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Worm Gearbox Bearing Life Prediction

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Abstract*: *Industrial gearboxes are designed for minimum 3 to 5 years of life considering normal working conditions. Theoretical life of gearbox system can be predicted by calculating the life of individual components of the gearbox like bearings, coupling and gear-pair. There is significant amount of research work done on estimating individual component life perdition however combined life prediction of a gearbox is a quite complex phenomenon. The challenge in developing a single life prediction model for variety of gearboxes available in today's market. There is a huge variety in their types, sizes, costs, application conditions (indoor, outdoor, marine, aerospace), safety requirements (domestic to hazardous). Providing a common solution that address all these is near to impossible. Major cause of early gearbox failure is wrong selection for application, improper installation, contaminants and excessive shock and impact loads. Vibration measurement gives an early indication of failure of rotating parts in gearbox which are primarily bearings and gear-pair. This research focuses on step by step approach to calculate the life of bearings in a gear box and gearbox life prediction models. The methodology followed can be used for other types of industrial gearboxes.

Key words*: *American Bearing Manufacturers Association (ABMA), American Gearing Manufacturers Association (AGMA), Fast Fourier Transform (FFT), Rolling Contact Fatigue (RCF)

I. INTRODUCTION

Gearbox life depends upon minimum life of its components. Gear pair is designed for higher life as per the application for which it is selected [1], [2]. Bearings are the most critical components in deciding overall life of gearbox. Hence it is important to understand causes of bearing failures and the life prediction models of bearings[2], [3]. Both ABMA standard and ISO 281 give detailed life prediction procedure. Theoretically it can be proven that bearings can last almost infinitely for a given application if they are properly mounted, lubricated and kept free from contaminants. However in real-life conditions this is not possible and bearings early due to various reasons[4], [5] and [6]. Bearing failures can be broken down in to two basic categories, one premature failure where bearings don't even run for minimum life for which they are selected and second is the fatigue failures where bearing run for sufficient time and then fail. Many a times premature failures can be catastrophic hence it is important to understand their root causes and make a plan to avoid them in applications[3].

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A. *Causes of bearing failure –*

i. Inadequate lubrication – about 43% of the bearings fail due to inadequate lubrication which results due to insufficient or interrupted supply of lubrication or many a times wrong selection of lubricants. Proper heat transfer from bearing hot races to lubricant needs to be ensured, else bearings can fail due to overheating[3].

ii. Improper installation – about 29% of the bearings fail due to improper installation. The reasons for this could be shaft or mounting defects, misalignment, incorrect fit of inner race with shaft or outer race within housing, excessive clearance, surface finish of mounting surface or housing, excessive axial play, false brinelling due to vibrations, end shield or bearing seal damage while mounting[3].

iii. Improper sealing – about 18% bearings fail due to ineffective seals. This can be due to contaminants from inside the lubricant or outside the bearings, moisture in the lubricant, corrosion[3].

iv. Subsurface fatigue - about 8% bearings fail due to subsurface fatigue. This can be due subsurface crack development coming out of Rolling Contact Fatigue (RCF) [8], [9]. The details of RCF are explained in section B.

v. Miscellaneous reasons – about 2% bearings fail due to other miscellaneous reasons which can be due to an electric current passing through bearings which are not properly insulated, quality of bearing steel and impurities in bearing steel processing. Vacuum degassed 52100 steel are most popular bearing material. Also incorrect selection of bearing can also result in premature failure. Bearing failure statistic shown in table 1 gives the common causes of failure and their % occurrence based on general industrial bearing failure analysis[3].

