Pinion Failure Analysis of a Helical Reduction Gearbox in a Kraft Process

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Abstract: This paper reports investigations related to addressing the cause of pinion teeth deformation of a helical reduction gearbox in a kraft process. The American Gear Manufacturers Association (AGMA) design methodology was employed to determine the safety factor under bending and surface fatigue strengths of a pinion and gear at two operating loads (3 and 3.75 MW). In addition, finite element analysis (FEA) of the pinion and gear assembly was also performed to check the misalignment (due to deformation) at 3.75 MW load. Based on the investigations presented herein, it is found that the pinion portion near the thrust disc became excessively deformed at the axial thrust corresponding to the 3.75 MW load, causing misalignment that resulted in the plastic deformation of the pinion teeth.

Keywords: helical reduction gearbox; AGMA design methodology; pinion teeth deformation; safety factor; finite element analysis

1. Introduction

Gearboxes are widely employed for mechanical power transmission in automobiles, windmills, machine tools, plant machinery, and many other mechanical systems. Enhancement and reduction of the speed of the output shaft are achieved using gearboxes at reduced and increased torques, respectively. From the economical perspectives, the efficient and reliable functioning of gearboxes of machinery is extremely vital. The failure of gearboxes in industrial plants can cause malfunction in machinery, which leads to production loss, a costly affair in terms of loss of profit. It also involves the additional cost of repair/replacement of failed gears. It is worth noting here that gear failure may occur due to design error, manufacturing error (e.g., poor machining, faulty heat treatment of parts, poor gear-set assembly), or improper installation and/or operation. Design errors mainly include improper gear geometry, poor selection of materials, and inadequate lubrication arrangement [1,2]. However, improper installation and operation involve faulty mounting, inadequate cooling, improper lubrication, and poor maintenance [3–6].

When errors are present, as described in the previous paragraph, gears fail and show the symptoms of scoring, wear, pitting, plastic flow, and teeth breakage/fracture. Scoring occurs because of lubrication failure, which causes asperity-to-asperity contact between the teeth of the pinion and gear [7–9]. This results in micro-welding at the tip of the asperity followed by tearing, which leads to the continuous and rapid removal of material from the teeth surface. However, wear is a type of damage that occurs owing to the progressive removal of material from the interacting surfaces. Material removal in wear may be due to adhesion, abrasion, corrosion, or a combination of these. Wear causes increased surface roughness, enhanced clearance at the mating interface, and weakened teeth. Pitting occurs at the gear teeth surface because of repeated loading that results in the contact stress exceeding the surface fatigue strength of the material. Pitting failure takes place over millions of
cycles of gear revolutions. During the pitting process, if material is removed from the mating surfaces of teeth in the form of a flake, it is called flaking/spalling. High contact stresses at the interfaces of teeth under rolling/sliding motion can cause the plastic flow of teeth surfaces as well. This type of failure is normally found in softer gear materials. Moreover, teeth fracture/breakage also takes place because of high overload arising from either impact loading or static load itself [10,11].

In many situations, the inspection of failed parts and analysis of data do not provide information regarding the cause of gear failure. In this case, gear design checks and laboratory tests are required to develop an understanding on the probable cause(s) of failure. It has been reported that if failure is influenced by the gear geometry, then checking for design and metallurgical defects should be performed. It is always recommended to perform non-destructive tests (e.g., measuring the surface hardness and roughness, magnetic particle inspection, acid etch inspection, gear tooth accuracy inspection) before performing any type of destructive tests (e.g., micro-hardness measurement, microstructural determination using acid etches, determination of grain size, inspection for non-metallic inclusions, SEM microscopy) on failed gears/components. Over the past several years, analyses have been conducted on the failure of gears of different mechanical systems. It has been reported that the majority of gear failures have occurred because of manufacturing and operational error of gears and gearboxes, respectively [12–18]. However, bending fatigue [19,20], wear and surface contact fatigue [21,22], and design faults [23,24] have also contributed significantly to failure.

