ANTENNA POINTING MECHANISM WITH INTEGRATED HOLD DOWN AND RELEASE FUNCTION

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ABSTRACT

EADS Astrium has developed an Antenna Pointing Mechanism (APM) for Geo-stationary orbits which operates successfully in space since December 2003. Typically launch loads are partly or completely off-loaded from the APM with Hold Down and Release Mechanism HD(RM.
For small antennas in the range of up to 2 kg it was the objective to completely eliminate the HDRM with the help of a **P**re-loaded **M**echanical **E**nd-stop (PME) that would together with the APM actuators serve as a secondary load path. The PME is loaded and released with the actuator torque.

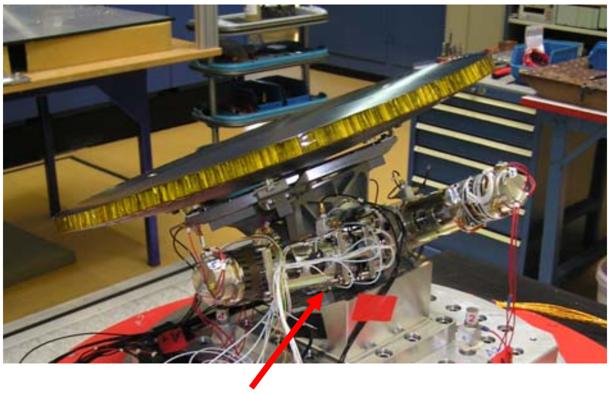

Fig. 1: APM with PME Vibration Test Set-up

- Unlimited repeatability of the lock- and un-lock function
- No mechanical interaction with the unit for locking and un-locking

The paper will present the development process from initial simplified tests, the design and analysis process up to the complete dynamical verification.
Specifically it will concentrate on:
- Mathematical models and model correlation to simulate the actuator's static and dynamic lock- and unlock function (Matlab)
- the APM entire dynamic behaviour (FEM)
- The effect of accepted gapping at the PME when exposed to launch loads on the unit's damping characteristic
- PME cup cone surface effects
- Lessons learned wrt:
  - mechanism characterisation in dynamic environment
  - prediction of system behaviour and holding loads with mathematical models
  - errors in the development process

1. INTRODCUTION

The key advantage of an APM with integrated hold down and release function would be:
- Simple APM to S/C I/F configuration
- Increased reliability
- No additional hold down and release mechanism
- No release shock
- No pyro harness

2. GENERAL IDEA AND INITIAL TESTS

The general idea of a "self locking" APM came from the un-powered self locking capability of the existing APM actuators. If the associated loads would be applied to push the APM in both rotation directions against a mechanical end-stop which must be physically located in between both rotation axes it should stiffen the system such that it could sustain typical launch loads for small antennas. This idea has been verified in a simple test by pushing one of the APM gimbal corners with the help of a bar to the PME (Fig. 2) which was built up from a steel ball acting against a brass cup (both with MoS2 surface, Fig. 3).

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The measured first and critical Eigenmode (60Hz) was well met by the mathematical model (FEM) and the load prediction from this model were in the range of acceptable loads levels for the APM and the actuators.