B. *Rolling Contact Fatigue –*

Fatigue failures are due to rolling contact fatigue (RCF). RCF theory for bearing material is not new, however it's interpretation and implementation in real-life bearings is important [7]. Rolling contact fatigue failure is made up of two dominant mechanisms, one is subsurface originated spalling [9] and the other is surface originated pitting. First step to avoid fatigue failures is to test the bearing steels using rolling contact fatigue type testing. These type of testing equipment are available and can be customized as per customer needs. Most popular types of machines are rotary tribometers which press the pins or balls on a rotating disk. Different combinations like Ball-on-disk [24], Cylinder-on-disk, Pin-on-disk, Cross cylinder, Block-on-ring, Block-on-cylinder, Ball-on-ring, Linear Reciprocating (Ball/Pin on Plate), ball on rod types are available [3]. A constant load is applied on the material to be evaluated. The material is either stationary or rotating based on machine configuration.

Load is applied using a standard 52100 vacuum degassed steel material either in the form of balls or a flat plate.
The test continues till the first spall is observed. There are weak points distributed randomly in the material, constant loading and simultaneous rotation results in to development of cracks at such weak points due to orthogonal shear stress at particular death. Minimum of 20 such readings are taken and the graph is plotted on Weibull curve [7]. The slope of the Weibull regression lines & the spread give idea about the probable L10 life and acceptance bands. Steel samples from same heat lot are used for RCF testing to eliminate lot to lot heat variations. Once the steel qualifies the RCF test then the heat parameters are set by the steel foundry to produce same quality of steel over a longer period. However RCF testing is done periodically to ensure the good quality of bearing steel supplied [3].

C. Bearing life prediction programs –

There are commercially available software like Cobra [10]and Mesys [11]. These use the standard bearing formulae for bearing analysis. There are several application specific programs [12], methodologies using standard finite element analysis [13], [16] developed to analyze the bearings. Other than these traditional life prediction methods, there are non-conventional methods like using Artificial Neural Network [14], [15], [18], [20], X-Ray diffraction method [17], Damage mechanics [19], health state assessment analysis [21].

II. EXPERIMENTAL SETUP

From failure causes mentioned in section I-A, root cause of majority of the gearbox failures is vibration that starts with bearings, increases the temperatures of bearings, damages gear teeth that results in failure of gearbox. Hence it is important to measure the vibrations and temperatures at all bearing locations along with housing.

A. *Test set-up.*

Fig. 1 Experimental Se-up

1. Drive side bearing Plummer block, (Ball bearing)
2. Gearbox output side Plummer block (Ball bearing)
3. Output shaft end support Plummer block (Ball bearing)
4. Motor LK-3701, Godrej Lawkim make, (1/2 hp, single phase, foot mounted, frame B-56)
5. Lovejoy coupling (bore size 30 mm)
6. Worm gearbox Gateej make (20 reduction ratio, input solid shaft diameter 35 mm, 50 mm extended, output shaft diameter 25 mm, 210 mm extended)
7. Vibration sensor QMVT2, Banner make (bi-axil vibration sensing and temperature sensing capability with RS-485 to USB cable)
8. ”J” type thermocouple (plug and play type with cable and display)
9. Non-contact type speed sensor with mounted bracket and cable
10. Hanger bearing for vertical radial load application (ball bearing)
11. Metal support structure with rubber pads to absorb vibrations
12. Metal frame for radial loading
13. Load cell for measuring radial load
14. Threaded rod & nut arrangement for radial loading
15. Load digital display
16. Temperature digital display iTherm make
17. Speed digital display

The test rig consists of a test bed that holds a gearbox and supports both driving & loading mechanisms. Various types of Gearboxes can be mounted using adaptors keeping the driving & loading mechanisms same. The gearbox is driven by ½ hp motor using a coupling. Radial load is applied on output shaft using a specially designed hanger bearing-screw arrangement. Precision loading is possible with this type of loading arrangement. Temperature & vibration readings are taken at various points to make sure vibrations are within limits specified by ISO 2372. Bearing life is calculated using ISO: 281 and gearbox life is predicted for different scenarios.

