Applied Mechanics and Materials, Volumes 7 - 8, 2007, Pages 95-100

Theoretical and experimental working life comparison for a helical gear under linear pitting failure

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Keywords: Theoretical life, failure modes, linear pitting, BS-ISO standards.

Abstract. For machine components like gears and bearings, working life calculation is one of the complex issues to deal with. It is because the mechanics of their operation is depending on many parameters, like loading, friction and lubrication etc. Also the influence of these parameters on the component failure modes cannot be perfectly idealized. But in this regard, standards like AGMA (American gear manufactures association); ISO (International standard organization) and BS (British standards) are quite useful on the basis of which theoretical working life for machine components under a specific failure mode can be predicted. In this paper with linear pitting failure mode assumptions, theoretical working life calculation has been made for a helical gear. BS-ISO 6336-2 standard is used for the gear theoretical life calculations. Further more a wear debris analysis based experiment has been performed for the validation of theoretical calculation. A back to back gear testing rig has been used for the experimental validation. The experimental results show that the theoretical life calculation made on the basis of BS-ISO 6336-2 standards is fairly accurate.

Introduction

If limits of the surface durability of the meshing flanks are exceeded, particles will break out of the flanks, leaving pits. The extent to which such pits can be tolerated (in size and number) varies within wide limits, depending largely on the field of application. In some fields, extensive pitting can be accepted; in other fields any appreciable pitting is to be avoided [1].

So, criticality of gear pitting is depending upon the nature of application. To decide the unavoidable surface degradation limit of gear due to pitting, before its ultimate failure, is still a very vague concept to express. Researchers like P.Dempsey [2] had performed thorough predictive analysis for gear pitting failure. But her research is unable to decide a proper threshold for unavoidable surface degradation limit.

To develop a proper definition and threshold for gear unavoidable surface degradation, as a first step, below in this paper a theoretical and experimental effort is explained. In this effort, BS-ISO 6336-2 standard for pitting failure is used to estimate theoretical working life (i.e. gear life before avoidable surface degradation occurs) for a helical gear. In life estimation theoretical calculations, such values and assumptions

for critical operating parameters like loading, speed, lubrication and gear physical dimensions have been selected that can lead a linear pitting failure on selected helical gear. To check the accuracy of this theoretical life estimation, a wear debris analysis based experimental effort was also performed by using a back to back gear failure testing rig. During comparison study it was found that standard based pitting failure theoretical life estimation may provide an aid to develop a proper definition for gear unavoidable surface degradation.

Theoretical life calculation

As per BS-ISO 6336-2, calculation of helical gear pitting failure is based on the contact stresses, σ_H at the pitch point of the meshing gears, or at the inner point of single pair tooth contact. σ_H shall be less than its permissible σ_{HP} for preventing failure and vice versa. Before detailing the calculation for σ_H, pre selected operational and gear physical dimension parameters are detailed below.

- Type of gears — Helical
- Helix angle — 17.75°
- Pressure angle — 20°
- Centre to centre distance — 113 mm
- Module of gear — 3
- Number of teeth on gear — 35
- Face width — 15 mm
- Tip diameter — 116.25 mm
- Pitch diameter (Also selecting as a reference diameter) — 110.25 mm
- Applied tangential loading (for pitting) — 12104 N
- Lubricant — Mobile ATF 200
- Gear Material — En32,Case hardened low carbon steel

In case of helical gear σ_H will be determined on pitch point of gear when overlapping ratio 'ε_β' of meshed gears is ≥ 1. While when ε_β is less than 1, then σ_H will be determined by linear interpolation between two limit value, i.e. σ_H for spur gears and σ_H for helical gears with $\varepsilon_\beta = 1$.
As per our selection of gear physical dimensions calculated value of ε_β is 0.485. So in further calculation we have used $\varepsilon_\beta = 1$.
The formula of contact stress for the pinion gear is;

$$\sigma_H = z_B \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \leq \sigma_{HP}$$

Where,

Z_B = pinion single pair tooth contact factor

σ_{HO} = nominal contact stress at the pitch point

K_A = application factor (It takes into account the load increment due to externally influenced variations of input or output torque)

K_V = dynamic factor (It takes into account load increments due to internal dynamic effects)

$K_{H\beta}$ = face load factor for contact stress (It takes into account uneven distribution of load over the face width, due to mesh misalignment caused by inaccuracies in manufacture, elastic deformations etc)

