DISC-DISC GEOMETRY AS AN ALTERNATIVE TO THE FZG TEST METHOD

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Abstract: *This work has intends to show the methodology developed and the results obtained so that the Twin Disk machine of the Metrology Laboratory of the Federal University of Rio de Janeiro can be used as an alternative to theFZG gear-testing machine. The regime of elastohydrodynamic lubrication found in gears can be obtained in the TwinDisk machine, provided that the operational parameters are adjusted so as to represent the phenomenon that occurs in thelubrication zone of gears in operation. Basically, the FZG machine determines the load limit that can be used onlubricating oil without the occurrence of failures by scuffing. The twin disk machine, not only achieves failure by scuffing, but also allows the adjustment of parameters such as, rolling speed, slide roll ratio and Hertz pressures.*

Keywords: *Elastohidrodinamic Lubrication, FZG machine, Scuffing, Wear, Twin Disk machine.*

1. Introduction

Typically, elastohydrodynamic (EHD) lubrication happens in roller bearings, cams and gears. Due to the high contact pressures, the lubricant film thickness is less than 1 µm. The rupture of the film causes high wear of the surfaces involved, bringing about a reduction in the working life of the equipment. A lot of models were developed to predict the film thickness under elastohydrodynamic conditions, and all of them predict that the film thickness depends on the speed between the surfaces, the geometric and elastic parameters of the materials involved and the behavior of the lubricant. Besides the lubricant film thickness, protection against wear is also influenced by other factors, such as, the use of high-pressure additives, that can protect the surfaces under high Hertz pressures.

Therefore, in order to develop a lubricant for EHD conditions, its necessary use experimental methods that permit the assessment of the wear protection capacity of the oils. Before being used in the field, the oil must be tested in situations that reproduce the elastohydrodynamic conditions. There are many types of tribological tests, such as, the Four-Ball, a test that simulates the lubrication of roller bearings and the FZG, that uses gears to evaluate oils. The FZG test has a methodology based on the wear in the gear, however, such gear teeth must have a heat treatment and manufacture, which makes the oil evolution very expensive.

As an alternative to the FZG test, a Twin Disk machine was developed in a joint project between the Metrology Laboratory of the UFRJ and the Management of Lubricants and Special Products of the CENPES/PETROBRAS. The results obtained indicate that the use of the disk geometry permits the evaluation of the oil wear protection capacity, becoming an alternative to the FZG test, with the benefit of reducing costs for the manufacture of test samples.

2. Elastohydrodynamic Lubrication (EHL)

The condition of EHD lubrication occurs when the film thickness is of the same magnitude as the composition of the roughness of the materials involved (Kragelsky, 1981). Some authors developed mathematical models to forecast the film thickness.

The models of film thickness in EHL have different experimental constants depending on the conditions of load, velocity, material and lubricant that make up the tribosystem. The use of the equations with dimensionless parameters, makes possible to summarize the film thickness equations of the EHL regime in a single general equation, Eq. 1:

$$\bar{h} = Z.(g1)^m.(g3)^n \qquad (1)$$

g_1 is the parameter that represents the viscosity of the lubricating fluid and g_3 is the elasticity of the materials. The coefficients of Eq.1 are set out in Tab. 1, which lists the different regimes of lubrication with their respective coefficients.

Table 1. Coefficients of film thickness for different EHL regimes (Johnson, 1970)

Authors	*EHL Regimes*	*Z*	*m*	*n*
Martin	Rigid-Isoviscous (R-I)	4.90	-	-
Blok	Rigid-Piezoviscous (R-V)	1.66	2/3	-
Herrebrugh	Elastic-Isoviscous (E-I)	3.10	-	0.80
Dowson-Higginson	Elastic-Piezoviscous (E-V)	2.65	0.54	0.06

On applying the models described in Table 1 to rough materials, a new unmentioned element arises and influenced the lubrication process, the roughness.

According to the Olver (2002) division, the effects of roughness in EHL can be divided into 3 categories:

1 – *Micro elastohydrodynamic lubrication* (Micro-EHL): there is a continuous fluid film, but the pressure and film thickness are subject to local fluctuations arising from the surface roughness;

2 – *Mixed lubrication*: the fluid film is discontinuous with some or many of the crests of roughness, which are subject to solid-solid or semi-solid contact;

3 – *Boundary lubrication*: The lubricating fluid, if present at all, is confined to the valleys and at a low pressure, thus supporting an insignificant portion of the total load.