B. *Gearbox specification details*

i. Gearbox type: Worm reducer, foot mounted.
ii. Center distance between input and output shaft: 1.75 inch
iii. Reduction ratio– 20, Input shaft – solid diameter1.18 inch
iv. Pressure Angle (ϕn): 21.5º
v. Lead Angle (λm): 8.688º
vi. Effective Diameter of the Worm (dm): 1.38 inch
vii. Effective Diameter of the Worm gear (Dm): - 6.12 inch
viii. Hand: Right
ix. Output shaft: solid extended dia1”, length 8.125 inch
x. Input shaft speed: 1485 rpm, Output shaft speed: 74.25 rpm
xi. Motor hp: 0.5, Input shaft bearing span:7.5 inch
xii. Output shaft bearing span:7 inch
xiii. Input shaft over hanging load (OHL): 0 lb, (solid input shaft extension 50 mm), Output shaft extension OHL– 0 lb xiv. Input shaft bearings: SKF 6206, Output shaft bearings: SKF 30205
xiv. Gearbox output torque capacity:573 in-lb (from catalogue)
xv. Gearbox temperature range: (-30 to 110 ° C)

C. *Gearbox cross section details*

Standard worm reducer is used for the test set-up with worm on input shaft supported by ball bearings and worm gear on output shaft supported by taper roller bearings.

Fig. 2 Worm gear box cross section details

III. GEARBOX AND BEARING LIFE PREDICTION.

Step by step methodology is developed to calculate gear forces and bearing life using the actual parameters of gearbox referring industrial standards AGMA and ISO.

A. *Gear force calculations–*

Worm gearbox forces are calculated based on industry standard ANSI AGMA 6022-C93 and ANSI AGMA 6034-B92 [5] Worm gearbox life is calculated as per reference AGMA [5] Motor 0.5 hp and output torque 573 in-lb is selected based on mechanical output torque as per product catalogue. 7-step procedure is followed for calculating gear forces.

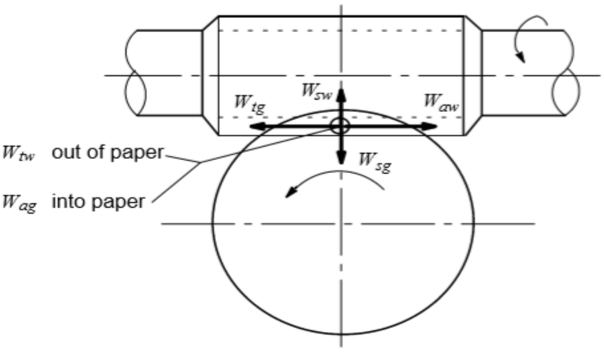

Fig. 3 Worm and worm gear forces right hand [5]

Fig.3 shows forces coming on worm-worm gear considering anticlockwise rotation.

Where Wsw = Wsg= Mesh separating force

Wtg = Waw = Worm gear tangential force = Worm Thrust force

Wtw=Wag=Worm gear thrust force = Worm tangential force

Steps used in gear force and bearing life calculation are mentioned below.

i. Sliding Velocity (V) at mean worm diameter: -

$$V = \frac{(\pi \times dm \times n)}{(12 \times COS(\lambda))} = \frac{(\pi \times 1.38 \times 1450)}{(12 \times COS(8.688))} = 529 \text{ ft/min}$$

ii. To calculate the coefficient of friction (μ): - this depends upon velocity range & can be calculated using formulae-

a) If V = 0 ft/min,Then μ = 0.150

b) If V = 0 to 10 ft/min,

Then $\mu = 0.124 \times e^{(-0.074 \times V^{0.645})}$

c)If V > 10 ft/min,

Then $\mu = 0.103 \times e^{(-0.110 \times V^{0.45})} + 0.012$

So for Sliding Velocity (V) >10 ft/min,

∵ Coefficient of friction $\mu = 0.103 \times e^{(-0.110 \times V^{0.45})} + 0.012$

$\mu = 0.103 \times e^{(-0.110 \times 529^{0.45})} + 0.012 = 0.02818$

iii.Efficiency Calculation (η): -

As per AGMA 6022-C93, Worm driving Efficiency,

$\eta_w = \frac{(\cos \phi_n - \mu \tan \lambda_m)}{(\cos \phi_n + \mu \cot \lambda_m)}$ where

η_w is the worm driving efficiency;

η_g is the wormgear driving efficiency;

λ_m is the lead angle of the worm at meandiameter (8.688 °)

ϕ_n is the normal pressure angle (21.5 °)

μ is the coefficient of friction (from AGMA6034-B92).

