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TEST RESULTS AND FLIGHT EXPERIENCE OF BALL BEARING MOMENTUM AND REACTION WHEELS

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ABSTRACT

The required satellite mission durations and levels of reliability have been considerably increased: While in the beginning of the 70's 3- to 5-year missions were planned, the standard is now 10 years with an expansion to 15 years and more for such programs as INTELSAT VII. Based on a 20-year test and flight experience with basically the same design, ball bearing momentum and reaction wheels with the required 15-year mission capability can be provided.

INTRODUCTION

The required satellite mission durations and levels of reliability have been considerably increased in the course of the last decade and so.

While for such programs as SYMPHONIE, APPLE, IRAS AND OTS, 2 to 5 years were planned, the specification for ECS, INTELSAT V, TV-SAT etc. called for a 7-year useful life. Presently, wheels are delivered for such programs as EUTELSAT 2 and DFS-KOPERNIKUS, calling for a 10-year operation capability.

For future missions, as for the INTELSAT VII satellites, a 15-year mission duration is required!

Besides design considerations and development efforts, the experience gained during testing and actual flight with wheels built essentially to the same procedures is of vital importance for providing that level of confidence necessary for the planning of such long missions.

WHEEL DESIGN

Design Description

A modular design was selected to cover a wide range of applications and specifications. With one ball bearing size three wheel diameter classes are provided which are given in Table 1 together with main parameters and/or parameter ranges:

Table 1. - Wheel Parameters

Wheel Diameter	Height	Angular Momentum	Mass
22 cm	7.5 cm	1.5 to 10 Nms	2.5 to 4 kg
26 cm	8.5 cm	5 to 20 Nms	3.5 to 6 kg
35 cm	12 cm	12 to 80 Nms	5 to 8.5 kg

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In the following, the mostly utilized 35-cm diameter wheel (Fig.1) is discussed in some detail:

It consists of five subassemblies:

- Housing with Center Stud Bolt
- Flywheel Mass with Spokes, Damper Rings and Hub
- Ball Bearing Unit
- Motor Rotor
- Motor Stator with Commutation Electronics

These subassemblies are completely independent of each other and can be built and tested separately. Thus, performance deficiencies or unacceptable tolerances can be determined at the subassembly stage and, if necessary, corrected. After successful individual tests, the subassemblies can be "accepted" and assembled to the complete wheel.

This modular design approach allows also for relatively simple adaptations to different specifications without losing the heritage. For instance, the flywheel mass can be equipped with rings of different moments of inertia, or, the motor designed for lower or higher torque capability, or the bearing unit adapted to different speeds or speed ranges.

o Housing

The lightweight housing provides a defined environment for the delicate "interior". It consists of the upper and lower housing, both spun from sheet metal. In the latter, the base plate is fixed by electron beam welding. Both parts are soldered to each other via a tear strip.

The special curvature together with the center stud bolt is the reason for the capability to withstand easily the air pressure during and after evacuation.

o Flywheel Mass

To achieve a favorable inertia-to-mass ratio, the flywheel rim is linked to the hub by lightweight but stiff double-T-shaped spokes made of sheet metal. If this flywheel were excited by vibrations, introduced to the hub, resonance step-ups of 80 or so would result. Because no load relief is foreseen, these loads would lead to ball bearings of a size not acceptable with regard to mass and friction. Therefore, the flywheel mass is damped by two damper rings which are in friction contact with the spokes. The reasonance step-ups are now more than one order of magnitude lower.

o Ball Bearing Unit

This unit is decisive for life and reliability. Two ball bearings are rigidly preloaded to each other by spacer sleeves and nuts. The material selection and geometrical properties assure a practical constant preload over temperature and temperature gradients.

The retainers are designed to operate stable under all speed, temperature and lubrication conditions. The initial amount of lubricant in the bearings is sufficient for providing an EHD-film for years of running. To assure a multi-year operation capability, a lubrication reservoir, activated by centrifugal force, is located between the outer rings. After a certain accumulated running time, this reservoir starts to bleed oil to the outer rings of the bearings.

o Motor Rotor

A brushless and ironless DC motor is used. Depending on the torque requirement, either one of the two cylindrical surfaces of the U-shaped rotor are covered with samarium cobalt permanent magnets.

o Motor Stator with Commutation Electronics

The stator coils are inserted into the holes of the cylindrical support of the armature. The stator is mounted between the above mentioned rotor surface. Thus, the permanent magnet flux is "cutting" the wires of the windings. To avoid eddy currents, stranded wire is employed in most applications.

The commutation sensors together with a shutter ring controls via electronics the currents in the coils of the motor. The sequence of switching provides also tachometer signals, and if necessary, also a speed direction indication.

