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FRICTION ASSESMENT OF LUBRICATED STEEL SURFACES I.

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Abstract: The coefficient of kinetic friction is a random variable. Its value is governed by a huge number of circumstances. Studying some widely known methods, one of them was improved and made applicable to analyse the rolling resistance of antifriction bearings. The rolling resistance depends mainly on the load, the type of supplied lubricant, the number of speed and the service temperature. Although the suggested test rig is currently under development, it seems to be able to select the proper lubricant for the given installation environment and operation conditions.

Keywords: *sliding friction, rolling resistance*

1. INTRODUCTION

There are many ways to interpret the coefficient of fiction. The Coulomb’s law of friction might be said to be true, of course. According to this law the coefficient of friction depends on the quality of surfaces are in contact with and moves relatively to each other. This well-known attitude has been refined by many researchers and with many considerations during the last centuries. Only a few questions of them would be mentioned in our paper, noting that the tests were restricted to steel surfaces.

1. Is there any medium (fluid, solid ‹solid coating› or both) between the surfaces?
2. In what physical-chemical conditions are the surfaces?
3. What is the roughness of the surfaces and the value of the normal force acting between the surfaces? What friction state is made by the fluid, the coating and the relative sliding speed together?
4. What is the ambient pressure and temperature?
5. What additional processes (wear, temperature increase) occur and to what extent?

Taking these simple issues into account may be possible through the collaboration of several disciplines. Numerous professional books, e.g. [1], scientific theses, e.g. [2] were completed on the subject, using the opportunities for measuring and modeling at the technical standard of their era. The objectives of our paper are to demonstrate some general and special experimental possibilities to clarify the friction conditions between machine parts and to contribute modestly to the extension of measurement experience.

2. Determination of the friction coefficient

The measurement of the friction coefficient is based on any physical principle. The method may be general or may be specialized to a kinematic pair. *Figures 1* and *2* show well-known general principles for the determination of static and dynamic friction coefficient, respectively. These principles are based on angular and force measurement, respectively. *Figure 3* shows a principle for clarifying the dynamic friction between the kinematic pairs of a power screw [3]. It is used to determine the torque need for raising or lowering the load stress load, and according to the well-known calculating model, the dynamic friction coefficient of the screw-nut relationship. The force vector $\vec{F}$ at the *Figure 3* includes the weight of the screw, rope drum and load. The torque generated due to the incrementally added m masses on the rope drum with diameter D, overcomes the friction torque of the screw and starts its rotation. In the case of a right-hand screw, the outlined figure corresponds to the case of raising the load. The torque equilibrium expressed by (1) is

$$Dmg = F\frac{d_2}{2}tg(\rho' + \alpha), \tag{1}$$

of which the static coefficient of friction, μ_0 is countable, knowing the mean diameter, d_2, the lead angle, α and the thread profile angle, β, i.e.

$$\mu_0 = \cos\left(\frac{\beta}{2}\right) tg\left[arctg\left(\frac{D}{d_2}\cdot\frac{2mg}{F}\right) - \alpha\right]. \tag{2}$$

