BS-ISO helical gear fatigue life estimation and debris analysis validation

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Abstract

Lack of representative theoretical models for wear phenomena in gears causes difficulties in their useful lifetime prediction even under controlled operational conditions. Critical operating parameters such as loading and lubrication affect the wear process in a very complex manner and lead the theoretical modeling to an imperfect zone of assumptions.

Complexities in gear wear mathematical modeling allow approximations to predict its useful lifetime. Based on modeling approximations and assumptions, organizations like AGMA (American Gear Manufacturers Association) and BS (British Standards) provides standards for gear useful lifetime formulations. In these standards the useful lifetime values are estimated by means of experimentation controlled with known gear operating conditions and physical dimensions. But for useful lifetime estimation and validation these standards have not considered any experimental approach that represents the actual gear wear.

In this paper an effort is made to validate the competency of standard's gear useful lifetime formulation by using an approach that is able to provide the idea about actual gear wear. During the effort BS-ISO 6336-2 standard formulation is used for helical gear useful life time estimation under linear pitting fatigue conditions. The used formulation estimation is validated by using wear quantitative features analysis which is

able to provide actual gear wear quantitative trends. The obtained wear quantitative trends fairly validate the lifetime estimation formulation of BS-ISO 6336-2 standard.

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

Abbreviations

AGMA	American Gear Manufacturers Association
ATF	Automatic Transmission Fluid
BS	British Standards
ESDU	Engineering Sciences Data Unit
ISO	International Standards Organization
PODS	Portable Oil Diagnostics System

Symbols

d	Debris density
rsr	Recorded size range
vsr	Volume of size range
CHRQ	Current hour recorded quantity
PHRQ	Previous hour recorded quantity
K_A	Application factor
$K_{H\alpha}$	Transverse load factor for contact stress
$K_{H\beta}$	Face load factor contact stress
K_V	Dynamic factor
Z_B	Pinion single pair tooth contact factor
Z_L	Lubrication factor
Z_{NT}	Life factor for contact stress
Z_R	Roughness factor
Z_V	Velocity factor
Z_W	Working hardening factor
Z_X	Size factor
S_{Hmin}	Safety factor
ε_β	Overlapping ratio
σ_H	Contact stresses
σ_{HO}	Nominal contact stresses at the pitch point
σ_{HP}	Permissible contact stresses at the pitch point
σ_{Hlim}	Allowable stress number

Introduction

Gearing of different sizes and with different manners of loading are indispensable components of machines and devices [1]. Their failure is one the major cause of complete failure and stoppage of a mechanical system. Due to this reason gear failure studies are quite worthwhile to perform and understand.

Gear failures can be classified in two major categories. One is tooth breakage failure and other is surface failure. Each of them has different causes and driving conditions as shown in table-1 [2-5].

In gear failures as provided in table-1, tooth breakage failure largely occurs due to misalignment problems. These problems can easily be reduced or eliminated by adopting the correct gear mounting procedures during assembly. However, gear surface failures normally occur due to critical operational parameters such as external loading, lubrication and operating speed [3-5]. These operational parameters are actually stimulating the generation of a complex wear mechanism on the gear surface which ultimately leads a surface failure during gear operation.

Wear mechanism is a primary source of gear surface failures. And to estimate a useful lifetime of gear, before the discussed failure occurs, it is necessary to develop a mathematical formulation that can completely represents the wear mechanism and its effects on the desired lifetime estimation. But following are some major difficulties in developing a wear mechanism formula [6].

- the wear process itself changes the composition and properties of the surface and near-surface regions;
- surface topography normally changes during the wear process;

- the mechanisms by which wear occurs are often complex and can involve a mixture of mechanical and chemical processes;
- the arbitrary nature of surface roughness needs complicated analytical considerations.

Without involving in the complex wear mechanism assumptions organizations like AGMA, BS and ISO have developed standards contains mathematical formulations that can use to determine useful lifetime of gears. These formulations are based on their gear design, manufacturing, operational and experimental expertise. But for useful lifetime estimation and validation these standards have not considered any experimental approach that represents the actual gear wear features.

In this paper BS-ISO 6336-2 standard is used to estimate useful lifetime under linear pitting fatigue conditions for a helical gear. During the performed research effort those values and assumptions for critical operating parameters such as loading, speed, lubrication and gear physical dimensions are selected that can lead linear pitting on gear surface. The standard's useful lifetime estimation is validated by using gear wear quantitative features based experimentation. The validation experimentation is performed at two different loading conditions that show credibility of validation results.

