A RECIRCULATING BALL SCREW MECHANISM FOR A TELESCOPIC SPACE APPENDAGE

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ABSTRACT

The aim of this research is to design and verify a new kind of linear telescopic space appendage through the utilisation of recirculating ball screw mechanism. The high reliability, already widely checked in the aeronautical and industrial production, make this mechanism particularly attractive for a new generation of space applications. The requirements imposed by the space environment as well as by mechanical stresses have led to a first geometrical configuration and material choice which considers the highest performances of the mechanism, in terms of lower contact stresses and wear, smoother contact pressure distribution and longer fatigue life.

1. RECIRCULATING BALL SCREW

The recirculating ball screws are mechanisms which can transform rotary to linear motion and viceversa, transmitting power. The principal elements of the mechanism are the screw and the nut, both with threaded zone, and the balls. Contact between the screw threads and the nut threads is mediated by the balls which roll along suitably shaped race-tracks. At the end of the path allowed by the race-track, the balls are re-circulated by a variety of systems (trunnions) which bring them back to the beginning of the working path. Their main characteristics are summarized through the following points: 1) balls roll in the threading between screw and nut and the screw angle provide the axial movement; 2) the missing cage causes contact stresses between balls which are normally not present in usual ball screws [1], [2], [3]; 3) in the recirculating part, which is the most critical point of the design, balls are unloaded and pushed along by the following; 4) contact pressures are generally higher on screw than on nut because of the different diameter but nut threading, which is always loaded, are more subjected to fatigue; 5) nut deformability provides a non uniform distributed load between threads; 6) screw open air exposure makes particularly critical some aspects as cleaning and lubrication content.

Recirculating ball screws represent a basic component in a wide variety of both aviation-related and industry systems. In the aviation field, they are used essentially to impart motion onto movable wing surfaces (flap and slat) and to the stabiliser. Specific characteristics of the ballscrews are: high precision, low or zero backlash and extremely high efficiency (90-94%); for aeronautical purposes hollow screw have been developed. In a telescopic structure, each element shows both an internal and an external threading representing, at the same time, a screw (length l) and a nut (length s) (Fig. 1) and consisting in a set of elements, with circular cross section, nested inside each other, with a variable overlapping length (s).

In the deployment mechanism the system provides a deployment through a Telescopic Ball Screw (TBS) which may operate either 1) in dry conditions or 2) lubricated via solid paste. Dry mechanisms allow simplicity of design and represent a good opportunity for non inspectionable space applications which require long-life affordability [4], [5], [6], [7]. In fact in the case of low cycle number, the absence of lubricant and the utilisation of an Alternate Design (AD) characterised by ceramic and steel balls with different diameters are the ideal solution. The combination of new stainless steel (Cronidur 30) screws and ceramic balls in alternate configuration has been adopted on many airplanes. 7000 cycles in landing and taking off have been reached in a test without lubricant before exceeding the upper limit values for backlash

