FAILURE ANALYSIS OF WAGON GEAR

This project report is submitted to

Yeshwantrao Chavan College of Engineering
(An Autonomous Institution Affiliated to Rashtrasant Tukdoji Maharaj Nagpur University)

In partial fulfillment of the requirement

For the award of the degree

Of

Bachelor of Engineering in Mechanical Engineering

By

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Under the guidance of

Prof. A.W Oke

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YESHWANTRAO CHAVAN COLLEGE OF ENGINEERING,
(An autonomous institution affiliated to Rashtrasant Tukadoji Maharaj Nagpur University, Nagpur)
NAGPUR – 441 110

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CERTIFICATE OF APPROVAL

Certified that the project report entitled "FAILURE ANALYSIS OF WAGON GEAR" has been successfully completed by ANKITA S. CHORE, AKSHAY JOSHI, BHARAT KOLHE, SREEHARSHA YARRAM under the guidance of PROF. A.W.OKE in recognition to the partial fulfillment for the award of the degree of Bachelor of Engineering in Mechanical Engineering, **Yeshwantrao Chavan College of Engineering*(An Autonomous Institution Affiliated to Rashtrasant Tukdoji Maharaj Nagpur University)***

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To whom So Ever It May Concern

This is to certify that the following students of Mechanical Engineering Department, Yeshwantrao Chavan College of Engineering have done project titled "Failure Analysis of Loco Gear" for Kinetic Gears, MIDC, Hingna, Nagpur.

The Students have really worked very hard and their efforts need to be appreciated.

We wish them the very best for their future.

The names of the students are

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For Kinetic Gears

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Date:- 14/04/2014
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DECLARATION

I certify that

a. The work contained in this project has been done by me under the guidance of my supervisor(s).
b. The work has not been submitted to any other Institute for any degree or diploma.
c. I have followed the guidelines provided by the Institute in preparing the project report.
d. I have conformed to the norms and guidelines given in the Ethical Code of Conduct of the Institute.
e. Whenever I have used materials (data, theoretical analysis, figures, and text) from other sources, I have given due credit to them by citing them in the text of the report and giving their details in the references. Further, I have taken permission from the copyright owners of the sources, whenever necessary.

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An Industrial project, this project encompasses failure analysis of a spiral bevel gear. Problem statement is the following; on wear-out of a spiral bevel gear which was the part of part of the hydraulic turbo reversing transmission of a Ventra Locomotive, the client placed an order of re-fabrication with M/S Kinetic Gears pvt. Ltd. After making the gear according to the client's specification, the gear was found cracked during hand press fitting with the shaft. The project strives to find out the causes of failure and its rectification, the constraint being the changes cannot be made in design of the gear or dimension of shaft.

CHAPTER 1
INTRODUCTION

CHAPTER ONE

INTRODUCTION

1.1 OVERVIEW

The main component to be analyzed is a spiral bevel pinion. This pinion along with the gear is used for axle gear drive box. It is used in Ventra LOCO 335HP shunting locomotive. It is the part of the hydraulic turbo reversing transmission.

1.2 DIMENSIONS

The dimensions of the spiral bevel pinion are tabulated for the ease of the reader.

No.	Parameter	Value
1	No. of Teeth	30
2	Transverse Module	10
3	Ratio	1.5
4	Pitch Circle Diameter	300
5	Outer Diameter	316.75
6	Face Width	80
7	Tooth Form	Spiral
8	Hardness	207-275 BHN

Table 1.2.1 Dimensions

1.3 MANUFACTURING

Manufacturing of the gear was done at Kinetic gears PVT ltd. The gear was made according to dimensions sent forth by the client. Standard manufacturing techniques were used.

