Thrust Cone Bearings Provide Increased Efficiency for Helical Gear Units at Moderate Speed Levels

Indications for possible energy saving potential in an expanded field of application

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Introduction

Thrust cone bearings are an elegant option to handle the axial forces generated by the torque transmission in helical-toothed gear stages. They have proven as an efficient and reliable bearing concept for integrally geared compressors but are nearly unknown in other fields of gearbox engineering. The presented investigations consider three aspects which appear relevant to extend the field of possible applications for thrust cones towards gearboxes constructed with roller bearings. Based on simulations and experiments design parameters were identified, which enable a significant reduction of the necessary velocity for full film lubrication. For a single stage test gearbox noticeable increases in efficiency were achieved by replacing tapered roller bearings with a combination of thrust cone and ball bearings, especially during partially loaded operation. The resistance to wear and the determination of limits for the bearable loads under mixed friction conditions for various thrust cone design configurations are investigated in a third test series. It appears that the few limit values known so far might be exceeded significantly for future applications.

Helical gears are a common solution to reduce noise and increase the transmittable torque in the construction of gearboxes. Unfortunately the pair of contact forces between the meshing tooth flanks is not perpendicular to the axis of rotation of the gear shafts, due to the helix angle. Transmitting torque between pinion and gear leads to an axial force component, which usually has to be transferred through the gears, the shafts and axial bearings into the housing of the gearbox (Fig. 1, left) or offset by the use of double-helical gears.

Thrust cone bearing concept. Figure 1 (right) presents an alternative-bearing concept, i.e. — the "thrust cone" bearing. Conical rims — denoted as thrust cones — are attached to

