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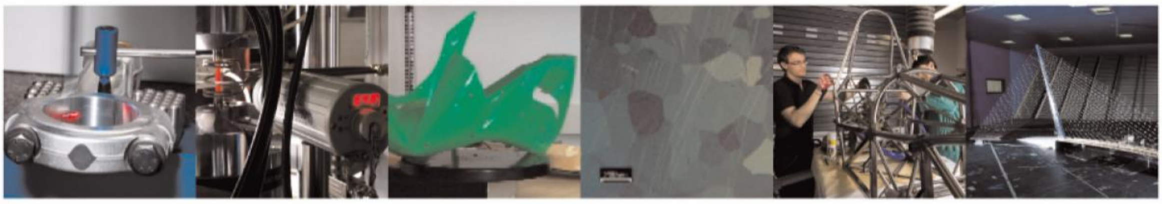

Tempered Wire Fatigue Testing

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Abstract

A new bench for the rotating bending fatigue tests of tempered steel wires is presented. The new bench is used to check the spring wire just before it is finally winded to realize a spring. The bench is basically a four-point bending machine. There are two main differences with respect to current bending machines. The first one is that the focus is on semi-finished components (more than 1 meter long), rather than standard small-scale specimens. The second one is that there is a non-linear configuration of the tested component due to its length.
The bench design has provided some unreferenced features that make the bench quite accurate and effective in producing quick fatigue assessments. A rotor-dynamic study has allowed to perform tests at 50Hz. As a preliminary application, some fatigue bending tests of tempered steel wires are described and discussed.

Introduction

It is of crucial importance checking the fatigue properties of wires before springs are manufactured, in order to assess and monitor the process performance. Such a specific attention on fatigue is due to the fact that springs (both helical and straight torsion bar) are supposed to undergo "very high cycle fatigue" [1], even in the order of giga-cycles [2][3]. Moreover, the current research on lightweight and compact construction solutions imposes the development of spring steels with higher mechanical and fatigue properties. Such conditions imply a high sensitivity of the material to defects. The correctness of each step of the manufacturing process must be checked and recorded. In particular, it is crucial that the steel wire from which the spring will be realized is (nearly) free from defects and provides the supposed fatigue properties. In fact, defects (either surface, sub-surface or inclusions) can lead to very high local stress concentration, detrimental for fatigue life [4][5].

Only the direct test of wire sections as provided by the manufacturing process can allow to assess the quality of the semi-finished component before the spring is actually realized. Referring to the testing of steel wires, different attempts to realize testing benches can be found in the literature[16]. In all of the presented papers, steel wires are tested under rotating bending load conditions. Such configuration allows for simple and effective assessment of the fatigue properties of the wire, while requiring a relative simple hardware. In [7][8], a Haigh–Robertson rotating–beam machine [9] is used for the characterization of wires in rotating bending conditions. The machine was designed to apply a constant bending moment to long span wires by tilting the two supports at the extremities. At one extremity, a motor is connected to rotate the wire. Given the high clamping forces at the motor connection, most of the ruptures were localized at the clamping. To avoid such a problem, wires with reduced section in the middle of the span were employed. In this way, however, a machining of the specimen was necessary.
In [10], a four point rotating bending machine has been presented. The machine was composed by three supports while the fourth bending point was realized directly by the motor connection. This, in our opinion, causes a too high stress concentration at the motor side of the wire. Wires were not directly connected to the support but a vinyl chloride bushing was interposed between each bearing and the wire in order to reduce stress concentration and reduce the occurrences of ruptures in correspondence of the supports. Commercial four points rotating bending machines for wires are available [11][12]. In such cases, four orientable bearings are used as supports, while the motor is mounted on a tilting mechanism and follows the movement of the wire extremity. Again, we think the wire is overloaded. Special bushings with concave profile are used to avoid stress concentrations at the support locations.
In this paper, a new test bench for rotating bending of wires[16] is presented. The test bench is similar to [10]-[12] with some relevant differences.
The paper is organized as follows. In the first section, the test bench designed for the durability test of tempered steel wires is presented. Then, the design of the plastic bushings and the dynamic behavior of the test bench are discussed. Finally, some preliminary experimental tests are presented.

