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[1.] Holger HOTA, [2.] Şerban BOBANCU, [3.] Péter DANI, [4.] Botond-Pál GÁLFI, [5.] Ildikó Renata MUNTEANU, [6.] Ioan SZÁVA

NEW TESTING BENCH FOR THE ROLLING FRICTION COEFFICIENT'S EVALUATION

[1.] Transilvania University of Brasov, 18 Lorena Street, 305400 Jimbolia, ROMANIA
[2.] Transilvania University of Brasov, Department of Mechanisms and Machine Elements, 29 Eroilor Avenue, 500030 Brasov, ROMANIA
[3-6.] Transilvania University of Brasov, Department of Mechanical Engineering, 29 Eroilor Avenue, 500030 Brasov, ROMANIA

Abstract: The authors conceived and realized one original testing bench, destined to evaluate the friction coefficient of the rolling resistance for cylindrical bodies on plane surfaces. Two, self-conceived and high-sensitivity, electric strain gauges assure the monitoring of the applied pressure load to the subassembly and of the horizontal force, destined to overcome the rolling friction resistance for the tested cylindrical rolling elements. The original electronic device allows a half-automatic testing procedure and consequently increasing its efficiency. This original testing bench allows a high-accuracy evaluation of the phenomenon regarding on the starting moment of the rolling elements (after overcoming the rolling resistance of the conjugated surface) and during the normal rolling process, too. There are described the working principle of the testing bench, the calibration of its main components, respectively some useful results of tests. In the next period the authors will perform, on different pairs of materials, some high-accuracy investigations. Using this testing bench, the authors performed several experimental investigations on steel-steel elements. These results are destined in the next period to establish, with a better accuracy, this coefficient for the rolling bearings' elements.
Keywords: rolling friction coefficient, testing bench, electronic driving device, electric strain-gage transducers

1. INTRODUCTION

It is well-known fact, that the friction is a complex physical-chemical phenomenon. In the case of the dry friction, the surface roughness and the components' deformability are decisive. In the present case, the authors studied the case of four cylindrical rolling elements disposed on plane surfaces.

Based on the literature [1-11], the following main aspects can be mentioned:

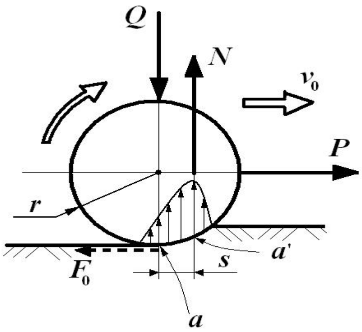

Figure1. The rolling resistance phenomenon

- The rolling resistance depends on:
  - the elastic behaviours of the materials;
  - the conjugate bodies behaviours;
  - the curvature of the contact surface, and
  - the magnitude of the applied load;
- To overcome the rolling resistance, it is spend a mechanical work, destined to deform the contact surfaces;
- That way will be obtained a non-uniform stress distribution in the contact zone, with its resultant N, disposed with the small distance *s* [*mm*] in front of the cylinder vertical axis (Figure 1);
- This distance *s* represents the arm of the rolling friction force, called *the friction coefficient of the rolling resistance*; that way, the so called *rolling friction's momentum*

$$M_f = N \cdot s = Q \cdot s, \tag{1}$$

is produced, which puts up resistance to the applied force's momentum

$$P \cdot r; \tag{2}$$

- The condition of an uniform rolling of the cylinder without sliding is

$$P \cdot r = Q \cdot s, \tag{3}$$

from where

$$P = s \cdot \frac{Q}{r}; \tag{4}$$

≡ Starting from the condition $P = F_0$, where F_0 represents the sliding friction force ($F_0 \leq Q \cdot \mu_0$, where μ_0 represents *the adhesive friction coefficient*).

On can obtain the following cases of:

≡ the pure rolling (without sliding) $\mu_0 \rangle \frac{s}{r}$;

≡ the pure sliding (without rolling) $\mu_0 \leq \frac{s}{r}$, respectively

≡ the rolling with sliding $\mu_0 = \frac{s}{r}$.

