ISO 281:2007 — Caveat Emptor!

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Introduction

In the July 2010 issue of the Society of Tribologists and Lubrication Engineers, *Tribology and Lubrication Engineering*, *TLT*, there is an article (Ref. 1) written by its staff entitled, "ISO 281:2007 Bearing-Life Standard — And the Answer Is?" The lead caption to the article is, "Every major industrial nation in the world accepts the new bearing-life standard (ISO 281:2007) (Ref. 2) except the U. S.

Why? And what does it mean for the industry?"

For this article the *TLT* staff interviewed three persons who were involved with the development of the standard. These were Myron McKenzie, chief engineer, American Roller, Morgan, NC; Dan Snyder, industry consultant, who had retired as director of application engineering for SKF Industries, Lansdale, PA and Martin Correns, director, advanced engineering analysis and simulation, INA – Schaeffler KG, Herzogenaurach, Germany. The *TLT* article (Ref. 1) is well written and, I believe, accurately reflects the opinions of the three interviewees who were in favor of adoption of ISO 281:2007 Bearing-Life Standard by the United States through ANSI/ABMA. In the article, I was cited as a major opponent of this standard, which is correct. For those interested in this subject, the article provides a reasonably good history and technical background and can be obtained online.

Myron McKenzie, Dan Snyder and I have been serving as voting members to two ANSI/ABMA Standards Committees. One committee — ANSI/ABMA B-3 — votes directly to adopt and/or modify bearing standards in the United States. The other committee — U. S. TAG (Technical Advisory Group) — advises the United States representative to ISO how they should vote on various issues, including adopting and modifying ISO Standards. Both U. S. ANSI/ABMA committees compromise members from bearing producers, users and generally interested parties — such as me.

ISO 281:2007 is one of the issues on which we disagree; this ISO 281:2007 standard was not recommended nor adopted by a majority in either committee.

The American Standards Institute (ANSI)/American Bearing Manufacturers Association (ABMA) standards 9 and 11 (Refs. 3–4) are used for the load ratings and life prediction of ball and roller bearings, respectively. These standards, with various updates through the years, were adopted by the ABMA in 1953. (ABMA changed their name from the Anti-Friction Bearing Manufactures Association [AFBMA] in 1993.)

Background

ANSI/ABMA Standards 9 and 11 (Refs. 3–4) are based on the Lundberg-Palmgren life model published in 1947 (Ref. 5) and partially revised in 1952 (Ref. 6). However, the life model of Gustaf Lundberg and Arvid Palmgren dates back to 1924. At that time, Palmgren (Ref. 7), who had been working at SKF in Sweden since 1917, published a paper in German outlining his approach to bearing life prediction (Ref. 7). He presented an empirical formula based on the concept of an L_{10} life, or the time that 90 percent of a bearing population would equal or exceed without rolling-element fatigue failure. This 1924 paper by Palmgren (Ref. 7) is the first time in the literature that a probabilistic approach to life prediction of a machine element was formulated (Ref. 8).

Even where a ball or roller bearing was properly designed, manufactured, installed, lubricated and maintained, rolling-element fatigue will limit the useable life of the bearing. In the life equations that Palmgren (Ref. 7) presented, he incorporated a "fatigue limit," or load below which no failure will occur, as well as a time or "location parameter" before which time no failure should occur.

Over the next 12 years Palmgren evolved his bearing life prediction formulae, eventually recanting his earlier idea of a fatigue limit. In 1936 Palmgren (Ref. 9) published the following:

"For a few decades, after the manufacture of ball bearings had taken up on modern lines, it was generally considered that ball bearings, like other machine units, were subject to a fatigue limit, i.e. that there was a limit to their carrying capacity beyond which fatigue speedily set in, but below which the bearings could continue to function for infinity. Systematic examination of the results of tests made in the SKF laboratories before 1918, however, showed that no fatigue limit existed within the range covered by the comparatively heavy loads employed for test purposes. It was found that so far as the scope of the investigation was concerned, the employment of a lighter load invariably had the effect of increasing the number of revolutions a bearing could execute before fatigue set in. It was certainly still assumed that a fatigue limit coexisted with a certain low specific load, but tests with light loads finally showed that the fatigue limit for infinite life, if such exists, is reached under a load lighter than all of those employed, and that in practice the life is accordingly always a function of load."

In other words, Palmgren (Ref. 9) in 1936 concluded that for bearing steels, and more specifically, for AISI 52100 steel, no fatigue limit existed as a practical matter.

What's the Issue?

I was invited by Tom Astrene of *TLT* to write a response to the July 2010 *TLT* article (Ref. 1). My rebuttal — "In Search of a Fatigue Limit: A Critique of ISO Standard 281:2007" — was published in *Tribology and Lubrication Engineering*, *TLT*, August 2010 edition (Ref. 10). While this article is also available online, I will attempt to summarize the essence of my response.