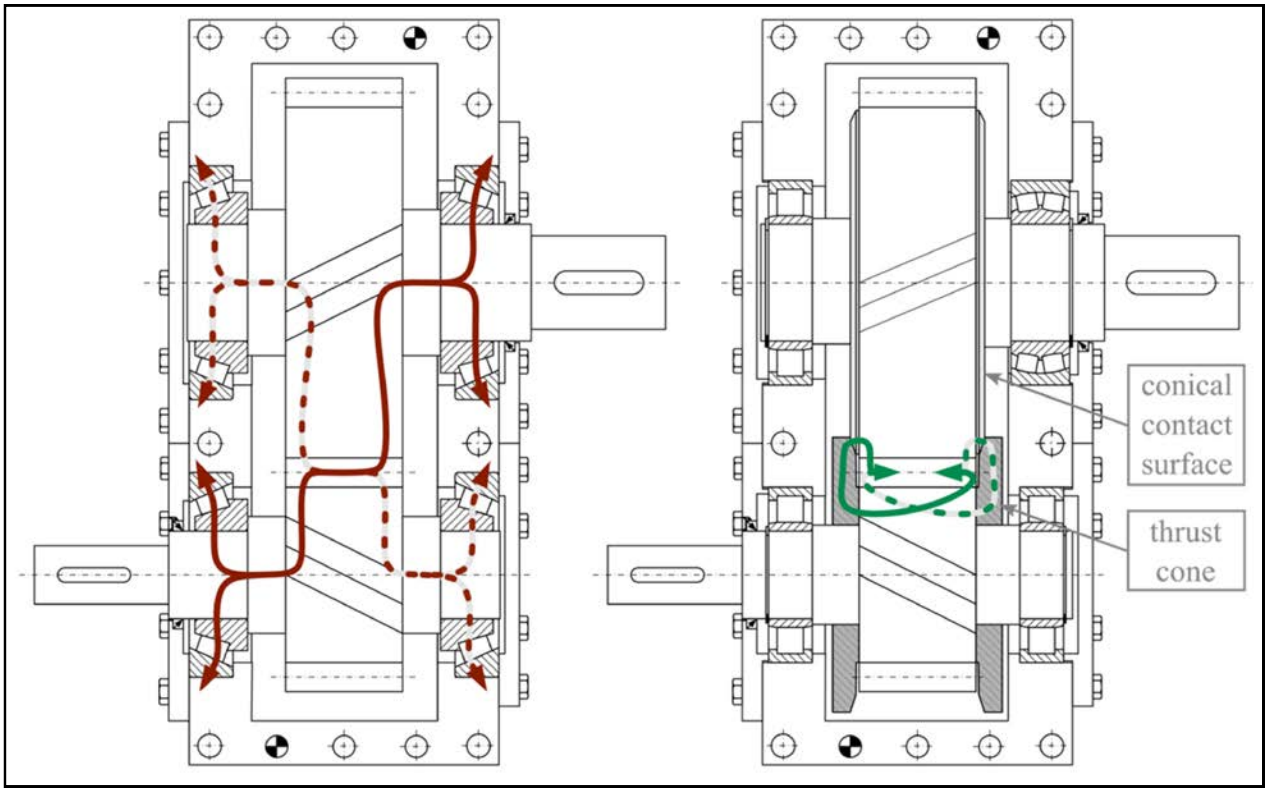

Figure 1 Axial force components in helical gear boxes: with conventional bearing concept (left) and thrust cone bearing setup (right) (Ref. 6).

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both sides of the pinion and flank the opposing wheel. Their conical shape and the contact surfaces ground to the sides of the wheel lead to a narrowing gap in the overlapping area. Lubrication fluid, sticking to the surfaces, is transferred into this gap and generates a hydrodynamic pressure film that separates thrust cones and the contact surfaces on the wheel. The axial force generated on the pinion tooth is transferred through the thrust cone and the fluid film onto the conical contact surface of the wheel; here it meets the axial force component created on the wheel's tooth. Since both force components obtain the same value — but with opposing directions — they "cancel" each other and no axial force is transferred to the shafts or the housing (Fig. 1, right). This load reduction enables a lighter construction for the machine components; the pinion shaft can be designed without an axial bearing, while the axial bearing on the wheel shaft operates only as positioning, i.e. — without load from the helical gear pair.

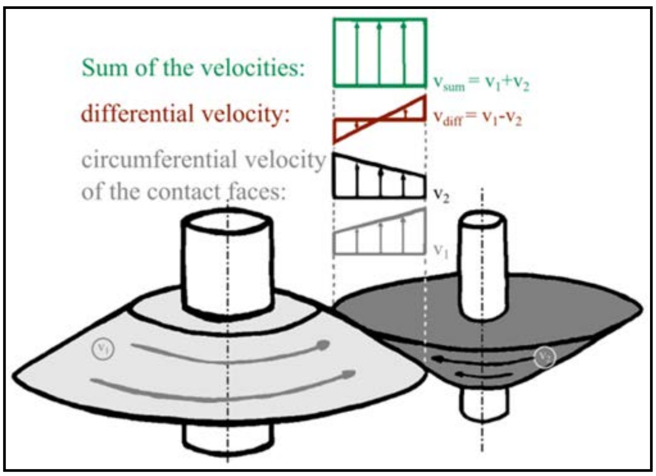

Figure 2 Contact velocities for two rotating cones: close to the pitch point favorable cinematic conditions for hydrodynamic lubrication occur (low differential velocity and high sum of surface velocities) (Ref. 6).

The frictional losses in sliding bearings rise proportionally to the square of the differential velocity between the contacting surfaces. In a typical axial sliding bearing, the differential velocity equals the circumferential velocity of the running surface. In a thrust cone bearing, a lower differential velocity occurs between the contacting surfaces since their contact region is located close to the pitch point of the gear pair (Fig. 2).

Aspects concerning an expanded field of application. Currently the main application for thrust cone bearings is in the field of integrally geared compressors (IGCs), which are characterized by a very high rotational speed on the pinion shaft (more than 10,000 rpm) and a nearly constant torque load at their point of operation. Since the first patent in 1924 (Ref. 2) the thrust cone bearing has proven to be an appropriate, alternative bearing concept for helical gear pairs. (Langer, Ref. 4) stated that a reduction of bearing-related frictional losses for thrust cone concepts to 10–20% compared to classical concepts based on tilting pads. Nevertheless, in special operational situations, such as emergency shutdown sequences, difficulties occur if the hydrodynamic carrying capacity of the lubricant film is not sufficient to separate the contact surfaces. Apart from IGCs the application of thrust cones in modern engineering is limited and nearly negligible.