For wheels which must be able to operate cw and ccw, two sets of sensors with adequate electronics are used.

Life Prediction

For the life prediction, the ISO/R281/1 is used. It takes into account newer research results. The traditionel AFBMA formula

$$L_{10} = \left(\frac{C}{P}\right)^3 \text{ million revolutions}$$

is extended to

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \left(\frac{C}{P}\right)^3 \text{ million revolutions,}$$

where

L_{na} = modified nominal life with the probability n in percent that a ball bearing fails before L_{na} is reached,

a_1 = life adjustment factor for achieving or exceeding a life time with a probability other than 90 percent,

a_2 = life adjustment factor for bearing material

a_3 = life adjustment factor for nonconvential operating conditions,

C = dynamic load rating and

P = equivalent load.

The essential message is that even a non-failure life L_{NF} can be expected, provided there is "sufficient" lubrication.

For the utilized bearing size ISO 04 (20x42x12 mm³), an equivalent load of 85 N, the load rating of 7800 N, the defined $a_1 = 0.05$ for achieving L_{NF} and a factor $a_2 \cdot a_3 = 2.5$ (resulting from material, cleanliness, surface asperity relative to lubrication film thickness), an L_{NF} of about 18 years results, when a speed of 6000 rpm is assumed. This life potential is beyond the present requirements of 15 years and valid for the fatigue life aspects.

Other failure mechanisms are:

1. Corrosion: Avoided by employing stainless steel and sealing of the housing.

2. Wear: Good surfaces, extreme cleanliness and a sealed housing are provided.

3. Retainer damage: A well proven material is used which is able to bear loads equivalent to 40,000 rpm for this bearing size continuously and a stable retainer movement.

4. Inadequate lubrication: A high-viscosity lubricant is used; surfaces wettability checked; a failsafe lubrication system is implemented.

TEST RESULTS

Lubrication Tests

After a theoretical and an experimental trade-off, an oil lubrication was favored; the generally employed grease lubrication for long life applications has some disadvantages: Excessive friction torque changes and spikes, especially over the temperature range and while employing temperature gradients. Furthermore, no simple mechanism for the replenishment of grease - without too much influence on the friction torque behavior - could be defined.

To assure a low friction torque over temperature and time, an oil film lubrication with the mineral oils SRG-60 and KG-80 was developed - similar to that employed in high performance gyro bearings. This task led to intensive investigations concerning materials, cleanliness, wettability, retainer dynamics, migration control, molecular seals, and oil distribution.

One test series was devoted to the determination of the minimum lubricant quantity for the generation of a full EHD-film. A main criterion for the quality of such a film was the electrical resistance as a function of time at different speeds. Because two bearings were mounted in an original bearing unit, one of the bearings was equipped with balls made from ceramics.

The retainers were vacuum impregnated and subsequently centrifuged to such an extent that no additional oil should come out during running. The oil amount, distributed to the steel parts was sequentially decreased from about 9 mg in steps down to about 2 mg and finally to about 1 mg. The torque over speed and time was recorded, together with a monitoring of the electrical resistance. It turned out that 2 mg are still sufficient for flawless operation while at 1 mg signs of starvation were evident: The resistance over time showed more breakdown spikes.

Therefore, it was concluded that the minimum amount of lubricant is 2 mg. At this lubrication condition, the constant friction torque level was reached shortly after switch-on. Also over temperature ranges (-10 °C to +55 °C), the torque values only changed some 10%.

With higher amounts of lubricant the run-in time increased accordingly from an hour to some ten hours but the finally achieved torque levels were practically the same as those got for minimum lubrication.

With a bearing unit, equipped with molecular seals and anticreep barriers, and with an initial amount of oil in the bearings of about 10 mg, years of flawless running are possible.

In the course of the initial development phase for wheels, due to schedule reasons, no testing for mission duration (at that time 5 years) could be planned. Therefore, a lubrication reservoir was conceived which starts to bleed oil to the bearings after some delay time. The working principle of this device is explained in the following to a certain extent.

The reservoir, situated between the two outer rings of the ball bearings on the outer spacer sleeve, is subjected to the centrifugal force during running. The lubricant - the base oil, stored as grease in a ring type chamber - can be centrifuged out through some orifices. The rate is limited by the thickener of the grease which forms a microporous filter in and in the vicinity of these orifices.

To avoid an early bleeding to the bearings, the centrifuged oil enters a coaxially situated oil reservoir, made from a porous material. This oil reservoir is vacuum impregnated with oil and afterwards centrifuged to an extent that in the nominal speed range no oil is bleeding to the bearings. Only after some operating time, when the oil from the grease chamber has re-saturated the oil reservoir, the bleeding to the bearing starts with a low rate.

