Capacitances and lubricant film thicknesses of motor bearings under different operating conditions

E. Wittek, M. Kriese, H. Tischmacher, S. Gattermann, B. Ponick, G. Poll

Abstract – **Motor bearings in industrial converter-fed three-phase motors are mostly equipped with grease-lubricated roller bearings. It is known that under certain operating conditions bearing currents flow that can damage the bearings. When estimating the danger of damaging bearing currents it is important to look at the thickness and the capacitance of the lubricant film as well as the electrical parameters of the drive system. Calculating the thickness of the lubricant film for grease-lubricated bearings is problematical. This paper discusses how this quantity can be determined based on the bearing capacitance. After a brief introduction into the theory of lubricant film formation, the results of extensive tests are presented regarding the dependency of the bearing capacitance on temperature and speed.**

Index Terms--**Induction motor, bearings, bearing currents, lubrication, capacitance measurement, induction motor drives**

I. Nomenclature

A_{Hertz}	Size of the Hertzian contact area
a	Semi-axis perpendicular to the motion direction
b	Semi-axis parallel to the motion direction
C	Capacitance
C_{bearing}	Bearing capacitance
C_{Hertz}	Capacitance of the Hertzian contact area
C_{K}	Contact capacitance
C_{total}	Total capacitance of both bearings
d_{i}	Inner race diameter
E	Reduced modulus of elasticity
F_{ax}	Axial load
h_0	Central lubricant film thickness
h_{exp}	Film thickness by measurement
k_{C}	Correlation factor between C_{K} and C_{Hertz}
P	Contact load
R	Main radius of curvature
R_{x}	Main radius of curvature in the motion plane
R_{L}	Charging resistance
R_{P}	Parallel resistance
u	Rolling velocity
U_{L}	Charging voltage
Z	Number of loaded rolling elements per bearing
α_{P}	Pressure coefficient of the viscosity
Δt	Time difference between two voltage limits
ε_0	Electrical field constant
ε_{r}	Relative permittivity
η_0	Dynamic oil viscosity at contact entry point
$\kappa = a/b$	Ratio of the semi-axis contact ellipse
τ	Time constant

II. Introduction

BEARING CURRENTS are mainly classified as rotor-ground currents, circulating currents and EDM currents. Low-frequency circulating currents are caused by inhomogeneities and anisotropies of the stator lamination. This kind of current is relatively easy to control through constructive measures and is thus not discussed in this paper. In the case of an inverter supply, an additional high-frequency circulating current appears often at low speed when there is a mostly resistive path between the rotor and the stator. Both kinds of circulating currents - high and low frequency - can be prevented effectively by using coated bearings.

EDM currents are the consequence of the breakdown of the lubricant film capacitance. When a limit voltage is exceeded, the bearing voltage breaks down and discharges in the form of an arc. As a result of the high energy density, melting and vaporisation processes can occur in the bearing raceways. The consequences range from gray frosted bearing raceways up to corrugation, which essentially destroys the bearing so that it can no longer be used. A more detailed description is given in [5].

The objective of motor manufacturers is to predict critical operating parameters where EDM currents can occur in order to apply countermeasures. Ongoing work can be found in [4] and [5].

One key quantity to predict bearing currents and to calculate the electrical field strength is the thickness of the lubricant film.

III. Basics of Lubricant Film Formation and Measurement

Generally, it is easy to calculate the lubricant film thickness in oil-lubricated roller bearings. However, the build-up of a lubricant film in grease-lubricated roller bearings is subject to additional phenomena that are difficult to acquire.

The mechanisms on how a lubricant film is formed, determining the capacitance and converting it into the lubricant film thickness are described in the following.

