Nutation damping theory and computer implementation

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NUTATION DAMPING

THEORY

AND

COMPUTER IMPLEMENTATION

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UCN, Almelo, January 1993
Abstract

The movement of spinning satellites is often disturbed by nutationally and antenna movement induced motions. These (unwanted) motions have to be damped and a tube-with-endpots damper can be used for this purpose.

At UCN Almelo the nutation damping theory for the tube with endpots damper was spread about many different reports. In this report all the theory necessary for the design of a tube with endpots damper is collected.

Expressions are given for the fluid displacement in the endpots, the fluid velocity in the fluid tube and the damper performance. In order to test the damper on earth, scaling is necessary. The scaling relations are derived for this purpose. The computer implementation of the theoretical model is described.

Results were calculated for a practical model. These results were compared with test results and an accurate agreement was found for small nutation angles. The model is found to be inaccurate for large nutation angles. Therefore an update of the theory is recommended.
Symbols

\( a \) Tube radius of damper \([m]\)

\( a_0 \) Amplitude of forcing acceleration \([m/s^2]\)

\( a_{0\text{max}} \) Forcing acceleration along the fluid tube at the beginning of a testrun \([m/s^2]\)

\( a_{0\text{min}} \) Forcing acceleration along the fluid tube at the end of a testrun \([m/s^2]\)

\( a_c \) Centrifugal acceleration of the rotor of the testing apparatus \([m/s^2]\)

\( a_{sp} \) Acceleration as a result of the spin velocity of the damper on the satellite \([m/s^2]\)

\( a_t \) Tangential acceleration of the rotor of the testing apparatus \([m/s^2]\)

\( a_z \) Forcing acceleration \([m/s^2]\)

\( b \) Endpot radius of damper \([m]\)

\( b_{\text{eff}} \) Effective endpot radius in case of testing a flight model \([m]\)

\( c \) Constant defining the relation between \( a_{0\text{min}} \) and \( s \) \([Nm/rad]\)

\( d \) Distance between outer radii of fluid tube and vapour tube \([m]\)

\( f_n \) Frequency ratio \( \Omega/\omega_c \) \([-]\)

\( F \) Radius dependent fluid velocity \([m/s]\)

\( F_s \) Amplitude of tube-wall velocity \([m/s]\)

\( g \) Gravity \([m/s^2]\)

\( g_f \) Equivalent for the gravitational force on the fluid \([m/s^2]\)

\( G \) Torsional stiffness of the rotor of the testing apparatus \([m/s^2]\)

\( h_0 \) Mean height fluid level \([m]\)

\( h_{\text{eff}} \) Effective mean height fluid level in case of testing a flight model \([m]\)

\( I \) Moment of inertia of the rotor of the testing apparatus \([kg/m^2]\)

\( I_i \) Moment of inertia with respect to \( i \)-axis \([kg/m^2]\)

\( L \) Tube length of damper \([m]\)

\( \hat{L} \) Angular momentum axis of satellite
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<td>$R$</td>
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<td>Angular deviation of the rotor at the end of a testrun [rad]</td>
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<tr>
<td>$\dot{\omega}$</td>
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</tr>
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<tr>
<td>$\omega_{\text{pm}}$</td>
<td>Frequency of the rotor of the testing</td>
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<td>apparatus</td>
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**Appendix A:** Design example  
**Appendix B:** Pascal input files
1. Introduction

At UCN tube-with-endpots nutation dampers are designed and made. In order to be able to design such a damper, software has been developed for the necessary calculations. The theory behind the software was spread about many different reports. For further applications, it is more convenient that all the necessary information is found in one report.

This report is oriented to one kind of passive type nutation damper for single spin satellites. No derivation of the satellite equations of motion will be given. All parameters which follow directly from these equations of motion will be considered postulated.

This report is a collection of the theory. It contains a description of the software and gives an example on how to design a nutation damper including the test-model.

The derivations of the damper theory are all printed in a separate report [12].
2. Satellite motion

2.1. Satellite nutation

The attitude motion of a completely rigid, rotating object in space free of all external forces or torques is considered. In describing this motion, four fundamental axes or sets of axes are important [4].

Geometrical axes are arbitrarily defined relative to the structure of the spacecraft itself. This is the reference system which defines the orientation of the satellite in inertial space. The angular momentum axis $\hat{L}$ is the axis through the centre of mass parallel to the angular momentum vector. The instantaneous rotation axis $\hat{\omega}$ is the axis about which the spacecraft is rotating at any instant. The principal axis is any axis $\hat{P}$ such that the resulting angular momentum is parallel to $\hat{P}$ when the spacecraft rotates about $\hat{P}$.

These four sets of axes may be used to define two types of attitude motion called pure rotation and nutation (fig 1).

---

**Figure 1: two types of attitude motion**

Pure rotation is the case in which the rotation axis, a principal axis, a geometrical axis and the angular momentum vector are all parallel or anti-parallel (fig 1(a)). These four axes will remain parallel as the object rotates.

Nutation is rotational motion for which the instantaneous rotation axis is not aligned with a principal axis (fig 1(c)).
In this case, the angular momentum vector, which remains fixed in space, will not be aligned with either of the other physical axes. Both $\hat{P}$ and $\hat{W}$ rotate about $\hat{L}$. $\hat{P}$ is fixed in the spacecraft. Neither $\hat{L}$ nor $\hat{W}$ is fixed in the spacecraft. $\hat{W}$ rotates both in the spacecraft and in inertial space, while $\hat{L}$ rotates in the spacecraft, but is fixed in inertial space. The angle between $\hat{P}$ and $\hat{L}$ is a measure of the magnitude of the nutation, called the nutation angle, $\theta$.