Figure 2 shows the long-term bleed test of a grease chamber. At the beginning, a high bleeding rate is experienced which levels out after some thousand hours of centrifuging. The high rate of bleeding at the beginning allows for a good definition of the delay time of the lubrication reservoir. An overall bleed characteristic of the lubrication reservoir is indicated too. In this example, the delay time would be 7,500 hours, the oil amount for saturation of the porous oil reservoir 900 mg and the total supply of oil, fed to the bearings, 170 mg in 120,000 hours.

The bearing unit is designed in a way that "surplus oil" is not leaving the lubrication area "for ever" but is stored in the vicinity of the bearing. This could be also demonstrated by testing. The advantage is that this oil can creep back to the bearings. The disadvantage is that one may experience overlubrication with excessive friction torques.

To prove that no critical overlubrication can take place, an accelerated overlubrication test was performed. The test setup is shown in Figure 3. An original bearing unit operating with its spin axis in the horizontal position was modified. The grease chamber's inner cover was removed so that oil could be directly dropped into the open grease chamber. The results of this test are interpreted by using Figure 4. Over a time period of about 90 days, the maximum oil quantity, centrifuged out of a grease chamber in a mission, was added. The acceleration factor was therefore in the order of 40 for a 10 year mission.

The bearing unit was allowed to run in for 6 days. The current (a measure for the friction torque) decreased from 310 mA to 215 mA. Now oil was added in steps at a relatively high rate between day 7 and 15. Because this amount of oil was absorbed by the porous oil reservoir, no lubrication status change of the bearings had occurred.

The oil added afterwards led to a small increase of the current (day 22 to 31) with the beginning of more torque noise on day 32 and an increase of the current to 294 mA. During the days 33 to 41 no oil was applied to allow for stabilization and run in. In this period the current decreased to about 210 mA with heavy torque spiked (shaded areas).

The adding of oil was resumed on day 42. Two days later again signs of overlubrication appeared with increased torque noise and a subsequent current increase to 320 mA on day 52. (The last oil increment was applied on day 48.) The following run-in period brought the current again down to about 210 mA on day 62.

To simulate a stop-start sequence, the bearing unit was stopped on day 77. Due to the high oil amount in the bearings, the current was 440 mA after the start on day 83; run-in occurred within about 3 days to a current between 210 and 218 mA. The oil amount of 100 % was reached on day 93.

A further addition of oil to about 108 % was performed on day 105 with a "response" of the torque on day 107 to about 330 mA. Now the bearing unit was stopped to simulate a stop-start sequence in an over-lubrication status. The switch-on (day 108) led to a longer run-in period but finally on day 122 the current was again down to 220 mA.

The test was the proof that a running bearing unit can distribute additional oil when applied in small quantities (actually continuously with a small rate) without excessive torques. Therefore,the feed rate of the lubrication reservoir is not so critical.

Storage Test

A wheel (SYMPHONIE qualification model) was qualified, integrated into the qualification model of a satellite and subjected to the qualification test of the satellite. After a storage period of five years it was possible to receive this wheel back for investigations. The only deterioration was an increase of the current which could be traced to a housing leakage. After reevacuation, the current was practically the same as during qualification.

Subsequently, the wheel was disassembled. The appearance of all sub-assemblies and parts was "as new". The bearings were well lubricated. The measurements of the roundness, surface parameters and so on confirmed that no degradation took place.

It was concluded that storage periods of a wheel, which could accumulate to say 15 to 20 years (integration + storage on ground + cold redundancy during a mission) should have no negative influence on the operational readiness of such a wheel.

Life Tests

A total of 103 years (status Nov. 1989) of testing were accumulated. Of special interest are two momentum wheels (3,000 rpm and 3,800 rpm) which have been in continuous operation - after full qualification - since October 1973. These wheels were also subjected to thermal cycling in half-year intervals to simulate eclipse periods. The currents/torque levels remained constant - within ± 10 % - after run in.

To increase the "torture" on the wheel, running at 3,800 rpm, it was mounted in a sunlight-shaft in March 1981. This was a cheap way to subject this wheel to seasonal and daily temperature cycling. As an example (Figure 5), the temperatures and currents over time are indicated for two days each in February, August and November 1985; the temperature was measured in the vicinity of one of the ball bearings. The temperature ranges from 3.5 °C to 30.5 °C. Daily changes occur with peak-to-peak values of 2.5 °C to 7 °C. The friction torques remain remarkably constant (0.012 Nm to 0.013 Nm).

