Table D3 Discussers predicted lives based upon Lundberg and Palmgren analysis

Test no.	A2 Material life factor	A3 Lubrication life factor	$Q/Qcvb$ Determined by discussers	Discussers L10 predicted (hour)	L10 test (hour)	Discussers life ratio: predicted/test data
1	12	2.35	0.2776	22.3	52.20	0.43
2	12	2.35	0.2776	22.3	23.70	0.34
3	12	0.21	0.2776	2.0	5.77	0.94
4	12	0.21	0.2776	2.0	1.73	1.15
5	12	0.21	0.2552	8.2	3.54	2.32
6	12	0.21	0.2972	1.4	22.10	0.06
7	12	2.35	0.2776	22.3	25.40	0.88
8	12	2.35	0.2776	22.3	31.00	0.72
9	4.5	2.35	0.2776	8.4	56.10	0.15

lated h/σ and life assuming a slope of 1.1 and a load/life ratio exponent of 3.0. These calculations are presented in Table D3. The discussers predicted L_{10} life is found in column 5 of this table.

The discussers predicted L_{10} life differs significantly from that presented by Professor Harris. The predicted lives would have been even smaller if the factors for contact angle and traction force had been included. The discussers average ratio of predicted life to actual test life is 0.70. This compares to that obtained by Professor Harris using the Lundberg-Palmgren method of 16.3. Could Professor Harris account for this difference?

The Ioannides-Harris fatigue lives could not be verified as the values and derivation for the A_{IH} factor was not presented. Could Professor Harris present these values and how they are obtained?

We could not verify the author's Eq. (24) from his Eq. (23). Are these equations correct?

Author's Closure

The lengthy opinion expressed by Messrs. Zaretsky and Poplawski concerning the absence of a fatigue limit stress to influence rolling bearing fatigue life in similar manner to the endurance limit universally recognized as affecting the fatigue endurance of structural engineering components is a matter of long-standing record. That Arvid Palmgren (1924) discarded the fatigue limit concept and never returned to it only attests to the substantial lack of cleanliness of the through-hardening steels used to manufacture ball and roller bearings during his professional lifetime. This low cleanliness level must be unfavorably compared to that of modern steels such as CVD 52100 and VIMVAR M50 having oxygen contents (hard particle oxides) and other impurities less than 10 ppm. Consequently, the microstructural stress concentrations, which are represented by the fatigue limit concept, are today substantially reduced as compared to what existed during the four decade period from 1920–1960, for example. In Palmgren's era, a group of 30 ball bearings loaded to a maximum Hertz stress of 3300 Mpa could be endurance tested within a week producing 30 failures. In 1980 at the time the Ioannides-Harris theory was being formulated, a group of similar bearings, similarly tested on the same basic machines, were running without any failures after more than 20 weeks of operation. It is suspected that Palmgren would not have discarded the fatigue limit concept for modern bearings.

It is true that the load-life exponent in Eq. (25) appears to be a function of the Weibull distribution shape parameter (slope). The bearing lives used to determine the exponent were established statistically. In their graphical illustrations of load versus life (see Fig. 8, Lundberg and Palmgren (1947)), LP used the L_{50} (median) life to establish the value 3 for ball bearings. Considering this, it would be reasonable to assert than the distribution of bearing lives did not need to conform to Weibull format to establish the median value; therefore, it may also be asserted that the load-life exponent is independent of the Weibull slope.

It is apparent that the exponents "c" and "h" were determined assuming that the maximum orthogonal subsurface shear stress is the failure-initiating stress. Therefore, assuming that the maximum octahedral shear is the culprit warrants a reconsideration of these exponents which are linked to the bearing material. This is the subject of a current investigation.

It is possible to take any group of endurance data sets and curve-fit a multi-parameter equation to the data. In that case, a load-life equation having a variable exponent could be obtained. For heavy loads and a given bearing dynamic capacity, the exponent would be small; e.g., 2. For lighter loads the exponent would be larger; e.g., 3.6 or 4 as cited by Zaretsky and Poplawski or even larger. (This concept was also presented previously at the 1981 ASTM meeting in Phoenix, Arizona.) The IH theory is more elegant relying only on a fatigue limit stress to achieve the same effect. As the load becomes lighter the applied stress approaches the fatigue limit, the effective stress; i.e., $\tau - \tau_1$ becomes very small, and bearing lifetime becomes very long. In general, the defining equations for physical behavior in nature are rather simple, similar to the IH theory.

It is true that residual stresses may be significant in their influence on bearing endurance. Residual stresses are, however, related to applied stresses as opposed to the fatigue limit stress which must be considered as a material strength. There is substantial evidence that the surface-finishing processes, used in the manufacture of balls and raceways, introduce compressive and tensile stresses into the subsurface. These stresses, which vary with subsurface depth, can act to augment or diminish the fatigue strength (limit stress). This too is under current investigation.

Zaretsky and Poplawski incorrectly state that the fatigue limit affects life only through its effect on the stressed volume. The terms $(\tau - \tau_1)^c$ and ΔV are multiplicative in Eq. (26). A stress less than τ_1 may be applied to the differential volume; ΔV remains but no failure occurs within.

With regard to the comments of Pinel and Rehman, it is not clear how they arrived at the data in their table; therefore no comment on their results can be made. Regarding their query on Eq. (24) however, it does not derive directly from Eq. (23), as they have assumed. Rather, it derives from Eq. (22). In that case, the parameters a^*, b^*, $\Sigma\rho$, (T_1/T) etc. relate to the ball/v-ring contact geometry.

This paper was the initial effort to demonstrate that the bearing component life prediction can be related to bearing life prediction providing the life prediction is based on stresses and not on life factors. That the use of the factor-based method significantly overstated ball endurances in Table 3 should not be surprising since bearing, and hence the derived ball, dynamic capacities are based on applied loading of approximately 25 to 50 percent of rated dynamic capacity. This loading pertains to maximum Hertz stresses in the 1400–2000 MPa range whereas the ball/v-ring tests were conducted at 4000 MPa. Similarly, in the Harris and McCool (1995) study, much of the loading was substantially less than 0.25C causing the standard life prediction method to underestimate fatigue life.