Useful lifetime estimation

According to BS-ISO 6336-2, such pitting that involves formation of pits and increases linearly or progressively with time under unchanged service conditions is termed as linear pitting. Calculation for time estimation when linear pitting occurs is based on the 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.

In case of helical gear σ_H is determined at the pitch point of a gear when the ε_β of meshed gears is greater than or equal to 1. But when ε_β is less than 1, then σ_H is determined by linear interpolation between two limit values, i.e. for spur gears and σ_H for helical gears with ε_β is equal to 1. According to the selected gear physical dimensions, as given in Table 2, the value of ε_β is calculated as 0.485. So in further calculations of σ_H the value of ε_β is considered as 1.

The formula of contact stress [4] for a pinion gear is;

$$\sigma_H = z_B \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \leq \sigma_{HP} \qquad \text{(Eq.1)}$$

In equation-1, the whole formula is based on three user selected parameters as described below:

(Z_B, K_A, K_V, $K_{H\beta}$, $K_{H\alpha}$) = Largely dependent on user selected gear physical dimensions

σ_{HO} = Proportional to user selected external loading

σ_{HP} = Largely depends on gear material

Now the formula for permissible contact stress ‘ σ_{HP} ’ [4] is:

$$\sigma_{HP} = \frac{\sigma_{Hlim} z_{NT} z_L z_V z_R z_W z_X}{S_{Hmin}} \qquad \text{(Eq.2)}$$

Similarly like the formula for contact stress, the whole permissible stress formula as given in equation-2, is also based on selected parameters as described below:

σ_{Hlim} = Proportional to gear material Ultimate tensile strength

(Z_L, Z_V, Z_R, Z_W, Z_X) = Depends upon lubricant, operational speed and gear manufacturing process

S_{Hmin} = Depends upon the gear application (like Aerospace, Manufacturing)

To proceed in calculations by using above equations, such values of parameters like gear geometry, gear material, external loading, lubricant viscosity, operational speed are needed to select that can cause pitting fatigue on helical gear. For this all the mentioned parameters are selected and provided in Table 2. Furthermore, to increase the reliability of results, both standards' fatigue life estimation and gear wear quantitative analysis validation is performed on two different loading conditions.

By using values of Table 2 the variables required for σ_H in equation 1 are calculated as provided in Table 4.

Now using Table 4 variables values in equation.1 the contact stress values on different loading conditions are give below.

- For loading condition-1:

σ_H for spur gear (at $\varepsilon_\beta = 0$) = 1093.27 Mpa

σ_H for helical gear (at $\varepsilon_\beta = 1$) = 373.59 Mpa.

Now by using interpolation the required σ_H (at $\varepsilon_\beta = 0.485$) = 744.08 Mpa

- For loading condition-2:

σ_H for spur gear (at $\varepsilon_\beta = 0$) = 943.2 Mpa

σ_H for helical gear (at $\varepsilon_\beta = 1$) = 346.23 Mpa.

Now by using interpolation the required σ_H (at $\varepsilon_\beta = 0.485$) = 653.55 Mpa

Similarly variables required to calculate σ_{HP} in equation 2 are calculated as:

σ_{Hlim} = 420 Mpa , Z_L = 0.92 , Z_V = 0.975, Z_R = 0.99 , Z_W = 1.211 , Z_X = 1 and S_{Hmin} (from ESDU 88033 [2], consider as industrial application gear) = 1.1.

Therefore, from equation 2, σ_{HP} = 420.04 (Z_{NT}) Mpa

As a general rule of material failure, pitting will start on the gear flank as soon as the contact stress becomes equal to the permissible stress and hence by equating their values the value for the Z_{NT} parameter can be obtained.

For loading condition-1:

$$Z_{NT1} = \frac{744.08}{420.04} = 1.77$$

For loading condition-2:

$$Z_{NT2} = \frac{653.55}{420.04} = 1.55$$

From BS-ISO 6336-2 standard the operation life cycles values at life factor 1.6 and 1.3 are 6×10^5 and 1×10^7 respectively. Now as an estimation the operation life cycles for above calculated life factors, where pitting failure will start, are given below,

Useful lifetime for loading condition-1 = $(6 \times 10^5) \times (\frac{1.6}{1.77})$ = 542372.88 cycles

= 542373 cycles

Useful lifetime for loading condition-2 = $(6 \times 10^5) + (1.55 - 1.6)(\frac{10^7 - 6 \times 10^5}{1.3 - 1.6})$