To improve the reliability of thrust cone bearings and widen the field of possible applications, the following three main topics are within the scope of a research project currently conducted at the Institute for Mechanical Engineering (IMW):

Low-speed, full-film lubrication. Aiming for higher gearbox efficiency, it is a crucial condition that the friction losses in the thrust cone bearing are lower than the reduction of losses in the bearings of the housing. Since the friction coefficient under full-film lubrication is significantly lower than in the mixed-friction regime, the importance of the required reduction of the transition velocity becomes obvious.

Proof of efficiency. In ICGs, thrust cone bearings are usually combined with journal bearings where the axial load compensation supersedes one axial bearing. In gearboxes based on roller bearings, the benefit in efficiency is gained by a change of bearing types. Instead of tapered roller bearings, which are distinguished by their high axial load-carrying capacity but generate a relatively high-energy consumption, more efficient ball bearings might be used if axial loads are compensated by thrust cones. To prove that the suggested change in bearing types outweighs the additional friction in the thrust cone contact, a comparative examination of gearbox efficiency for both concepts is presented.

Determination of bearable load under mixed-friction conditions. Even though the transition velocity is reduced, there will remain situations (starting or breaking maneuvers, for example) with insufficient speed for full-film lubrication. A successful thrust cone design must safely withstand these mixed-friction situations during the product's life cycle. Unfortunately, at this writing there is as yet no available information on bearable loads for thrust cone bearings in open literature. To enable a wider use of thrust cone bearings as a resource-efficient machine element in gearboxes, a description of possible design influences on limiting load values under mixed-friction conditions is required.

Reduction of the Required Velocity for Full-Film Lubrication

Since full-film lubrication achieves efficient operation and nearly eliminates wear effects on the contacting surfaces, research activities were initially focused on influences on the transition velocity. In general the fluid film in a thrust cone bearing increases with the rotational speed and reduces with additional load — but for a certain operation point (combination of load and speed) various thrust cone designs generate different fluid film thicknesses. To predict the effect of design parameters on film thickness, a hydrodynamic fluid film simulation was developed. The algorithm — inspired by the work of (Barragan de Ling, Ref. 1) — allows solving the Reynolds differential equation for a thrust cone bearing, calculates the hydrodynamic pressure distribution, and determines the minimum gap size between the elastically deformed contact surfaces.

Figure 3 illustrates some design variations for thrust cone bearings, influencing the transition behavior. Besides variations of cone angle and slip value (depending on the distance between pitch point and contact surface), macroscopic shape variations for the running surface geometry were within the scope of our examinations.

To validate the predictions made by simulation, experiments on the thrust cone test bench (Ref. 7) were carried out. For various thrust cone specimens, representing the shape designs used in the simulations earlier, the transition velocities at different load steps were determined. In the full-film lubrication region, contact surfaces are completely separated by the lubrication fluid. Due to its low conductivity, high electric resistance can be observed between the specimens. If the velocity is reduced the film thickness shrinks for a constant axial load. A drastic drop of the electrical resistance is notable when first metallic contact occurs between the peaks of the rough surfaces. The current combination of load and velocity is regarded as a transition point to the mixed-friction regime. Some of the determined transition points are plotted (Fig. 4) as an example of the experimental work. The data illustrates borderlines between mixed- friction and full lubrication for flat thrust cones with different inclination angles and slip values. Since low- transition velocities are desired for early full-film lubrications, optimal design configurations can be found in the lower region (Fig. 4). In accordance with simulations, low slip values and cone angles lead to an earlier separation of contact surfaces; nevertheless, a cone angle greater than zero is necessary to generate even the slightest hydrodynamic lubrication.

Figure 4 shows that a small geometric variation in the bearing design can have a great influence on transition behavior. The transition velocity for the best design (inclination angle: 0.5°; slippage: 10%) is about 50% lower than for the worst in the Fig. (inclination angle: 1.0°; slippage: 20%). Assuming constant acceleration during a starting maneuver, the distance to be run under mixed-friction conditions is reduced by the factor 4.

