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# 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):

$$T = \frac{P \times 60}{2 \times \pi \times NP} \quad (1)$$

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)

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

$\sigma_o$ = 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 ($\sigma_o$) we obtain the value of $\sigma_o$

Note: The allowable static stress ($\sigma_o$) for a steel gear is approximately one-third of the ultimate tensile strength $\sigma_u$ 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, $\sigma_e$ (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: $\sigma_{es}$ = Surface endurance limit $N/(mm)^2$,

$\phi_N$ = 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, $\sigma_{es}$ $N/(mm)^2$) for steel may be obtained from:

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

## 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.

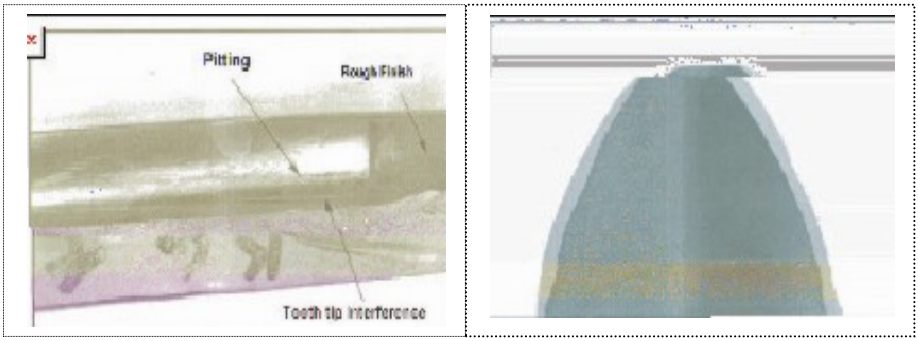



**Figure 1: Evidence of "tooth-tip *interferences***



**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<br>$W_W > W_D$ 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<br>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<br>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<br>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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