Essay

A Method for Monitoring Lubrication Conditions of Journal Bearings in a Diesel Engine Based on Contact Potential

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Abstract: Wear of the journal bearings in a diesel engine is usually caused by asperity contact. Increased contact potential is caused by the asperity contact between the journal bearing and the shell. This paper analyzes the relationship between the contact potential and asperity contact and presents a method based on contact potential to monitor the bearing wear caused by asperity contact. A thermo-elastic hydrodynamic lubrication (THL) model of the journal bearing on the test bench was established and was verified by measuring its axis orbit. The asperity contact proportion was calculated based on this THL model, and its relationship with the measured contact potential was determined. The main contribution of this paper is to present a new method for monitoring the lubrication conditions of journal bearings in a diesel engine based on contact potential. The results showed that (a) when the minimum oil film thickness was less than 5 µm, asperity contact occurred between the bearing shell and the journal, which led to a sharp increase in contact pressure and a rapid increase in friction power consumption. Further, (b) there was a positive correlation between contact potential and asperity contact. The contact potential was greater than 0.75 mv when asperity contact occurred. These results proved that asperity contact could be accurately monitored using the contact potential, and the feasibility of using the contact potential to monitor the lubrication condition of a bearing was verified.

Keywords: diesel engine; bearing wear; monitoring; contact potential

1. Introduction

Approximately 25% of total friction losses in machines are caused by bearings, especially in certain high-power internal combustion engines, where the proportion of the total loss may reach as high as 40% [1]. When a friction pair is severely worn, a series of serious effects will occur, which will affect the economy, power, and reliability of the internal combustion engine. According to statistics on ship claims from the Swedish Club (2010–2016), in main marine engines, compensations for main bearing failure account for 33.5% of total compensations, with an average compensation of 1.6 million US dollars per claim. In addition, the number of compensations increased by 110% across the data period, now ranking first among single causes of compensation across all part failures [2]. Thus, research analyzing and monitoring the lubrication conditions of these bearings is of great significance. Nowadays, many scholars have studied the hydrodynamic lubrication of journal bearings. Recently, many research studies on the two aspects of simulation analysis of bearing lubrication conditions and bearing condition monitoring have been published.
For simulation analysis of bearing lubrication conditions, Tucker et al. presented a flexible CFD modelling approach for the prediction of temperature in a journal bearing. The results showed that circumferential variations in shaft temperature are localized close to the shaft surface and are not significant for bearing performance when the shaft position is steady [3]. Then, taking into account the time dependence of temperature in lubricant film, Tucker predicted the thermal design problem for a generic two-axial groove bearing [4]. However, these studies are steady-state analyses of bearing lubrication that did not consider the influence of external load changes. Shahmohamadi et al. considered the influence of dynamics in their CFD model and analyzed the power loss and minimum film thickness at different crank angles [5]. However, the effect of surface roughness was not taken into account in the studies above. In order to ensure the accuracy of the calculation, the effect of surface roughness on asperity contact is considered in the calculation model in this paper.

For bearing condition monitoring, the traditional methods for testing or monitoring the lubrication conditions of bearings include the temperature method, the vibration method, and the oil analysis method. Some researchers have analyzed bearing temperature data to generate bearing fault warnings. Azevedo et al. proposed analyzing the evolution of temperature in a bearing to know whether the temperature increased because of a fault or due to higher load [6]. Kusiak et al. used a neural network model to analyze bearing temperature data and evaluate the lubrication conditions of bearings [7]. Yilmaz et al. used the temperature signals to diagnose motor bearings [8]. However, because of the thermal inertia of the temperature sensors, their response speed is not as good as that of the vibration method. Other researchers used the vibration method to monitor the lubrication conditions of the bearings, such as Kanematsu, who used the vibration method to diagnose motor bearing faults [9]; Chen, who used vibration signals to diagnose faults in big end bearings [10]; Xu, who analyzed the vibration characteristics of different clearances of bearings to monitor the lubrication of bearings [11]; and Mao, who proposed a method to monitor bearing failure based on vibration [12]. However, the vibration method is difficult to apply due to the many excitation sources, complex transmission paths, signal interferences, and low signal-to-noise ratio. Some researchers used the oil analysis method to monitor the conditions of bearings, such as Bai, who used ferrography and spectrometric analysis technology to monitor the conditions of bearings [13], and Shah, who described a method to monitor main bearing health using periodically collected lube oil samples [14]. Because the collected lubrication oil flows through various friction pairs within a diesel engine, it is difficult to determine which friction pair is faulty.