2. THE DESCRIPTION OF THE TESTING BENCH

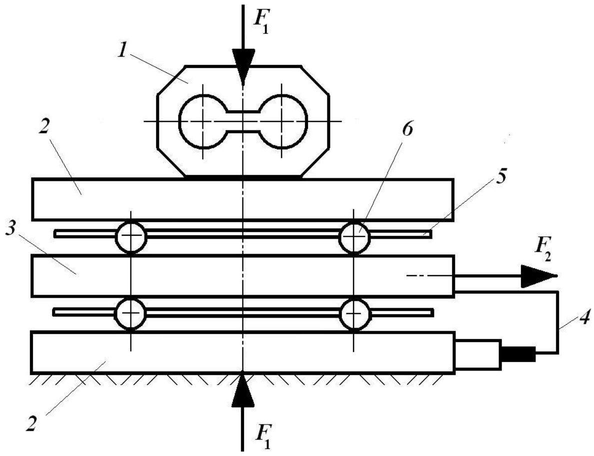

Figure 2. The scheme of the original stand

The authors conceived a simply, but useful testing bench, shown in Figure 2. Its main components are presented below. Using two special cages *5*, four cylindrical rolling elements 6 are disposed between two very rigid steel plates (*2* and *3*). These plates and cylinders are manufactured from OLC 45, having adequate number of pairs with uniformly hardness, obtained by mean of heat treatments. The intermediate rigid plate *3* assures a symmetrical loading and symmetrical behaviour of the system. The special cages *5* serve to keep the rolling elements at the same distances during the experiments.

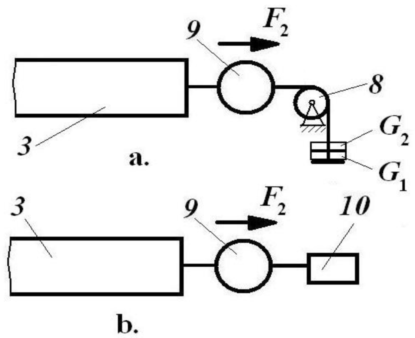

Figure 3. The proposed variants of the loading system: *a.* using some calibrated weights; *b.* using the below-described electronic device

This subassembly is disposed between the jaws of one universal (tensile/compression) testing machine and the applied load $F_1 = F_V$ is evaluated with high accuracy by means of the octagonal electric strain gage transducer *1*. By means of a special, self-conceived inductive transducer *4*, fixed between the plates *2* and *3*, is monitored both the starting moment of the subassembly *3-5-6* and its total displacement in one respectively in the opposite horizontal direction (based on the action of the horizontal force F_2. During the tests, the assembly (shown in Figure 2) is loaded with force F_1 by means of the universal testing machine. In the next step, by means of a special loading system (see Figure 3,a or Figure 3,b) the force $F_2 = F_h$ is applied to overcome the rolling friction resistance for the cylinders *6*.

One has to mention the fact that the first technical solution of this loading system (shown in Figure 3,a) uses some calibrated weights $G_1, \ldots, G_n$, which transfer their load by means of a nylon cord through the special, high-accuracy, ring-shaped, electric strain gage transducer *9* and the pulley *8* to the plate *3*. The electric strain gage *9* is necessary to eliminate the friction's losing effect on the pulley in the force's monitoring, respectively to assure the instantaneously data acquisition.

This solution presents between others, two main disadvantages, namely:

≡ only some well-defined, discrete loads can be applied;

≡ allows to analyze the phenomenon only in a single direction (e.g. in that indicated by force $F_2 = F_h$, I means: in right direction).

In order to eliminate these disadvantages, respectively to improve the system's efficiency, the authors conceived the second technical solution (Figure 3,b), where both of these disadvantages are eliminated successfully.

In this case the shuttle movement (backward and forward movement)of the intermediate plate *3* is assured with the desired rhythm (time-step) and with the same loading ratio by means of a small electric motor.

Also, one has to mention the fact that there are involved two electric strain-gauges for the F_2 horizontal forces' monitoring, disposed in left and right side of the subassembly *3-5-6*.

In Figure 4 is presented the electric motor controlling unit. In order to fulfil the electromechanical requirements (to ensure the cyclical reversing functions and counting the number of cycles), a simple control circuit was designed and built.. The requirements are not special (speed, accuracy, etc.). Consequently, a simple relay-based automation circuit was designed and built-up. This solution meant reduced design and financial effort.

In principle, the controlling unit consists of in a simple power supply circuit, having a 230/12V-100VA mains transformer, a fuse, and a 50A/400V bridge rectifier. The bridge rectifier is oversized, to resist at repeated current shocks and voltage spikes, generated by the DC electric motor (automotive windshield wiper type). The cyclical reversing automation is made with relay *Rel* (which has three pairs of normally open - normally closed contacts) and micro-switch contacts. Displacement limits are signalled by micro-switches *Mi1* and *Mi2* to cause change direction of displacement. Relay *Rel* with contacts *K1* and micro-switches *Mi1* and *Mi2* form a typical ON/OFF circuit with latch function, due to *K1:P1-S1*normal open contacts. *K2* contacts, operated also by *Rel*, realize the voltage polarity switching function at motor supply connectors *Mot1* and *Mot2*. The counting the number of cycles is made using an electromagnetic counter *L1* which senses the ON/OFF transitions of relay *Rel*.