Based on the literature review, it is found that there is no published work regarding the failure analysis of high-speed helical gearbox (pitch velocity of pinion > 55 m/s) having provision of axial thrust sustaining between a disc mounted on the pinion and the blank of gear. Therefore, the objective of this paper is to investigate the failure of pinion teeth of a helical reduction gearbox used in kraft process. It is understood that the information provided in this paper may be useful to practising engineers and researchers.

2. Pinion Failure and Data Collection

A single-stage helical reduction gearbox was employed in a kraft process plant to reduce the speed (8350 rpm) of a steam turbine to operate an alternator at a speed of 1500 rpm, such that electricity (3 MW) would be generated for use in the plant. This corresponds to pinion and gear speeds of 8350 rpm and 1500 rpm, respectively. The gearbox was intended to operate continuously for 20 years (excluding break periods for maintenance). The data pertaining to the pinion and gear were measured/collected at the site of gearbox failure, which are listed in Table 1. The photographic view of the helical gearbox is shown in Figure 1 along with the names of key components. However, Figure 2a shows the key dimensions of the pinion and gear of the helical gearbox. Moreover, the photographic view of pinion is also shown in Figure 2b for its geometrical visual to the readers. Because of the requirement of greater electricity generation in the plant, the reduction gearbox was operated at 3.75 MW (instead of 3 MW) with the permission of the gearbox designer. However, when the gearbox was operated at high load (3.75 MW), the pinion teeth became plastically deformed near the thrust disc. The photographic views of healthy and failed pinions are shown in Figure 3a,b, respectively. Thus, it was vital to examine the cause(s) of failure of the pinion teeth.
Table 1. Configuration data of pinion and gear collected at the site of the gearbox failure.

| S. No. | Parameters                              | Pinion | Gear  |
|--------|----------------------------------------|--------|-------|
| 1      | Helix angle (°)                        | 15.6   | 15.6  |
| 2      | Pressure angle (°)                      | 20     | 20    |
| 3      | Face width (mm)                        | 190    | 190   |
| 4      | No. of teeth                           | 20     | 111   |
| 5      | Pitch circle diameter (mm)             | 131.9  | 732   |
| 6      | Dia. of dedendum circle (mm)           | 116    | 716.1 |
| 7      | Dia. of addendum circle (mm)           | 144.6  | 744.7 |
| 8      | Tooth thickness at root (mm)           | 14     | 16    |
| 9      | Tooth thickness at PCD (mm)            | 12     | 14    |
| 10     | Material of gears and shaft            | 17CrNiMo6 | 17CrNiMo6 |
| 11     | Hardness at flank and face (HRC)       | 60 ± 2 | 60 ± 2 |

Figure 1. (a) Photographic view of helical gearbox with names of vital components; (b) photographic view of pinion’s axial thrust sustaining arrangement.

Figure 2. (a) Key dimensions of pinion and gear in schematic view; (b) photographic view of pinion.
Figure 3. (a) Healthy pinion teeth (without deformation) when operated at 3 MW; (b) deformed teeth of pinion near thrust disc when operated at 3.75 MW.

3. Investigation on Gear Design and Alignment

Based on the visual evidence, the type of failure (plastic deformation) that took place in the teeth of the pinion (refer to Figure 3b) revealed that it might have been due to either design error or misalignment. Thus, it was decided to check the safety factor of the pinion and gear from the strength perspective (in bending and contact stress modes) at both loads (3 MW and 3.75 MW) using American Gear Manufacturers Association (AGMA) design methodology [25–27]. Because the failure of pinion teeth occurred at a localised position near the thrust disc (refer to Figure 3b), it was decided to check the misalignment between the pinion and gear at high load (3.75 MW), employing finite element analysis (using ANSYS software). The design and alignment (deformation) investigations are discussed below.