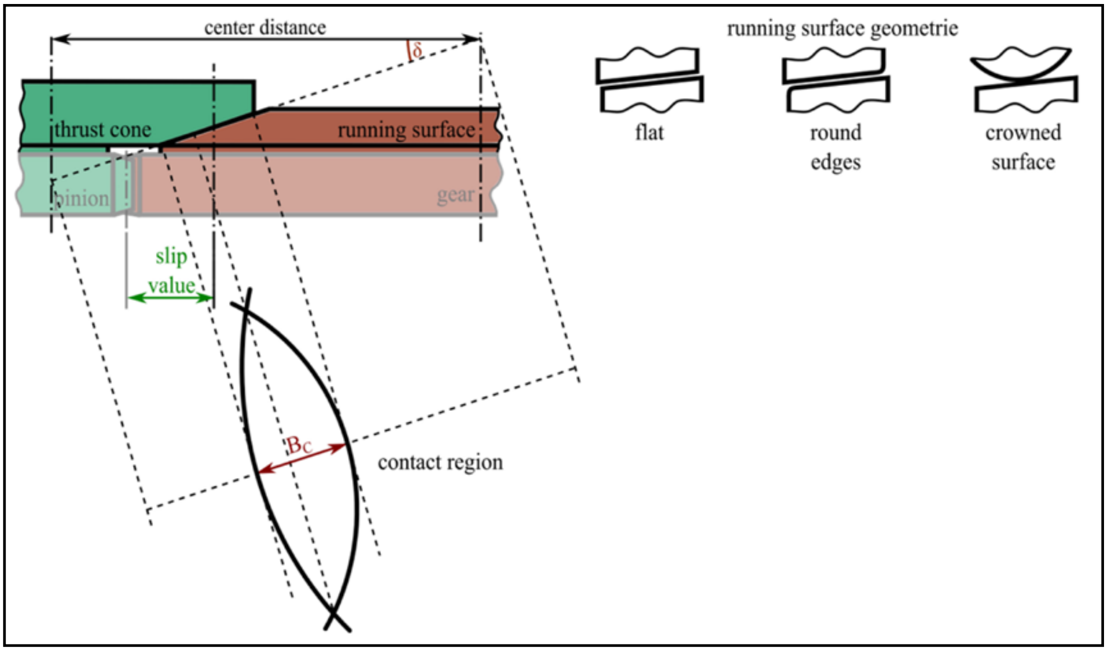

Figure 3 Design variations for thrust cone bearings (Ref. 6).

What's more, it becomes obvious that the transitions observed in the test occur within a speed range that is significantly lower than the operational velocities for thrust cone bearings in ICGs (> 100 m/s), and even relevant for gearboxes running with roller bearings.

Gearbox Efficiency in Dependency of the Bearing Concept

To investigate the thrust cones' ability to reduce friction losses in gearboxes based on roller bearings, a test gearbox (Fig. 5) has been set up. The gearbox is driven by an inverter-fed, asynchronous machine; a torque load can be applied by an adjustable mechanical break coupled to the output shaft. Input and output torque and the rotational speed are