To assure that the worm gear will drive, putting the values in above equation we get η_w= 83%

iv. Calculation for the Input Torque (Ti): -

$$T_i = \frac{T_O}{(\text{Ratio} \times \eta)} = \frac{\mathbf{573}}{(\mathbf{20 \times 0.83})} = 34.5 \text{lb.in}$$

v. Worm Tangential Force (Wtw): -

$$Wtw = \frac{(2 \times T_i)}{d_m} = \frac{(2 \times 34.5)}{1.38} = 50\ lb$$

vi. Worm Axial or Thrust Force (Waw):-

$$Waw = \frac{(2 \times T_O)}{D_m} = \frac{(2 \times 573)}{6.12} = 187.25 \text{ lb}$$

vii. Worm Radial or Separating Force:-

$$W_{sw} = \frac{W_{aw} \times TAN(\phi_n)}{\cos \lambda_n} = \frac{187.25 \times TAN(21.5)}{\cos (8.688)} = 74.61 \text{lb}$$

B. *Bearing life calculations*

i. Bearing reaction calculation –

OHL Component: -Nil since we don't have overhung load

Details of Bearings: -(From SKF Bearing Catalogue)

Deep groove ball bearing, Designation: - 6206

Inner Diameter1.18", Outer Diameter2.44", Basic Dynamic Load Rating4563 lb, Basic Static Load Rating2517 lb, Width of Bearing 0.63"

Assuming input shaft Rotates clockwise viewed from Right Considering Forces in Vertical Plane (refer figure 4)

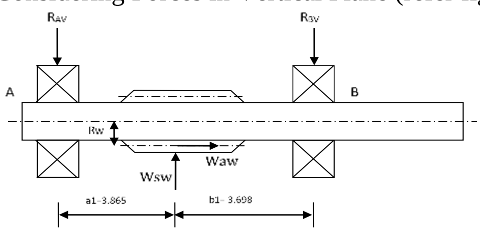

Fig. 4 Bearing reactions

Taking Moment @ A and considering Equilibrium,

$$R_{BV} = \frac{((Waw \times Rw) + (Wsw \times a1) - (F_{rv} \times (a1 + b1 + c1)))}{(a1 + b1)}$$

Putting the values,since $F_{rv} = 0$

$$R_{BV} = \frac{((187.25 \times 0.69) + (74.61 \times 3.865) - 0)}{(3.865 + 3.698)} = 55.21 \text{ lb.}$$

Considering equilibrium

∵RAV + RBV + Frv= Wsw

So,RAV = Wsw - Frv - RBV

= 74.61-0- (55.21)= 19.4 lb.

So the reactions in vertical plane are,

RAV = 19.4lb

RBV = 55.21lb

Similarly forces in horizontal plane are calculated that gives

RAH= 22.4 lb

RBH = 23.4 lb

Resultant Radial load acting on the Bearing: -

At Bearing 'A', $R_A = \sqrt{{R_{AV}}^2 + {R_{AH}}^2}$

$= \sqrt{19.4^2 + 22.4^2} \quad = \quad 29.63$ lb.

At Bearing 'B'

$R_B = \sqrt{{R_{AH}}^2 + {R_{AH}}^2} = \sqrt{((55.21)^2 + (23.422))^2} = \sim 60$ lb.