The transition from full EHL to the category micro-EHL and afterwards to mixed lubrication is characterized by the rate of decrease of the film thickness taking into account the modifications of the roughness peaks. This transition characterizes the presence of more than one lubrication mechanism, in which the effects of the roughness are added.

In 1959, Dowson and Higginson proposed an inverse hydrodynamic method as a solution. The results were grouped in four dimensionless parameters, widely used in EHD problems, as described in Eqs. 2:

$$H = \frac{h_{min}}{R'}, \qquad W = \frac{w}{E'R'}, \qquad U = \frac{\mu_0 u}{E'R'}, \qquad G = aE' \qquad (2)$$

Where h is the minimum film thickness, R' is the radius of equivalent curvature, w is the load per unit of length, E' is the equivalent module of elasticity, μ_0 is the viscosity at room temperature and pressure, u is the velocity and a is the coefficient of pressure-viscosity.

Through these parameters, the numeric solution of Dowson and Higginson for a line of contact, can be represented by Eq. 3:

$$H = 2.65\, G^{0.54}\, U^{0.7}\, W^{-0.13} \qquad (3)$$

H is the dimensionless film thickness, W the dimensionless load, U the dimensionless velocity and G the dimensionless material.

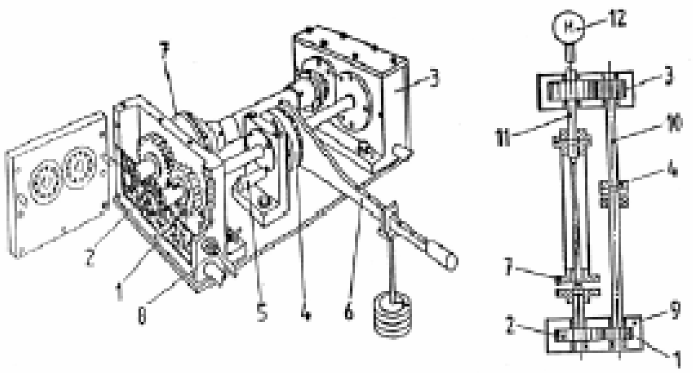
3. Tribological tests

3.1 Four-Ball Test

The Four-Ball test is used to determine the extreme pressure property of lubricating fluids (ASTM D 2783, 1998), and consists of a pot containing three steel spheres immersed in a test fluid and held stationary. A fourth sphere is positioned so as to rotate under a controlled load. It intends to determine the load at which the spheres weld or to measure the scar diameter of the wear caused by the movement between the spheres.

3.2 FZG Test

The testing of lubricating oils in the FZG machine with a scaled load increase (DIN 51354, 1990), intends to determine the load limit of each oil, characterized by the occurrence of scores and corrosion points. The method consists of immerse the gears in a lubricating bath to be tested at a constant speed and with the initial oil temperature pre-established. Gradually, the load on the gears is increased and, at each new stage, the surface alterations of the teeth flanks are visually checked. In Fig. 1 the FZG test machine is shown with its power circuit and load application.



1. Test pinion	7. Torque measurement clutch
2. Test wheel	8. Temperature sensor
3. Drive gears	9. Heat system
4. Load clutch	10. Shaft 1
5. Locking pin	11. Shaft 2
6. Load lever and Weights	12. Drive motor

Figure 1. FZG Machine (DIN 51354, 1990)

There are 12 possible stages in this machine, each of them characterized by an increase in load, that is, an increase in the normal force between the gear teeth. The stage of failure takes place when the total sum of the pinion wear (all the scores and corrosion points) extends 20 mm, which is the width of the gear teeth. The material of which the gears are made has a high manganese and chrome content (20 MnCr5) and the thermal treatments used aim not only increasing the surface resistance, but also at guaranteeing that the gears do not fail.

3.3 Twin Disk Test Machine

With the objective of obtaining the EHD conditions in a simpler and more economic device, a test bench was built where two disks are placed in contact, it is possible to drive one independently of the other and also to apply a controlled load upon them. Besides this, the Twin Disk machine was designed with a test box that contains the test disks, their shafts, roller bearings, and the oil to be tested. It was necessary to immerse the disks in a bath of oil in order to guarantee a sufficient quantity of the fluid under test.