$K_{H\alpha}$ = transverse load factor for contact stress (It takes into account uneven load distribution in the transverse direction resulting, for example, from pitch deviation)

As per formulation provided in BS-ISO 6336-2,

Z_B (for spur gear) and Z_B (for helical gear) = 1
σ_{HO} (for spur gear) = 1598.74 N/mm^2, σ_{HO} (for helical gear) = 1482.189 N/mm^2
K_A (As we have uniform loading, for spur as well as helical gear) = 1
K_V (for spur as well as helical gear) = 1.031
$K_{H\beta}$ (for spur as well as helical gear) = 1.00294
$K_{H\alpha}$ (for spur as well as helical gear) = 0.8826

Therefore, σ_H for spur gear = 1527.3 N/mm^2, σ_H for helical gear = 1415.96 N/mm^2
By using a linear interpolation σ_H @ $\varepsilon_{\beta = 0.485}$ = 1470 N/mm^2
Now for permissible contact stress 'σ_{HP}' the formula is: $\sigma_{HP} = \frac{\sigma_{Hlim} Z_{NT} Z_L Z_V Z_R Z_W Z_X}{S_{Hmin}}$

Where,

σ_{Hlim} = Allowable stress number
Z_{NT} = Life factor for contact stress
Z_L = Lubrication factor
Z_V = Velocity factor
Z_R = Roughness factor
Z_W = Work hardening factor
Z_X = Size factor
S_{Hmin} = Safety factor

As per formulation provided in BS-ISO 6336-2,
σ_{Hlim} = 420 N/mm^2
Z_{NT} (for spur/helical gear@ $\leq 10^5$ load cycles) = 1.6
Z_L = 0.92
Z_V = 0.975
Z_R = 0.99
Z_W = 1.211
Z_X = 1
S_{Hmin} (from ESDU 88033 [3], consider as industrial application gear) = 1.1

Therefore, σ_{HP} = 795 N/mm^2
As $\sigma_{HP} < \sigma_H$, so the theoretical life of selected helical gear for limited pitting is less than 10^5. By performing a very gross assumption, as $\sigma_H = 1.85\ \sigma_{HP}$, so start time of limited pitting will be $10^5/1.85$ = 9.01 hours.

Note: Also we have selected pinion and wheel gears of same physical dimensions. So the calculation and value of σ_H will be same for both of them.

Experimental setup, observation and analysis for theoretical life verification

As per the details given above, a pair of case hardened low carbon steel gears with a face width of 15mm and having 35 teeth's was selected for gear pitting failure test. The gears were tested for 21 hours on a back to back gear rig as shown in figure 1. In which after 12 hour operation visible micro pitting was identified. After every 3 hours visible inspection of gears was done and loaded gear teeth's images were captured by using micro imaging capable digital camera. After every one hour wear debris bottle sampling had been done at the sampling point that is provided in the gear rig oil piping before the filter as shown in figure 2. Magnetic plug wear debris sampling was also put under observation after every one hour. Two magnetic plugs at different positions in the tested gear box were used during the experiment. Due to handling problems hour 3 bottle and magnetic sample data could not be recorded.

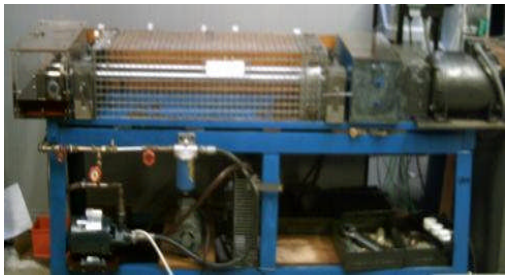

Figure 1 Back to back gear testing rig

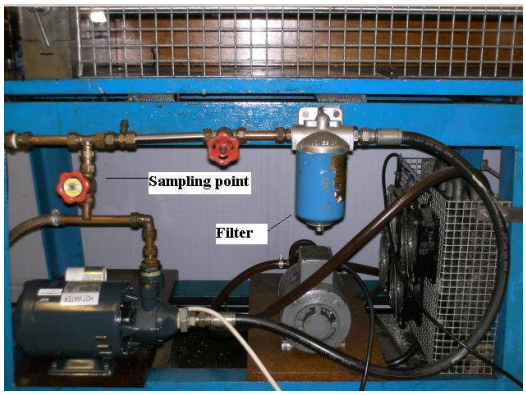

Figure 2 Sampling point and filter arrangement on Back to back gear testing rig

For wear debris quantitative measurement, as when collected in bottle and magnetic oil samples, Arti's Portable Oil Diagnostic System (PODS), as shown in figure 3, was used.