1.3.1 AUTOCAD DRAWING

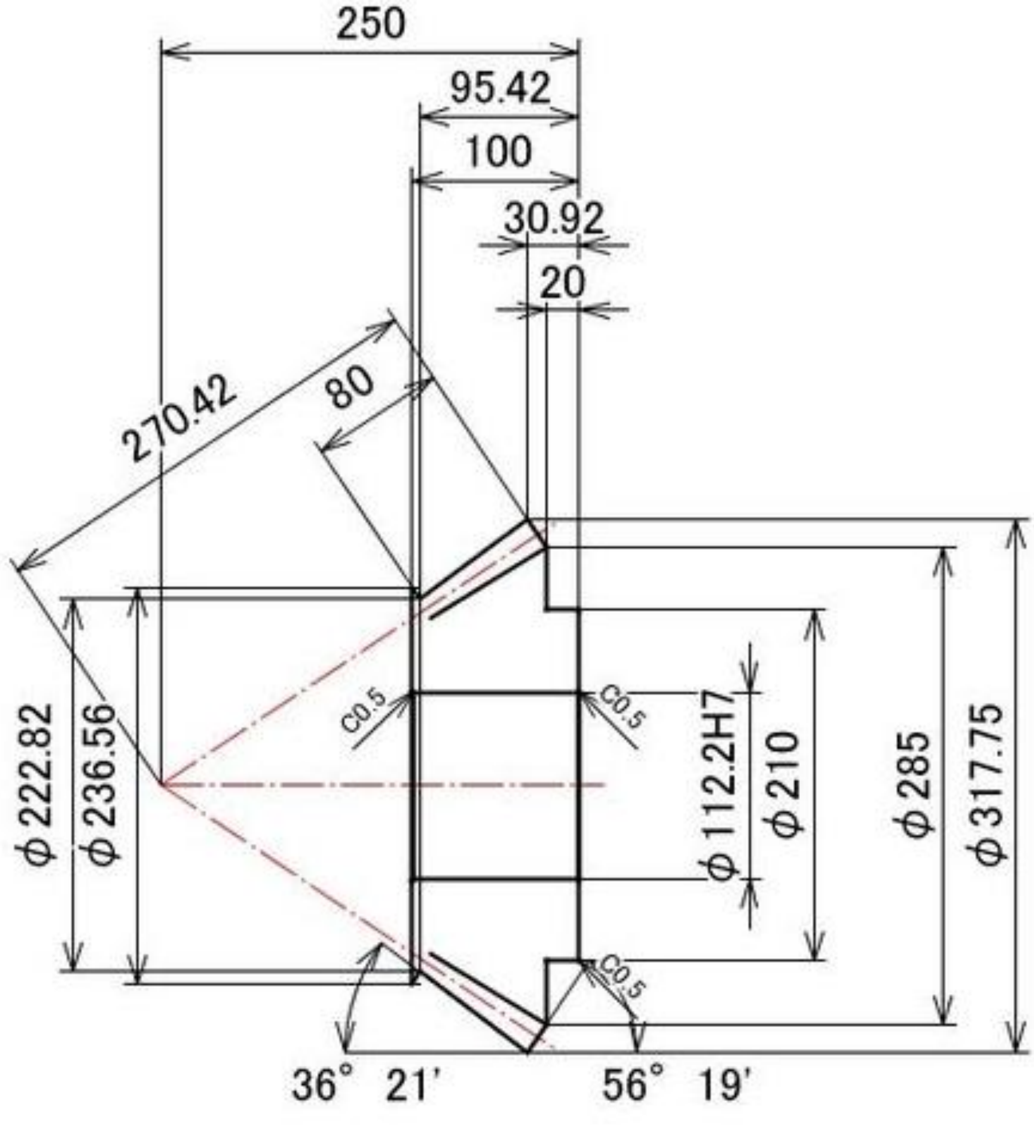

Figure 1.3.1 AutoCAD Drawing of pinion

1.3.2 PROCESS CHART

No.	Process Name / Operations	Actual Dimensions
1	HS Raw Material	EN 9 forging
2	Rough Turning	Ø 109 -20
3	Finish Turning	Ø 316.7 * 120.67
4	HOBBING	
5	Run out of Gear Blank	AUM 0.056
6	Grinding	Ø109.80

Table 1.3.1 Process Chart

1.3.3 FITTING DIMENSIONS

No.	Parameter	Value
1.	Taper of pinion bore	1:40
2.	Fit Type	Interference
3.	Fit Value	0.5 mm
4.	Inner Diameter	109.7 mm
5.	Outer Diameter	111.8 mm

Table 1.3.2 Fitting Dimensions

1.4 MANUFACTURING PLANT LAYOUT:

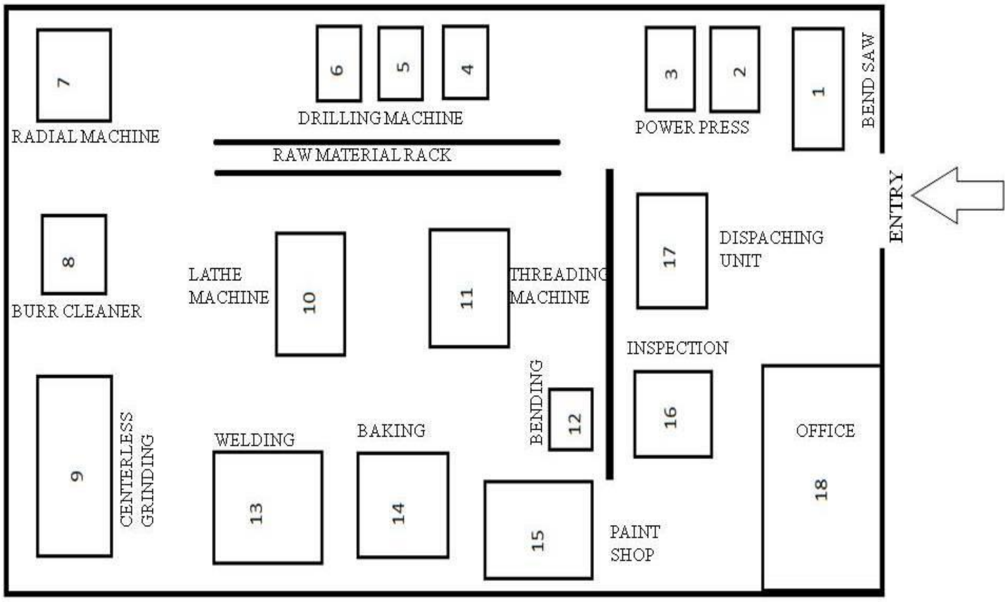

Figure 1.4.1 Manufacturing Plant Layout

1.5. MANUFACTURING PROCESS:

1.5.1 Rough turning:

The Rough Turning operation allows you to specify:

- Longitudinal, Face and Parallel Contour roughing modes
- External, internal or frontal machining according to the type of area to machine
- Relimitation of the area to machine
- Various approach and retract path types
- Various lead-in and lift-off options with specific federate.
- Recess machining
- Various contouring options with specific federate.

