Performance evaluation of a Ball Screw mechanism through a multibody dynamic model

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BERTOLINO Antonio Carlo[1, a *], DE MARTIN Andrea[1,b] and SORLI Massimo[1,c]

[1] Department of Mechanical and Aerospace Engineering, Politecnico di Torino, Torino, Italy

[a]antonio.bertolino@polito.it, [b]andrea.demartin@polito.it, [c]massimo.sorli@polito.it

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Abstract. Ball screws are mechanism to convert the rotational into linear motion and viceversa and are widespread in a variety of different sectors. A detailed high-fidelity dynamic mathematical model of such component is paramount in several fields and, in particular, in the definition of a PHM system for flight control EMAs in order to increase their reliability. In fact they can be used as a virtual test bench on which inject artificial defects and study their effect on specific indicators. This paper presents a MBD model of a single-nut ball screw with internal recirculation able of describing the full dynamic of each internal component allowing a more in-depth understanding of the system behavior and poses the basis for PHM-oriented analyses on different degradations.

Introduction

During last decades, a trend towards more electric aircraft has arisen. Several efforts have been made to replace currently wide-spread electro-hydraulic actuators with electro-mechanical ones (EMAs) for primary flight controls. However, the principal drawback is the increased jamming probability which makes this technology not suitable for safety critical applications in its base configuration. To overcome this issue a possible solution is to use a simple architecture EMA equipped with an efficient prognostic and health management system (PHM). To train this PHM algorithm and to understand the correlation between different parameters and the performance deterioration, a high-fidelity dynamic model is paramount: in fact it can be used as a virtual test rig on which inject artificial defects to study the system under nominal and degraded conditions and to extract meaningful features correlated with faults' extent, in a cost-effective way.

According to literature FMECAs, the ball screw is the most critical subcomponent of an EMA. For this reason, this component is the focus of the current research. The ball screw is the last and key component of the EMA's mechanical transmission, converting the rotational motion coming from the electric motor into a linear displacement. This component can reach extremely high efficiencies replacing the sliding with rolling friction. Its comprehension is essential to avoid performance degradations and catastrophic failures, such as jamming. However, experimental analyses to understand the internal behavior are fairly difficult because of its closed nature; therefore, several theoretical studies analyzed the system from different perspectives, such as internal kinematics [1,2], no-load drag torque [3,4] and recirculating units degradation [5,6].

Hitherto, most of the models used a quasi-static approach describing the motion of the element in steady state conditions. Unfortunately, these models are not suitable for PHM. To fill this gap, the authors developed a set of dynamic models with increasing complexity and realism in order to be used for analyses with different levels of required details. A unidimensional lumped parameter dynamic model was first created. Being the ball screw mainly a 3D mechanism, a MBD model was developed in Simscape Multibody environment [7] taking into account the full dynamic of each subcomponent as well as their mutual interactions through a dedicated contact model considering grease lubrication [8]. Several degradation model were inserted and investigated [9].

However, no recirculating channel model was present, hence the capability of the MSC ADAMS software to handle contacts between arbitrary-shaped bodies was exploited analyzing a ball screw with a simple end-cap type axial recirculation [10].

Differently from [10], an internal recirculation type with three liner circuits is considered in this work, representing a novel contribution in the modelling of recirculation, usually disregarded in literature studies. This paper presents a detailed three-dimensional multibody dynamic model of a ball screw with the aim to understand the underlying dynamic of internal elements and to make a further step towards a model-based PHM for this mechanism. The MBD model can accurately describe the kinematics and dynamics of each element within the mechanism and allows to understand the complex interaction between the gothic arch helical grooves and the spheres under time-variant speed and external loads. The spheres are restrained between the grooves only by the Hertzian contact with the grooves, while the motion is transmitted by friction. Each sphere can freely move within the grooves, creating either backlash or multiple contact points with variable contact angles. The capabilities of such model are presented by evaluating the overall performance and internal component motion. The considered ball screw refer to the one mounted on the experimental test bench which is being built in the laboratories of Politecnico di Torino [11,12] with the aim of validating the results of the current and previous models and getting more in-deep insights for prognostic purposes. The proposed model poses the basis for a complete understanding of the internal dynamics of the mechanism considering also the internal recirculating system.

Model definition

The developed model has been developed in the MSC ADAMS environments to exploit its general shape body contact feature. The ball screw under study has an only rotating screw shaft and a only translating not preloaded single nut, on which an external force is applied axially as disturbance. The motion is imposed to the screw shaft as speed law.

