

A power recirculating test rig for ball screw endurance tests

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Abstract. A conceptual design of an innovative test rig for endurance tests of ball screws is presented in this paper. The test rig layout is based on the power recirculating principle and it also allows to overtake the main critical issues of the ball screw endurance tests. Among these there are the high power required to make the test, the lengthy duration of the same and the high loads between the screw and the frame that holds it. The article describes the test rig designed scheme, the kinematic expedients to be adopted in order to obtain the required performance and functionality and the sizing procedure to choose the actuation system.

1 Introduction

The mechanical screw transmissions concern the transforming of a rotational movement into a linear one and vice versa. Usually the input motion into the transmission is rotational and driven by an electric motor. According to the required application the movement is applied to the screw or to the nut. The two most important features of these mechanisms are the heightened precision in positioning and increased load capacity. For these reasons it is widely used in a large number of applications: from automatic machines to robotic systems and more generally in applications in which high precision linear movements are required.

Ball screw mechanisms are closed systems and it is difficult to experimentally investigate what happens between balls, screw and nut during the movement. Several mathematical models are developed to describe the behaviour of this mechanical transmission. The most important studies have been done by Levit [1][2] and in general these approaches are based on the settled theories developed for ball bearings [3]. More recently a revision of Levit's work has been proposed and more generally to the classical approach to the study of ball screw kinematic [4-7] introducing pseudosliding in nut-screw contact. Indeed the analysis of the movement of the balls during the working cycle has shown microslipping in the contact areas. This phenomenon has great effects both in the mechanical system design phase and in its use. In particular it causes a reduction of the maximum theoretical mechanical efficiency that is possible to obtain from the transmission [5] and links efficiency to the load operating conditions. The non-traditional approach to the study of recirculating ball screw mechanisms is not supported by a wide experimental analysis available in scientific literature. Thus it is difficult to characterize the parameters on which this

approach is based and the behaviour of the system related to their variation. The object of this work is to propose a new layout of endurance test rig capable of performing endurance tests under real load conditions and measure physics values useful in developing the mathematical models set out above.

In particular this paper shows the conceptual design of this device. The proposed layout is well suited for tests on large size ball screws (diameter to 50 mm) but for description of the system and the analysis of its performance a 25 mm diameter maximum ball screw has been taken into account. The originality of the system consists in the recirculating power principle applied in the ball screw test. This article sets out the functional groups constituting the test rig, with reference to the design choices carried out. The principal forces acting on the mechanical organs and on the drive system are evaluated by means of a simple mathematical model in order to define all the elements necessary for the design of the system.

2 Test rig description

The test rig uses the recirculating power principle borrowed from gear test benches concept [8]. To simplify it is possible to schematise a gear test rig, called "close cycle system" in two identical pairs of gears held at the end of two shafts. The gears are preloaded twisting the two shafts. This torsion is obtained by giving relative rotation to the two elements constituting of a joint by means of a predetermined torque. A motor moves a shaft introducing only the necessary power to overcome the friction while strong forces are transmitted among the gears due to the preload.

The same idea is duplicate for recirculating ball screws mounting two nuts on the same screw which are

pushed or pulled against each other by means of spring pack, as shown in figure 1.

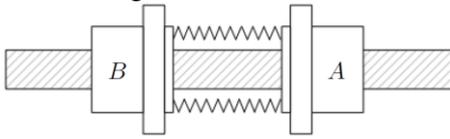


Figure 1. Recirculating power scheme for recirculating ball screws.

In this way the same force acts on the two nuts and the mechanical power resulting from these forces is nullified because they are equal in modules but opposite in direction. The inertia forces, the friction of the nut-screw contact, the friction of the slider-guide contact, the friction of the bearings that hold the screw are the only loads in action. It is necessary to point out that this layout, by nature, is particularly suitable for testing large size ball screws.

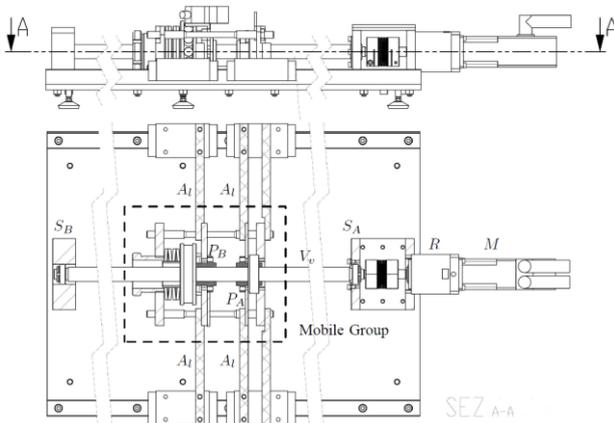


Figure 2. Test rig cross section.

