PRELIMINARY RESULTS ON PASSIVE EDDY CURRENT DAMPER TECHNOLOGY FOR SSME TURBOMACHINERY

Robert E. Cunningham
NASA Lewis Research Center
Cleveland, Ohio 44135

ABSTRACT

Some preliminary results have been obtained for the dynamic response of a rotor operating over a speed range of 800 to 10,000 rpm. Amplitude frequency plots show the lateral vibratory response of an unbalanced rotor with and without external damping. The mode of damping is by means of eddy currents generated with 4 "c" shaped permanent magnets installed at the lower bearing of a vertically oriented rotor. The lower ball bearing and its damper assembly are totally immersed in liquid nitrogen at a temperature of -197°C (-320°F). These preliminary results for a referenced or baseline passive eddy current damper assembly show that the amplitude of synchronous vibration is reduced at the resonant frequency. Measured damping coefficients were calculated to $\zeta = 0.086$; this compares with a theoretically calculated value of $\zeta = 0.079$.

INTRODUCTION

- Improved performance and durability of SSME turbopumps will depend, to a large degree, on an effective and predictable mechanism for dissipating vibrational energy in the rotors.
- Properly designed dampers located at or near bearing supports can be effective in controlling vibrations produced by rotor unbalance, commonly known as synchronous whirl.
- Dampers can also be effective in controlling non-synchronous whirl, the type often produced by shaft internal hysteresis, hydrodynamic seals, interference fits, tie bolts, etc.
- Dampers can reduce the dependence on ultra precision balancing or the need for frequent rebalancing.
- Dampers along with properly designed flexible bearing supports can reduce magnitude of transmitted forces thru the ball bearings to the casing, and thus extend ball bearing life.
- A unique method of damping is by means of eddy current generation in a conductor caused to vibrate in a magnetic flux field.
- The low temperatures encountered in the turbopumps actually increase the available energy dissipation and thereby make this mechanism an attractive candidate for SSME turbomachinery.
An objective of this work is to verify by experimentation the derived relationship that define the eddy current damping coefficient.

A rotating rest apparatus was designed and fabricated in order to properly evaluate candidate damper designs over a speed range of 800 to 10,000 rpm while operating in liquid nitrogen at -195°C.

Measured rotor response to unbalance forces with and without applied damping are compared to theoretical results from a computer code.

**SUMMARY AND CONCLUSIONS**

Demonstrated the successful attenuation of synchronous vibration at the first system resonance using passive eddy current damping in liquid nitrogen.

Obtained reasonable agreement between theoretical and measured response of an unbalanced rotor for both the undamped and damped cases. A measured damping coefficient of $\zeta = 0.086$ was obtained while the predicted damping coefficient was calculated to be $\zeta = 0.079$.
STATING FARADAY'S AND LENZ'S LAWS IN EQUATION FORM

IT CAN BE SHOWN THAT:

(eq. 1) \[ F_M = \frac{B^2 \ell^2}{R} \] \( V \) (NEWTONS)

WHERE: \( B \), FLUX DENSITY, \( \frac{\text{WEBERS}}{M^2} \)

\( \ell \), CONDUCTOR LENGTH, M

\( R \), RESISTANCE, ohms

\( V \), VELOCITY, M/sec

OBTAINING \( F_M \) IN TERMS OF MATERIAL CONDUCTIVITY WHERE:

(eq. 2) \[ R = \rho \frac{\ell}{A} \] \( \) WHERE \( \rho \), ohms-meters

AND

(eq. 3) \[ F_M = \left( \frac{B^2 \ell A}{\rho} \right) V \] \( \) (NEWTONS)
THE CONSTANT OF PROPORTIONALITY IS KNOWN AS THE DAMPING COEFFICIENT, $C_D$ THEREFORE:

\[ (eq. \ 4) \quad F_d = C_D \cdot V \quad (NEWTONS) \]

EQUATING $F_M = F_d$ AND SOLVING FOR $C_D$ WE HAVE

\[ (eq. \ 5) \quad C_D = \frac{B_g^2 \ell A}{\rho} \left( \frac{NEWTONS-sec}{M} \right) \]