A. Influencing quantities

Elastohydrodynamic lubricant films are formed in roller bearings between the raceways of the bearing rings and the rolling elements. In operation, these separate the surfaces and as a result decrease the friction and wear. The lubricant film thickness H_0 for oil lubrication that is obtained at the elliptical point of contact can be approximately calculated using the following formula:

$$H_0 = \frac{2{,}69 \cdot G^{0{,}49} \cdot U_0^{0{,}68}}{W_0^{0{,}067}} \cdot \left(1 - 0{,}61 \cdot e^{-0{,}73 \cdot \chi}\right) \quad (1)$$

with:

$$G = \alpha_p \cdot E \quad (2)$$

$$U_0 = \frac{\eta_0 \cdot u}{E \cdot R_x} \quad (3)$$

$$W_0 = \frac{P}{E \cdot R_x^2} \quad (4)$$

$$H_0 = \frac{h_0}{R} \quad (5)$$

On closer examination of (1) it can be seen that parameter U_0 has the highest influence on the lubricant film thickness. In turn, this is dependent on the rolling velocity u and the temperature-dependent dynamic viscosity η_0. An additional parameter that changes in operation is the contact load P; however, parameter W_0 has only a low influence on the lubricant film thickness under normal load conditions. For grease lubrication, an essential influencing quantity on the lubricant film thickness is additionally effective, which is the so-called starvation. This describes the lack of lubricant at the contact entry point in the manner that there is not sufficient grease that can flow back to this entry point between two rollover processes. Such an effect occurs at high speeds. When starvation occurs the lubricant film thickness is significantly reduced. The operating point at which starvation starts can hardly be predicted. For this reason the experimental determination of the lubricant film thickness is necessary. A characteristic curve is a function of the following parameters:

- Speed
- Temperature
- Bearing load

B. *Test rig*

The roller bearing tests to measure the capacitance are performed on an axial test rig (Fig. 1).

Fig. 1. Test rig with temperature control

In this case, two type 6008 deep-groove ball bearings are axially tensioned against each other using plate-type springs. As a result of the pure axial load, uniform contact relationships are ensured for all of the rolling elements. Using a process thermostat, the bearings can be kept at a constant temperature between 20°C and 80°C. The schematic test rig design with the electrical equivalent circuit is shown in Fig. 2. Each of the two test bearings is considered to be a parallel circuit comprising capacitance C_L and ohmic resistance R_P. Both vary depending on the lubricant film thickness and the operating state. The two test bearings are connected in parallel.

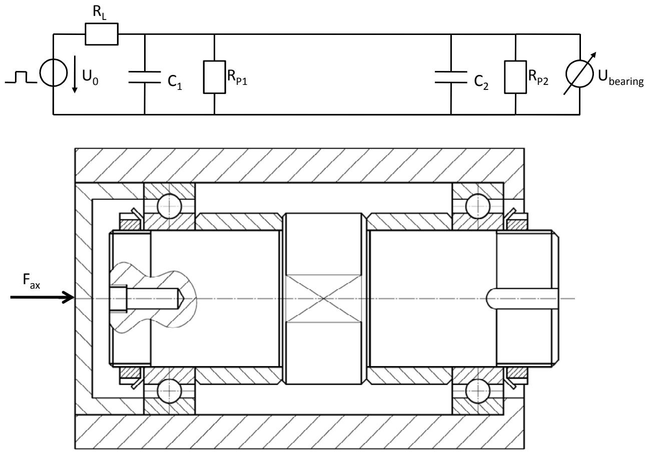

Fig. 2. Schematic of the test rig with the electrical equivalent circuit

C. *Determining the bearing capacitance*

The lubricant film thickness in the roller bearing is determined on a capacitive basis. A constant voltage measurement leads to the bearing capacitance which is represented by two capacitors connected in parallel between the test shaft and the housing. The charging process of these capacitors starts by applying a voltage step across the bearings. This process is realized through a defined charging resistance R_L. By measuring the capacitance voltage, a charging characteristic is defined using its time constant τ (Fig. 3).