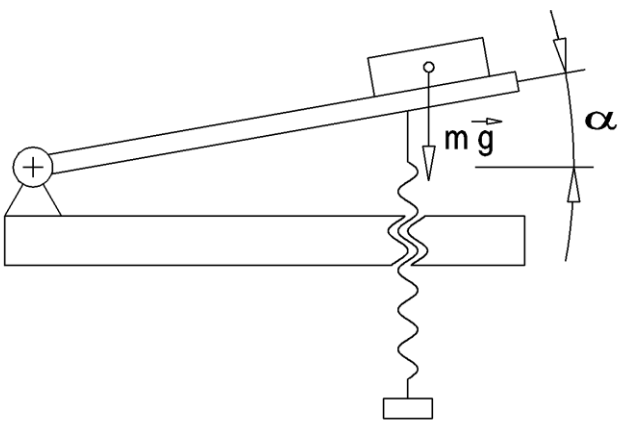

Figure 1. Measurement of static friction coefficient

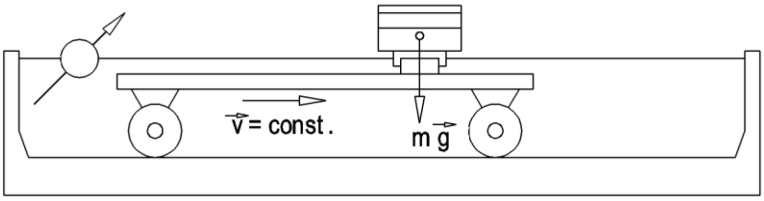

Figure 2. Measurement of dynamic friction coefficient

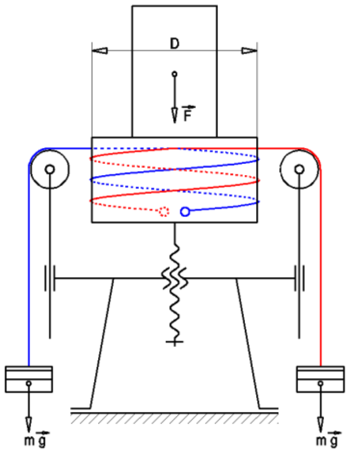

Figure 3. Measurement of friction torque between screw and nut [3]

There are other methods for measuring the coefficient of friction such as the principle of harmonic motion [3]. Let's rush through the known process. *Figure 4* shows two pulleys, rotating oppositely to each other, at a given distance, *c* apart from each other. The horizontal straight rod lying on the pulleys has a reciprocating motion. The force vectors designate that of acting to the rod.

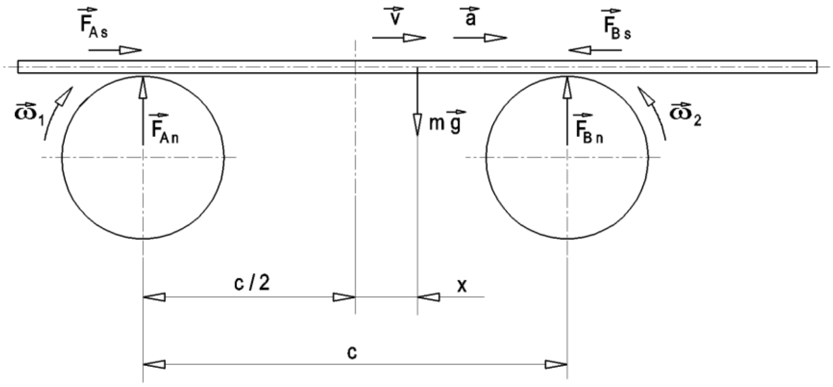

Figure 4. Kinematic and dynamic relations of the system [3]

3. THE USED PRINCIPLE OF THE MEASUREMENT

A homogeneous second order linear differential equation,

$$\ddot{x} + \frac{2\mu g}{c} x = 0 \, . \tag{3}$$

applies for the problem of Figure 4., where the coefficient of dynamic friction, μ is unknown, the acceleration of gravity, g and the centre distance of the pulleys, c are known. The momentary displacement of the centre of gravity of the horizontal rod, x and its momentary acceleration, $\ddot{x}$ are in the equation. Designating the multiplication factor of x by ω^2, the general solution is

$$x(t) = \alpha \cos(\omega t) + \beta sin(\omega t). \tag{4}$$

This $x(t)$ function is really the solution of the equation (3) where considering the initial conditions [at $t = 0$, x(t) = α and $\dot{x}(t) = 0$ cause $\beta = 0$], the solution of the simple harmonic motion is

$$x(t) = \alpha \cos(\omega t) \, , \tag{5}$$

where α is the amplitude of vibration and that of the circular frequency is

$$\omega = \sqrt{\frac{2\mu g}{c}} \, . \tag{6}$$

Measuring the periodic time of the vibration, T, the coefficient of dynamic friction is

