Prototype of a Torsion Fatigue Machine Based on Unbalanced Rotation.

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Abstract. The fatigue test machine was designed and built to torsional loading. The design is relatively simple, low cost manufacturing, allowing its construction in conventional machining, easy maintenance and low operating cost. The machine is powered by an electric motor and a variable frequency drive allows controlling of your speed. To avoid undesired fluctuations in rotation during the test, the movement is transmitted to the unbalanced rotating disc with the use of two synchronous pulleys and a timing belt. The torque generated in the specimen and/or component under test originates in an unbalanced rotating disc and the magnitude of this torque is controlled by varying the unbalanced mass or the motor rotation. Tests can be performed on standard specimens or components adaptable to the machine fixing system. The tests can be applied to metallic or nonmetallic materials and the torque amplitude can be constant or variable in time with controlled R (ratio between the maximum and minimum stress). The torque is measured with a reaction torque meter that is coupled at one end of the specimen and is monitored throughout the test. The tests indicate that the designed and built fatigue machine works satisfactorily for cyclic loading tests with constant and variable amplitude.

Key words: fatigue, torsion, test machine.

1. INTRODUCTION

Mechanical testing machines usually used in fatigue tests are mechanical or servo-hydraulic. Servo-hydraulic equipment is multifunctional, but high cost machines with complex operation and maintenance. As the fatigue tests are long and numerous, the cost of use servo-hydraulic equipment is extremely expensive, Franco *et al.* (2011); Kusaba *et al.* (2012). On the other hand, mechanical equipment is constructively simple, of easy maintenance and low cost; but limited in functionality.

The results of the fatigue tests are not homogeneous, so it is necessary to perform numerous tests to estimate the fatigue limit of the material and thus obtain reliable results. According Kusaba *et al.* (2012) in these cases it is very important to develop equipment from an appropriate methodology for the fatigue tests in an attempt to avoid the use of hydraulic machines that besides being expensive and have a high consumption of electric energy.

In order to avoid the use of a hydraulic machine a rotational unbalanced based torsional loading fatigue testing machine has been designed and constructed to perform tests of metallic and non-metallic specimens. This fatigue test machine is designed to adapt different geometries of the specimens; varying load conditions alternating or pulsed; it is possible to read and store the test data; have a low operating cost and to vary the loading frequency and amplitude.

The main feature of the unbalanced rotational-based Torsion Fatigue Machine is the unbalanced rotation system used to produce the alternating torque on the sample. Due to this characteristic, it is possible to perform tests with constant or variable torque amplitude, controlling only the angular velocity of the electric motor. Two helical extension springs allow to control the value of R (ratio between the minimum torque and the maximum torque) and to perform tests with varying average torque in both directions of torque. The mathematical formulation and manufacturing details of this equipment are shown in the sequence.