Scholars have analyzed the relationship between contact potential and wear. Priestner et al. [15,16] studied a fatigue wear test-bed using a Sapphire, established an elastic–hydrodynamic lubrication (EHL) model that considered the asperity (Greenwood and Tripp model), and verified the precision of the model using the temperature parameters on the back of the shell. The effects of the load and the lubricant brand on friction power consumption, maximum peak pressure, and minimum oil film were analyzed. The relationship between the contact voltage and asperity contact was analyzed. However, the relationship between the contact voltage,asperity contact, minimum oil film, and axis orbit was not further analyzed. Zhu [17,18] designed a thermoelectric sensor and built a thermoelectric monitoring system, which verified that the thermoelectric signals of a diesel engine were repeatable. The effectiveness of the thermoelectric method was confirmed when monitoring the wear of the main bearings in medium-speed marine diesel engines. Contact voltage plays an important role in the bearing monitoring method based on the thermoelectric method. If the bearing enters the hydrodynamic lubrication state, the potential should be stable. However, if the bearing enters the mixed lubrication state, the bearing shell and journal may be in contact and result in a change of contact voltage, which can be used to monitor the lubrication state of the bearings [17]. However, the relationship between the contact potential and the lubrication state was not studied based on this mechanism. The connection between the contact potential and contact was not studied systematically. The lubrication state of the journal bearings was restricted by many factors, and the signals could not
be investigated independently on an actual machine. To study bearing wear monitoring based on a contact potential method, a single-factor fault simulation test is needed.

This paper establishes a thermo-elastic hydrodynamic lubrication (THL) model of a bearing in a Sapphire test bench. The THL model is based on the Reynolds equation, the viscosity temperature characteristics, and the Greenwood/Tripp asperity theory. Single-factor tests with different speeds and loads were performed to verify the accuracy of the model through the axis orbit. The relationship between the asperity contact and the minimum oil film thickness was analyzed by using the calculation model, which demonstrated that the minimum oil film thickness was very small when asperity contact occurred. The bearings were in the mixed lubrication condition, and the bearing lubrication condition degraded. The contact potential obtained through the test also had the same characteristics as the asperity contact percentage. The contact potential could reflect whether the bearing journal and shell were experiencing asperity contact and whether the bearing had entered the mixed lubrication condition. Thus, the contact potential could be used to monitor the lubrication condition of the journal bearing.

2. Lubrication Theory of a Journal Bearing

Unlike the solution of EHL lubrication, the THL model solution is based on the temperature distribution of the lubricating oil film according to the energy equation. When solving the Reynolds equation, changes in the viscosity value due to temperature and pressure should be considered. Moreover, the temperature distribution is affected by the pressure distribution and the interface boundary conditions. To solve the THL model, the EHL lubrication, the interface boundary conditions, and the energy equation should be calculated simultaneously.

2.1. Multibody Dynamic Theory

When establishing the shafting model, a simplified bearing is adopted to facilitate the calculation. The modal compression method is applied to the solution process in order to compress the journal and the shell in the same plane.

\[
M \ddot{u} + C \dot{u} + K u = F \quad (1)
\]

In multibody dynamic analysis, the Guyan/Craig-Bampton method is used to reduce the freedom degrees of the crankshaft [19,20], and Equation (1) can be rewritten as follows:

\[
\begin{bmatrix}
M_{ii} & \cdots & M_{ir} \\
\vdots & \ddots & \vdots \\
M_{ri} & \cdots & M_{rr}
\end{bmatrix}
\begin{bmatrix}
\ddot{u}_i \\
\vdots \\
\ddot{u}_r
\end{bmatrix}
+ \begin{bmatrix}
C_{ii} & \cdots & C_{ir} \\
\vdots & \ddots & \vdots \\
C_{ri} & \cdots & C_{rr}
\end{bmatrix}
\begin{bmatrix}
\dot{u}_i \\
\vdots \\
\dot{u}_r
\end{bmatrix}
+ \begin{bmatrix}
K_{ii} & \cdots & K_{ir} \\
\vdots & \ddots & \vdots \\
K_{ri} & \cdots & K_{rr}
\end{bmatrix}
\begin{bmatrix}
u_i \\
\vdots \\
u_r
\end{bmatrix}
= \begin{bmatrix}
F_i \\
\vdots \\
F_r
\end{bmatrix}. \quad (2)
\]

