Journal of Innovative Research in Engineering and Science 1(1), October, 2010
Maiden Edition

Gearbox Failure Analysis *(pp. 62-74)*

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Abstract: An analysis of the failures of gearbox units after about 10,000 service hours when the design service life of 45,000 hours was expected was performed to establish the cause of failure. The gear tooth characteristics, namely, beam strength, maximum allowable dynamic load, allowable static load and limiting tooth wear load were analysed for ability to cope with the duty load. The failed gear components were examined visually and metallurgically. Lubrication oil used in the gearboxes was examined for consistency with manufacturer's standards. The observed failures were due to design and manufacturing errors: for instance, the duty load on the gear teeth exceeded the maximum tooth beam strength, allowable dynamic load and limiting wear load. The out-of-tolerance surface finish also contributed as it resulted in the misalignment of pinions and gears. To prevent these failures required adequate sizing of the gear teeth and face width as well as better choice of materials with higher allowable static stresses to bear the loads imposed. The surface roughness should be within tolerance.

Key words: gearbox, failure analysis, gear, gear-tooth strength, beam strength

NOTATIONS

ASME	American Society of Mechanical Engineers
AGMA	American Gear manufacturers Association
b	Gear Face width in mm
BHN	Brinell Hardness Number
C	Deformation or Dynamic Factor
Cs	Service Factor
Cv	Velocity Factor
E_G	Young's Modulus for Gear Material
ISO	International Standards Organization
K	Load Stress Factor
m	Module in mm
N	Speed in rpm
P	Power Transmitted in watts
Q	Gear Ratio or Velocity Factor
T	Torque Transmitted in N-m
T_E	Formative or Equivalent Number of Teeth for the Pinion
T_P	Number of Teeth on the Pinion
v	Pitch Line Velocity
VR	Velocity Ratio
W_D	Total Dynamic Load

W_I	Increment Load due to dynamic action
W_S	Static Load
W_T	Steady Load due to Transmitted Load
W_W	Maximum or limiting wear tooth load
y'	Tooth form factor or Lewis Factor

INTRODUCTION

The gearbox is a speed-reducing gear unit. It consists of a 16-tooth helical pinion (driver) meshing with the driven gear. The driven gear has a nominal rotational speed of 530 rpm. Both pinion and gear are case-hardened and ground: it was the pinions of this train which in most cases failed. The electric motor operates at 1500 rpm, direct coupled to pinion shaft. Tables 1 and 2 show tested gearbox units technical and running data respectively.

Ten gearboxes, each with a design service life expectancy of five years (i.e., 45,000 run hours) were installed in a gas re-injection facility. Unexpectedly, less than eighteen months later (i.e., after about 10,000 run hours), the gearboxes became noisy and began to fail. Eight of the gearboxes eventually failed and were replaced, and in the subsequent two years, four of the replacement ones have also failed. Thus, out of eighteen gearboxes installed, twelve failed in service, i.e., well short of their design life expectancy. The phenomenon aroused interest which led to this study.

2 DESIGN REQUIREMENTS FOR HELICAL GEAR-DRIVE

2.1 Design Requirements for Helical Gear-Drive

The gear teeth should be able to transmit the maximum power required; posses sufficient strength so that they will not fail under static or dynamic loading during normal running conditions; and have wear characteristics so that their lives are satisfactory. The alignment of the gears and deflections of the shafts must be considered because they affect the performance of the gears. The lubrication of the gears must be satisfactory (Khurmi and Gupta, 2006).

2.1.1 Gear-Tooth Strength or Beam Strength (W_T)

Calculations will be based on ISO 6336 Part 1 Standards. The failures have all occurred on the pinions. As the pinion and gear are manufactured from the same material, subsequent calculations will be based on the pinion, which is the weaker component. The transmitted torque, T is given by (Khurmi and Gupta, 2006):

$$T = \frac{P \text{ x } 60}{2 \text{ x } \pi \text{ x NP}} \qquad (1)$$

where: T = *Torque transmitted in Nm*
P = Power transmitted in Watts

N_p = Pinion speed in r.p.m.

The duty tangential tooth load (service load) can be obtained from the power transmitted and the pitch line velocity by using the following relation (Khurmi and Gupta, 2006):

$$W_T = \frac{P}{*v} \times C_s \qquad (2)$$

Where W_T = Permissible tangential tooth load (N)
P = Power transmitted (Watts)

$$*v = \text{Pitch line velocity (m/s)} = \frac{\pi D.N}{60} \qquad (3)$$

D = Pitch circle diameter (m),

N = Speed (r.p.m.)