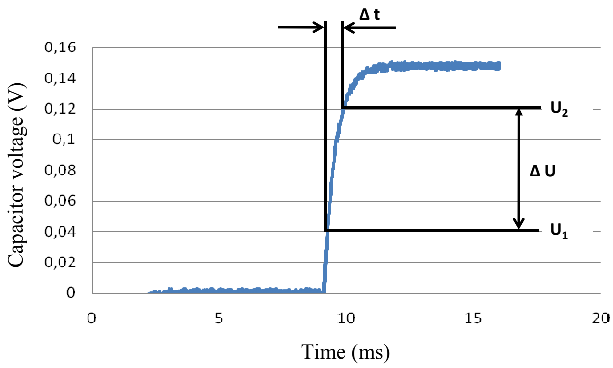

Fig. 3. Determining the time constant from the charging characteristic

The bearing capacitance is obtained according to the following equation by evaluating the time constant of this charging characteristic.

$$C = \tau / R_L \quad (6)$$

Determining τ based on the initial gradient of the charging characteristic turned out to be difficult. Because of this a time interval Δt is determined between the two defined voltage limits U_1 and U_2 during the charging process. Using the example shown in Fig. 3, the time constant is:

$$\tau = \frac{\Delta t}{\ln \dfrac{U_2 / U_0}{U_1 / U_0}} \tag{7}$$

As a result of the contact micro roughness between the rolling elements and the raceways for extremely thin lubricant films, there is an ohmic resistance R_P parallel to the bearing capacitance that changes. This is taken into account as follows when calculating the capacitance:

$$C = \tau \cdot \left(\frac{1}{R_L} + \frac{1}{R_P} \right) \tag{8}$$

D. Converting the capacitance to lubricant film thickness

The capacitance is an important electrical quantity to predict bearing currents. To calculate the electrical field strength in the lubricant gap, the lubricant film thickness is additionally required. The relationship between the bearing capacitance and lubricant film thickness is described in [1].

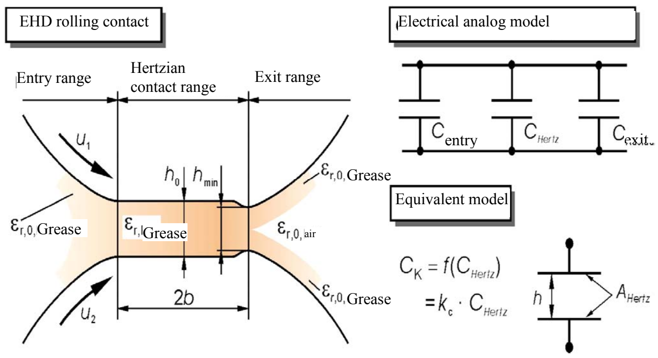

Fig. 4. Model to determine the capacitance in an EHD rolling contact according to Brüser [2]

The capacitance C_{Hertz} of the Hertzian contact range is the decisive capacitance to evaluate the lubricant condition. In fact the contact capacitance C_K is larger than C_{Hertz}. There is a correlation factor k_C between both quantities.

$$C_K = f(C_{Hertz}) = k_C \cdot C_{Hertz} \tag{9}$$

Based on his tests, which were performed under comparable test conditions, *Bartz* determined a factor $k_C = 3.5$ [3], which was used to evaluate the tests that have been carried-out. The capacitance C_{Hertz} can, to a good approximation, be interpreted as a plate-type capacitor and can be described using the following equation (10). A_{Hertz} can be calculated with the load in the contact and the adaption between the running traces and the rolling elements.