3.1. Investigation on Gear Design Using the AGMA Approach

The design check of a gearset is a reasonably challenging task as it involves the satisfaction of many design constraints. However, it is widely understood that gear teeth may fail through bending fatigue at its root and pitting fatigue on its surface. Thus, it was decided to check the safety factor against bending and pitting of pinion and gear, mainly at high load 3.75 MW employing AGMA standards. AGMA has proposed the following relations for finding the safety factor under bending and surface fatigues [25–27]:

AGMA bending stress relation:

\[
\sigma_b = \frac{W_f}{F_{mf}} \cdot \frac{K_a}{K_v} \cdot K_s K_b K_I
\]

AGMA bending fatigue strength relation:

\[
\sigma_{fb} = \frac{K_L}{K_T K_R} S_f' b
\]

Safety factor relation for bending fatigue:

\[
N_b = \frac{S_{fb}}{\sigma_b}
\]

AGMA surface stress relation:

\[
\sigma_c = C_p \sqrt{\frac{W_f}{F_{ld}} \cdot \frac{C_a C_m}{C_v} \cdot C_s C_f}
\]
AGMA surface fatigue strength relation:

\[ S_{fc} = \frac{C_L}{C_T C_R} S'_{fc} \]  \hspace{1cm} (5)

Safety factor relation for contact fatigue:

\[ N_c = \left( \frac{S_{fc}}{\sigma_c} \right)^2 \]  \hspace{1cm} (6)

Symbols appearing in Equations (1)–(6) are named in the nomenclature. The design calculations based on AGMA standards are presented in the “Results and discussion” section.

3.2. Investigation on Deformation and Stress Using the FEA Approach

In view of the location of failure of pinion teeth, it was planned to conduct the investigation from the misalignment perspective. Thus, the deformation and stress in meshed gear pairs were investigated using finite element analysis. The geometric models of pinion and gear pairs were prepared in SolidWorks software. Tetrahedron element was used to mesh the model in ANSYS Workbench. A total of 100,710 elements and 173,987 nodes were used to mesh the model pair after performing the mesh independence test. The transmitted loads by pinion and gear were calculated based on the input data. Tangential, radial, and axial loads were applied to compute the deformation and stresses in the static analysis at 3.75 MW. In the analysis, tip-loading (instead of highest point single tooth contact (HPSTC)) conditions were assumed, considering the large magnitude of the face width (> 30 m) of the pinion. Except for the rotation around the shaft axis (i.e., \( x \)-axis), the remaining degrees of freedom (\( U_x, U_y, U_z, ROT_Y \), and \( ROT_Z \)) were removed at the bearing locations.

In the design of this helical gearbox, it can be seen (refer to Figure 1) that the provision for sustaining the axial thrust coming onto the pinion shaft is made through the interface of the thrust disc and gear. The thrust load is transmitted from the pinion to gear and, finally, it is sustained by the thrust washer bearing, mounted on the gear shaft towards the left side. Figure 4a,b illustrate the provision of the axial load transmission. Looking at the load on the thrust disc, it was thought to check the deformation of pinion teeth near the thrust disc at the load arising from 3.75 MW. The axial thrust applied in the analysis (tangential and radial loads are not shown) is illustrated in Figure 4c.
Figure 4. (a) CAD model of meshed pinion and gear; (b) schematic demonstration of axial thrust acting at the interface of gear and thrust disc; (c) magnitude of axial thrust acting at 3.75 MW.

4. Results and Discussion

The design calculations for checking the strengths (against bending and surface pitting) and safety factor of the pinion and gear at two loads are presented in Table 2. Gear design checks were performed using AGMA standards. It can be seen in this table that the safety factor against the bending failure of pinion teeth is less than 1 at high load (3.75 MW). Thus, the pinion teeth are susceptible to failure at high load in the bending mode. However, the gear is safe against bending with a safety factor of 1.31. The teeth interface of the pinion and gear against surface pitting is also safe (factor of safety is 1.27).