2. PROTOTYPE

Figure (1) shows the mechanical prototype of the torsional fatigue test machine. This prototype is an evolution of the machine proposed by Joaquim et al. (2009) and its main components illustrated in Figure (2) are: electric motor (1), timing belt pulley (2), timing belt (3), rotating disk (4), helical extension spring (5), test specimen (6), torque meter (7), sliding base (8) and linear guide (9).
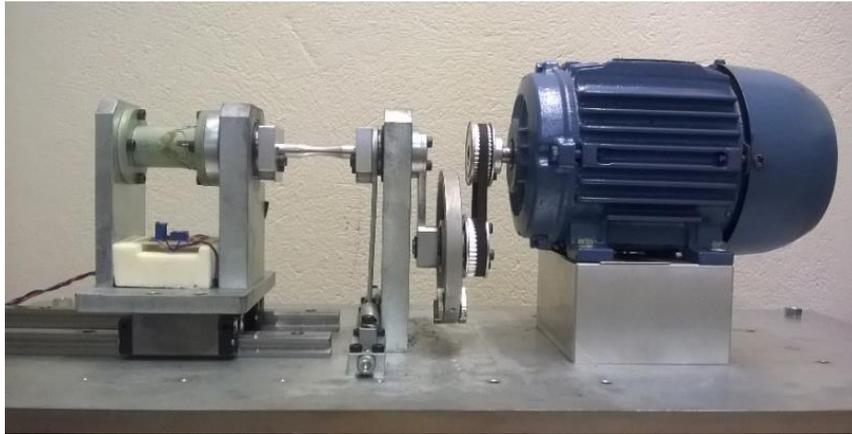
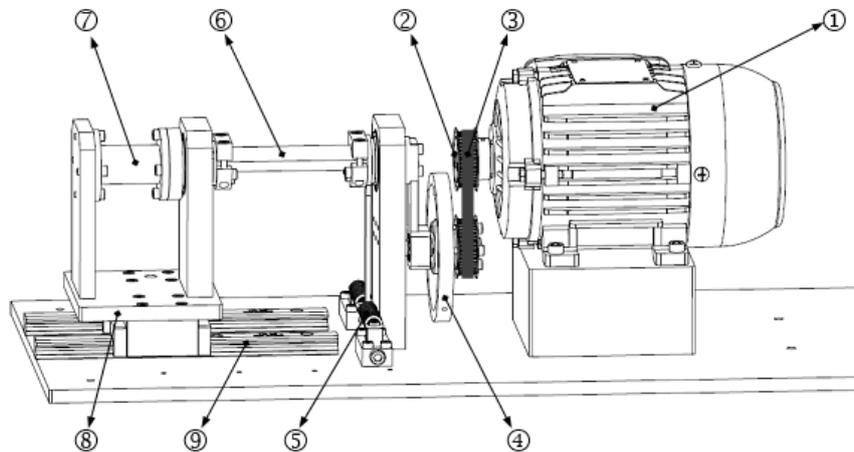**Figure 1 - Profile of the torsional fatigue test machine.**

Figure 2 - Main components of the torsional fatigue test machine.

2.1. Control and Power System

The machine is driven by a 1/4 hp electric motor and a variable frequency drive, not shown in Figs. (1) and (2), is used to control and vary the angular velocity of the electric motor.

To avoid fluctuation in disc rotation during the test with constant torque amplitude the movement is transmitted to the unbalanced rotating disk using synchronized pulleys and a timing belt with transmission ratio 1:1.

2.2. Torque measurement

The reaction torque meter shown in Fig. (3) was constructed using the full Wheatstone bridge of 120Ω strain gages. The torque meter does not respond to axial or bending loading and the output voltage is linear with respect to torque (Murray, Miller, 1992; Lee, et al., 2005; Khan, Wang, 2001). Torque can be measured and stored throughout the test.

Figure 3 – Reaction Torque meter.

2.3. Mechanism of torque transmission to the specimen

Figure (4) illustrates the model with one degree of freedom (rotation axis of the specimen) used by Joaquim *et al.* (2009) to explain that the loading torque acting on the specimen. When the motor is turned on, the rotating disk rotates at the same angular velocity of the motor, ω, and the unbalanced mass fixed at the rotating disc produces a centrifugal force of amplitude $F = me\omega^2$.
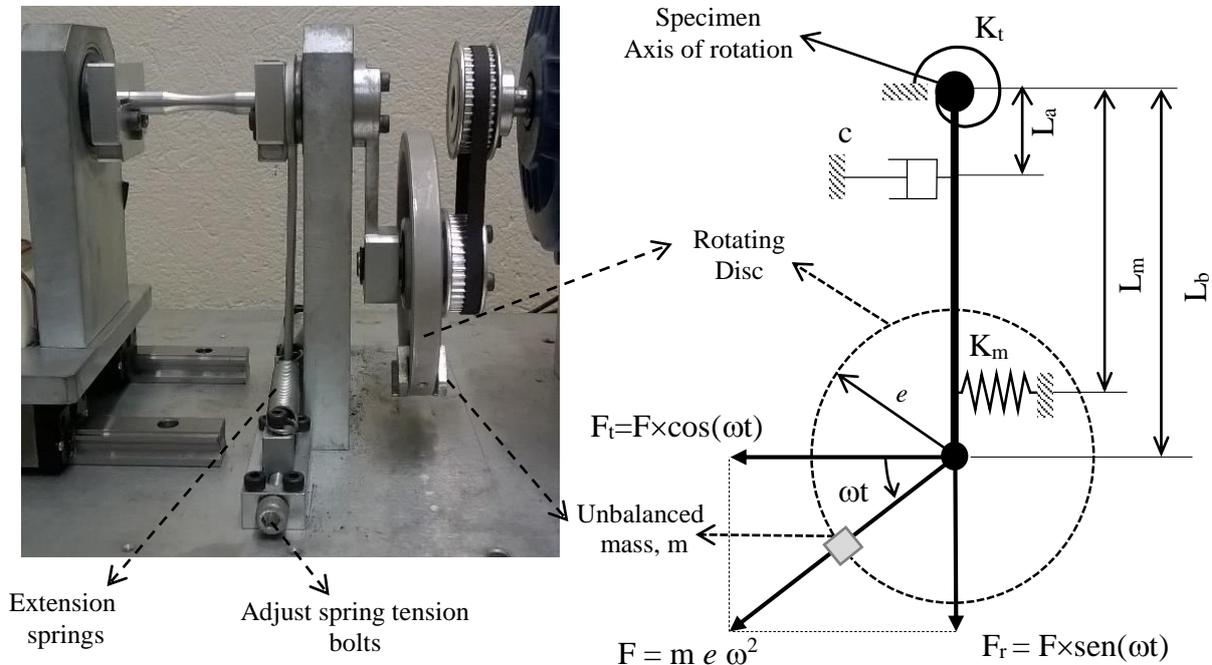