To obtain a physical feel for nutation, it is noted that in inertial space, $\hat{W}$ rotates about $\hat{L}$ on a cone of half-cone angle $(-\theta)$ called the space cone, as illustrated in figure 2 (left) for $I_1 > I_3$. Similarly, $\hat{W}$ maintains a fixed angle, $\xi$, with $\hat{P}$ and, therefore, rotates about $\hat{P}$ on a cone called the body cone. Because $\hat{W}$ is the instantaneous rotation axis, the body is instantaneous at rest along the $\hat{W}$ axis as $\hat{W}$ moves about $\hat{L}$. Therefore, we may visualize the motion of the spacecraft as the body cone rolling without slipping on the space cone (see figure 2).

Figure 2: Motion of a nutating spacecraft, the body cone rolls on the space cone for $I_1 = I_2 > I_3$ (left) and $I_1 = I_2 < I_3$ (right)
Satellite motion

The space cone is fixed in space and the body cone is fixed in the spacecraft. Figure 2 left is correct only for objects, such as a tall cylinder, for which $I_1$ is greater than $I_3$. If $I_3$ is greater than $I_1$, as in the case of a thin disk, the space cone lies inside the body cone, as shown in figure 2 right.

2.2. Satellite motion due to antenna movement

A satellite can have another nutation-like motion. This motion is caused by the movement of antennas. When a satellite is positioned, the antennas will oscillate due to their moment of inertia and their flexibility. This oscillation of the antennas will cause the satellite to oscillate. Various modes of antenna movement of a satellite with four antennas are illustrated in figure 3 [11].

The symmetric antenna mode will not cause the satellite to oscillate unlike the antisymmetric antenna mode. So the frequency of this latter mode will be important to design the damper.
**Satellite motion**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Description</th>
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<tr>
<td><strong>EA1 Mode</strong></td>
<td>- Equatorial antisymmetric</td>
</tr>
<tr>
<td></td>
<td>- Uncoupled</td>
</tr>
<tr>
<td><strong>EA2 Mode</strong></td>
<td>- Equatorial antisymmetric</td>
</tr>
<tr>
<td></td>
<td>- Uncoupled</td>
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<td></td>
<td>- Local</td>
</tr>
<tr>
<td><strong>MS1 Mode</strong></td>
<td>- Meridian symmetric</td>
</tr>
<tr>
<td></td>
<td>- Uncoupled</td>
</tr>
<tr>
<td><strong>MS2 Mode</strong></td>
<td>- Meridian symmetric</td>
</tr>
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<td>- Meridian antisymmetric</td>
</tr>
<tr>
<td></td>
<td>- Coupled</td>
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*Figure 3: different modes of antenna movement*
Nutation damping

3. Nutation damping [1]

3.1. Why nutation damping?

One method to stabilize the attitude of a satellite in space, is by giving it a rotational movement around the spin-axis. With this rotational movement the satellite gains a gyroscopic stiffness which allows it to be controlled accurately. Since observational instruments need to be aligned to the spin-axis or positioned under a certain angle with this axis for scanning purposes, any unwanted oscillations caused by attitude and orbit control need to be damped out. Therefore such a satellite is equipped with one or more nutation dampers.

3.2. History of nutation dampers

Nutation dampers were first developed for the stabilization of aircraft gyroscopes. Further work on nutation dampers was done for spin-stabilized rockets that were propelled beyond the earth atmosphere where fin stabilization becomes ineffective because of the absence of air. The first nutation damper flown in a missile was built at the Naval Ordnance Test Station, U.S.A., in 1950 for the stabilization of the gyroscope in the Sidewinder missile. This damper consisted of an annular damper that was partially filled with mercury. The sloshing of the mercury dissipated the nutational energy. A similar mercury filled annular damper was used in the Pioneer-I lunar probe in 1958 and became the first nutation damper to be used in space.

Since then many types of nutation dampers have been designed for spin-stabilized spacecrafts ranging in size from small scientific satellites to large communication satellites. Although communication satellites are generally 3-axis stabilized in geostationary orbit, they are spin-stabilized during their transfer orbit.
4. Nutation damper

4.1. Introduction

4.1.1. Oblate versus prolate spacecrafts

A distinction has to be made between oblate and prolate spacecrafts. The spacecraft is oblate and stable, if the spacecraft has a nominal spin-axis which is the principal axis with maximal moment of inertia. Otherwise the spacecraft is prolate and unstable.

When energy is dissipated, the angular momentum vector becomes aligned with the principal axis with the largest moment of inertia. So if a passive nutation damper is used on a prolate spacecraft, the nutation angle will increase as a result of the energy dissipation. The solution for this problem is the use of an active nutation damper [4].

Figure 4: This spacecraft is prolate because the nominal spin axis $P_3$ is the axis with minimum moment of inertia

4.1.2. The coupling between satellite and damper

In this paragraph a short note will be given on the coupling between satellite and damper.

Because the damper mass is small in comparison to the satellite mass, the solution of the equations of motion can be simplified by neglecting the effect of the moving damper mass
Nutation damper on the motion of the satellite body. The motion of the satellite is then decoupled from the motion of the damper mass. The satellite equation of motion can now be solved separately from the damper equation of motion.

\[ \text{torque free rigid body motion} \]

\[ \text{nutation energy of satellite} \]

\[ \text{periodic forcing acceleration on damper mass} \]

\[ \text{change of nutational energy} \]

\[ \text{damper equation of motion} \]

\[ \text{averaged energy dissipation over 1 nutation cycle} \]

\[ \text{nutation angle time history} \]

\[ \text{Figure 5: The energy sink method} \]

With the satellite equations of motion, the nutational energy is calculated as well as the periodic forcing acceleration on the damper mass. The energy, which is dissipated in the damper during one nutation cycle, is calculated by substitution of the forcing acceleration in the damper equation of motion. In this calculation the forcing acceleration is assumed to remain constant. Now the over one cycle averaged dissipated energy is subtracted from the total.
energy to obtain the nutational energy after one (dissipated) nutation cycle.

This method is called the energy sink method and is schematically shown in figure 5. This method is only valid if the nutational cycle time is small in comparison to the total damping time. If the nutation frequency is small, the cycle time will be large and the energy sink method will not be valid.