$$C_{Hertz} = \varepsilon_0 \cdot \varepsilon_r \frac{A_{Hertz}}{h_0} \tag{10}$$

The total capacitance C_{total} is measured over the two test bearings in a parallel circuit configuration. In turn, the test bearings have a parallel circuit comprising Z series circuits of contact capacitances $C_{K,i}$ and $C_{K,a}$ at the inner (index i) and outer rings (index a):

$$C_{total} = 2 \cdot C_{bearing} = 2 \cdot \sum_{1}^{Z} \frac{C_{K,i} \cdot C_{K,a}}{C_{K,i} + C_{K,a}}$$

$$= 2 \cdot Z \cdot k_C \cdot \varepsilon_0 \cdot \frac{\left(\varepsilon_{r,i} \cdot \dfrac{A_{Hertz,i}}{h_i} \right) \cdot \left(\varepsilon_{r,a} \cdot \dfrac{A_{Hertz,a}}{h_a} \right)}{\left(\varepsilon_{r,i} \cdot \dfrac{A_{Hertz,i}}{h_i} \right) + \left(\varepsilon_{r,a} \cdot \dfrac{A_{Hertz,a}}{h_a} \right)} \tag{11}$$

Generally, corrugation can occur at the inner bearing ring as well as at the outer ring. Both were seen at the bearing test stand in [5] with a radial load and also in field operation.

Simplifying, in this paper, it is assumed that the inner and outer ring have the same lubricant film thicknesses. The lubricant film thickness h_{exp} is calculated as follows:

$$h_{exp} = \frac{Z \cdot k_C \cdot \varepsilon_0 \cdot \varepsilon_r \cdot A_{Hertz}}{C_{ges}} \tag{12}$$

After determining the bearing capacitance and the lubricant film thickness on the test stand, the results must be transferred to different motor bearings. The results obtained in this way are used as input data for the bearing current model according to [4].

One dominating influence on the lubricant film thickness is the velocity u from the hydro-dynamic perspective. In a first approximation, the ratio of the inner ring raceway diameters $d_{i,2}/d_{i,1}$ can be taken as a measure for the ratio of the velocities u_1/u_2 in two bearings with different sizes but with the same bearing speed. This approach has the decisive advantage that the effects of different lubrication grades can be detected at the test rig - and transferred to the motor bearing to be calculated for comparable load condition. As a consequence, the real condition of the lubricant can be far better emulated in a motor bearing than in earlier work as [6], where only calculated lubricant film thicknesses were used.

The calculated lubricant film thicknesses from the test bearing were transferred to the required bearing size. This was used as basis to calculate the capacitance of the bearing being considered. In this case, the load direction has to be taken into consideration. While the bearing in the test stand was subject to a pure axial load, the load for the motor bearings in [4] is predominantly the weight of the rotor in the radial direction. In this particular case, only the rolling elements in the loading zone are in charge and therefore decisive for the bearing capacitance. Based on the previously calculated Hertzian contact area A_{Hertz} and the measured lubricant film thickness h_{exp}, the capacitance of the motor bearing can be calculated according to (9) and (10).

IV. Measurement Results

A. *Characteristic grease values*

To calculate the lubricant film thickness on the basis of the measured capacitance the dielectric coefficients ε_r of the lubricating greases were determined (Fig. 5). The values are in the range between $\varepsilon_r = 2.88$ and $\varepsilon_r = 4.76$. There is no significant temperature dependency.

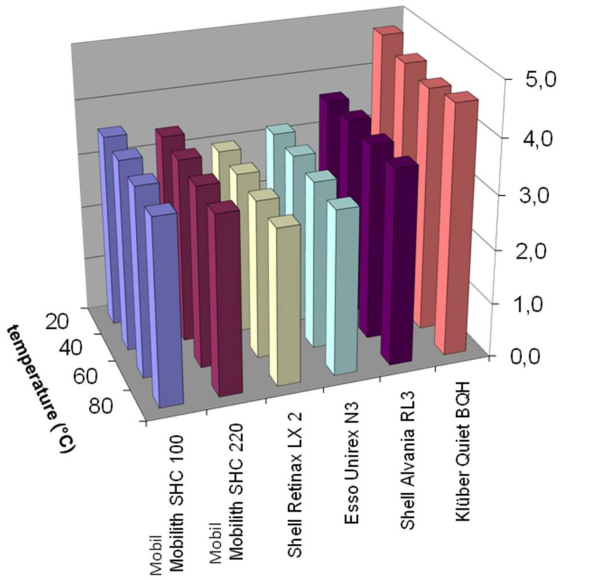

Fig. 5. Experimentally determined dielectric coefficients of the test grease

B. *Bearing capacitances*

The bearing capacitances of both test bearings in parallel were determined at various speeds and temperatures on the test rig (Fig. 1). For an axial load of $F_{ax} = 100$ N this values are shown in Fig. 6.