Figure 4 - Equivalent system with one degree of freedom (Adapted from Joaquim et al., 2009).

The tangential component of unbalanced force, F_t, produces the torque on the specimen axis given by $F_t \times L_b$, where L_b is the distance from specimen axis to the center of rotating disc. Thus, the amplitude of torque on specimen axis generated by the unbalanced force is proportional to $me\omega^2 \times L_b$ and varies in time as $\cos(\omega t)$, that is, corresponds to the alternating torque.

The midrange torque is generated on specimen axis using the force adjustment on the helical extension springs. Thus, the value of $R=\tau_{min}/\tau_{max}$ is controlled by the force on springs and de amplitude of the unbalanced force.

Figure (5) shows examples of fatigue tests that can be performed with the proposed prototype. Figures (5a) and (5b) represents tests with constant amplitude with R different and in Figure (5c) the amplitude of load varies (ω variable) with R = -1.

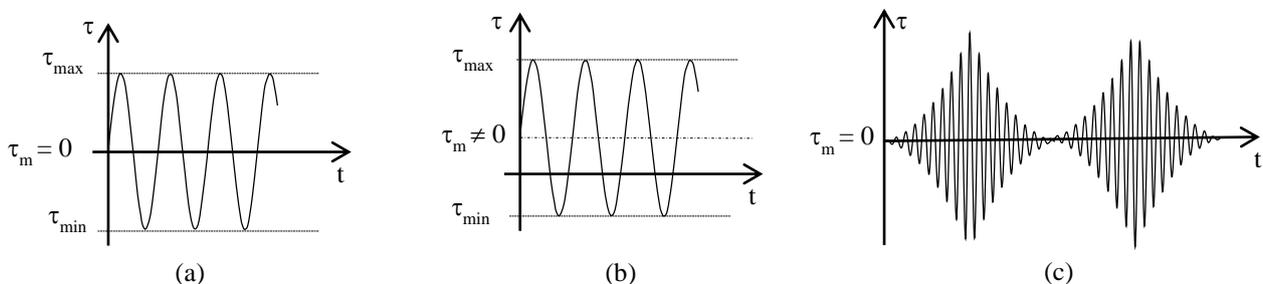The radial component of the unbalanced force does not produce torque on specimen axis.

Figure 5 - Loads in time, (a) constant amplitude with R = -1, (b) constant amplitude with R ≠ -1 and (c) variable amplitude with R = -1.

2.4. The Mathematical Model

Figure (6) illustrates the equivalent system used to obtain the mathematical model of the fatigue machine which allows only rotation around the axial axis of the specimen.

The mathematical model using only one degree-of-freedom was successfully used by Joaquim et al. (2009) to obtain the dynamic parameters of similar fatigue machine and for this reason, this model was also used in this work. The degree of freedom used in the model is the rotation around the axis of the specimen, $\theta(t)$.

