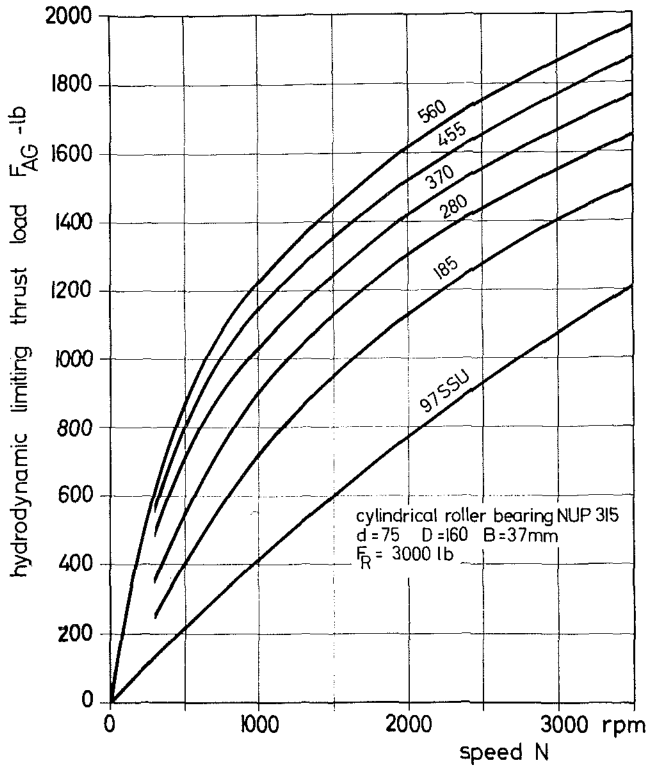

Fig. 12 Hydrodynamic limiting thrust load for cylindrical roller bearing NUP 315

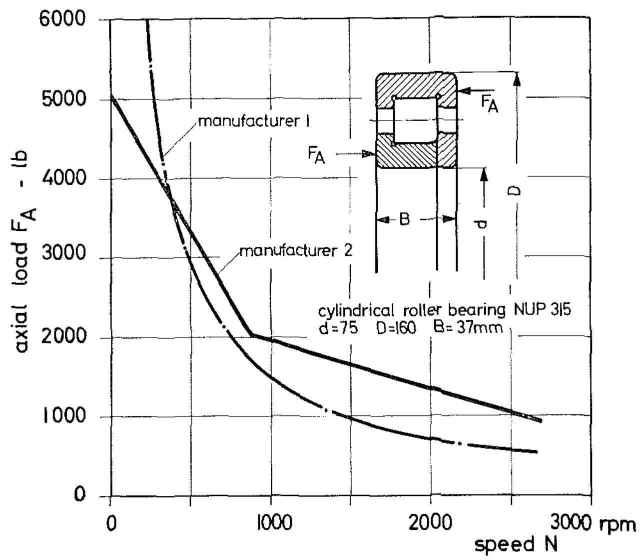

Fig. 13 Safe thrust load by conventional calculation

Conclusions

In designing for the flange contact of radial cylindrical roller bearings large equivalent curvature radii should be provided for contact point between roller end and flange. The distance of this point from the raceway should be as small as possible. With an adequate lubricant such bearings can carry high thrust loads over a wide range of speeds with full hydrodynamic lubrication at the flange. Under full hydrodynamic lubrication friction, wear, and temperature rise are correspondingly low.

The procedures so far used in the determination of the safe thrust load have to be revised and completed. Oil viscosity, for instance, should be introduced as an important parameter. Consideration should be given the fact that contrary to current opinion the hydrodynamic limiting load at the flange increases with increasing speed.

Presently, the hydrodynamic limiting loads are being determined for more differentiated roller/flange contacts. This is necessary for more accurate comparisons with theory. The results can then be generalized and a new, modern calculation method developed.

More testing is being performed in the region of lower speeds to obtain more accurate knowledge on the feasible limiting load under conditions of mixed friction at the flange. Special attention will be devoted to conditions existing with grease lubrication and lubricants with additives.