C_s = Service factor.

The pinion pitch velocity, $V_p = \frac{\pi D_p N_p}{60}$ The applicable equation is the modified Lewis equation (Khurmi and Gupta, 2006), it is given by:

$$W_T = (\sigma_o \times C_V)\, b.\pi.m.y' \qquad (4)$$

Where: W_T = Tangential tooth load (N)

σ_o = Allowable static stress N/(mm)^2

C_V = Velocity factor

b = Face width (mm)

m = Module in mm

y' = Tooth form factor or Lewis factor (Khurmi and Gupta, 2006), corresponding to the formative or virtual or equivalent number of teeth T_E.

From standard gear design tables, the table for values of allowable static stress (σ_o) we obtain the value of σ_o

Note: The allowable static stress (σ_o) for a steel gear is approximately one-third of the ultimate tensile strength σ_u i.e., $\sigma_o = \frac{\sigma_u}{3 . \sigma_o}$ for Cast Steel will be used in the calculations because this is the material grade identified in the gear-material compositional analysis. T_E is the formative or equivalent number of teeth and the governing relationship is (Khurmi and Gupta, 2006):

$$T_E = \frac{T_p}{cos^3 \alpha} \quad (5)$$

$$C_V = \left(\frac{15}{15+v}\right) \quad (6)$$

For peripheral velocities (v) from 10 m/s to 20 m/s,

$$C_V = 0.5441$$

The value of y′ in terms of the number of teeth may be expressed as follows:

$$y' = 0.175 - \left(\frac{0.841}{T_E}\right) \quad (7)$$

2.1.2 Gear-Tooth Dynamic-Load

The dynamic tooth-load on helical gears is given by the Buckingham equation, (Buckingham, 1949):

$$W_D = W_T + W_I \quad (8)$$

Where: W_D = Total dynamic load (N)

W_T = Steady load due to transmitted torque (N)

W_I = Incremental load due to dynamic action (N)

Due to other variables influencing the W_I, the above equation reduces to (Buckingham 1949):

$$W_D = W_T + \left(\frac{21v\,(b.C\,Cos^2\alpha + W_T)cos\alpha}{21v + \sqrt{b.C\,cos^2\alpha + W_T}}\right) \quad (9)$$

where:

v = Pitch line velocity (m/s)

b = Face width of gears (mm)

C = A deformation or dynamic factor (N/mm).

C, depends upon the error, e, in action (mm), between the teeth, the method of cut for the gear, the tooth form, and the material of the gears. The maximum allowable tooth error (e) in action, also depends upon the pitch line velocity (v) and the method of cut of the gears. From engineering handbook on values of deformation factors, C is obtained as 476 for tooth error (e) in action of 0.04 mm (the pinion and gear are made of steel).

2.1.3 Gear-Tooth Static-Load

The static tooth load or endurance strength of the tooth is given (Buckingham, 1949) by:

$$W_S = \sigma e \times b \times \pi \times m \times y' \quad (10)$$

For steel, the flexural endurance limit, σ_e (in $N/(mm)^2$), may be obtained [Buchingham,1949] by using the following relation:

$$\sigma_e = 1.75 \times B.H.N. \quad (11)$$

2.1.4 Gear-Tooth Maximum or Limiting Wear Load

This is given as (Buckingham, 1949):

$$W_W = \frac{D_P \times b \times Q \times K}{Cos^2\alpha} \quad (12)$$

where W_W = Maximum or limiting load (N) for wear

D_P = Pitch circle diameter (mm) of the pinion

b = Face width (mm) of the pinion

Q = Ratio factor

$$= \frac{2 \times V \times R.}{V \times R - 1} = \frac{2 \times T_G}{T_G - T_P} \quad (13)$$

For internal gears

V.R. = Velocity ratio = $\frac{T_G}{T_P}$

K = Load stress factor (material combination factor) (N(mm)^2).

According to Buckingham, (Buckingham, 1949) the load stress factor, K, is given by the relation:

$$K = \left[\frac{(\sigma_{es})^2 \sin \phi N}{1.4}\right]\left[\frac{1}{E_P} + \frac{1}{E_G}\right] \quad (14)$$

where: σ_{es} = Surface endurance limit N/(mm)^2,

ϕ_N = Normal pressure angle

E_P = Young's modulus for the material of the pinion N/(mm^2)

E_G = Young's modulus for the material of the gear in N/(mm)^2.

