# Estimating friction parameters in reaction wheels for attitude control

**Valdemir Carrara[1] and Hélio Koiti Kuga[2]**

[1] INPE, Av. dos Astronautas, 1758, São José dos Campos, SP, CEP12227-010, Brazil

[2] INPE, Av. dos Astronautas, 1758, São José dos Campos, SP, CEP12227-010, Brazil

Abstract:

The ever-increasing use of artificial satellites in both the study of terrestrial and space phenomena is demanding a search for increasingly accurate and reliable pointing systems. Attitude control systems rely on sensors and actuators that follow these requirements, but its cost is still high, with little tendency to fall. It is common nowadays to employ reaction wheels for attitude control that provide wide range of torque magnitude, high reliability and little power consumption. The wheels are composed by a Brushless DC motor (BLDC) whose rotor is attached to a flywheel. The low torque generated by the motor and the high inertia causes the wheel to accelerate or decelerate at very low rates. However, the bearing friction causes the response of wheel to be non-linear, which may compromise the stability and precision of the control system as a whole. This work presents a characterization of a reaction wheel of SunSpace Co., maximum capacity of 0.65 Nms, in order to estimate their friction parameters. It used a friction model that takes into account the Coulomb friction, viscous friction and static friction, according to Stribeck formulation. The parameters were estimated by means of a nonlinear batch least squares procedure, from data raised experimentally. The results have shown wide agreement with the experimental data, and were also close to a deterministic model, previously obtained for this wheel. This model was then employed in a Dynamic Model Compensator (DMC) control, which successfully reduced the attitude steady state error of an instrumented one-axis air-bearing table.

***Keywords: attitude control, reaction wheel, parameter estimation***

## INTRODUCTION

This paper presents a Dynamic Model Compensator (DMC) control of reaction wheel in current control mode. The error is compensated for by means of a mathematical model of wheel dynamics and bearing friction. Reaction wheels are actuators largely employed in attitude control subsystems in order to provide attitude pointing and stability of artificial satellites. They consist of a Brushless DC motor (BLDC) coupled to a high inertia flywheel. The torque applied to the wheel is sensed by the satellite in the opposite direction, allowing the attitude control based on information of inertial sensors like gyroscopes, sun sensors, magnetometers and star sensors. Reaction wheels are devices that must operate continuously for several years in vacuum conditions, subject to wide variations in temperature and high radiation doses. So, its reliability and quality are essential to the satellite health. These requirements pose great challenge to a reaction wheel design, which makes such equipment highly complex and expensive. Reaction wheels are classified according to its capability of storing angular momentum; from the small ones employed in micro-satellites to large ones appropriated for orbital stations and communication satellites. Normally reaction wheels are operated either in current (or equivalently torque) mode or in speed mode. In current mode the electronics delivers the necessary current to the motor in order to achieve the commanded torque. In speed mode a secondary outer control loop regulates the current to eliminate the error between the commanded angular speed and the flywheel speed, which is measured by some sort of rate sensor (usually Hall effect sensor or optical incremental encoder). The speed mode control avoids the bearing friction effects, which causes a non-linear behavior in the current control mode. However, speed control introduces more complexity in the electronics and also causes some delay in wheel response. In order to assure linearity in the current mode, and eventually disregard the speed control mode, this work suggests mitigating the effects of friction by adopting a DMC controller in current control loop. This compensator was applied to an off-the-shelf reaction wheel that operates in both current and speed mode. The friction model includes Coulomb, viscous and static or breakaway torques. With the aim of evaluating the control performance, the static friction was replaced by the Stribeck friction, which, unlike the previous one, does not present discontinuities when the motor reverses its rotation sense. All friction parameters and the motor coefficient were obtained by a least squares fit of data collected from several experiments performed with the wheel in current mode. The experiments consisted of a continuously varying current command in order to stimulate the wheel through various speeds and sense inversions, so as to assure correct parameters identification and model fidelity. The DMC was then introduced in the attitude control loop of an air-bearing table that emulates the frictionless conditions found in space. The table has a fiber optic gyroscope for measuring angular rate (that provides the reference for the attitude after integration), the reaction wheel, a system of radio-modem for reaction wheel telemetry and command, and a power supply battery. A small computer fan was placed in the air-bearing table, so as to yield a small torque, which shall be duly balanced by the attitude control procedure. By proper selection of the initial conditions plus the fan torque, the wheel will be forced by the attitude control to reverse its direction of rotation. The results show that there is a significant gain when the DMC is implemented in the control loop, when compared with the simple current mode control, with control performance comparable to the speed mode. It is also presented in this study a comparison of the statistical method for determining friction and motor parameters with a deterministic method, in which each parameter has been obtained from a dedicated experiment to highlight its influence.

