FAILURE ANALYSIS FOR HIGH STRESS GEAR

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Abstract-Gear failure is one of the most headache problems in the industry particularly when it is occurred during the critical operation season of the industrial systems. One of these situations was occurred during the season of sugar industry. The gear was manufactured from carburized steel DIN standard No.1.6587 (18CrNiMo7-6). The gear, module is 5.08 mm, pressure angle is 20º, No of teeth is 16, width is 85 mm and helix angle is 8.5º Right with mating gear No of teeth is 70 teeth. The failure was occurred within 1 month of continuous operation. The gear is driven by 315 kW motor at 1450 rpm. A visual investigation has been carried out, a set of photos were taken and the working conditions were reviewed. The visual investigation showed that the pinion has undergone some broken teeth at the pinion and pettings in the mating gear. The operating conditions have been revised during consecutive intervals of operation. The design data has been reviewed according to standard gear design recommendations. Material analysis and micro hardness measurements followed by a micro structure have been also investigated. The micro hardness survey through the teeth tips and roots revealed that the carburization case depth was not adapted with the standards recommendations. The case depth was, 2.3 mm at teeth tip and 1mm at teeth root, and that is not suitable and more than the standards recommendations to the gears of module 5.08 mm. The micro hardness measurements and micro structure ensured that the case hardness at gear tips was larger than the standards values and not conformed with the module which resulted a full hardening at the teeth tip portion make it brittle. This refers to an excessive carburizing time. Also the big difference between case depth at the teeth, top and the root, occurred due to bad circulation of the salt bath around the pinion. Under, the high cyclic bending stress $\sigma_F = 400$ N/mm^2 and contact stress $\sigma_H = 1300$ N/mm^2 at the pinion teeth while the design values are 259 N/mm^2, 723 N/mm^2 respectively so the teeth started to fail. Thick layer of the carburized case may lead to more probable brittle case at teeth tips which lead to cracks initiation. Therefore, it is recommended to select the carburizing parameters matching with standards recommendations of the carburizing layer with correct case depth and maintaining its thickness difference at both the teeth tips and roots at minimum as possible. Also high stress required to redesign the gearbox again for suitable stresses levels according design standards.
Keywords – Gear failure, Sugar industry, high stress gears, Case depth, Hardness, 18CrNiMo7-6.

1. INTRODUCTION

Gear failure is one of the most serious problems causing a headache to gears designers, manufacturers and users. The industrial processes do not need an unplanned shut down due to any reasons because it cost a lot of money and waste of time and effort. Gear failure have many reasons to occur; it may happen due to a lot of error sources in gear process production such as: design, manufacturing, heat treatment, maintenance or random so, the problem of gear failure must be analyzed, investigated, and clearly solved for avoiding its bad effects in the operation.

The present case investigates a pinion gear failure in sugar industry which was subjected to a high stress due to, high power drive motor 315 kw, single stage reduction of 16/70, module 5.08 and width 85 mm. the gear box is driving a circulation fan which is critical in the industry process. This circulation fan must work 24 hrs / day. The environmental and operating conditions were reviewed carefully searching for corrosion, erosion or overload causes. The routine maintenance was carried out. Material selection and heat treatment specifications were analyzed and compared to gear standard design recommendations. Visual inspection and a set of photos were taken for documenting the failure case as shown in figures (1, 2, 3). Micro hardness survey was carried out to the teeth and the area between teeth at the root sections. Also the gear microstructure was investigated looking for the microstructure and compare it with the target microstructure of the standards. The contact stress and bending stress on the teeth are calculated regarding the operating conditions. The gear design criteria were reviewed concerning the design standards procedures and recommendations. The results of the above investigations were discussed concerning the design and standards requirements. Also a comparison with other works was done. Finally conclusion and recommendations were drawn.