The differential equation for small oscillations of $\theta(t)$ is given by (Rao, 2009; Inman, 2013):

$$\underbrace{I\ddot{\theta}(t)}_{\text{inertia torque}} + \underbrace{cL_a^2\dot{\theta}(t)}_{\text{viscous damping torque}} + \underbrace{(K_t + L_m^2 K_m)\theta(t)}_{\text{elastic torque}} = \underbrace{- m e \omega^2 L_b \cos(\omega t)}_{\text{Unbalanced torque}} \tag{1}$$

where I is the moment of inertia around the axis of rotation, c is the viscous damping constant, K_t is the equivalent torsional spring constant os specimen, K_m is the equivalent constant of the helical springs, L_m is the length of the springs arm, L_a is the length of damping arm and the product $me\omega^2 L_b$ is the amplitude value of the excitation torque.
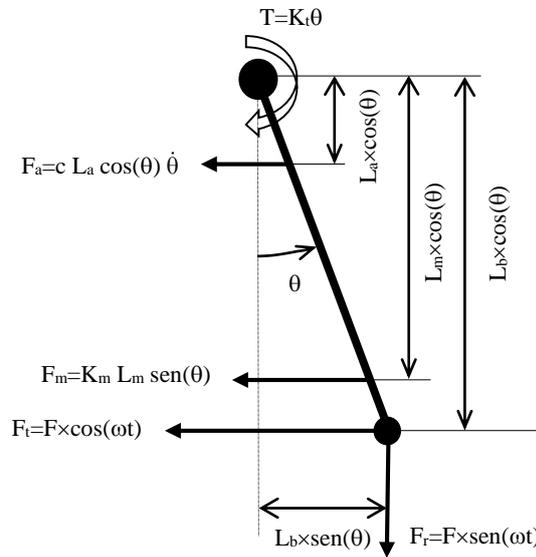

Figure 6 - Equivalent forces for displaced system (Adapted from Joaquim *et al.*, 2009).

The natural frequency of system, ω_n, given by (Rao, 2009; Inman, 2013):

$$\omega_n = \sqrt{\frac{K_{eq}}{I}} = \sqrt{\frac{(K_t + L_m^2 K_m)}{I}} \tag{2}$$

and from the particular solution, we find the amplitude value of the angular motion and the phase angle, given by:

$$\theta_0 = \frac{\theta_e}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}} \tag{3}$$

and

$$tg(\alpha) = \frac{C_{eq}\omega}{K_{eq} - I\omega^2} = \frac{2\zeta r}{1-r^2} \tag{4}$$

where r is the frequency ratio given by $r = \omega/\omega_n$, ζ is the damping factor, $\zeta = c_{eq}/2I\omega_n$, and θ_e denote the amplitude value of the angular displacement when $\omega = 0$ due to the static displacement .

Thus, the torque acting on the specimen is composed by the initial torque (midrange torque) that is controlled by the forces on extension springs and by the unbalanced mass rotation. The torque amplitude produced by the unbalanced mass, T_{cpd}, can be estimated using the frequency response of the system (Rao, 2009; Inman, 2013) and obtained with the following expression:

$$\frac{T_{cpd}}{m\,e\,L_b} = \omega^2 \sqrt{\frac{1 + (2\zeta\,\omega/\omega_n)^2}{[1 - (\omega/\omega_n)^2]^2 + (2\zeta\,\omega/\omega_n)^2}} \tag{5}$$

The values for ζ and ω_n vary with the type of material, geometry of the specimen and the springs stiffness. Their values can be determined experimentally (showed in results).

3. DINAMIC MODEL PARAMETERS

The impact test was the experimental method used to identify the natural frequency and damping of the system. The hammer impact load is applied as indicated in Fig. (7) and the axis specimen torque is used to evaluate the parameters of the system.

Figure 7 – Hammer impact point.

Figures 8 and 9 show the results for the system with and without the extension springs. The peaks and valleys with their respective values of the torque, T, given in Nm are highlighted as a function of time, t, in seconds.

