EDDY CURRENT DAMPER SIMULATION AND MODELING

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ABSTRACT

In spring loaded and passive deployment systems, the need for high reliability, predictable energy absorption is required to contour the deployment, and dissipate system potential energy in a desired manner. Eddy Current Dampers (ECD's) offer higher reliability and more stable performance over fluid dampers. One major advantage of ECD's is their linearity and predictable performance. With the proper math model and determination of system constants and variables, it is possible to accurately model performance of an ECD deployment system over temperature and torque variations. It is the aim of this paper to present the equations and testing methods to properly predict and size an ECD for a passive deployment system.

1. INTRODUCTION

There are different constructions and designs for ECD's, depending on the manufacturer. While the design and "temperature factors" may vary from manufacturer to manufacturer, the basic principle of using a high speed magnetic damper, and a gearbox to increase damping rate and torque capacity is almost universal. Both the high-speed magnetic damper and the gearbox will contribute to the static friction and damping characteristics of the ECD assembly. To properly model the characteristics and performance, the engineer should separate the various reaction torque properties of the high-speed damper and the gearbox, and reflect these values to the high torque, low speed, input shaft of the ECD. Lubrication methods and techniques, and system load variations and temperature characteristics must also be considered. We will first introduce and discuss the various elements and their contribution to performance, and then we will provide a specific example with empirical testing to validate the modeling technique.

2. HIGH-SPEED MAGNETIC DAMPER

As mentioned, the basic construction of the high-speed magnetic damper will vary, but you may characterize the basic properties of these magnetic dampers, regardless of the construction. For consistency and clarity, we will refer to the high-speed magnetic damper as the high-speed damper, and the overall assembly of the high-speed magnetic damper plus the gearbox as the ECD.

2.1 Damping Rate of the High-Speed Damper

The damping rate of the high-speed damper is obviously the first characteristic one needs to determine. In this paper, we refer to the damping rate of the high speed damper as **BD**, and it is rated at +25˚ C. The damping rate is defined as the change in reactive torque of the damper, over the change in speed. This sounds obvious, but it is important to note that this value looks at the slope of the reactive torque line, rather than taking the ratio of the torque to speed at a particular point. This subtlety will be discussed in further detail later. The damping rate of the high-speed damper will change with temperature. The magnitude of the change will vary with damper design and materials, but with CDA's design; you may calculate 0.4% decrease in damping rate per ˚C increase in temperature. The damping rate of the high-speed damper at a temperature (**T**) is designated as $\mathbf{BD_T}$.

It is also important to know the maximum linearity of damping for a particular high-speed magnetic damper. There will be a maximum torque and speed point of the high-speed damper where the linearity will start to roll off. This value is typically much higher than a system would ever be employed, and it will be much higher in value than a fluid damper could achieve.

2.2 Coulomb Friction of the High-Speed Damper

The next important characteristic of the high-speed magnetic damper is the coulomb (or static) friction, designated **FD** herein. The main contributor of this term is the residual magnetic "memory" or coercive force of the soft iron steel used in the high-speed damper. As you introduce a permanent magnet field inside a soft iron structure, you will magnetize the structure to an extent that when you try to rotate one with respect to the other, the residual magnetism in the iron will try to resist the movement of the magnetic field. Some damper designs keep these two members stationary, lowering the residual magnetism effect. However, these designs tend to be less linear, and have lower effective damping rates for the same volume. Other components of the **FD** are the bearing preload and mechanical friction term of the rotating elements. It is not important to know the exact proportion of these frictions, only the combined total of the magnitudes. This is easily determined with a torque watch or similar device. The **FD** term does not include lubrication effects, which may change with temperature.