DISCUSSION

W. J. Derner[2]

The author is to be commended for his detailed investigation into a much neglected aspect of the range of usefulness of the cylindrical radial roller bearing. This reviewer's company has published catalogue ratings for the thrust carrying ability of the cylindrical radial roller bearing in which the concept of constant energy has been the governing factor for the dependence of thrust ability on speed. The author's findings of significantly greater thrust ability at higher speed may enhance the application of the already versatile cylindrical, radial, roller bearing.

We question whether the author's test arrangement with its

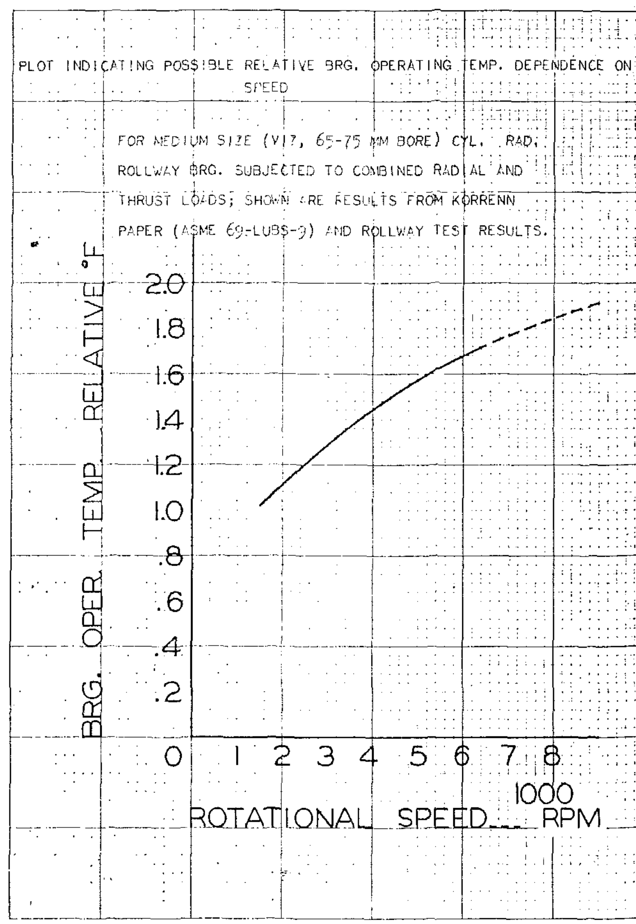

Fig. 14

[2] Rollway Bearing Co., Inc., Syracuse, N. Y.

idealized radial and thrust loading mechanism reflects practical limitations.

Practical applications will seldom provide anywhere near the alignment attained in the test.

The author's use of loose flange plates, in lieu of inner and outer rings having integral flanges, is understandable, but we suggest that the bearing with the loose flange plate represents a more flexible system. We would anticipate less thrust carrying ability in the integral flange type bearing.

Noting that the author's test bearings had radial loads represented by a $C/P \cong 13$, we offer the opinion that the observed torque (which included radial load and thrust load effects on test bearing) is seemingly very high for the conditions of the test; viz. considering bearing size, speed, and loads. Rollway has conducted similar tests on a basic No. 213 radial bearing, under 3000 lb radial load and variable thrust loads up to 2500 lb, at 1750 rpm, and has found torque values only $^1/_5$th of those demonstrated by the author, even accounting for the small size difference. Since the Rollway radial loads represented a $C/P \cong 6$, and since other (earlier) Rollway work has shown a marked dependence of friction *coefficient* on the C/P ratio, we think the high torque observed by the author may be partially explained by these forementioned points.

By combining some operating temperatures reported by the author with similar data from Rollways tests, we would like to suggest the following qualitative relationship between temperature and speed for thrust loaded cylindrical radial roller bearings (illustrated in Fig. 14).

We have also investigated the effects of different oil flow rates on bearing operating temperature. It may be of interest to the author that this temperature is significantly affected by relatively small changes in oil flow rate, as well as changes in differential oil temperature (viz., oil outlet minus oil inlet).