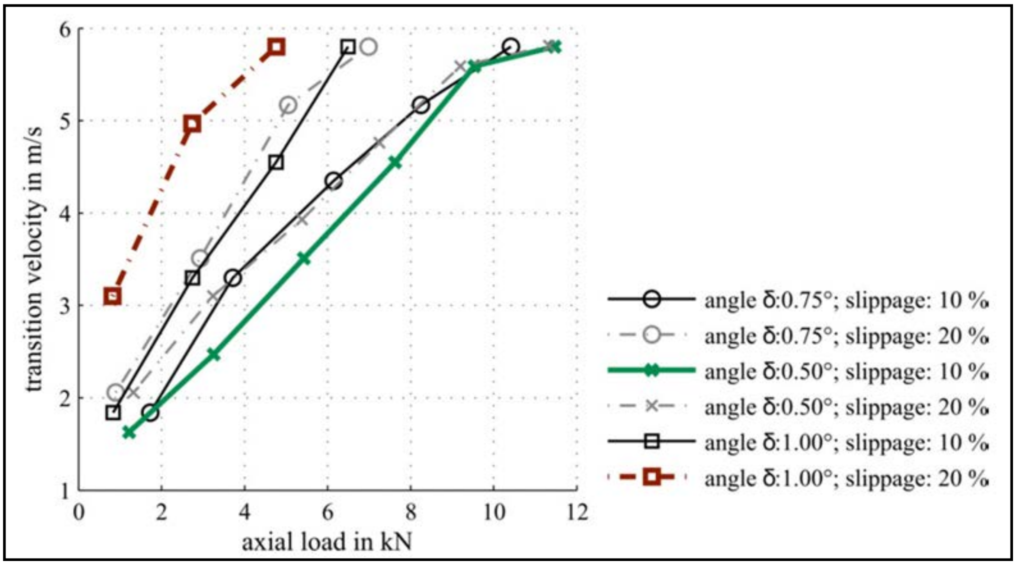

Figure 4 "Border lines" for the transition between mixed friction and full film lubrication for different flat thrust cones based on the experimental data.

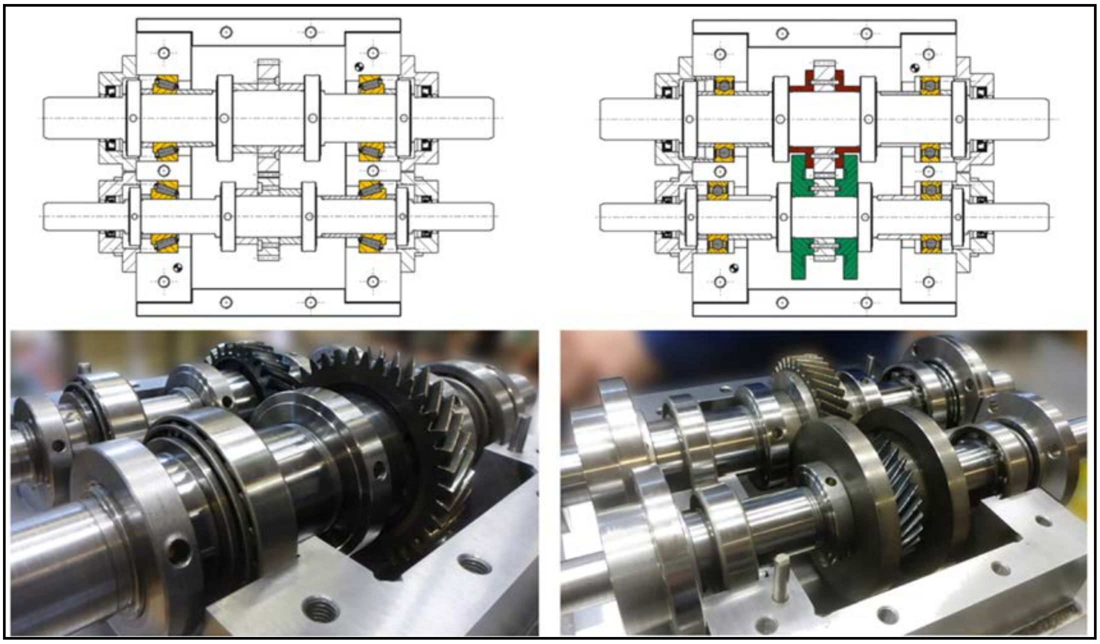

Figure 5 Test gearbox — equipable with tapered roller bearings (left side) or a combination of thrust cone and ball bearings (right side) as concept drawings (top) and assembled (bottom) (Ref. 6).

recorded via measurement shafts with integrated rotary encoders. Both power input and output can be calculated from the captured data. The difference between input and output is regarded as system losses of the gearbox and the reciprocal quotient as its efficiency. The system losses are the sum of losses caused by several gearbox elements. To evaluate the influence of the bearing concept on the system losses, the gearbox can be equipped with either a set of tapered roller bearings in O-arrangement, or with a combination of thrust cone and ball bearings. All other components contributing to the system losses — gearing, seals, oil level, etc. — are kept constant during the tests.