2. LITERATURE SURVEY

For avoiding failure; the gear loads and stresses must be calculated correctly and design parameters such as the dimensions, accuracy, material, heat treatment and finishing specifications to be selected according to the standards and design guidance

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literatures. The gear design specifications are identified, applied and reviewed before it launched. Also manufacturing specifications should be selected accurately for verifying the design target data.

Rossino et al [1] found that the surface contact fatigue failure was resulted from the excessive carburizing case depth which led to an intergranular carbon-rich brittle cementite. This causes a crack initiation when contact stress is applied with cyclic load followed by crack propagation until intergranular fracture occurred.

Saber [2] observed that although the material and the heat treatment have satisfied the recommended values but it was noticed that the pinion which subjected to more cyclic load had case hardness less than the gear. Pinion is subjected to more cyclic loads. So, the hardness selection may be done for safety or for replacing the smaller one at failure but under fatigue, the pinion teeth were failed and the broken fragments start to initiate pitting and cracks at both pinion and gear teeth.

Starzhinskii et al [3] stated that, to increase the gear surface life, there are many parameters must be controlled for ensuring optimum; tempered martensite microstructure, carburizing depth, proper hardness and absence of; micro-cracks, decarburization, corrosion, and machining defects.

Tobie et al [4] had shown that the case hardness depth can Influencing both bending and surface (contact) load capacity but in different manners. Maximum load capacity is achieved for an optimum value of case depth, but optimum values for maximum tooth root bending strength and pitting resistance of gear not necessarily be at the same parameters. An unfavorable case depth, smaller or larger than the optimum, leaded to a reduction of attainable load capacity.

Netpu and Srichandr [5] found that the gear teeth stress could cause failure if it increases the allowable design strength of the gear materials. So the contact stress started pitting or/and cracks on the surface, which developed to fatigue cracks causing cracks propagation till failure. Also failure occurred due to the excessive stress when the driving power was increased from 300 kW to 600 kW without re-evaluate the stresses.

Yuruts and Aleksandrov [6] had stated that the micro mechanism of fatigue failure is depending on the fatigue stress amplitude.

Koshy and Taji [7] concluded that high stresses initiated cracks at the tooth root because this section had the maximum tensile stress.

Fernandes and Mcduling [8] indicated that contact fatigue damage would first had occurred in the dedendum of the smaller gear suffered more revolutions, and therefore each tooth will experience a larger number of stress cycles. So, it is recommended that the smaller gear must be harder than the mating gear.

3. PROBLEM DEFINITION

Sugar industry is a seasonal industry so, the production line components must be stable and reliable through the production season to avoid a lot of loss. In the present case, a circulation fan gearbox with the following data:

Electric motor power is 315 kw, The input speed is 1450 rpm, The reduction ratio is 1:4.375, Single stage 16/70.

Operation 24 hrs, Equipped with oil cooling system, The structure is rigid.

The gears teeth design data were reviewed from the design drawing. The gears material was carburized steel DIN standard No.1.6587 (18CrNiMo7-6) and heat treatment specifications were:

Case hardening depth: 1.2- 1.4 mm

Case hardness Hv30 = 700-720 kg/mm2

Tempering 2hrs at 200 Co

The geometrical design data were summarized in the following table:

Table (1) Geometrical design data

Parameter	Pinion	Gear
No of teeth	16	70
Module	5.08	
Pressure angle	20°	
Helix angle	8.5° R	8.5° L
Profile displacement factor	0.492	0.541
Profile displacement	2.5	2.75
Tip diameter	97.34	375.3
Pitch diameter	82.182	359.55
Root diameter	73.67	351.23
Span distance no of teeth	3	9
Span distance	40.38	134.39
Center distance	226	

The gear box worked for 1 month and failure was occurred in the pinion gear as shown in the pictures of figures (1, 2, 3). The failure was studied to discover the failure reasons to avoid them in the future. Failure analysis for that failed gears showed that

these failures may occurred due to errors of: design, manufacturing, heat treatment, assembly, overload, vibration, maintenance or/and random effects.