4.1.3. Equatorial versus meridional configuration

The damper can be mounted in two different configurations: equatorial and meridional (figure 6). The tube of the equatorial damper is curved with the centre of curvature on the spin-axis due to the curvature of the acceleration field [1].

![Figure 6: equatorially (left) and meridionally (right) mounted damper](image)

4.1.4. Tube-with-endpots damper

The tube-with-endpots damper consists of two endpots connected by two tubes (figure 7). The damper is located on the satellite such that oscillatory nutational motion produces an acceleration component along the damping tube. This causes the fluid to move in the lower tube (denoted as fluid tube)
and viscous friction in the fluid converts mechanical energy into heat. This energy conversion provides the damping action [5]. The gas (upper) tube guarantees fluid movement without pressure raise in the gas above the fluid. Required is that no fluid will enter this vapour tube.

\[\text{figure 7: Schematic drawing of a tube with endpots damper}\]

One of the advantages of this damper is its almost zero dead-band. This means damping will occur until nutation angles close to zero are reached.

The damping performance is measured by the power dissipation normalized with the amplitude of the forcing acceleration \((P/a_0^2)\). This performance is a function of the frequency of the forcing acceleration and shows an extreme at its resonance frequency. In general the damper will be designed such that its resonance frequency is close to the middle of the range of expected forcing frequencies.

4.2. Time constant

The decay of the nutation angle can be characterized by its time constant. This time constant can be established unam-
Nutation damper

biguously if the nutation angle changes exponentially with time. It is defined as the time in which the nutation angle has changed with a predefined factor.

In the area of small nutation angles \( \theta \), the time dependence of \( \theta \) can be described with:

\[
\theta(t) = \theta_0 e^{-\frac{t}{\tau}}
\]

in which \( \tau \) is the time-constant.

The time-constant is calculated with [1]:

\[
\tau = \mathcal{F} \frac{1}{\omega_z^2} \left( \frac{P}{a_0^2} \right)^{-1}
\]

where \( \mathcal{F} \) is a function, which depends on the geometry of the satellite and the mounting configuration (equatorial or meridional) of the damper. This function is not damper-dependent and its derivation is beyond the scope of this report, for this one is referred to [1], [2] and [3].

The time-constant is dependent on \( (P/a_0^2) \) as can be seen from (4.2). \( (P/a_0^2) \) is called the performance of the damper.

### 4.3. Performance theory [7]

In order to be able to derive a formula for the damper-performance, the next assumptions are made:

- The fluid is incompressible;
- The fluid is isoviscous;
- The flow is laminar and parallel to the \( z \)-axis of the tube (length-axis);
- A steady state response is regarded;
- The acceleration field is harmonic;
- Entrance/exit effects of the fluid in the endpots are neglected;
- The energy dissipation in the endpots is neglected.

Two different elaborations of the same model have been developed by Fokker and ESA. The first part of both elaborations are basically the same, but some different steps are made in the last parts of the two elaborations. Both elabo-
Nutation damper

Equations are described completely in [12]. Below a qualitative description is given of both models with some important results.

Both models are based upon the Navier-Stokes equation. A solution is derived by separation of variables. With a parameter transformation a Bessel equation of order zero is derived. This Bessel equation has a standard solution. From this point the elaborations of Fokker and ESA differ.

4.3.1. Elaboration Fokker

Fokker describes the fluid velocity. With the matching boundary condition, the fluid velocity as a function of time and radius is derived [12]:

\[ u_z(r,t) = \frac{a_0}{\sqrt{\Omega}} \left( \frac{\gamma^2}{\gamma^2 + \beta} \right) \left( \frac{J_0(\epsilon)}{J_0(\epsilon_a)} - 1 \right) e^{-i\Omega t} \quad [m/s] \]  

(4.3)

with \( \omega_0 = a \sqrt{\frac{2q}{L}} \), \( \gamma = \frac{\Omega}{\omega_0} \), \( \nu = \frac{\mu}{\rho} \), \( \beta = \frac{2J_1(\epsilon_a)}{\epsilon_a J_0(\epsilon_a)} - 1 \),

\[ \epsilon_a = \frac{a}{\sqrt{1/\gamma}} \quad \text{and} \quad \epsilon = \frac{r}{\sqrt{1/\gamma}}. \]

in which \( \omega_0 \) is the resonance frequency of the damper.

Next a formula for the power dissipation of the damper is derived using the real part of the velocity. This results in the next equation:

\[ P = \frac{\pi \rho L a^2 \gamma^4}{a_0^2} \left| (\gamma^2 + \beta) J_0(\epsilon_a) \right|^2 \Omega \int_0^1 |J_1(\xi \sqrt{1/\gamma})|^2 d\xi \quad [kg.s] \]

(4.4)

with \( \xi = \frac{r}{a} \), and \( \sigma = \frac{a^2 \omega_0}{\nu} \).

This is an equation for the power dissipation averaged over one cycle.

An expression for the fluid displacement is derived integrating the fluid velocity. The expression for the amplitude of the fluid displacement is:
This amplitude, which represents the maximum fluid displacement in the endpot, is of importance for the calculation of the distance between the fluid-tube and the vapour-tube.

4.3.2. Elaboration ESA

ESA describes the absolute fluid velocity. With the according boundary condition, the fluid velocity as a function of time and radius is derived as [12]:

\[
\begin{align*}
    u_x(r, t) &= \Re \left\{ \frac{F_x \left[ \Omega^2 J_0(\epsilon) + \omega_0^2 J_2(\epsilon_a) \right] e^{-i \omega t}}{\Omega^2 J_0(\epsilon_a) + \omega_0^2 J_2(\epsilon_a)} \right\}
\end{align*}
\] (4.6)

The formula for the power dissipation is derived using the formula for the fluid velocity. The integral over the tube area is transformed into the integral over the wall by utilizing the theorem of Stokes. This results for the power dissipation:

\[
\frac{P}{a_0^2} = -\frac{\pi}{2} a^2 \rho \Re \left\{ \frac{i \Omega J_2(\epsilon_a)}{\omega_0^2 J_2(\epsilon_a) + \Omega^2 J_0(\epsilon_a)} \right\}
\] (4.7)

ESA didn't derive an equation for the fluid displacement.