1.5.2. Finish turning:

The Groove Finish Turning operation allows you to finish a groove by means of downward profile following. You can specify:

- The type of machining according to the orientation of the groove profile to be machined
- Relimitation of the profile by start and end elements
- Various approach and retract path types
- Linear and circular lead-in and lift-off options with specific federates
- Various corner processing options with specific federates
- Local federates for individual elements of the machined profile
- Tool output point change
- Cutter compensation

1.5.3. Hobbing:

Hobbing is a machining process for gear cutting , cutting splines, and cutting sprokets on a hobbing machine, which is a special type of milling machine. The teeth or splines are progressively cut into the workpiece by a series of cuts made by a cutting tool called a hobs. Compared to other gear forming processes it is relatively inexpensive but still quite accurate, thus it is used for a broad range of parts and quantities. It is the most widely used gear cutting process for creating spur and helical gears and more gears are cut by hobbing than any other process since it is relatively quick and inexpensive.

1.5.4. Hobs:

The hob is a cutting tool used to cut the teeth into the workpiece. It is cylindrical in shape with helical cutting teeth. These teeth have grooves that run the length of the hob, which aid in cutting and chip removal. There are also special hobs designed for special gears such as the spline and sprocket gears.

The cross-sectional shape of the hob teeth are almost the same shape as teeth of a rack gear that would be used with the finished product. There are slight changes to the shape for generating purposes, such as extending the hob's tooth length to create a clearance in the gear's roots. Each hob tooth is relieved on the back side to reduce friction.

Most hobs are single-thread hobs, but double-, and triple-thread hobs increase production rates. The downside is that they are not as accurate as single-thread hobs. Depending on type of gear teeth to be cut, there are custom made hobs and general purpose hobs. Custom made hobs are different from other hobs as they are suited to make gears with modified tooth profile. The tooth profile is modified to add strength and reduce size and noise of gears.

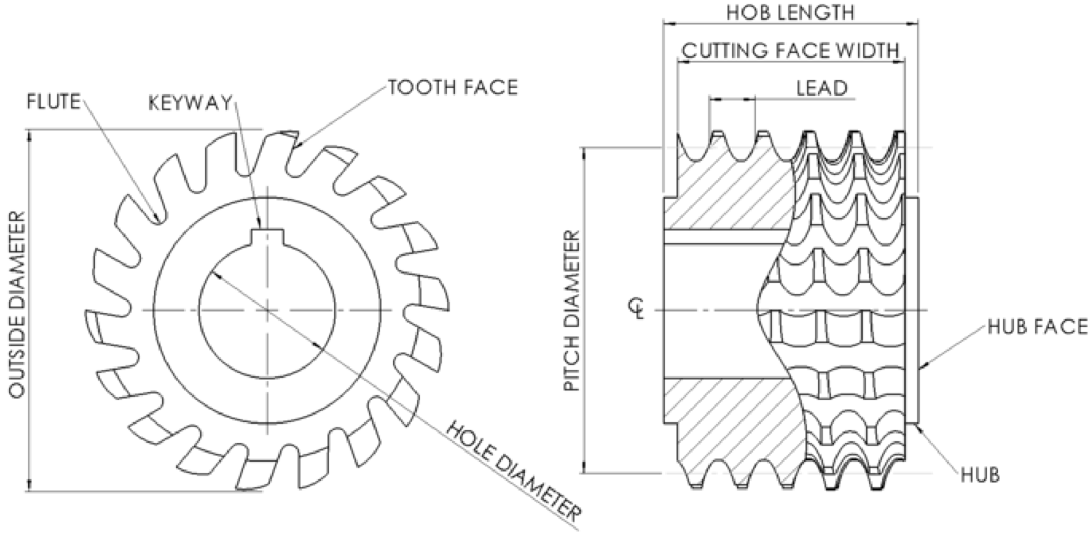

Figure 1.5.1 Hobs

1.4 OBJECTIVES OF PROJECT

The main objective of project is to identify the defect in gear to reduce fracture and cracks generated in pinion gear and to eliminate this defect by applying changes in material, dimensions, and design specifications.

1.5 SCOPE OF WORK:

Project deals with failure of gears and analysis. It studies the gear, determines its causes of failure and tries to rectify the defects and suggest changes so that the defect will not occur again.

CHAPTER 2
LITERATURE SURVEY

CHAPTER TWO

LITERATURE SURVEY

2.1. THEOROTICAL BACKGROUND

To transmit power from one component of a mechanical system to another gears are the efficient and effective machines. There are many designs of gears which are manufactured depending upon their functionality, system requirements and operating conditions. These include spur gears, hypoid gears, spiral and straight bevel gear.

Many gear failures occur due to design errors, manufacturing faults, maintenance, inspection, inevitable repetitive stresses resulting in surface fatigue, wear and deterioration of lubricant properties. The gears generally fail when tooth stress exceeds the safe limit. Analysis of different aspects of gear failure is studied thoroughly in this project.

2.2. TYPES OF GEARS

2.2.1. Spur gear:

Spur gear or straight cut gears are the simplest type of gears. Their general form is cylinder or disk. The edge of each tooth is straight and aligned parallel to the axis of rotation. These gears can be meshed together correctly only if they are fitted to parallel shafts.

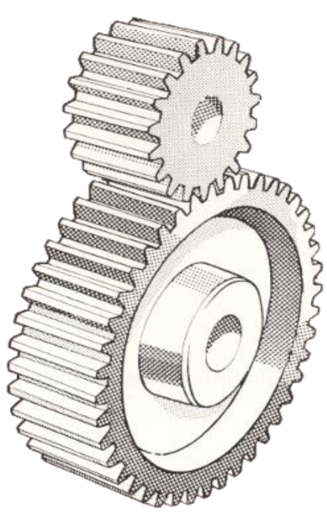

Figure .2.2.1. Spur gear

2.2.2. Helical gears

The leading edges of the teeth are not parallel to the axis of rotation, but are set an angle. Since the gear is curved, this angling causes the tooth shape to be a segment of helix. Helical gears can be meshed in parallel or crossed orientation. The angled teeth engage more gradually than do spur gear teeth, causing them to run more smoothly and quietly.