Four point rotating bending

We focus on simple hardware, high test frequency and a wide range of maximum stresses. Four point rotating bending is usually employed for the fatigue test of wires. Figure 1 shows the scheme of our four point rotating bending machine.

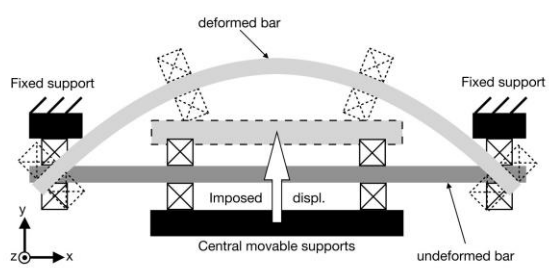

Figure 1. Scheme of the proposed four point rotating bending bench.

The wire is deformed by moving the central supports which, in this configuration, are rigidly connected. By this arrangement, different displacements of the central supports are avoided assuring a constant bending region of the wire. In [12], the central supports are independent and a load cell is connected to each support to monitor the actual reaction.

When rotating, the surface points of the wire will experience a cyclic load between the maximum deformation in compression and the maximum deformation in traction. As pointed out in [13], the rotating bending test differs from a pure bending fatigue test for the effect of the hysteresis cycle that causes a phase angle between the deformation and stress plane, while in pure bending the two planes coincide. However, the effect of the hysteresis depends on the stress amplitude and on the material. In this paper, wires of high strength steel for which the elastic limit is close to the ultimate mechanical strength are considered. In this case, the plastic deformation is small and the hysteresis effect is negligible.

In Figure 2 top, the reaction forces and the bending moment acting on the wire when subjected to a four point bending loading scheme are reported. Region (2) represents the zone of the wire where a constant bending moment is present. Any point of this region has the same probability of failure. The actual failure will occur in correspondence of inclusions or defects on either the surface or the sub-surface region of the wire. It has to be empathized that on the borders of region (2), i.e. where the reaction loads *P1* and *P2* are applied, the bending moment reaches the maximum value. It means that, to avoid localized ruptures in the correspondences of the supports, the supports must be suitably designed in order to avoid any stress concentrations.

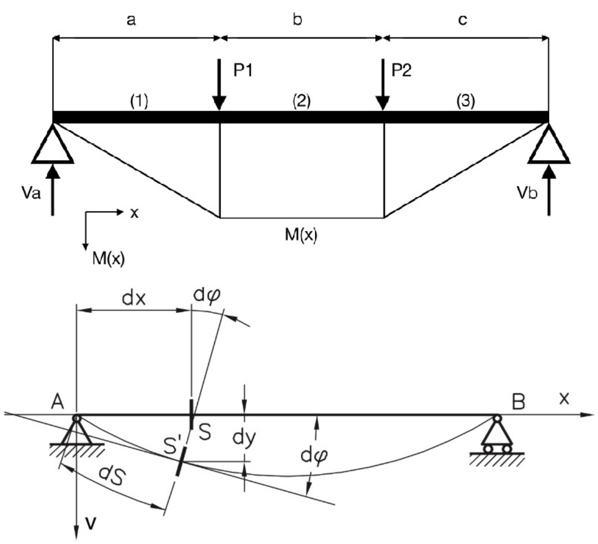

Figure 2. Four point rotating bending test. Top: reaction forces and bending moment. Bottom: notation for the computation of the deformation of a beam.

By considering the notations of Figure 2 bottom, the curvature of the beam can be computed as

$$\frac{M(x)}{EJ} = \frac{d^2y/dx^2}{[1+(dy/dx)]^{3/2}} \quad (1)$$

where *M(x)* is the applied bending moment, *E* the elastic modulus of the material, *J* the (constant) moment of inertia of the section and *y(x)* is the transversal displacement of the wire, with first derivative *dy/dx* and second derivative (curvature) d^2y/dx^2. Eq. 1 refers to the curvature moment equation of the beam for large displacements. The equation could be simplified for small displacements by considering *dy/dy≈1* if the transversal displacement is less than the 10% of the span of the beam. In the present case, if high values of stress are reached, this condition is not met and the formulation for large displacement has to be considered.