Figure 2 shows a cross section of the design test rig. The screw (V_v), the supports that hold the end of the screw (S_A, S_B), the electric motor (M) and the reducer (R) that provide the movement of the screw, the mobile equipment and the two nuts are all shown. The screw is held by two SKF 6003 bearings: the one nearest to the motor is axially fixed while the other is free to move in such direction. Two steel plate (P_A, P_B) are joined to the two nuts. Two holes are made at the ends of the plates in order to fix four aluminium beams (Al_1, Al_2, Al_3, Al_4). These elements, hereafter referred to as wings, rest on four ball bearing slides (HIWIN HG-H-30-CA2R-1160-ZA-HII) capable to blocking the rotation of the nut and ensuring its translation.

Figure 3 shows a cross section of the mobile equipment. The two nuts mounted on the same screw, among which the load generate by the springs M_k , is shown. The spring block has been placed outside the two nuts in order to install the load system without disassembling the ball screw. We have chosen the belleville springs to generate the load because they are able to produce high force with small displacement. On the one hand the springs transmit to the plate (P_B) joined with nut B and on the other hand to plate P_A connected to nut A via four beams (Tr_1, Tr_2, Tr_3, Tr_4). The spring force passes through to SKF 811109 TN bearing before

discharging on plate P_B while the other passes through plate P_C , the beams, plate P_D and bearing SKF GAC 40 F before discharging on plate P_A . In this way the two nuts are pushed against each other and the screw between them is compressed.

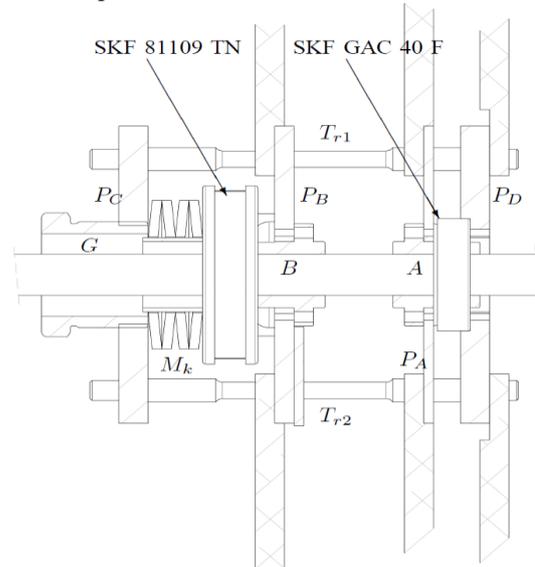


Figure 3. Mobile group cross section.

In order to have an axial load along the screw without load components in any other direction, the following expedients were taken into account: 1) insertion of an axial bearing (SKF 81109 TN) between the belleville springs and the plate joined to the nut B . 2) insertion of an orientable bearing (SKF GAC 40 F) between the nut A and the plate P_D .

In this way the two nuts are free to rotate one respect to the other along the screw axis and the system that generates the load is able to auto-align with the two nuts.

By means of the screwing or unscrewing of the ring nut G it is possible to modify the springs deflection and consequently to adjust the load that acts on the two nuts. Note that the plate P_D is joined to the beams and transmits the load directly to the orientable bearing and the latter does the same with nut A . This plate is equipped with two tiny wings. These are used only to keep the position of the mobile equipment during the setup phase of the machine in order to facilitate this task. Finally the screw is actuated by means of a motor-reducer group adequately sized and joined to the screw by a coupling. When the mobile group moves in steady state conditions, the only power required by the electric motor is that to overcome the friction forces within the screw-nut contact, within the sliders that block the rotation of the nuts and within the ball bearings that hold the screw. A device with adequate instrumentation as set out before, allows one to carry out the endurance test on the screws with pay load requiring the least possible mechanical power and thus making the trial intrinsically faster and cheaper.

3 Mathematical model

A mathematical model of the system is necessary to evaluate the required motor torque during the endurance

test and the reaction forces on the anti-spin system. This model has to be as simple as possible and must accurately estimate these forces, to be use in the design phase of the test rig. It will be noted that the system is not simple to analyse because it is hyperstatic. Every nut is fixed to the ground both by the screw, which blocks the translations in all directions except along the screw axis, and by two tiny wings, which block the spinning around the screw. The motor ``sees" the translating block, in which the recirculating power takes place, as a black box dissipating mechanical power due to the non ideal transmission efficiency. To evaluate the power dissipation it is necessary to enter into this black box and to analyse what occurs internally.