The surface endurance limit, σ_{es} N/(mm)^2) for steel may be obtained from:

$$\sigma_{es} = (2.8 \times \mathrm{B.H.N.} - 70)\ \frac{\mathrm{N}}{\mathrm{mm}}^2 \quad (15)$$

3 METHODOLOGY

The gear-tooth characteristics: tooth beam strength, maximum allowable dynamic load on each tooth, allowable static load, and limiting tooth wear load were analysed for ability to carry the duty transmission load imposed on the gear unit. The failed gear-components were examined visually and metallurgically. The lubrication oil before and after use were examined and compared with recommendations and standards. The major limitation to this investigation was that overload and safety factors used by the manufacturer in the design of the gearbox units were not available.

Table 1: Tested Gearbox Technical Data.

Power transmitted by the pinion	P	= 270 kW
Speed of pinion (Driver)	Np	=1500 rpm

Speed of gear (Driven)	N_G	= 530 rpm
Gear ratio	G	= 2.83
Velocity ratio	VR	= 2.83
Number of teeth on the pinion	Tp	=16
Pitch circle diameter of pinion	Dp	=160 mm
Pressure angle (stub involute)	φ	= 20^0
Helix angle	α	= 25^0
Gear face width	b	= 125 mm
Pinion/Gear material	40Ni2CrMo28, Case Hardened	

Table 2: Gearbox Units Running Data

Gear Case Study Unit	**Run Hours**	**Number of Starts**	**Behaviour**
1	10,880	40	4 Electrical Trips (no. overload trips)
2	12,332	18	11 Trips (1 overload trip)
3	18,720	26	19 Trips (4 overload trips)
4	9,504	15	6 Trips (1 overload trip)
5	10,780	25	3 Electrical Trips (no. overload trip)
6	17,110	29	25 Trips (4 overload trips)
7	14,889	17	6 Trips (5 electrical and 1 overload)
8	10,293	31	6 Trips (2 stalled electric motors, 2 single-phase, 2 overload)

3. DESIGN REQUIREMENTS FOR HELICAL GEAR-DRIVE

The gear teeth should:

- Be able to transmit the maximum power required;
- Posses sufficient strength so that they will not fail under static or dynamic loading during normal running condition and

- have wear characteristics so that their lives are satisfactory.The alignment of the gears and deflections of the shafts must be considered because they affect the performance of the gears.
- The lubrication of the gears must be satisfactory (Khurmi and Gupta, 2006).

4 COMPUTATIONS FOR GEAR PARAMETERS

The appropriate relations of section 2 are used with values of tables 1and 2to evaluate the gear function parameters using equations 1-15 and presented in tables 6-9.

Design Factors for the Gears

To remedy the failures from the stand-point of design, it is good to match the duty load with the tooth's critical specifications. We shall stick to the material identified in the compositional analysis performed (cast-steel: case hardened). This is because the choice of different materials with higher allowable static–stresses, e.g., steel grade 35Mn2Mo28 or 35NiCr60 may have cost implications.

5 VISUAL AND METALURGICAL EXAMINATIONS

The pinion teeth, with debris for different failed gearbox units, were taken for visual and metallurgical examinations.

Visual Examination: The Pinions

- Contact ensued predominantly towards the "lead-in" edges of the teeth.
- Evidence of "tooth-tip interferences" was seen (See Figures 1).
- Outside of the contact area, and in particular towards the "lead–out", the surface roughness of the gear teeth became coarser.

Metallurgical Examination

- A chemical composition analysis (see Table 8) showed that the cast-steel gears were different in composition from that of 40Ni2CrMo28 specified**.**
- Metallurgical examination of pitted regions on the profile identified fatigue cracks, which extended into the bulk of the material.

  Martensitic microstructure was observed in the surface-hardened region, whilst a bainitic microstructure was evident in the core.