Also a reaction wheel life test is performed with one daily triangular speed cycle between -3,500 rpm and +3,500 rpm. In this case, too, each half year temperature cycling takes place. Figure 6 shows the friction torque over speed at the start of the test (after run in) and after 10 years of testing. There is practically no change; this is also true for the results at lower and higher temperatures.

No ball bearing failure occurred during all life tests.

FLIGHT EXPERIENCE

Flight Programs

There are 19 satellite programs (40 satellites, 84 wheels) flying this basic wheel design. A total of 212 operating years was accumulated, not counting the cold redundant wheels.

The OTS wheel is that with the longest operation time of 11.5 years (status November 1989).

No ball bearing failures had been experienced. The two SYMPHONIE wheels showed signs of electronics degradation long after the planned mission duration of 5 years. The wheels served 9 years after the satellites were decommissioned, long after other failures (batteries, transponders etc.).

Due to the demonstrated reliability and long-life characteristics, 240 wheels were manufactured or are under contract.

Flight Operation Results

Because there were no failures reported, we found it rather difficult to get information concerning the performance of the wheels in orbit. However, it was possible to receive some results from the OTS, INTELSAT V, TV-SAT 2 and DFS-KOPERNIKUS Satellites. Data of the latter are presented at the Symposium only.

o **OTS-B**

This satellite has been 11.5 years in operation. In Table 2 some acceptance test and flight results are presented which show the motor current over speed, temperature and time.

Table 2 - Test and Flight Result OTS-B

Speed	Acceptance Test Nov. 1976 22 °C	-5 °C	45 °C	Flight 18th Oct. 1989 4000 rpm 31 °C
3,600 rpm	220 mA	284 mA	186 mA	min 206 mA
4,000 rpm	230 mA	292 mA	192 mA	max 292 mA
4,400 rpm	234 mA	294 mA	198 mA	avge 248 mA

The attitude control utilizes an integrator in the loop. Therefore, a "massive" differential control component is necessary. This leads to a noisy current, as can be seen also in Figure 7 which shows the current over time on the 7th Sep. 1989 at a speed of 4,000 rpm and a temperature of 31 °C.

o **INTELSAT V**

Acceptance test, pre-launch and flight results can be compared in Table 3.

Table 3. - Test and Flight Results INTELSAT V

	Launch	Acc.Test (Amp)	Pre-Launch (Amp)	Post-Launch (Amp)	July 85 (Amp)	May 88 (Amp)
F 1	Dec 80	0.17	0.21	0.19	0.19	0.19
F 2	May 81	0.23	0.22	0.20	0.19	0.19
F 3	Dec 81	0.20	0.23	0.20	0.19	0.20
F 4	Mar 82	0.21	0.20	0.20	0.21	0.20
F 5	Sep 82	0.20	0.21	0.20	0.19	0.19
F 6	May 83	0.19	0.18	0.17	0.19	0.18
F 7	Oct 83	0.21	0.25	0.20	0.21	0.21
F 8	Mar 84	0.20	0.18	0.17	0.20	0.20
F10	Apr 85	0.17	0.21	0.20	0.20	0.17
F11	Jul 85	0.17	0.20	0.20	0.20	0.17

Temperatures were 20 °C to 35 °C, speeds in the nominal range.

o **TV-SAT 2**

In Figure 8 the wheel temperature, motor voltage (a measure for the speed) and motor current is plotted for the 28 of Sep. 1989, about at maximum eclipse (launch: Aug. 1989). The satellite's position is 11° West which can be recognized in the temperature plot as a relatively high temperature gradient. The current, quantisized by TM, is not influenced by these moderate temperature changes. It compares favorably with the acceptance test result of 0.11 Amp at normal temperature and nominal speed.

CONCLUSION

A ball bearing momentum/reaction wheel of modular design with a special lubrication system has proved its ability of reliable long life operation during tests and in orbit. With this heritage, the specifications for a 15-year mission can be met with a high confidence level.