2.3 Lubrication Effects of the High-Speed Damper

We have separated the lubrication effects from the other characteristics, because their contribution will vary with lubrication technique, and will also change with temperature, if a wet lube system is employed. It is interesting to note that the effects on performance due to lubrication contribute very little (if any) to the damping characteristic. The main contribution of lubrication is to offset the static friction at the output as temperature decreases. Here again, there will be significant non-linearitites if the temperature gets too cold for the wet lube selected. Dry lube may also have unintended secondary effects depending on the method selected. It is impossible for us to discuss all of the possibilities here, but we will show a specific example and the effects. The component for lubrication effects is designated as $\mathbf{FL_T}$ because it is a friction component, which is a function of temperature, **T**. The $\mathbf{FL_T}$ is actually an easy function to establish. If you measure the change in coulomb friction of the ECD assembly over the temperature range, you can easily estimate the linear relationship for the change of coulomb friction over temperature.

3. GEARBOX CONTRIBUTION TO OVERALL DAMPING AND PERFORMANCE

The gear ratio has a squared effect on the damping from the high-speed damper to the low speed input shaft. This squared effect results in a tremendous increase in the overall damping of the ECD by the inclusion of the gearbox. But what is often overlooked is the fact that gearing efficiency, or more properly stated "gearing inefficiency", will also contribute to the damping rate of an ECD assembly.

3.1 Dynamic Torque In a Gearbox

In the conjugate action of gear mesh, the reactive torque generated in the gearbox will be a function of speed as well as torque. That is to say, the efficiency of a gearbox varies with speed and torque. The resultant characteristic is a true reactive torque generation which is proportional to input speed. Therefore we say that there is a damping rate of the gearbox. It is important to note that this is a mechanical phenomenon, not associated with the lubrication system. Figure 1, which helps illustrate this point, shows the friction of a gear mesh with respect to pitch line velocity. Notice the linearity of the friction with respect to speed. We are not suggesting that a poorly designed gearbox is desirable. In fact, it is critical to have a high quality gearbox in order to control the magnitude and consistency damping torque contribution, while maintaining low coulomb friction.

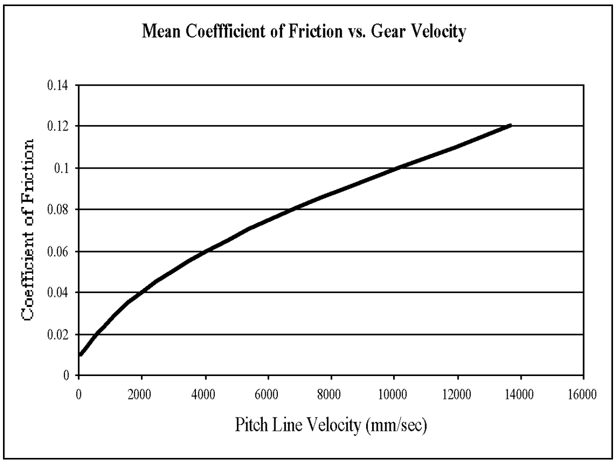

Figure 1

For analysis purposes, it is convenient to rate the damping rate of the gearbox reflected to the high-speed damper as about 20 to 30% of the damping rate of the high-speed damper itself. Damping rate of the gearbox is designated as **BG**, and is referred to the high-speed damper.

3.2 Coulomb Friction of a Gearbox

The gearbox will also contribute to the coulomb friction of the overall ECD. The main contribution of this term is usually bearing preload and friction. Again, it is not important to know the individual components of this term, but rather it's combined effects. This paper assumes that there are matched coefficients of thermal expansion (CTE) in the gearbox, and there will be no change in levels over the operating temperature range of the system. Designs which use "floating plane" planetary gearsets, or materials with unmatched CTE's may experience significant changes in gearbox friction and damping levels over temperature and time. Coulomb friction of the gearbox is designated as **FG**, and is referred to the high-speed damper.

3.3 Lubrication Effects in the Gearbox

Lubrication effects on the gearbox are usually insignificant and may be ignored for analysis purposes. As the contribution of the friction due to lubrication is generally greatest at the high-speed device, one can group this into the single category of $\mathbf{FL_T}$. Once again, we stress that this characterization is not for every manufacturer of dampers, but rather the design utilized by CDA InterCorp. Derivation of functions may be necessary for some manufacturers' gearboxes.