4.3.3. Comparison of the Fokker elaboration versus the ESA elaboration

Fokker describes the fluid-velocity relative to the fluid tube whereas ESA describes the absolute velocity, but this difference is only expressed in the reference system in which all variables are described. A different reference system doesn't have any influence on the power-dissipation.

Fokker calculates the power-dissipation with an integral with respect to the tube radius, where ESA calculates the power-dissipation with a boundary condition for the fluid velocity.

Both models give a different expression for the damper-performance, but because the models are based on the same
Nutation damper

theory, the two expressions should calculate the same results. The software check of the computer programs show that the models calculate equal results (see chapter 6.3).

4.4. Simultaneous excitations

In this paragraph the model is extended to simultaneous harmonic excitations. Such a state of composed motion may occur in space, in case antenna modes and nutation modes are acting at the same time. For instance, at the early operational phase directly after the antenna deployment. The different acceleration fields of each motion may simply be superimposed. In [12] the theoretical derivations are reported.

The extension of the model uses the elaboration according to Fokker. A superposition of different acceleration fields is executed. These acceleration fields differ in amplitude and in frequency. The dissipated power is calculated as a function of time, because the energy sink method is not valid for all cases. When two accelerations with almost equal cycle times act simultaneously, the resulting cycle time can become large in comparison to the total damping time. So the power dissipation is not averaged over one cycle. The next equation results:

\[ P(t) = 2\pi \rho L a^2 \sum_i \sum_j a_{ij}^2 \int_0^1 P_{ij}(\xi) \xi d\xi \]  

In this equation \( n \) is the number of different excitations. \( P_{ij}(\xi) = C_{rr}(\xi) + C_{ri}(\xi) + C_{ri}(\xi) + C_{rr}(\xi) \)

\( \rho \) is the dissipated power for simultaneous excitations with excitation \( i \) and \( j \).

\[ C_{rr} = \frac{\gamma^2 \gamma^2 J_1(\xi e_{a_i}) J_1(\xi e_{a_j})}{2 \sqrt{\Omega_i \Omega_j (\gamma^2 + \beta^2)(\gamma^2 + \beta^2) J_0(\xi e_{a_i}) J_0(\xi e_{a_j})}} \cos(\Omega_i t) \cos(\Omega_j t) \]

\[ C_{ri} = \frac{\gamma^2 \gamma^2 J_1(\xi e_{a_i}) J_1(\xi e_{a_j})}{2 \sqrt{\Omega_i \Omega_j (\gamma^2 + \beta^2)(\gamma^2 + \beta^2) J_0(\xi e_{a_i}) J_0(\xi e_{a_j})}} \sin(\Omega_i t) \sin(\Omega_j t) \]
In order to be able to compare the dissipated power with the performance at single excitation, the power-dissipation is now modified into a representative performance function [8]:

\[
P = \sum_{i=1}^{n} \sum_{j=1}^{n} P_{ij}(a_{0i}, a_{0j}, f_{m_i}, f_{m_j}) \frac{\sum_{i=1}^{n} a_{0i}^2}{a_0^2} \tag{4.9}
\]

With this function the performance can be calculated as a function of time and amplitude of the forcing acceleration.

4.5. Effective tube length

The tube length is defined as the length between the two endpoints in the previous paragraphs. But this is not the length experienced by the fluid, since it will not stop its forward movement immediately after the tube is left. In practice the fluid will move a bit further as if there was still a tube present. (see figure 8)

In order to account for this effect the effective tubelength is defined as:

\[
L_{\text{eff}} = L + 2\alpha b \tag{4.10}
\]

in which \(\alpha\) is a factor which defines a measure for the distance in the endpoint that is used to leave the tube. This factor can be found empirically and is found to be \(0 < \alpha < 1\).
**4.6. Fluid surface tilt**

In the previous paragraphs it is assumed that the fluid surface in the endpot remains parallel to the tube-axis. In practice this is beyond the truth. The nutation of the spacecraft causes the fluid surface to tilt. This is illustrated in figure 9.

![Figure 9: Fluid surface tilt](image)

In this figure $a_z$ denotes the component of nutational acceleration along the damping tube and $a_p$ denotes the acceleration which is experienced as a result of the spin velocity of the spacecraft. The fluid surface will be normal to the resulting acceleration.

A tilted fluid surface effects the pressure at the end of the tube. The pressure at the entrance of the damping tube is reduced for the pot with the lower mean level (negative net mass flux) and increased for the pot with the upper mean level (positive net mass flux) (see figure 10). Thus for a given forcing acceleration $a_0$, the pressure gradient influenced by fluid surface tilt will be larger. The derivation of the effect is given in [12].

![Figure 10: Fluid surface tilt in two endpots](image)

Due to the fluid surface tilt the driving force is gained with the factor:
Nutation damper

\[
\left(1 + \frac{2b}{L_{\text{eff}}} \right)
\]  \hspace{1cm} (4.11)

in which \(L_{\text{eff}}\) is the effective tube length.

Since \(a_0\) is linearly dependent on the driving force, the expression of the damper-performance \((F/a_0^2)\) has to be multiplied with

\[
\left(1 + \frac{2b}{L_{\text{eff}}} \right)^2
\]  \hspace{1cm} (4.12)

The performance increases by the tilted fluid surface when it is referred to the nominal acceleration.