$$\mu = \left(\frac{2\pi}{T}\right)^2 \frac{c}{2g} \, . \tag{7}$$

4. TEST RIG IMPLEMENTATION

The practical implementation of the conceptual assumption presupposes the lateral guidance of the vibrating horizontal rod. A circular cross-section rod suitable to move on the disks (pulleys) made of the inner ring of a deep groove ball bearing. The curvature relations are similar to that of the ball bearing, shown in *Figure 5*. On the inner ring of the bearing 6312, the main curve radii of the ball race are 36.4 mm and 11.5 mm, the balls have a diameter of 22.2 mm. Accordingly to this geometry, a rod with diameter of up to 22 mm is suitable for the 72.8 mm diameter discs. It is advisable to increase the centre distance of the disks to make the rod more balanced. Based on the description of the principle of measurement and the result of formula (7), it follows that under Earth conditions, in the case of fixed rotary axes, the result of the measurement depends only on the centre distance, c. It is independent of the speed of the discs, so they can be different. The truth of this state is probable at dry friction, low speed and under normal atmospheric and temperature conditions. However, due to the simplicity and the later extension of the measurement conditions, the speed of rotating discs are chosen to be the same.

Lining up four gears of the same number of teeth, the opposite direction of rotation and the same speed of the pulleys can be assured [3]. Revolving the pulleys in the proper direction the rod is moved inwards.

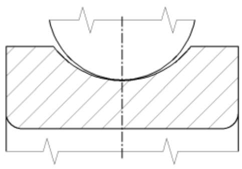

Figure 5. Curvature relations in a deep groove ball bearing

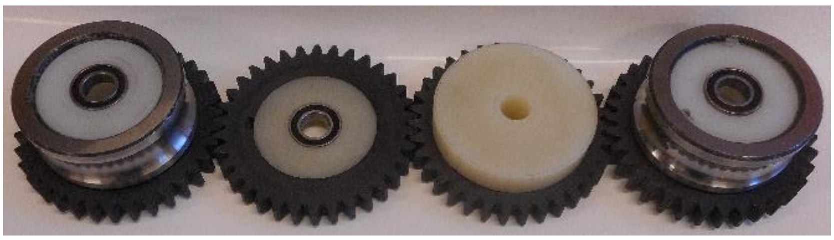

Figure 6. Powertrain of the pulleys

In the drive arrangement shown in *Figure 6*, the third gear (counted from left) is rotated clockwise to produce the desired motion conditions. The amplitude of reciprocating motion can be set to $0 < \alpha < c/2$ according to the initial condition. The amplitude cannot have a value close to the limits of the range, partly to make it easier to detect vibration movement and partly to balance the rod.

In case of dry friction, the friction coefficient is expected in the range of $10^{-1} < \mu < 10^{1}$, depending on the ambient temperature and pressure [1]. Let the periodic time of motion be in the range of $2{,}5s > T > 0{,}25s$. This periodic time allows the accurate detection of vibration, even visually. Due to equation (7), this corresponds to a $c = 10{,}3106\ m$ centre distance. *Figure 6* shows uncorrected (without any profile correction) spur gears. The number of teeth, $z = 34$ and the module, $m = 3\ mm$. The centre distance, c of the two pulleys is $c = 3a = 3zm = 306\ mm$, which fulfils the former criterion.

5. POSSIBLE TEST CIRCUMSTANCES

The measuring device is able to determine the sliding friction coefficient, originally in the case of dry friction. The drive is also insensitive to the speed and the rolling resistance of the bearings. Doe to the practical value of the friction coefficient, the relatively large periodic time makes the air resistance to be negligible. Any circumstance that reduces friction coefficient, e.g. lubrication, increases the periodic time, T which improves the accuracy of the given test method.

In the case of dry friction, the presence of low temperature and vacuum influence more strongly the value of the sliding friction coefficient. The effects of sliding speed and the curvatures are not significant. For lubricated surfaces, however, all other factors that may affect the lubricating condition should be considered. The most important factors are:

- The phase of lubricant
  - Gas,
  - Liquid,
  - Consistent (greases),
  - Solid (dry).
- Other properties of the lubricant
  - Viscosity,
  - Consistency,
  - Time-dependent rheological properties.
- Mixing additives to the lubricants,
- Roughness and pattern of the ball race of disc and the rod surface,
- The curvature of the disc and rod,
- The rotational speed of the disks,
- The loads on the surfaces,
- Possible solid surface coatings.