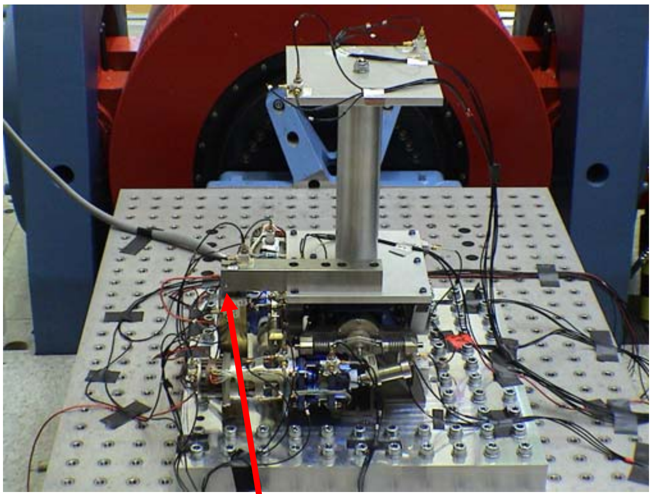

Fig. 2: APM PME initial verification model

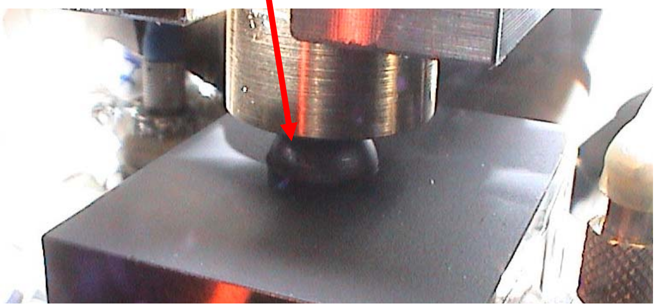

Figure 3: Mechanical End Stop

3. THE APM ENGINEERING MODEL

According to these positive test results an APM QM with integrated PME has been designed, analysed and built in order to verify the function of the PME and the dynamic system behaviour completely for all three vibration axes (Fig. 1). The PME was located at the lower end of one of the rotating brackets (Fig 1 and 4). This location had the advantage that only minimum design changes were required to the existing APM but the disadvantage that the PME location was relatively close to the APM pivot thus generating high PME loads requiring high actuator pre-loads to prevent or minimise gapping. The PME itself was designed and built as in the previous model as a brass sphere acting against a steel ball, both coated with MoS_2 (Fig. 5 and 6).

In order to simulate the dynamical behaviour of the APM a Finite Element Model (FEM) has been built up in NASTRAN.

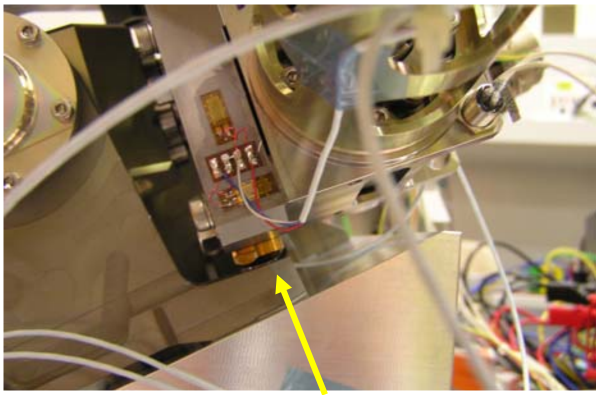

Fig. 4: PME Position (Detail)

Fig. 5: PME Ball (steel with MoS_2 - polished)

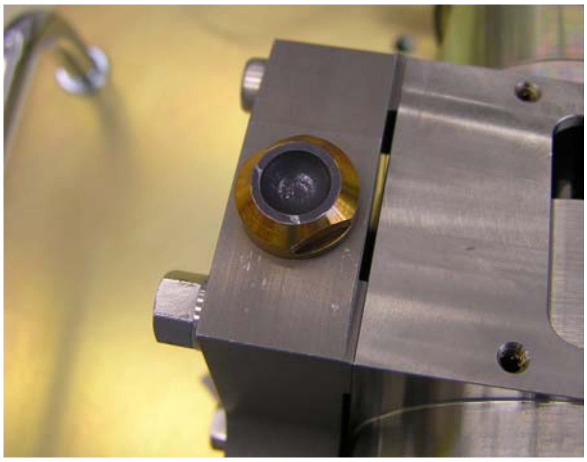

Fig. 6: PME Cup (Brass with MoS_2 - sprayed)

The PME was simulated as a contact point that allowed transfer of vertical and lateral loads. In principal this interface cannot transfer tension loads as it would gap. Gapping principally should have been avoided by applying sufficient pre-load that must be larger than the maximum expected load from sine and random excitation.