The fluid displacement has to be multiplied with:

\[
\left(1 + \frac{2b}{L_{\text{eff}}} \right)
\]  \hspace{1cm} (4.13)

4.7. Matching the theory with practice [8]

The model as described above is not exact. Some deviation appears between the calculated results and the measured results. In order to match the theory with practice, two match factors are introduced: \(\alpha\) and \(n\). These are based on effective length and fluid surface tilt. The match factor \(\alpha\) is already described in paragraph 4.6. The match factor \(n\) is derived from the fluid surface tilt. The factor with which the performance is multiplied is changed into:

\[
\left[1 + \frac{2b}{L_{\text{eff}}} \right]^n
\]  \hspace{1cm} (4.14)

with: \(L_{\text{eff}} = L + 2ab\)

By adjusting the match factors the theoretical performance is matched with practice. A change in \(\alpha\) shifts the resonance frequency of the damper and a change in \(n\) changes the magnitude of the power dissipation. How the match factors are derived is outlined below.

First a damper is designed with both match factors arbitrarily chosen. This damper is tested. The test results are compared with the calculated results and the match factors are
calculated. Then the damper is redesigned with the new factors and a test is performed again. The cycle is conducted until no significant difference is found between the test results and the calculated results.

The match factors will be different for every other damper.

4.8. Conclusion

As can be seen from the equations for the performance, according to this model, the performance is not dependent on the nutation angle $\theta$.

Due to an excitation with simultaneous accelerations, the calculation of the performance becomes dependent on the amplitude of forcing acceleration. This is due to the coupled acceleration fields.

Both the effects fluid surface tilt and effective tube length have an increasing effect on the performance. Besides the effective length induces a shift of the resonance frequency to lower frequencies.

Results from this model were compared with scale model test results and the agreement was found to be excellent. Of course this is due to the match factors which match the theory with practice.
Performance tests

5. Performance tests

5.1. Introduction
In order to verify whether the damper meets the requirements, it has to be tested. In this chapter a description of the test-facility used for the two dampers of the Cluster-satellite is given, the definition of the scale model is made and the test parameters are defined.

5.2. Test facility
5.2.1. Test equipment

Figure 11: Topview of the testing apparatus. On each side of the rotor a damper is mounted. Two masses placed on top of the rotor are used for adjustment of the moment of inertia.

The damping performance of the PTM's is determined with a testing apparatus sketched in figure 11. The testing apparatus consists of an airbearing borne rotor which rotates around the vertical axis. The damper is mounted on the rotor at a certain distance from the axis of rotation. By giving the rotor a spring controlled oscillating motion, the motion of the damper mounted on the satellite is simulated.

5.2.2. Test procedure
At the beginning of a testrun, the required pendulum time is adjusted and the rotor is given a predefined amplitude. The motion of the rotor is recorded until the required minimum angular amplitude has been reached. Then the damping is calculated from the recorded decrease of amplitude in time.

The damping thus found represents the damping due to the damper and the test equipment itself. In order to determine the damping performance of the damper alone, an additional run
Performance tests

(dummy run) is made, during which the damper is replaced by a dummy weight with the same mass as the damper. Finally the performance of the damper is found by subtracting the performance of the dummy run from the damped run.

5.3. Definition of the performance test models

5.3.1. Scaling

In the testing apparatus the centrifugal acceleration of the satellite is replaced by gravity, causing the need to scale the flight models into performance test models. The purpose of scaling is to gain comparable fluid behaviour for flight model and performance test model in the different environments of earth and orbit. For this comparable fluid behaviour a straight tube of the damper is necessary due to the rectangular shape of the gravitation field. The scaling theory is outlined in [12].

5.3.2. Scaling relations

In [12] the scaling relations for important parameters are derived. The results are listed below. The index \( f_m \) denotes flight model and the index \( ptm \) denotes performance test model.

Scale factor for the geometry:
\[
\frac{L_{ptm}}{a_{fm}} = \frac{L_{ptm}}{L_{fm}}
\]  

(5.1)

Scale factor for the pendulum time:
\[
\frac{T_{ptm}}{T_{fm}} = L_f^2
\]

(5.2)

Scale factor for the forcing acceleration:
\[
\Psi_{\text{max}} = \frac{a_{0f}}{a_{0}} \left( R - h_0 \right)
\]

(5.3)

Scale factor for the power dissipation:
\[
\left( \frac{P}{a_0^2} \right)_{ptm} = L_f^5 \left( \frac{P}{a_0^2} \right)_{fm}
\]

(5.4)
5.3.3. Testing apparatus performance calculations

Time constant

The time constant \( \tau \) is a key parameter in the result calculation of a test run. From \( \tau \) the performance \( P/a_0^2 \) is calculated [12].

The time constant \( \tau \) can be expressed as:

\[
\tau^{-1} = -\frac{1}{\psi_{\text{max}}} \frac{d\psi}{dt}
\]

(5.5)

The time constant can be calculated with this equation from the test results, since \( \psi \) is recorded versus time.

Performance calculation

The power dissipation \( P/a_0^2 \) can be calculated from the measured \( \tau \), using the energy-sink method. In [12] an expression for \( P/a_0^2 \) is derived. The resulting expression for the power dissipation function yields:

\[
\left( \frac{P}{a_0^2} \right)_{\text{ptm}} = \frac{I}{\tau} \left( \frac{T_{\text{ptm}}}{2\pi R_p} \right)^2
\]

(5.6)

5.4. Test set-up analysis

5.4.1 Minimum duration of a testrun

Because it is required that the transient response is vanished at the end of a testrun, the testrun has a minimum duration. This minimum duration can be calculated by requiring that at the end of a testrun, the transient response is negligibly small in comparison with the steady state response. The minimum duration is derived in [12] and results:

\[
T_{\text{test}} > 2 \frac{a^2}{3\nu} \ln \left( \frac{a_{\text{min}}}{a_{\text{max}}} \right)
\]

(5.7)

In order to establish this minimum duration, this criterium is rewritten in the more useful form:

\[
\frac{T}{R_p} > 2 \frac{P}{3\nu} \left( \frac{2\pi a}{T_{\text{ptm}}} \right)^2
\]