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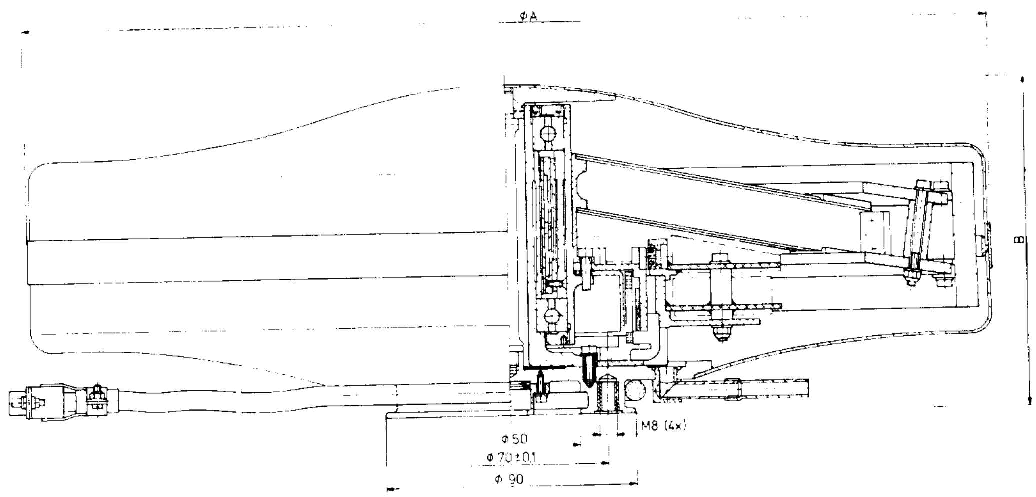

Figure 1. Wheel

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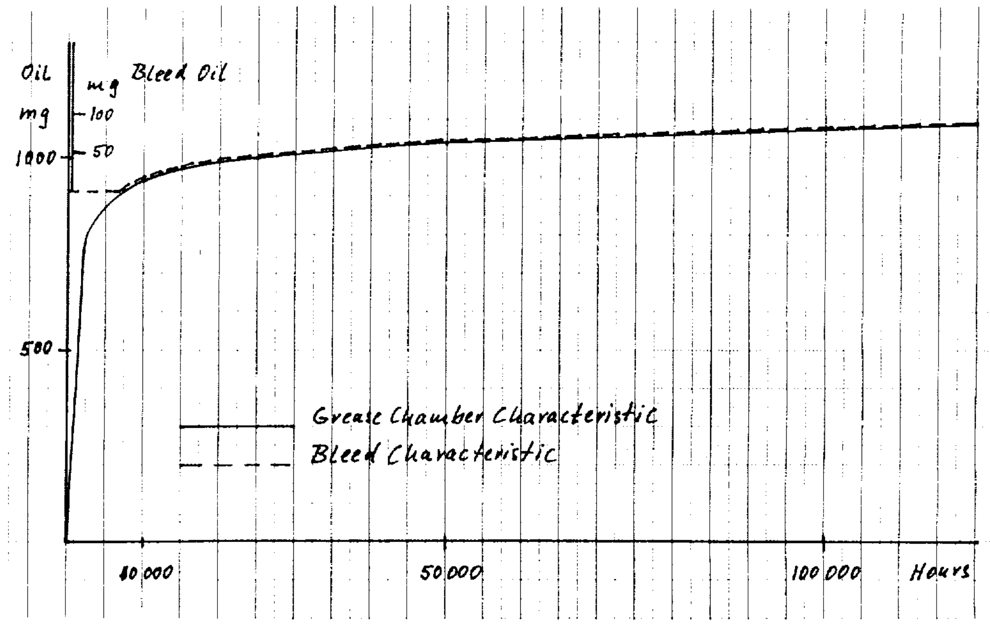