Axial load acting on the Bearing is Fae = Pa + Fa,

Fa = External Load acting on the bearing, in this Case Fa = 0 lb, So,Fae = Waw, = 187.25 lb

As per SKF catalogue, axial load is 0.25 times static load capacity, in this case it is 0.25 X 2517 = 629.25, hence axial load of 187.25 lb is acceptable.

ii. Equivalent load calculation –

Select maximum load between two bearings A and B so that we get a worst-case life.R_A= 45.96 lb, R_B = 60lb, So we consider 60 as load P.

iii. Modified life calculation -

L_{nm}modified life of the bearing is given by

$$L_{nm} = a_1 \times a_{ISO} \times (\frac{C}{P})^3$$

Wherea_1=Life modification factor for reliability Table I

$a_{ISO} = f \times (\frac{e_C C_u}{P}, k)$ where f = size factor

Table ILife modification factor for Reliability [6]

Reliability %	Lnm	a1
90	L10m	1
95	L5m	0,64
96	L4m	0,55
97	L3m	0,47
98	L2m	0,37
99	L1m	0,25
99,2	L0,8m	0,22
99,4	L0,6m	0,19
99,6	L0,4m	0,16
99,8	L0,2m	0,12
99,9	L0,1m	0,093
99,92	L0,08m	0,087
99,94	L0,06m	0,080
99,95	L0,05m	0,077

For ball bearings, C_u is given by

$C_u = \frac{C_0}{22}$, where $C_u$is the fatigue load limit

where pitch circle diameter of bearing is ≤ 100 mm,

C_o = Static load rating which is 11.2 kN

$C_u = \frac{11.2 \text{ kN}}{22}$ = 509 N = 114.42 lb

e_C = Contaminant factor = 0.5 for normal cleanliness for pitch circle diameter of bearing is ≤ 100 mm refer TableII.

P = Dynamic equivalent radial load given by P = XF_r+ YF_a, where X and Y factors are given in Table III

P = 1(59.97) + 0 (187.25) = 59.97lb as resultant radial load is 59.97lb

k = viscosity ratio which is the ratio of the actual kinematic viscosity, ν, to the reference kinematic viscosity, ν1. The kinematic viscosity, ν, is considered when the lubricant is at operating temperature.

Table II Contamination factor [6]

Contamination factor	Ec	
Level of contamination	Dpw≤100 mm	Dpw>100 mm
Extreme cleanliness Particle size of the order of lubricant film thickness;laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed	0.8 to 0.6	0.9 to 0.8
Normal cleanliness, Oil filtered through fine filter; conditions typical of bearings greased for life and shielded	0.6 to 0.5	0.8 to 0.6
Slight contamination Slight contamination in lubricant	0.5 to 0.3	0.6 to 0.4
Typical contamination Conditions typical of bearings without integral seals; coursefiltering; wear particles and ingress from surroundings	0.1 to 0.3	0.2 to 0.4
Severe contamination, Bearing environment heavily contaminated and bearingarrangement with inadequate sealing	0.1 to 0	0.1 to 0
Very severe contamination	0	0

$k = \frac{v}{v_1}$,The reference kinematic viscosity, ν1, can be estimated by means of the diagram in Fig. 5, depending on bearing speed and pitch diameter, Dpw ([mean bearing diameter 0,5 (d + D)]

Table III Values of X and Y for radial ball bearings [6]

X & Y factors for single row radial ball bearing						
earing typ	Relative axial load		Single row ball bearing			
	$\frac{f_0 F_a}{C_{0r}}$ c	$\frac{F_a}{iZ{D_w}^2}$	$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$	
			X	Y	X	Y
Radial contact ball bearings	0,172	0,172	1	0	0.56	2,3
	0,345	0,345				1,99
	0,689	0,689				1,71
	1,03	1,03				1,55
	1,38	1,38				1,45
	2,07	2,07				1,31
	3,45	3,45				1,15
	5,17	5,17				1,04
	6,89	6,89				1

Putting value of mean diameter for 6206 as (30+62)/2 = 46 mm, speed 1450 rpm, v_1 is 20 mm^2/sec, kinematic viscosity of oil used in gearbox (Mobil Glygoyle HE-460) at room temperature 40 deg. Cent is 460 cSt which is 4.6 Stokes or 4.6 mm^2/sec.

