Failure investigation of Planetary Gear Train due to Pitting

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Abstract - Planetary gear train is a gear system consisting of one or more planet gears, revolving about a sun gear. And it is wildly used in industry, such as paper industries, wind mills, and sugar industries. An epicyclical gearing system is particularly well suited for achieving a high-reduction ratio in a relatively small, power dense package. It is widely recognized that the load sharing is not equal among the planetary gear meshes. Similarly, the stress distribution at each mesh point contains variability. Pitting is a surface fatigue failure of the gear tooth. It occurs due to -misalignment; wrong viscosity selection of the lubricant used, and Contact stress exceeding the surface fatigue strength of the material

As we know gear is one of the most critical component in a mechanical power transmission system, failure of one gear will affect the whole transmission system. Therefore it is necessary to find the root cause which results into failure of gear and try to eliminate these causes. Failure analysis is an engineering approach to determining how and why equipment or a component has failed. The goal of a failure analysis is to understand the root cause of the failure so as to prevent similar failures in the future.

Keywords - Planetary gear train, Pitting.

I. INTRODUCTION

Gears are used extensively for transmission of power. Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. Planetary gear trains are one of the main subdivisions of the simple epicyclic gear train family. The epicyclic gear train family in general has a central “sun” gear which meshes with and is surrounded by planet gears. The outermost gear, the ring gear, meshes with each of the planet gears. The planet gears are held to a cage or carrier that fixes the planets in orbit relative to each other.

Planetary gears have been in motion for many years. The advantages are the higher torque capacity, smaller size, lower weight and improved efficiency characteristics of a planetary design. The small size and modular construction of planetary gearboxes also means that they can be assembled in several stages, providing high reduction capability from a highly compact package. And it can be wildly used in industry, such as printing lathe, automation assembly and sugar industry.

Because a planetary gear box is smaller and lighter, up to half the size and 60% lighter than conventional heavy engineered gearboxes, it is tempting to suppose that it is not as strong. Companies are using planetary gears not just for the weight saving, the reduction capability and the compact size, but also, in some cases because the units are weight balanced, i.e. in instances where a conventional gearbox is used the shaft is not in line with the bulk of the gearbox and it’s casing, hence there is an overhang and unbalanced weight to deal with. Because planetary gearboxes operate around a central shaft, they can be used in-line with turbines, pumps and wheel drives.

Wear is nothing but progressive removal of metal from the surface. Consequently tooth thins down and gets weakened. Pitting is a surface fatigue failure of the gear tooth. It occurs due to misalignment; wrong viscosity selection of the lubricant used, and contact stress exceeding the surface fatigue strength of the material. Material in the fatigue region gets removed and a pit is formed. The pit itself will cause stress concentration and soon the pitting spreads to adjacent region till the whole surface is covered. Subsequently, higher impact load resulting from pitting may cause fracture of already weakened tooth.

Gears analysis in the past was performed using analytical methods, which required a number of assumptions and simplifications. In general, gear analyses are multidisciplinary, including calculations
related to the tooth stresses and to tribological failures such as like wear or scoring. Designing highly loaded spur gears for power transmission systems that are both strong and quiet requires analysis methods that can easily be implemented and also provide information on contact and bending stresses, along with transmission errors. The finite element method is capable of providing this information. The finite element method is very often used to analyze the stress state of an elastic body with complicated geometry, such as a gear.

TECHNICAL DATA :
1. TYPE :- HELICAL PLANETARY GEARBOX
2. INPUT :- FREE (SOLID MALE SHAFT) ,200 H.P./1000 rpm
3. MODEL :- 4KT-18HX
4. MOUNTING :- FOOT
5. REDUCTION RATIO :- 162.43: 1
6. OUTPUT SPEED :- 6.15 rpm at INPUT 1000 rpm
7. RATED TORQUE :- 823402 Nm
8. ORIENTATION :- HORIZONTAL
9. LUBRICATION :- OIL.
10. LUBRICANT :- ISO VG -320
11. GEARBOX WEIGHT:- 6.5 TON (approx.)
12. Material for Sun Gear:- 17CrNiMo6
13. Material for Planet Gear:- 17CrNiMo6
14. Material for Ring Gear:- En-19

Reduction Ratio according to stages:

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Stages</th>
<th>Red Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>KT-14 HX</td>
<td>2.05</td>
</tr>
<tr>
<td>2</td>
<td>KT-14</td>
<td>5.43</td>
</tr>
<tr>
<td>3</td>
<td>KT-16</td>
<td>3.82</td>
</tr>
<tr>
<td>4</td>
<td>KT-18</td>
<td>3.82</td>
</tr>
</tbody>
</table>

KAVITSU Transmission Pvt. Ltd., Satara is manufacturing Industrial GearBoxes, Electric Motors & Slew Bearing. Company is facing pitting failure problem in the 4KT-18HX model (Gear Box). As we know gear is one of the most critical components in a mechanical power transmission system, failure of gear will affect the whole transmission system. Therefore it is necessary to find the root cause of pitting failure in gears and try to eliminate these causes.