According to Rao (2009), the logarithmic decrement, δ, can be determined experimentally, since known two or more displacements (peaks or valleys) successive. The logarithmic decrement is expressed by $\delta = \frac{1}{n}\ln\left(\frac{T_i}{T_{i+n}}\right)$, where n is the number of periods between the peaks or valleys considered for analysis. The damping factor, ζ, may be a function of δ and is given by $\zeta = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}}$. The damped frequency, ω_d, is expressed by $\omega_d = \frac{2\pi n}{t_{i+n} - t_i}$, where t_i and t_{i+n} represent the times (in seconds), corresponding to their considered torques. Thus, the natural frequency, ω_n, as a function of ω_d and ζ, is expressed by $\omega_n = \frac{\omega_d}{\sqrt{1-\zeta^2}}$ in rad/s.
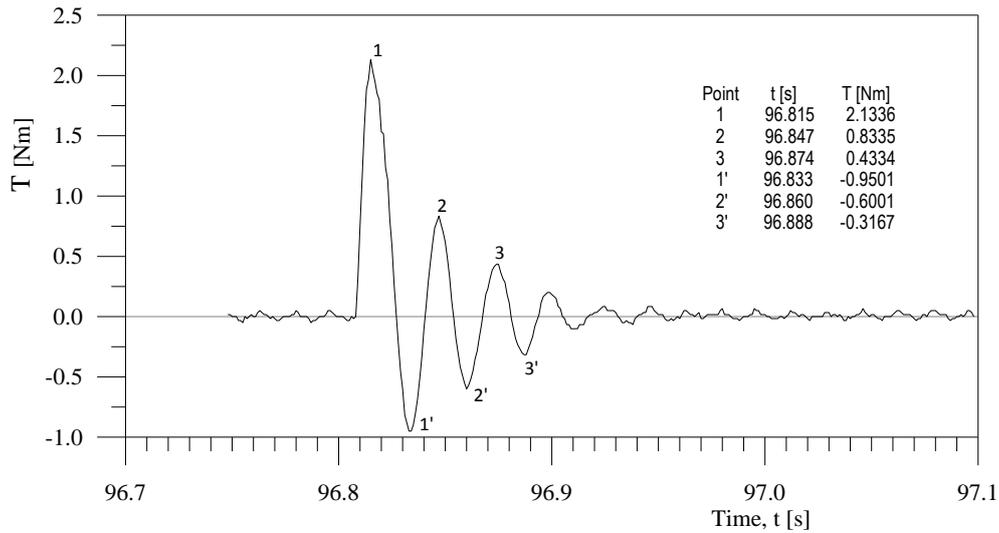
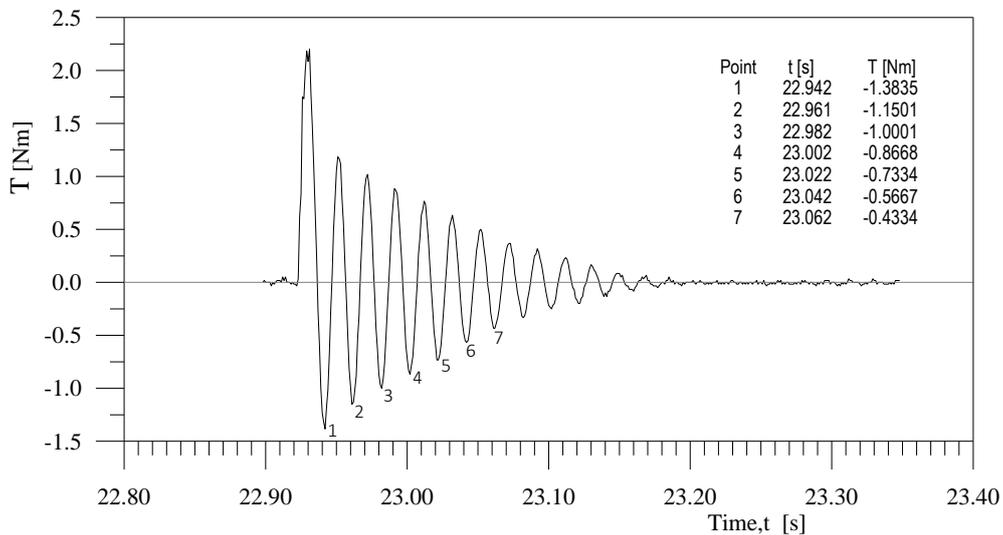**Figure 8 - Underdamped system behavior without extension springs.**

Figure 9 - Underdamped system behavior with extension springs.