Figure .2.2.2. Helical gear

2.2.3. Bevel gears

Bevel gears are essentially conically shaped, although the actual gear does not extend all the way to the vertex of the cone that bounds it. The angle between the shaft can be anything except 0 or 180.Bevel gears with equal no. of teeth and shaft axes at 90 are called as miter gears.

The teeth of bevel gear can be straight cut as with spur gears or they may be cut in variety of shapes. Spiral bevel gears that are both curved along their length and set at an angle. Zero bevel gears have teeth which are curved along their length, but not angled.

Figure .2.2.3. Bevel gear

2.2.4. Crown gears

A crown gear is a particular form of bevel gear whose teeth projects at right angles to the plane of the wheel; in their orientation the teeth resembles the points on a crown. A crown gear can only mesh accurately with another bevel gear, although crown gears are sometimes seen meshing with spur gears.

Figure 2.2.4. Crown gear

2.2.5. Worm gear

A worm is a gear that resembles a screw. It is a species of helical gear, but its helix angle is usually somewhat large and its body is usually fairly long in the axial direction; and it is this attribute which give it its screw like qualities.

2.2.6. Rack and pinion gear

A rack is a toothed bar or rod that can be thought of as a sector gear with an infinitely large radius of curvature. Torque can be converted to linear force by meshing a rack with a pinion; the pinion turns; the rack moves in straight line. Racks also feature in the theory of gear geometry, where the tooth shape of an interchangeable set of gears may be specified for the rack and tooth shapes for gears of particular actual radii then derived from that. The rack and pinion gear type is employed in a rack railway.

Figure 2.2.6. Rack and pinion

2.3. GEAR MATERIAL:

2.3.1. Material used: EN9

EN9 is an unalloyed medium carbon steel. It is supplied at the hardness obtained after hot rolling or cold drawing, with hardness normally within the range of 180 to 230HB. EN9 is available from stock in bar and can be cut to your requirements. We also offer EN9 plate flame cut to your required sizes and normalized.

TYPICAL ANALYSIS				
C.	Si.	Mn.	S.	P.
0.50%	0.25%	0.70%	0.05%	0.05%

Table 2.1 Typical Analysis

Density	Coefficient of Thermal Expansion	Modulus of Elasticity
kg/m	per c from 20 c	N/mm
7800	11.6x10-6	206000

Table 2.2 Properties of EN – 9

Characteristics:

- A 6% allowance should always be made for removal of surface defects during machining.
- Ideal for 45 ton tensile applications
- Not recommended for carburizing.
- Heat treatment only to limited ruling sections

Typical Applications:

- Sprockets
- Cylinders
- Cams

- Crankshafts
- Keys
- Small Gears
- Machine Tool

2.3.2. Cast iron

Cast iron is used in place of non--heat treated steel where good wear resistance combined with excellent machinability is required. Complicated blank shapes can be cast more easily from iron than they can be produced by machining from bars or forgings. Usually cast iron gears are not hardened. When required, furnace hardening is used for through hardening. Induction or flame hardening can be used when localized hardening is desirable.

2.3.3. Ductile (nodular) iron

Ductile iron can be through hardened or surface hardened.

2.3.4. Bronze

Bronze materials are used when corrosion resistance or non--magnetic properties are required.

2.3.5. Non-metallic

Some non--metallic materials offer advantages when loads are light and operating temperatures permit. Selection and specifications for these materials should be based upon the requirements of the application.

2.3.6. Heat treatment process

1. **Forging:**

   Heat slowly and uniformly to 1100°C.After forging cool slowly.
2. **Annealing:**

   The purpose of annealing is to soften the material and to relieve the internal stress.

It improves machinability as well but the strength properties such as yield stress, hardness, tensile stress, etc decreases. Heat uniformly to 700°C . Soak well and cool slowly in the furnace.

3. **Normalizing:**

   It is a process of restoring the metal to the original metallurgical state. The process of normalizing is to heat the specimen 60C to 70C above the upper critical temperature and to hold it at that temperature for a short period and then cool it in air. Normalize at 840-870°C, and cool in air.

4. **Hardening:**

   It is a process in which steel is heated to temperature above the critical point and then quenching it in water, oil or cold baths.

   In this case, surface temperature and core temperature widely differ and this may lead to quench crack. But the hardness obtainable is of high degree.

5. **Tempering:**

   The tempering process is heating the specimen to a temperature just below the critical temperature and then holding it in that temperature for a given period of time and then cooling it to the room temperature. It improves mechanical properties like toughness, grain structure, shock resistance etc.

6. **Stress relieving:**

   After rough machining tools should be stress relieved. Soaking time 2 hours is given after the whole piece has attained a temperature of approximately 675°C. Cool in furnace to approximately 500°C, and then freely in air.

2.3.7. Types of fits:

1. **Clearance:**

In a fit, this is the difference between the sizes of the hole and the shaft, before assembly, when this difference is positive. The clearance may be maximum clearance and minimum clearance. Minimum clearance in the fit is the difference between the maximum size of the hole and the minimum size of the shaft.

2. Interference:

It is the difference between the sizes of the hole and the shaft before assembly, when the difference is negative. The interference may be maximum or minimum. Maximum interference is arithmetical difference between the minimum size of the hole and the maximum size of the shaft before assembly. Minimum interference is the difference between the maximum size of the hole and the minimum size of the shaft.