Figure 2. Grease Chamber Characteristic and Bleed Characteristic

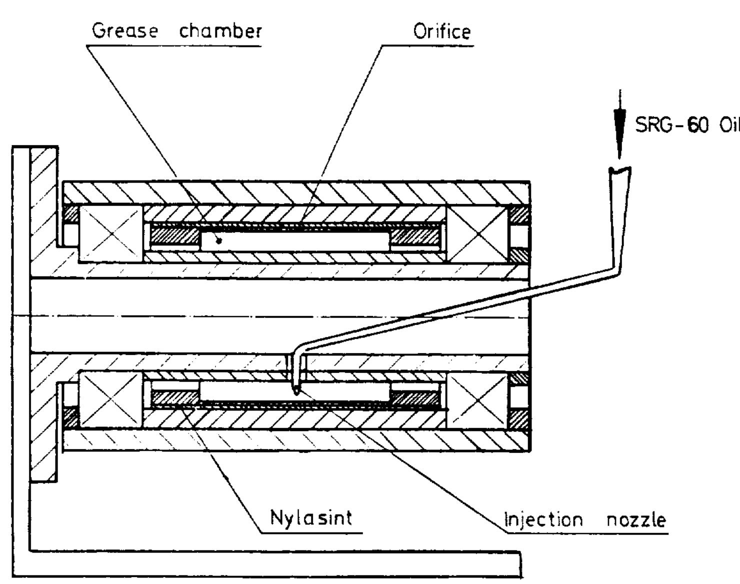
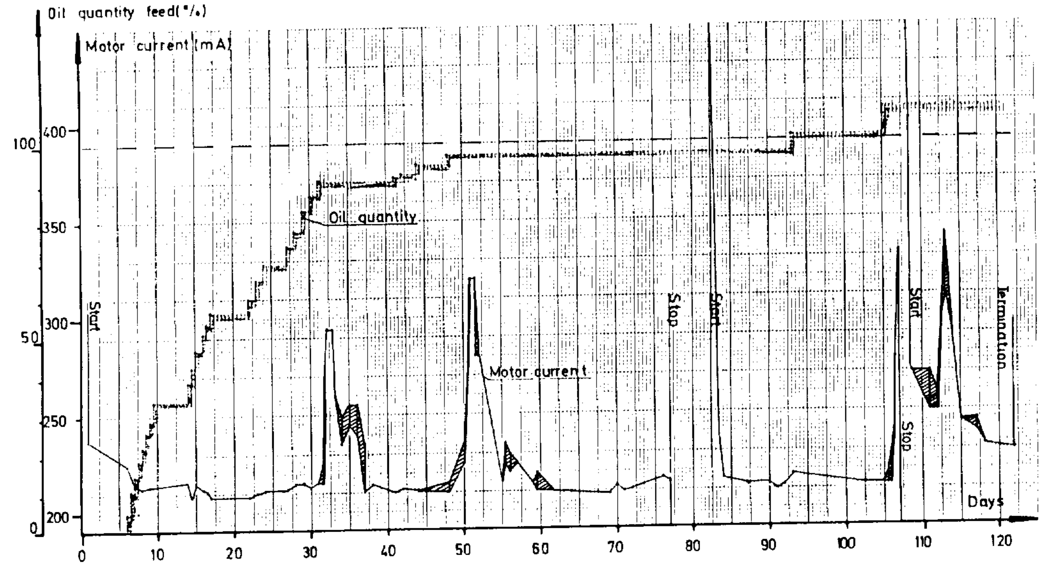