Geometry tessellation. The complex geometry of the screw shaft and nut bodies have initially been simplified by removing al the non-necessary features: the simpler the geometry, the less point are necessary for its accurate discretization, the faster the simulation. The resulting geometry, imported into ADAMS, is triangular tessellated using the RAPID algorithm in order to speed up the contact detection during dynamic simulation.

Contact detection. After a contact occurs, the contact point location is calculated. The solver computes the intersection volume, which may be single or, in general, multiple depending on the specific condition. Assuming an uniform density of the material, the centroid of the intersection volume is obtained, having the same center of mass of the intersection volume. This point represents the contact point, while the penetration depth, used as a penalization parameter to enforce the contact constraint, is calculated as the distance between the two points of each tessellated surface closest to the intersection volume centroid.

Normal contact model. The ADAMS solver contains a contact predictor that estimates the onset of contacts and modifies the integration time step accordingly. Theoretically, contacting bodies does not compenetrate. This can be usually obtained introducing a penalty regularization considering the penetration as a deformation of the bodies and enforcing the contact constraint with a reaction force, which can be calculated as the sum of an elastic and damping contributions where a smooth activation function modulates the damping contribution (of the order of 1 Ns/mm) with respect to the penetration itself, avoiding contact discontinuities due to non-null approaching speeds. The contact stiffness constant is calculated a priori from the knowledge of the geometry and exact curvature radii [13], and assuming a theoretical contact angle usually close to 45° [14], according to the explicit non-recursive method for Hertzian contacts proposed in [15].

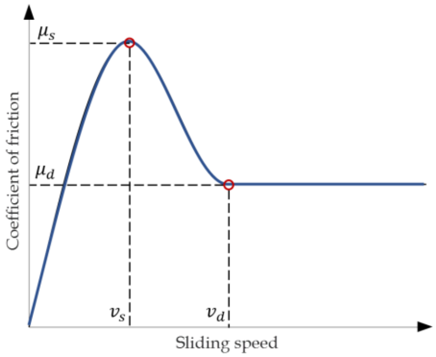

Figure 1 – Coefficient of friction model versus sliding speed.

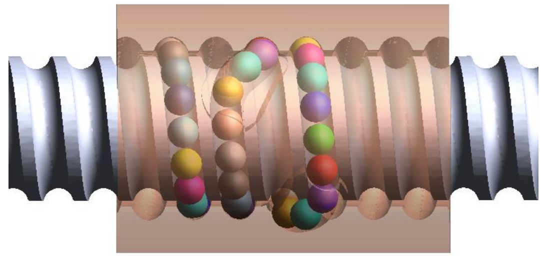

Figure 2 – ADAMS ball screw model with internal recirculating inserts.

Friction model. Once the contact point location is defined and the outward normal are identified, the normal and slip velocity can be easily obtained. The first is the previously cited approaching speed $\dot{g}$, while the second is the key parameter to calculate the tangential friction forces. ADAMS does not consider rolling friction resistance, but it only takes into account sliding and spin friction. There is no contact stiction in ADAMS, but a little amount of sliding speed is always required between the bodies in order to create a friction force: this is to ensure a zero force condition for the case of perfect adherence and to avoid numerical instabilities given by the abrupt transition between negative and positive speed values. Figure 1 depicts the coefficient of friction function versus the sliding speed: static and dynamic COF values and the relative threshold speeds must be chosen by the user to adapt the curve to the punctual problem. The shape of the COF function represents a lubricated contact with various lubricating regimes In this paper, the static and dynamic COFs have been set respectively to $\mu_s = 0.11$ ($v_s = 0.1$ mm/s) and $\mu_d = 0.06$ ($v_d = 0.15$ mm/s). The spin friction torque is applied to react to a relative angular speed about the contact normal axis and is calculated assuming an equivalent circular contact area.

Constraints. No ideal joints are imposed. Each rotation or translation to be constrained generate a reaction force/torque proportional to the constraint violation through an elastic-damping element, that has been constructed such that to allow also the presence of a little amount of backlash, considered null in the current paper. Two elastic joints have been imposed at the two extremities of the screw shaft to reflect the structural implementation of the experimental test bench which will be used to validate the current model [11,12].

Simulation results

This section presents the preliminary results of a first dynamic simulation of the presented model (Fig. 2), applied to a ball screw with a nominal diameter of 16 mm and lead 5 mm with internal recirculation. The three recirculating inserts are evenly spaced by 120° circumferentially.

Compliant joints have been considered for both the screw shaft and the nut. For the latter, only the rotational motion around the ball screw symmetry axis is constrained with an elasto-damping joint, while all the other rotations are not constrained. All the results will be presented referring to a fixed coordinate system, located on the ball screw symmetry axis and coincident with the lower face of the screw shaft, with the z axis correspondent to the ball screw symmetry axis.