Eq. 1 is a differential equation of the second order. The solution of Eq. 1 can be found by numerical integration along the axis of the beam. The resulting solution is function of:

- lengths of the sections of the wire (parameters a, b and c of Figure 2)
- imposed displacements of the supports u_1 and u_2
- parameters of the section of the wire (*J* and *E*)

Design of the test bench

In Figure 3 the scheme of the four point rotating bending test bench is reported, while Figure 4 shows the 3D drawing of the bench (top view). The bench consists of two fixed supports and two supports constrained to have the same displacement. The motor is connected to the wire by a special elastic joint and it is free to move to accommodate the deformation of the wire.

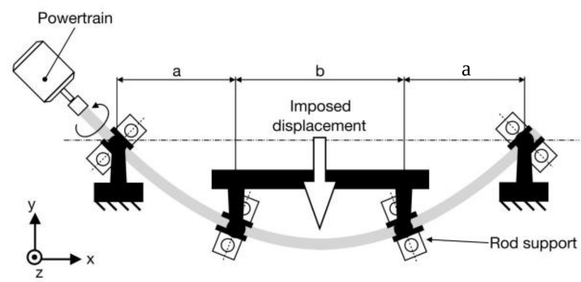

Figure 3. Scheme of the four point rotating bending test bench.

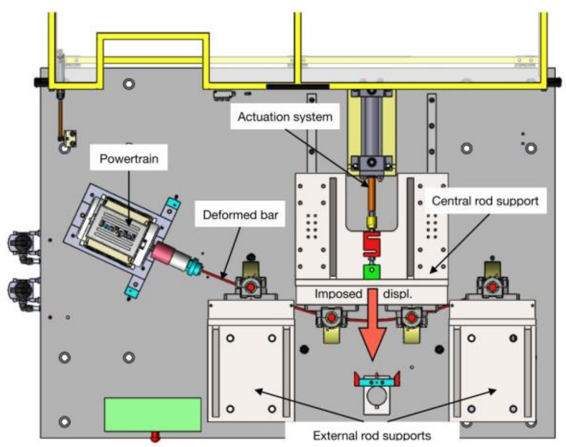

Figure 4. Drawing of the four point rotating bending test bench.

The supports of the wire (Figure 5) are free to rotate with respect to a vertical axis to follow the deformation of the wire. The two central supports are moved by a hydraulic actuator. The movement of the supports is guided by high precision linear rails. By this arrangement, the displacements of the two central supports is constrained to be equal, except for the mechanical tolerances.

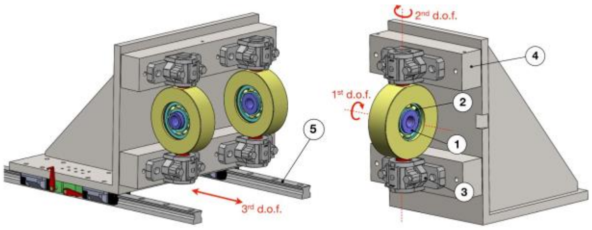

Figure 5. Drawing of the supports of the four point rotating bending test bench. 1: polymeric bushing. 2: rotating bearing. 3: fixed bearing. 4: movable bracket. 5: high precision linear rails.

The curvature of the wire in its deformed configuration is measured by an arcmeter (see Figure 6). The maximum stress in the wire can be related to the read quantity *h* as

$$\sigma = E \cdot \frac{8hd}{8h^2+2l^2-8hd} \qquad (2)$$

where *l* is the length of the arcmeter (see Figure 6) and *d* the diameter of the wire.

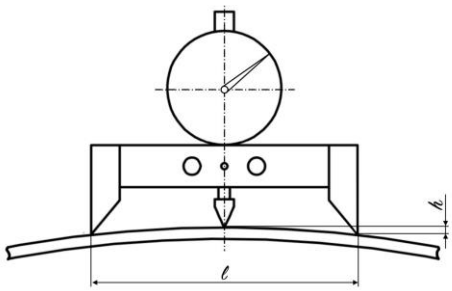

Figure 6. Arcmeter.

Finally, a safety system is built in the testing machine. A series of three proxies is positioned close to the inner border of the central zone of the wire. When the wire fails, the deformation is recovered and the wire moves apart from at least one proxy. As the signal of at least one proxy is zero, the motor is switched off and the rotation is arrested.

Particular attention has been devoted to the design of the polymeric material [14] of the bushings inside the supports. Actually, the increased stress of the wire inside the bushings must be properly minimized.