3.1 Model simplifications

The mechanical system is shown in a simplified way in figure 3. In particular the following components are taken into account: the screw, the two nuts and the tiny wings. All the masses of the components not taken into consideration in the scheme are adequately distributed among the previously components, in order to maintain their inertial contribution. Note that by reason of this simplification the position of the center of mass of each component is not retained, thus the moments of the weight forces on the constraints are neglected.

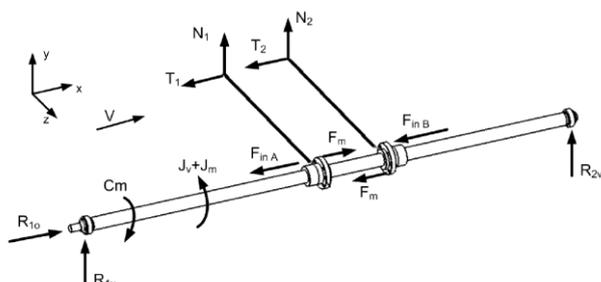


Figure 4. Simplified test rig scheme.

Figure 3 shows the external forces acting on the system, the constraint reaction forces on the slider that block the rotation of the tiny wings, the constraint reaction forces on the bearings, the motor torque and the inertia forces. Further simplifications have been effected to simplify the analysis of the system: 1) the weight force is neglected. This simplification is acceptable because the weight of the mobile pack is applied almost entirely on the sliders that hold the rotation of the nuts, hence it does not significantly modify the efficiency in the screw-nut contact; 2) a tiny wing of the anti-spin system for each nut has not been taken into consideration so as to make this constraint isostatic. The value obtained has to halved to consider the missing wing.

For the purposes of this mathematical model a complicated description of the thread-ball contact is not used. A recirculating ball screw has been studied as a normal screw where the friction force in the screw-nut contact is evaluated by the value of the efficiency supplied by the ball screw maker. The coupling of screw and nut can be represented by two wedges each with a single degree of freedom: The geometry of each wedge has to respect the tread characteristics (diameter and

pitch), and the relationship between the pitch and the slant of the thread is: $\tan(\alpha) = p / (2\pi r_v)$ where α is the slant of the thread, p is the lead of the screw and $r_v = D_p / 2$ is half pitch diameter (D_p).

Introducing the friction coefficient in the contact between the wedges it is possible to get the transmission efficiency (direct and indirect) by means the following equations:

$$\eta_{dir} = \frac{\tan \alpha}{\tan(\alpha + \varphi)}, \eta_{inv} = \frac{\tan(\alpha - \varphi)}{\tan \alpha} \quad (1)$$

It is possible to obtain the values of the screw efficiency from the manufacture's catalogue. In effect these values depend on several factors, for example, the load and the temperature. The friction coefficient and the friction angle φ are tied by the following equation:

$$\varphi = \arctan\left(\frac{f_G}{r_b \sin \alpha}\right) \quad (2)$$

From the literature [2] and from experimental tests [9] the size of the friction coefficient is $f_G = 0.001$ cm.

3.2 Non recursive system model

On the hypothesis that the friction force on the slider is omitted, it is possible to simplify the equations that allow one to obtain the required motor torque during an endurance test. In this manner the resolution method is not recursive. The inertia forces are defined completely when the mass of each component and their motion law are chosen. The test rig is divided into two subsystems named *A* e *B*. Each subsystem contains a nut and a part of screw. Inertia forces are applied to these and called F_{inA} , F_{inB} while the screw inertia torque $J_v \omega_v$ is divided into equal parts on each subsystems. The spring force (F_m) acting on the two screws is an external force taking into account the view of each subsystem. Due to this force the two nuts, that are fixed rotationally by means of the tiny wings, try to rotate the piece of screw between them in opposite directions. So the screw in this zone is subject to torsion. In the equivalent scheme based on the wedges this torsion is described as a resistant force opposite to the translation of the wedge-screw, as shown in figure 5. The module of this force is equal to C_v / r_v where C_v is the resistant torque and r_v is the pitch radius of the screw. Note that nut *A* is close to the motor while the other nut is called *B*. In the equivalent scheme based on wedge the nut *A* ``sees", in addition to the resistant force, another horizontal force due to the action of the motor (C_m / r_v).