Load is applied by means of a hydraulic cylinder that acts upon the bearing shaft of disk 1. This shaft is mounted over a dove-tail linear guide. The disks are driven by 2 motors of 5.5 kW each and controlled by frequency inverters. A screw/crown gear box was added after each motor with the purpose of attaining low rolling speeds, thus increasing the torque. Between the gear boxes and the shaft where the disks are mounted there is a torque transducer with a capacity of measuring torques of up to 500 Nm.

Having in mind the need for an accurate measurement of the force between the disks, a load cell with a capacity of up to 100,000N was installed between the hydraulic cylinder and the base of the bearings of disk 1. Encoders of 720 pulses/rotation measure the velocity of the shaft at each moment of the test. A thermal resistor PT-100 was installed near to the lubrication zone to measure the oil temperature. In Fig. 2, the configuration of the machine is shown.

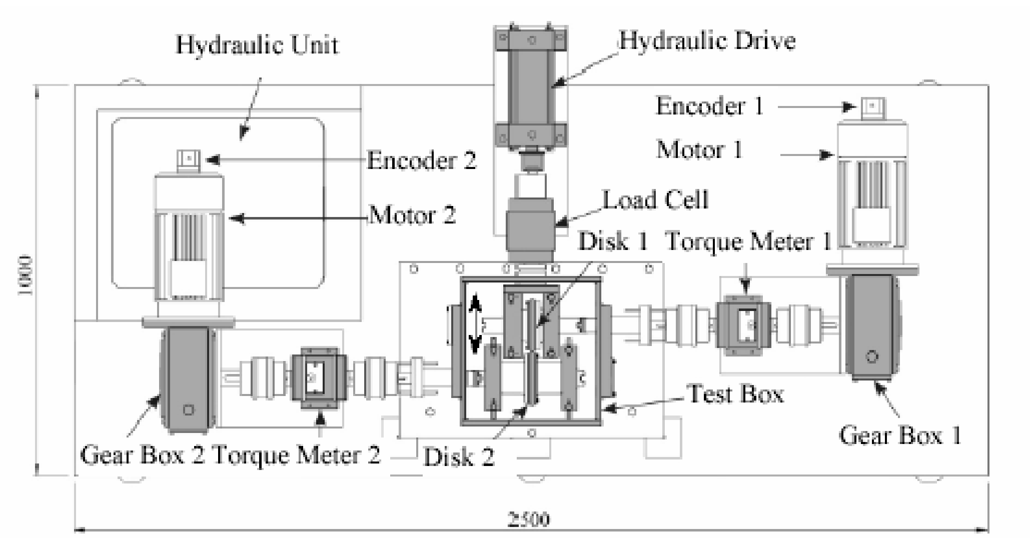


Figure 2. Configuration of the Twin Disk machine

A temperature control system for the lubricant was implemented, which is able to heat from room temperature up to 120°C controlled by a thermostat. The signals from all transducers are amplified and received in a LabView environment. In Fig. 3, there is a photograph of the Twin Disk machine.

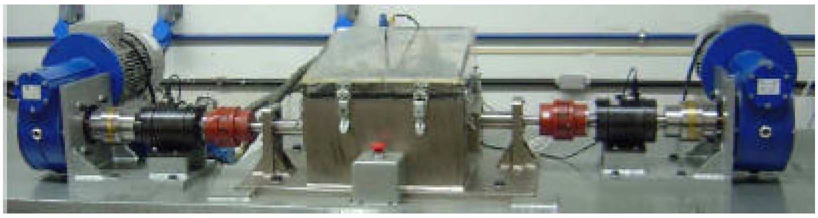


Figure 3. Twin Disk Machine of the Metrology Laboratory of UFRJ

4. Experimental configuration for Twin Disk geometry as an alternative to the FZG

On comparing contact between gears with the contact between the two disks, various factors should be taken into consideration, due to the fact that the gear geometry is quite different to the cylindrical geometry of the disks. In a pair of geared teeth with an involute profile, the radius of the curvature at the point of contact of the gear varies, in accordance with the movement of the contact point along the line of action. Thus, it is reasonable to conclude that the Hertz contact pressure will also vary; as a result, it is necessary to establish the rule for the choice of the curvature radius that will be used in the calculations. Considering the physical meaning of the involute profile as the unwinding of a string around the circumference of the base, we can infer that the curvature radius varies from zero (at the base of the tooth) to a maximum value (at the crown of the tooth), passing through an intermediate value in the primitive diameter. Due to the differences in the geometry of the two gears, we will have at the point of contact of the two involutes, different values for the curvature radius for each one of the gears. The radius of equivalent curvature both for the gears as well as for the disks can be calculated with Eq. 4:

$$1/? = (1/?_1 + 1/?_2) \qquad (4)$$

Where ? is the radius of the equivalent curvature.

In Table 2, the values of the curvature radii are shown for the FZG gears and for the disks of the Twin Disk machine.

Table 2. Radius of Curvature

	Radius [mm]	Radius of Curvature ? [mm]	Radius of Equivalent Curvature R´ [mm]
Pinion	36.6	14	8.4
Gear	54.9	21	
Disks	67.5	67.5	33.75

To made correlation between the phenomenon that occurs between the gear teeth and the contact between the disks, both the film thickness, and the slide roll rate speed must be equal. To calculate the appropriate rolling speed for the geometry of the disks, the Dowson-Higginson model was used, modified for the experimental conditions, as shown in Eq. 5.

$$h_{min} \sim R'^{0.43}\, u^{0.7}\, w^{-0.13} \qquad (5)$$

Where R´ is the equivalent curvature radius, u is the rolling speed and w is the load per unit width.

From this relationship of proportionality, it is possible to calculate what rolling speed should be used so that a lubricating film of the same thickness is in the FZG gears and the twin disk machine. The Hertz pressure contact in the 12 stages should be equivalent both in the FZG and in the Twin Disk geometry. Tab. 3 shows the force applied to the gear teeth of the FZG and its respective Hertz pressure as per DIN 51354 (1990), and the force applied to the disks for each stage.

Table 3. Loads and Hertz pressure for FZG and Twin Disk

Degree	Force applied in the FZG [N]	Hertz Pressure [N/mm^2]	Forced applied to the disks [kN]
1	99.0	146	-
2	407.0	295	-
3	1044	474	3.0
4	1800	621	5.0
5	2786	773	7.5
6	4007	927	11
7	5435	1080	14
8	7080	1232	19
9	8949	1386	25
10	11029	1538	30
11	13342	1691	36
12	15826	1841	43

To calculate the slide velocity, we used an average of the product of the normal force, “P”, between the gears teeth and the relative speed between the surfaces. To make the graph of how the product of the force P and the velocity vary along the line of action, an approximation in which the load is equally divided when two pairs of teeth are engaged. The contact ratio shows the proportion of engagement when two pairs of teeth are engaged. Using the theory of recovery degree (Nelson, 1980), we calculated that in 39% of the time of engagement two pairs of teeth will be engaged. The line of action is 18mm, then in approximately 11mm there will only be one pair of teeth engaged. Fig. 4 shows how these 11mm are distributed along the line of action.

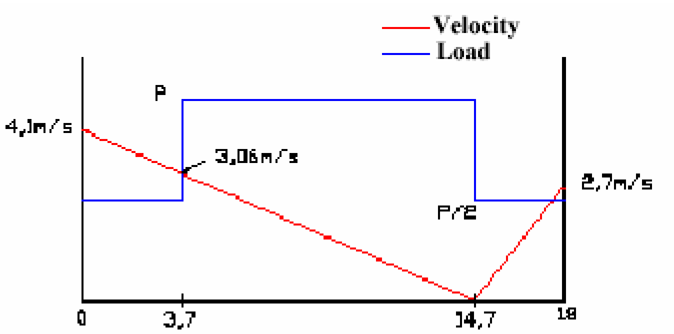


Figure 4. Load and velocity along the gear engagement.

As it not possible to continuously vary the slide speed in the Twin Disk machine, a constant slide speed was chosen, and this slide speed is proportionally to shaded area in Fig. 5.