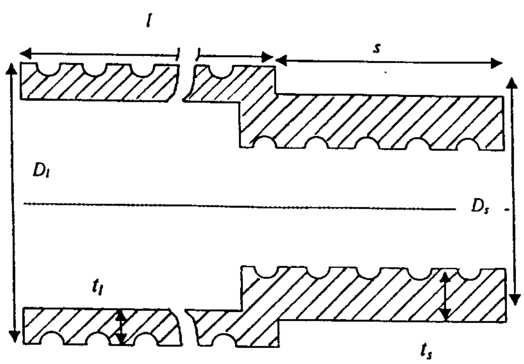

Figure 1: Generic element of a telescopic recirculating ball screw

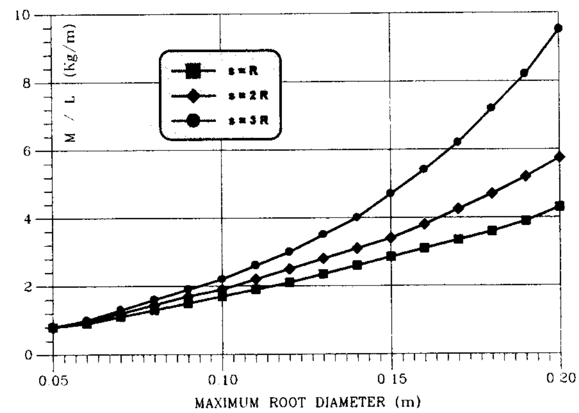

Figure 2: M/l ratio vs root element radius R

and the lower for efficiency. Despite these advantages, high deployment control and positioning - in radar interferometry, for example - require infinitesimal adjustments which may necessitate to decrease local wear through contact lubrication; in this case, Teflon, MoS_2 and solid paste could be utilised [8], [9], [10].

2. DESIGN CHARACTERISTICS

Possible utilisation of the proposed system may fall in different areas, ranging from platform to, mostly, payload applications: precise off platform positioning of devices and instruments and successive micrometric adjustments are the main goals for all the cases as either radar or GPS interferometry, sensor measurements or attitude control manoeuvres, etc.
Telescopic booms are generally preferred to tubular Stems when higher strength and stiffness, lower deployment length, precise positioning, good axial rigidity even during deployment are required. They appear particularly reliable and have been utilised as manipulator on Skylab and on the International Space Station to move the large Orbital Replaceable Units (ORU). The telescopic appendage [11], [12] is composed of several circular elements, which in stowed position are nested inside each other, and of a canister whose main role is to contain the whole stowed system. Its geometry and mechanical characteristics strongly depend upon stresses distribution both in launch and operating environments, but generally all the mission requirements can be summarised as follows: 1) providing the necessary stiffness to satisfy the minimum dynamic characteristics in stowed configuration, 2) reducing out-of-plane deployed oscillations to minimise overloads on ball screws, 3) deploying smoothly, avoiding sudden releases and shocks, 4) designing joints which can take care of different coefficients of thermal expansion, 5) limiting sub-components and parts to facilitate assembling and inspectioning, 6) providing the possibility to test the mechanism on ground.
Basing on the concept that each element is nested inside another, all the element have a different cross section, mass and inertia. The overall flexural rigidity of each element is

$$(EI)_{tot} = \frac{R_l \cdot R_s}{R_l + R_s} \tag{1}$$

where R_l is the flexural rigidity of the section of length l and R_s of the section of length s. These two rigidities are expressed by

$$R_l = EI_{yy} \cong E\pi(\frac{D_l}{2})t_l \qquad R_s = k_0 \cdot s \tag{2}$$

where D_l and t_l are respectively diameter and thickness of section of length l and k_0 a constant coefficient which provides a linear relationship between rigidity and length s. Usually all predimensioning operations for telescopic appendages require reducing 1) the transversal dimensions, which affects balls and threads geometry and 2) the longitudinal dimensions in stowed configuration which limit the maximum length $(l+s)_i$ of each element. Mass over length ratio vs the root diameter is plotted in Fig. 2, in which the overlapping length s is as parameter. From mission requirements, the total number of elements and overall mass are then computed. The coefficient k_0 in (1) is given by

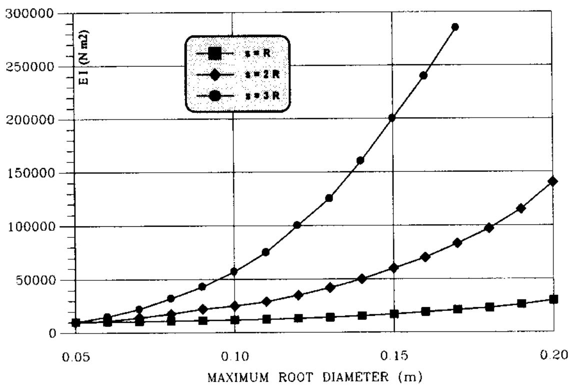

Figure 3: Element flexural rigidity *vs* root radius R

$$\frac{l}{k_0} = \frac{l}{k_1 s^2} + \frac{s}{3E\pi(\frac{D_s}{2})^3 t_s} \tag{3}$$