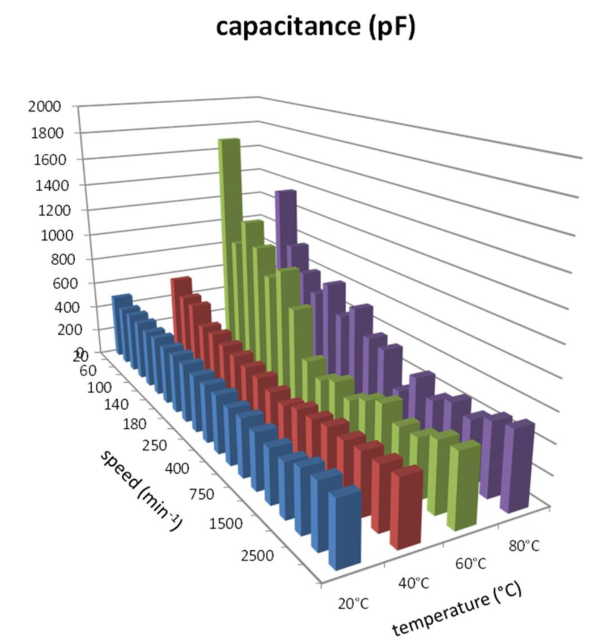

Fig. 6. Bearing capacitances at $F_{ax} = 100$ N (both bearings in parallel)

There is no significant speed influence on the capacitance in the temperature range from 20°C to 40°C. On the other hand, at 60°C and 80°C, the capacitance significantly increases in the low speed range. The reason for this is a thin lubricant film thickness that prevails at these temperatures. Depending on the lubricant film thickness, different levels of metallic contact between the roughnesses cusps of the surfaces in contact with the rolling elements are established. This effect causes an ohmic transition resistance R_P across the bearings, which changes depending on the operating parameters. A lower lubricant film thickness, leads to a higher metallic contact area and therefore to a lower resistance. Decreasing this resistance also decreases the charging voltage across the test bearings. The charging characteristics at four speeds are shown as example in Fig. 7. At $n = 20$ min^{-1}, no voltage is established. As the speed increases, the final voltage increases and the charging characteristics are significantly smoother.

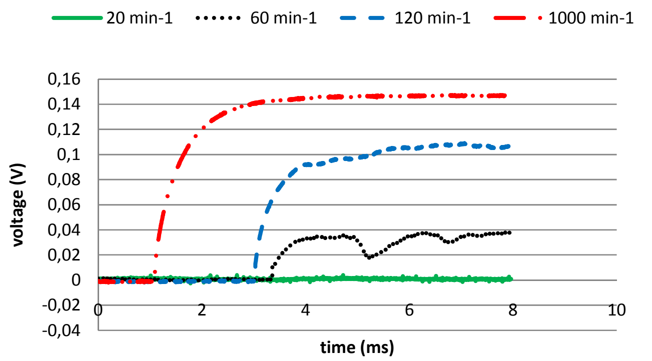

Fig. 7. Charging characteristics for various speeds

The final values of the bearing voltage for a charging voltage of $U_L = 0.15$ V are shown in Fig. 8. Analogous to the bearing capacitances, it is obvious that no separating lubricant film is established in the range of low speeds and high temperatures.