The load prediction to prevent gapping from the FEM was 780N at the PME equivalent to 1267N on each actuator.

This model has been equipped with an Antenna reflector and exposed to vibration levels in the range of 7g sine and 13g RMS random in all three axes (Fig. 1).

The following observations have been made during the vibration test or have been derived by correlation of the measured accelerations with the FEM results:

- The theoretical back driving limit of the actuators under vibration (no friction) of 1,400N at actuator level was exceeded by 1,800N.
- The APM remained stable and the actuator did not rotate during the vibration test thus did not release the pre-loading on the PME.
- The PME load was exceeded by 860N. So gapping occurred
- The cup/ball interface at the PME remained in good shape
- The APM could be "unlocked" with nominal power

These observations shall be discusses more in detail:

The Cup-Cone Interface showed wear or "hammering" effects on the surfaces but the layer itself was not removed (Fig. 7 and 8). Around the ball MoS_2 material was deposited to an extent not acceptable for a customer. Due to the selection of the steel/brass material combination even a complete removal of the MoS2 surface would not cause cold welding effects and thus in principal the coating on the brass cup would not be required.

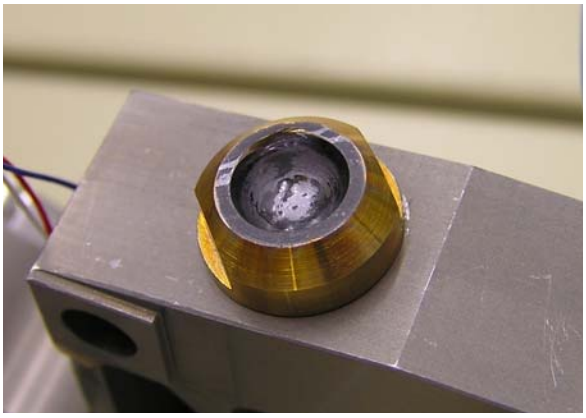

Fig. 7: Cup Surface after Vibration Test

Both the extrapolated PME loads as well as the PME visual inspection supported the assumption that the PME gapped. This can also be derived from the sine response plots (Fig. 9).

