own; however, since Mr. Derner's commentary is not accompanied by appropriate mathematical backup, the same cannot be done in the latter case. The writer suspects, however, that Mr. Derner's analysis does not consider the total effective roller length, notwithstanding his commentary to the contrary, and therefore a stiffer bearing results from his analysis.

Influence of Water on Fatigue-Failure Location and Surface Alteration During Rolling-Contact Lubrication[1]

J. M. HOPKINS.[2] The authors are to be complimented for conducting an investigation of value to bearing users, bearing investigators, and lubrication investigators. However, we feel that some comment needs to be made relative to the significance of the probability of failure of pole and equator area.

Conclusion "A," based on the failure analysis presented in Table 3, states "approximately 90% of failures per unit surface area of region occur on the poles and equator. . . . These regions have reduced fatigue strength."

Admittedly, a predominance of failure occurring in end fiber areas warrants suspicion; however, the "reduced fatigue strength" conclusion is not justifiable without a comparable life analysis of balls failed in end fiber and side fiber areas.

In previous work conducted by our laboratory,[3] we found the rolling contact fatigue life differences between balls failed in the end fiber and side fiber areas to be relatively insignificant in balls of standard manufacture. We also found variations in the failure probability of end fiber to side fiber failures in nonstandard manufactured balls.

We would also like to direct the authors' attention to the paper by Gulbransen and Copan[4] which we feel suggests an alternate failure mode to that proposed by the authors.

R. J. PARKER[5] AND E. V. ZARETSKY.[5] The authors have written an excellent paper on a well-conducted experimental program. The effect of very small water concentrations on roller-element fatigue life is clearly shown.

The work verifies and extends the work of Grunberg and Scott[6] which shows reduced fatigue life with several lubricants for larger concentrations of water than those reported by the author. For these tests,[6] no reduction in life was observed when stainless steel balls were used as specimens. These results[6] support the authors' corrosion hypothesis.

The stress-life relationship found by the authors is of particular interest. The exponents $n = 11.7$ and 11.5 are not surprising. Unpublished fatigue data from the discussers' laboratory indicate an exponent as high as 14 in the five-ball fatigue tester in the stress range of 650,000 to 800,000 psi maximum Hertz stress. The discussers' analysis of other data from a four-ball fatigue tester [3] indicated an exponent of 13 at maximum Hertz stresses between 600,000 and 800,000 psi. The accumulation of such data showing larger exponents than 9 or 10 indicate that the bearing materials may be more sensitive to stress changes at higher stress levels than has been generally accepted.

The authors' data tend to refute the contention put forth in the authors' closure of [3], "that *no power law with a single exponent fits the data well throughout the overload region* (high Hertz stress region, e.g., maximum Hertz stress greater than 600,000 psi), and that this is indicative of a change in mechanism of the fatigue failure phenomenon compared with that prevailing at lower loads where a power law fits very well." A single exponent does fit the authors' data in a very high stress region. In addition, the authors' results suggest two different failure mechanisms exhibiting the same stress-life relation.

The discussers would like to ask the authors if a recalculation of Hertz stress was made based on the altered surface geometry due to plastic deformation and wear? If so, what effect did these alterations have on the reported stress-life exponent?

FRED G. ROUNDS.[7] In order to find an explanation for the detrimental effect of water on fatigue life presented in an earlier paper [1], the authors examined the failed balls by a number of techniques. We have used many of the same techniques to examine failed balls but our purpose was somewhat different. In our case, we wanted to explain the effects of oil additives. The results of our surface analyses tend to parallel those presented by the authors. To study the effects of straight mineral oils, synthetic fluids and additive blends on fatigue failure location, 375 upper balls from 4-ball fatigue tests were etched. The results of our examinations indicated that a region about 15 deg from the equator was as prone to fatigue failure as the polar area on a failure rate per unit area basis. Thus we confirm the poor fatigue characteristics of the equatorial region of balls. Although our equatorial region was somewhat larger than that found by the authors, the difference in equatorial region size may merely reflect differences in the amount of upset that occurred during the initial cold heading operation used to make the balls.