where k_1 is the rigidity of the set balls+thread. Ball rigidity K, according to Hertz theory, is a non linear function of the load applied on it F_N, which, on turn, depends upon total axial load F_A, screw angle ϕ, contact angle α and total ball number N,

$$\begin{aligned} K &= \frac{F_N}{y_{s/p}} = \frac{F_N}{A\sqrt[3]{F_N^2}} = \frac{\sqrt[3]{F_N}}{A} \\ F_N &= \frac{F_A}{N \cdot \cos\phi \cdot \sin\alpha} \end{aligned} \tag{4}$$

in which $y_{s/p}$ is the deflection at contact area and A is a constant depending on the elastic characteristics of materials. It's evident that K can not be computed since F_N is precisely known; it's worthwhile noting that loading distribution on balls is highly non-uniform and maximum and minimum values differ significantly from the average. In a traction loaded model, characterised by two trunnions placed at 180°, positive and negative radial displacements occur respectively for balls at 180° and 90° involving different compression stresses; recirculating trunnions therefore appear as the main reason for non uniform stress distribution. A numerical 3-D procedure through which this calculation is made possible is described in next section. Some results of experimental and numerical experiments are reported in Fig. 3 which plots, with s as parameter, element flexural rigidity *vs* root radius.

A telescopic composite appendage with a deployed length of 5 $m.$, a sustainable tip mass P of 30 $Kg.$, a root radius of 0.15 $m.$ and a length s of $3R$ has the following properties:

Number of elements	9
Maximum radius R_{max}	0.15 $m.$
Minimum radius R_{min}	0.118 $m.$
Mass Q **(**$\rho = 1600$**)**	20.04 Kg
Q/P	0.66
1st **bending freq.** ν_1	2.09 Hz

Table 1: Properties of a telescopic boom; length 5 $m.$ and tip mass P 30 $Kg.$

2.1. Bi-material configuration

Specific rigidity and stress are the parameters mostly used to drive material choice for space applications. Maximum values are obtained with composite materials and especially with either borum or carbonium fibers. These appear particularly attractive also for their low sensitivity to high temperature gradients and for almost negligible electromagnetic impact on the overall space system. Their utilisation has been here tested modifying an aeronautical Recirculating Telescopic Ball Screw (RTBS) with the following recommendations: 1) grinding and thread operations wouldn't generate the necessary rolling surface quality and surface hardness, which affects surface wear and deterioration, would result significantly small in comparison either with steel or ceramic balls; consequently 2) stresses between thread and balls would generate too large strains which would seriously worsen screw efficiency. RTBS are, then, here designed coupling composites (carbon fibers + epoxy), as main structural element, and steel (cronidur 30), as external covering necessary for the rolling of the balls.

Among all the available technologies through which this coupling could be made possible, gluing has been considered suitable for the present purposes. Two composite cylinders, one inner and the other outer, have been glued to an aeronautical steel screw: both have the same dimensions - an internal diameter of 4 $mm.$ and an external diameter of 8.4 $mm.$ - as reported in Fig. 4.

Shrewdness has been adopted for its realisation: 1) In order to avoid thermal stresses on composite cylinders which may originate from temper process, the steel screw has been first tempered and then made hollow, operating the gluing of external and internal