The above factors have a very high impact to each other, so the test possibilities are very wide. Without the necessity of completeness, some interesting cases should be mentioned.

- The rod is very light, made of steel tube with very small wall thickness and the disk has a very high peripheral speed. An aerodynamic lubrication condition may occur between the contacting surfaces. The coefficient of friction may fall below $\mu = 10^{-1}$, which results a significant increase in the periodic time of the harmonic motion.
- In case of fluid friction, the peripheral speed is also important because the friction condition is influenced by the viscosity of the lubricant, the highest asperities of the rough surfaces and the loading of the surfaces.
- In the case of grease lubrication, at very low temperature, both the viscosity of the base oil, forming grease, and the consistency of grease are decreasing. There is a need for time to distribute the lubricant to the correct lubricity.

While the lubrication condition is defective, no safe separation of metal surface asperities is achieved.

– The friction-reducing effect of some soft-metal coatings is only applied in case of higher loads [4], therefore, it is necessary to create a gradual increase in load. The simplest way is the loading of the rod having properly stiff at its ends so that the symmetry of the loads remains. The increment of air resistance cannot be considerable if the conditions of simple harmonic motion should be provided later on.

The test rig was made from the elements shown at *Figure 6*. The shaft of the third gear is driven by a controllable speed step motor through timing belt drive. *Figure 7* shows a measurement compilation. The room temperature was $22\ °C$. The electric motor is driven through a power supply. The adjusted speed is about 500 rpm, the rod and the disks are lubricated slightly by rolling bearing grease, made of mineral oil, viscosity $\nu_{40\ °C} = 97\ mms^{-1}$ with barium complex soap thickener, NLGI grade is 2. The solid rod diameter, $d = 15\ mm$, its length, $l = 1.1m$. The measured periodic time was $T = 2.4\ s$.

A similar measurement was made in dry condition. The rod vas a hollow cylinder of outer diameter, $d = 16\ mm$, wall thickness, $s = 1\ mm$ and length, $l = 0.5\ m$. The other circumstances were as the previous one. The periodic time was $T = 2.0\ s$. The counted coefficients of dynamic friction, doe to formula (7) are 0.107 and 0.154, respectively.

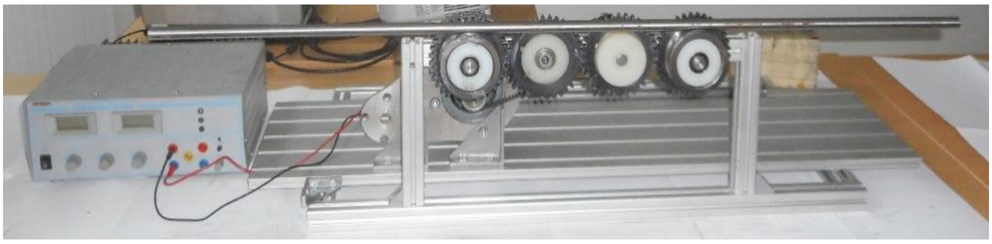

Figure 7. The test rig. Rod and disks are lubricated slightly by grease

6. Extension of the test principle

The principle shown in *Figure 4* and the experimental apparatus shown in *Figure 7* assume that the four gears with the same number of teeth determine unambiguously the speed of the discs on the first and last axes, and the average sliding speed between the disks and the reciprocating rod will be permanent. Let's modify this principle to make the test rig suitable for measuring rolling resistance of rolling bearings and for studying the effect of lubricant and ambient temperature on rolling resistance. *Figure 8* shows the modified principle.