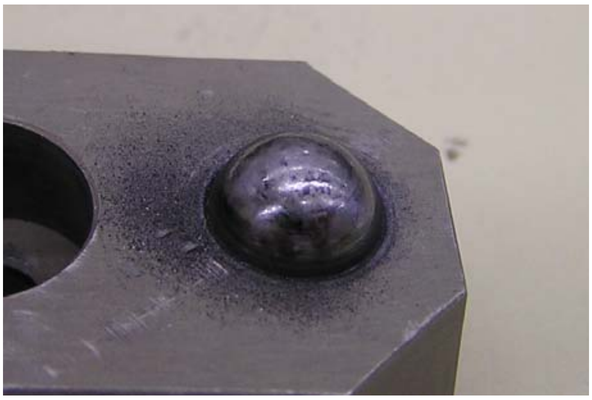

Fig 8: Ball Surface after Vibration Test

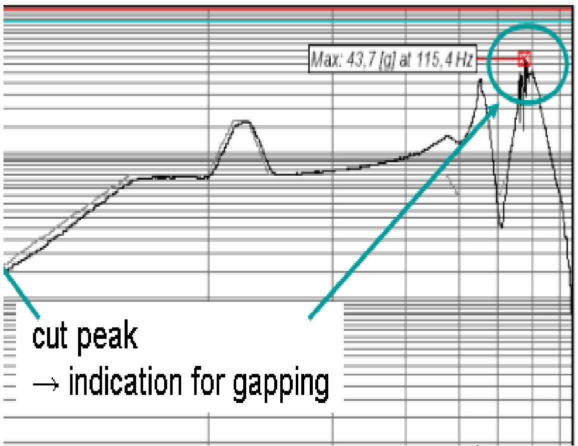

Fig. 9: Cut Peak at Sine Acceptance 2nd mode

In order to improve the correlation of the mathematical model with the test results the damping factor has been varied and it was found that 4% system damping provide a good prediction of the non-linear system behaviour.
The 4% is not from the gapping effect alone. Later variations of the APM with one Hold Down and Release Mechanism (HDRM) and a combination of a HDRM and PME also confirmed the 4% which can be assumed as a system typical value independent from the method of launch locking.

Though everybody was happy about the positive results of the vibration test, the question arose why the APM remained stable and the PME did not open though the theoretical holding force was exceeded.
The stepper motor used has a cogging torque (magnetic torque) of 14 mNm. The transmission factor from torque into output force is 100N/mNm. So the maximum un-powered holding force would be 1,400N assuming that vibration would eliminate all friction effects. Effectively a range of 3,200 N has been

observed during the vibration test without causing the actuator to rotate.
Based on these theoretical considerations the customer decided to skip the PME in favour of a HDRM. At that time no analytical model was available to explain what happened.

Principally there is a simple chance to prove also theoretically the holding capability of the APM by both using motors with stronger cogging torque and to improve the transmission factor. With motors as actually installed in flight units with 25mNm cogging torque and a transmission factor of 200 N/mNm as installed in the APM already in space the theoretical holding force would be 5,000 N and thus absolutely sufficient.

4. Simulation of the APM in Matlab®

4.1 General
Nevertheless the question was raised why against all theory the actuators didn't move during the EM vibration test.
In order to answer this question a mathematical model of the complete APM was established in Matlab/Simulink® (Fig. 10). In this model the single parts of the drive chain were simulated by independent sub-models and verified by tests on unit level. Special attention was put on non-linear contributors like gear-box play and the gapping of the PME. Additionally all friction effects could be set to zero in case of a dynamic excitation.

4.2 Motor Model
The motor model includes all mechanical, electro mechanical and magnetic effects of the stepper motor (Fig. 11).

It allows both to "drive" the system by providing power supply and to use it as generator or "brake" when mechanically driven from the output shaft.

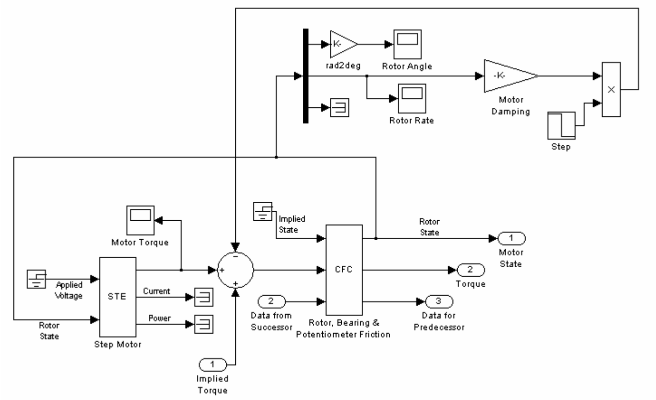

Fig. 11: Motor Model

The "brake" effect can be demonstrated by applying a constantly increasing torque which induces a step (rotation >1.8°) at 12mNm when all friction effects were zero (dynamic) and 16mNm with friction effects (Fig. 12)