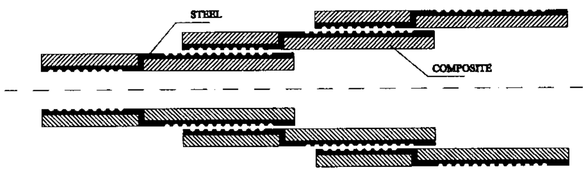

Figure 4: Geometry of a bimaterial screw

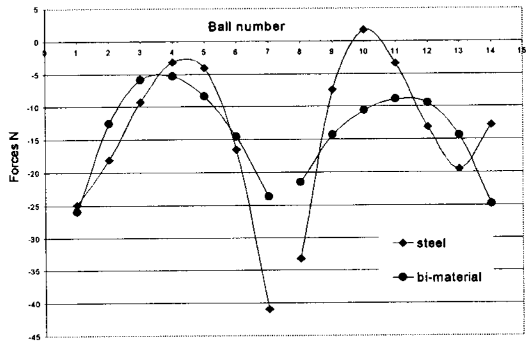

Figure 6: Comparison between a steel RTBS and a bi-material configuration

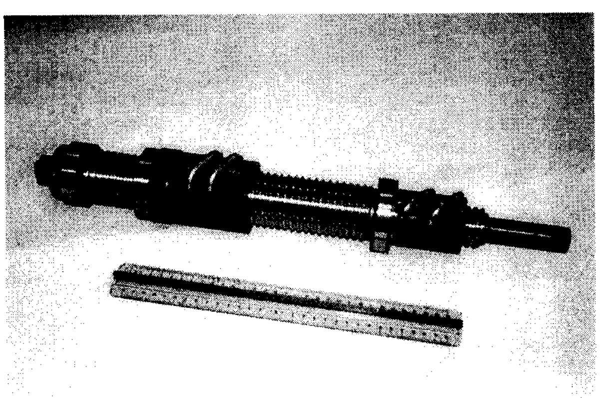

Figure 5: The realised bi-material screw element

cylinder subsequently; 2) considering that only 1.5 $mm.$ is the remaining thickness after grinding and threading operations on steel screw, these workings have been realised after the gluing of the composite cylinders which act as structural reinforcements; 3) recirculating trunnions have been positioned externally (Fig. 5) and this involved specific holes on composite cylinders.

A similar assembling, with two internal recirculating threads, has been subjected to fatigue life experiments and at the time of the publication of this work it has reached 10 million cycles. Simultaneously FEM static analyses have been performed to compare stresses distribution between the original RTBS, completely made of steel, and the present composite design. Inside each of the two internal threads 14 balls are present, 7 made of steel and 7 made of ceramic; the total number is therefore 28, 14 with larger diameter and 14 with smaller diameter (AD, Alternate Design). Steel balls, whose contact stresses computation is the objective of the analysis, have been simulated by gap elements connected with both screw and nut. Figure 6 shows forces distribution for the 7 balls of each internal recirculating threads: the composite design presents lower and smoother values on both threads resulting in a more uniform pressure and wear and, above all, in longer mechanical life. Together with the significant weight saving, the composite design appear here particularly favourable.

3. ON-BALL STRESSES

In a deployable recirculating telescopic ball screw mechanism [3] for space applications, which may not be inspected and must provide extremely high motion efficiency, contact stresses between balls and threads need to be analysed and optimised through numerical and experimental tests. Contact numerical analysis [13] is intrinsically non-linear because of boundary motions and friction on contact area and is generally performed through four main approaches: gap element method, Lagrange multipliers method, penalty method and non-linear programming. This last method is here adopted for all the 2-D and 3-D numerical analyses.

In order to calibrate the numerical model developed to compute on-ball stresses, 2-D point contact analysis, which observes a deformable cylinder and two deformable flats with the same mechanical characteristics, is performed and compared to Hertz theory [13]. This theory, which considers static conditions, small elastic strains and a limited contact area not influenced by either boundary conditions or bodies geometry, forecasts a contact area width a of 0.00116

