reasonable and appropriate torque margins for ECD inputs.

Another key element of ECDs is the gearbox construction standard. In most ECD applications, gearing is required to increase the effective damping rate at the low speed input. However, the type and construction of the gearbox plays a significant role in the magnitude of the static friction, and what torque margins should be applied.

## 2. COMPARISON OF ECD DESIGNS

The two primary types of ECD configurations are the Rotating Disk Type Damper and the Shorted Alternator Type Damper. Configurations and design principles are presented for the discussion of operating theory as well as presentation of the fundamental components of static friction (stiction) and magnetic coulomb torque. The following discussions of ECD designs refer to the high-speed damper modules. Gearbox contribution is also discussed in this paper.

### 2.1 Rotating or Conductive Disk Dampers

The design of Rotating Disk Dampers consists of one or more electrically conductive, non-magnetic disks rotating between a series of permanent magnets. When the disk is rotated, eddy currents are generated in the conductive disk. These eddy currents generate reaction torques in proportion to the rotational velocity. Refer to Figure1 for a schematical representation of a Rotating Disk Damper. For this design concept, the only component of the backdriving torque is the stiction of the preloaded bearings supporting the conductive disk.

### 2.2 Shorted Alternator Dampers

These types of dampers have several fundamental differences when compared to Rotating Disk Dampers. The *permanent magnets* rotate, relative to the magnetic, stationary, soft iron and shorted coils. Eddy currents are generated in the coils and, in turn, generate the reaction torque damping characteristics. See Figure 2 for a mechanical schematic representation of this type of ECD assembly.

Since the MAGNETS rotate with respect to the magnetic soft iron core, there will be an inherent Magnetic Coulomb Torque (also referred to as detent or cogging torque) resisting the angular motion, due to the soft iron material hysteresis and residual magnetism. The magnetic field couples to the soft iron, as the permanent magnets rotate. The residual magnetism of the soft iron wants to keep the magnets in place. Depending on the specific configuration of the soft iron core and permanent magnets, the magnitude and ripple (torque versus position) of the Magnetic Coulomb Torque will vary with the design.

It is easy to separate the Magnetic Coulomb Torque from the bearing stiction by replacing the magnet shaft assembly with a shaft with identical bearing preload and *no magnets.* Care should be taken to assure the bearing preload is as close as possible in both configurations. A simple torque watch or reaction torque stand may be used to determine the magnitude of damper bearing stiction and total magnetic coulomb torque by backdriving the dampers with each of these shaft configurations.

## 3. GEARBOX CONSIDERATIONS

Another critical component of evaluation of friction components is analysis of the gearbox construction. Since the gearbox is such a critical component of a robust ECD design, careful consideration should be given to the type and configuration of the gearbox assembly. The following sections describe different types of gearboxes, and detail the different aspects of friction, and how they vary with time, temperature, and gravitational orientation. Matched coefficient of thermal expansion materials is highly desirable in any ECD gearbox design, and should be considered when analyzing the requirements for an ECD application.

### 3.1 Floating Plane Planetary Gearboxes:

The advantage of a Floating Plane design is lower cost and high gear ratios in a compact axial length. The disadvantages include lower gearbox efficiency, variation of static friction and lower torque capacity per given diameter. Additionally, it is generally not advisable to use a Floating Plane Planetary in cryogenic or extreme high temperatures, due to the differential velocity contact area.

ECD assemblies that use these types of planetary gearboxes typically have more mechanical static friction when compared to "supported cage" type of planetary gearboxes. Not only will the magnitude of the static friction be greater, but the variation of the friction will vary with the gravitational orientation of the gearbox (or lack thereof). If the position of each plane of gearing shifts, the contact area of the rotating planet gears changes, resulting in increased friction in some gravitionational orientations. Subsequently, in a zero or micro-"g" environment, the planar location of gears will be random. Further, the magnitude of static friction offset will vary with input torque magnitude. As torque is applied to the input shaft of an ECD assembly, resultant pinion twist from application of torque will tend to drive the high speed floating planes into the lower speed floating planes, changing the total friction offset.

### 3.2 Supported Cage Planetary

Supported Cage Planetary Gearboxes offer higher efficiency, higher torque capacity per unit volume, lower mechanical friction values, and wider operating temperature ranges; when compared to Floating Plane Planetaries. The disadvantages of supported cage planetaries include higher costs and lower gear ratios for a given length.