As did the authors, we also examined the ball tracks by light microscopy at about 100× but the white bands noted by the authors were not seen even though no attempt was made to dry the oils prior to test. Do the authors have any ideas about the composition of these bands and how they are formed? When the ball tracks were examined at much higher magnification with an electron microscope, the general surface appearance of balls run at 1,200,000 psi Hertz max. closely resembled the surface appearance of thrust ball bearing races run at loads from 300,000 to 500,000 psi Hertz Max. in friction tests.[8] However, the high load test electron micrographs had one additional feature, fine cracks were often detected in the electron micrographs taken at points on the running track well away from the fatigue spall. These cracks are probably similar to those found by Reichenbach and Syniuta.[9]

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[1] By P. Schatzberg and I. M. Felsen, published in the April, 1969, issue of the Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 91, No. 2, pp. 301–307.

[2] Marlin-Rockwell Corp., Division of TRW Inc., Jamestown, N. Y.

[3] Hopkins, J. M., "Method of Producing Improved Bearing Components by Elimination or Control of Fiber Orientation," Marlin-Rockwell Corporation, Jamestown, N. Y., Oct. 15, 1963.

[4] Gulbransen, Earl A., and Copan, Thomas P., "Effect of Stress on Unusual Crystal Habits of Corrosion Products on Iron, Nickel, Chromium and Stainless Steel," Westinghouse Research Laboratories, Pittsburgh, Pa., 1958.

[5] NASA-Lewis Research Center, Cleveland, Ohio. Mr. Zaretsky is a Mem. ASME.

[6] Grunberg, L., and Scott, D., "The Acceleration of Pitting Failure by Water in the Lubricant," *Journal of the Institute of Petroleum*, Vol. 44, No. 419, Nov. 1958, pp. 406–410.

[7] General Motors Research Laboratory, Warren, Mich.

[8] Rounds, F. G., "Effects of Additives on the Friction of Steel on Steel Part 1. Surface Topography and Film Composition Studies," *ASLE Trans.*, Vol. 7, 1964, pp. 11–23.

[9] Reichenbach, G. S., and Syniuta, W. D., "An Electron Microscopy Study of Rolling Contact Fatigue," *ASLE Trans.*, Vol. 8, 1965, pp. 217–223.

The authors show that the presence of water in squalene has a detrimental effect on both fatigue life and wear. Some years ago, we investigated the effect of water contamination in automatic transmission fluids. It was found that the addition of 0.5% water to a number of commercial Type A Suffix A fluids had no significant detrimental effect on either fatigue life or wear. In our case, wear was determined by weighing the three lower balls both before and after the test. Thus the presence of water in an oil does not automatically mean a loss of fatigue life and an increase in wear.

To explain the reduction in life due to the presence of water in squalene, the authors suggest that water displaces the friction polymer that would normally be found on the contacting surfaces, thus facilitating microcrack penetration and surface corrosion. Bubbling dry or wet argon through the oil reservoir, as did the authors, would tend to restrict the amount of oxygen available in the test area. As a consequence, one wonders whether sufficient oxygen was available to form the normal amounts of friction polymer. If not, an unduly pessimistic picture of the effect of water on wear and perhaps life may have resulted from the authors' work.

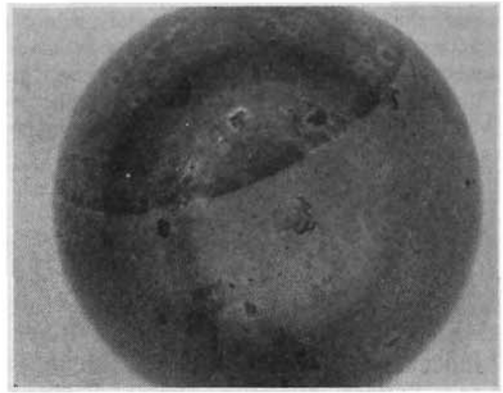

Fig. 1

T. E. TALLIAN[10] AND H. E. MAHNCKE.[11] In discussing this paper, certain comments are in order pertaining both to it and to its precursor [11].[12] These will be given without distinction as to which paper is at issue. This appears permissible since quite a few of the data in [11] are restated in the present paper.

Our first comment pertains to the chemistry of the lubricant used by the authors. The papers give rolling 4-ball test results using squalene, a highly purified non-polar paraffin. The "high" water content in the papers is only 100 ppm. This is no higher than water content in good conventional mineral oils.