In Carrara and Milani (2007) the friction parameters of a reaction wheel commanded in current are raised from experimental means. In that work the wheel is subject to specific commands in order to highlight a particular parameter. These are then calculated by manual curve adjustment, based on minimum quadratic variation. The model used took into account the Coulomb and viscous frictions. In Carrara (2010) the same model was used in an attitude controller of an air bearing table. The controller used command in current with dynamic compensation based only on Coulomb and viscous frictions. With this method it was possible to reduce the error during wheel rotation inversion by an order of magnitude. Later it was made a comparison between the two forms of control in Carrara et al. (2011), which showed that the dynamic compensator introduces an error comparable but slightly higher to control mode in the angular velocity of the wheel. The wheel friction parameters plus the Stribeck friction (in fact, a continuous and differentiable model of static friction or departure friction) were estimated by a Kalman filter in Fernandes et al. (2012), but with non-conclusive results, because of the scarcity of accurate experimental data. Few works in literature relate friction models with reaction wheels bearings (Moreira et al., 2005; Shenming and Cheng, 2006). On the other hand, several articles have friction models and estimation of parameters in rotors, as in Olsson et al. (1998) and Canudas and Ge (1997), including a dynamic model for friction by Canudas et al. (1997). This work proposes to estimate, by means of a nonlinear least squares procedure, the parameters of the friction of a reaction wheel, shown in the photo of Fig. 1, considering not only the Coulomb and viscous frictions, but also the Stribeck friction. The parameters so estimated are then compared to those obtained in Carrara and Milani (2007) and Carrara (2010). In the following sections will be presented the formulation of friction model and of the estimation of parameters. The experimental results appear next, together with the comparison between both methods: statistical and deterministic. The conclusions are presented in sequence. The reaction wheel used was manufactured by SunSpace (Engelbrecht, 2005) and acquired by the Space Mechanics and Control Division of INPE.

## MATHEMATICAL MODEL

For gathering data necessary for this work, a setup made by Carrara and Milani (2007) was used. In a bearing table system of one degree of freedom in rotation (Fig. 1) a reaction wheel with maximum capacity of 0.65 Nms commanded by current via serial interface, a fiber optic gyroscope of one axis (not used in this work), a command and telemetry electronics, a radio modem for communication with the equipment and a battery for power supply were installed. The programs needed to command the wheel and make the current readings and angular velocity were written in C++, and run on a computer that is external to the table.

**Fig 1 – Experiment mounted on the air-bearing table.**

The mathematical model of a reaction wheel is analogous to the model of a DC motor, which inertia includes, besides the rotor inertia, the inertia of the flywheel attached to the axis of the wheel. In the model considered here it was included the viscous friction, Coulomb friction and the friction of Stribeck. The differential equation describing the motion is:

$$T_w = J_w\,\dot{\omega} + b\,\omega + \operatorname{sgn}(\omega)\left[c + d\ \mathrm{e}^{-\omega^2/\omega_s^2}\right] \qquad (1)$$

where $T_w$ is the motor torque, the wheel's and rotor inertia is $J_w$, $b$ is the coefficient of viscous friction, $c$ is the Coulomb friction torque, $d$ is the starting torque, $\omega$ is the angular velocity of the wheel and $\omega_s$ is known as Stribeck speed (Olsson et al.,1998; Canudas and Ge, 1997). The torque model is displayed graphically in Fig. 2. The starting torque $d$ can be decomposed on the difference between the static torque $T_s$ and the Coulomb torque $c$, i.e. $d = T_s - c$. Neglecting nonlinear effects present in current to torque conversion, one can consider that the torque applied to the motor is proportional to the current in the stator, $I$, in the form:

$$T_w = k_m I \tag{2}$$

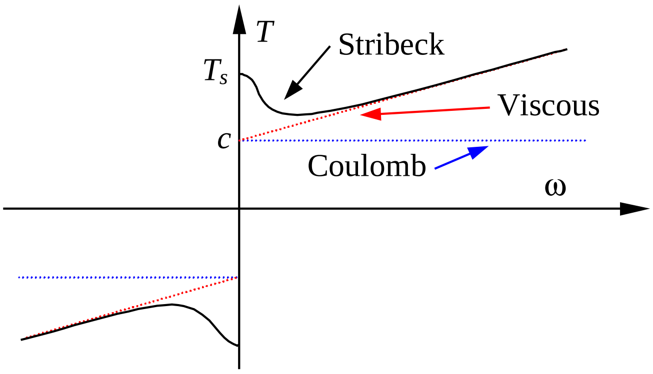



**Fig 2 – Friction torque model used in the parameter estimation.**

In current control mode, one commands the current $I$ on the wheel and get telemetry readings of angular velocity $\omega$ and current itself, which may be slightly different from that commanded due to the presence of an internal current control loop to the wheel. For the estimation of parameters by means of a least squares procedure, the state to solve for is composed by the angular velocity, the motor constant, viscous friction coefficient, Coulomb torque and static torque. Since the inertia of the wheel cannot be estimated independently of other parameters, the inertia value supplied by the manufacturer of $J_w = 1.5\ 10^{-3}$ kg m$^2$ was adopted. The state to be estimated is then

$$\begin{aligned}\mathbf{x} &= \left(\omega \quad k_m / J_w \quad b / J_w \quad c / J_w \quad T_s / J_w\right)^T = \\ &= \left(x_1 \quad x_2 \quad x_3 \quad x_4 \quad x_5\right)^T\end{aligned} \tag{3}$$

Stribeck speed $\omega_s$ could also be estimated, but preliminary tests showed that the noise present in the measurements at low speed, where this parameter is important, do not allow a good estimate of its value. In addition the estimated values of the remaining parameters are barely affected by $\omega_s$. As a result one adopted to this speed the 4 rpm value obtained indirectly through a mapping of the average current as a function of the angular velocity of the wheel at low speeds, using the speed control mode.

From the Eq. (1), the dynamical model for the estimation process is drawn:

$$\dot{x}_1 = x_2\, I - x_3\, x_1 - \operatorname{sgn}(x_1)\left[x_4 + (x_5 - x_4)\,\mathrm{e}^{-x_1^2/\omega_s^2}\right] \tag{4}$$

Once the dynamical part is represented by only one (time) variable (rotation $x_1$), and the remaining states are parameters, the non-null elements of the corresponding Jacobian matrix of partial derivatives are:

$$\frac{\partial \dot{x}_1}{\partial x_1} = -x_3 + 2\frac{x_1}{\omega_s^2}\operatorname{sgn}(x_1)\,(x_5 - x_4)\,\mathrm{e}^{-x_1^2/\omega_s^2} \tag{5}$$

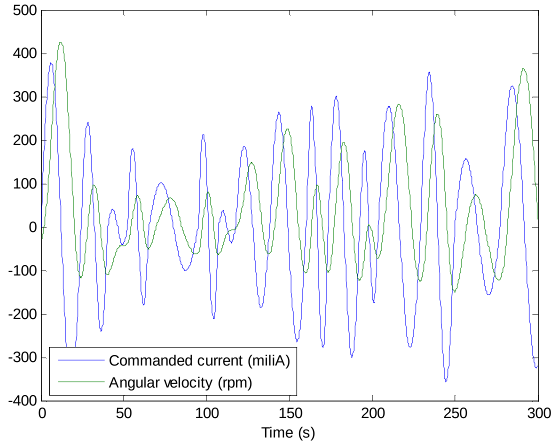
$$\frac{\partial \dot{x}_1}{\partial x_2} = I \tag{6}$$

$$\frac{\partial \dot{x}_1}{\partial x_3} = -x_1 \tag{7}$$

$$\frac{\partial \dot{x}_1}{\partial x_4} = -\mathrm{sgn}(x_1)\left(1 - e^{-x_1^2/\omega_s^2}\right) \tag{8}$$

$$\frac{\partial \dot{x}_1}{\partial x_5} = -\mathrm{sgn}(x_1)\, e^{-x_1^2/\omega_s^2} \tag{9}$$

The data were generated with the wheel subjected to two command profiles, both with small amplitude, so as to keep it at low speeds and with periodic reversals in the direction of rotation, as shown in Fig. 3. The first profile consisted of multiple sinusoidal cycles in which each period had the amplitude and the period chosen randomly within certain limits. The second profile had random amplitude, constant current in each actuation, and reverse direction every 30 seconds, similar to a square wave.



