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An experimental approach for investigating scuffing initiation due to overload cycles with a twin-disc test device

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ABSTRACT

This paper presents an investigation into the effect of unexpected overloading on scuffing initiation with a twin-disc test device. An existing twin-disc test device was modified to be suitable for applying overloading dynamically to the disc contact. Three test series including three pairs of discs in each series were subjected to combined normal force and rolling/sliding loading. First a base level for scuffing initiation was defined by increasing the normal load stepwise until failure, while keeping the rolling and sliding between the discs constant. In the subsequent test series, the overload cycles were applied at a significantly lower level in two different patterns leading to scuffing at both an earlier and later stage than was observed for the base level.

1. Introduction

Bevel gears in ship shafting lines often experience occasional overload cycles caused by propeller impacts on hard obstacles e.g. ice floes. The gear teeth have long been dimensioned against dynamic loads using load influence factors. However, occasional overload cycles are usually not noticed as a separate factor and therefore their role is neither visible nor accounted for in gear dimensioning.

The exact initiating mechanism of scuffing failure is not fully understood, although there appears to be a close relationship between a scuffing failure and lubricant film breakdown as noted in e.g. Ref. [1]. High surface pressure and sliding velocity with consequent frictional heating may affect the thickness of the lubrication film, whose decreasing viscosity leads to damaging asperity contact.

In addition, shear stresses play a key role in determining the film's thickness in an EHL contact between the rolling surfaces and the lubricant. However, a lubricant can only withstand a limited shear stress [2,3]. If this limit is exceeded and if the maximum allowable shear stress in the oil is more dependent on the oil's temperature than viscosity, a thinner oil film may result because less oil is dragged into the contact area [4].

Another factor to consider is that a transient load rise instantly reduces the lubricant film thickness at the same time increasing the friction coefficient [5], which increases the potential for asperity contacts. It has also been reported that a temperature rise in the lubricant film profile occurs under heavy loading, due to the compression and the viscous friction [6,7] increasing the risk for a scuffing failure.

The risk of scuffing is also known to be high during the initial settling of contacting surfaces often termed ‘running-in’, which is defined as the series of processes of wear rate and friction stabilisation for lubricated contacts [8]. In fact, the nature of scuffing is different than other gear failure modes in that it may be initiated after only a short period of severe load cycles in harsh conditions. Hence, rarely occurring occasional overload cycles may shorten the gear's life time significantly through a scuffing failure.

The flash temperature method is a widely-used approach to calculate the risk of scuffing. It was originally proposed by Blok [9] in 1937 and further enhanced by Jaeger [10] and Archard [11]. The basis of the theory is in the calculation of the maximum contact temperature t_c, which is an indication of the risk for a scuffing failure. This temperature is considered to be the sum of the bulk temperature t_b and the flash temperature $t_{f\,max}$ generated in the contact as follows:

$$t_c = t_b + t_{f\,\max} \tag{1}$$

One of the challenges with the above theory is the evaluation of the bulk temperature. In many cases this can be found by measurement, but an analytical approach based on loading is presented in Ref. [12]. Also recently Xue et al. proposed a method for predicting scuffing failure in spur gear pairs by means of a transient thermal solver [7]. This includes a temperature rise formula based on transient heat flux and a transient thermal elastohydrodynamic lubrication (TEHL) model, which takes into account the dynamic loads.

The experimental approach for finding the scuffing load-carrying capacity of a lubricant-material pair is traditionally defined by conducting FZG test procedures described in the standards [13]. Additionally, several scuffing studies have been performed with both

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the FZG device [1] and with bevel-gear-based test devices [14,15]. The focus has been on variating the used lubricant, the influence of the oil temperature and the applied loading, amount of the oil supplied and the effect of a coating on the contact surface. Despite the stability of the results delivered, the possibilities for investigating a wider range of parameters is somewhat limited; often requiring a new design for the gear wheels. In addition, several scuffing studies have been performed by variating the above mentioned parameters, but observing the behaviour of a piston ring and liner contact [16–18]. However, an obvious difference exists between the conditions in comparison to a gear teeth contact.

A new approach to the scuffing test has recently been presented [19]. This utilises a ball-on-disc test device with contra-rotating surfaces to limit the EHD films. This arrangement enables the determination of the boundary lubrication conditions over a wide range of sliding speeds.

With the twin-disc test device, the elliptical pressure distribution, which is seen between the gear teeth in e.g. spiral bevel gears, can be reproduced [20,21]. This is done through the precise definition of the discs' geometries. Furthermore, the structure of the device enables the actual loading of the contact to be highly adjustable. Over the years, the twin-disc test device has been used to study the behaviour of friction, pitting formation, wear prediction and the influence of surface roughness [22–25]. The effect of different surface conditions on scuffing initiation has been also investigated [26]. However, there has not been much research focusing on the effect of variable loading on scuffing failures, which is the main topic of this paper. The paper presents the first result of an investigation of the relationship between overload cycles and the initiation of scuffing using a twin-disc test device.