3.4 Lost Motion (or Deadband) in an ECD

Unlike a fluid damper, which typically has 15° or more of deadband, a quality ECD should have no more than 0.25° of lost motion. Values of this low magnitude may be ignored in the overall scheme of analysis. However, if desired, this characteristic may be included, for comparison sake.

4. COMBINING THE EFFECTS AND REFERRING THE PERFORMANCE TO THE LOW SPEED INPUT OF THE ECD

From the background given, we can establish some basic equations to combine the effects of the different components for an ECD. First, let us combine the components, and reflect them to the high-speed damper:

The total damping rate at temperature **T**:

$$\mathbf{BT_T = BD \bullet (1-.004(t-25)) + BG} \qquad \textbf{Eq. 1}$$

Total coulomb friction at temperature **T**;

$$\mathbf{FT_T = FD + FL_T + FG} \qquad \textbf{Eq. 2}$$

Referring these parameters to the low-speed input of the ECD (High-speed damper + gearbox):

Damping rate of the ECD, at temperature, **T**:

$$\mathbf{BECD_T = BT_T \bullet N^2} \qquad \textbf{Eq. 3}$$

Coulomb Friction of the ECD, at temp., **T**:

$$\mathbf{FECD_T = FT_T \bullet N} \qquad \textbf{Eq. 4}$$

Where "N" is the gear ratio. If $\mathbf{BECD_T}$ and $\mathbf{FECD_T}$ are known, we can start predicting performance in the system. For a given input torque at temperature, **T** ($\mathbf{TI_T}$), we can calculate the velocity of the ECD:

$$\mathbf{\omega I_T = (TI_T - FECD_T) \div BECD_T} \qquad \textbf{Eq. 5}$$

DO NOT assume that input torques ($\mathbf{TI_T}$) are constant over temperature. Even simple spring-loaded hinge mechanisms can have significant changes in torque magnitude over temperature if the friction levels and spring torque change appreciably. The best advice here is to empirically test the hinge over the desired temperature range, and establish a function to characterize the input torque. Also, in most systems, the input torque generated from the spring is a function of position. The input torque is typically greatest at the start of the deployment, and reduces linearly to the end of the deployment angle. Therefore, in simulating ECD performance and deployment time, it is important to characterize the torque over the deployment angle because TI_T is a function of position as well as temperature.

Because of the inherent offset due to the coulomb friction of the ECD ($\mathbf{FECD_T}$), the Dynamic Damping Rate ($\mathbf{DBECD_T}$), is not constant. This is most noticeable at very low loads. However, it should be noted that even though this non-linearity exists, the effect or magnitude of it is significantly lower for ECD's when compared to fluid dampers. Obviously the non-linearities for fluid dampers occur for very different reasons, but it is important to realize that one should not simply specify a damping rate for a damper because the value varies with load. For this reason, we at CDA have specified the damping rate at a specific point as the "Dynamic Damping Rate", calculated as follows:

$$\mathbf{DBECD_T = TI_T / \omega I_T} \qquad \textbf{Eq.6}$$

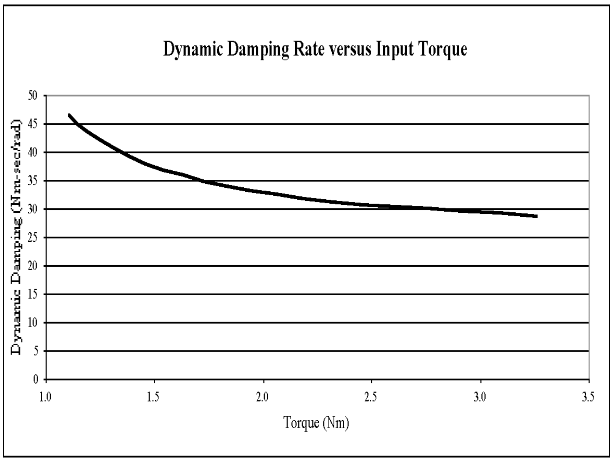

Figure 2

5. MODELING A SYSTEM

The next step in system modeling makes some basic assumptions and neglects stiffness and resonant frequencies. However, we will establish a baseline characterization that accurately models the deployment position, velocity profiles, and deployment time over temperature and torque within the linear range.