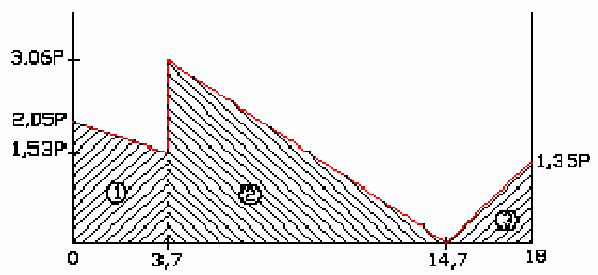


Figure 5. Product of the normal load times the velocity

As a result, we can conclude that the average slide roll ratio of the FZG is equivalent to the slide roll ratio of the Twin Disk machine. Therefore, using Eq. 5 and the established slide roll ratio, it is possible to select the rotation speeds of the test disks, which are shown in Table 4.

Table 4. Velocities of the FZG and the Disks

	FZG		Twin Disk	
	Pinion	Crown	Disk 1	Disk 2
Rotation of the shaft [RPM]	2170	1440	305	192
Rolling Speed [m/s]	3.18		1.79	
Slide Roll Ratio	0.45		0.45	

5. Experimental methodology

Besides using the some operating parameters used in the tests with FZG test rig, it is also necessary that the surfaces hardness and finishing are similar. Johnson and Spence (1991), carrying out their experiments in a twin disk machine, presented results that clearly show that disks ground in the circumferential direction have a coefficient of traction much greater than disks ground in the transversal direction. Therefore, to follow the direction of the asperities made in the gear grinding, it was necessary to adapt a tool sharpener to grind the disks in the transversal direction. The disks were made with 20MnCr5 steel, and, after milling, the disks were cemented so as to obtain a surface hardness of 60 to 62 Hc. After heat treatment, the disks were ground and their final roughness $\boldsymbol{R_a}$ was around 0.50 μm, the contact width between the disks being 10±0.2mm.

Five oils were used in the tests to evaluate the correlation between the results obtained with the Twin Disk machine and the traditional tribological tests. The first, called ***EGF 68-PS***, is a lubricating oil for enclosed gears and industrial gears boxes in heavy duty services under high loads. The second, called ***HR 68-EP,*** is recommended for hydraulic systems that operate in severe conditions of pressure and temperature, having mainly anti-wear and anti-corrosion additives. The third, called ***TR 68,*** is a lubricating for use in turbines, gear boxes and hydraulic systems in light services. The fourth and the fifth, denominated ***RL 133*** e ***RL 144*** respectively, are reference oils for FZG test machines. Table 5 presents the results of the main physical-chemical characteristics of the tested oils.

Table 5. Characterization of the Oils

Property\Oil	EGF 68-PS	HR 68-EP	TR 68	RL 133	RL 144
Density, g/cm^3	0.887	0.875	0.873	0.891	0.872
Viscosity at 40°C, cSt	70.36	66.11	63.20	103.20	49.71
Viscosity at 100°C, cSt	8.752	8,750	8.512	13.750	7.148
Viscosity Index	96	105	105	134	101

A new pair of disks was used in each test, each oil was tested twice in identical operational conditions, that is, each test was repeated with the purpose of evaluating the repeatability achieved with the equipment. At each change of lubricant, the whole test box was cleaned with the solvent n-heptane, and, to remove possible residues of the solvent, the box was flushed with the lubricant to be tested afterwards.

The tests started with a load corresponding to stage 3 of the FZG. The temperature system control, similar to the FZG machine, maintained the temperature at 90°C until stage 8, from then on the bath temperature increased until it reached 100°C at stage 12. The loads were increased consecutively with intervals of approximately 15 minutes each. Fig. 6 shows one of the tests performed.