By varying torque load and drive speed, characteristic efficiency maps were determined for both bearing concepts (Ref. 6). As an example of the results gained in the comparative tests, efficiency values derived from recorded data at several load levels for drive speeds of 1,200 rpm and 2,400 rpm are plotted over the power input (Fig. 6). Even though the recorded input power values are not exactly the same for both bearing concepts, a benefit of the thrust cone solution is clearly recognizable. Interestingly, the difference between the power losses of the two bearing concepts is hardly varying with the input power. That indicates a nearly load-independent loss mechanism being responsible for the gap between the compared power loss characteristics. Such a mechanism might be explained by drag losses in the bearings, which are highly dependent on the roller geometry and nearly independent from load.

For all configurations in Figure 6 we can see that the efficiency increases with the transmitted power, since load-independent friction components gain higher weight only in partially loaded operation regions. The highest differences between the bearing concepts occur for both velocities in the lower loaded region. Assuming that other loss mechanisms, e.g. — the friction in the tooth contact — raise with the load, the importance of the nearly static difference between the bearing concepts due to the drag losses is shrinking. Nevertheless the use of thrust cone bearings leads to an average rise in efficiency for the plotted graphs of three to four percent; in the partially loaded regions ascents of about seven percent can be observed.

Bearable Load Under Mixed-Friction Conditions

Despite the achieved reduction of the transition velocity, there remains a speed range in which mixed friction occurs. A functional gearbox design demands thrust cone bearings that reliably endure the estimated mixed -friction phases in the product's life cycle.

Since, as mentioned, the availability of data on acceptable loads for thrust cone bearings in open literature is limited, a

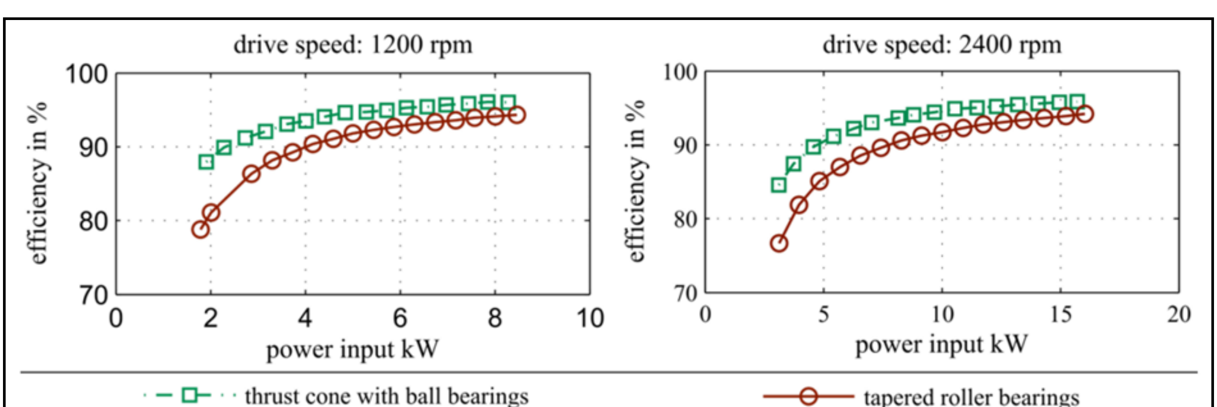

Figure 6 Comparison of the efficiency for a gearbox with tapered roller bearings and a gearbox with a combination of thrust cones and ball.

second test series shall provide further insight in bearable loads and influencing parameters. The tests, conducted as well on the thrust cone test bench, are carried out with a constant rotational velocity that belongs to the mixed-friction speed range for all specimens in the transition velocity tests. Starting with a minimum load of 2 kN, the axial force applied to the probes is raised every 50,000 turns by 2 kN until the destruction of the running surfaces is detected or the surfaces withstand the maximum load of 100 kN and passed 2.5 million turns. During the tests cone angle, slip value and contact width are varied for specimens made of 30CrNiMo8 and 34CrMo4. Nitriding of the running surfaces is another parameter in the tests. A high increase of axial accelerations or torque values indicates the surface destruction. To prevent fluid damages the temperature on the contact surface is monitored; if it reaches a critical value, the tests are interrupted for cooling; high temperature-time gradients might indicate a surface destruction as well.