Pitting is a surface fatigue failure of the gear tooth. It occurs due to –
- misalignment;
- wrong viscosity selection of the lubricant used, and
- Contact stress exceeding the surface fatigue strength of the material.

Material in the fatigue region gets removed and a pit is formed.

II. LITERATURE REVIEW

Ali Raad Hassan [1] studied contact stress analysis of spur gear teeth pair. He compared finite element analysis results with theoretical calculations. He developed a programme to plot a pair of teeth in contact. This programme was run for each 3° of pinion rotation from the first location of contact to the last location of contact to produce 10 cases. He made finite element models for these cases and stress analysis was done. By this study he concluded that, the maximum stress result obtained from AGMA stress calculation method and the maximum contact stress obtained from the finite element contact analysis was nearly same under the same conditions.

Seok-Chul Hwang et al. [2] presented two examples of spur and helical gears to investigate the respective variations of the contact stress in a pair of mating gears with the contact position. The strength determined from the AGMA and ISO standards is valid under the assumption that the load is uniformly distributed along the line of contact. However, in actuality, the load per unit length varies with the point of contact. The results obtained from finite element analysis are compared with the stresses yielded through AGMA standards. They have concluded that, the values calculated by using finite element analysis are below the contact fatigue strength of the material; hence, they yield the appropriate strength and safety.

Sunyoung Park et al. [4] investigated that the failure of planetary gear carrier of 1200HP transmission was caused by blowholes formed in the casting process. A systematic failure analysis of the defect planetary gear carrier of tracked vehicle transmission was carried out.
The analysis was focused on the metallurgical and mechanical points of views. Metallurgical analysis is composed of macroscopic, microscopic, chemical composition and spheroidal graphite rate analysis. The results showed that the stress distribution around the blowholes was irregular and the stress was concentrated on the blowhole regions as we expected. They found stress distribution of the defect planetary gear carrier is concentrated around blowhole regions and the maximum stress of the gear carrier is increased 26% more than the normal gear carrier.

Osman Asi [5] studied fatigue failure of a helical gear used in gearbox of a bus, which is made from AISI 8620 steel. An evaluation of the failed helical gear was undertaken to assess its integrity that included a visual examination, photo documentation, chemical analysis, micro-hardness measurement, and metallographic examination. Results indicate that teeth of the helical gear failed by fatigue with a fatigue crack initiation from destructive pitting and spalling. By this study, it was concluded that the primary cause of failure of the helical gear was likely a misalignment of the helical gear.

Waldemar Tuszynski [6] compared two different automotive gear oils from the point of view of vulnerability to deteriorating due to ageing their properties related to scuffing and pitting prevention. Gear oils of API GL-3 and GL-5 performance levels were tested. They used scuffing test and pitting test to compare these oils. The results of both test were GL-5 oils are less vulnerable to deterioration than GL-3 oils.

Hayrettin Duzcuoglu and Huseyin Imrek [7] investigated a new method for preventing premature pitting formation on spur gears. They studied the formation of pitting in the single tooth meshing region by decreasing the Hertzian surface pressure by increasing tooth width. By doing this, service life is increased by means of making a tooth width modification (F/b ratio change), thus decreasing the high load at the single tooth contact region. Finally they concluded that to minimize the pitting formation, gears having wider teeth can be manufactured.

Mr. Bharat Gupta et al. [8] studied contact stress analysis of spur gear. They analyze contact stress using Hertz theory which was originally derived for contact between two cylinders. Finally they Compared peak values of the contact stresses by considering different modules by Hertz Theory and ANSYS 13.0. By doing this study they concluded that maximum contact stress decreases with increasing module and for large power transmission spur gear with higher module is preferred.

III. Hertz Contact Stress (Involute Gear Tooth Contact Stress Analysis)

One of the main gear tooth failure is pitting which is a surface fatigue failure due to many repetition of high contact stresses occurring in the gear tooth surface while a pair of teeth is transmitting power. Contact failure in gears is currently predicted by comparing the calculated Hertz contact stress to experimentally determined allowable values for the given material. The method of calculating gear contact stress by Hertz’s equation originally derived for contact between two cylinders. Contact stresses between cylinders are shown in figure 1 and figure 2.

In machine design, problems frequently occurs when two members with curved surfaces are deformed when pressed against one another giving rise to an area of contact under compressive stresses. Of particular interest to the gear designer is the case where the curved surfaces are of cylindrical shape because they closely resemble gear tooth surfaces.