So k value comes out to be

$k = \frac{v}{v1} = \frac{4.60}{20} = 0.23\ mm^2/sec$

To calculate a_{ISO}, when values of k and $\frac{e_C C_u}{P}$ are known Fig.6 is used.

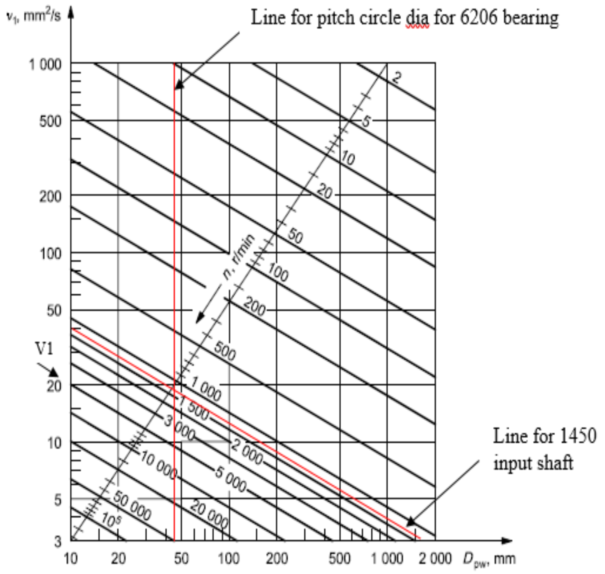

Fig. 5 kinematic viscosity v1for ball bearings [6]

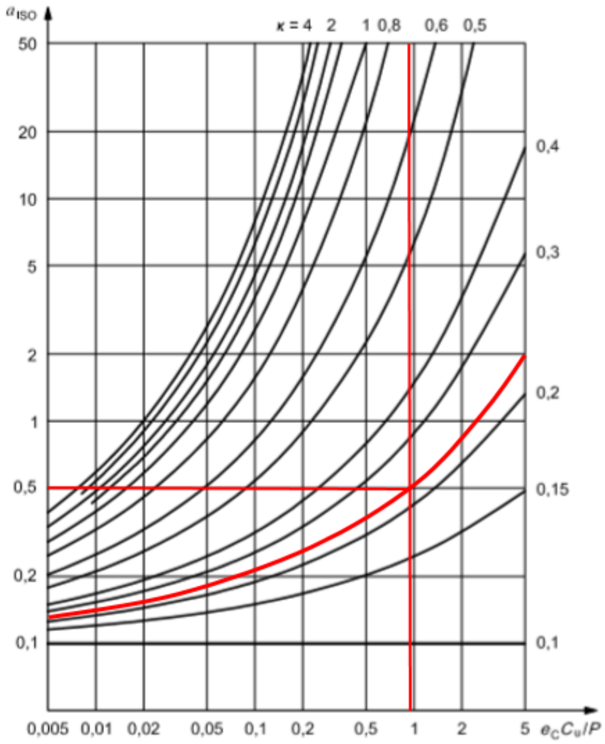

Fig. 6 Life modification factor, a_{ISO}, for radial ball bearings

Putting values of e_C, C_u and P,

$$\frac{e_C C_u}{P} = \frac{0.5 \times 114.42}{60} = 0.954$$

Putting these values in Fig. 6 we get a_{ISO} as approximately 0.5 Hence modified L_{10} life is given by

$L_{10} = a_1 \times a_{ISO} \times (\frac{C}{P})^3$, Putting the values of a_1 =1, a_{ISO}=0.5, C = Basic dynamic load rating given in bearing catalogue = 20.3 kN = 4563.6 lb and P = 60