Even though the test series is not yet finished, first observations are nevertheless available. Most of the tested surfaces did not pass the full test cycle without failure. Table 1 presents the maximum load reached in the experiments for 16 specimens arranged as fractional factorial design. Additionally, the equivalent Hertzian pressure is calculated for each test.

The equivalent Hertzian pressure is a common tool for the dimensioning of thrust cone bearings (Refs. 3 and 5). Therefore the flat, conical running surfaces are assumed to be a Hertzian contact of two parallel cylinders with a length that equals the contact width between the thrust cone and the surface on the wheel. The radii of the cylinders are the perpendicular distance between the center of the contact zone and the rotation axis of their shafts.

The only design value for the bearable load of flat thrust cones known from open literature is 50 MPa (Ref. 3). The 16 tests presented clearly exceeded that threshold. Only in one case surface failure occurred at a pressure below 80 MPa; in all other cases the surfaces withstood a load higher than 150 MPa and failed at values three to six times higher than the value known from literature. In four cases the specimen endured the maximum test load of 100 kN without failure. To illustrate the influence of the design parameters under investigation, a main effects diagram is presented (Fig. 7).

As expected, low slippage values and the nitriding of running surfaces have a clear, positive effect on the bearable equivalent pressure. The choice of material and the contact width seem to be less important for the contact pressure. Nevertheless the contact width is clearly connected to the absolute bearable load values, which can be seen by comparing the average axial force for both contact sizes (62.0 kN for the 10 mm tracks, and 87.5 kN for the 15 mm surfaces). The strongest effect on the equivalent pressure can be seen for the inclination angle. Interestingly, this parameter has a different emphasis than in the transition velocity tests described previously. While lower inclination angles appeared suitable for early full lubrication, contact with stronger inclined surfaces resists higher loads under mixed-friction conditions. A possible explanation for the observed effect could be seen in deformations induced by the high loads during the mixed-friction test. If the conical surfaces are flattened by the deformation, specimens with initially high inclination angles are hydrodynamically optimized, while the inclination disappears, or at least drops, under a critical minimum for specimens that already started with low angles. Further insight in interactions between the regarded parameters and a regression for their influence on the bearable loads shall be derived when enough tests are conducted to allow a full fractional analysis.

(Editor's Note: This paper was written in the Spring of 2017. Since that time, the series of tests referred to in the "Bearable Load Under Mixed-Friction Conditions section beginning on page 55 has been completed and results are now available at https://doi.org/10.21268/20170718-120141.)

It must be stated that the values gained thus far should be regarded as a design aid while considering possible extreme load situations for the contact. A permanent bearing operation under the tested conditions does not appear reasonable, since the higher losses in a thrust cone bearing under mixed-friction will not in any way contribute to an efficient gearbox. For "normal" load situations the bearing should be designed for

Table 1 Excerpt of experimentally determined failure loads under mixed friction condition for various thrust cone designs

parameters					results	
surface angle in °	track width in mm	material -	slippage in %	nitride surface -	max. axial force in kN	equivalent HERTZ'ian pressure in MPa
0,5	10	34CrMo4	5	yes	82	226.5
0,5	10	34CrMo4	10	no	40	158.2
0,5	10	30CrNiMo8	5	no	10	79.1
0,5	10	30CrNiMo8	10	yes	70	209.2
0,5	15	34CrMo4	5	no	92	195.9
0,5	15	34CrMo4	10	yes	60	158.2
0,5	15	30CrNiMo8	5	yes	100 (no failure)	204.2
0,5	15	30CrNiMo8	10	no	78	180.3
1,0	10	34CrMo4	5	no	76	308.3
1,0	10	34CrMo4	10	yes	64	282.9
1,0	10	30CrNiMo8	5	yes	100 (no failure)	353.7
1,0	10	30CrNiMo8	10	no	54	259.9
1,0	15	34CrMo4	5	yes	100 (no failure)	288.8
1,0	15	34CrMo4	10	no	70	241.6
1,0	15	30CrNiMo8	5	no	100	288.8
1,0	15	30CrNiMo8	10	yes	100 (no failure)	288.8

averaged equivalent HERTZ'ian pressure values:

parameter -	lower level -	averaged value in MPa	higher level -	averaged value in MPa
surface angle:	0,5°	176.4	1,0°	289.1
track width:	10 mm	234.7	15 mm	230.8
material:	34CrMo4	232.5	30CrNiMo8	233.0
slippage:	5 %	243.1	10 %	222.4
nitride:	no	214.0	yes	251.5