The variables of this subsystem are two: the screw resistant torque C_v and the torque required by the motor C_m . Figure 5 shows a summarized scheme of the two subsystem *A* e *B* with all the forces applied. Note that between the two wedges there is a contact force *S*. Without friction between nut and screw this force would be perpendicular to the contact surface otherwise a friction force *T* is present and opposite to the motion of the two wedges. The component of *S* perpendicular to the surface, called *N*, has a different direction in the two subsystems *A* and *B* as is the force (F_m) generated by the spring on them. Subsystem *A*, closes to the motor, has two variable C_v and C_m at the same time, while the

subsystem *B* has only the variable C_v (resistant moment). Firstly we can resolve the equilibrium equations of subsystem *B*, in order to obtain the common variable, and then those of the subsystem *A* to obtain the motor torque C_m required to move the mobile equipment.

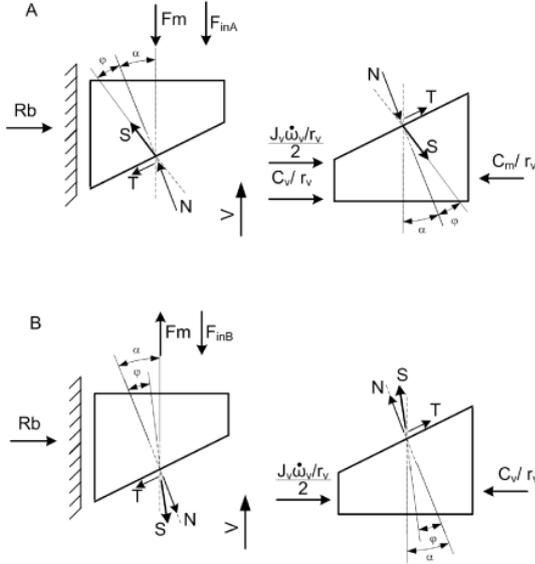


Figure 5. Subsystems A and B.

3.2.1 Solving subsystem B

Solving the equilibrium equations of subsystem *B* we obtain the variable C_v in the function of the spring force F_m applied on the two nuts. In particular we have:

$$\begin{cases} \frac{C_v}{r_v} = -N[\sin \alpha - \tan \varphi \cos \alpha] + \frac{J_v \dot{\omega}_v}{2r_v} \\ F_m - F_{inB} = N[\cos \alpha + \tan \varphi \sin \alpha] \end{cases} \quad (3)$$

From the second equation of 3 it is possible to calculate N while from the first equation we get C_v in function of the spring force F_m acting on the two nuts:

$$N = \left(\frac{F_m - F_{inB}}{\cos \alpha + \tan \varphi \sin \alpha} \right) \quad (4)$$

$$C_v = \frac{J_v \dot{\omega}_v}{2} - (F_m - F_{inB})r_v \tan(\alpha - \varphi) \quad (5)$$

As shown in figure 6, the wing fixed on subsystem *B* has to sustain the constraint reaction N_b equal to the ratio between the torque C_v and the gap l between the guide and the screw axis:

$$N_b = \frac{2C_v - J_v \dot{\omega}_v}{2l} \quad (6)$$

Obviously this value has to be divided in two in order to consider the presence of the two wings.

3.2.2 Solving subsystem A

By virtue of the scheme shown in figure 5 and knowing the value of C_v in respect to the spring force F_m , it is possible to resolve subsystem *A* and to obtain the value of C_m . The equilibrium equations are similar to those of subsystem *B*. The variable C_m , in function to the spring force F_m is stated as follows:

$$C_m = J_v \dot{\omega}_v - (F_m - F_{inB})r_v \tan(\alpha - \varphi) + (F_m - F_{inA})r_v \tan(\alpha + \varphi) \quad (7)$$

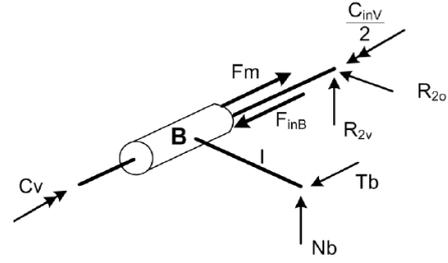


Figure 6. Subsystem B.

As shown in figure 7, the wing fixed on the subsystem *A* must sustain the constraint reaction N_a equal to the ratio between the resultant torque on the nut and the gap l between the guide and the screw axis:

$$N_a = \frac{(F_m + F_{inA})r_v \tan(\alpha + \varphi)}{l} \quad (8)$$

It can be noted that the two constraint reactions N_a and N_b , have different modulus and opposite direction according to the spring force that tends to rotate the piece of the screw between the two nuts in opposite directions.