= 966667 cycles

Experimental setup

Two pairs of case hardened low carbon steel gears with a face width of 15 mm and having 35 teeth were selected for two pitting failure tests at different loading conditions. All experimental parameters values, provided in Table-2, for loading, lubricant and operational speed were applied on the testing rig as shown in Figure-1. Same number of teeth's selection for both driver and driven gear was used to observe the wear status on teeth of any gear when pitting occurs on same location teeth of the other gear. During first test the gears were tested for 1.08×10^{6} operational cycles. While for the second test the second gear pair was tested for 1.44×10^{6} operational cycles. Visible inspection of the gears was undertaken after every 1.8×10^{5} operational cycles and images of the gear teeth's were captured by using a micro imaging digital camera. After every one hour wear debris bottle sampling was undertaken at the sampling point that is provided in the gear rig oil piping before the filter as shown in Fig 2.

Wear quantitative features measurements for collected oil samples, using an Arti's PODS, as shown in Fig 3, was performed and the results were recorded for wear particle sizes and their respective quantities.

Tests observations and debris analysis verification

During first test at loading condition-1 visible micro pitting i.e. 5mm to 10 mm [7] was identified on 3rd visual inspection i.e. after 5.4×10^{5} cycles of operation as shown in Fig 4. While for the second test at loading condition-2 the defined pitting was identified on 6th visual inspection i.e. after 1.08×10^{6} cycles of operation as shown in Fig 5. In both tests, at test completion, more than 50% of driven gear teeth's were observed with micro pitting. In contrast to driven gears, driving gears were observed less pitted.

For wear quantitative features analysis verification the recorded wear particle sizes and quantities for both tests are plotted as shown in Figs 6 and 7. During plotting cumulative formulas as given below in equation-3 and 4 are used . The aim for applying the mentioned formulas is to convert the PODS provided data into representative size and quantity numbers.

$$\text{Size number} = rsr \times vsr \times d \times 1000 \qquad \text{(Eq.3)}$$

$$\text{Quantity (for any hour)} = CHRQ + PHRQ \qquad \text{(Eq.4)}$$

Note: In gear pitting failure majority of generated wear particles are platelet in shape [8]. To approximate their volume, the 'rsr' value is considered as the length and width of the particle. For particle thickness a value 10% of length is considered [9].

Wear particle size and quantity feature can be used as a representative of wear severity and wear rate at the gear contact surfaces [10]. In case of pitting failure it is anticipated that as soon as the pitting fatigue starts at the contact surfaces, large quantity of particles will come out from the contacting gear flanks and leave a pit hole in the gear surface. Due to this release of particles at the time of pitting fatigue a sudden increase in wear particle size and quantity feature can be detected in an oil sample. From size and quantity plots as shown in Figs 6, 7 it is clear that wear severity and wear rate reaches high values at 4.8×10^5 cycles and 1.02×10^6 cycles of gear operation for test-1 and test-2 respectively.

The visual observations and wear quantitative features indications shows the possible pitting start time for test-1 is in between 4.8×10^5 to 5.4×10^5 cycles and for test-2 it is in between 1.02×10^6 to 1.08×10^6 cycles. The estimated useful lifetimes on both loading conditions are 5.42×10^5 cycles and 0.96×10^6 cycles and are fairly close to their respective ranges as observed and measured during the experimentation.

In above mentioned experimentation the lubricant Mobil ATF 200 having viscosity as mentioned in Table 2 was used. As gear surface failures are also dependent on lubrication and lubricant properties. So the results obtained during the experiment are only valid for Mobil ATF 200. Use of any other lubricant might make a difference with the results achieved during this research effort.

Conclusion

The above discussed calculations, test observations and wear quantitative features analysis validation shows that the mathematical formulae defined in the BS-ISO 6336-2 standard can be used to predict the lifetime of helical gears. On the basis of this approach, research on the life estimation for unavoidable surface degradation of helical gear due to pitting, before its ultimate failure, is planned for the future.