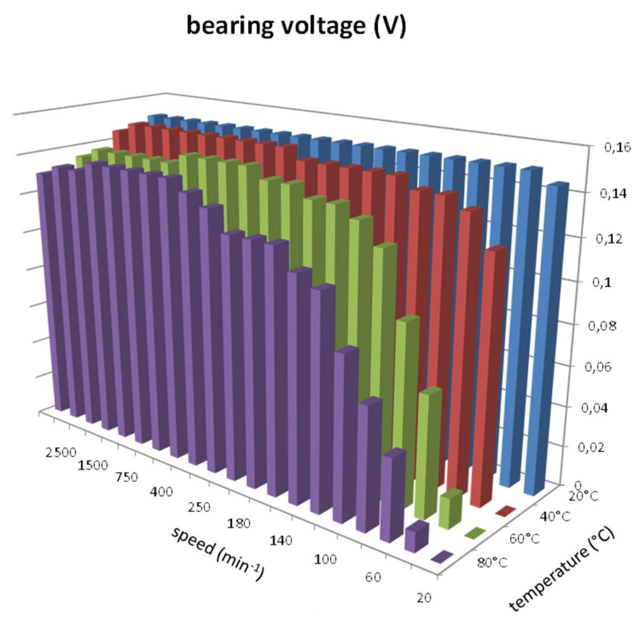

Fig. 8. Final bearing voltage values at $F_{ax} = 100$ N

According to (12), the lubricant film thickness is calculated based on the bearing capacitances from Fig. 6. The results are shown in Fig. 9. These values can be subsequently scaled to different bearing sizes so that they can be used in the simulation model for bearing currents [4].

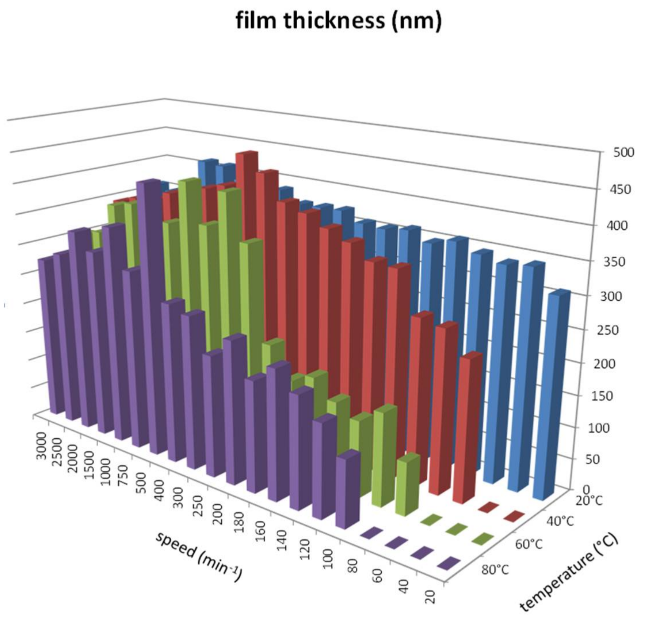

Fig. 9. Experimentally determined lubricant film thicknesses, h_{exp} at F_{ax} = 100 N

In the range of low speeds and high temperatures a supporting film of lubricant has still not been established in the bearing. Further, at high temperatures above speeds $n \approx 1000\ \mathrm{min}^{-1}$, the effect of starvation begins. The consequence is that the lubricant film thicknesses decrease. Both of these influences have still not been taken into account in the analytical calculation in [6].

V. Discussion

In this paper, it has been shown how the bearing capacitance of deep-groove ball bearings depends on speed and temperature carried out at tests on a roller bearing test rig. The lubricant film thickness has then been calculated from this data. Further, a way has been shown how to transfer the results gained from this test rig to different bearing sizes. The results were used as input quantity for the simulation model for bearing currents developed in [4]. In addition to the simulation, bearing current measurements were carried-out in [4] on a three-phase converter-fed motor. The comparison between simulated and measured bearing currents shows a significant improvement. The simulation results based on the capacitances and lubricant film thicknesses, which are determined in this paper (Fig. 10), show a good correlation with the measured values. In contrast to this, the comparison with the estimated values shows a poorer correlation (Fig. 11). The initial peak of the EDM current in Fig. 10 is closer to the measured value than in Fig. 11. Even this initial peak with the highest amplitude is decisive for bearing damage due to electric currents.