Table (1) shows the estimated values for the damping factors obtained from the impact test. The data correspond to the underdamped behavior of the system shown in Fig. (8-9).

Table 1 - System damping factors.

Coupled Extension Springs	Damped frequency ω_d [Hz]	Logarithmic decrement (δ)	Damping factor (ζ)	Natural frequency, ω_n [Hz]
Yes	50,000	0,194	0,031	50,024
No	36,363	0,549	0,087	36,502

Therefore, it was possible to estimate the value of the natural frequency of the system. The operating band of the fatigue machine must be different from the natural frequency of the system, $\omega_n \cong 36$ Hz for torsional fatigue test without the use of the traction springs and $\omega_n = 50$ Hz with the use of traction springs, as seen in Tab. (1). Knowing that if the frequency of operation of the fatigue machine is equal to the natural frequency of the system, the phenomenon of resonance occurs, causing undesired vibrations in machine and premature failure of specimen.

4. EXAMPLE TEST

In Figures (10) to (12) are shown the results of a fatigue test with constant amplitude for specimen of A356 aluminum alloy. The test was performed with loading frequency equal 20,5Hz (below the natural frequency of the machine) and R ≅ -1.

Figure (10) shows the details of the start of the fatigue test. After approximately 8 seconds of starting, the electric motor the torque reaches the value specified for the test.

The result of the signal is a sinusoidal curve, with approximately 20,5 cycles per second, as shown in Fig. (11). Figure (11) illustrates the intermediate signals of the fatigue test and we note that the average torque is zero and the amplitude remains constant over time.

As the test proceeds at the end of the specimen life, the alternating torque increases, as observed in Fig. (12), which shows details of the final region of the torsional fatigue test. The increase of the alternating torque can indicate the presence of a fatigue crack in the specimen, probably indicating the end of Stage II fatigue lifer.

According to Joaquim et al. (2009), the alternating torque increases just before failure due to loss of test specimen rigidity due to the presence and growth of fatigue crack that causes changes in the frequency response curve (FRF). The point B represents the moment of rupture of the material and in sequence, the alternating torque decreases rapidly. This fatigue test resulted in 29,190 cycles with alternating torque of 2,0645 Nm and midrange torque of - 0,0095 Nm, approximately.
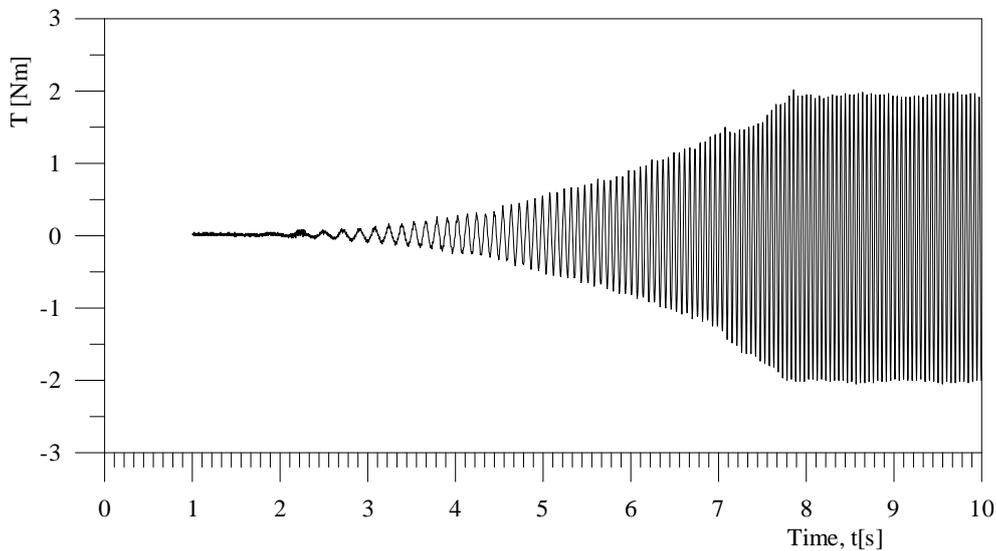