The authors report that there was distinct corrosion—outside the running tracks—when the squalane with 100 ppm water was run. No similar corrosion would be expected when running with conventional oils, and the reason for this difference must be the displacement of water from the steel surfaces by polar organic material content of the latter oils, either from polar impurities or from additives. Thus, the authors' results, from the point of view of lubricant chemistry, seem to show primarily that if one (*a*) removes all organic polar material from a hydrocarbon, and (*b*) adds even traces of water, then the mixture is corrosive to steel. They also show, very validly, that the corrosion leads to corrosive wear in the rolling tracks of their test balls and that concurrently—probably as a direct result—life is reduced. It does not, of course, follow that oils containing polar organic materials would act likewise on steel if 100 ppm water were added or, more correctly, that these oils would in any way be improved if the water now present were removed.

Our second comment pertains to the elastohydrodynamic film condition in the tests. Due to the low speed-viscosity product, the elastohydrodynamic film thickness in these tests is so low that the surfaces were necessarily in heavy asperity contact in both the low and the high load tests [12]. Under such conditions, surfaces fail in *surface fatigue*. Spalling is secondary to the formation of plastically worked and microcracked surface material [13]. The authors well recognize this and hypothesize that water enters these microcracks and promotes propagation, either mechanically or chemically. This is a more general form of the Way theory of hydraulic crack propagation which is, among many recently published papers, discussed in a paper by Foord, et al. [14], in this Conference. Since one of us (T. E. Tallian) has, in the past, expressed reservations regarding the Way theory, it is perhaps opportune to summarize our position as follows:

1 We have read the arguments in Way's favor and feel that they revolve principally around these findings:

(*a*) Surface initiated spall *propagation* rates are dependent on lubricant viscosity and are very low if there is no lubricant. Thus, lubricant entering the cracks may be an influential factor [15].

(*b*) In soft steels, when run with sliding, the *lagging* surface spalls before the *leading* surface. Cracks run away from the surface at an angle and are pointed forward in the lagging and rearward in the leading surface. Oil may more readily be trapped in cracks running *forward* and exert greater effect [15].

There are many other detailed arguments, among which is a showing that significant pressures can be calculated at crack tips, given suitable assumptions, and that a hole drilled into a crack stops propagation. These, however, appear less urgent than (*a*) and (*b*).

2 As to (*a*) in the foregoing, it is a well established fact that two given surfaces, when run with a given oil, fail much earlier when the asperity interaction is severe (thin elastohydrodynamic film). This fact governs not only the initiation of a spall, but also the propagation of spalls. In the presence of low film thickness, the *entire surface* in contact is often cold worked and microcracked and spall systems can, therefore, propagate rapidly by being reinitiated over and over again as this distressed surface continues to be subjected to more asperity interaction. Fig. 5 shows a ball with many *unconnected* small spalls arising under such conditions. Thus, while there *may be* a hydraulic crack propagation effect, it is not *required* to explain the rapid propagation of cracks under asperity interaction conditions. The asperity interactions explain this phenomenon directly.

To us, the experience reported in [12], viz., that either thin or thick oils give long life if the film thickness/roughness ratio is high, and short life if the ratio is low, is best reconciled by a theory basing failure rate on the stress fields set up at asperities when they interact. The present authors (and others before them, e.g., [16]) have amply shown that these stress fields depend greatly on lubricant chemistry, and we feel this dependence is due to the chemical action of the lubricant over the entire contact surface and not necessarily inside microcracks only.

3 On argument (*b*) for the Way theory, attention is called to the fact that in rolling contact fatigue tests, deep subsurface plastic phenomena (deformation banding) take place unidirectionally: bands rise towards the surface at an acute angle with the rolling velocity vector [17]. While the mechanics of this phenomenon are by no means clear, it is *not* a surface cracking phenomenon. Rather, it reflects the fact that the plastic flow in rolling contact as described by Merwin and Johnson [18] is directional with respect to the rolling velocity vector. As long as such lack of symmetry

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[11] Supervisor, Chemistry Section, SKF Industries, Inc., Engineering and Research Center, King of Prussia, Pa.

[12] Numbers in brackets designate Additional References at end of discussion.

exists in rolling contact at depths greatly beneath the surface, where no cracks have penetrated, no symmetry should be expected in the propagation rate of surface cracks, whether or not they are hydraulically propagated.