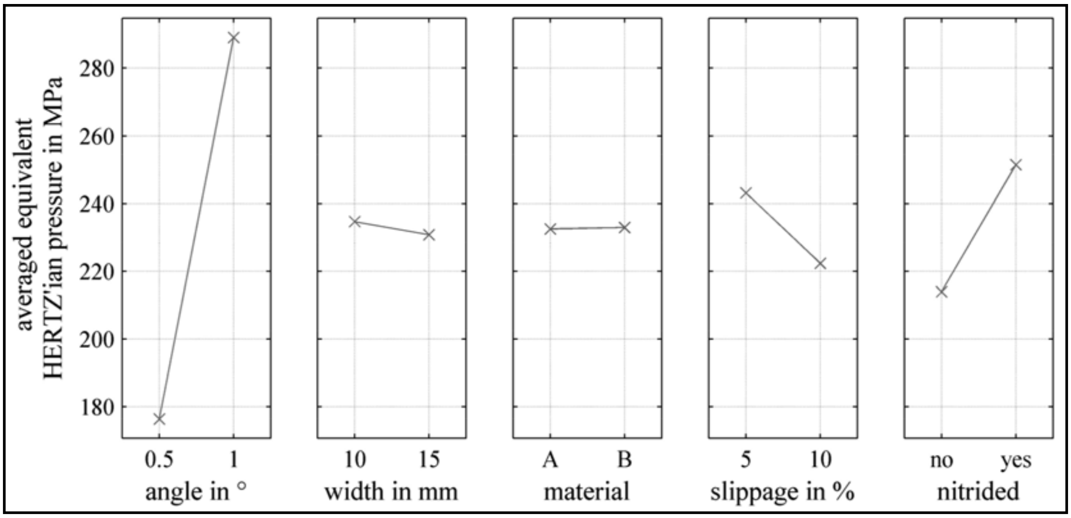

Figure 7 Main effects on the bearable equivalent Hertzian pressure under mixed friction conditions based on data given in Table 1 for fractional factorial design of experiments. Note on material abbreviation: A stands for 34CrMo4; B represents 30CrNiMo8.

full-film lubrication by using numerical simulations. The presented tests reveal a remarkable load reserve for thrust cone bearings running temporarily under mixed-friction conditions — which is important in enabling a reliable application design.

Conclusions

The conducted experiments on the transition behavior to full lubrication have shown that the transition velocity is highly influenced by geometrical parameters. For flat running surfaces, low inclination angles, and low slippage support early full lubrication.

Comparative efficiency tests on a gearbox revealed a remarkable energy-saving potential for lower-load situations by replacing tapered roller bearings with a combination of thrust cone and roller bearings.

Under mixed-friction conditions thrust cone specimens withstood higher loads than expected by common design values. Nitriding of contacting surfaces and low slippage are suitable to increase the bearable equivalent Hertzian pressure. The surface inclination angle generates an interesting influence, since significant higher load values could be reached with greater angles under mixed-friction conditions. This observation makes a constant value for the equivalent Hertzian pressure, as proposed by literature to date, a doubtable design base. The presented mixed-friction experiments reveal a need for geometry-dependent limits. Furthermore, since lower angles encourage full-film lubrication, an optimization problem results for the design engineer, which might be solved by numerical simulations.

(For more details on the work presented in this article, particularly those pertaining to numerical simulations, can be go to: Hess, Marcel, 2018. Einsatz von Druckkämmen zur Effizienzsteigerung von schrägverzahnten Getrieben. Dissertation TU-Clausthal. ISBN 978-3-86948-624-6.)

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