3. Transition:

It is between clearance and interference, where the tolerance zones of the holes and shaft overlap.

2.4. NOMENCLATURE OF SPIRAL BEVEL GEAR:

1. **Dedendum of pinion**:
   It is the depth of the tooth space of the pinion below the pitch cone at the mean cone distance.
2. **Addendum of gear:**
   It is the height from the top of the gear tooth to the chord subtending the circular thickness arc at the mean cone distance in a plane normal to the tooth trace.
3. **Back angle distance:**
   It is the distance from the intersection of the gear axis and the mounting surface to a back cone element, for gear and pinion respectively.
4. **Face angle distance:**
   It is the distance from the intersection of the gear axis and the mounting surface to a face cone element, for gear and pinion respectively.
5. **Mean radius of curvature:**
   It is the radius o f curvature of the tooth surface in the lengthwise direction at the mean cone distance.
6. **Backlash allowance:**

It is the amount by which the tooth thicknesses are reduced to provide the necessary backlash in assembly. It is specified at the outer cone distance.

7. **Number of teeth in gear:**

   It is the number of teeth contained in the whole circumference of the gear pitch cone.

8. **Number of teeth in pinion:**

   It is the number of teeth contained in the whole circumference of the Pinion pitch cone.

9. **Pitch mean circular:**

   It is the distance along the pitch circle at the mean cone distance between corresponding profiles of adjacent teeth.

10. **Dedendum angles :**

    It is the sum of dedendum angles for tilted root line taper.

11. **Dedendum angles:**

    It is the sum of dedendum angles for uniform depth.

12. **Dedendum of gear:**

    It is the depth of the tooth space of the gear below the pitch cone at the mean cone distance.

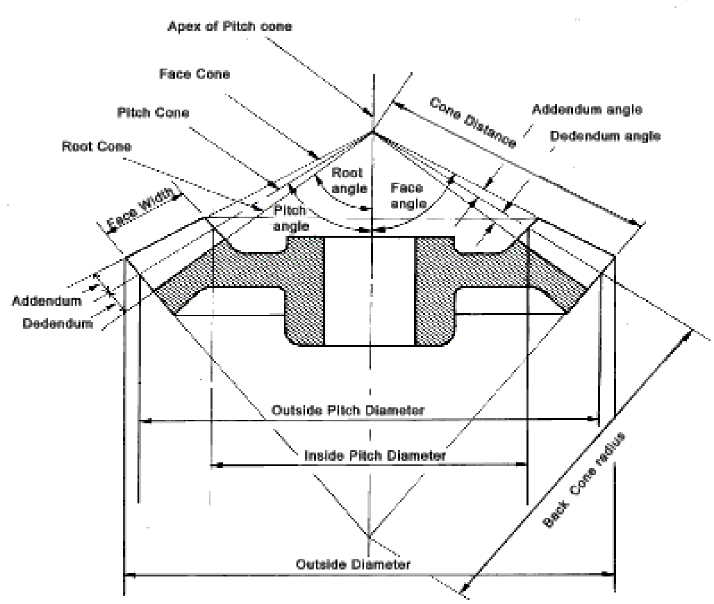

Figure .2.4.1. Spiral bevel gear nomenclature

2.4. KEY ASSEMBLY PARAMETERS:

Several parameters contribute to proper assembly so that the gearbox operates smoothly and efficiently. The most important are:

• Mounting distance.

• Matched teeth.

• Backlash

Bevel gears of AGMA quality 8 or better are normally manufactured and tested in sets. Then, the manufacture marks the preferred values for each of the three parameters on individual gears.

Frequently, the manufacturer also marks a gearset number on each of the two gears in a set. Gears manufactured in sets should only be assembled with their mates. Similarly, if one gear in a set fails, the entire set should be replaced.

2.4.1. Mounting distance:

The distance from a locating surface on the back of one gear to the centerline of a mating gear is called the mounting distance. This is the most important parameter for ensuring proper operation. The manufacturer establishes the optimum value for this distance by running the gearset and adjusting its position to obtain a tooth contact pattern that is consistent with smooth running and optimum load distribution between mating gear teeth. Because of dimensional variations between parts, each gear in a set has a unique value for the mounting distance and, in most cases; the manufacturer permanently marks this value on each gear.

It is possible to manufacture all gears to the nominal mounting distance specified on the drawing. But, the additional cost to do so is usually not warranted.

In cases where mounting distances have not been marked on the gears, assembly technicians must resort to a costly trial and error method of marking teeth with gear compound and adjusting the gears to obtain a suitable tooth contact pattern. The lack of marked values may also indicate that the gear set was not manufactured for high capacity or consistency.

2.4.2. Matched teeth:

After optimum tooth contact is obtained in the running tests, the manufacturer marks mating teeth in engagement for identification. These marks usually consist of x’s or dots, on two adjacent teeth of one gear and the mating tooth of the other gear. When assembling the gear set, position the single marked tooth in the space between the two adjacent marked teeth.

2.4.3. Backlash:

The third most important parameter for a bevel gearset is the space between mating gear teeth, called backlash. Unless otherwise specified, backlash is measured normal (perpendicular) to the tooth surface, and not in the plane of rotation. In most cases, the manufacturer marks the normal backlash value on the gear.

Measurements in the plane of rotation, called transverse backlash, are as much as 40% larger, and should not be used for assembly purposes.