4 For the foregoing reasons and others, it is felt that the Way theory, while having great heuristic value and worthy of being vigorously tested in the future, has not been vindicated and should not at this time be taken for granted in explaining rolling contact failure. This point is relevant to the present papers because it circumscribes the field of applicability of their results. The results are applicable to conditions of severe asperity interaction (low speed, high load) and only there. This, again, is as expected from past results: lubricant chemistry is dominant for low elastohydrodynamic film thickness but not for high elastohydrodynamic film thickness, where boundary lubrication does not take place.

Our final comments pertain to the question of polar regions and fiber flow in balls. We agree that polar regions are areas of lower contact fatigue strength. The argument, however, that this is due to "end grain," i.e., the emergence of "fiber flow" at the poles, is not decisive. There are results on hand in the discussors' laboratory to show that end grain is not of lesser endurance than "side," i.e., surface parallel grain. The real (i.e., large) difference in endurance is between center ingot and surface material. Ball poles represent the center portion of the ball wire; so, to a lesser degree, does the equatorial band.

A word of caution may be in order regarding an interpretation of "fiber flow" in steel as representing elongated inclusions. While any elongated inclusions *will* follow the fiber flow, it is formed by alloy and carbide segregation bands and *not* inclusions (it exists, very visibly, in vacuum melted steel with no large, stringer-type inclusions). End grain wear will corrode (or etch) preferentially, due to the fact that there is in it a larger concentration gradient of alloying elements. Due to this, it is reasonable that life of polar areas should be reduced preferentially by corrosion.

The authors have issued a timely warning regarding the need to investigate trace contaminants in lubricants for their potential effect on rolling contact life. Batch-by-batch variability of lubricants has often been blamed for rolling contact endurance scatter between tests. Trace impurities may well be a substantial contributor to this effect. Further tests with lubricants that do have the usual polar material content and, hopefully, at much lower stress levels where bulk plastic effects do not predominate,[13] are highly desirable. In this regard, it is regretted that shortage of time or funds prevented the authors from running their lower-load tests long enough to produce fatigue life data.

Additional References

11 Schatzberg, P., and Felsen, I. M., "The Influence of Water and Oxygen on Rolling Contact Lubrication," Symposium on Chemistry of Lubrication, Division of Petroleum Chemistry, Inc., American Chemical Society, San Francisco, April 2–5, 1968.

12 Valori, R. R., Tallian, T., and Sibley, L. B., "Elastohydrodynamic Film Effects on the Load-Life Behavior of Rolling Contacts," ASME Paper No. 65—Lubs-11, 1965.

13 Tallian, T. E., "On Competing Failure Modes in Rolling Contact," *ASLE Trans.*, No. 10, 1967, p. 413.

14 Foord, C. A., Hingley, C. G., and Cameron, A., "Pitting of Steel Under Varying Speeds and Combined Stresses," to be presented at the ASME-ASLE Lubrication Conference, Atlantic City, October 8–10, 1968.

15 Dawson, P. H., "Present Position in the Series of Tests on the Pitting of Lubricated Rolling Surfaces," *Proceedings of the Institute of Mechanical Engineers*, Vol. 180, Part 3B, 1965–1966.

16 Rounds, F. G., "Effects of Base Oil Viscosity and Type on Bearing Ball Fatigue," *Trans. ASLE*, Vol. 5, 1962.

17 Martin, J. A., Borgese, S. F., and Eberhardt, A. D., "Microstructural Alterations of Rolling Bearing Steel Undergoing Cyclic Stressing," *Journal of Basic Engineering*, Trans. ASME, Series D, Vol. 88, No. 3, Sept. 1966, p. 555.

18 Merwin, J. E., and Johnson, K. L., "Analysis of Plastic Deformation in Rolling Contact," *Proceedings of the Institute of Mechanical Engineers*, No. 177, 1963.

[13] It should not be inferred that the existence of *any* power law between load and life is assurance that plastic effects are negligible. The law must hold with the *same exponent* when load is reduced to stress levels below 400 ksi or lower. In the high-load tests of the paper, the exponent is *too high* to be compatible with the lower exponent found in full scale rolling bearing tests at lower stress levels.