2.2. Reynolds Equation

During the working time of the internal combustion engine, the crankshaft bears an alternating load with periodic changes in size and direction. Stress and deformation are produced on the surface of the bearing journal and shell, which can change the thickness of the oil film. This difference in thickness will result in different contact forms between the journal bearing and shell. The lubrication forms for the journal bearing are boundary lubrication, mixed lubrication, and dry friction. The journal surface and bearing shell mainly experience dynamic pressure lubrication. When there is mixed lubrication, the thickness of the oil film is only a few micrometers, and the influence of asperity contact on the lubrication cannot be ignored. The equation used to solve for the lubrication is shown below [21]:

\[
\frac{\partial}{\partial x} \left[ \frac{h^3}{12 \eta} \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[ \frac{h^3}{12 \eta} \frac{\partial p}{\partial z} \right] = \left( \frac{u_1 - u_2}{2} \right) \frac{\partial}{\partial x} \left( \frac{\partial h}{\partial x} + \frac{\partial \partial \sigma}{\partial x} \right) + \frac{\partial (\partial h)}{\partial t}. \quad (3)
\]
2.3. Asperity Contact Model

Studies have shown that the lubrication state of the journal bearings in a diesel engine is a state of mixed lubrication between hydrodynamic lubrication and dry friction. The asperity contact model should be considered when studying the lubrication performance of a diesel engine journal bearing. The surface roughness distribution of the bearing journal and shell is approximately a Gauss distribution. The Greenwood/Tripp contact model was used to define the dry contact pressure of the bearing [22], as shown below in Equations (4) through (7):

\[ P_a = \frac{16 \sqrt{2\pi}}{15} (\alpha\bar{\eta})^2 \frac{\sigma^2}{\sqrt{\beta}} E^* f(h), \]  
\[ \frac{1}{E^*} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}, \]  
\[ f(h) = \begin{cases} 
4.4086 \times 10^{-5} (4 - \frac{h}{\sigma})^{6.804}, & \frac{h}{\sigma} < 4 \text{ (dry friction)}, \\
0, & \frac{h}{\sigma} < 4 \text{ dynamic pressure lubrication}, 
\end{cases} \]  
\[ \sigma = \sqrt{\sigma_1^2 + \sigma_2^2}. \]

2.4. Thermal Boundary Equation

The thermal boundary conditions are divided into oil film with journal thermal boundary conditions and oil film with bearing shell thermal boundary conditions, both of which undergo solid and liquid heat transfer processes. Consideration should be given not only to the heat conduction between the oil film and the bearing journal and shell, but also to the convective heat transfer between the oil film and the contact surface of the journal and the shell by the lubricating oil flow. Therefore, the Cauchy boundary condition \( i \) was adopted for the oil film on the journal and the oil film on the bearing shell. Their thermal boundary conditions are shown in Equations (8) and (9), respectively.

\[ a_j \frac{\Delta T}{\Delta n} + b_j(T - T_j) = 0 \]  
\[ a_s \frac{\Delta T}{\Delta n} + b_s(T - T_s) = 0 \]

2.5. Viscosity Temperature Characteristic Equation of Lubricating Oil

The viscosity temperature and pressure characteristics of the lubricating oil should be comprehensively considered in the calculation process because the dynamic viscosity of the lubricating oil changes according to temperature and pressure. Based on the Vogel model and Barus model [21], the viscosity relation of the lubricating oil was obtained as follows:

\[ \eta(p, T) = A e^{(\frac{p}{RT} + \Delta p)}. \]

3. Multibody Dynamic Model of a Journal Bearing Wear Test Bench

In order to reduce the number of computer calculations and time required, a modal reduction of the finite element model of the bench was performed, and the main nodes of the model and their freedom degrees were retained. In order to ensure the uniform transmission of force and moment between nodes, five Rbe2 elements were applied at the center node of the small end of the connecting rod, the large end of the connecting rod, and the support for the bearing journal. The bearing grid was evenly distributed into five layers. AVL-Excite software was used to build a flexible multibody dynamic model [23], as shown in Figure 1.
Figure 1. Multibody dynamics model: (a) 2D Excite model; (b) 3D model.