In 1982 H. K. Lorosch (Ref. 11), of FAG Bearing Company (now part of INA-Schaeffler KG), published results of fatigue tests on three groups of vacuum-degassed, 7205B-size AISI

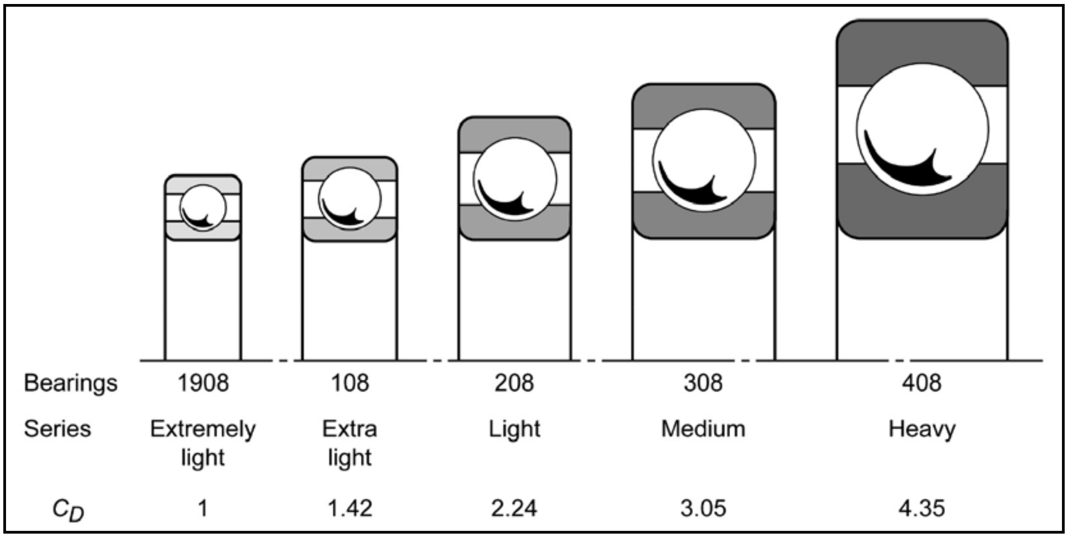

Figure 1 Effect of bearing series on relative sizes and dynamic capacities, C_D, of 40-mm deep-groove ball bearings (Ref. 20).

52100 inner races at maximum Hertz stresses of 2.6, 2.8, and 3.5 GPa (370, 406, and 500 ksi), respectively. These were very highly loaded bearings. From these tests Lorosch concluded that, "Under low loads and with elastohydrodynamic lubrication, there is no material fatigue, thus indicating that under such conditions bearing life is practically unlimited."

O. Zwirlein and H. Schlicht (Ref. 12), also of FAG Bearing Company, in a companion paper published concurrently in 1982 with that of Lorosch, and using the same 7205B-size bearing inner races, reported large amounts of compressive residual stress due to the transformation of retained austenite into martinsite. Bearing research performed at the General Motors Research Center in Warren, Michigan in the 1950s and early 1960s showed that these compressive residual stresses can significantly increase bearing life (Refs. 13-16).

Lorosch, Zwirlein and Schlicht (Refs. 11-12) failed to account for the presence of these significant, induced compressive residual stresses in their bearing raceways. Instead they assumed that the large increases in life that they reported were due to a "fatigue limit." Zwirlein and Schlicht (Refs. 12) concluded that, "Contact pressures (maximum Hertz stresses) less than 2.6 GPa (370 ksi) do not lead to the formation of pitting within a foreseeable period. This corresponds to 'true endurance.'" However, their observation is not supported by rolling-element fatigue data in the open literature for maximum Hertz (contact) stress levels below 2.6 GPa (370 ksi). If Lorosch, Zwirlein and Schlicht (Refs. 11-12) were correct, no bearing in rotating machinery applications would fail due to classical rolling-element fatigue.

Based on the FAG criteria, for a ball bearing the fatigue limit occurs at a maximum Hertz stress of 2.0 GPa (292 ksi). For roller bearings the fatigue limit occurs at a maximum Hertz stress of 1.4 GPa (205 ksi). It is difficult for me to reconcile that for the same bearing steel there are two separate fatigue limits — one for ball bearings and the other for roller bearings — that are so significantly different (Ref. 10).

In 1985, based on the results reported by Lorosch, Zwirlein and Schlicht (Refs. 11-12), Stathis Ioannides and Tedric A. Harris (Ref. 17) at the SKF Engineering and Research Centre in Nieuwegein, The Netherlands, applied Palmgren's 1924 concept (Ref. 7) of a "fatigue limit" to the 1947 Lundberg-Palmgren equations. However, in their 1985 paper (Ref. 17) Ioannides and Harris either did not know or, if they knew, did not reference that Palmgren, also from SKF, discarded the concept of a fatigue limit in 1936 (Ref. 7).

Subsequently, according to Myron McKenzie (Ref. 1), the ISO (in Europe) began to shift its focus on the use of the Ioannides and Harris fatigue life model incorporating a fatigue limit into their bearing life predictions. According to Martin Correns (Ref. 1), a German (Institute for Standardization) DIN standard was published in Germany in 2003 that incorporated a fatigue limit that became part of ISO 281:2007 standard (Ref. 2) four years later. The fatigue limiting maximum Hertz (contact) stress corresponds to 1.5 GPa (218 ksi). This essentially means that were you to run a ball or roller bearing at or below this stress, rolling-element fatigue life would be infinite, or the bearing should not be expected to fail from rolling-element (contact) fatigue. Furthermore, at higher contact stresses, fatigue life would be significantly increased by a reduction in the magnitude of the critical sub-surface shearing stress that causes fatigue.