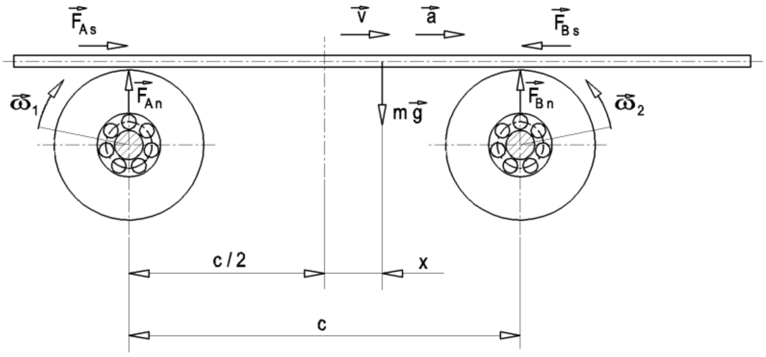

Figure 8. Modified principle. The disks are revolved only by the rolling resistance

In contrast to the original task, only the inner ring of the bearings that support the disks are rotated. The outer rings and the discs rotate only due to rolling resistance. The rolling resistance is influenced by the part of mass of the rod, the mass of outer ring of bearing and the mass of the disks. The friction force between the disks and the rod is considerably larger than the rolling resistance of the bearings, and the friction force acts at much greater arm than the rolling resistance, so the rod will not slip on the discs. If the rod accelerates, the angular acceleration of the disks can be linked unambiguously. It can also be clarified that the angular accelerations of the two disks are equal in magnitude and direction, due to the clear rolling between the disks and the rod. Not only the rod, but the disks perform harmonic motion, and it is expected that the vibration will be unattenuated as well. To formulate the motion equation for the left-hand disk, *Figure 9* illustrates the diameters d and D, the mass moment of inertia, J, the outer forces, F_s and F_g, and the angular acceleration, ε.

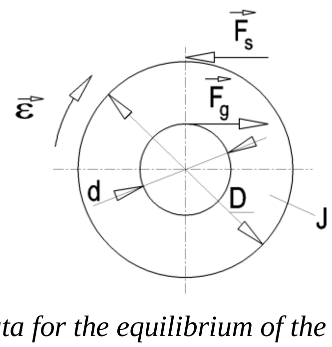

Figure 9. Data for the equilibrium of the left hand disk

The normal force acting to the disk from the rod, due to *Figure 8* is

$$F_n = \left(\frac{1}{2} - \frac{x}{c}\right) mg. \tag{8}$$

The normal force acting upwards from the balls to the outer ring of the bearing is completed by the sum of the disk and the outer ring of bearing, $m_t g$. The equation of the torques from the external forces and the acceleration moment in the case of the left disc, with the symbols of *Figure 9* is

$$\frac{d}{2}F_g - \frac{D}{2}F_s = J\varepsilon. \tag{9}$$

from which the friction force F_s can be expressed and its value is much smaller than that of the problem of *Figure 4*. The rolling resistance from the supplemented equation (8) is

$$F_g = \mu_g\left[\left(\frac{1}{2}-\frac{x}{c}\right)mg + m_t g\right]. \tag{10}$$

To obtain the friction force acting at the circumference of the left side disc, the instantaneous acceleration of the rod, $\ddot{x} = \varepsilon D/2$ should be considered, and equation (10) should be substituted into equation (8), i.e.

$$F_s = \frac{d}{D}\mu_g g\left[\left(\frac{1}{2}-\frac{x}{c}\right)m + m_t\right] - \frac{2}{D}J\varepsilon. \tag{11}$$

The formula of the friction force acting at the circumference of the right side disc is similar. The angular acceleration of this disc is the same, both in magnitude and direction as that of the left side disc, i.e.

$$F_s = \frac{d}{D}\mu_g g\left[\left(\frac{1}{2}+\frac{x}{c}\right)m + m_t\right] + \frac{2}{D}J\varepsilon. \tag{12}$$

The motion equation of the horizontal rod, obtained from Newton's second law, similarly to equation (4) is a homogeneous second order linear differential equation, i.e.

$$\ddot{x} + \frac{\frac{d}{D}\frac{2\mu_g g}{c}}{1+\frac{8J}{D^2 m}}x = 0\,. \tag{13}$$

Comparing this to the simpler equation (4), there are some extra influencing factor, as the mass of the rod, m, the diameter of the ball race at the outer ring of the investigated bearing, d or the diameter, D and the mass moment of inertia, J, of the discs, as *Figure 9* shows. J also involves that of the outer ring of the analysed deep groove ball bearing and the hollow disc between the bearing and the pulley of the horizontal rod.