(5.8)
5.4.2. Torsional stability of the rotor

The ideal testrig should be infinitely stiff in order to exclude unwanted oscillation phenomena, but in practice the rotor twists due to the moment exerted by the dampers. In figure 12 it is shown that the twist of the rotor effects the acceleration $a_0$ along the fluid tube with the component $g \sin(\delta)$. A criterium for the torsional stability of the rotor can be found by the requirement that this component must be smaller than one percent of the maximum forcing acceleration:

$$g \sin(\delta) < 0.01a_{\text{max}}$$

(5.9)

An expression for the required torsional stiffness $G$ is derived in [12].

$$G > 100 \pi \rho g^2 b_m^2 (L_{fr} + 2b_{fr}) c_{fr} L_r^4$$

(5.10)

5.4.3. Mounting radius range

The acceleration field to which the PTM's are exposed changes with the mounting radius $R_f$ of the dampers. The minimum mounting radius is prescribed by a criterium based on the quotient of the centrifugal and the tangential acceleration of the rotor. By requiring

$$\left| \frac{a_c}{a_t} \right| < 0.01$$

(5.11)

and by use of equation (5.3) a criterium for $R_f$ can be obtained

$$R_f > 100 \frac{a_{\text{max}} (R-h_0)}{g L_r^2 f_m^2}$$

(5.12)
When a flight model is tested on earth, the damper geometry might need to be modified if the performance is calculated. Due to the rectangular shape of the gravitation field on earth, the fluid surface on earth differs from the one in orbit. The fluid surface in the endpots is now tilted, see figure 13. This will effect the tuning of the dampers and moreover the allowable fluid displacement in the endpots. The theoretical predictions must therefore be corrected for these geometry changes and the tests must be carefully defined to avoid fluid entering the vapour tubes.

Figure 13: Tilt of the fluid surface in the endpots

Mounted on the testing apparatus, the fluid surface in the circular endpot becomes elliptically. For performance calculations this elliptical endpot surface must be translated into an effective circular endpot area with endpot radius $b_{eff}$. Also the allowable fluid displacement $s$ and the distance $h_0$ between the centre of the fluid tube and the fluid surface are changed.
6. Computer implementation

6.1. Introduction

At UCN two different programs, which calculate the damper-performance, are available. One is made by Fokker and is written in the language Fortran. The other is made by UCN and is written in the language Pascal. In this report a description is given only of the Pascal program of UCN, which is moreover a software package. This package didn’t give accurate results at first, so it was improved until the results were correct.

With this software package calculation of the damper-performance is possible, together with calculation of the fluid-displacement in the endpots, the fluid-velocity and the time-dependent performance and fluid-displacement at simultaneous excitations. Also calculation of a scaled test-model with its scaled behaviour is possible.

In appendix A an example is given on how to design a damper with this software.

6.2. Software package

6.2.1. Units

The improved software package is built around three units. The first unit contains basic mathematic procedures and functions, the second unit takes care of the properties of the damper and the third unit contains the performance-specific procedures and functions.

6.2.2. Program

Together with these three units there are at the moment two programs. One to design the damper and the other to design the performance test model and to calculate the test apparatus performance criteria. These two programs use procedures which are listed in the units.

The performance is calculated in two different manners. The formula of ESA and the formula of Fokker are implemented. The performance is calculated as a function of the frequency-ratio
\(\Omega/\omega_n\). Necessary for these calculations are three Bessel-functions \(J_1\), \(J_2\) and \(J_3\). These Besselfunctions are infinite series and can be represented by a sum. The sums which have to be used in these programs are derived in [12]. The fluid-displacement is calculated as a function of \(\Omega/\omega_n\) and the fluid-velocity is calculated as a function of the tube radius \(r\). The formulas which are used in this program to calculate the velocity and the displacement are based on the Fokker elaboration. The simultaneous excitations performance is calculated as a function of time.

The calculation of the performance test model is based on the theory outlined in chapter 5. This program calculates the scale model geometry and the performance in test-conditions. It also calculates the requirements of the test-apparatus.

### 6.2.3. Input

The input of the damper-design program is separated in two files. One file contains the values of the geometry of the damper, the other file the values for the excitation that the damper experiences. Examples of these files are given in appendix B.

The input of the PTM-program is separated in two files. One file contains the flight model information. This is the same file which is used for the damper-design program. The other program contains the requirements which define the test series for the PTM. Examples of these files are given in appendix B.

### 6.2.4. Output

The output of all programs is written in columns of ascii-text characters. These columns can be read by standard programs in order to draw diagrams of the performance, fluid-velocity etc.
6.3. Software check

In order to verify the correctness of the software it needs to be checked. Five different checks were performed.
- Check of the damper performance according to Fokker;
- Check of the damper performance according to ESA;
- Check of simultaneous excitations;
- Check of fluid-velocity;
- Check of PTM calculation.

The first issue of the Pascal program made by UCN, was checked, modified and improved. After this improvement the results were equal with the results of the Fokker program. In order to verify whether the agreement of the two programs was not coincidental, the check was extended for different dampers in different circumstances. All the results showed satisfactory agreement. The fluid-displacement and the forcing acceleration were checked in the same manner.

The calculation of the performance according to the ESA elaborations was checked by comparison with the newest (improved) UCN program. Also this check was performed for different dampers and different circumstances. The check showed that the two methods calculate equal results for the performance.

The calculation of simultaneous excitations was checked by using equal frequencies for the different excitations. So it could be compared since the total amplitude of the different excitations was made equal to an amplitude at which the performance was calculated with the single excitation performance routine. The average performance calculated with simultaneous excitations must be equal to the performance calculated with single excitation. This check showed that the results were equal.

The calculation of the fluid-velocity could not be verified with another computer program, because such a program was not available. The velocity profiles were printed and similar profiles were found in literature [13] [14].
The check which was used for the program which calculates the PTM, was performed by setting the scale-factor to 1. The results from the PTM should be equal to the results calculated with the normal performance routine. This check was performed and the results agreed.

All these checks show that the Pascal program package calculates correct results for the different functions. All the results agreed within at least 1%.
7. Conclusion and recommendations

The updated UCN program proved to be reliable. It calculates the damper-performance correctly. With this program the next calculations can be performed.