In Fig.1 two gear teeth are shown in mating condition at the pitch point. Referring to Fig.2, the area of contact under load is a narrow rectangle of width B and length L. The stress distribution pattern is elliptical across the width as shown in figure 3.
Fig. 3: Ellipsoidal–prism pressure distribution Value is given by,

\[ p_{c_{\text{max}}} = \frac{4F}{L} \times \frac{B}{B} \times L \]

Where,

\[ B = \frac{1}{1 - v_1^2} \frac{1}{1 - v_2^2} E_1 E_2 \]

Here, F is the applied force, \( v_1 \) and \( v_2 \) are Poison’s ratio of the two materials of the cylinders with diameters \( D_1 \) and \( D_2 \), and \( E_1 \) and \( E_2 \) are the respective modulii of elasticity.

Putting the values of B and assuming a value of 0.3 to poison’s ratio in Eqn. 3.1, and by replacing diameters by respective radii,

\[ p_{c_{\text{max}}} = \sqrt{0.35 \times \frac{F}{L} \times \frac{R_1 + R_2}{R_1 R_2}} \]

The Hertz equations discussed so far can be utilized to calculate the contact stresses which prevail in case of tooth surfaces of two mating spur gears. Though an approximation, the contact aspects of such gears can be taken to be equivalent to those of cylinders having the same radii of curvature at the contact point as the load transmitting gears. Radius of curvature changes continuously in case of an involutes curve, and it changes sharply in the vicinity of the base circle.

Fig. 4: Equivalent contacting cylinder

Now by putting following equations in Eqn. 3.1

\[ F = \frac{F_t}{\cos \alpha}, \quad L = b, \quad R_1 = \frac{d_1 \times \sin \alpha}{2}, \quad R_2 = \frac{d_2 \times \sin \alpha}{2} \]

Where \( F_t \) is the tangential force or transmitted load, b is the tooth width, \( R_1 \) and \( R_2 \) are the radii of curvature at pitch point, and \( d_1 \) and \( d_2 \) are the pitch circle diameters of the gears.

Putting,

\[ E = \frac{E_1 E_2}{E_1 + E_2} \quad \text{and} \quad \mu = \frac{d_2}{d_1} \]

We get,

\[ \frac{1}{R_1} + \frac{1}{R_2} = \frac{2}{\sin \alpha} \times \left( \frac{1 + \frac{1}{d_1} + \frac{1}{d_2}}{\sin \alpha} \times \frac{2}{d_1 \times \sin \alpha} \times \left( \frac{u+1}{u} \right) \right) \]

3.3

Inserting these values in Eqn. 3.2 we get the expression for the maximum contact pressure at the pitch point

\[ p_p = \sqrt{0.35 \times \frac{F_t}{\cos \alpha} \times \frac{E}{b} \times \frac{1}{\sin \alpha} \times \frac{1}{d_1} \times \frac{u+1}{u} \times \frac{1}{\cos \alpha \times \sin \alpha}} \]

Now by considering service pressure angle, \( \alpha_w \)
To simplify calculations, is written in the form

\[ P_p = \frac{0.35 \times \frac{1}{b \sqrt{\frac{E_1}{d_1}} \times \frac{u+1}{u} \times \frac{1}{\cos \alpha \times \tan \alpha_w}}}{FOS} \]

Allowable Maximum Pp = \( \frac{P_p}{FOS} \)

From the above equation, Pressure or Stress is obtained at pitch point in which factor of safety will be included by dividing \( P_p \) with factor of safety which can be taken from ANSYS result or may be from other FOS tables. After that the allowable max. Pressure or Stress at pitch point answer will be obtained. Ultimately minimum factor of safety is taken from ANSYS result in order to get an accurateresult of allowable max. Pressure or Stress at pitch point.

\( y_m \) is the material coefficient and \( y_p \) is the pitch point coefficient, which are given by

\[ y_m = \sqrt{0.35 \times \frac{2 \times E_1 \times E_2}{E_1 + E_2}} \]
\[ y_p = \sqrt{\frac{1}{\cos \alpha \times \tan \alpha_w}} \]

**IV. MODELING OF PLANETARY GEAR TRAIN:**

Modeling of planetary gear train is carried out using UG NX 7, is an advanced CAD/CAM/CAE software.