THE DAMPING FORCE IS VELOCITY DEPENDENT AS IS VISCOUS OIL SQUEEZE FILM DAMPING. MOST IMPORTANTLY HOWEVER THE DAMPING COEFFICIENT $C_D$ VARIES AS THE SQUARE OF THE MAGNETIC FLUX DENSITY, $B_g$, AND INVERSELY AS THE CONDUCTOR MATERIAL RESISTIVITY $\rho$. 
ELECTRICAL RESISTIVITY OF COPPER AS A FUNCTION OF TEMPERATURE

REF: NBS CRYOGENIC DATA MEMORANDUM NO. M-11

$\rho_{273} = 1.545 \times 10^{-6}$ OHMS-cm

<table>
<thead>
<tr>
<th>TEMP., °C</th>
<th>RESISTIVITY, ohms-m</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>$1.6 \times 10^{-8}$</td>
</tr>
<tr>
<td>-197</td>
<td>$2.0 \times 10^{-9}$</td>
</tr>
<tr>
<td>-253</td>
<td>$3.09 \times 10^{-11}$</td>
</tr>
</tbody>
</table>

ELECTRICAL RESISTIVITY RATIO, $\rho/\rho_{273}$

TEMPERATURE, °K

CS-84-0496
REVERSIBLE TEMPERATURE CHANGES
IN REMNANCE FLUX AND COERCIVITY

REF: ARNOLD ENG'R

$\Delta\% = 0.05\% \text{ PER } 0\degree C$

BOR H, %

TEMPERATURE, 0°C
THE EQUATION FOR DAMPING COEFFICIENT $C_D$ MUST BE MODIFIED TO ACCOUNT FOR:

(A) FINITE CONDUCTOR GEOMETRY, INDUCTIVITY, $L_i$
TYPICAL VALUE MAY BE, $0.27 \mu_0$

$\mu_0$, MATERIAL PERMEABILITY

(B) FLUX DISTRIBUTION OVER FINITE POLE FACE
GEOMETRY, $f$
TYPICAL VALUE, $f = 0.66$

• BOTH OF THE ABOVE REDUCE THE MAGNITUDE OF DAMPING COEFFICIENT THEORETICALLY AVAILABLE

\[
C_D' = \frac{(f B_g)^2 \ell A L_i}{\rho \mu_0} \left( \frac{N\text{-sec}}{M} \right)
\]

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THE FLUX DENSITY ACROSS AN AIR GAP, REF. FIG. 1, IS DETERMINED BY THE MAGNET MATERIAL AND ITS HYSTERESIS CURVE AND THE MAGNET GEOMETRY. THE FOLLOWING RELATIONSHIP IS GIVEN IN MAGNET DESIGN MANUALS.

\[ B_g = \left( \frac{A_m}{A_g} \right) \frac{B_R}{\sigma} \]

WHERE:  
- \( A_m \), CROSS SECTIONAL AREA NORMAL TO MAGNETIC AXIS  
- \( A_g \), CROSS SECTIONAL AREA OF POLE FACES  
- \( B_R \), REMNANCE FLUX  
- \( \sigma \), FLUX LEAKAGE

\( \left( \frac{A_m}{A_g} \right) \), FOCUSING EFFECT

\( \sigma \), FLUX LEAKAGE IS A FUNCTION OF GEOMETRY.

TYPICAL VALUES FOR "C" SHAPED MAGNET  
2.5 TO 4
PERMANENT MAGNET DESIGNS FOR EXPERIMENTAL EVALUATION IN LIQUID N\textsubscript{2} AND LIQUID H\textsubscript{2}

"c" MAGNET MATERIAL, ALNICO V

**DIMENSIONS**
- $D = 2.4\text{ in}$
- $d = 1.5\text{ in}$
- $T = .5\text{ in}$
- $Lg = .25\text{ in}$
- $W = 1.2\text{ in}$

**THEORETICAL DAMPING COEFFICIENTS**
- **ROOM TEMP. (20\degree C)**
  - $C_D = 1.31\text{ lb-sec/in}$
- **LIQUID N\textsubscript{2} (-197\degree C)**
  - $C_D = 1.02\text{ lb-sec/in}$
- **LIQUID H\textsubscript{2} (-253\degree C)**
  - $C_D = 66.2\text{ lb-sec/in}$
- **OPTIMIZED GEOMETRY, COMPUTER PROGRAM**

"c" MAGNET MATERIAL, ALNICO V
POLE FACES, SAMARIUM COBALT

**DIMENSIONS**
- $D = 3.0\text{ in}$
- $d = 1.75\text{ in}$
- $T = .313\text{ in}$
- $Lg = .156\text{ in}$
- $W = 1.2\text{ in}$
NASA EDDY CURRENT DAMPER TEST APPARATUS
LIQUID N2 SYSTEM-15,000 RPM DESIGN, STIFF SHAFT

- ROTOR CROSS SECTION -

Ht = 17.5 LB  Lt = 18.5 IN.