Figure 3 Arti's PODS

Observations of quantitative wear debris measurement for both bottle and magnetic sampling were recorded. By using these observations pitting failure diagnostic trends are plotted as shown in below figure 4 and 5. For each sampling mode (i.e. bottle and magnetic) two diagnostics trends are shown in figures 4 and 5. One of them is plotted on real values and other one is plotted by using a 3-degree polynomial in order to perform pitting failure diagnostics. As from the literature knowledge [4] it is well known that in gear operating life cycle, we have three operating life zones. On the basis of debris quantity trends, all of them are can be easily marked on quantity vs. time curve, as marked on below figure 4 and figure 5. By the help of the available trends, rough estimation of gear life operating zones has been made as:

- Running in zone: 0 hrs – 5.5hrs
- Normal working zone: 5.5hrs – 9.8 hrs
- Failure zone : 9.8 hrs (pitting start time) — Onwards

This rough estimation of gear operating life zones times are also verified by means of visual inspections and camera imaging as shown in figure 6.

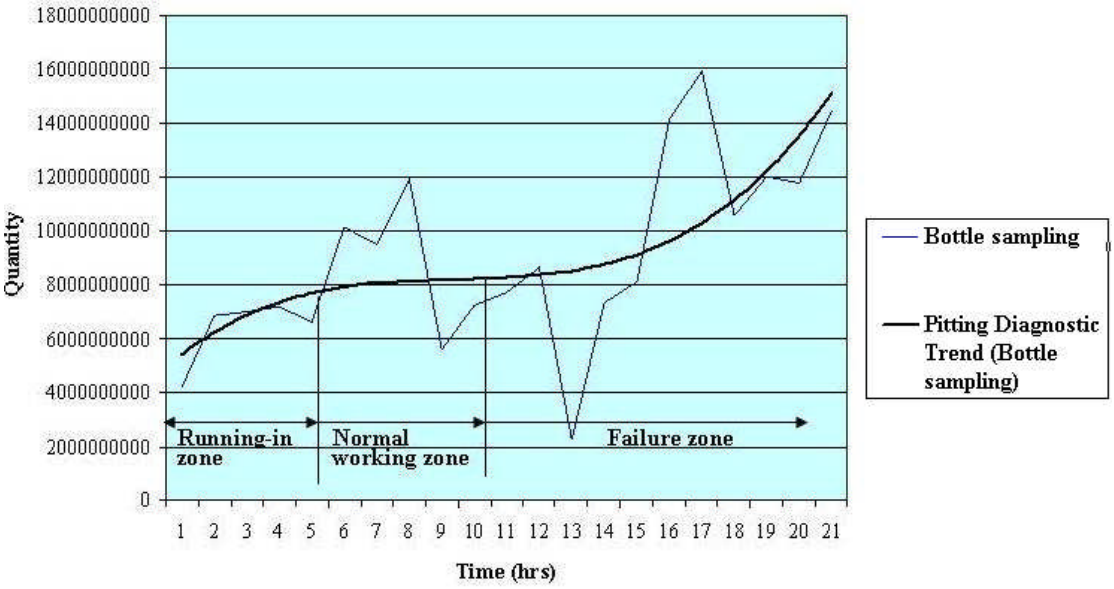

Figure 4. Pitting diagnostics trend from bottle sampling

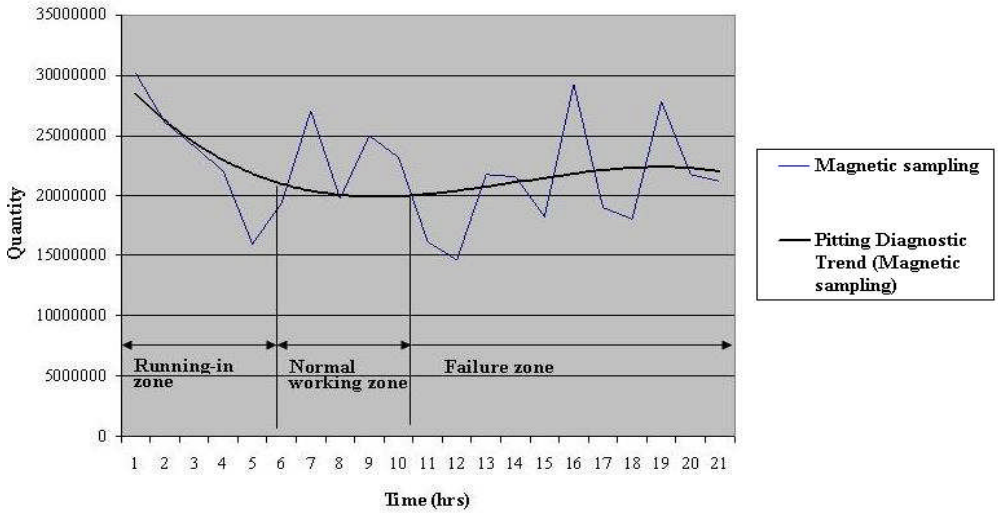