The investigation procedure was surveying the assembly, operation and maintenance of the gearbox. Also, the environment was surveyed looking for dust, heat or corrosion sources. The operating conditions also reviewed for finding any overloads.

Material selection was considered and heat treatment specifications were reviewed. The heat treatment cycle parameters were also checked. The micro-hardness measurements survey for the teeth sections and also at the teeth roots at the area between teeth. The microstructure investigations were applied to the teeth and the area between teeth. The results of the previous steps were discussed and conclusions are withdrawn.

4. THE RESEARCH METHODOLOGY

The procedure followed in this paper for analyzing the present case failure can be summarized in a set of steps [9, 10, 11] such as:

- Observe visually the failure gears
- Review the gears' manufacturing data
- Check the assembly procedure
- Analyze the working conditions
- Investigate the hardness and the case depth
- Investigate the gears microstructure
- Compare the results of the original gear and the manufactured one
- Compare the results of both the original and manufactured gears with the standards specs.
- Then the failure causes can be carefully identified and recommendations can be given to avoid the future failures.
- Stress analysis review for the gears loading rates concerning bending and contact stresses following the metric AGMA procedure.

5. OBSERVATIONS AND REVIEW

The visual observation of the pinion gear after failure had shown that many teeth were broken at the tip section of the teeth and pitting was noticed through the teeth. see figures (1, 2, 3). Notice that most of pinion broken teeth at the tip.

Fig. (1) Pinion gear teeth failure

Fig. (2) Pinion gear section failure

Fig. (3) Pinion gear pitting and failure

Fig. (4) mating gear teeth pitting

The mating gear teeth had pitting as shown in figure (4) but no complete tooth was broken.

By surveying the operating conditions, there were not any abnormal conditions such as overload sources, excessive heat, shocks or vibration. The gearbox cooling system was working properly. It was noticed that no assembly faults or misalignment and neither sources of abrasion nor sources of corrosion were found.

The design data for the pinion and the gear were reviewed compared to the actual gears data and it were satisfied.

6. INVESTIGATION

The laboratory investigations were carried out which included: material analysis and micro hardness measurement survey across the teeth and the root between teeth followed by a microstructure scan. The following subclasses will show the results of the investigations.

6.1 Material Analysis

The spectra and chemical analysis results ensured that the material was confirmed to the German Standard (DIN 1.6587) (18CrNiMo7-6) for both pinion and gear.

6.2 Hardness Analysis

A micro hardness $HV_{0.1}$ kg (kg_f/mm^2) test was carried out for pinion through both the teeth section and the area between teeth at the root at which has the maximum bending stress position. The results were then tabulated and presented in the following table (table 2) and figure (5) which show the measured data of the pinion gear.

Table (2) Hardness values through the gear tooth sections for the gear

Depth (μm)	On tooth surface $HV_{0.1}$ (kg/mm^2)	Between teeth roots $HV_{0.1}$ (kg/mm^2)
20	715	664
100	697	657
200	689	641
300	689	619
500	672	585
600	657	-
700	-	553
800	649	-
1000	626	530
1100	-	503
1200	616	492
1500	598	
1700	578	
2000	530	
2300	513	
2400	497	
Teeth core hardness $Hv_{0.1}$ = 401 kg/mm^2		

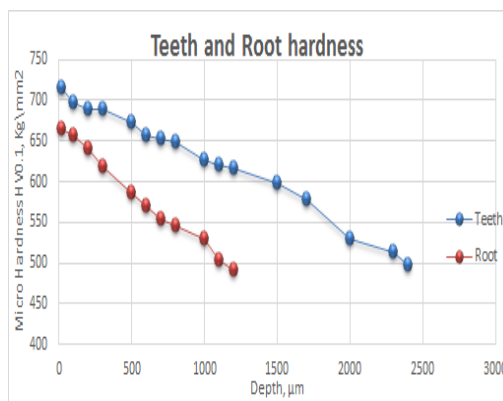*Fig. (5) Pinion gear micro-hardness*

The surface hardness HV0.1 of the Gear = 715 Kg/mm², while the core hardness of the original gear = 401 Kg/mm².
The case depth on the teeth tip of the original gear = 2.3 mm, while the case depth at the teeth root = 1.05 mm
The effective carburizing depth is considered at HRc=50 or HV= 513 (Kozlovskil et al [12]).