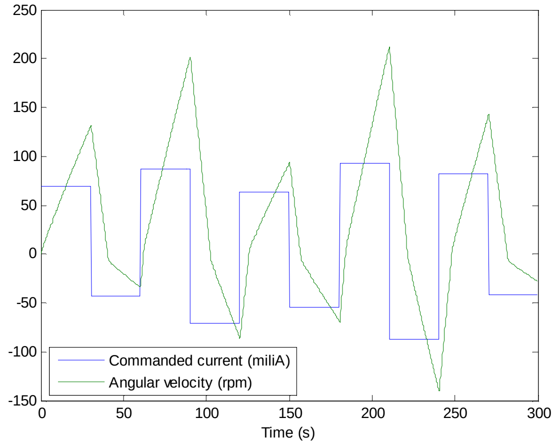




**Fig 3 – Sinusoidal (left) and stepwise (right) profiles commanded to the reaction wheel.**

The bearing temperature and atmospheric pressure inside the reaction wheel were monitored during the entire run of the profiles, with duration of 300 seconds each. Although it is plausible that the temperature influences on friction and, as a consequence, also in the behavior of the wheel, this influence has not been taken into account in this model, since the variation of both during the experiment was small, less than 1$^{o}$ C in temperature. Note that, particularly in Fig. 3 right, the Coulomb torque causes changes of inflection in the curve of the angular velocity when it reverses its direction of rotation. This is an indication that these experiments are able to provide information for this and other estimation parameters, which will be presented in the following section.

## ESTIMATION PROCEDURE

The procedure of parameter estimation from Eq. (1) was based on the batch least squares method. The weighted loss function $J$, considering a-priori information, in norm notation, is given by:

$$J = \left\|\mathbf{y} - \mathbf{H}\mathbf{x}\right\|^2_{\mathbf{R}^{-1}} + \left\|\hat{\mathbf{x}}_o - \mathbf{x}\right\|^2_{\mathbf{P}_o^{-1}} \tag{10}$$

where $\|\cdot\|$ represents the norm of a matrix or vector, $\mathbf{y}$ is the vector containing $m$ measurements, $\mathbf{H}$ is the $m \times n$ matrix that relates the measurements to the state $\mathbf{x}$ of $n$ elements, $\hat{\mathbf{x}}_o$ is the a-priori state value, $\mathbf{R}$ is the $m \times m$ covariance matrix of measurement errors, and $\mathbf{P}_O$ is the covariance matrix of the errors on the a-priori state. Initially the loss function is in the form:

$$J = \left\| \begin{bmatrix} \left(\mathbf{P}_O^{-1}\right)^{1/2} \hat{\mathbf{x}}_o \\ \left(\mathbf{R}^{-1}\right)^{1/2} \mathbf{y} \end{bmatrix} - \begin{bmatrix} \left(\mathbf{P}_O^{-1}\right)^{1/2} \\ \left(\mathbf{R}^{-1}\right)^{1/2} \mathbf{H} \end{bmatrix} \mathbf{x} \right\|^2 \tag{11}$$

where $\left(\cdot\right)^{1/2}$ represents a square root matrix of $\left(\cdot\right)$. Utilizing an orthogonal transformation **T** of e. g. Householder, that does not change the norm, one triangularizes the system so that:

$$J = \left\| \mathbf{T}\begin{bmatrix} \left(\mathbf{P}_O^{-1}\right)^{1/2}\hat{\mathbf{x}}_o \\ \left(\mathbf{R}^{-1}\right)^{1/2}\mathbf{y} \end{bmatrix} - \mathbf{T}\begin{bmatrix} \left(\mathbf{P}_O^{-1}\right)^{1/2} \\ \left(\mathbf{R}^{-1}\right)^{1/2}\mathbf{H} \end{bmatrix}\mathbf{x} \right\|^2 = \tag{12}$$
$$= \left\| \begin{bmatrix} \mathbf{y}_1 \\ \mathbf{y}_2 \end{bmatrix} - \begin{bmatrix} \mathbf{H}_1 \\ \mathbf{0} \end{bmatrix}\mathbf{x} \right\|^2 = \left\|\mathbf{y}_1 - \mathbf{H}_1\mathbf{x}\right\|^2 + \left\|\mathbf{y}_2\right\|^2$$

Note that after the orthogonal transformation, $\mathbf{H}_1$ is a $n \times n$ triangular matrix, **0** is a $m \times n$ matrix of zeros, and $\mathbf{y}_1$ and $\mathbf{y}_2$ are vectors of sizes $n$ and $m$, respectively, resulting from the application of the orthogonal transformation **T**. Therefore the minimum of the loss function is simply:

$$J_{\min} = \left\|\mathbf{y}_2\right\|^2 \quad \text{if} \quad \left\|\mathbf{y}_1 - \mathbf{H}_1\mathbf{x}\right\| = 0\,, \tag{13}$$

which is the least squares solution according to Lawson and Hanson (1974). Once the matrix $\mathbf{H}_1$ is triangular, the resolution of:

$$\mathbf{y}_1 = \mathbf{H}_1\hat{\mathbf{x}} \tag{14}$$

is trivial (back substitution), and $\hat{\mathbf{x}}$ is the estimated state vector. This approach was coded in Fortran and adapted (Kuga, 1989) to solve the non-linear problem of estimation of friction parameters. From an initial condition $\hat{\mathbf{x}}_o$ (a-priori), the solution is obtained iteratively and converges quickly in few iterations.