- Damper-performance as a function of the frequency ratio $\Omega/\omega_n$,
- The forcing acceleration as a function of the frequency ratio in case of nutation (satellite dynamics),
- Fluid displacement as a function of the frequency ratio,
- Fluid velocity as a function of the tube radius,
- Damper-performance and fluid displacement as a function of time for simultaneous excitations.

When the test results are compared with the theory, a difference can be remarked. The theory predicts a constant performance as a function of the forcing acceleration, where the test results show an increasing or decreasing performance. The values of the test results for small forcing accelerations are equal to the predicted results.

In order to predict the performance at large forcing accelerations it is recommended to regard the next points:
- As an entrance effect only the effective tube length is reckoned with.
- At large nutation angles the flow becomes turbulent.
8. Literature


Appendix A: Design example

Introduction
In this chapter an example is given of the complete damper design trajectory, including the design of the performance test model and the definition of the test plan. This example is based on the design of the dampers for the Cluster-satellite. These dampers were designed to damp out the movements of the satellite caused by the antenna and nutation modes.

Performance requirements
The performance requirements define the minimal performance for a damper on a given frequency range and are drawn in figure A.1.

\[
\begin{array}{c|c|c|c}
P/\alpha & \text{EA} & \text{MA} \\
\text{[kg.s]} & & \\
\end{array}
\]

\text{frequency-ratio}

\text{figure A.1: performance requirements}

This figure shows that two dampers are needed to meet the requirements. Each damper takes care of one mode (EA or MA). Together they need to take care of the middle frequency range (N-mode).

An other particularly important requirement is that the mass of each damper shall be minimised. The dampers are tuned by adjustment of their endpot radius, so only the fluid tube radius and the fluid tube length are free to be chosen in the
design process. The fluid tube radius influences the power-dissipation to the fourth power.

\[ \frac{P}{a_0^2} = a^4L \]

This equation shows that L should be maximised if possible, to obtain the highest \( \frac{P}{a_0^2} \). Besides, a long fluid tube will lead to a minimised mass as well (see figure A.2)

![Figure A.2: The mass and endpot radius of the tuned dampers versus the tube length L.](image)

Because the tube length L was limited up to 278 mm, this length was chosen for the dampers.

**Design trajecotry**

The design of a damper was conducted by following the flow-chart in figure A.3. Started was with matchfactors arbitrarily chosen at zero. Then a damper and its PTM were designed. Because the theory didn’t model the test results, the matchfactors were obtained by curve fitting. With these match factors a new damper was designed and the cyclus was followed until the theory agreed with the test results and the requirements were met. The separate actions which were taken are described in separate chapters.
The dampers were designed by varying the design parameters until the performance was just a little above the required value. The required performance was given in a range of frequencies (see figure A.1). The damper was tuned at this range, which means that the peak of the performance curve is in this range. The performance will be above the required value when the performance at the two edges of the frequency range are above the required values. Now a damper was designed by "designing" the performance of the two edge-frequencies above the requirements.

The influence of the design parameters on the performance is given in figures A.4 to A.7.
With the aid of these figures iteratively the damper was designed by respectively changing parameters, calculating the performance and comparing the requirements with the calculated values. It is convenient when first the damper is tuned to the right frequency and then the height of the curve is changed.

As can be seen from figure A.7, the fill level has little influence on the performance. So it can't be used as a design parameter. The fill level has a minimum requirement which is based on the theoretical worst case fluid displacement in the endpots for a damper subjected to one mode at the time. Extra fluid needs to be added to compensate for the reduction of the effective fluid level in the endpots caused by the tilted endpots $s_{e}$, the fluid surface tilt $s_{f}$ and the alignment tolerance $s_{a}$:

$$h_{0} = a + s_{max} + s_{e} + s_{f} + s_{a}$$
Design example

In the example the next damper set resulted:

<table>
<thead>
<tr>
<th></th>
<th>a [mm]</th>
<th>b [mm]</th>
<th>h₀ [mm]</th>
<th>L [mm]</th>
<th>R [m]</th>
<th>α</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>EA</td>
<td>7.0</td>
<td>48.6</td>
<td>14.0</td>
<td>278.0</td>
<td>1.37</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>MA</td>
<td>14.0</td>
<td>38.2</td>
<td>36.5</td>
<td>278.0</td>
<td>1.37</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

*Table 1: first damper set in the iteration process*

The calculated performance curve is shown in figure A.8. In this curve the requirements are drawn as straight lines.

**Total performance**

![Total performance graph](image)

*Figure A.8*

Also the fluid displacement and fluid velocity as a function of the radius were calculated. These are shown in figures A.9 to A.11.
Design example A.6

Fluid displacement

![Graph of fluid displacement](image)

**Figure A.9**

**Fluid velocity, EA-damper**

![Graph of fluid velocity, EA-damper](image)

**Figure A.10**

**Fluid velocity, MA-damper**

![Graph of fluid velocity, MA-damper](image)

**Figure A.11**

The performance and displacement as a function of time at simultaneous excitations are given by figures A.12 and A.13.

**Performance at multiple excitations**

![Graph of performance at multiple excitations](image)

**Figure A.12**

**Fluid displacement at multiple excitations**

![Graph of fluid displacement at multiple excitations](image)

**Figure A.13**
PTM design

As a result of the different environmental conditions in space and on earth, a scale model needs to be designed in order to be able to test the damper. The scale factor for the test model geometry needs to be defined and was calculated with:

\[ L_x = \sqrt[3]{\frac{(R-h_0)\omega_x^2}{g}} \]

The dimensions and characteristics of the PTM were calculated. The PTM geometry is given in table 2.