The first step is to determine what the load input torque is over the deployment range. Using spring manufacturers published data is a starting point. Since the spring torque is fairly linear, a simple proportional approximation based upon position will yield good results. As mentioned, it is a good idea to verify the load torques and frictions over the operating temperature range. After establishing these values, we assign these variables:

Deployment Angle: $\boldsymbol{\Delta\theta}$ (in radians)
Position at start of Deployment: $\boldsymbol{\theta s}$
Position at end of Deployment: $\boldsymbol{\theta e}$

Torque at start of Deployment: **Ts** (in Nm)
Torque at end of Deployment: **Te** (in Nm)
System Friction over temperature: $\mathbf{FS_T}$ **(inNm)**

If the torque is linear over the deployment angle, the input torque at any position, **θ**, may be expressed as:

$$T\theta = \theta \div \theta s \bullet (Ts-Te)+Te \qquad \textbf{Eq. 7}$$

The next step is to determine the velocity at the start and end of deployment (ωs & ωe):

$$\omega s = (Ts - (FECD_T + FS_T)) \div BECD_T \qquad \textbf{Eq. 8}$$

$$\omega e = (Te - (FECD_T + FS_T)) \div BECD_T \qquad \textbf{Eq. 9}$$

Since ECD's are so linear, we can assume an average velocity (ϖ), and a constant deceleration (α):

$$\varpi = (\omega s + \omega e) \div 2 \qquad \textbf{Eq. 10}$$

Where our deployment time (**Δt)** is simply:

$$\Delta t = \Delta\theta \div \varpi \qquad \textbf{Eq. 11}$$

And our negative acceleration is:

$$\alpha = (\omega e - \omega s) \div \Delta t \qquad \textbf{Eq. 12}$$

Now we use the kinematic equation for motion to determine position at time "t":

$$\theta_t = \theta s - (\omega s \bullet t) - 0.5\alpha \bullet t^2 \qquad \textbf{Eq. 13}$$

As we mentioned before, we are ignoring the initial time to accelerate the ECD and load inertia up to speed, since it is typically an insignificant percentage of the deployment. At this point we have all the equations to model the system.

6. TEST DATA OF ECD SYSTEM

In an effort to demonstrate the equations above, CDA produced an ECD assembly to the following specifications:

ECD Type: Cc-16
Mass: 0.185 kg (With Resolver)
Length: 63 mm (With Resolver)
Diameter: 25 mm
Option(s): Integral *OnAxis* Resolver

The following constants and variables were determined and reflected to the high-speed-damper:

BD = 1.3 E-04 Nm•sec/rad
FD = 7.9 E-04 Nm
FL_T = -2.12E-06•T+2.2E-04 Nm
N = 380:1
BG = BD • 0.26
FG = 4.5 E-04 Nm

Which provides the following characteristics, reflected to the low speed input of the ECD Assembly, at +25˚ C:

$FECD_{25}$ = 0.52 Nm
$BECD_{25}$ = 24 Nm•sec/rad

And the load description (within the range of operation) was determined as follows:

Ts = 3.73 Nm
Te = 1.58 Nm
FS_T = -0.0044 • T+0.58 Nm
Δθ = 1.57 Radians

The ECD assembly was mounted to the spring loaded assembly as shown in figure 3. This simple deployment mechanism was designed to easily fit inside a temperature chamber to test deployment characteristics over a temperature range of -40˚ C to +100˚ C. The output of the *OnAxis* Resolver was converted to a dc analog signal and recorded on a digital storage oscilloscope to record position versus time of the deployment at each test temperature.