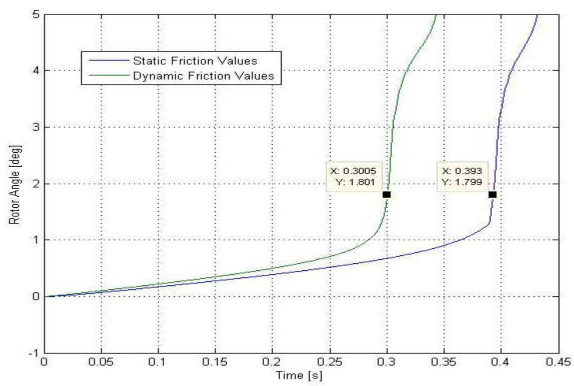

Fig. 12: Rotor Angle vs. Time for a Linearly Increasing Input Torque

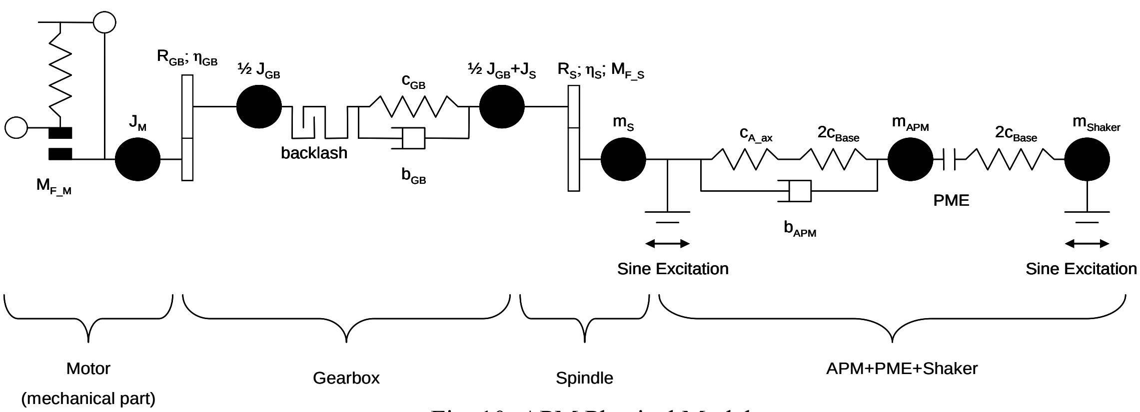

Fig. 10: APM Physical Model

4.3 Gear Box Model

The gear box is also completely modeled in its mechanical characteristics including backlash or play (Fig. 13).

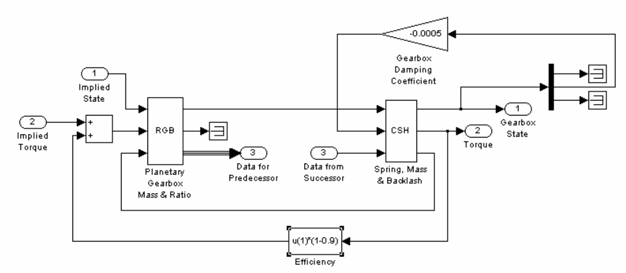

Fig. 13: Gear-Box Model

The backlash is simulated by a composite spring hook (CSH) element provided in Matlab.

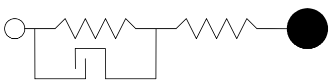

Fig. 14: CSH Element to Simulate Backlash

Setting the spring element in parallel to the hook to zero the contact positions at both ends of the hook can be specified to simulate contact and thus the backlash. The effect of this element can be shown by transferring a sine pulse via this element. The velocity plot in Fig. 15 shows that the gear-box mass velocity follows the input velocity with a certain delay (backlash) and starts to oscillate according to the gear-box mass/spring relationship.