In 2012, two companion papers (Refs. 18-19) were published in the *International Journal of Fatigue* by researchers from the SKF Engineering and Research Center rationalizing the presence of a fatigue limit in through hardened bearing steels, and the use of a fatigue limit in the ISO 281:2007 standard (Ref. 2). You, the reader, can make up his or her mind regarding the technical contents of these papers and whether, based upon the preponderance of their data, that content supports and justifies the application of a fatigue limit in the standard.

What is the Advantage or Disadvantage of the Fatigue Life Limit?

In order to answer this question, the following hypothetical example is presented. Assume that a gearbox manufacturer designs and manufactures a 2-to-1 ratio speed reducer comprising a high-speed input shaft supported by two medium series, deep-groove ball bearings, and a low-speed output shaft also supported by 2 medium deep-groove ball bearings of the same size as those on the input shaft. (Relative bearing sizes and their respective dynamic load capacities, C_D, are illustrated in Figure 1 (Ref. 20). Further assume that, using the ANSI/ABMA life calculation method from the standards (Refs. 3–4), the calculated L_{10} lives of each of the bearings on the input shaft are 100,000 hours each and the calculated L_{10} lives for each of the bearings on the output shaft are 50,000 hours. Using "strict series reliability" (Ref. 21), the L_{10} bearing system life will be 18,800 hours. The system life is always less than the lowest lived component in the system. It is assumed for this example that the gears will not fail.

Subsequently, the gearbox manufacturer recalculates his bearing life using the ISO 281:2007 standard (Ref. 2) that contains a fatigue limit. During the course of his calculations he discovers that he can substitute, in this case, a light series (smaller) bearing having a smaller outside diameter at lower cost and, theoretically, retaining the same life and reliability as the larger, medium series bearing based on the ANSI/ABMA calculations. With the assumption of a fatigue limit the calculated bearing lives and system life remain the same. The gearbox manufacturing costs are reduced and the weight and size of the gearbox structure can be marginally reduced. The gearbox manufacturer goes with the ISO 281:2007 standard, the fatigue limit, and the small series bearing.

As a marketing inducement, if a gearbox (bearing) failure occurs, the manufacturer warrants the gearbox with a new replacement for one year or 2,000 hours of operation — whichever comes first after purchase and delivery. Life calculations using the ISO 281:2007 standard, together with Weibull statistical analysis, predict that for every 1,000 gearboxes manufactured, 9 gearboxes will be returned as a result of a failed bearing the first year. This means that 9 bearings out of the 4,000 bearings in service, or less than a quarter of one percent, will fail during this time period. However, "If a fatigue limit does not exist" for the bearings in service, the predicted bearing system L_{10} life is reduced from 18,800 hours to 7,520 hours. As a result the gearbox warranty claims would be expected to increase from 9 gearboxes to 24 gearboxes in their first year of service. This also means that 24 out of the 4,000 bearings in service, or 0.6 percent of the bearings, would have failed from fatigue.

For purposes of this hypothetical example, assume further that a large utility purchases 1,000 of these gearboxes to attach to cooling system pumps. The gearbox usage is projected for each pump at approximately 2,000 hours-per-year. The utility wants to project the number of gearbox repairs and/or replacements they can expect over a 5 year period. Based upon the gearbox manufacturer's ISO 281:2007 standard calculations and an 18,800 hour bearing system L_{10} life, approximately 50 gearboxes are projected to be repaired and/or replaced over the 5 year period. However, assume that the bearing calculation does not incorporate a fatigue limit. The resultant bearing L_{10} life system is 7,520 hours. For 10,000 hours of operation (5 years), 130 gearboxes would project to being repaired and/or replaced.

Rolling-element bearing failure time (life) is not deterministic, but probabilistic. The rolling-element bearing life standards are meant to allow the engineer to predict the probability of fatigue failures occurring. Hence, the calculated L_{10} life is the time beyond which 90% of a bearing population will be expected to survive, and before which time 10% will be expected to fail from fatigue. You cannot determine the life of a single individual bearing out of a population — only its probability of survival under its designated operating conditions. But in this writer's opinion, the bearing standards also allow the engineer to assess risk, plan for maintenance and replacement, and perhaps reduce costs. While the above example is hypothetical, it is meant to illustrate that the specific standard used can have significant economic impact. The question that the customer needs to ask the product manufacturer is, "How did you make your life and reliability calculations?" Regarding the above example, in this writer's opinion, a reasonably prudent engineer should use the 130 gearbox replacement projection for planning purposes. In the end — as always — Caveat Emptor! "Let the buyer beware!" **PTE**

(*Editors' Note: Do you have an opinion or question regarding the above? The author would love to hear from you. Please send your questions/comments to: jmcguinn@powertransmission.com.*)

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