4. Sapphire Test Bench

The "Sapphire" test bench consists of mechanical, hydraulic, and auxiliary systems. Its schematic diagram is shown in Figure 2.

Figure 2. Schematic diagram of the "Sapphire" test bench.

The mechanical system consists of one test bearing and two support bearings. The outer diameter of the test bearing is 56.426 mm, and the bearing radius clearance is 0.06 mm. The hydraulic system consists of two parts. One is the loading system, which is used to produce a sine load on the test bearing, where the maximum load can reach 150 MPa. The other is the lubrication system. The lubricating oil is heated by the heat exchanger and used to lubricate the three bearings. The main shaft is connected to the motor through a coupling, and the maximum speed is 3000 r/min.

The test bench ascertained the thickness of the oil film, the thermoelectric signals, the bearing load, and the temperature of the shell back. The oil film thickness was measured by two eddy current displacement sensors (with a range of 1 mm and a resolution of 0.1 µm) to synthesize the axis orbit, which had a distribution of ±45°. The temperature of the bearing back was measured using a K-type
thermocouple. The load of the experimental bearing was measured using a strain gauge installed on the connecting rod and was the key input parameter of the simulation calculation. A contact voltage sensor was installed on the end to monitor the contact potential of the friction pair between the bearing journal and shell.

The test was performed according to the scheme illustrated in Table 1. The oil inlet temperature was kept at 70 °C (±1 °C), and the oil inlet pressure was kept at 5 bar (±0.1 bar) during the working time. The bearing lubrication state was determined based on the bearing shell back temperature. The oil film thickness sensor signals and bearing load signals were recorded after the pad back temperature was stable for two hours. Figure 3 shows the pressure curve obtained in the test process, which was also the input of the simulation calculation.

Table 1. Calculation and test scheme.

| Item                        | Load (MPa) | Rotating speed (r/min) |
|-----------------------------|------------|------------------------|
| Different rotating speed    | 0          | 1000                   |
|                             |            | 2000                   |
|                             |            | 3000                   |

| Item                        | Rotating speed (r/min) | Peak Load (Mpa) |
|-----------------------------|------------------------|-----------------|
| different load              | 3000                   | 20              |
|                             |                        | 40              |
|                             |                        | 60              |

Figure 3. Load curves under the different working conditions.

5. Analysis of Test and Calculation Results

When the test bench was in operation, the rotating action of the bearing journal subjected the oil film to the wall shear force, creating hydraulic pressure. Sufficiently large hydraulic pressure could lead to elastic deformation of the main shaft and the bearing pedestal, making the position of the elastic deformation uncertain, which would complicate measuring the minimum oil film thickness. In other words, the plane where the eddy current displacement sensors were installed was not the plane of the minimum oil film thickness. Therefore, the model was verified by analyzing the axis orbit of the measuring point plane and calculating the results in this study.
In order to evaluate the calculation accuracy, the track correlation coefficient was used to evaluate the curve correlation:

\[ r(X, Y) = \frac{\text{Cov}(X, Y)}{\sqrt{\text{Var}(X)\text{Var}(Y)}}, \]  
(11)

\[ X = \sqrt{x_1^2 + y_1^2}, \]  
(12)

\[ Y = \sqrt{x_2^2 + y_2^2}. \]  
(13)

5.1. Analysis of Measurement Data and Simulation Calculation Data at Different Speeds

Under the no-load condition, the test and calculation were carried out according to Table 1, and the results are shown in Figure 4.

![Figure 4](image_url)

Figure 4. The axis orbits of the tests and calculations at different speeds: (a) Measurement and calculation results at 1000 r/min; (b) Measurement and calculation results at 2000 r/min; (c) Measurement and calculation results at 3000 r/min; (d) Correlation coefficient of calculation and measurement results at different speeds.