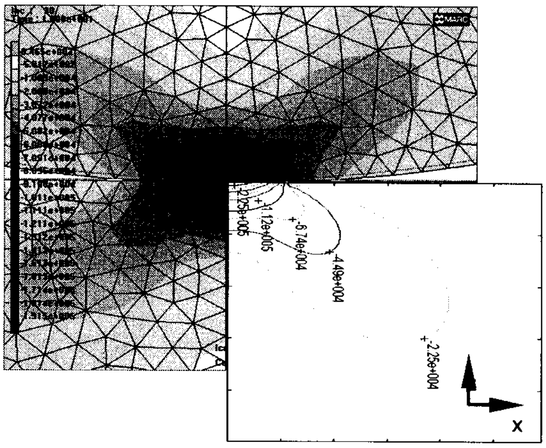

Figure 7: Comparison between numerical results and Hertz theory for σ_x

$cm.$ and a stress distribution (Fig. 7) which is perfectly reproduced by a mesh size h that follows the equation

$$\frac{a}{h} \geq 4 \tag{5}$$

The numerical model here adopted considers a ball radius of 2.778 $mm.$, a thread radius of 2.889 $mm.$, a compressive force P of 25 N and a material given by stainless steel Cronidur 30. Friction and contact stresses between balls and thread are computed combining numerical and analytical results through the following procedure: 1) elliptical contact, existing between balls and elliptical thread, is numerically modelled and is utilised to obtain ball radial deformation (δ=0.687 μm); 2) this value has been then inserted in the theoretical formulation of point contact to obtain the equivalent contact pressure between spheres. Hertz equations modified for contact between spheres are the following

$$a^3 = \frac{3PR^*}{4E^*} \qquad \Rightarrow \qquad a^2 = R^*\delta \tag{6}$$

$$\begin{aligned} p(x) &= p_0\left(1 - \left(\frac{R^*}{a}\right)^2\right)^{1/2} \\ p_0 &= \frac{3P}{2\pi a^2} \end{aligned} \tag{7}$$

in which R^* and E^* are respectively the equivalent curvature and stiffness, p_0 is the maximum contact pressure and $p(x)$ is the pressure distribution along contact area; 3) through p_0, stress distribution (σ_r, σ_θ, σ_z), given in a polar reference system with the origin on the contact point and z axis along balls centre, is easily determined via theoretical expressions. This distribution is plotted on balls surface (Figg. 8,9,10) and along z axis that means inside the balls (Figg. 11,12,13). From the first three graphs, it can be noted that half of the contact area width results approximately 0.03 $mm.$ and the maximum stress value, corresponding to an initial compressive force on screw of 25 N, is around 1500 MPa. Inside the balls, stresses σ_r and σ_θ vanish within 0.04 $mm.$ from the contact point, while σ_z has a significant value even beyond 0.08 $mm.$.

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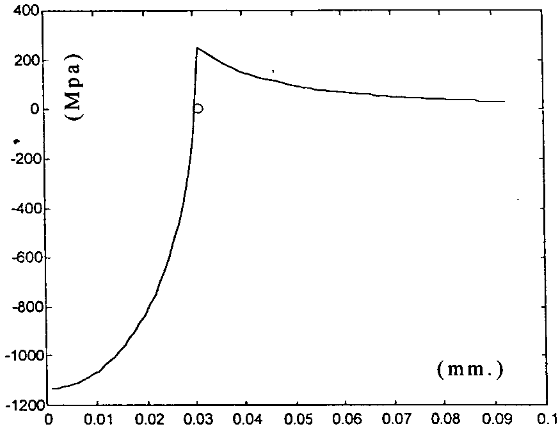