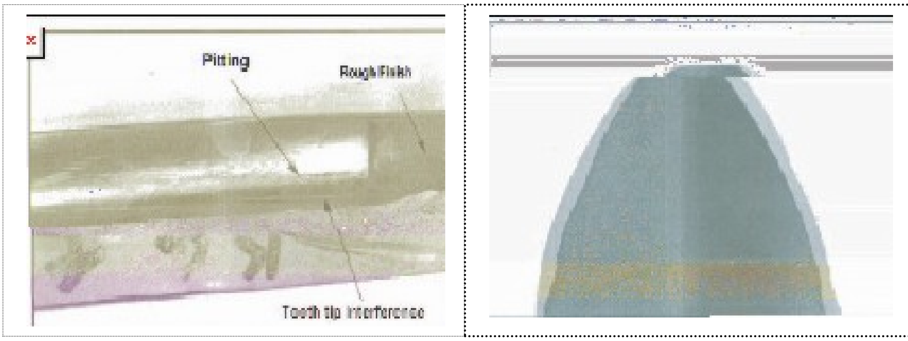

Figure 1: Evidence of "tooth-tip *interferences*

Figure 2: Martensitic microstructure

Table 6: Typical Results for Gearbox units 1 to 8

Gear Unit Parameter	Permissible Gear- Tooth Design Load (kN)	Minimum Required Permissible Design Load for Acceptable Service Life (kN)	Comments
Gear Tooth Beam Strength, W_T	29	33	Gear tooth over loaded. Responsible for gear tooth fracture. Inefficient design.
Gear Tooth Dynamic Load, W_D	63	68.7	Gear tooth over loaded. Inefficient design
Gear Tooth Static Load, W_S	228.9	228.9	Acceptable
Gear Tooth Maximum Wear Load, W_W	63.9	> 68.7	Inefficient design. Led to pre-mature wear.
$W_S \geq 1.25\ W_D$	228.9	85.9	Meets requirements
$W_W > W_D$	63.9	> 68.7	W_W is less than W_D . W_W should be > 68.7 kN

Table 7: Chemical Composition of Gear Teeth Materials for Gear Box Components

Element	Gearbox Material (% by mass)	40Ni2CrMo28 Material (% by mass)
Carbon	0.35	0.35-0.45
Silicon	0.30	0.10-0.35
Manganese	0.01	0.40-0.70
Sulphur	0.006	<0.035
Nickel	0.00	1.25-1.75

Chromium	0.00	0.90-1.30
Molybdenum	0.00	0.20-0.35
Iron	Remainder	Remainder

6 CONCLUSIONS

- Design errors were responsible for gear tooth overloading and tooth tip interference.
- Manufacturing errors led to misalignment due to gear profile mis-match, and created uneven loading of the gear teeth.
- The gears have premature failure due to surface-contact fatigue.
- The results of the chemical compositional analysis (shown in Table 8) of the gears were
  different from the grade specified, i.e., cast-steel composition rather than the 40Ni2CrMo28 specified.
- No evidence of lubrication deficiencies were observed with respect to the gears contacting surfaces.
- From the calculations done in this work, the recommended values are set out in table 9 and table 10 below:The surface roughness Ra for the gear tooth face should be < m, and for the tooth□0.5 flank < m.□0.6

Table 8: Recommended Gear Unit Permissible Tooth-Load for Acceptable Service Life

Gear Unit Parameter	Permissible Design Value (kN)	Duty Load Value (kN)	Recommended (based on calculations) Permissible Design Value for Acceptable Service Life (kN)
Gear Tooth Beam Strength, W_T	29	33	41.8
Gear Tooth Dynamic Load, W_D	63	68.7	87

Gear Tooth Static Load, W_S	210.5	210.5	228
Gear Tooth Maximum Wear Load, W_W	60.4	68.7	76.7
$W_S \geq 1.25\ W_D$	210.5	85.9	108.8
$W_W > W_D$	60.4	> 68.7	76.7 $W_W > W_D$ as required.

Table 9: Recommended Values of Vital Gearbox Units Parameter

Gear Unit Parameter	Permissible Design Value	Duty Load Value	Recommended (based on calculations) Permissible Design Value for Acceptable Service Life
Gear Module (mm)	10		12
B.H.N.	245		270 For adequate surface wear resistance.
Gear Face Width (mm)	125		150
Gear-Tooth face Surface Roughness (µm)	0.70	0.20 – 0.60	0.20 – 0.60 Tolerance for helical gears for v > 12 m/s.
Gear-Tooth flank Surface Roughness (µm)	2.05	0.20 – 0.70	0.20 – 0.70 Tolerance to avoid gear profile mis-match.
Minimum Teeth Number on Pinion to avoid Interference	16		17, To avoid tooth tip interference

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