6.3 The Microstructure

Optical microscope with X500 magnification power was used in investigating the microstructure; Nital Etching 2% of the specimen. The microstructure of the pinion gear at the case of the teeth are shown in figures (6).

Figure (6) shows tempered martensite with small amount of retained austenite.

Fig. (7) Pinion gear teeth root microstructure

The microstructure which reveals small amount of martensite; black points in the microstructure.
The microstructure of the manufactured gears, as shown in figure (7) at teeth root reveals an amount of tempered martensite with small amount of retained austenite.

6.4 Stress Analysis

Many standards, books and researchers [13, 14, 15, 16] worked in the area of gears stress analysis such as ISO 6336, DIN 3990 and AGMA 2101-C95 (metric system). The followed procedures are almost similar. The present study uses stress analysis based on AGMA standards [13, 14].

There are mainly five equations controlling the design of bending and contact stresses applied to the gear tooth beside the tangential force equations and a set of constants are determined according to the case study operating conditions, geometry, loading, heat treatment, materials, reliability and safety.

The following equations are presenting the general stresses to the gear tooth.

1- The contact stress equation is:

$$\sigma_H = Z_E \sqrt{F_t\, K_o K_v K_s\, \frac{K_H}{d_{w1}\, b}\, \frac{Z_R}{Z_I}} \quad (1)$$

2- Tangential transmitted force equation is:

$$F_t = 1.91 \times 10^7\, P/(w_1 d_{w1}) \quad (2)$$

3- Allowable design stress is:

$$\sigma_H \leq \frac{\sigma_{HP}}{S_H}\, \frac{Z_N}{Y_\theta}\, \frac{Z_W}{Y_Z} \quad (3)$$

4- Bending stress equation is:

$$\sigma_F = F_t\, K_o K_v K_s\, \frac{1}{b\, m_t}\, \frac{K_H K_B}{Y_J} \quad (4)$$

5- Allowable bending stress equation is:

$$\sigma_F \leq \frac{\sigma_{FP} Y_N}{S_F Y_\theta Y_Z} \quad (5)$$

Where:

σ_H is contact stress, N/mm^2
σ_{HP} is allowable contact stress, N/mm^2
Z_E is elastic coefficient [N/mm^2]$^{0.5}$
F_t is transmitted tangential load, N
K_o is overload factor
K_v is dynamic factor
K_s is size factor
K_H is load distribution factor
Z_R is surface condition factor for pitting resistance
b is net width of narrowest gear, mm
Z_I is pitting resistance geometry factor
dw_1 is operating pitch diameter, mm = 2a/*(u+1)
a is center distance, mm
u is gear ratio>1
Z_N is stress cycle factor for pitting
Z_W is hardness ratio factor for pitting resistance
S_H is safety factor for pitting resistance
Y_Θ is temperature factor
Y_Z is reliability factor

According to the case study operating conditions and data, using AGMA standards [13, 14], the following parameters and constants were obtained and tabulated as shown in table (3):

Table (3) Obtained parameters and constants of gears

Parameter	Value
P	315 kw
w_1	1450 rpm
dw_1	84.09 mm
m_t	5.136 mm
B	85
F_t	49373 N
Z_E	190 [N/mm^2]$^{0.5}$
Z_I	0.25
Z_N	0.7
Z_R	1.1
Z_W	1
Y_J	0.48
Y_N	0.8
Y_Z	1.25