## ESTIMATION OF FRICTION PARAMETERS

Some of the friction parameters of this wheel were estimated by Carrara and Milani (2007) and Carrara (2010) in previous works. Because very specific methods for individual computation of the friction parameters were used in those works, they were named deterministic methods, in contrast with the statistical methods employed in this study. By the deterministic methods there were obtained the viscous friction coefficient $b$ = 5.16 $10^{-6}$ Nms, the Coulomb friction $c$ = 0.8795 $10^{-3}$ Nm and the motor constant $k_m$ = 0.0270 Nm/A.

In the parameter estimation procedure the departure state vector was set to:

$$\mathbf{x}_0 = \begin{pmatrix} 0 & 18 & 0.00344 & 0.5863 & 0 \end{pmatrix}^T \tag{15}$$

which correspond to the values of the deterministic methods, as defined by Eq. (3). The profile 2 was used to estimate the values of the parameters $x_2, x_3, x_4, x_5$. The profile 1 (sinusoidal) was used to validate the estimated parameters. In the least squares procedure one assumed that rotation measurements had a standard deviation of about 5 rpm. The state vector after convergence of the procedure was

$$\hat{\mathbf{x}}_0 = \begin{pmatrix} 0 & 15.205 & 0.00322 & 0.5863 & 0.6037 \end{pmatrix}^T. \tag{16}$$

Considering the inertia value $J_w$ = 1.5 $10^{-3}$ kg m$^2$, the friction parameters result in $b$ = 4.83 $10^{-6}$ Nms, $c$ = 0.8795 $10^{-3}$ Nm and $k_m$ = 0.0228 Nm/A. Table 1 shows the results obtained herein. It is realized that the highest difference was encountered in the motor constant, which was 15% below the deterministic method. The Coulomb torque did not present meaningful difference within the accuracy tolerance adopted in its computation.

**Table 1 – Friction parameters of the reaction wheel**

| Parameter | | Deterministic | Statistical |
|---|---|---|---|
| Motor constant | $k_m$ | 0.0270 | 0.0228 |
| Viscous coefficient | $b$ | 5.16 $10^{-6}$ | 4.83 $10^{-6}$ |
| Coulomb torque | $c$ | 0.8795 $10^{-3}$ | 0.8795 $10^{-3}$ |
| Static torque | $T_s$ | - | 0.9055 $10^{-3}$ |

Figure 4 shows to profile 1 a comparison of the measured (Fig. 3) and estimated speeds by the deterministic and statistical methods. Note that both methods present similar results, however the error with respect to actual measurements is still relatively high.

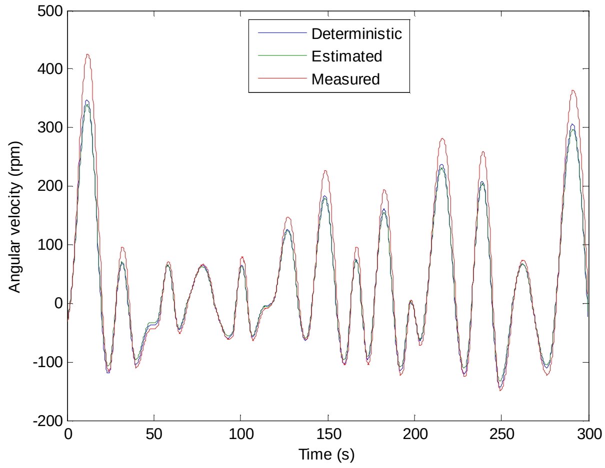



**Fig 4 – Results of deterministic and statistical methods in the profile 1.**

Figure 5 shows the same results for the stepwise profile 2 (Fig. 4). Note clearly the better closeness of the statistical adjustment (compared to the deterministic one) to the experimental rotation measurements.

Figure 6 shows the measurements (in red), and the residuals between the measured rotations and the values estimated by both methods (deterministic and statistical), for the sinusoidal profile 1. Increasing residuals near the zero crossings are verified, where the friction models has lower performance.

Figure 7 show the residuals between measured and estimated rotations for the stepwise profile 2. It is also quite pronounced the best performance of the statistical adjustment at low speeds (50-100 rpm).