Here the periodic time, T of the simple harmonic motion is measureable, so the circular frequency of the vibration is countable, i.e. $\omega = 2\pi/T$. At last, the coefficient of rolling resistance of the investigate bearing is

$$\mu_g = \frac{D}{d}\left(1 + \frac{8J}{D^2 m}\right)\omega^2 \frac{c}{2g}. \tag{14}$$

The only uncertainty in the calculations is caused by the determination of the mass moment of inertia, J of the elements between the balls of the analysed bearings and the horizontal rod.

7. PRACTICAL APPLICATION OF THE USED TEST PRINCIPLE

The elements shown in *Figure 6* or the test rig at *Figure 7*, following the theoretical solution of *Figure 4*, is suitable to test the sliding friction between cylindrical and toroidal surfaces which has point contact and both the speed and load may be varied. The Hertzian stress calculated at the maximum adjustable load (to be described later) in the contact between the inner ring of a deep groove ball bearing 6312 of *Figure 5* and a 22 mm diameter cylindrical bar is approximately 320 MPa, well below the allowable practical values. The measurement of dynamic friction coefficient is possible with or without lubrication. The favoured lubricant is grease. Supplying and removing the lubricant on or from the surfaces, respectively can be carried out quickly and easily. The free choice of lubricant type is limited by load and temperature range. Any type of oil lubrication would require a closed gearbox.

The toroid surface of the disks (ball race of the bearing) is made of quenched chromium steel. The material quality and roughness of the rod surface can range from cold drawn normalized unalloyed tool steel to grinded, hardened high alloy steel. Increasing the load is possible with symmetrically located masses at the ends of the horizontal rod. The magnitude of the masses is limited by the usual degree of allowable deflection of the axles. Let this value be $y_{all} = 0.3\ mm$ in the case of a cylindrical steel bar with a diameter of $22\ mm$ and a length of $0.9\ m$, for which considering the centre distance, c the overhanging ends can be loaded up to a maximum of 3 kg mass, in addition to the own mass of the rod of almost 3 kg.

Reduction of the load can be achieved by choosing a cold-drawn precision steel tube of $1\ mm$ wall thickness, in this case its own weight is below $0.5\ kg$, so it is possible to test lubricating greases with less consistency and lubricating oils of lower viscosity.

The ambient temperature of the test does not deteriorate the accuracy of the measurement within a permissible limits, so it is possible to detect degradation of the lubricant caused by the lower temperature ranges.

According to the principle shown in *Figure 8*, the rolling resistance of rolling bearings can be tested. The practical implementation of the measurement requires further considerations since it is necessary to solve both the easy and quick insertion of the bearing and the changing of tested lubricant in the measuring rig.

8. FURTHER TASKS OF THE MEASURING RIG

The rotation of the disks is carried out by a variable speed electric motor. In case of the need for high sliding speed changes, the replacement of the electric motor and the gear unit can be solved, too. The number of revolutions of the discs can be measured using a mechanical counter or an inductive encoder and a recapitulative device.

The total number of vibrations carried by the rod can also be measured by an inductive transducer, receiver and summing counter. The magnetic encoder may be attached to the centre of the rod and the receiver head is fixed to the stand. The summing counter also measures the time t of the test. The summing counter connected with the receiver counts the passes, N at the equilibrium position. The periodic time, T is twice the ratio of the usual one, i.e. $T = 2t/N$.

9. SUMMARY

By studying a well-known sliding friction measurement principle, the authors have been able to clarify the basic principles of the operation of a device applicable to test the rolling resistance of a given deep groove ball bearing supplied by any type of lubricating grease, at an adjustable operating temperature, thereby detecting a special ability of a particular grease.

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