Y_J is bending strength geometry factor

σ_F is bending stress, N/mm^2

σ_{FP} is allowable bending stress, N/mm^2

Y_N is stress cycle factor for bending

S_F is safety factor for bending strength

K_B is rim factor

m_t is transverse metric module, mm

P is transmitted power, kw

w_1 is pinion speed, rpm

Y_Θ	1
S_F	1.2
S_H	1.2
K_B	1
K_H	1
K_o	1
Ks	1.1
K_v	1.4
σ_{FP}	485 N/mm^2
σ_{HP}	1550 N/mm^2

According to the above procedure and case study data, the analysis calculation results are:

- The contact stress: $\sigma_H = 1300$ N/mm^2

- The bending stress: $\sigma_F = 400$ N/mm^2

- The contact design stress term: $\dfrac{\sigma_{HP}}{S_H}\dfrac{Z_N}{Y_\theta}\dfrac{Z_W}{Y_Z} = 723$ N/mm^2

- The bending design stress term: $\dfrac{\sigma_{FP}Y_N}{S_F Y_\theta Y_Z} = 259$ N/mm^2

7. DISCUSSION OF RESULTS

The above investigation results concerning material analysis, microhardness survey, microstructure and the stress analysis will be discussed in this section compared to; standards, designs and researches literatures results considering the case study operating conditions.

7.1 The Material, Microhardness And Microstructure

The material analysis confirmed suitable material as required, microhardness survey of the pinion had revealed that a carburizing case depth at the teeth was not confirming the recommended values of references [17,18] because the depth equal 2.3 mm and it is much greater than the recommended suitable to the case data which causing the carburizing case is full hardened portion at the tip of the teeth. So, this portion miss its toughness and could not resist impact load. Also too much case depth makes the gear teeth more brittle with the tendency to shatter off the top of teeth [19]. All the above circumstances caused initiation of cracks and it was propagated until failure is occurred. This type of failure is classified as tooth bending fatigue failure [20].

The microstructure showed that the case microstructure consists of a tempered martensite and a small amount of the retained austenite which found to be adequate and confirmed the requirements.

7.2 The Stress Analysis

The results of the stress analysis had shown there were high level of contact and bending stress than the design values. The contact stress was $\sigma_H = 1300$ N/mm^2 while the design value must within the value of $\dfrac{\sigma_{HP}}{S_H}\dfrac{Z_N}{Y_\theta}\dfrac{Z_W}{Y_Z} = 723$ N/mm^2.

Also the bending stress was $\sigma_F = 400$ N/mm^2 while the bending design stress must be within the value of $\dfrac{\sigma_{FP}Y_N}{S_F Y_\theta Y_Z} = 259$ N/mm^2.

Then there are two main problems in this case study; the big layer of carburizing case has to be carried out with the correct value and, the high level of; contact and fatigue stresses lead to the fast failure of the gears and must be investigated for solving it in the future.   It is clear that high stresses come due to an error of design or an error of the gear box modification specially, it was used a single stage reduction while the gear casing has a multi stages axis as shown in figure (8).

Fig. (8), a multi stage casing

8. CONCLUSIONS

The gears have mainly two problems; first is the big carburizing case depth which leads to brittle teeth tip and this could be changed to the suitable depth and, second is the high contact and bending stress levels and this required reanalysis of the gear box deign specially the gearbox casing has multi stage gearing instead of the used single stage gearing.
The next recommendations are drawn:

- ✓ Carburizing depth 2.3 is very big which lead to brittleness and crack initiation lead to failure [21].
- ✓ The dynamic factor K_v and surface condition factor Z_R are highly affected by the pitch variation accuracy so; the gear teeth must be grinded. AGMA2101-C95
- ✓ The geometry factors Y_J, Z_I must be improved AGMA 908 - B89
- ✓ The profile displacement of gear larger than the pinion it is against the DIN 3960 recommendations [22].
- ✓ The size factor could be improved by increasing the module [23].
- ✓ Take care from increasing the pinion number of teeth because that has a disadvantage concerning the bending stress [24].
- ✓ Redesign the gear box using the available reduction stages in the gearbox casing.

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