<table>
<thead>
<tr>
<th>Type</th>
<th>Sun Gear Data</th>
<th>Planet Gear Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of Teeth</td>
<td>22</td>
<td>20</td>
</tr>
<tr>
<td>Module</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Pressure Angle</td>
<td>20°</td>
<td>20°</td>
</tr>
<tr>
<td>Correction Factor</td>
<td>-0.237</td>
<td>-0.237</td>
</tr>
<tr>
<td>Pitch Circle Dia.</td>
<td>330</td>
<td>300</td>
</tr>
<tr>
<td>Major Dia.</td>
<td>352.89-0.5</td>
<td>337.11</td>
</tr>
<tr>
<td>Minor Dia.</td>
<td>285.39</td>
<td>269.61</td>
</tr>
</tbody>
</table>

**Table No. 1: Gear Data**

In this paper work module, pressure angle, numbers of teeth of planetary gear train are taken as input parameters.

**V. FEM RESULTS FOR CONTACT STRESS OF PLANETARY GEAR TRAIN BY ANSYS 13.0**

Following results are obtained by changing element size.

<table>
<thead>
<tr>
<th>Element size (mm)</th>
<th>No. of Nodes</th>
<th>Stress Max. (Mpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>872669</td>
<td>1565.5</td>
</tr>
<tr>
<td>11</td>
<td>684194</td>
<td>1038.8</td>
</tr>
<tr>
<td>12</td>
<td>574583</td>
<td>1551.2</td>
</tr>
<tr>
<td>13</td>
<td>475392</td>
<td>1804.2</td>
</tr>
<tr>
<td>14</td>
<td>413646</td>
<td>1771.8</td>
</tr>
<tr>
<td>15</td>
<td>345625</td>
<td>1548</td>
</tr>
</tbody>
</table>

**Table No. 2: No. of Nodes vs. Stress**
Graph plotted for No. of Nodes Vs Max. Stress and it is found 1804.2 MPa is genuine peak. This contact stresses is higher than the surface endurance limit of the material which is 1761.2 MPa. Therefore it is found that higher contact stresses are the cause of pitting failure.

Current planetary gear train is of module 15, but above analysis result shows that contact stresses are higher than the surface endurance limit.

Standard values of module are as shown.

<table>
<thead>
<tr>
<th>Preferred (1)</th>
<th>Choice 2 (2)</th>
<th>Choice 3 (3)</th>
<th>Preferred (1)</th>
<th>Choice 2 (2)</th>
<th>Choice 3 (3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>125</td>
<td>10</td>
<td>4</td>
<td>7</td>
<td>0</td>
</tr>
<tr>
<td>1.25</td>
<td>1.125</td>
<td>10</td>
<td>4</td>
<td>7</td>
<td>6</td>
</tr>
<tr>
<td>1.5</td>
<td>1.375</td>
<td>12</td>
<td>6</td>
<td>11</td>
<td>13</td>
</tr>
<tr>
<td>2</td>
<td>1.75</td>
<td>16</td>
<td>16</td>
<td>14</td>
<td>18</td>
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<tr>
<td>2.5</td>
<td>2.25</td>
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<tr>
<td>4</td>
<td>3.5</td>
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<tr>
<td>5</td>
<td>4.5</td>
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<td>36</td>
<td>36</td>
<td>38</td>
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<tr>
<td>6</td>
<td>5.5</td>
<td>40</td>
<td>40</td>
<td>40</td>
<td>42</td>
</tr>
</tbody>
</table>

The module given under choice 1 is always preferred. If that is not possible under certain circumstances module under choice 2, can be selected.

From above table module 16 is selected for gear and according to this new planetary gear train is designed.
Graph 2: No. of Nodes Vs Max. Stress(module 16)

By studying the above graph it is found that 1549 MPa is genuine peak and which is lower than the surface endurance limit of 1761.2 MPa.

Pitting is the surface fatigue failure which occurs due to many repetitions of Hertz contact stresses. The failure occurs when the surface contact stresses are higher than the endurance limit of the material. The failure starts with the formation of pits which continue to grow resulting in the rupture of the tooth surface.

17CrNiMo6: Chrome-Nickel-Moly Carburising Steel generally supplied annealed to HB 229max. Carburised and heat treated it develops a hard wear resistant case to HRC 60-63.

We can improve the surface endurance limit by increasing hardness of the material by heat treatment.

Case Hardening:
- Heat to 780°C – 820°C
- Quench in oil

After heat-treatment hardness of the material is 63 HRC. Surface endurance limit after heat-treatment is 1904 MPa.

VI. CONCLUSION

KAVITSU Transmission Pvt. Ltd., Satara is facing gear pitting failure problem in their 4KT-18 HX model. For this purpose contact stress is found out using ANSYS Work-Bench 12. This contact stress is higher than the material surface endurance limit of 1761.12 MPa. This result shows that contact stress is the reason for pitting failure.

The module is important geometrical parameter during the design of gear. Therefore selection of proper module size is an important factor before designing gear. Maximum contact stress decreases with increasing module.

Also Hear treatment of gear is also an important factor.

VII. REFERENCES