Where mounting distances have been marked on one or both gears, adjust these distances first. Then, use the normal backlash value to verify proper assembly. The backlash value should not be measured until after assembling at least one of the two gears, preferably the pinion, at its marked mounting distance. The reason is simple. Because bevel gears are conical in shape, they can be assembled in an almost infinite number of positions (most of which cause poor performance) and still obtain the required backlash.

Frequently, a casual user assembles bevel gears to obtain a specific amount of backlash without regard to the mounting distance. This is especially true for low quality or lightly-loaded bevel gears. Although this method occasionally works, it is a risky approach where gears are loaded to maximum capacity. It usually causes shorter life and poor performance. Only at the proper mounting distance will a gear set run correctly and still have the right amount of backlash.

2.5. GEAR ASSEMBLY:

In most bevel gear sets, particularly those with ratios above 2:1, the pinion position (mounting distance) effects tooth contact (the most important parameter for good performance) to a larger extent than the gear position. Conversely, the gear position has a larger effect on backlash. For this reason, the manufacturer may have marked the mounting distance only on the pinion. In such cases, be sure to accurately position the pinion according to its marked mounting distance before adjusting backlash. This can be accomplished either by gauging or by direct measurement. Gages are used mostly for large gear production runs, and are based on the direct measurement method. For this reason, only the measurement method is described here.

Note that when mating gears are adjusted to their optimum position, their back angles will probably not be flush with each other. Do not attempt to position bevel gears by making the back angles flush.

2.6. Measurement method:

A typical gearbox contains both an overhung and a straddle-mounted gear. The procedure for assembling the bevel gears in this gearbox by the measurement method is simple:

• On the housing, measure the distance from locating surface to bore centerline in both horizontal and vertical directions (HMD and VMD).

• Record the gear and pinion mounting distances (MDG and MDP).

• Measure the gear and pinion thicknesses (WG and WP).

• Assemble the gears into their subassemblies.

• Measure the length that controls gear position in each subassembly (MAG and MAP).

• Calculate the distance from shim mounting surface to crossing point for each subassembly (MDV and MDH).

• Calculate the required shim thicknesses and assemble the gearbox with shims in place.

To aid the assembly technician, mark the housing measurements on the housing after it is machined and inspected.

Measurements involving the pinion and gear are taken on the subassemblies, which consist of gear, shaft, bearing, and gear mounting components. This minimizes any variations due to fits between these components and greatly reduces the number of measurements required. In this example, only the gear thickness is measured before putting together the subassembly. Subtracting this distance from the mounting distance lets the technician measure from the front of the assembled gear.

2.7. Verifying proper assembly:

Use the measurement method to assemble both gears, making sure that the matched teeth are properly engaged. Only after positioning both gears at their mounting distances should you measure backlash to verify proper assembly. In cases where only the pinion is marked with a mounting distance, position the pinion at its mounting distance first, then adjust the gear position for the required backlash.

To obtain further verification, you can perform an optional contact pattern check. This involves painting the tooth surfaces on one gear with a thin marking compound and rotating the assembled gears under light load. The compound transfers from one gear to the other and shows how the teeth contact. Numerous publications contain charts with typical contact patterns (ANSI/AGMA 2008-B90 and ANSI/AGMA 2005-B88). The patterns seldom look exactly like the published examples and frequently require some interpretation.

Technicians familiar with contact pattern checking may believe that this technique is the preferred method of assembly. However, the manufacturer has already performed this check as part of the testing (to determine mounting dimensions) and interpreted the contact patterns so you don't have to.

2.8 FORCES ACTING ON THE GEAR

2.8.1 Tangential:

The tangential forces on a bevel gear (member with larger number of teeth) is given by:

$WtG = 2TGDm$

where

WtG is tangential force at mean diameter on the gear, lb (N);

TG is torque transmitted by the gear, lb in (Nm).

The tangential force on themating pinion is given by: $WtP = WtG \cos \psi P \cos \psi G = 2TPdm$

Where *WtP* is tangential force at mean diameter on the pinion, lb (N).

2.8.2. Radial:

The values of radial force, *Wr*, on bevel gears are given in the following formulas. When using the formulas the tangential force, spiral angle, pitch angle, and pressure angle of the corresponding member must be used:

For a concave load face:

$Wr = Wt \cos \psi \, (\tan o \cos \gamma - \sin \psi \sin \gamma)$

For a convex load face

$Wr = Wt \cos \psi \, (\tan o \cos \gamma + \sin \psi \sin \gamma)$

Where *Wr* is radial force, lb (N).

A positive sign (+) indicates direction of force is away from the mating member. This is commonly called the separating force.

A negative sign (--) indicates direction of force is toward the mating member. This is commonly called the attracting force.

2.8.3. Axial:

The values of axial force, *Wx*, on bevel gears are given in the following formulas. The symbols in the formulas represent the values (e.g., tangential force, spiral angle, pitch angle, and pressure angle) for the gear or pinion member under consideration:

For a concave load face:

$Wx = Wt \cos \psi (\tan o \sin \gamma + \sin \psi \cos \gamma)$

For a convex load face:

$Wx = Wt \cos \psi \, (\tan o \sin \gamma - \sin \psi \cos \gamma)$

where

Wx is axial force, lb (N);

Wt is tangential force, lb (N);

Ô is normal pressure angle. This is the pressure angle on the loaded side of the tooth (depending upon direction of rotation);

γ is pitch angle of pinion or gear on bevel gears.