The deformations computed based on the finite element analysis of a pinion and gear pair are presented in Figures 5–7. Total deformation is maximum on the pinion teeth towards the thrust disc, as can be observed in Figure 5a. The maximum deformation occurs near 78 µm. Figure 5b,c show the magnified view of deformation contours for better visualisation. However, Figure 5d demonstrates the localised sectional view, indicating the maximum deformation at the tip of the pinion teeth. The comparison of Figures 3b and 5d reveals that the locations of teeth failure and maximum deformation are almost at the same positions. The deformations obtained in the “x-direction” and “y-direction” are also shown in Figures 6a–c and 7a,b, respectively. Based on these results, it is understood that because of poor geometrical conformity of the pinion teeth in comparison to gear teeth, large deformation is a possibility. Moreover, Figures 8 and 9 show the magnitude and location of the calculated contact and bending stresses. The values of the calculated stresses exceed the material strengths of the pinion. This indicates a failure of the pinion in both modes, i.e., bending and surface pitting. It is understood that at high load (i.e., 3.75 MW) excessive deformation in the pinion
teeth in the vicinity of the thrust disc became constrained by the robustness of the gear teeth, which led to the generation of high stresses, causing failure owing to plastic deformation.

Figure 5. (a) Deformation computed using FEA of gearset; (b) isometric view of gearset with deformation contours; (c) magnified view of deformation contours; (d) sectional view of teeth deformation towards thrust disc.
Figure 6. (a) Deformation computed in gearset in x-direction; (b) isometric view of gearset with deformation contours; (c) magnified view of deformation contours.

Figure 7. (a) Deformation computed in gearset in y-direction; (b) isometric view of gearset with deformation contours.
Figure 8. von-Mises contact stress at the interface of pinion and gear teeth.

Figure 9. Isometric view of gear pair illustrating the normal stress in y-direction.

Table 2. Design checks of pinion and gear as per AGMA for assessing the safety factor.

| Steps | Case-I Gearbox Operated at 3 MW | Case-II Gearbox Operated at 3.75 MW |
|-------|---------------------------------|------------------------------------|
| 1     | $N_g = 111, N_p = 20, m_G = N_g / N_p$ | $N_g = 111, N_p = 20, m_G = N_g / N_p$ |
|       | $= 111/20 = 5.55$                 | $= 111/20 = 5.55$                  |
| 2     | Torque on the pinion shaft:       | Torque on the pinion shaft          |
|       | $T_p = P / \omega_p = 3.0 \times 10^6 / (2 \times \pi \times 8350 / 60)$ | $T_p = P / \omega_p = 3.75 \times 10^6 / (8350 \times 6.28 / 60)$ |
|       | $= 3430 \text{N} \cdot \text{m}$  | $= 4290.78 \text{N} \cdot \text{m}$ |
| 3     | Output torque:                    | Output torque:                      |
|       | $T_q = m_G \times T_p = 5.55 \times 3340 = 19,036.5 \text{N} \cdot \text{m}$ | $T_q = m_G \times T_p = 5.55 \times 4291 = 23,815.05 \text{N} \cdot \text{m}$ |
| 4     | Transmitted load:                 | Transmitted load:                   |
|       | $W_i = T_p / (d_p / 2) = 3430 / (0.1319 / 2)$ | $W_i = T_p / (d_p / 2) = 3430 / (0.1319 / 2)$ |
|       | $= 52 \times 10^3 \text{N}$       | $= 52 \times 10^3 \text{N}$         |
| 5     | Velocity factor ($K_v$):          | Velocity factor ($K_v$):             |
|       | Pitch line velocity $V_i = (d_p / 2) \omega_p$ | Pitch line velocity $V_i = (d_p / 2) \omega_p$ |
|       | $= (0.1319 / 2) \times (2 \pi \times 8350 / 60) = 57.6 \text{m/s}$ | $= (0.1319 / 2) \times (6.28 \times 8350 / 60) = 57.6 \text{m/s}$ |
|       | $K_v = (78 / (78 + (200 \times V_i)^{0.5}))^{0.5} = 0.65$ | $K_v = (78 / (78 + (200 \times V_i)^{0.5}))^{0.5} = 0.65$ |
### Table 2. Cont.