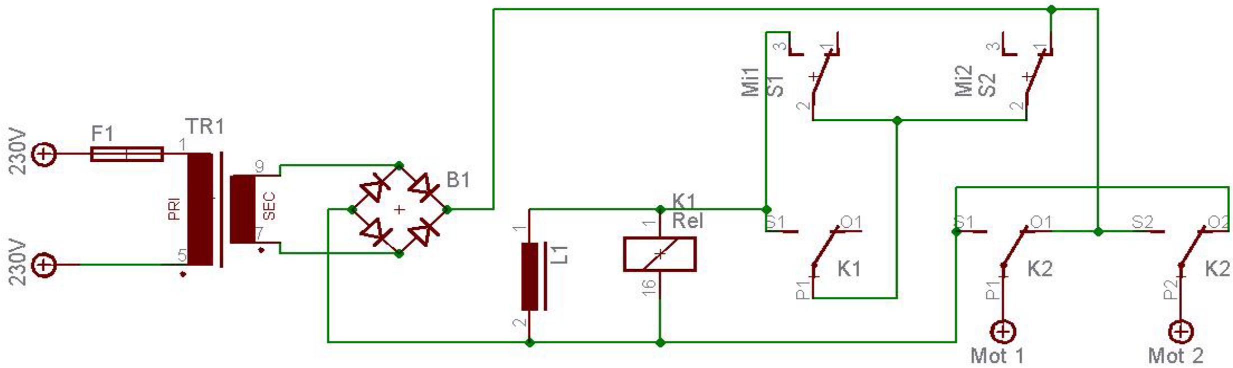

Figure 4. The schema of the controlling unit

By means of a continuously monitoring of the subassembly's ***3-5-6*** movement, beginning from the starting moment (instant) and during the mentioned shuttle movement, for a predefined number of cycles, became possible to be achieved the changing of the applied vertical load F_1 and of the horizontal ones F_2, too. These values will offer finally, based on schema from Figure 5, and using some simply Static-calculus from Mechanics, the rolling friction coefficients (corresponding to the starting moment and to the constant speed rolling process), too.

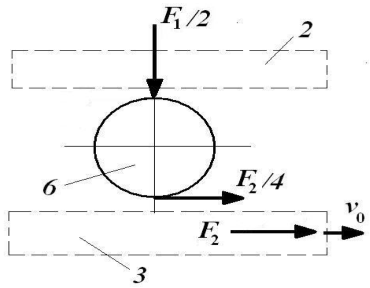

Figure 5. The rolling element solicitation

3. CALIBRATION AND PRELIMINARY TESTING RESULTS

In Figure 6 is offered the calibration curve for the self-conceived octagonal force transducer. One has to mention the fact that it presents a very good linearity of the signal.

Similarly, in Figure 7 is presented the calibration curve of the original, self-conceived inductive transducer, which also presents an acceptable linearity of the signal.

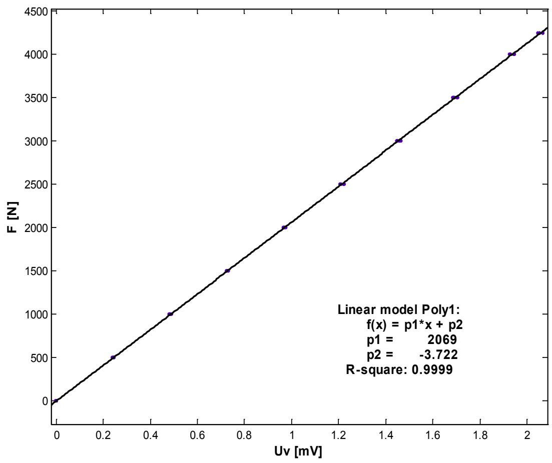

Figure 6. The octagonal transducer's calibration curve

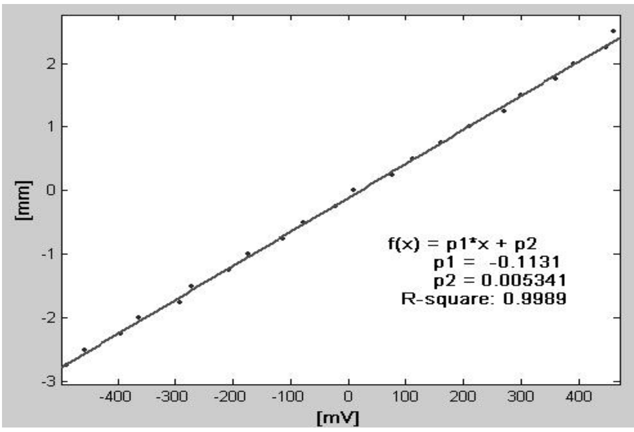

Figure 7. The inductive transducer's calibration curve

In order to illustrate the testing bench efficiency, in Figure 8 is presented the overall statistical evaluation of measurements performed on a set of 15 identical pairs of rolling elements and plates, having all of them the same 37 HRC hardness.