Deployment Test Fixture

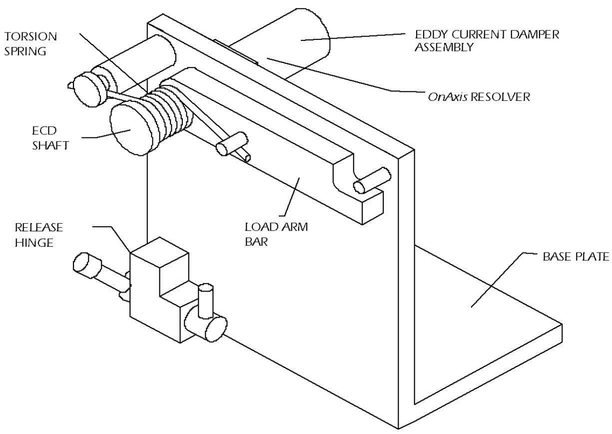

Figure 3

Using the established equations, we may evaluate the deployment position, **θ**, versus time, **t**, at any temperature, **T**, within the operating temperature range noted. The actual test data and the simulation estimates at +25˚ C, -40˚C and +100˚C are shown in figures 4, 5 and 6. As you can see the simulation yields excellent results. However, it must be noted that our original estimates were off by 10 to 20% of deployment time at the temperature extremes, until the system friction (FS_T) was properly characterized.

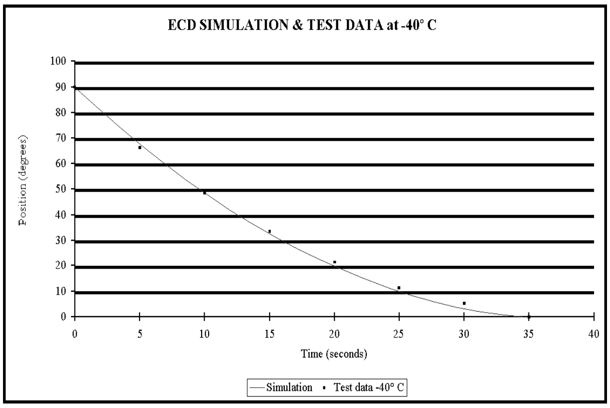

Figure 4

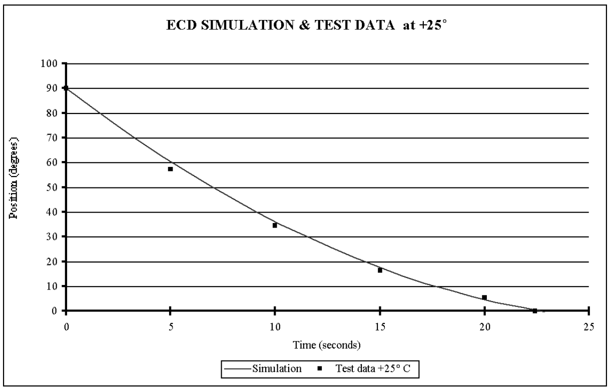

Figure 5

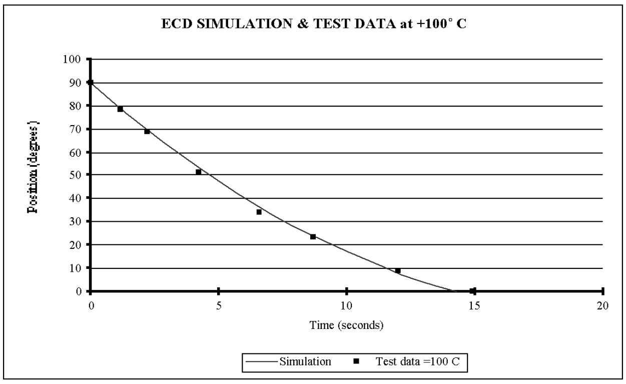

Figure 6

Additionally, to demonstrate the reliability and consistency of the ECD assembly, we conducted over 500 deployments and plotted the deployment profile at the temperature extremes. Results yielded a deployment time within 0.5 seconds of the original tests, at each temperature.