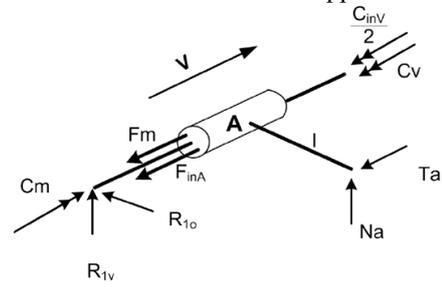


Figure 7. Subsystem A.

3.2.3 Inversion of the rotation of the screw

The developed mathematical model takes into consideration a certain sense of rotation and so a translation direction of the mobile equipment as shown with letter v in figures. The reversal of the sense of rotation implicates a different reply from the system because of the friction forces changing direction. Indeed friction forces are always opposite to the direction of the velocity. Revisiting the mathematical steps presented in previous sections it is possible to obtain the equations necessary to calculate the motor torque and the value of the slider constraint reaction in such working conditions.

4 Motor reducer sizing

4.1 Motion law definition

The nominal life of a ball screw is defined as the number of revolutions that it can reach. The lifetime is closely linked to the payload and accordingly two different load conditions are defined: 1) Dynamic Load C_{da} , the constant unidirectional axial load that a certain number of equal ball screws are able to bear without problems for 1 million of revolutions 2) Static load C_{0a} , the constant

unidirectional axial load applied in static condition that generates, in the maximum stress area, a permanent deformation of 0.0001 times the diameter of the rolling ball. If we apply to the screw the load C_a its lifetime is 10^6 revolutions. When loads are smaller the lifetime is longer and it follows the equation: $n=(C_d/C_e)^3 10^6$. Where the variable C_e represents an equivalent load applied on the screw during the working cycle.

Having chosen the applied load, lifetime of the screw only depends on the motion law or in other words the rise h and the time t_a for the rise h . In particular for the case study the value of the rise h is the movement of the nuts mounted on the screw and it must be at least equal to the use length of the nut and no greater than the distance between the two nuts. The limit to the maximum rise is due to the fact that every nut must work over its piece of screw without involving the screw used by the other nut. It can be noted that during the design of the test rig the distance between the two nuts is an important parameter and it must be at least 1.25 times the length of a nut subject to test. The time t_a for the rise depends on the maximum velocity of the motor, the reducer and the screw, values available on the manufacture's catalogues.

4.2 Motor reducer sizing

Generally the choice of the motor and the reducer has to be made at the same time. The continuum operation area in the speed-torque performance curve of a brushless motor can be schematized as a rectangle limited by the torque C_n whereas the maximum torque curve $C_{m,max}$ defines the short term operation area [10].

Usually C_n is constant up to the maximum allowed velocity $\omega_{m,max}$ whereas $C_{m,max}$ decreases starting from a certain value of the velocity ω_m . The value of the torque C_n is obtained by thermal evaluation. Supposing that the dissipated power of the actuator and transformed into heat, is proportional to the square of motor torque C it is necessary that root mean square of motor torque $C_{m,q}$ be inferior or equal to the value of C_n . For a proper sizing of the motor not only should $C_{m,q} \leq C_n$ but also the time t_a must be less than the motor thermal constant τ_{th} . The optimal choice of motor and reducer is a complex task [11], because the following conditions must be verified together: 1) Maximum torque limit: $C_{max} \leq C_{m,max}$. 2) Root mean square torque limit: $C_{m,q} \leq C_n$, 3) Maximum velocity limit: $\omega_{max} \leq \omega_{m,max}$

A sizing procedure capable of considering all the aspects before described is presented in [11][12], applied in [13] and it has been used for choosing motor and reducer of this test rig.

5 Conclusions

The paper presents a conceptual design of an innovative test rig for endurance tests on recirculating ball screws. The originality of the layout is the use of the recirculating power principal by which the test rig can perform tests with a minimum required power, in a short time and by means of a simple machine. Due to these qualities the proposed solution can be used for large size screw. The

paper shows the key elements of the test rig analysing the main construction aspects and through the development of a simple mathematical model regarding the sizing of actuation system. The conceptual design of this new layout has been developed and for design convenience a 25 mm diameter ball screw has been taken into account, but this layout, by nature, is particularly suitable for testing large size ball screws.

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