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Tables:

Failure mode	Failure cause	Failure drive parameter
Tooth breakage Fatigue evidence , cracks , fractures and plastic deformation	Caused by bending stress exceeding the fracture strength or fatigue strength of the gear material. Mostly occur due to misalignment problem	External applied loading
Surface failure Pitting	Caused by the contact stress exceeding the fatigue strength or the crushing strength of the gear material.	External applied loading, lubrication surface interaction and lubricant properties
Surface failure Scuffing , scoring and abrasive wear	Caused by oil film failure, inadequate lubrication or dirt.	Lubrication flow and properties , operational temperature and pressure

Table 1 Gear failure modes, causes and driving parameters [2-5]

Type of gears	Helical
Helix angle	17.75º
Centre to centre distance	113 mm
Number of teeth on gear	35
Face width	15 mm
Pitch diameter (Also selecting as a reference diameter)	110.25 mm
Applied tangential loading (for pitting) Loading condition-1 Loading condition-2	 14465.26 N 4347 N
Lubricant	Mobil ATF 200 (Viscosity: 78.31 Centistokes at 25ºC)
Gear Material	En32,Casehardened
Testing Speed	1000 rpm

Table 2 Gear physical dimensions and operating parameters for pitting failure life estimation and validation

Loading conditons	Calculated variables
Loading condition-1	• Z_B (spur gear) = 2.65 and Z_B (helical gear) = 1 • σ_{HO} (spur gear) = 1801.69 Mpa, σ_{HO} (helical gear) = 1529.87 Mpa • K_A (for uniform loading, for spur as well as helical gear) = 1 • K_V (spur) = 0.03 and K_V (helical gear) = 0.02 • $K_{H\beta}$ (spur) = 1.00 and $K_{H\beta}$ (helical gear) = 1.00 • $K_{H\alpha}$ (spur) =1.71 and $K_{H\alpha}$ (helical gear) = 3.41
Loading condition-2	• Z_B (spur gear) = 2.65 and Z_B (helical gear) = 1 • σ_{HO} (spur gear) = 987.65 Mpa, σ_{HO} (helical gear) = 838.65 Mpa • K_A (for uniform loading, for spur as well as helical gear) = 1 • K_V (spur) = 0.04 and K_V (helical gear) = 0.03 • $K_{H\beta}$ (spur) = 1.00 and $K_{H\beta}$ (helical gear) = 1.00 • KHα (spur) = 3.60 and KHα (helical gear) = 6.32

Table 3 Variables calculated for σ_H by using BS-ISO 6336

Figures:

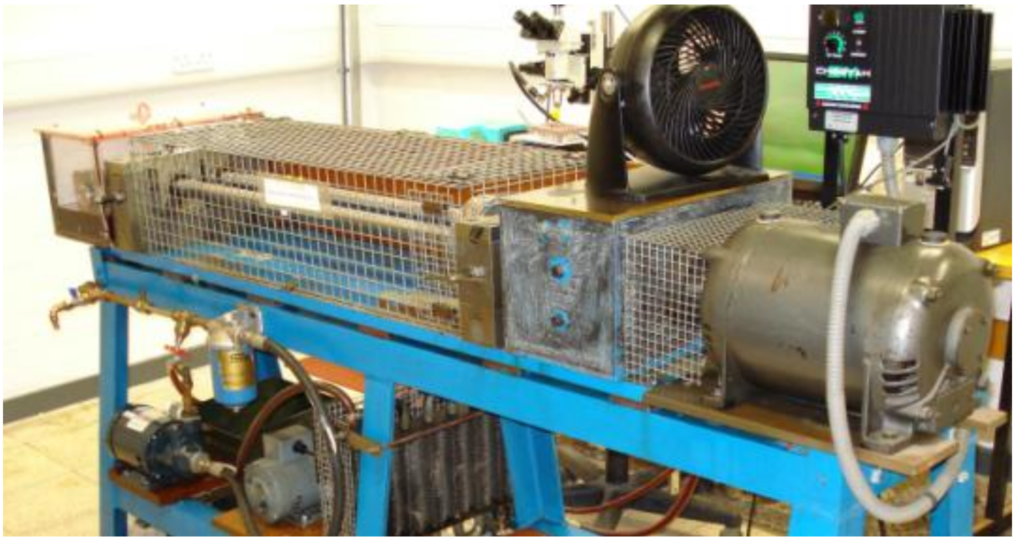

Figure 1 Back to back gear testing rig

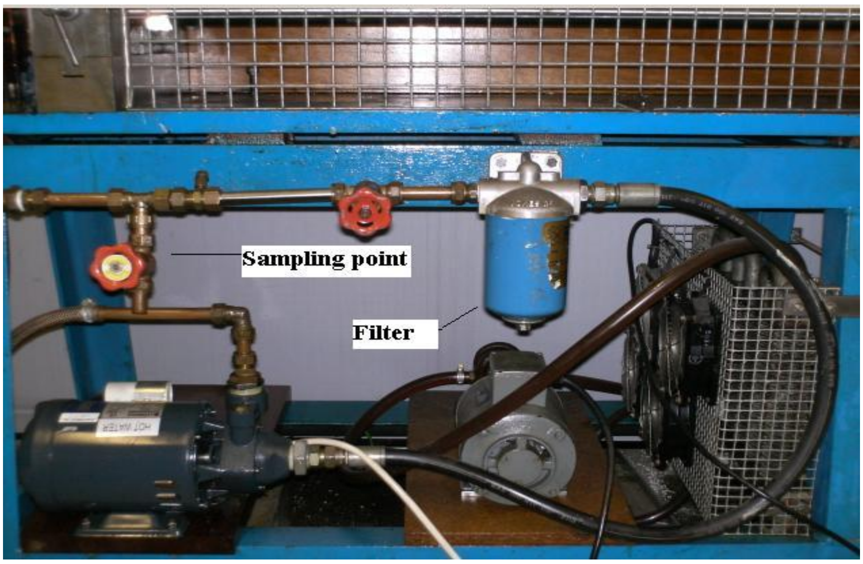

Figure 2 Sampling point and filter arrangement on back to back gear testing rig

Figure 3 Arti's PODS

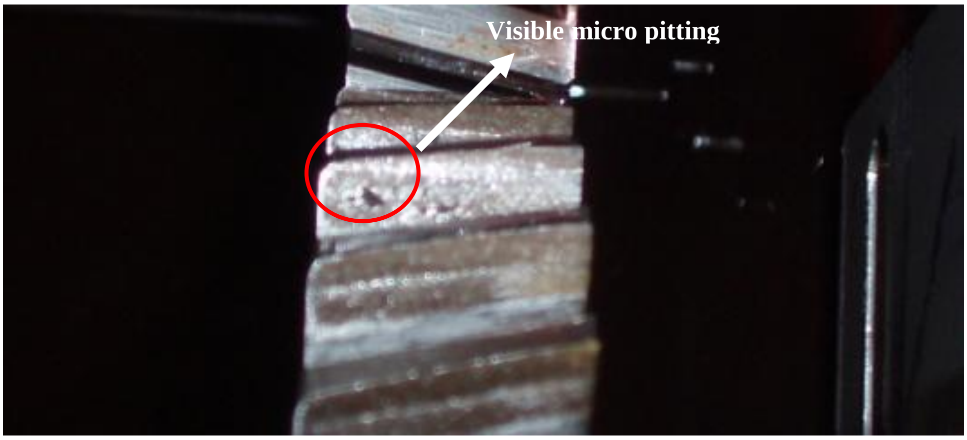

Figure 4. Test-1 gear images showing pitting fatigue