Similarly, in the case of a set of other 15 identical pairs of rolling elements and plates, having the same 50 HRC hardness, the results of the statistical evaluation is offered in Figure 9.In this last case, one can put in evidence an asymptotic variation law, corresponding to the vertical force magnitude greater than $F_1 = 5500\,N$. This kind of information can be very useful in designing of different rolling friction elements, taking into the consideration both the vertical load and the horizontal one, which is necessary to start up the rolling elements.In this sense, one can mention the case of the several axial bearings, which have only periodical motion (they are started only after several minutes or hours and they after than have some relative long standstills.

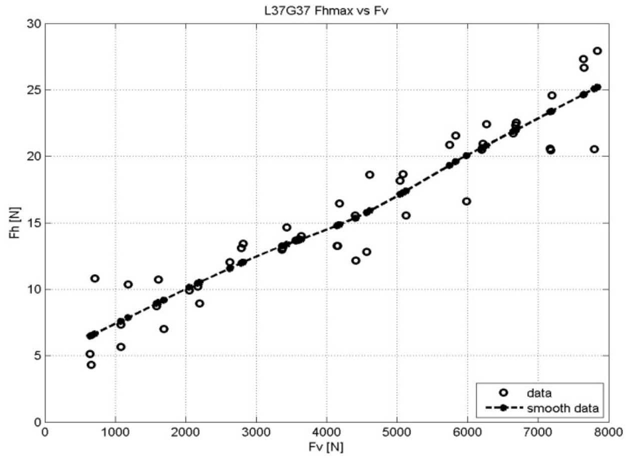

Figure 8. The variation of the maximal horizontal force in the case of the 37 HRC-plates with 37 HRC cylinders

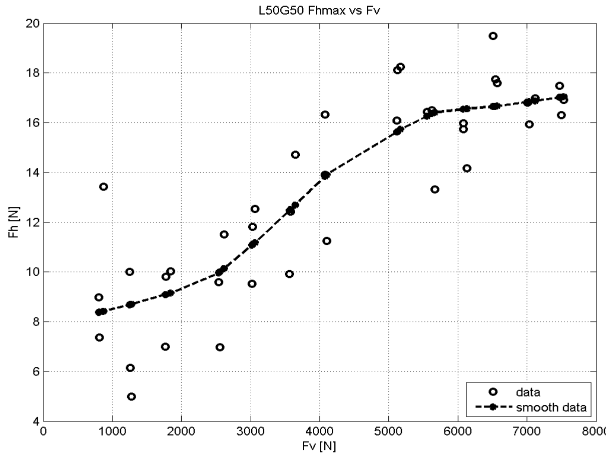

Figure 9. The variation of the maximal horizontal force in the case of the 50 HRC-plates with 50 HRC cylinders

4. CONCLUSION

The authors conceived and realised one original testing device for establishing the rolling friction coefficient both in the starting moment and for continuously rolling of some cylinder-shape elements. By means of this data acquisition becomes possible to establish for each (increasing) preloading value F_1 both the starting moment's rolling resistance force and that, which corresponds to the continuously rolling of the cylinders.

Based on these experimental results, becomes possible to estimate better both the predictable load-bearing capacity and the lifetime of these rolling elements.

From the literature on can be expected higher values for the starting moment's rolling resistance force versus some lower ones, corresponding to the constant rolling speed of the cylinders, which are demonstrated by means of these small preliminary experimental results.

In the next period, the authors intend to investigate different kind of materials for the components *2-6-3* (steel, copper-covered steel, respectively polymer-covered steel), which will represent some preliminary results for different engineering application (rolling bearings, Biomechanics and others), where the rolling friction coefficient's reduction presents high interest for increasing the lifetime of these products.

Also, there are presented two of the calibrations curves of the self-conceived force transducer, respectively of the inductive transducer. Their linearity is acceptable for the foreseen measurements.

By comparing the results obtained for two different harnesses, one can conclude that the maximal values for $F_2 = F_h$ was obtained for the lower hardness of the pairs, and minimal values belongs to the higher hardness of the elements.

ACKNOWLEDGEMENT

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