7. BACKING INTO THE SIMULATION

If you have an existing Eddy Current Damper that you would like to characterize, it would be difficult to establish all of the variables and constants reflected in this paper, since we characterized the performance from the high-speed damper. This was done in order to utilize manufacturers' catalog information to compare performance of various damper and gear ratio combinations. However, it is not too difficult to establish the characteristics of the overall ECD assembly with a simple test fixture. Ideally, what you would like to establish is the Damping Rate of the ECD assembly over temperature ($BECD_T$), and the coulomb friction of the ECD assembly over temperature ($FECD_T$).

Figure 7 shows a test rig used for testing ECD assemblies at CDA. It is important to design such an assembly with as little friction as possible, while maintaining high torsional and radial stiffness. Once the test fixture and mounting is established, simply apply various load masses on the flywheel, and record the time it takes for a given displacement (say 180˚ or 3.14 radians). When you plot Torque versus Velocity of the ECD you will see the linear relationship, and the coulomb friction offset. Take these measurements over several temperatures to establish the function of damping and coulomb friction with temperature. Make sure you consider any errors introduced by the test rig such as friction over temperature.

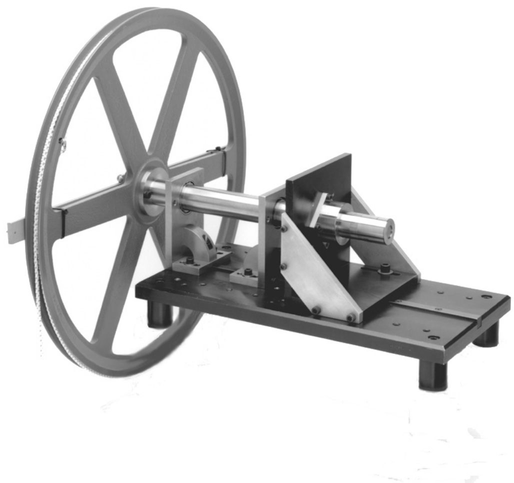

Figure 7

The Figure 8 is a composite performance you would find for the ECD used for this demonstration. This result is typical for an ECD assembly. Regardless of the desire to fully model a system, taking the time to characterize the performance, linearity, and consistency of an ECD assembly is well worth the effort.

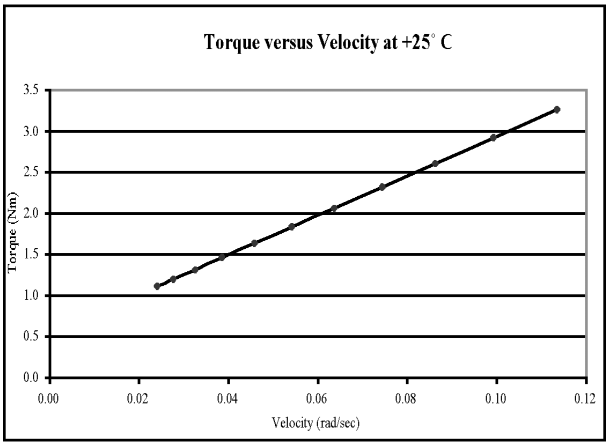

Figure 8

8. CONCLUSION

We have demonstrated both analytically and empirically the performance of an ECD in a passive deployment system. It is imperative to establish both the load and the ECD characteristics over operating temperature, as well as determine linear limits. Once these parameters have been established, simple linear analysis yields excellent simulation results. If an engineer is sizing a new application, it is preferable to model the characteristics reflected to the high-speed damper before reflecting the performance to the low speed input. If an engineer is trying to model an existing ECD assembly, empirical testing of the ECD assembly is required. Testing methods and fixturing should be robust, with friction levels minimized and characterized.

When defining the requirements for a damper assembly, the system engineer should consider at the desired deployment and load characteristics, rather than simply define a damping rate and maximum torque capacity.

References:

[1] InterCorp Engineering: Eddy Current Damper Engineering Reference Data, CDA InterCorp © 2000

[2] InterCorp Engineering: ECD vs. Fluid Damper Simulation Software CDA InterCorp © 2001