The results indicate problems in the wheel response at low angular rates, mainly in transitions crossing the zero level. Nevertheless, the model obtained by the statistical estimation of parameters behaves better in this range. In practical terms, the use of this model in a control system provides a smooth transition through zero, and can eliminate the need to define a dead zone, facilitating the design and implementation of the control system. On the other hand it should be noted that the mathematical model used in both methods is symmetrical with respect to the direction of rotation. However there is evidence (Canudas and Åström, 1987) that bearings may be asymmetric, although the degree of asymmetry is in general small.

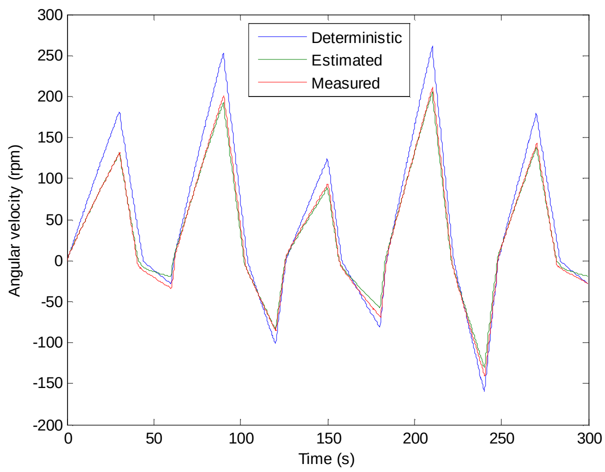



**Fig 5 – Results of deterministic and statistical methods in the profile 2.**

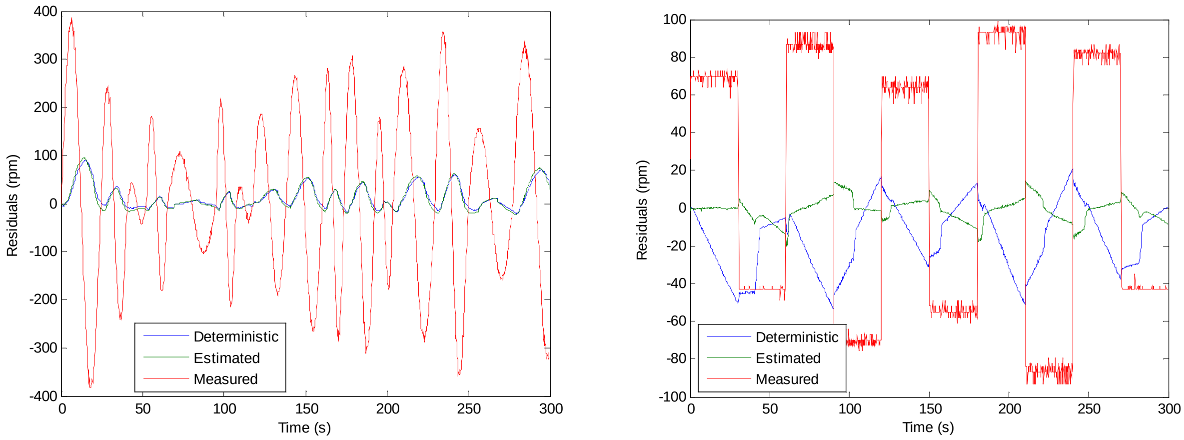



**Fig 6 – Estimation residuals of profile 1 (left) and 2 (right).**

## TORQUE MODE CONTROL

In order to emphasize the non-linear friction effect in the controller performance, a cooler fan was attached to the air-bearing table and oriented in such a way that introduces a small but constant torque. The initial velocity of the wheel was adjusted so that a zero-speed crossover occurs during the control action. A PID controller was used to control the attitude of the air-bearing table, based on the integrated signal of the FOG gyro. The PID gains were adjusted to minimize or to avoid the overshoot response in attitude, and were kept constant during the whole experiment. The air-bearing table dynamics can be modeled as a one-axis rigid body with inertia $J$ and the fan disturbance torque $T_d$:

$$J_w \dot{\Omega} = T_w + T_d \tag{17}$$

where $\Omega$ is the table angular velocity, as measured by the FOG gyro, and $T_w$ is the wheel's reaction torque.

Figure 7 shows the attitude error with a null reference signal, while Fig. 8 shows commanded current, equals to the PID signal, i. e. $I = u$ where $u$ is the PID output. The maximum attitude error occurs during wheel's reversion, at elapsed time of 230 seconds, approximately. The torque generated by the fan could be estimated based on the angular momentum variation, resulting 0.46 $10^{-3}$ Nm, and is practically constant. The attitude error reaches 1.5 degrees after zero-speed crossing, followed by an error of 0.2 degree in steady state. From controller viewpoint, this means that the pointing requirement is no longer accomplished during zero-speed crossing. Most of the error is due to the long time the integral controller takes to compensate the fast changing in the friction torque during wheel reversion. As it will be shown, the DMC controller changes the control signal as quick as the friction torque, allowing the PID to respond only to the external disturbance torque.