NO. OF STATIONS = 14
NO. OF BEARINGS (&SEALS) = 2

UN DAMPED SYNCHRONOUS SHAFT NODES
Ht = 17.5 LB  Lt = 18.5 IN.

MODE 1 FREQUENCY = 86 Hz ( 6480 RPM)

NO. OF STATIONS = 15
NO. OF BEARINGS (&SEALS) = 2
THEORETICAL RESPONSE OF UNDAMPED VERSUS DAMPED ROTOR

UNBALANCE, 0.149 oz-in.; THEORETICAL DAMPING COEFFICIENT $\zeta$, 0.084

**Graphs:**
- **Undamped**
- **Damped**

**Axes:**
- **Y-axis:** Amplitude, mil
- **X-axis:** Shaft, rpm

**Graph Details:**
- The undamped rotor shows a sharp peak at a certain speed.
- The damped rotor has a smoother response with a lower peak.

**Note:**
- The graphs illustrate the effect of damping on the response of a rotor system.
MEASURED RESPONSE OF UNDAMPED VERSUS DAMPED ROTOR
UNBALANCE, 0.149 oz-in.; MEASURED DAMPING COEFFICIENT ζ, 0.079

NDAMPED

AMPLITUDE, mil

DAMPED

SHAFT, rpm

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### SUMMARY OF THEORETICAL VS MEASURED RESPONSE FOR LN₂ TEST ROTOR

**ROTOR UNBALANCE, .149 OUNCE - INCHES**

**SPRING CONSTANT, \( K_s = 11,100 \text{ lb/in} \)**

<table>
<thead>
<tr>
<th></th>
<th>THEORETICAL RESPONSE</th>
<th>MEASURED RESPONSE</th>
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<tbody>
<tr>
<td></td>
<td>UNDAMPED</td>
<td>DAMPED</td>
</tr>
<tr>
<td><strong>RESONANT FREQ.</strong></td>
<td>5400</td>
<td>5400</td>
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<tr>
<td>(CPM)</td>
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<tr>
<td><strong>MAX. AMPLITUDE</strong></td>
<td>11.9</td>
<td>6.2</td>
</tr>
<tr>
<td><strong>AT RESONANCE</strong></td>
<td></td>
<td></td>
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<tr>
<td>(MILS)</td>
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<td></td>
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<tr>
<td><strong>DAMPING COEFFICIENT</strong></td>
<td>≈.002</td>
<td>.084</td>
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</tbody>
</table>
### SUMMARY OF THEORETICAL VS MEASURED RESPONSE FOR LN$_2$ TEST ROTOR

**Rotor Unbalance, .149 Ounce - Inches**

**Spring Constant, $K_s = 11,100$ lb/in**

<table>
<thead>
<tr>
<th></th>
<th>Theoretical Response</th>
<th>Measured Response</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Resonant Freq. (CPM)</strong></td>
<td>UNDAMPED: 5400</td>
<td>DAMPED: 5400</td>
</tr>
<tr>
<td></td>
<td>UNDAMPED: 5200</td>
<td>DAMPED: 5600</td>
</tr>
<tr>
<td><strong>Max. Amplitude at Resonance (Mils)</strong></td>
<td>UNDAMPED: 11.9</td>
<td>DAMPED: 6.2</td>
</tr>
<tr>
<td></td>
<td>UNDAMPED: 16.1</td>
<td>DAMPED: 7.3</td>
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<tr>
<td><strong>Damping Coefficient</strong></td>
<td>UNDAMPED: .002</td>
<td>DAMPED: .084</td>
</tr>
<tr>
<td></td>
<td></td>
<td>DAMPED: .079</td>
</tr>
</tbody>
</table>