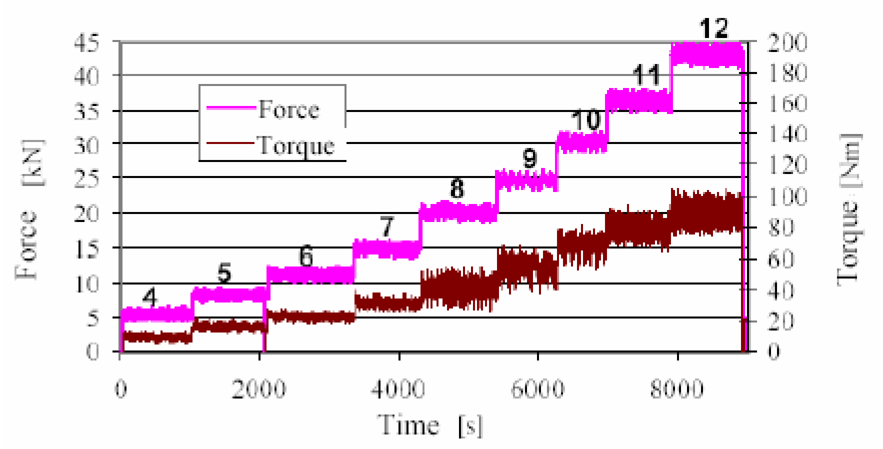


Figure 6. Test of the oil EGF 68-PS

It can be noted that a dynamic load appeared in the measure of the normal force, and the same in the measured torque. This dynamic component of the force is caused by the disalignment occurred by the sets of shafts and disks. Despite appearing inconvenient, this vibration makes the Twin Disk test more similar to the FZG test rig, which presents vibrations due to the variations of load in the contacts between the gear teeth.

6. Results

In order to investigate the lubricating power of the five oils employed, laboratory tests were made which results are presented in Tab. 6. The first was the Four-Ball wear test, where the scars on the spheres were measured after 1 hour of testing under a load of 40kg at 1200RPM. In this test, the smaller the scar is, the greater the wear resistance will be. In a FZG test, on the other hand, the higher load stage is, the better the lubricant will be.

Table 6. Results of the Four-Ball, FZG, Twin Disk.

Tribological Test	EGF 68-PS	HR 68-EP	TR 68	RL 133	RL 144
Four-Ball(1)	0.29	0.47	0.61	0.33	0.64
FZG	12	11	7	11	6
Twin Disk	12	10	9	11	8

(1) The values refer to the diameter of the measured scar, in mm.

From the quantitative point of view, the results proved satisfactory for the evaluation of lubricants as to their capacity of wear protection, since they are similar to those obtained with the FZG. Also all of the tests presented results corresponding to the performance of the oils. In the oils where better results in the FZG tests were obtained (EGF 68-EP and RL 133) less wear with the Four-Ball was observed, this shows the coherence of the adopted methodologies.

Below we present two photographs that show a failure by scuffing both in a gear of the FZG and in a tested disk. Before the disks suffer severe damages it is possible to verify the pitting that happens because of fatigue of the material, which is different from the scuffing that occurs due to the failure of the lubricant in separating the surfaces.


Figure 7. FZG Gear standard scuffing (Höhn, 2004)

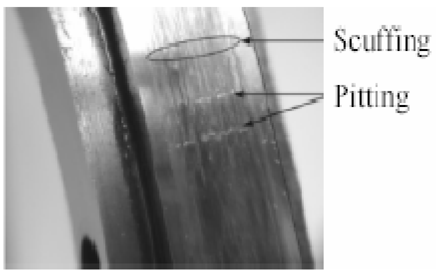


Figure 8. Tested disk scuffing and pitting

7. Conclusions

The Twin Disk testing machine built at the Metrology Laboratory of UFRJ is capable of working within the levels of load and velocity needed for the correlation with the FZG machine. The 12 load stages were tested without problems by the EGF 68-PS oil, however, due to the robustness of the load application system, it is not possible to have an accurate control of loads below 3,000N. For this reason, the tests in the Twin Disk machine were initiated with a load equivalent to stage 3.

The use of the Twin Disk machine simulating the FZG tests, proved itself to be a tool capable of differentiating the protection that the lubricating oils offer against wear, thus becoming an economic alternative in the development of new formulations of oils.

Despite the fact that the contact is a line for the disks and a point for the Four-Ball spheres, the wear results in the Twin Disk machine accompanied in a qualitative manner the results obtained in the Four-Ball machine.

The failure stage in the Twin Disk machine is clearly shown by the torque signals of the two shafts and not by visual means, as it is the case of the FZG. Another advantage of this procedure of failure determination is that it is not necessary to open the equipment to verify the scuffing marks.

It should be pointed out that these are preliminary results, and that with future work the intention is to improve the correlation developed between the FZG testing machine and the Twin Disk testing machine.

The operating parameters of the Twin Disk machine may be adjusted to modify the tests, thus, altering the thickness of the film or the Hertz pressure contact, as desired to adjust the working conditions.

8. Acknowledgements

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