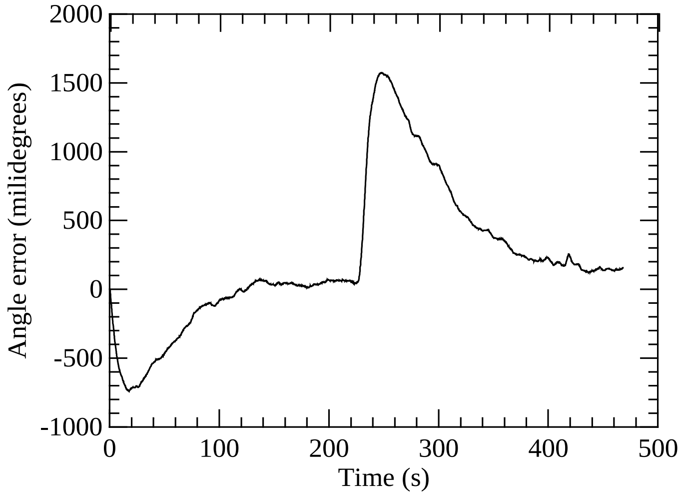



**Fig 7 – Attitude error during zero-speed crossing and with external disturbance (cooler fan).**

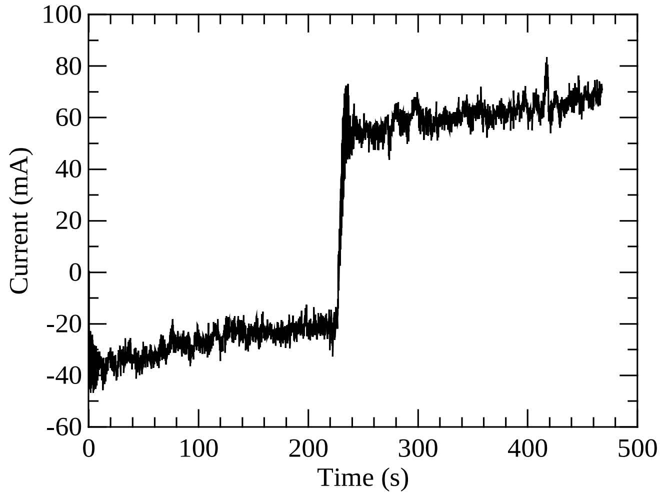



**Fig 8 – Control signal (motor current) under external disturbance.**

## DYNAMIC MODEL COMPENSATOR CONTROL

Using a nonlinear controller to handle the zero-speed problem of the reaction wheel is a natural consequence of the fact that the mathematical model represents the behavior of the wheel reasonably well. It is, therefore, straightforward to use this model as a nonlinear compensator for the controller, and to make the wheel action directly proportional to the PID signal (Canudas De Wit and Lischinsky, 1997). Since the table responds to only an acceleration of the wheel, the control command shall be in the form:

$$I = u + \frac{b}{k_m}\omega + \frac{\operatorname{sgn}(\omega)}{k_m}\left[c + d\, e^{-\omega^2/\omega_s^2}\right], \tag{18}$$

where $u$ is the PID control signal. For a null wheel angular velocity the compensator takes the form:

$$I = u + (c + d)\operatorname{sgn}(u) / k_m . \tag{19}$$

Figure 9 shows a simplified block diagram of the dynamic model compensator (DMC) control. The new controller was tested under the same condition as the torque mode control, but incorporating the dynamic compensation. As can be seen in Fig. 10 and 11 (analogous to Fig. 7 and 8) the error was almost negligible, with a maximum deviation of only 0.1 degree during wheel reversion and it took about 20 s to reach the steady state. The large error of almost 0.6 degree is due to the initial step response of the control at the beginning of the experiment, and shall not be considered as a steady error. The control signal is shown in black in Fig. 11, and separated in its two components: the friction dynamic compensator (in red) and the PID signal (blue curve). It is clear in this graph that the PID control is approximately constant, as it would be expected due to the disturbing torque of the fan. The PID controller gains were kept identical to those used previously, although they could be adjusted in order to achieve a better performance, since the dynamics is now almost linear due to the model compensator.