Unlike the Floating Plane planetary gearboxes, each stage of planetary gears are *supported by its own set of preloaded carrier bearings*. Regardless of orientation, the change in surface contact area is minimal, and you do not have differential velocities of gears and surfaces sliding against each-other.

In order to achieve ratios greater than about ten to one, multiple stages of planetaries are required. Since each stage is independently supported by a set of bearings the length is longer than a like ratio Floating Plane design. Alternatively, a spur-planetary combination can provide high ratios in compact lengths.

### 3.3 Spur Gearboxes

Spur gearboxes can offer high gear ratios in a compact length. However, their use should be limited to the high speed end of the gearbox. The low speed, high torque stage should be accommodated with a high capacity planetary gearbox.

Spur designs utilize gear clusters, supported on a set of bearings. Each pass, or level, of gearing is separated from the others, and like the Supported Cage Planetary, static mechanical friction is independent of orientation.

### 3.4 Harmonic Drive Gearboxes

Harmonic drive gearboxes offer high ratio on a single module, however, they are notorious for very high stiction torque, and the performance varies greatly with temperature. Harmonic Gearboxes should be avoided in ECD applications.

## 4. APPLICATION OF TORQUE MARGINS

One must consider the basic construction of the high speed ECD module when considering applicable torque margins for deployable systems. Rotating Disk type dampers have inherently less zero speed backdriving torque when compared to Shorted Alternator type dampers. This is due to the magnetic coulomb torque generated as a result of the rotating magnetic field with respect to the magnetic return path. However, Shorted Alternator type designs are more effective in generating damping torques per unit volume. Therefore, to obtain a specific damping rate at the low speed input, a Rotating Disk damper usually requires a higher gear ratio than a comparable size Shorted Alternator design.

The rule-of-thumb in determining minimum torque margin for ECDs is to apply a 3x torque margin ratio to zero speed static friction (200%). While this magnitude is appropriate for pure mechanical friction, it is "over margining" to apply a 3X torque ratio to the Magnetic Coulomb torque. Magnetic Coulomb of the rotating magnet with respect to the stationary soft iron is *material contact free torque, which cannot change over* *time or repeated cycles.* There is a slight increase in the magnetic coulomb torque at cryogenic temperatures, due to the magnetic flux density increase as temperature decreases. Samarium Cobalt typically has a temperature coefficient of Remanence of -0.03% /K. Therefore, as the temperature goes down, the magnetic Coulomb Torque increases, although not by that entire magnitude; as the air gap helps stabilize that minimal affect.

CDA has characterized an ECD with a Shorted Alternator type design, with a Supported Cage Planetary Gearbox. We tested the ECD assembly zero speed torque with a magnet assembly and with a plain shaft assembly to duplicate the preload and high speed gear mesh friction. The following characterizes the maximum static zero speed input torque.

A. Magnet Assembly Test: This test quantifies the total static friction of an ECD assembly. This includes the Magnetic Coulomb Torque and total mechanical static friction.

+100° C Torque = 0.416 Nm
+25° C Torque = 0.445 Nm
-100° C Torque = 0.494 Nm

B. Tests with plain shaft (no magnet): This test determines the total mechanical static friction. It separates the Magnetic Coulomb Torque from the test above.

+100° C Torque = 0.031 Nm
+25° C Torque = 0.035 Nm
-100° C Torque = 0.048 Nm

Clearly, the main component of the total zero speed input torque is the magnetic coulomb torque. The mechanical static friction may be characterized in test condition B above. It is important to conduct Mechanical Static Friction tests in multiple gravitational orientations.

When torque margins are analyzed it is entirely appropriate to apply the 3 X torque margin ratio to the worst case *mechanical static friction*, and apply a lower magnitude torque margin ratio to the Magnetic Coulomb Torque; if a Shorted Alternator type damper is used. It is recommended to use 1.25 to 1.50 X torque margin on the Magnetic Coulomb Torque component. More conservative applications could use as much as 2 X torque margin for the Magnetic Coulomb component of the total static friction, and use a 3X or even a 4X torque margin ratio for the maximum mechanical static friction. This yields a more realistic and appropriate analysis of torque margin for an ECD. MIL-A-83577 B [1] suggests 100% torque margin (2.0 X torque margin ratio) for rotating elements, at the acceptance test or qualification phase of a program, if actual torque values are measured.