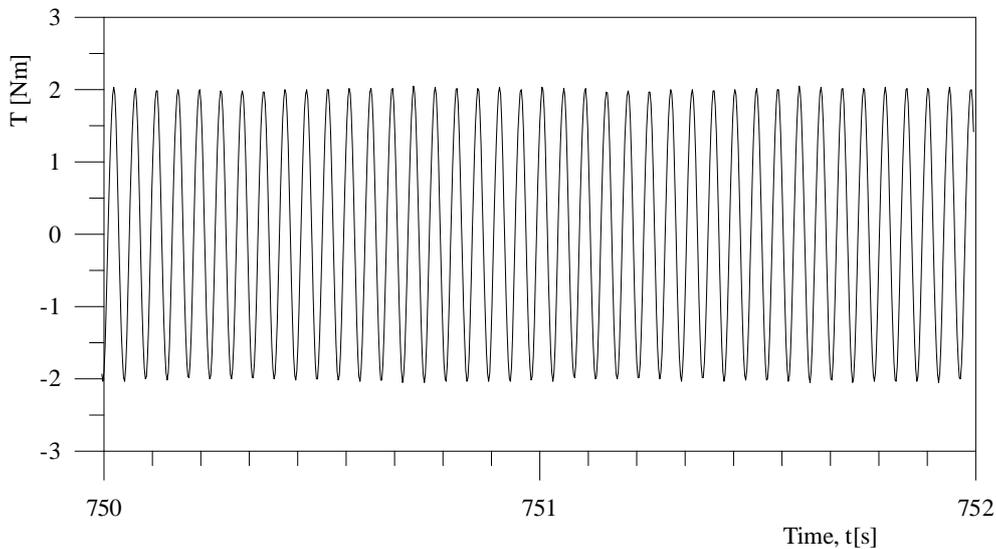**Figure 10 - Detail of the beginning of the torsional fatigue test.**

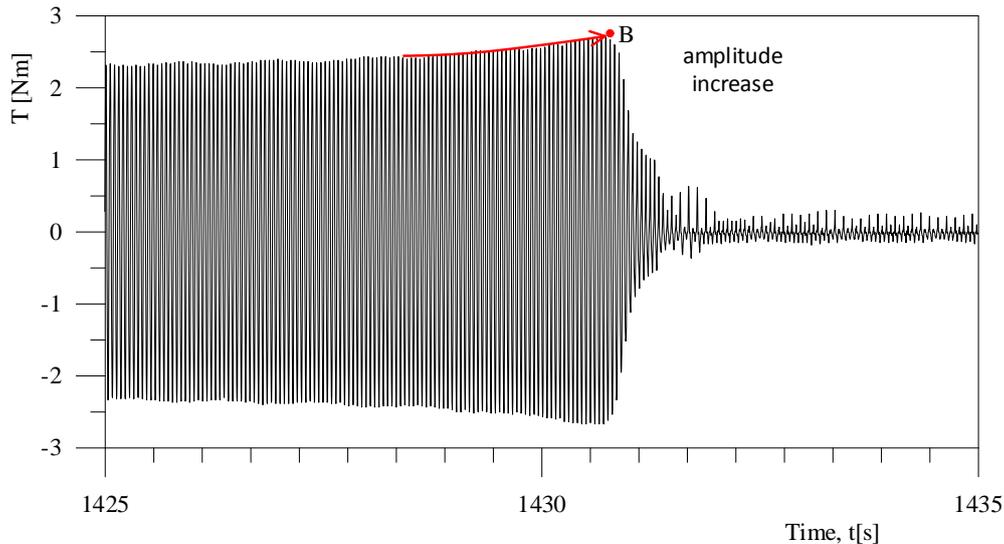**Figure 11 - Details of the intermediate region.**

Figure 12 - Details of the final region of the torsional fatigue test.

5. CONCLUSION

The Torsional Load Fatigue Testing Machine described in this article is appropriate to perform tests with variable amplitude and variable R. The main characteristic of the equipment is the alternating torque generated by an unbalanced rotating mass. The maintenance of the machine is simple and low cost, has low energy consumption, low noise operation, easy control and application of load. Due to the system of sliding guides, the specimens can be made with different sizes and geometries.

The Rotational Imbalance-based Torsion Fatigue Machine works satisfactorily, and responds to its objectives, so that it generated a National Invention Request to the National Institute of Industrial Property (INPI), on June 21, 2016, process number: BR 10 2016 014593 7.

6. ACKNOWLEDGEMENTS

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