Figure 3. Test Setup for Overlubrication Test



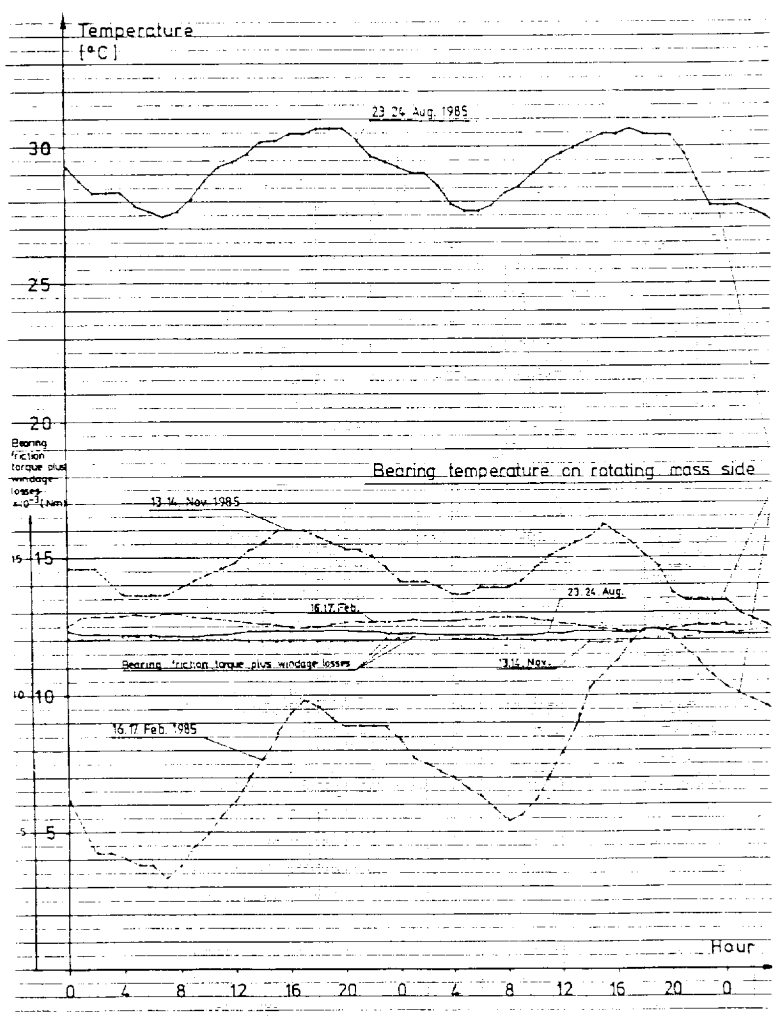


Figure 4. Results of the Overlubrication Test

Figure 5. Life Test Results, Momentum Wheel

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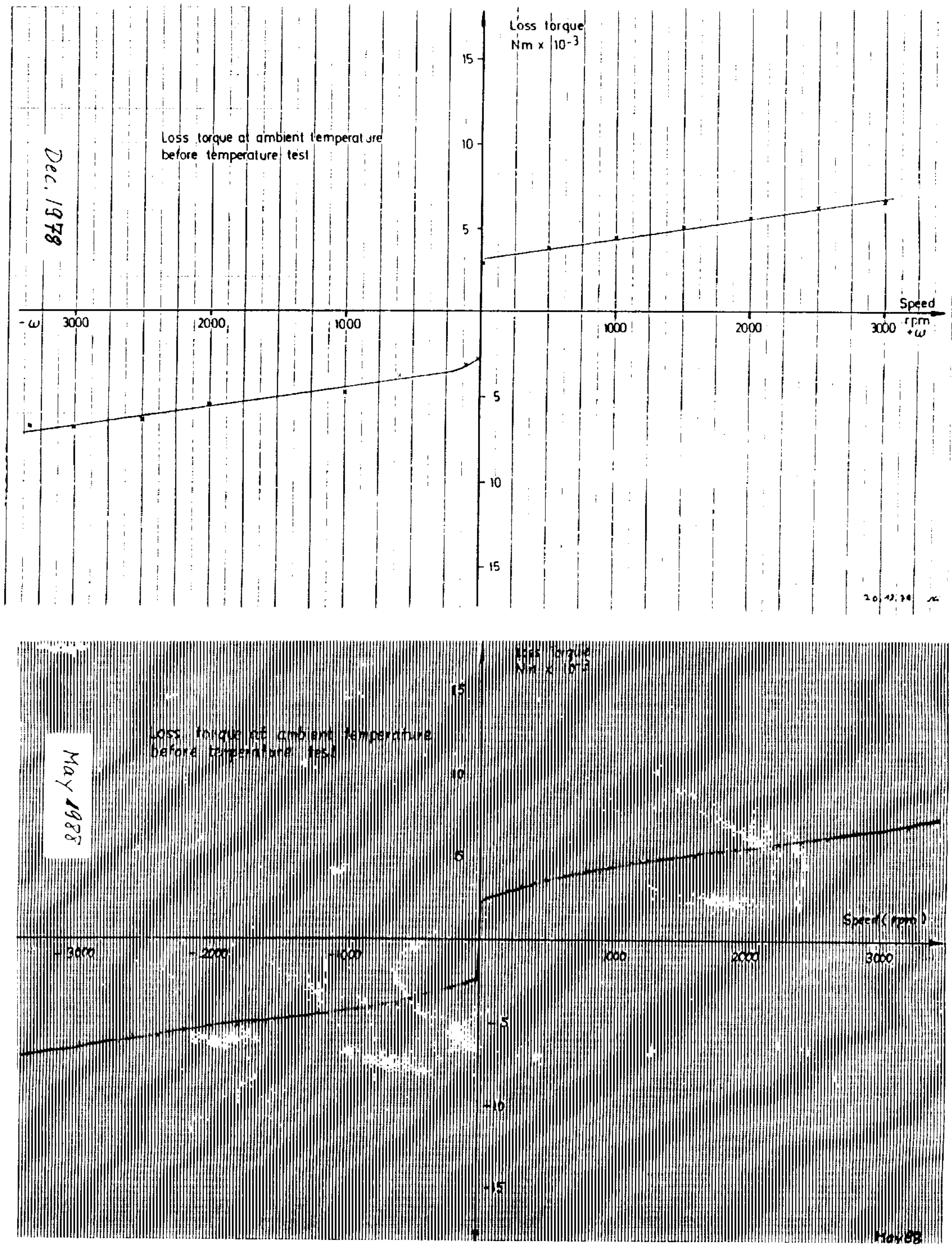