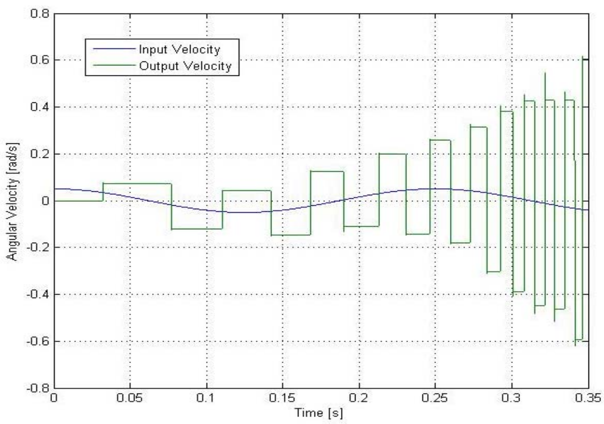

Fig. 15: Velocity Response of the gear-box with backlash

4.4 Spindle Model

Principally the spindle model could be equivalent to the gear-box one. Two exceptions have to be taken into account:

1. no backlash as the spindle is backlash-free by design
2. the load path from the APM has to be split into an axial load part that is transferred via the spindle bearing back into the APM and a rotational part that rotates the gear-box and finally the motor.

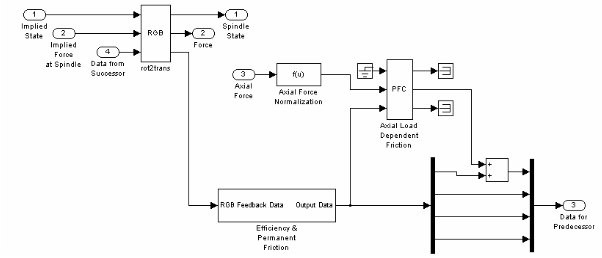

Fig. 16: Spindle Model

The static characteristic of the spindle model is shown in Fig. 17.

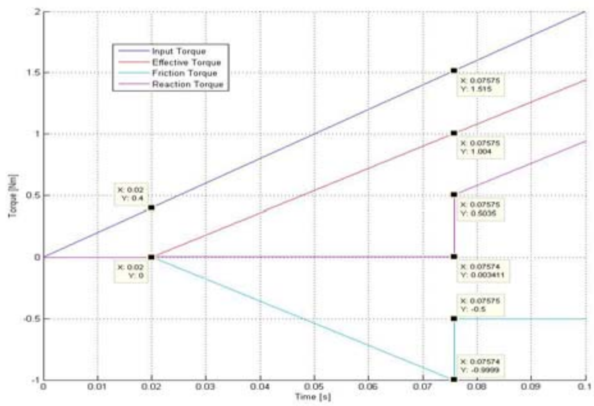

Fig. 17: Static Torque Transfer Characteristic of the Spindle.

4.5. APM and PME

The APM is a simple mass spring model adjusted to have it's Eigenmode at 100Hz with a mass equivalent to the APM inertia. As special feature the CSH element as used in the gear-box has been implemented to simulate the gapping at the PME.

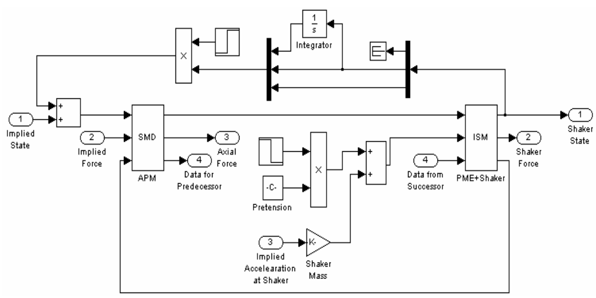

Fig. 18: APM/PME Model

5. Simulation Results

The effect of a stable APM as observed at the EM test could not be verified with the model.
Setting all friction to zero in the mathematical model the entire drive chain is accelerated from the pre-loaded configuration when externally excited. Due to the inertias the system doesn't stop when the expected cogging torque is achieved but continues rotating to a load value around 325N and oscillates with the excitation frequency.