Figure 4 illustrates that the calculation results approximated the test results. With an increase in rotation speed, the movement area of the axis orbit became smaller. Thus, the bearing operation was more stable. However, a considerable difference of 0.04–0.06 mm can be seen on the x axis. This is because this area corresponds to the crank angle of the oil return angle. The pressure changes greatly at this time, which causes the axis trajectory to oscillate, and the oil film that occurs in this area is the minimum for the whole working cycle.

It is thus shown that with an increase in the rotating speed in the test, the oscillation in the oil return angle decreased. The test bench became more stable, and the correlation coefficient of
the calculation and measurement increased. In particular, the test data of the working condition at 3000 r/min reflected the simulation calculation results.

5.2. Analysis of Measurement Data and Simulation Calculation Data at Different Loads

Under the working condition of 3000 r/min, the test and calculation were performed according to Table 1. The results are shown in Figure 5.

![Figure 5. Axis orbits of tests and simulation calculations at different loads: (a) Test and calculation results at 20 MPa; (b) Test and calculation results at 40 MPa; (c) Test and calculation results at 60 MPa; (d) Comparison of calculation and test results under different loads.](image)

Figure 5 demonstrates that the calculation results approximate the measurement results, and the correlation coefficient is high. According to Figure 5, by increasing the load, the movement area of the axis orbit becomes larger, and the horizontal movement amplitude increases significantly. Thus, the lubrication conditions of the bearing worsened.

6. Discussion

6.1. Comparison between the Minimum Oil Film Thickness and the Asperity Contact Percentage

While the engine is operating, the lubricating oil separates the bearing shell and journal to prevent dry solid friction. The lubricating oil negates the heat generated by the fluid friction work. When asperity contact occurs, the bearing friction work increases sharply and results in a temperature increase in the oil film, which causes a temperature increase and a viscosity decrease in the lubricating oil. Furthermore, the bearing capacity and the minimum oil film thickness decrease, which eventually leads to bearing wear failure. Therefore, asperity contact significantly influences the lubrication state of the bearing.
In Figure 6a, the calculation results show that asperity contact occurred between the bearing shell and journal within a crank angle range of 120°–350°, which corresponds to the minimum oil film thickness of less than 5 µm. With a decrease in the minimum oil film thickness, the asperity contact percentage increases, which leads to a sharp increase in contact pressure and a rapid increase in friction power consumption, as demonstrated in Figure 6b.

![Figure 6](image.png)

Figure 6. Calculation results at 3000 r/min with 60 MPa (peak load): (a) Minimum oil film thickness and asperity contact percentage; (b) Friction power consumption and asperity contact pressure.

As shown in Figure 7a, the total pressure is approximately 40 MPa before asperity contact occurs. According to Figure 7b, the total pressure is approximately 70 MPa when asperity contact occurs. As Figure 7c illustrates, the total pressure approaches 80 MPa when the asperity contact percentage reaches its maximum. It is thus shown that with the occurrence of asperity contact, the working condition of the bearing worsens. This indicates that the bearing is in the mixed lubrication state.

![Figure 7](image.png)

Figure 7. Specific analysis results for the test bearing simulation at 3000 r/min with 60 MPa (peak load). (a) Total pressure at 110.07 deg; (b) total pressure at 120.07 deg; and (c) total pressure at 250.07 deg.
Therefore, the asperity contact percentage indicates whether the bearing is in the mixed lubrication state. The occurrence of any asperity contact percentage should be avoided in the lubrication design as much as possible.

6.2. Comparison of Contact Potential and Asperity Contact Percentage

The contact voltage measured by the contact voltage sensor shows that the bearing shell and journal are in partial contact within the range $95^\circ - 349^\circ$ in Figure 8, which is consistent with the calculated asperity contact percentage curve. This indicates that the contact voltage can reflect the change in the asperity contact percentage, and the contact voltage can be used to monitor whether the bearing is in the mixed lubrication state.

![Figure 8](image.png)

**Figure 8.** Contact percentage compared with the contact voltage (3000 r/min and 60 MPa).

In order to evaluate the relationship between the asperity contact percentage and contact voltage, T1 (the duration angle with an asperity contact percentage greater than zero) and T2 (the duration angle with a contact voltage greater than 75 mv) were extracted.