Authors' Closure

In their discussion Tallian and Mahncke suggest that the absence of any polar organic materials in the highly purified paraffin lubricant used was primarily responsible for the effect of dissolved water, since they assume that in conventional oils polar organic additives would displace water from the steel surface. That this is not so was shown by Grunberg and Scott[14] who studied the effect of additives on water-induced pitting of bearing balls. They found that the detrimental effect on life of water contents in the 100 ppm range was not counteracted by the presence of several different polar organic compounds. They did find that isoamyl alcohol had a significant inhibiting effect on the influence of water. In a subsequent paper[15] it was shown that the addition of an imidazoline derivative counteracted the effect of water. Thus, while there are undoubtedly some additives that inhibit the detrimental influence of water on rolling-contact fatigue life, which has also been indirectly suggested by Rounds in his discussion of our paper, it does not follow that polar organic molecules normally present in lubricating oils will serve that purpose. It appears, rather, that special additives are required.

It was our purpose in this investigation to reduce material and environmental variables to clearly demonstrate the influence of small amounts of water on rolling-contact fatigue and wear. This resulted in the use of a highly purified hydrocarbon lubricant with a limited capacity to dissolve water and consequently low water contents. Nevertheless, the effects were clearly demonstrated. We agree that experiments at lower stress levels should be conducted and preferably with higher water contents, such as might be encountered in a marine environment.

Recent experiments at this laboratory[16] were conducted using No. 208-type angular contact AISI 52100 steel ball bearings at 1800 rpm and a conventional-type paraffin lubricating oil (80 cs at 100 deg F) containing a dispersant detergent, a sulfonate emulsifier, corrosion and oxidation inhibitors. The applied thrust load resulted in a mean Hertz compressive stress level of 200 ksi. The presence of 0.1 percent emulsified water maintained in a colloidal dispersion in the oil caused a reduction in the 10 percent and 50 percent fatigue lives of as much as 60 percent. These results clearly indicate that the detrimental effect of water on fatigue life demonstrated by our four-ball experiments has been reproduced under conditions more characteristic of normal bearing operations and in the presence of various polar organic additives.

It appears, therefore, that the broader applicability of our four-ball results, which has been challenged by the comments of Tallian and Mahncke, is reasonably well established. It further appears that the severity of the rolling-contact conditions in the four-ball apparatus constitutes, at least with regard to the effect of water on fatigue life, a difference in degree rather than kind when compared with normal bearing operations.

[14] Grunberg, L., and Scott, D., "The Effect of Additives on the Water-Induced Pitting of Ball Bearings," *Journal of the Institute of Petroleum*, Vol. 46, No. 440, Aug. 1960, pp. 259–266.

[15] Scott, D., "Further Data on the Effect of Additives on the Water-Induced Pitting of Ball Bearings," *Journal of the Institute of Petroleum*, Vol. 48, No. 457, Jan. 1962, pp. 24–25.

[16] Marzani, J. A., "Effect of Water on Fatigue Life of Angular Contact Ball Bearings," Naval Ship Research and Development Laboratory MATLAB Report 300 (in press).

Comments regarding the relative fatigue strength of different regions of the bearing balls were made by Tallian and Mahncke, Hopkins, and Rounds. The discussers differ among each other regarding the interpretation of our data, there being some agreement and some disagreement with the authors. In this case, as well as in others, it was our primary purpose to demonstrate the effect of small amounts of water in reducing the fatigue strength of bearing steel. We believe the results in Table 3 demonstrate this effect.

We appreciate the comments of Parker and Zaretsky concerning the constancy of the stress-life relationship we found for a high stress region. This constancy tends to enhance the validity of rolling-contact fatigue experiments at elevated stress levels. We did not recalculate Hertz stress based on the running track's altered surface geometry.

The authors express their appreciation to all the discussers for their detailed and penetrating comments.

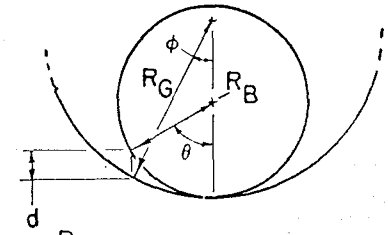
Contact Conformity Effects on Spinning Torque and Friction[1]

SUBHASH K. BATRA[2] The authors must be complimented for their keen insight in isolating an important factor contributing to friction losses in ball bearings. They have presented valuable experimental results by studying the influence of a component of motion in the bearing assembly, that of ball spin, that is often overlooked. In the process, of course, they have raised some intriguing questions about the elastohydrodynamic effects due to the spin motion. But first a few comments about the data obtained under unlubricated conditions may be in order.