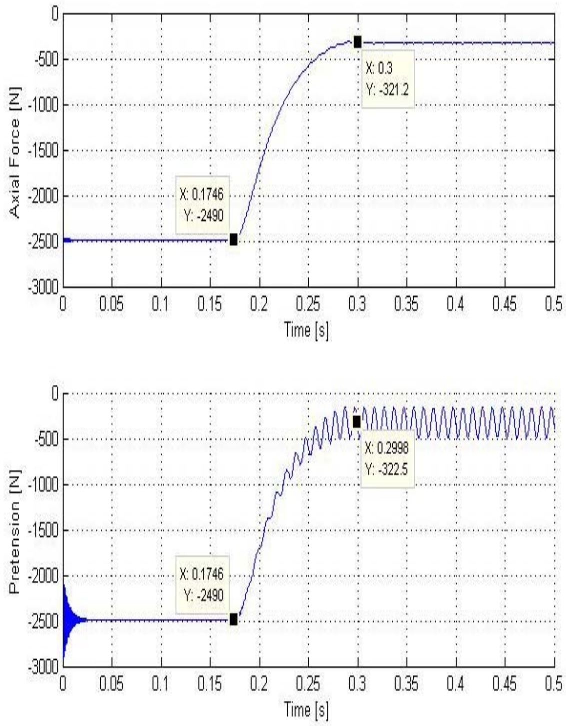

Figure 19: Axial Force and Pretension for Actuator Holding Force Simulation

In order to further verify the model a single actuator model was tested by simply driving the actuator against a hard end stop and exciting the entire unit dynamically.

In all cases the actuator could be statically driven to force values as expected from the motor torque, the transmission factor and the losses. Values up to 4kN could be achieved. Independent from the pre-load in all cases the system was back-driven whenever the excitation started and stopped typically at around 1,200N as expected from the motor cogging torque. So obviously some effects must have prevented a motor rotation beyond the first cogging torque peak.

An extensive variation of parameters in reasonable ranges has been performed in order to at least verify the single actuator test. A good combination was found by applying

- Potentiometer friction[2] of 4 mNm**,**
- Motor damping of 2.5% and
- Measured spindle efficiency of 64%. The spindle is preloaded such that in case of vibration a relative movement of the engaged threats is unlikely to occur.
- Spindle friction 40mNm

With these values the actuator pre-load dropped down to 1,325N which is in line with the actuator tes (Fig. 20).

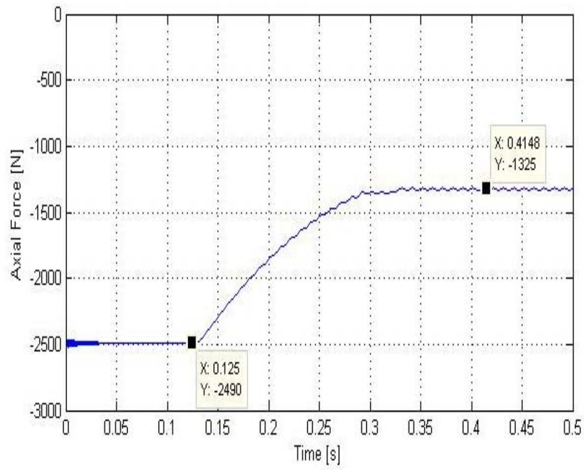

Figure 20: Axial Force vs. Time for Modified Dynamic Loss Parameters

But this result is still not in line with the observations made at the QM test.

An additional effect which could help to explain the contradiction between theory and praxis is the inclined orientation of the actuator in the launch position which causes remarkable shear loads on the spindle and spindle guiding which could cause clamping effects.

This has been implemented into the mathematical model and also verified by test (Fig. 21). Both the model and the test resulted in a holding force increase about 1kN which is still insufficient as another kN is missing to explain why the QM test was successful and the actuators didn't move though the load was beyond the one that could be held in a friction-less system.

[2] A potentiometer is mounted on the motor shaft for motor step counting.