A finite element model of the wire and the support bushings has been realized in order to design the internal curved profile of the bushings. The model considers the wire in contact with the bushings. The central bushings are then displaced in order to apply the desired deformation to the wire. Wire diameters from 8 mm to 20 mm have been considered.

Frequency analysis

During the test, the wire is rotated in order to apply a sinusoidal fatigue load. To reduce the testing time, a high rotational velocity is required. To avoid vibrations of the wire, its first eigenfrequency should be higher than the rotational frequency. The eigenfrequency of a multi-span beams with pinned intermediated supports can be estimated as [15]:

$$f_i = \frac{\lambda_i^2}{2\pi L^2}\sqrt{\frac{EJ}{m}} \cdot d \qquad (3)$$

where λ_i is a numerical coefficient depending on the number of the considered resonance frequency and on the boundary constraints. For the first eigenfrequency of a four pinned intermediate support beam with a pinned end and a free end, $\lambda_i=1.539$. E is the elastic modulus, L the length of each beam span, J the moment of inertia of the section of the beam and m the beam mass per unit of length. For a circular wire with $J=\pi d^4/64$ and $m=\pi d^2/4$, Eq. 3 can be rewritten as

$$f_i = \frac{\lambda_i^2}{8\pi L^2}\sqrt{\frac{E}{\rho}} \cdot d \qquad (4)$$

with d diameter of the wire and ρ density.

According to Eq. 4, the first eigenfrequency of the wire is proportional to the diameter of the wire. The most critical situation is given by the wire with the smallest diameter to be tested with the test bench. In the considered test bench, the smallest wire diameter is 8 mm, which gives a first eigenfrequency of 62.8 Hz. This frequency is higher than the maximum frequency of the two poles motor used for the test bench, which has a maximum frequency of 50 Hz with European electric service.

To check the results of Eq. 4 and to take into account the nonlinear effects of the contacts of the bushing, a finite element model of the system has been considered. The FEM model is depicted in Figure 7 top. The model considers a wire with a diameter of 8 mm and the four support bushings. In the middle, the static deformation of the wire for a displacement of 62.6 mm of the central supports is shown. This displacement corresponds to a maximum stress of 1000 MPa on the wire. At the bottom, the first eigenfrequency deformation is reported. The first eigenfrequency has been computed for different displacements of the central supports, corresponding to different values of maximum stress. The results are reported in Table 1. The results show that the first eigenfrequency grows with the wire deformation and is close to the frequency computed by applying eq. 4.

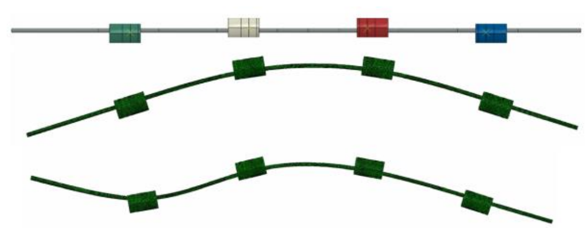

Figure 7. FEM model of the wire and supports. Top: model. Middle: static deformed shape of the wire, displacement 62.6 mm (maximum stress:1000 MPa). Bottom: first eigenfrequency deformation (68.2 Hz).

Table 1. Computed first eigenfrequency for different displacements of the central supports for a wire with diameter of 8 mm.

Imposed displacement [mm]	Maximum stress [MPa]	First eigenfrequency [Hz]
3	47	60.7
12.2	193	61.8
25.1	394	62.8
44.3	689	64.7
65.1	1000	68.2

Preliminary experimental tests

Preliminary fatigue tests have been performed in order to check performance. Two sets of wires for springs have been tested to verify the position and properties of the crack surface. Failure has to happen between the two central supports.

The test bench during a fatigue test is reported in Figure 8.

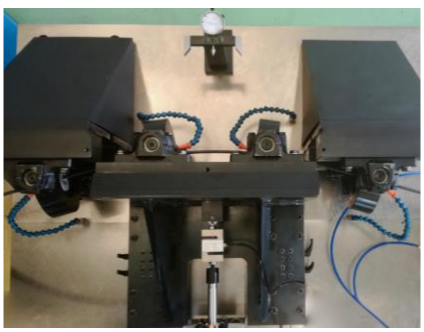

Figure 8. Test bench during a fatigue test.