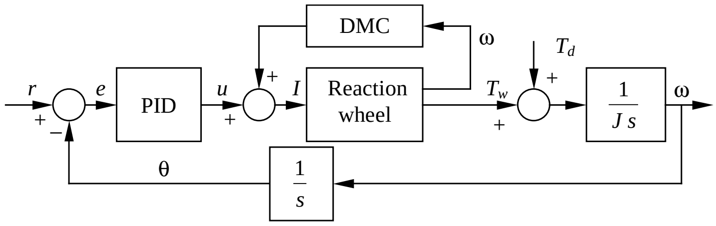



**Figure 9 – PID controller with RW dynamic model compensation.**

The effect of the Stribeck friction is barely seen in Fig. 11, which indicates that this friction is not so significant for the wheel's behavior. In fact, the same experiment was carried out without the Stribeck friction model in the DMC (not shown in this paper), which showed similar results. However, it is not recommended to simple neglect the Stribeck factor, since it introduces some sort of hysteresis that should be important during motor starting and reversing.

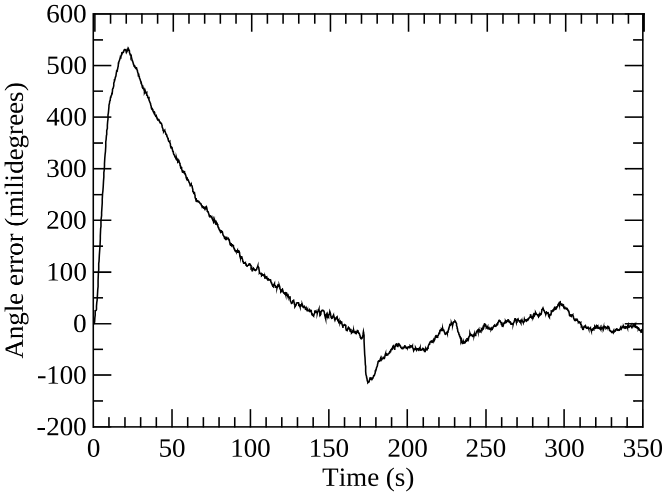



**Figure 10 – Attitude error of the air-bearing table with DMC control.**

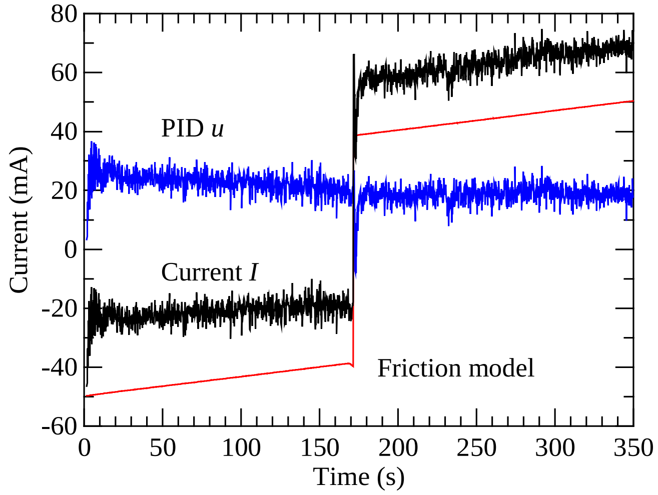



**Figure 11 – PID controller with RW dynamic model compensation.**

## CONCLUSIONS

This article presented a computational and mathematical model for a reaction wheel of SunSpace (Engelbrecht, 2005), obtained from non-linear models of Coulomb, viscous, and Stribeck frictions, based on testing and experimental measurements of the behavior of the wheel. Previous work did not include Stribeck friction, and the values of the parameters of friction (Coulomb and viscous) were obtained deterministically (Carrara and Milani, 2007; Carrara, 2010). Based on this more complete model, it was accomplished a non-linear estimation of states and parameters by the method of least squares, using data from two experiments: one with sinusoidal profile, and another with positive and negative levels, where the transitions by zero was exercised numerous times (24 times in the sinusoidal profile 1and 9 times in the stepwise profile 2). As expected, degraded performance of the models in the crossings by zero was noted, but with better fit of statistical method. A nonlinear Dynamic Model Compensator (DMC) for the reaction wheel control was then implemented in order to make the wheel behavior linear. The controller showed improved performance in this new condition and reached the maximum error of only 0.1 degree at zero-speed crossing. The DMC presented also smooth responses near zero, as expected, with errors smaller than the ones presented with the deterministic parameter estimation method (Carrara et al., 2011). Due to this, the compensator significantly reduced the nonlinear effects that occur in the response of the wheel during the reversals of direction, avoiding the model discretization and decreasing the complexity of the control synthesis in this type of actuator. Future works suggest the use of this model in a control system of position (angle) or angular velocity and corresponding performance comparisons in terms of response time, performance and accuracy.

## ACKNOWLEDGMENTS

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