T. A. Harris[3]

The limited axial load-carrying ability of cylindrical roller bearings has been recognized for some time. Palmgren[4] gives a formula relating the allowable axial load to bearing dimensions and speed; e.g.:

$$F_{a_{\max}} = f_a Z d_w^2 (3.5 - f_v N d_w)$$

where Z is the number of rollers per row, f_a is a coefficient for maximum permissible pressure between the roller ends and flange, and f_v is a coefficient for sliding speed. The thrust loads permitted by Palmgren's formula are relatively small compared to radial load. No account is taken of improvement of hydrodynamic roller end-flange lubrication with increasing speed.

To improve the thrust-carrying ability of cylindrical roller bearings, it was found necessary to provide spherical surfaces on the roller ends and slight angles on the guide flanges. Fig. 15, 16, and 17 taken from U. S. Patent 3,268,278[5] illustrate the SKF "Expediter" railroad axle bearing which employs these features. It may be noted in the patent that the flange angle and roller end sphere radius are specific functions of bearing geometry and load. Also, the patent specifies that the loci of contact points between the roller ends and the flanges are circles whose diameters are not too different from the raceway diameters.

Since the development of the patent some further analytical investigation has revealed that only rollers which carry radial load can also carry thrust load. On the other hand, the zone of loading may be expanded under the effect of applied thrust loading depending on mounted clearance in the bearing. For example, Fig. 18 taken from an unpublished SKF report indicates the increase in load zone for a 309 bearing. Together with the load zone expansion, the author has also neglected to indicate the effect of thrust load on fatigue life. Fig. 19 taken from the same

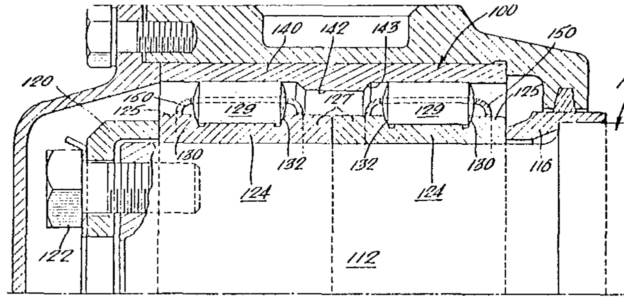

Fig. 16

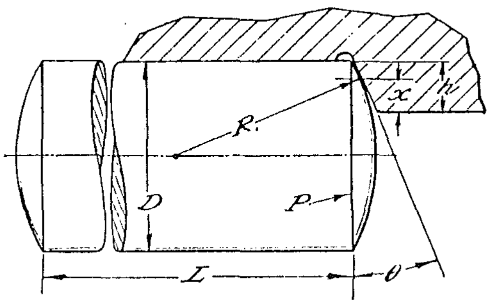

Fig. 17

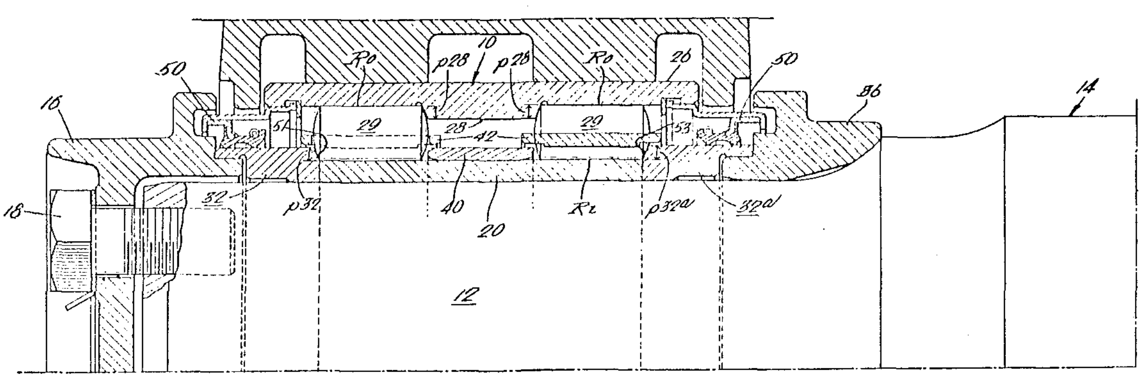

Fig. 15

[3] Manager, Analytical Services Dept., SKF Industries, Inc., King of Prussia, Pa.