Twenty-nine tempered wires for the realization of helical springs have been tested. The wires had two different diameters (8 and 11.5 mm) and were made from two different high resistance steel alloys (both with ultimate tensile strength greater than 1600 MPa). The applied stress ranged from 550 to 1000 MPa. Of the 29 wires, only one has shown a rupture located outside the central section, although very close to one of the supports of the central section.

In Figure 9, the illustrative fractographic image of the fracture surface of the specimen number 2 is shown. This specimen has a diameter of 8 mm and was subjected to a maximum stress of 1000 MPa. From the picture, it can be observed the relatively large final failure zone (indicated with “O” in the picture) with respect to the beach marks (“B”) zone, clear indication of a high applied stress. The presence of different ratchet marks (“R”) and multiple crack initiations indicates that the stress on the surface of the specimen was pretty uniform.

The location of the fracture zones and the fractographic image prove that the test bench presented in this paper is actually able to provide a four point rotating bending load situation, without relevant stress concentration at the supports.

Figure 10 shows an illustrative fatigue testing campaign that was completed in less than two days on precise ground steel bars.

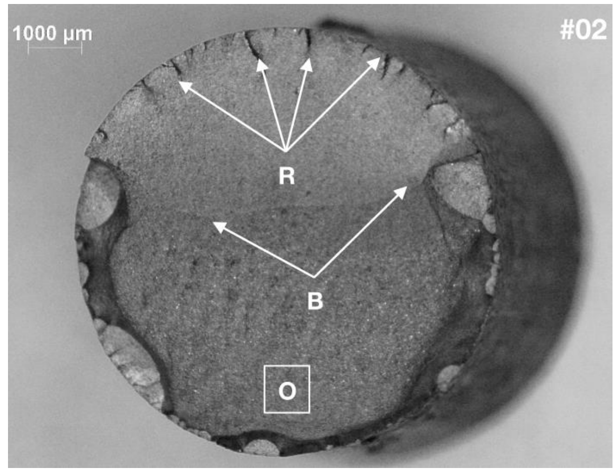

Figure 9. Specimen #02 (diameter 8 mm, applied load 1000 MPa), fractographic image. O = final failure zone; B = beach marks; R = ratchet marks.

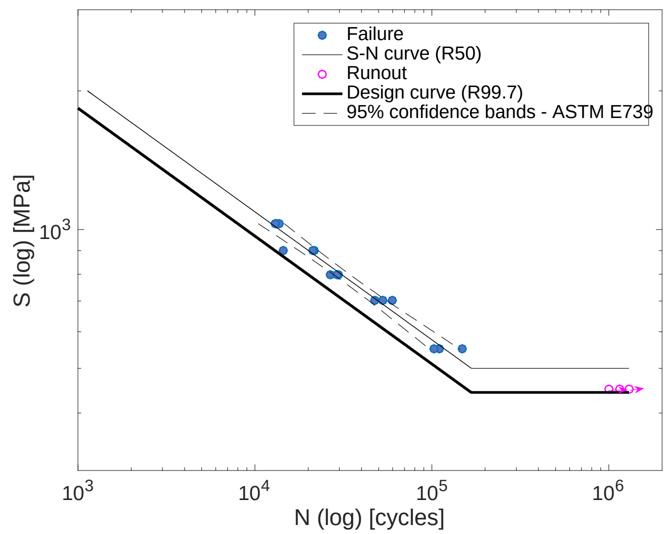

Figure 10. Results of the tests performed on precise ground bars.

Conclusions

This paper has been devoted to the design and realization of an innovative test bench for the four-point fatigue testing of tempered wires. The test bench has been designed in order to guarantee a constant bending moment in the central part of the wire and, most important, to avoid stress concentrations at the supports. Special polymeric bearings with a curved inner surface have been designed. The vibrational behavior of the test bench has been simulated to allow tests at frequencies up to 50 Hz.
Preliminary experimental tests have been conducted. Standard semi-finished bars (tempered wires) have been used. Tests were fully satisfactory, the breaking surface of the wire was found in the central part of the wire and not outside the supports. Out of 29 tested wires, in 28 cases the wires have broken in the central part, one in the support.

A complete fatigue testing campaign of the addressed components can be performed in a couple of days.

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