<table>
<thead>
<tr>
<th></th>
<th>a [mm]</th>
<th>b [mm]</th>
<th>h₀ [mm]</th>
<th>L [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>EA</td>
<td>5.0</td>
<td>34.7</td>
<td>10.0</td>
<td>198.6</td>
</tr>
<tr>
<td>MA</td>
<td>10.0</td>
<td>27.3</td>
<td>26.1</td>
<td>198.6</td>
</tr>
</tbody>
</table>

Table 2: PTM geometry of the first damperset in the iteration process

It is obvious that the damper couldn't be tested at all the frequencies where it was calculated. So a selection of frequencies had to be made. Because the damper had to perform well at the required frequencies, it was convenient to chose frequencies in these ranges as testing frequencies. Most of the testing frequencies were situated around a peak in the performance curve. Because the testing apparatus exhibits internal friction, which causes extra damping, three dummy-runs had to be made. See chapter 5.2.
Design example

Test results

The test-results showed a tendency which is illustrated in figure A.14.

Horizontally the amplitude of the damper displacement (which is a measurement for the forcing acceleration) is plotted and vertically the measured damper performance. Curve 1 was measured at a frequency ratio higher than that of the peak in the performance curve (see figure A.8). Curves 2, 4 and 7 were measured on top of the performance curve. Curve 8 and 9 were measured at a lower frequency ratio. When the damper results are compared with the calculated results, they need to be scaled back. Now a surface plot can be made of the performance dependent on two parameters, being the forcing acceleration and the frequency ratio. Such a plot is given in figure A.15. In this plot only the MA-mode is drawn.
Design example

After testing the performance proved to be different from the calculated performance, so the two new match-factors had to be calculated. Then the whole design trajectory was conducted again. This was done in the same manner as the first design sequence. Thus a set of two match-factors was chosen, the new performance was calculated and the results were compared with the test-results. This sequence was repeated until the damper just met the requirements.

Final design

After repeating the design trajectory, the final design of the flight model was determined at:
Conclusion

In this appendix it is shown how a damper is designed.
- Collect and interpret the performance requirements;
- Design a damper with arbitrarily chosen matchfactors;
- Design its performance test model;
- Test the performance test model;
- Check theoretical results against test results;
- In the event of no sufficient agreement, curve fit new match factors and repeat the design sequence;

The final damper set which was calculated in this example proved to perform well in the qualification tests and will be mounted on the Cluster-satellite in 1993.
Pascal-inputfiles

Appendix B: Pascal inputfiles

Damper inputfile

EA-damper
  equatorial
  0.00  alfa
  0.00  mn
  7.00e-3  a
  48.6e-3  b
  14.0e-3  h0
  278.0e-3  L
  1370.0e-3  R
  0.5  z
  15  sr
  3  fluid

MA-damper
  equatorial
  0.00  alfa
  0.00  mn
  14.00e-3  a
  38.2e-3  b
  36.5e-3  h0
  278.0e-3  L
  1370.0e-3  R
  0.5  z
  15  sr
  3  fluid

Performance requirements inputfile

0.311  fn_start  , P/a0^2 plot x-axis: fn_start to fn_end
1.120  fn_end
29  N_points  , curve grid of P/a0^2 graph
1.08  fv  , frequency ratio of fluid velocity calc.
100  R_points  , curve grid of velocity graph
28  T_test  , temperature of test
0.010472  Theta  , initial nutation angle
1000  N_transient  , curve grid of transient graph
4  N_freq  , number of simultaneous excitations
0.422  fn(1)  , a0[1], frequency & amplitude of
0.898  fn(2)  , a0[2], simultaneous excitations
1.024  fn(3)  , a0[3],
1.087  fn(4)  , a0[4],

!!!!!!!Filenames and dampertype in damperprogram!!!!!!!
### PTM requirements inputfile

**EA-damper**

<table>
<thead>
<tr>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00500</td>
<td>a, tube radius of PTM</td>
</tr>
<tr>
<td>2.00</td>
<td>Ra, mounting radius on rotor</td>
</tr>
<tr>
<td>190</td>
<td>Ip, moment of inertia of rotor</td>
</tr>
<tr>
<td>9.81</td>
<td>g, gravity</td>
</tr>
<tr>
<td>20</td>
<td>T_test, temperature of test</td>
</tr>
<tr>
<td>5</td>
<td>N_points, Number of tests</td>
</tr>
<tr>
<td>3</td>
<td>Noa, Number of modes</td>
</tr>
<tr>
<td>1.0</td>
<td>fn_start, Total frequency range from</td>
</tr>
<tr>
<td>1.1</td>
<td>fn_end, fn_start to fn_end</td>
</tr>
<tr>
<td>0.010472</td>
<td>theta, initial nutation angle</td>
</tr>
<tr>
<td>0</td>
<td>2.8e-5, 6.43e-3, freq, a0min, a0max</td>
</tr>
<tr>
<td>0.5</td>
<td>1.97e-4, 1.17e-1, frequencies at which the new acceleration</td>
</tr>
<tr>
<td>1.0</td>
<td>1.8e-4, 2.42e-2, requirements start and the acc. requirements</td>
</tr>
</tbody>
</table>

**MA-damper**

<table>
<thead>
<tr>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01000</td>
<td>a, tube radius of PTM</td>
</tr>
<tr>
<td>2.00</td>
<td>Ra, mounting radius on rotor</td>
</tr>
<tr>
<td>190</td>
<td>Ip, moment of inertia of rotor</td>
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<tr>
<td>9.81</td>
<td>g, gravity</td>
</tr>
<tr>
<td>20</td>
<td>T_test, temperature of test</td>
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<tr>
<td>5</td>
<td>N_points, Number of tests</td>
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<tr>
<td>1.0</td>
<td>fn_start, Total frequency range from</td>
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<td>fn_end, fn_start to fn_end</td>
</tr>
<tr>
<td>0.010472</td>
<td>theta, initial nutation angle</td>
</tr>
<tr>
<td>0</td>
<td>2.8e-5, 6.43e-3, freq, a0min, a0max</td>
</tr>
<tr>
<td>0.5</td>
<td>1.97e-4, 1.17e-1, frequencies at which the new acceleration</td>
</tr>
<tr>
<td>1.0</td>
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