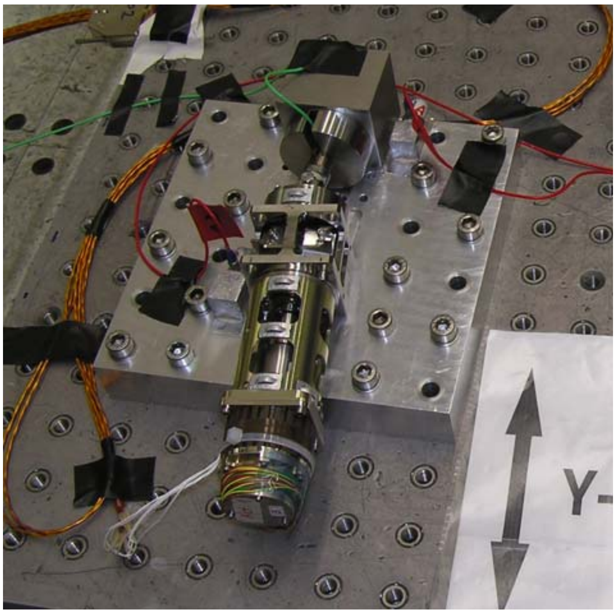

Fig. 21: Single Actuator Vibration Test Set-up with inclined orientation

Test No.	F_{PME} [N]	F_{ax} [N]	F_{rad} [N]	Vibration Level	Remark
01	3500	3530	490	-6dB	F_{act} dropped to 2300N
02	3400	3430	480	-12dB	F_{act} dropped to 2700N
03	3000	3030	420	-12dB	F_{act} dropped to 2500N

Table 2: Radial Force Test Results

6. Outlook

The APM QM test has proven the principal of a "self-locking" system which could be launched without additional external means to fix it.

Two basic concerns have to be eliminated:

- PME interface that shall not generate debris
- Actuator unit that both theoretically and practically ensures the locking force in static and dynamic environment.

A potential solution for the first point would be a pure brass cup without additional coating acting against a steel ball with polished MoS2 layer.

As already pointed out in chapter 3 the simplest way to overcome the theoretical proof of the actuator holding capability would be to take

1. A motor with larger cogging torque and
2. A lower spindle pitch.

But if for some reasons those solutions are not feasible there is still another option for the existing configuration:

3. Providing a constant DC voltage to the motor coils.

Excitation Amplitude [g]	Excitation Frequency [Hz]	U_{app} [V]	K_M [Nm/A]	a_{APM_max} [g]	F_{ax_max} [N]
0.5	97	0.5	0.15	9	-2800
0.7	97	0.75	0.15	10	-3300
2	97	1	0.15	13.5	-4200
2.5	97	1.25	0.15	17.5	-5000

Table 3: Holding Force vs. Applied Voltage

A 1.5V alkaline cell would fit to hold the actuator for a typical launch phase.

An issue may be how to switch on and off this power supply and to decouple it from the motor electronics

7. Lessons Learned

Lesson 1:

The theory of zero friction in dynamic environment could not be verified.

Certain elements in the drive chain obviously provide some friction. This specifically applies for the pre-loaded spindle. A specific torque test with a spindle in dynamic environment has not been performed in the frame of that development but should be done to verify this assumption.

Lesson 2:

Controlled gapping is a potential method to reduce dynamic loads and to cut resonance frequencies in mechanism. Contact surfaces can "survive" even large gapping forces when carefully selected with correct material mix and adapted shapes. Debris generation can be avoided by appropriate surface coatings.

Lesson 3:

Prediction of mechanism dynamic behaviour with Finite Element Programs and dynamical functional behaviour with simulation programs like Matlab is a very complex task due to the large variety of parameters to be considered.

4% system damping is an appropriate value to cover the energy "loss" effects in mechanism like the APM (4 gimbal bearings, 4 drive chain bearings, 2 gear boxes, 2 spindles, 2 motors)

Lesson 4:

The necessity to characterise the components of the drive chain in detail and to perform a dynamical functional analysis prior to the QM test has been underestimated.

It is more favourable to predict correctly expected system behaviour than to explain it after a test.