In Fig. 3 the authors present experimentally measured spinning torque M_s as a function of total load. These data are compared with the torques predicted (for $f_s = 4$) by the formula developed by Poritsky et al., in reference [4]. This formula was developed for Hertzian contact conditions and assumed a flat contact ellipse. Under the loads imposed and in view of the calculations shown in Table 1 (based on the geometry in the undeformed state) for low values of C it seems reasonable to expect that the area of contact will wrap around the ball over an arc. As a consequence even the pressure distribution may depart from the Hertzian profile. It may, therefore, be more useful to first derive a value for f_s from $C \rightarrow \infty$ case, where Poritsky's assumptions are valid, and then use it to predict M_s for the other values of C. Such calculations provide an estimate of the influence of geometry on the analysis of reference [4], assuming that coefficient of kinetic friction will remain constant throughout the contact zone.

The spinning torque M_s versus load under lubricated conditions is shown in Fig. 7 of this paper. It is clear that the slope of the curve for C as 60 percent is the largest, that for 51 and 55 percent is smaller, and that for $C \rightarrow \infty$ is the least. It seems to suggest that perhaps geometry is playing a significant role in this elastohydrodynamic situation; for it is conceivable that there is a critical value of conformity where M_s versus load curve will have the maximum slope (if indeed it remains linear). It would therefore seem reasonable to explore a wider spectrum of conformity values to confirm or deny such a conjecture.

The calculated value of f_s in the lubricated case (where full film is assumed to exist) is at best a measure of the average shear stress acting over the contact area; its estimation can also be affected by the geometrical influences due to conformity levels.



$$c = \frac{R_G}{2R_B} \times 100 \text{ (percent conformity)}$$

$$\frac{d}{R_B} = (1 - \cos\theta) - \frac{c}{50}\left(1 - \sqrt{1 - \left(\frac{c}{50}\sin\theta\right)^2}\right)$$

Fig. 1

Its physical significance, however, is not very clear; for it is highly dependent on the lubricant properties, geometry of contact and other physical variables. As far as the discounting of viscous drag effect in accounting for the measured spin torques is concerned the authors may have made a hasty judgment. From the estimated Hertz stress it would seem that elastohydrodynamic effects are definitely present; but the analysis used in estimating the viscous drag does not take this into account. It is possible that a more comprehensive elastohydrodynamic analysis may explain the experimental data more adequately.

Table 1 Gap between the ball and the groove in the meridional plane (R_B = 0.25 in.)

θ (degrees)	d (C = 51%) (μ-in.)	d (C = 55%) (μ-in.)	d (C = 60%) (μ-in.)	d ($C \rightarrow \infty$) (μ-in.)
0	0	0	0	0
5.0	18.7	86.77	159.01	951.3
10.0	75.58	349.88	640.43	3,798.06
20.0	313.99	1,446.1	2,633.0	15,076.8
30.0	754.7	3,442.6	6,211.4	33,493.0

Maximum deflection for 100,000 psi maximum Hertz pressure is of the order of 1 mil.

A. B. JONES[3] AND T. A. HARRIS.[4] The subject of friction torque in ball bearings must yet be regarded as a substantially unknown quantity insofar as prediction of bearing adequacy is required. Too little testing effort has been devoted to this topic by researchers and therefore this paper is a welcome sign that serious consideration is being given to better definition of this important factor affecting bearing performance. The authors are to be commended on their presentation with the anticipation that further material will follow.

The discussers, however, wish to assert that establishment of spinning friction coefficients and torque values is only a small portion of the overall problem. Rather, frictional phenomena in ball bearing contacts are the result of combined rolling, sliding and spinning actions. In establishing a meaningful method to predict bearing friction, it would appear that rolling-sliding phenomena are more significant than spinning owing to the ability to develop elastohydrodynamic lubricant films thereby. Fig. 10 of the paper gives an indication of this significance when the sharp decline in friction coefficient is noted in the area of 1.7 percent reciprocal conformity; i.e., $f = 0.59$ where $f = r/d$. It appears that only inside this region of conformity is it possible to establish a satisfactory lubricant wedge and hence generate a load-supporting lubricant film. Since the rolling-sliding interaction is much more effective in generating a lubricant film, it is here anticipated that

[1] By M. W. Dietrich, R. J. Parker, E. V. Zaretsky, and W. J. Anderson, published in the April, 1969, issue of the JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 91, No. 2, pp. 308–313.

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