There is a revision to NASA Goddard's publication "General Environmental Verification Specification for STS and ELV" (GEVS-SE) that is being considered, to include separation of torque margin requirements for known quantifiable resistive torques that do not change over the operating life of the unit.[2] Magnetic Coulomb Torque is an ideal example of this torque classification. The revision change is still in-process at the time of this publication due date.

## 5. DAMPER SCHEMATIC CONFIGURATIONS

Figure 1 is a schematic representation of a Rotating Disk Eddy Current Damper. The damping reaction torques are produced by eddy currents generated in the electrically conductive, non-magnetic rotating disk. The predominance of the zero speed backdriving torque with this design is the bearing preload, and the mechanical stiction of the bearings. There is little to no residual magnetism effect on the zero speed torque, since the rotating element is non-magnetic.

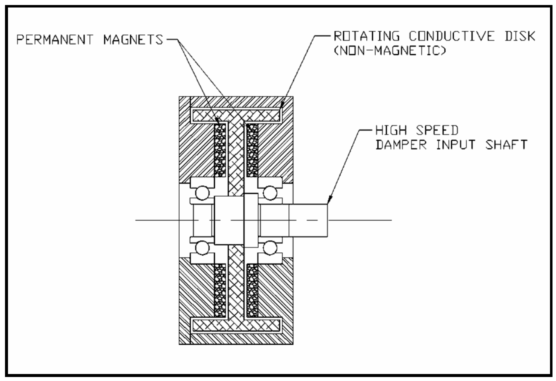


**Figure 1**

Figure 2 is a schematic representation of a Shorted Alternator Eddy Current Damper. The damping reaction torques are produced, as the rotating input shaft with permanent magnets, generate eddy currents in the copper coils embedded inside the stationary soft iron stator core. The predominance of the zero speed backdriving torque with this design, is the residual magnetism interaction from the stationary core to the rotating permanent magnets. The magnetic coulomb torque is a mechanical contact-free torque that does not generate wear or particles. The mechanical stiction of the bearings is a small portion of the zero speed backdriving torque for this design.

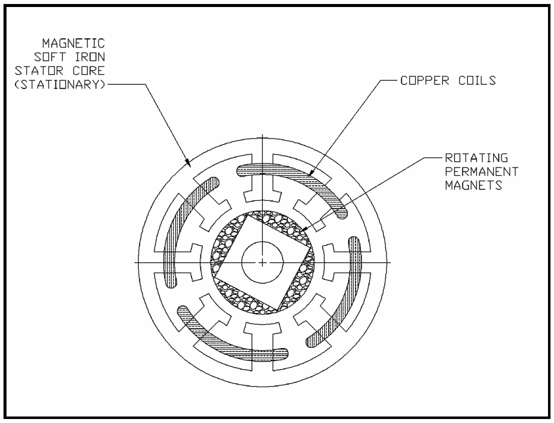


**Figure 2**

## 6. CONCLUSION

All Eddy Current Damper (ECD) designs are not equal. It is not appropriate to categorically apply fixed torque margin ratios to all ECD designs. Careful consideration of the fundamental damper design principles and empirical testing of the damper is required before determination of minimum torque margin ratio is defined. Magnetic Coulomb Torque (or residual magnetism) should be separated from Mechanical Static Friction when analyzing any ECD design. Application of torque margin ratios should be separated from the component of torque that is *material contact free*, and that which consists of true *mechanical static friction.* Testing should be conducted with a replacement shaft with identical bearing preload, in order to separate out the residual magnetism affect.

Gearbox design is another key factor in consideration of energy absorption devices. Deigns that have unpreloaded gear structures that are orientation specific should be used with caution. ECD assemblies should be tested in all gravitational orientations to determine maximum total static friction.

## 7. ACKNOWLEDGEMENTS

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## 8. REFERENCES

[1] MIL-A-83577 B, Military Specification General Specification for Assemblies, Moving Mechanical, for Space and Launch Vehicles 31 May, 1996

[2] GEVS-SE Rev A. General Environmental Verification Specification for STS & ELV. Payloads, Subsystems and Components. NASA Goddard Space Flight Center, June 1996

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