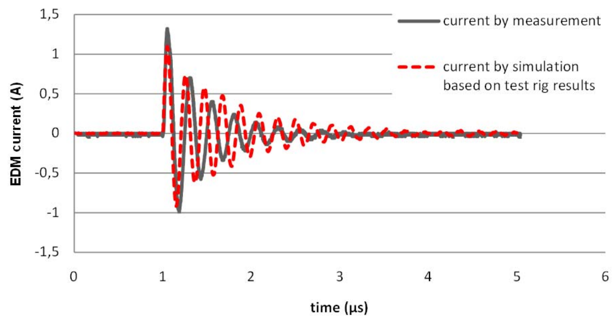

Fig. 10. EDM currents measured – simulated with calculated values

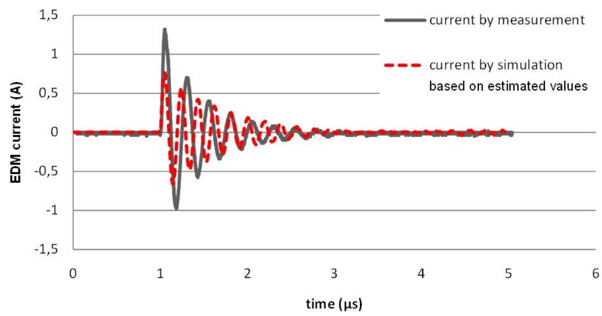

Fig. 11. EDM currents measured – simulated with estimated values

As a consequence, this work significantly improves the calculation of bearing currents. As a consequence, dangerous operating conditions can be identified more exactly by simulation.

VI. Outlook

A converter-fed motor in field operation is usually loaded by dynamic forces, which cause the bearing elements to vibrate. Due to this the lubricant film thickness changes dynamically. Because of this, further investigations have to take this into account. For ongoing work an appropriate test rig (Fig. 12) has already been designed, assembled and is presently undergoing trials.

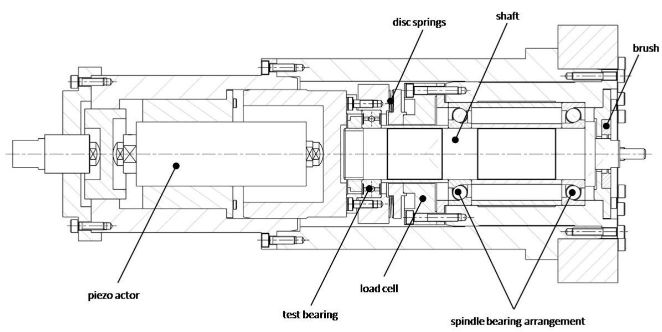

Fig. 12: Dynamic test rig

In addition, tests at different static loads are provided. This will allow an even more detailed set of characteristic curves to be obtained for the simulation model in [4] for a wide range of mechanical lubricant gap loads.

VII. ACKNOWLEDGMENT

The results presented in this paper were obtained in a close and intensive cooperation with Siemens AG, Large Drives in Nuremberg and the Institute for Drive Systems and Power Electronics of the Leibniz University Hanover.

VIII. REFERENCES

Dissertations:

[1] H. Baly, "Friction of grease-lubricated ball bearings" (in German), Dissertation. University of Hanover: Institute for Machine Elements, Engineering Design and Tribology (IMKT), 2005

[2] P. Brüser, "Investigations of the elastohydrodynamic film thickness in elliptical hertzian contacts (in German)," Dissertation. University of Braunschweig. 1972"

[3] M. Bartz, "Lubricant film formation in grease lubricated high speed spindle bearings (in German)," Dissertation. University of Hanover: Institute for Machine Elements, Engineering Design and Safety Engineering (IMKS), 1996