Figure 5. Pitting diagnostics trend from magnetic sampling

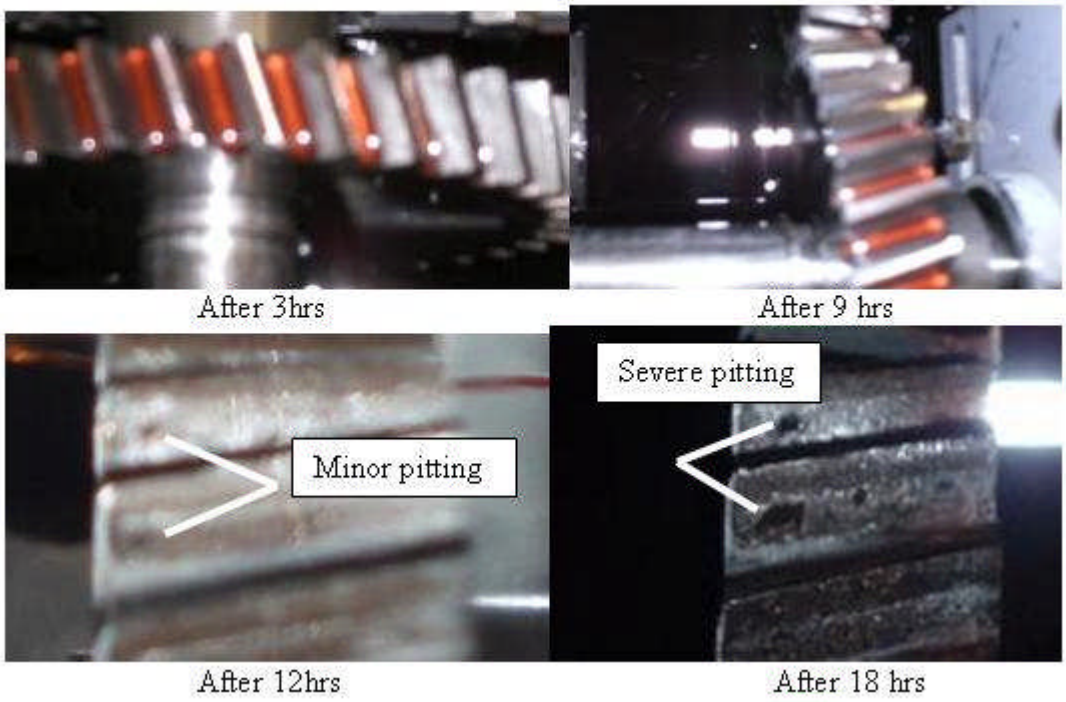

Figure 6. Gear images before and after failure

Conclusion

On the basis of BS-ISO 6336-2 standard the theoretical life for the selected helical gear when limited pitting is permissible is slightly above than 9 hours. While from performed experiment and diagnostics analysis it is clearly identified that pitting was started on the selected pair of helical gear between hour 9 and hour 10. This shows that theoretical formulations available in BS-ISO 6336-2 standard are reliable to calculate helical gear theoretical life. On the behalf of this fairly accurate approach for helical gear life estimation, research related to unavoidable surface degradation limit of helical gear due to pitting, before its ultimate failure, is in plan to perform in future.

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2007-12-31T00:00:00Z

http://dx.doi.org/10.4028/www.scientific.net/AMM.7-8.95

Muhammad Ali Khan and Andrew G Starr. Theoretical and experimental working life comparison for a helical gear under linear pitting failure. Applied Mechanics and Materials, Volume 7-8, 2007, pp95-100

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