Figure 6. Life Test Results, Reaction Wheel

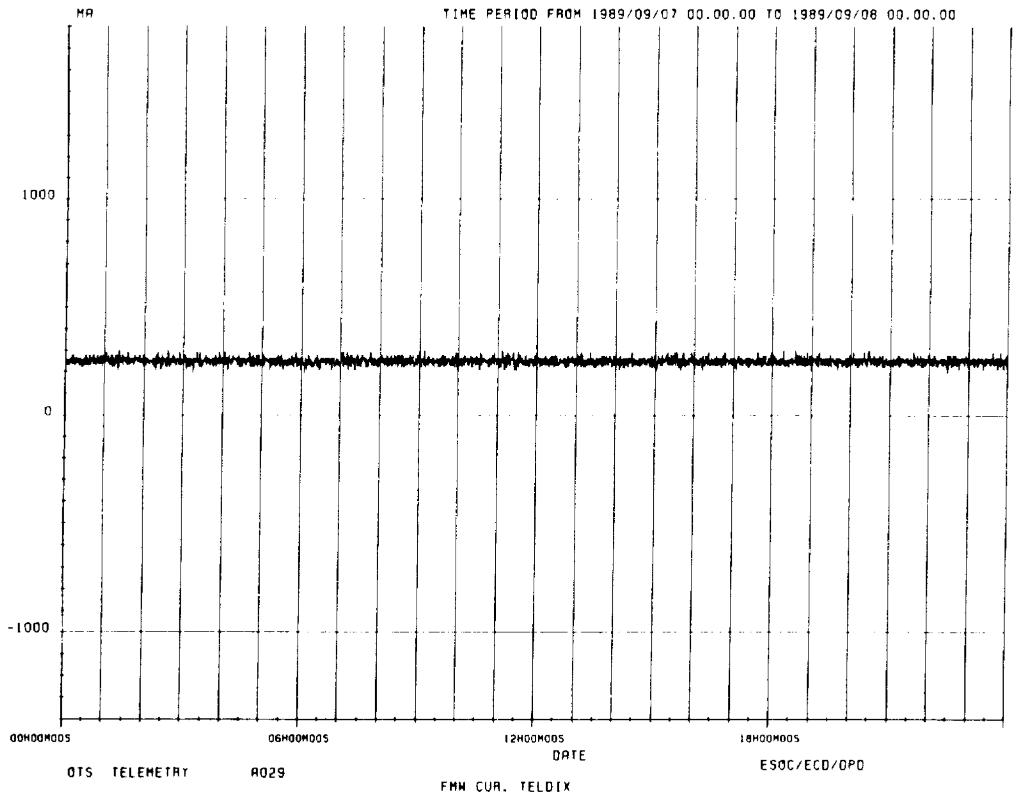

Figure 7. OTS-Current

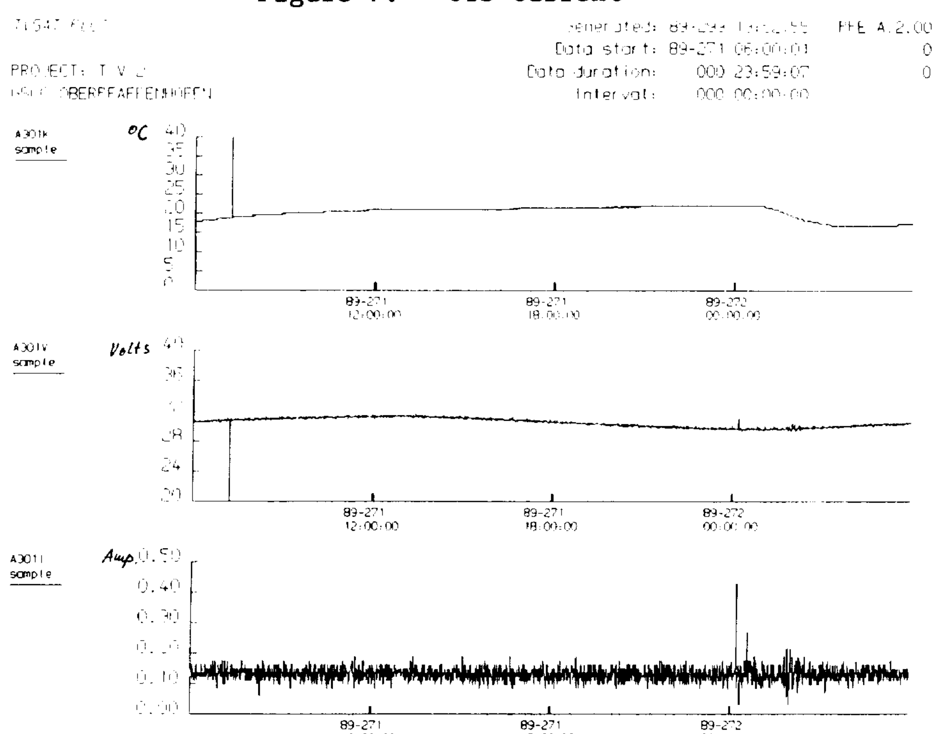

Figure 8. TV-SAT 2 - Current/Voltage/ Temperature

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