2.Test equipment

The tests are conducted with an existing in-house-built twin-disc test device as shown in Fig. 1. The original version of the device is described in detail in Ref. [27]. However, the device was updated in order to be able to apply dynamic loading on the contact. The hydraulic cylinder and its control valve, which place a variable normal force F_N on the discs, were renewed. In addition, the valve control system hardware has been updated and reprogrammed. The load frame with the test discs are illustrated in Fig. 2. The applied normal load is measured with a force transducer situated behind the normal force cylinder and its signal is used as an input for the valve controller.

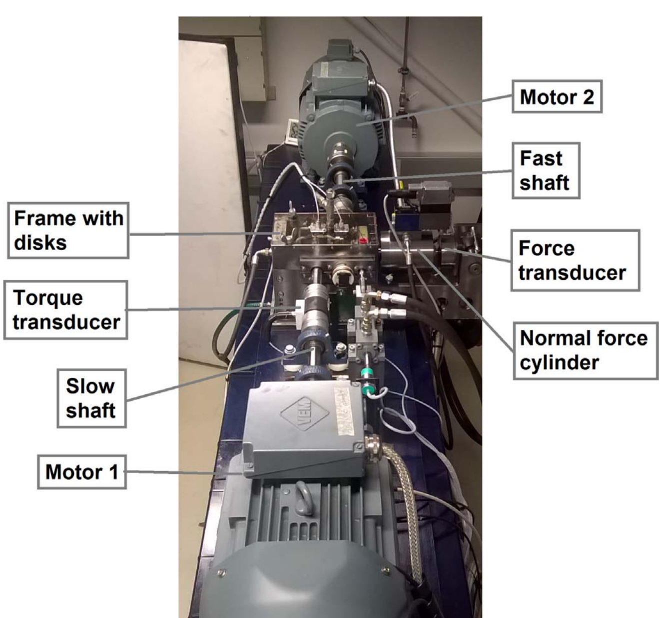

Fig. 1. The twin-disc test device.

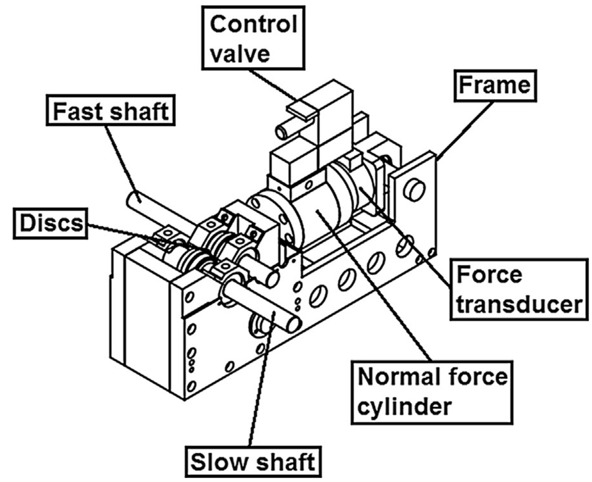

Fig. 2. The load frame.

The adjustable rotation speed, direction and slide-to-roll ratio are controlled with frequency converters, which drive the electric motors. These are joined to the shafts via couplings allowing minor misalignment between the axles.

A separate hydraulic unit provides lubricant for the disc contact injected from the inlet side. The same unit also lubricates the support rolling bearings inside the test device. Stepless adjustment of the lubricant flow is provided by means of a frequency-converter-driven pump motor and manual valves. The lubricant inlet temperature is automatically controlled by a separate microcontroller unit.

The test device is highly adjustable and automated enabling the possibility for unmanned operation. Some of the performance values are listed below:

- Rotation speed: Max 6000 RPM
- Normal load: Max 50 kN
- Lubricant temperature: 25…120 °C
- Lubricant flow rate: 0.5…20 l/min

2.1 Measurements

The measurement system collects data on the above-mentioned normal force, lubricant flow and inlet temperature, as well as the shaft rotation speeds and the torque on the slow shaft. In addition, the test disc bulk temperature on the fast shaft was recorded using a telemetry system and a thermocouple located approximately 3.5 mm underneath the surface at the disc contact.

The torque signal T_m measured by the torque transducer mounted on the slow shaft, can be separated into the torque T_r originating from the bearings and the shaft seals, and the torque T_c, which originates from the disc contact. These two values have to be separated in order to analyse the contact behaviour more precisely. The actual torque at the disc contact can be calculated with the following equation:

$$T_c = T_m \pm T_r \tag{2}$$

The T_r value is negative, when measuring the braking torque on the slower shaft and positive in the opposite situation.

Typical values for the torque T_r were found by measuring the shaft torque as a function of the normal force. This was done by rotating the shafts with the same speeds 175.8 RPM, resulting in a pure rolling condition. The operating temperature was 60 °C and three different disc pairs were used. A load-dependent equation was fitted to the measured values shown in Fig. 3:

$$T_r = 0.0801 \cdot \ln(F_N) - 0.0265 \tag{3}$$