[4] Palmgren, A., *Ball and Roller Bearing Engineering*, 3rd Ed., Burbank, 1959, pp. 99–101.

[5] Purdy, G., "Rolling Bearings," U. S. Patent 3,268,278, August 23, 1966.

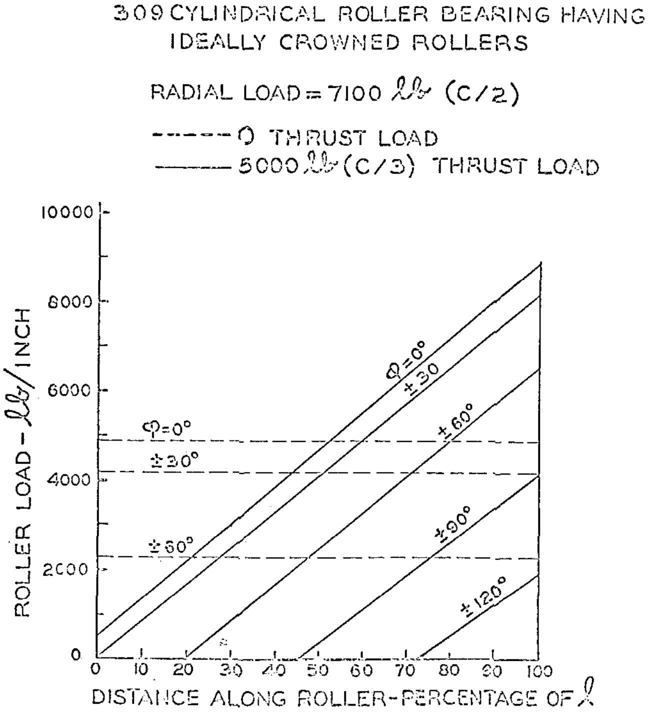

Fig. 18 Roller loading versus axial and circumferential location

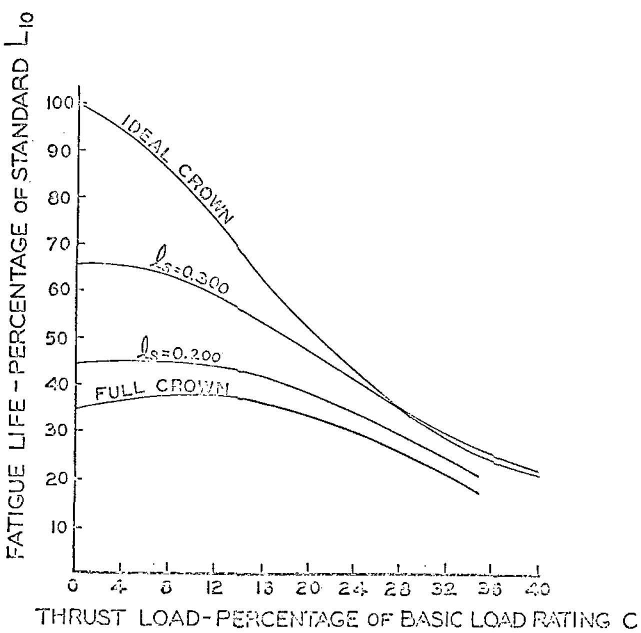

Fig. 19 Fatigue life versus thrust load (309 cylindrical roller bearing; radial load = 7100 lb (c/2))

SKF report illustrates the reduction in fatigue life to be expected with applied thrust load.

The author states that the contact between roller ends and flanges is subjected to pure sliding. This could only be true if the flange-roller contacts occurred at the roller axes. Since contact occurs close to the instant center of rotation; i.e., near the raceway, some degree of rolling occurs, especially for rollers mating with angled flanges.

The writer believes that the author's presentation can be enhanced by a more general analytical approach which permits dedetermination of the load zone extension. Assumptions related to roller loading and tilting (twist) can thereby be eliminated.