$L_{10} = 1 \times 0.5 \times (\frac{4563.6}{60})^3$=~88000 millions of revolutions.= 88000 × $\frac{1000000}{(60 \times 1450)}$= 1029278hrs

With similar approach, taper roller bearing life on output shaft is calculated as L_{10}= 3799316 hours. These lives are higher as gear box is very lightly loaded in lab environment. In actual applications, there are overhung loads, higher duty cycles & harsh environmental conditions that reduce the bearing life significantly. As per ANSI/AGMA 6013--A06, L_{10} life of any ball bearing or roller bearing selected for industrial application has to be minimum 5000 hours which comes out to be approximately 1 year considering 24 hours of working per day for 200 working days.

IV. GEARBOX SYSTEM LIFE PREDICTION MODEL

Gearbox internal forces, rotations of gear mesh, shafts, bearings are a complex phenomenon and can't be simulated in one model. Although there are industry standards like American Gear Manufacturers association (AGMA) and American Bearing Manufacturers association (ABMA) to evaluate life of gears and bearings individually, there is no standard that combines all these in to a single life prediction model. Based on the research done on this subject and the testing done, it is concluded that there are two life prediction models possible. One based on weakest component life assuming normal working conditions and other based on application where the gearbox is used[3], [7]

A. *Life prediction model based on weakest component life*

In this prediction model, life doesn't depend upon the application conditions as the worst-case scenarios are simulated within the model. However, it is assumed that gearbox is used in normal working conditions without shock/impact loads with normal duty cycles. Weakest component in current worm gearbox is bearing under no lubrication condition and under thrust load in worst case conditions. For this prediction these assumptions are made –

i. Selection of Gearbox for normal, peak, shock and impact loads is proper.
ii. Duty cycle of gearbox is considered during selection.
iii. Operating temp. is within range (-30 to 100 deg, C)
iv. No leakage of oil observed during working temperature.
v. No contaminants are observed during working.
vi. Next weak component is the coupling, coupling life depends upon flexible material used like rubber or steel. As per John. A. Mancuso, [24] Lovejoy coupling is elastomeric compression type coupling with rubber material has a life of about 3 to 5 years (refer annexure-3) under moderate speeds, torques and misalignments. Excessive misalignments reduce life of couplings further.
vii. Gear pair is designed for 3 to 5 years, however a gear wear factor of about 0.6 to 0.9 is considered for worst to normal
workingconditions.

B. *Life prediction model based on application*

In this model, life prediction changes significantly as per application conditions. Weakest component depends upon application that the gear box is subjected to and it can be anything from coupling, seal, gear-pair or bearing. For example, if there are excessive contaminants in lubricant then the contamination factor can go down up to 0.1 as shown in table II and bearing life is reduced significantly. On the other hand if the conditions are similar to lab environment then the bearing life can be high as calculated for the test set-up in experiment.

C. *Comparison of Gear Box life prediction models*

Table VII Comparison of Gearbox life predicting model

Weakest Component-based model	Application based model
Application details are not needed for life prediction as worst-case situations are used for calculations.	Application details should be known for accurate estimation.
Less accurate as worst-case conditions are assumed.	More accurate as actual application conditions are used to calculate the life.
Easy, quick and cost effective	Time consuming and might be costly as field data needs to be captured.
More conservative, hence often gives lesser life	More practical, hence often gives more life.
Can be used for initial ballpark estimates before buying Gearbox	Can be used for remaining life prediction for already installed Gearbox.

V. CONCLUSIONS

From the calculations using actual parameters of gearbox system and the worst case assumptions, gearbox lives can be predicted. However the accuracy of prediction depends upon actual application conditions that the gearbox is put in to industry. Hence two life prediction models are proposed. Weakest component based model can be used in catalogues by gearbox manufacturers clearly stating the assumptions and application based life prediction can be used by condition monitoring/services functions where actual field data can be captured using data acquisition systems.

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