| Steps | Case-I Gearbox Operated at 3 MW | Case-II Gearbox Operated at 3.75 MW |
|-------|----------------------------------|-----------------------------------|
| 6     | Various factors: Size factor $K_x = 1.0$ | Various factors: Size factor $K_x = 1.0$ |
|       | Rim thickness factor $K_y = 1.0$ | Rim thickness factor $K_y = 1.0$ |
|       | Load distribution factor $K_m = 1.8$ | Load distribution factor $K_m = 1.8$ |
|       | Application factor $K_a = 1.25$ | Application factor $K_a = 1.25$ |
|       | Idler factor $K_j = 1.0$ | Idler factor $K_j = 1.0$ |
|       | Geometry factor $I_{pinion} = 0.428$ | Geometry factor $I_{pinion} = 0.428$ |
| 7     | Pinion-tooth bending stress: $e_{hp} = (W_t \times p_d / (F \times f))$ | Pinion-tooth bending stress: $e_{hp} = (W_t \times p_d / (F \times f))$ |
|       | $\times (K_m \times K_a \times K_y) / K_j$ | $\times (K_m \times K_a \times K_y) / K_j$ |
|       | $= (72,050 \times 151.63) / (0.19 \times 0.428) \times (2.25 / 0.65)$ | $= (65,060 \times 151.63) / (0.19 \times 0.428) \times (2.25 / 0.65)$ |
|       | $= 335.95 \text{ MPa}$ | $= 419.92 \text{ MPa}$ |
| 8     | Gear tooth bending stress: $e_{gb} = (W_t \times p_d / (F \times f))$ | Gear tooth bending stress: $e_{gb} = (W_t \times p_d / (F \times f))$ |
|       | $\times (K_m \times K_a \times K_y) / K_j$ | $\times (K_m \times K_a \times K_y) / K_j$ |
|       | $= (52,050 \times 151.63) / (0.19 \times 0.61) \times (2.25 / 0.65)$ | $= (65,060 \times 151.63) / (0.19 \times 0.61) \times (2.25 / 0.65)$ |
|       | $= 235.71 \text{ MPa}$ | $= 294.64 \text{ MPa}$ |
| 9     | Length of action: $Z_{pg} = (((r_p + a_p)^2 - (r_p \cos \Phi)^2)^{0.5}$ | Gear tooth bending stress: $e_{gb} = (W_t \times p_d / (F \times f))$ |
|       | $+ (r_p + a_p)^2 - (r_p \cos \Phi)^2)^2 - C_{pg} \sin \Phi$ | $\times (K_m \times K_a \times K_y) / K_j$ |
|       | $= (0.07232 - 0.06959 \times \cos (20.7))^{0.5}$ | $= (0.07232 - 0.06959 \times \cos (20.7))^{0.5}$ |
|       | $+ (0.37232 - 0.366 \times \cos (20.7))^{0.5}$ | $+ (0.37232 - 0.366 \times \cos (20.7))^{0.5}$ |
|       | $- 0.432 \times \sin (20.7) = 0.0312 \text{ m}$ | $- 0.432 \times \sin (20.7) = 0.0312 \text{ m}$ |
| 10    | Transverse contact ratio: $m_{ppg} = p_d \times Z_{pg} / (3.14 \times \cos (20.7)) = 1.6$ | Transverse contact ratio: $m_{ppg} = p_d \times Z_{pg} / (3.14 \times \cos (20.7)) = 1.6$ |
| 11    | Axial contact ratio: $m_f = F / p_d \times \tan \Phi / 3.14$ | Axial contact ratio: $m_f = F / p_d \times \tan \Phi / 3.14$ |
|       | $= 0.19 \times 151.63 \times 0.28 / 3.14 = 2.57$ | $= 0.19 \times 151.63 \times 0.28 / 3.14 = 2.57$ |
|       | $p_x = p \times \cot \Phi = 0.0207 \times \cos 15.67 = 0.074 \text{ m}$ | $p_x = p \times \cot \Phi = 0.0207 \times \cos 15.67 = 0.074 \text{ m}$ |
| 12    | Normal pressure angle and helix angle: $\phi_n = 20^\circ$, $\phi_h = 14.7^\circ$ | Normal pressure angle and helix angle: $\phi_n = 20^\circ$, $\phi_h = 14.7^\circ$ |