D. P. Townsend[6] and E. V. Zaretsky[7]

Dr. Korrenn presents a very interesting paper which defines a means to induce axial load carrying capacity in a cylindrical roller bearing. The usefulness of the paper can be increased if the author can either give the derivation of his equations in an appendix or the references from which they were obtained. The origin of the equations is important because the author appears to use hydrodynamic theory where elastohydrodynamic (EHD) theory should, in the discussors' opinion, be used. Could the author clarify this contention?

In the text, the author mentions pure sliding. In accordance with EHD theory there must be a component of rolling where

$$h \propto (\eta_0 \bar{V})^{1/2}$$

and

h = film thickness
η_0 = viscosity
$\bar{V}$ = average rolling velocity

If there was no rolling present, there is a question whether an EHD film could be developed.

The author states that e, the distance between the contact point and the raceway, must be small. Is there an optimum value for e which depends on the Hertz stress at the contact point between the roller and the race flange?

While Dr. Korrenn defines F_{AG} as the limiting load, a statement is made that the ratio of F_{AG}/F_A should be more than 1 at high speeds. Does this imply that F_{AG} is not the limiting load at high speeds?

Could the author clarify his implication that the beneficial effect of increasing speed is offset by "the increasing rate of wear and the temperature rise." If an EHD film was present it would be large enough to prevent wear and the increasing speed would enhance this capability. However, if insufficient cooling were provided for the bearing, the then increased speed would increase lubricant temperature. In this manner, only, would the increased speed effect be detrimental by decreasing the EHD film and cause predominant boundary lubrication.

Author's Closure

Mr. Derner fears that the idealized test arrangement and the use of loose flange plates could limit the practical applicability of the test results.

We have compared the running performance of bearings on the test rig with many practical applications. In some cases, the bearings, having run on a test rig, were afterwards mounted into gears. The conformity between test run and practice was so good that in our opinion there is no doubt of the applicability of the test results to practice.

Mr. Derner further compares our test results with his own tests and points to the comparatively high friction found by us.

The author concedes that the dependence of the friction coefficient on the ratio C/P partly explains this difference. The main reason, however, is certainly the lubrication and the oil viscosity. In Fig. 20, the friction torque E_L of the test bearing is plotted versus speed, oil viscosity μ being the parameter. Oil sump lubrication was provided. It is clearly seen how, at a constant viscosity μ, the friction of $E_{L\,\min}$ increases with speed, the difference $(E_{L\,\max} - E_{L\,\min})$ mainly proceeds from the increasing whirl losses in the oil sump. At a higher viscosity μ the friction E_L and, hence, the difference $(E_{L\,\max} - E_{L\,\min})$ increases even more.

At an equal outer load, the differences of friction, depending only on speed, oil viscosity, and quantity, are so high that the differences of 1:5 mentioned by Mr. Derner may be explicable mainly with these facts.

[6] NASA-Lewis Research Center, Cleveland, Ohio.
[7] NASA-Lewis Research Center, Cleveland, Ohio. Mem. ASME.

Mr. Harris confirms that the formulas so far given for the calculation of the thrust load-carrying ability of cylindrical roller bearings don't take into account the improved hydrodynamic flange lubrication at increased speed. He further mentions practical executions of SKF-radial cylindrical roller bearings equipped with rollers featuring spherical ends, the flanges being tapered.

The author wishes to make it clear that he did not intend to present thus fitted cylindrical roller bearings—which have been used already by other rolling bearing manufacturers—as a novelty. He rather intended to report about new, so far unknown concepts of the mechanism of lubrication at the flange and about their consequences regarding the introduction of realistic modern calculation procedure. When determining the safe thrust load-carrying capacity of course, the reduction of the fatigue life, mentioned by Mr. Harris, will be taken into consideration.

Mr. Harris is right in saying that there is no pure sliding at the contact points near the raceways. Considering, however, that optimum flange angles are less than one deg, the share of rolling motion within the relative motion at the flange amounts to one percent at the most and therefore can be neglected in relation to the predominant sliding motion.

The author also believes that a more general analytical approach to this problem would be useful. For the substantiation of the theoretical statements, further methodical experimental work is indispensable.