Figure 9 illustrates that T1 and T2 have the same change trend and that the measured contact voltage follows the same rule as the calculated asperity contact percentage. This indicates that the duration angle of the contact voltage increases in conjunction with an increase in load, whereas with an increase in the rotating speed, the duration angle of the contact voltage first increases and then decreases. This is because the bearing crosses the inflection point of the Stribeck curve, leading to a change in the bearing lubrication state. The contact potential can be used to monitor whether asperity contact exists, which makes it possible to monitor the lubrication state of the bearing.

![Figure 9](image.png)

**Figure 9.** Eigenvalue comparison diagram: (a) Comparison of eigenvalues under different loads; (b) Comparison of eigenvalues under different rotating speeds.
7. Conclusions

A journal bearing test bench was the research object of this paper. A THL model of the journal bearing was established based on the Reynolds equation, the viscosity temperature characteristics, and the Greenwood/Tripp asperity theory. The correlation coefficient of the trajectory was used to evaluate the accuracy of the model. It was found that the correlation coefficient of the calculation results and the measurement results increased in relation to the rotating speed. The correlation exceeded 0.8 under a working condition of 3000 r/min at the different loads, verifying the model’s accuracy.

The relationship between the minimum oil film thickness and the asperity contact percentage was analyzed. The calculations showed that when the minimum thickness of the bearing oil film was less than 5 μm, asperity contact would occur in the bearing and cause an increase in the friction consumption of the bearing, leading to an increase in the oil film pressure. Thus, the lubrication condition of the bearing worsened. Therefore, the lubrication of the journal bearing was greatly affected by the asperity contact.

The measured contact potential followed the same rule as the calculated asperity contact percentage. Asperity contact occurred when the potential exceeded 75 mv. It increased in relation to an increase in load. The measured contact potential decreased, followed by an increase in rotational speed. Therefore, it is possible to monitor whether the bearing experiences asperity contact using the contact potential. Thus, the contact potential could be used to monitor the lubrication condition of a journal bearing.

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List of Nomenclature

| Symbol | Description |
|--------|-------------|
| M      | Mass matrix of a flexible body | E₂ |
| C      | Damping matrix | σ₁ |
| K      | Stiffness matrix | σ₂ |
| F      | Force | aₗ |
| u      | Generalized coordinates of the crankshaft and bearing | bₗ |
| θ      | Oil filling rate | Tₗ |
| h      | Oil film thickness | aₖ |
| ϕₓ     | Pressure flow factor of the X direction | b |
| ϕₙ     | Pressure flow factor of the Z direction | Tₙ |
| ϕₖ     | Shear flow factor | T |
| η      | Dynamic viscosity of the oil | n |
| Symbol | Description |
|--------|-------------|
| $u_1$  | Axial speeds of the journal A, B, C |
| $u_2$  | The axial speed of the journal and shell |
| $P_a$  | Dry contact pressure $E_x$ |
| $\beta$ | Average radius of curvature of the rough peak $E_Y$ |
| $E^*$  | Comprehensive elastic modulus $x_1$ |
| $v_1$  | Poisson ratio of the surface materials of the bearing journal $y_1$ |
| $v_2$  | Poisson ratio of the surface materials of the shell $x_2$ |
| $E_1$  | Elastic modulus value of the surface materials of the bearing journal $y_2$ |
| $i$    | Internal freedom degrees $r$ |
| PC     | Personal computer Rbe2 |
| PLC    | Programmable logic controller AVL |
| CFD    | Computational fluid dynamics MOFT |
| ACP    | Asperity contact percentage |
| C     | Viscosity temperature coefficients |
| $a$    | Viscosity pressure coefficient |
| $E$    | Oil film thickness in the horizontal direction |
| $E$    | Oil film thickness in the vertical direction |
| $\beta$ | Coordinate in the horizontal direction of the axis orbit by measurement |
| $\beta$ | Coordinate in the vertical direction of the axis orbit by measurement |
| $\beta$ | Coordinate in the horizontal direction of the axis orbit by calculation |
| $\beta$ | Coordinate in the vertical direction of the axis orbit by calculation |
| $\beta$ | Freedom degrees reserved |
| $\beta$ | Rigid bar element |
| $\beta$ | Approved vendor list |
| $\beta$ | Minimum oil film thickness |

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