2.9. DESIGN OF GEAR TEETH:

The process of designing gear teeth is somewhat arbitrary in that the specific application in which the gear will be used determines many of the key design parameters. Recommended design practices are published in the AGMA standard 2005-D03, Design Manual for Bevel Gear Teeth. This design standard illustrates all aspects of bevel gear tooth design, starting from preliminary design values and progressing towards a finished design ready to be analyzed. Not only does it give recommended practices for design, it also covers manufacturing considerations, inspection methods, lubrication, mounting methods, and appropriate drawing formats. While this is certainly an invaluable tool published in order to provide one guideline for the design of bevel gears across all industries, it does not always properly differentiate design parameters that should be used for one industry versus another. For example, the automotive industry typically uses cast iron gears in transmission applications whereas a cast iron gear would not be feasible in a helicopter transmission because of the high level of loading and occurrence of peak loads that have the potential to be significantly higher than the load at normal operating conditions. As a result, a gear designer must have significant experience in the appropriate industry and be able to make intelligent decisions based on the specific application, which may or may not agree with the AGMA recommendations. In addition, many of the recommendations are based on spiral bevel gears meshing at a shaft angle of 90 degrees, whereas in this application, the bevel gears mesh at a shaft angle of 57 degrees.

Spiral angle and pressure angle are two design parameters that help determine the shape of a spiral bevel gear tooth. Common design practices have determined that for spiral bevel gears, a pressure angle of twenty degrees and a spiral angle of thirty five degrees should be used. Following this common practice for selection of spiral angle establishes a good face contact ratio which maximizes smoothness and quietness during gear mesh. In regards to the selection of a pressure angle, a lower pressure angle increases the transverse contact ratio, a benefit which results in increased bending strength, while also increasing the risk of undercut which is a major concern. Lower pressure angles also help to reduce the axial and separating forces and increase the top lands and slot widths. These factors help to strengthen the gear teeth because the increased slot widths allow the use of larger fillet radii, resulting in increased bending strength. The contact stress is reduced however, as a result of the larger fillet radii, so close consideration is required to ensure the correct pressure angle is chosen for the intended application.

In addition, spiral bevel gears are designed such that the axial thrust load tends to move the pinion out of mesh. This helps to avoid the loss of backlash, defined as the clearance between mating components. While a lot of backlash is not desirable, small amounts of backlash are required to allow for proper lubrication, manufacturing errors, deflection under load, and differential expansion between the gears and housing.

As previously mentioned, the gear addressed throughout this paper is replacing a similar gear that operated in the fleet for many years. The main difference between the two gears is the number of teeth on the pinion which helps to achieve the proper gear reduction ratio to reduce the speed at the tail rotor. As a result, the design of this gear was simplified because not everything had to be developed from scratch. Important geometric design parameters remained constant between the old gear and the new gear in order to be able to use the existing bearings, transmission housings, seals, and other hardware. Minimal changes were made to the values for diametric pitch, pitch diameter, pitch angles, and face width, which are the basis for calculating the necessary geometric design parameters.

2.10 CONCLUSION

Thus in this chapter we studied the basic of gears , their forces, their design parameters and the material used in their manufacture.

CHAPTER THREE ACCIDENT AND ANALYSIS

CHAPTER THREE

3.1. INTRODUCTION:

This chapter discusses the accident that lead to the failure of the gear, its causes and who was responsible for the failure of the gear

3.2 THE ACCIDENT

The accident occurred at the workplace of the client. The gear manufactured by Kinetic gears was to be mounted on the shaft to be later assembled in the Ventra locomotive transmission system. The fitting type being interference fit, the gear was to be pushed on the shaft as it is generally required. However when the gear was initially simply loaded over the shaft, a gap of 12 mm was present. This gap itself is way too much. This, however was not noticed.

Figure 3.2.1 Too much Gap

Then the shaft was forcefully pushed down inside the gear hole to achieve the fitting. This lead to gathering of material on the shaft inner diameter step which should have altered the workmen of extra engagement of material and the fitting should have been stopped. This was not noticed and the shaft was kept on being pushed.

Figure 3.2.2 Deposition of excess material on shaft collar

This lead to stretching of material on the shaft up to 10 mm due to excess pressure applied.

Figure 3.2.3 Stretching of material on the shaft

Soon after this the gear cracked from its weakest point at the gear back face.

Figure 3.2.4 Crack along gear backface

3.3. VISUAL OBSERVATIONS:

The gear was damaged along the entire circumference of the gear. The crack started at the 12.7 mm diameter hole on the gear back face. From there it travelled along the entire gear face.

Due to excessive pressure applied during assembly there were pressure marks inside the gear hole.

3.4 Probable Causes

3.4.1 Taper

The gear could not properly fit inside the shaft. The excessive pressure marks on the gear-hole surface and on the shaft suggests that the taper may have been wrong.