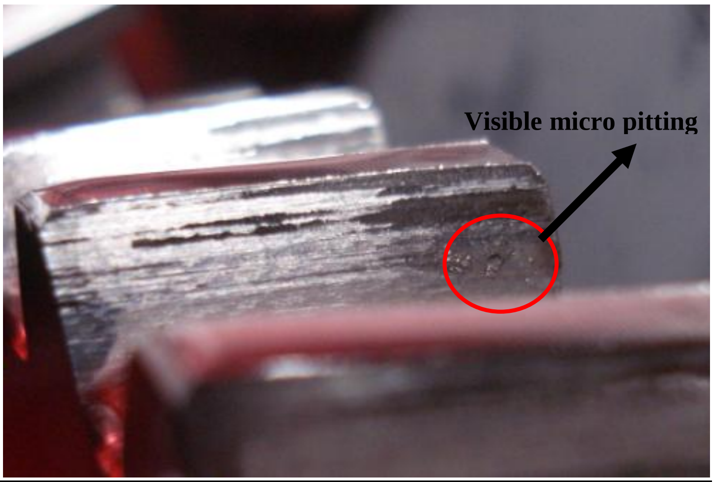

Figure 5. Test-2 gear images showing pitting fatigue

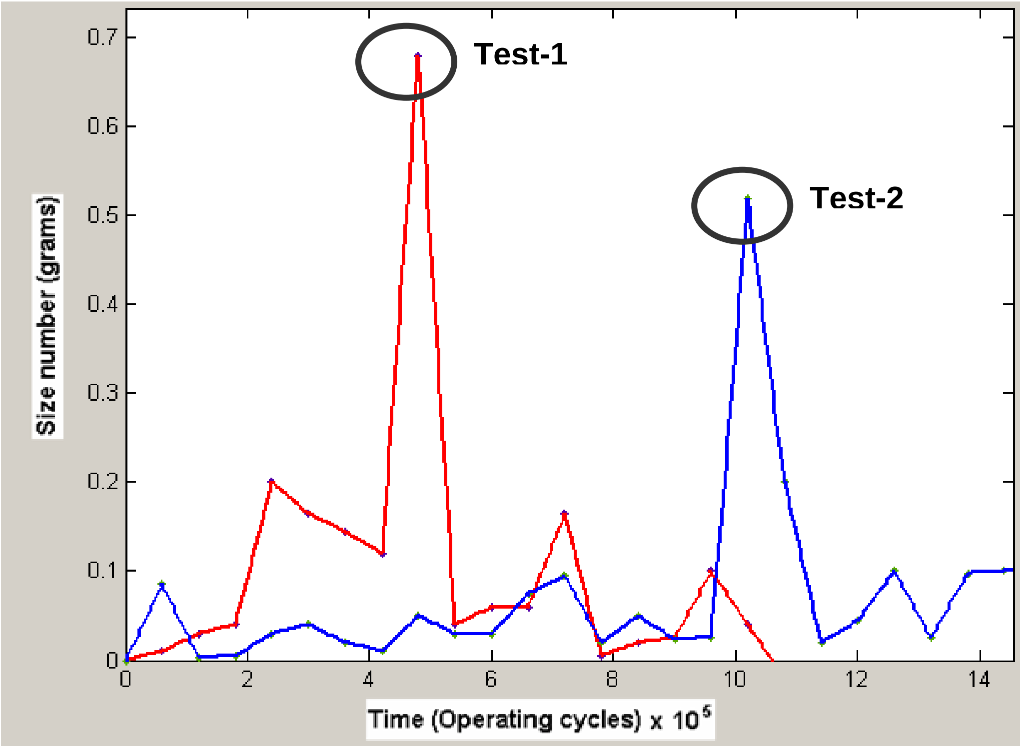

Figure 6. Size trend for pitting diagnostics

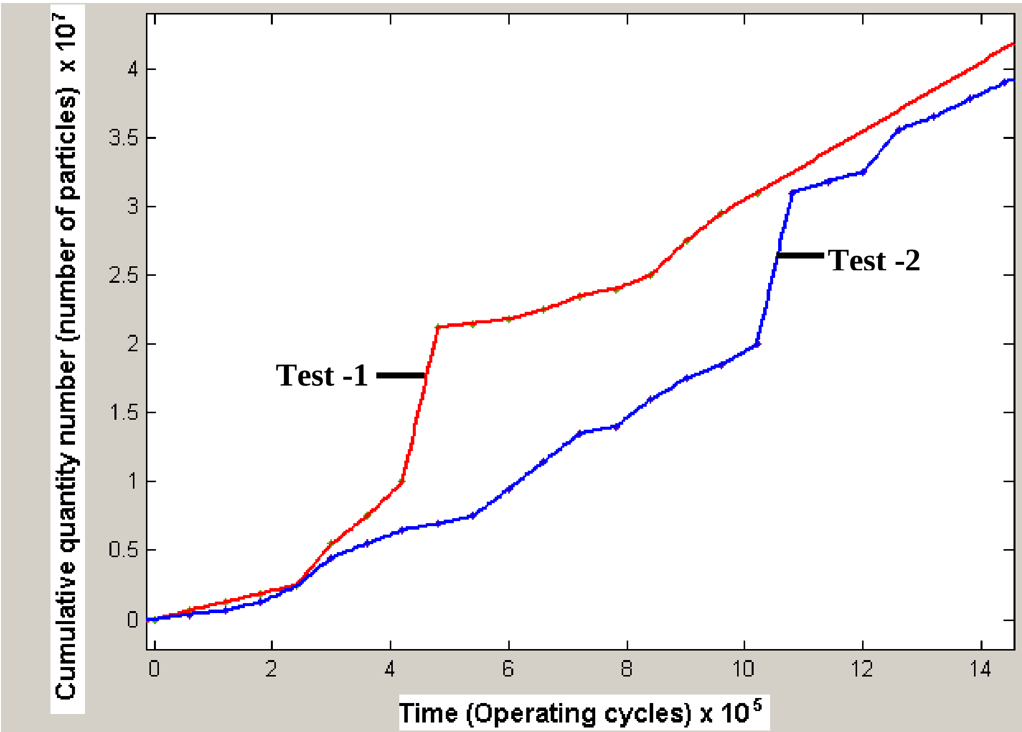

Figure 7. Quantity trend for pitting diagnostics