The simulation has been performed imposing to the screw shaft a smooth speed step of -500 rpm from 0.15 s to 0.25 s while applying an axial external force as disturbance on the nut in the negative direction of the z axis, opposing to its imposed speed. Figure 3 depicts the nut linear speed and that of the screw shaft reported in the linear domain through the transmission ratio. Figure 4 shows the external force and the torque required on the screw shaft to realize the

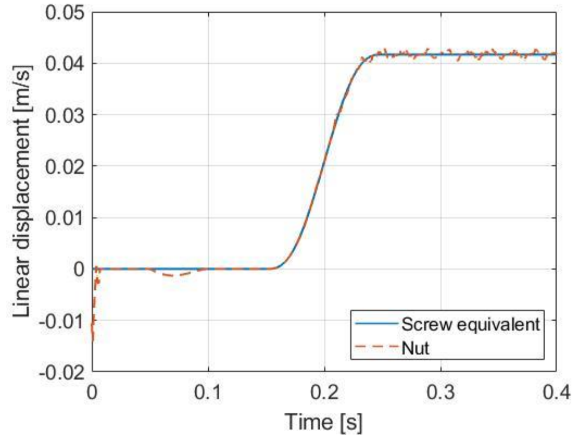

Figure 3 – Linear speed of the nut and screw shaft.

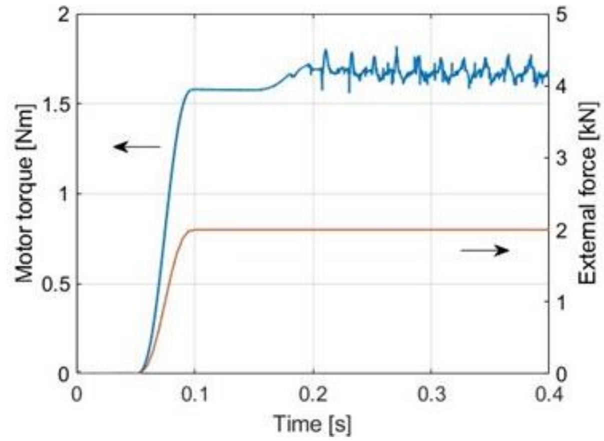

Figure 4 – External force on the nut and torque on the screw shaft.

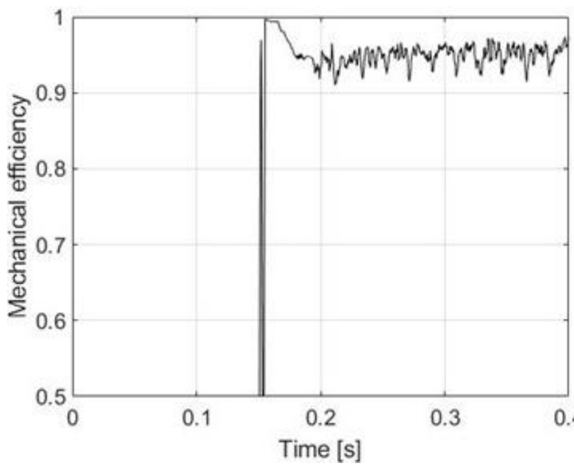

Figure 5 – Mechanical efficiency.

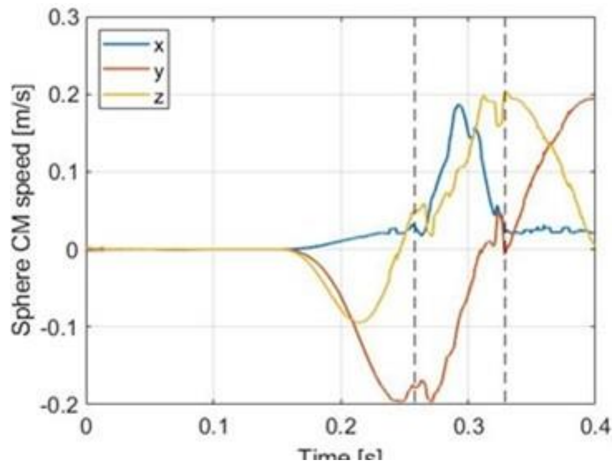

Figure 6 – Speed of the centre of mass of the selected sphere.