Figure 8: σ_r distribution along ball surface

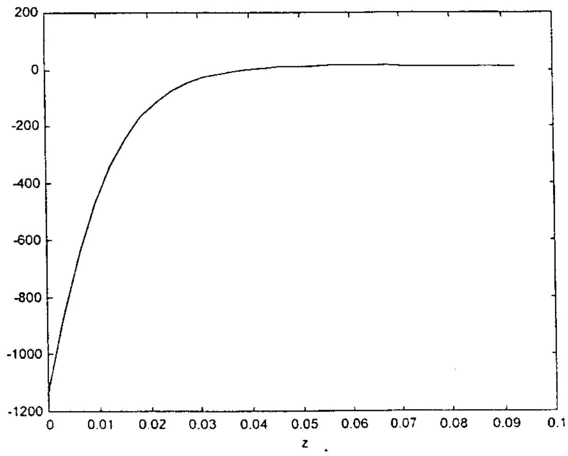

Figure 11: σ_r distribution along z axis

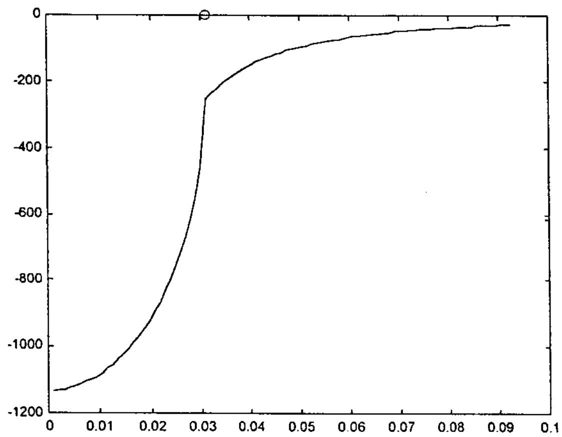

Figure 9: σ_θ distribution along ball surface

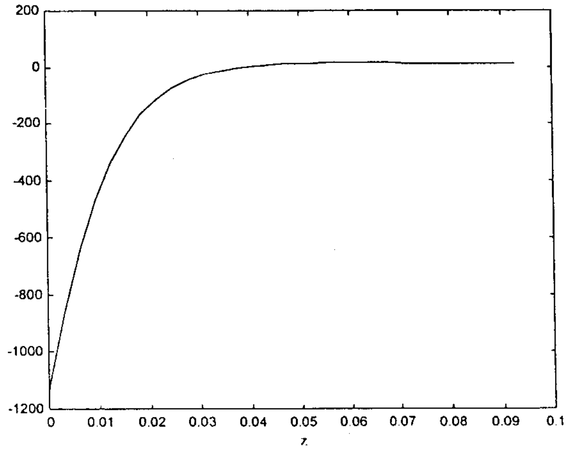

Figure 12: σ_θ distribution along z axis

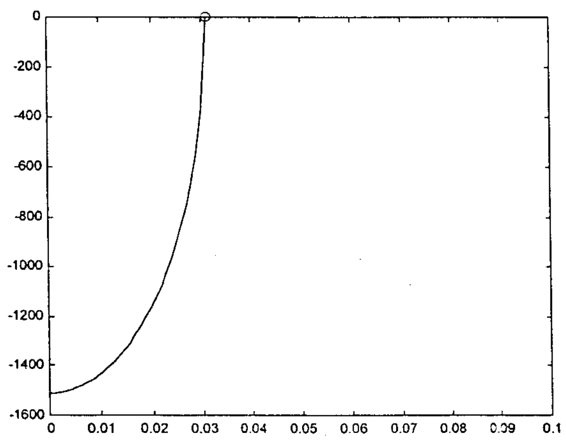

Figure 10: σ_z distribution along ball surface

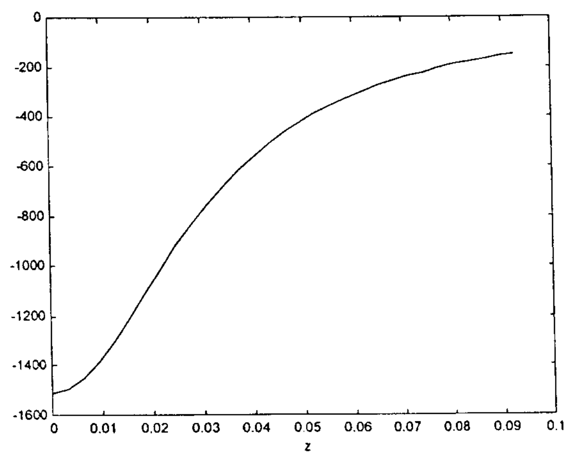

Figure 13: σ_z distribution along z axis