| 13    | Min. length of the lines of contact: $n_{pg} = $ Fractional part of $m_{ppg} = 0.6$ | Min. length of the lines of contact: $n_{pg} = $ Fractional part of $m_{ppg} = 0.6$ |
|       | $n_a = $ Fractional part of $m_f = 0.57$ | $n_a = $ Fractional part of $m_f = 0.57$ |
|       | $l_{minpg} = m_{ppg} \times (1 - n_a) / (1 - n_{pg}) / \cos \phi_h$ | $l_{minpg} = m_{ppg} \times (1 - n_a) / (1 - n_{pg}) / \cos \phi_h$ |
|       | $= (1.6 \times 0.19 - (1 - 0.57)(1 - 0.6) \times 0.074)$ | $= (1.6 \times 0.19 - (1 - 0.57)(1 - 0.6) \times 0.074)$ |
|       | $= \cos 14.7 = 0.301 \text{ m}$ | $= \cos 14.7 = 0.301 \text{ m}$ |
|       | $m_{npg} = F / l_{minpg} = 0.19 / 0.301 = 0.63$ | $m_{npg} = F / l_{minpg} = 0.19 / 0.301 = 0.63$ |
| 14    | Radii of curvature of teeth: $\rho_p = ((0.5(r_p + a_p) + (C_{pg} - r_p - a_p))^2$ | Radii of curvature of teeth: $\rho_p = ((0.5(r_p + a_p) + (C_{pg} - r_p - a_p))^2$ |
|       | $- (r_p \cos \Phi)^2)^{0.5} = (0.00435 - 0.003805)^{0.5}$ | $- (r_p \cos \Phi)^2)^{0.5} = (0.00435 - 0.003805)^{0.5}$ |
|       | $= 0.0223 \text{ m}$ | $= 0.0223 \text{ m}$ |
|       | $\rho_g = C_{pg} \sin \Phi - 0.432 \times \sin (20.7) - 0.0233$ | $\rho_g = C_{pg} \sin \Phi - 0.432 \times \sin (20.7) - 0.0233$ |
|       | $= 0.129 \text{ m}$ | $= 0.129 \text{ m}$ |
| 15    | Pitting geometry factor: $I_{pg} = \cos \Phi / ((1 + p / 1 + p) / d_p m_{npg} = 0.935 / (50.67 \times 0.1319 \times 0.63)$ | Pitting geometry factor: $I_{pg} = \cos \Phi / ((1 + p / 1 + p) / d_p m_{npg} = 0.935 / (50.67 \times 0.1319 \times 0.63)$ |
|       | $= 0.222$ | $= 0.222$ |
| 16    | The elastic coefficient: $C_p = (3.14 \times ((1 / \nu_p^2) / E_p + (1 - \nu_p^2) / E_g))^{-0.5}$ | The elastic coefficient: $C_p = (3.14 \times ((1 / \nu_p^2) / E_p + (1 - \nu_p^2) / E_g))^{-0.5}$ |
|       | $= 191.63$ | $= 191.63$ |
| 17    | Surface stress at mesh: $\sigma_{cp} = C_p (W_{tC} C_{Cm} C_{Cg} / (F_{tpd} d_p C_p))^{0.5}$ | Surface stress at mesh: $\sigma_{cp} = C_p (W_{tC} C_{Cm} C_{Cg} / (F_{tpd} d_p C_p))^{0.5}$ |
|       | $= 191.63 ((52,050 \times 1.25 \times 1.8 \times 1.0 \times 1.0) / (0.19 \times 0.222 \times 0.1319 \times 0.65)^{0.5}$ | $= 191.63 ((65,060 \times 1.25 \times 1.8 \times 1.0 \times 1.0) / (0.19 \times 0.222 \times 0.1319 \times 0.65)^{0.5}$ |
|       | $= 1090.5 \text{ MPa}$ | $= 1219 \text{ MPa}$ |
Steps | Case-I Gearbox Operated at 3 MW | Case-II Gearbox Operated at 3.75 MW
--- | --- | ---
18 Corrected bending-fatigue strength: $S_{fb}') = 6,235 + 174HB - 0.126HB^2$ | Corrected bending-fatigue strength: $S_{fb}') = 6,235 + 174HB - 0.126HB^2$
$6235 + 174 \times 600 - 0.126 \times 600^2$ | $6235 + 174 \times 600 - 0.126 \times 600^2$
$65,275 \times 6890 = 450 \text{ MPa}$ | $65,275 \times 6890 = 450 \text{ MPa}$
Service life = 20 years, continuous run, Operating temperature = 70°C | Service life = 20 years, continuous run, Operating temperature = 70°C
No. of cycles during service $= 8350 \times 20 \times 365 \times 5 \times 24 \times 60 = 8.78 \times 10^{10}$ | No. of cycles during service $= 8350 \times 20 \times 365 \times 24 \times 60 = 8.78 \times 10^{10}$