In his thesis "Sliding Friction and Load Limits at the Flanges of Tapered Roller Bearings" [1], the author dwelt thoroughly on the lubrication and the load-carrying capacity at the flanges of tapered roller bearings. Basically, there are the same conditions as at the flanges of cylindrical roller bearings.

Spherical roller ends and tapered or curved flanges show "point contact" and it is certain that, for the explication of the mechanisms associated with the occurring flange loads, the EHD theory has to be used. To compare the results of our test runs with complete bearings, especially the paper [7] "Lubrication at Point Contacts," by J. F. Archard and M. T. Kirk, *Proceedings of the Royal Society*, Series A, Vol. 261, 1961, pp. 532–550, was used.

The author does not believe that under the described geometry of the flange contact there must be a component of rolling to develop an EHD film.

The author has pointed out that a small distance between raceway and contact point favors hydrodynamic carrying and reduces the friction. Also, the Hertz stress is reduced at a smaller distance of the contact point. The reduction of e is practically limited by the raceway undercuts and the roller chamfers between roller end and roller OD. The hydrodynamically carrying surface around the contact point must be in condition to develop unrestrainedly.

The hydrodynamic load limits F_{AG} at the flange are reached with the inset of metal-to-metal contacts at the contact point of

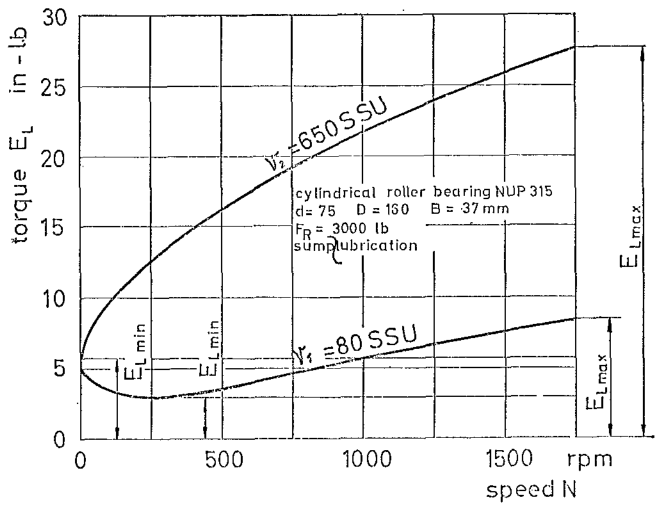

Fig. 20 Bearing friction of different viscosities and speeds

the flange. F_{AG} is increased by adequate geometry by increasing speeds and oil viscosity and by the smallest possible roughness of the surfaces of flange and roller end. The experimentally found relationship between F_{AG}, N, and ν is shown in Fig. 12. The curves apply to a constant minimum gap width at the contact point; its size depends on the roughness of the flanges and roller end, given by the present manufacturing standards. As long as the thrust load F_A at a given speed and viscosity is less than the hydrodynamic load limit F_{AG} found through test runs and theory, consequently, $F_{AG}/F_A > 1$, there is a fully hydrodynamic lubrication at the flange. If F_A exceeds the coefficient F_{AG}, hence $F_{AG}/F_A < 1$, there is mixed friction. Whether a bearing running at increasing speed with a load ratio $F_{AG}/F_A < 1$ enters into the desired range $F_{AG}/F_A \geqq 1$, depends essentially on how much the ratio F_{AG}/F_A was less than 1 initially, hence on the magnitude of the metal-to-metal contact in relation to the amount of hydrodynamic carrying in the contact point. In the most unfavorable case, the temperature rise associated with increasing speed reduces the viscosity of the oil so much that in spite of the now more effective hydrodynamic speed, the gap width is reduced and thus the portion of the load carried by metal-to-metal contact is augmented.

These considerations make it evident that, particularly at higher speeds, a fully hydrodynamic lubrication at the flange should be obtained. This requirement is satisfied whenever $F_{AG}/F_A > 1$.

The author wants to thank the discussers for the valuable suggestions which enabled him to present his work in this field still more in detail. I gratefully feel that with my presentation I have been able to create attention and interest among the experts dealing with problems of bearing engineering.