Papers Presented at Conferences (Unpublished):

[4] M. Kriese, E. Wittek, H. Tischmacher, S. Gattermann, G. Poll, B. Ponick, "Prediction of Motor Bearing Currents for Converter Operation," ICEM 2010

[5] H. Tischmacher, S. Gattermann, "Bearing currents in converter operation," ICEM 2010

Papers from Conference Proceedings (Published):

[6] O. Magdun, A. Binder, "Calculation of Roller and Ball Bearing Capacitances and Prediction of EDM Currents," IEEE 2009, Preprint of IECON 2009 Proceedings, pp. 1047 - 1052

IX. BIOGRAPHIES

Eike Christian Wittek was born in Hameln, Germany, in December 1981, studied mechanical engineering at the Leibniz University of Hanover, and graduated with a diploma degree in 2008. Since 2008 he is active in the field of bearing fatigue and grease lubrication at the Leibniz University of Hannover as research associate at the Institute for Machine Elements, Engineering Design and Tribology. He focuses on lubricant film analysis both in bearing test rigs and on a model ball-disc test rig.

Michael Kriese was born in Hanover, Germany, in October 1982, studied electrical engineering at the University of Hannover, and graduated with a diploma degree in 2007. Since 2008 he is active in the field of electrical drives at the University of Hanover as research associate at the Institute for Drive Systems and Power Electronics. He focuses on the investigation of bearing currents in inverter-fed induction machines. He does also theoretical, constructive and experimental work in developing electrical motors

Hans Tischmacher was born in Wolfsburg, Germany, in August 1966. He received the Dipl.-Eng. degree in electrical drives from the Technical University of Braunschweig, Germany, in 1994. He is currently with Siemens AG, Industry Sector, Drive Technologies Division, Large Drives, Nuremberg, Germany. He has 15 years experience in the development and testing of electrical motors and power electronic converters. He holds several patents in this field. His scientific interests include electric drive systems, power electronics, electrical machines especially the effects of the power electronic converters on electrical machines, e.g. bearing currents, noise emission, insulation stress etc.

Sven Gattermann was born in Stadtoldendorf, Germany, in November 1972. He received the Dipl.-Eng. degree in materials science from the Technical University of Clausthal, Germany, in 2001. Since 2004 he has been with Siemens AG, Industry Sector, Drive Technologies Division, Large Drives, Nuremberg, Germany. One of his favorite fields of interests is the investigation of the effects of bearing currents in electric drives.

Bernd Ponick was born in Großburgwedel, Germany, in May 1964. He received his Dipl.-Ing. degree in electrical power engineering from the University of Hanover in 1990 and his Dr.-Ing. degree for a thesis on electrical machines in 1994. After 9 years with the Large Drives Division of Siemens as design engineer for large variable speed motors, head of electrical design and Technical Director of Siemens Dynamowerk Berlin, he is since 2003 full professor for electrical machines and drive systems at Leibniz Universität Hannover. His main research activities are calculation and simulation methods for electrical machines, prediction of and measures against important parasitic effects such as magnetic noise, additional losses or bearing currents, and new applications for electric machines, e.g. for electric and hybrid vehicles.

Gerhard Poll was born in Schweinfurt, Germany, in February 1954. He received his Dipl.-Ing. degree in mechanical engineering from the University of Aachen in 1977 and his Dr.-Ing. degree for a thesis on mechanical wear at the Institute for Railway Vehicles and materials handling of the RWTH Aachen in 1983. After twelve years with SKF as team leader in the field of drive systems, project leader in the field of tribology and head of research and product development of the CR-Industries (SKF-group) in Elgin (Illinois), he is since 1996 Professor (C4) for Machine elements and Engineering design (successor of Prof. Paland) and head of the Institute for Machine Elements, Engineering Design and Tribology of the "Leibniz Universität Hannover." His main research activities are rolling bearings and lubricants, dynamic seals, synchronizers, continuously variable transmissions (CVT) and railway vehicles.