Life factor $K_L = 1.3558(8.78 \times 10^{10})^{-0.0178} = 0.86$ | Life factor $K_L = 1.3558(8.78 \times 10^{10})^{-0.0178} = 0.86$
Temperature factor $K_T = 1.0$ | Temperature factor $K_T = 1.0$
Reliability factor $K_R = 1.0$ | Reliability factor $K_R = 1.0$
Corrected bending fatigue strength, $S_{fb} = (K_L \times S_{fb}')/(K_T \times K_R)$ | Corrected bending fatigue strength, $S_{fb} = (K_L \times S_{fb}')/(K_T \times K_R)$
$= (0.86 \times 450) / (1 \times 1) = 387 \text{ MPa}$ | $= (0.86 \times 450) / (1 \times 1) = 387 \text{ MPa}$
19 Corrected surface-fatigue strength: $S_{fc}' = 27,000 + 364 \times HB$ | Corrected surface-fatigue strength: $S_{fc}' = 27,000 + 364 \times HB$
$= 27,000 + 364 \times 600$ | $= 27,000 + 364 \times 600$
$= 245,400 \times 6890$ | $= 245,400 \times 6890$
$= 1691 \text{ MPa}$ | $= 1691 \text{ MPa}$
$C_L = 1.4488(8.77 \times 10^{10})^{-0.023} = 0.81$ | $C_L = 1.4488(8.77 \times 10^{10})^{-0.023} = 0.81$
$C_T = 1.0, C_R = 1.0, C_{RH} = 1.0$ | $C_T = 1.0, C_R = 1.0, C_{RH} = 1.0$
$S_{fc} = C_L \times C_{RH} \times S_{fc}'/(C_T \times C_R)$ | $S_{fc} = C_L \times C_{RH} \times S_{fc}'/(C_T \times C_R)$
$= 0.81 \times 1691 = 1370 \text{ MPa}$ | $= 0.81 \times 1691 = 1370 \text{ MPa}$
20 Safety factor against bending failure: $N_{bpinion} = 387/335.95 = 1.15$ (Safe) | Safety factor against bending failure: $N_{bpinion} = 387/419.92 = 0.91$ (Unsafe)
$N_{bgear} = 387/235.71 = 1.64$ (Safe) | $N_{bgear} = 387/294.64 = 1.31$ (Safe)
21 Safety factor against surface failure: $N_{fpinion} = (1370/1090.5)^2 = 1.57$ (Safe) | Safety factor against surface failure: $N_{fpinion} = (1370/1219)^2 = 1.26$ (Safe)

5. Conclusions

Based on the design and deformation checks presented herein, the following conclusions are drawn:

- The AGMA design approach yielded a sufficient factor of safety against the bending and surface pitting in the pinion at the 3-MW load. However, the factor of safety against bending fatigue strength was reduced to 1 at the load of 3.75 MW, which is unsafe.
- The FEA results show large deformation in the pinion teeth at the location of the thrust disc. It is understood that a portion of the pinion teeth near the thrust disc is more loaded because of constrained deformation of the pinion teeth by the gear. This led to an increase in stresses, resulting in the plastic deformation of teeth.
- The provision made for sustaining the axial thrust between the thrust disc and gear is not a sound method. In place of this arrangement, a thrust bearing should have been employed. It is worth noting here that the load carrying capacity of the bearing formed between two parallel surfaces (gear and thrust disc) is always poor, because effective film formation does not take place without a physical wedge.
- The pinion and thrust disc should be redesigned for sustaining the enhanced loads to avoid deformation in the pinion teeth at a 3.75-MW load.

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