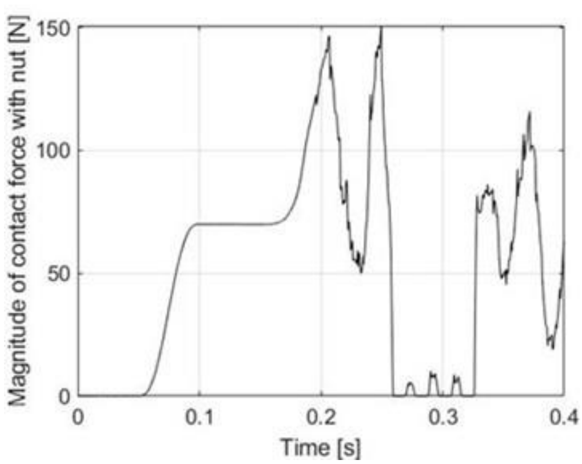

Figure 7 – Contact force between the nut and the analyzed sphere.

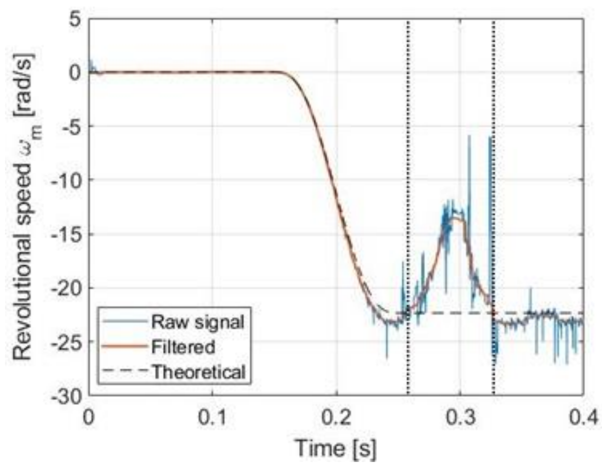

Figure 8 – Revolutional speed of the selected sphere.

imposed motion profile. It can be seen that when the external force is applied, though the screw shaft is steady, the nut slightly move because of the elastic deformations in the sphere/groove contacts. Furthermore, the presence of internal backlash is visible in the very beginning of the simulation on Fig. 3: in fact the initial configuration of the various bodies assumes the spheres and the nut in the nominal positions but, as the simulation starts, the gravity makes the system to reach a settle position. The motor torque rises to the level required to contrast the external force and after shows a little increment necessary to compensate the internal friction and guarantee the required motion. The system shows a realistic mechanical efficiency value of $\approx$ 93% (Fig. 5), typical of such mechanisms.

In order to clearly present the results, only the signals of one of the spheres are shown. Figure 6 illustrates the speed components of its center of mass. It is worth to be highlighted that the speeds of the spheres assume a sinusoidal shape when engaged in bearing load between the grooves while they varies when the spheres enters the recirculation. It can be seen around 0.3 s in the area between the two vertical dashed lines. A similar behavior is observable in Fig. 8, in which the revolution speed of the sphere with respect to the screw shaft around the ball screw axis is shown. The dashed lines represents the ideal revolution speed calculated according to [2]. When the sphere enters the recirculation, this speed sensibly decreases because its revolving motion is slowed down by the necessity of pass over the thread head to come back to the beginning of its motion path.

As observable in Fig. 3, because of the lack of other nut constraints but the anti-rotation around the symmetry axis combined with the slight asymmetry of the mass distribution and the presence of backlash, the nut oscillates and, hence, this reflects on the magnitude of the contact forces with the spheres. Figure 7 shows the contact force with the nut of the selected sphere. The contact forces originate with the application of the external force, remain steady until the motion starts and falls to near zero into the recirculation channel. The asymmetry of the nut causes a slight tilt of the nut, hence not every sphere bears the same load fraction. However, the mean value corresponds to the expected theoretical one [2]. The nut oscillation reflects also in the torque signal and finally in the mechanical efficiency. Because of the contact conditions, the spheres does not enter the recirculating inserts simultaneously on each recirculating circuit and this contribute to generate vibrations and nut oscillations.

Conclusions

A multibody dynamic model of a ball screw is presented in this paper. The innovation of such a model is the possibility to dynamically simulate the recirculation inserts analyzing the motion of the spheres within them, allowing the accuracy of the system to be sensibly increased. Firstly the model configuration is explained and finally the results of a preliminary simulation are presented. Further work includes the study of the friction increase in one or multiple recirculating circuits up to jamming, and the investigation on the effect of extraneous particles between the spheres. Last but not least, the experimental validation of the present model is scheduled as soon as the test bench will be fully functional. The current ball screw modeling is one of the activities included in the broader framework of PHM of electromechanical actuators ongoing within the authors' research group.

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