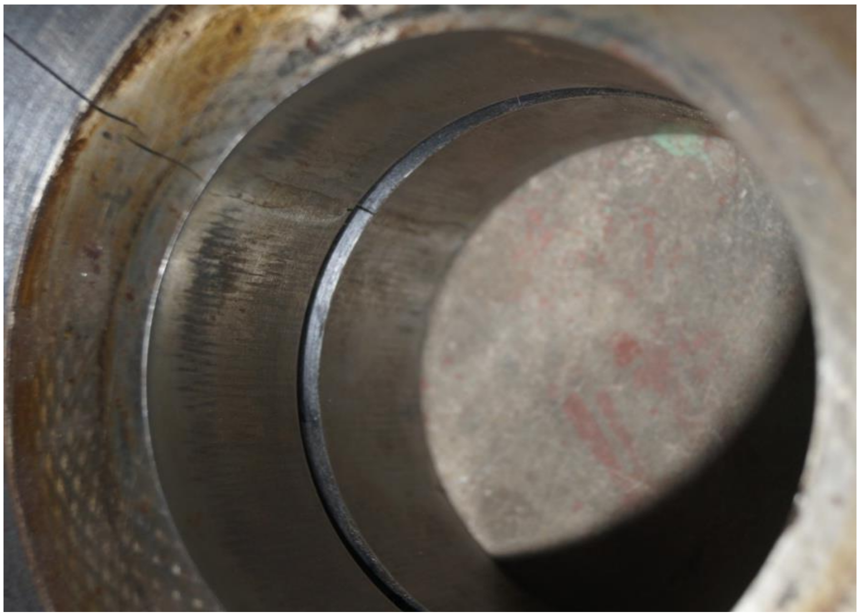

Figure 3.4.1 Excessive pressure marks on gear

3.4.2 Hole on gear back face

The weakest point on the gear, the hole was the origin of the crack. It is the prima face cause of failure of the gear. Hence it is a cause of failure of the gear

3.4.3 Material

The ease with which the gear broke down on simple manual assembly suggests the the material selection may have been wrong.

3.5 Conclusion

Thus three probable causes of failure have been identified.

CHAPTER FOUR
ANALYSIS AND RECTIFICATION

CHAPTER FOUR

4.1 INTRODUCTION

In this chapter, we discuss the taper of the gear, its significance in causing the failure and how the taper was rectified. The taper was responsible for the interference fit. Thus the wrong taper would lead to a wrong interference fit. The analysis of taper was done using the press fit formulae. The hole on the gear back face was also studied. The material of the gear was analyzed and then appropriately changed.

4.2 PRESS FIT

4.2.1 INTRODUCTION

In a press fit, the pressure *pf* is caused by the interference between the shaft and the hub. This pressure increases the radius of the hole and decreases the radius of the shaft. There is a radial displacement of the hub and a radial displacement of the shaft. The pressure *pf* is internal to the hub and external to the shaft.

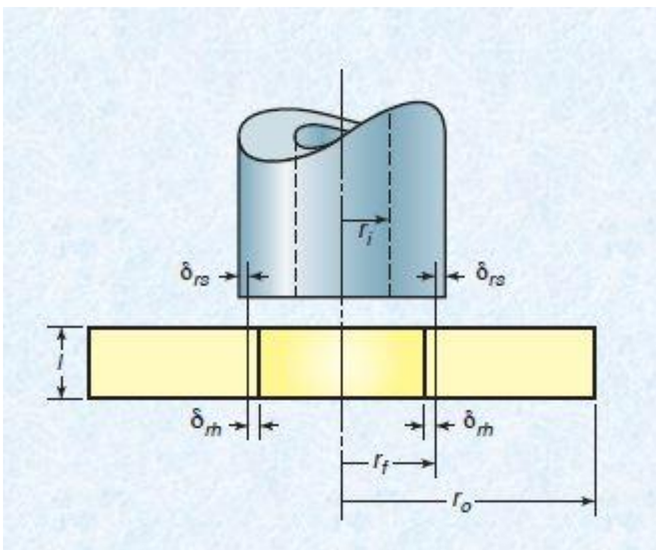

Figure 4.2.1 Press Fit Terminology

4.2.2 PRESS FIT FORMULAE

The radial displacement of the hub

$\delta rh = (rf/Eh)*(\sigma t - v*\sigma r)$

where

Eh = Modulus of elasticity of the hub

σt = Permissible circumferiential stress

σr = Permissible radial stress

v = Poisson's ratio of hub

δrh = radial displacement

The radial displacement of the shaft

$\delta rs = (rf/Es)*(\sigma t - v*\sigma r)$

where

Es = Modulus of elasticity of the hub

σt = Permissible circumferiential stress

σr = Permissible radial stress

v = Poisson's ratio of hub

δrs = radial displacement

The radial interenference

$\delta r = \delta rh - \delta rs$

Torque that can be transmitted

$$T = Pmax* rf$$

$$= 2\Pi \mu rf^2 l*pf$$

4.2.3 PRESS FIT CALCULATIONS – Original

Paramter	**Value**
Inner Diameter	109.2 mm
Outer Diameter	111.7 mm
Taper	1:40
Material	EN 9

ign parameters

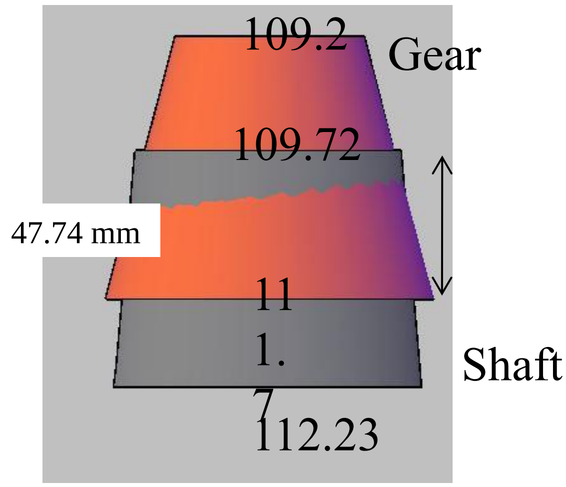

Figure 4.2.2 Taper 1:40

We calculated the interference displacement for original design.

δ_{rs}	0.00155
δ_{rh}	0.00151
δ_r	-0.0004
Force	106.94 N
Torque	5907.16 Nm

Table 4.2.2 Orignal interefernce caluclations

Here δ_{rs} is greater than δ_{rh} .

This means that the radial displacement of shaft in inward direction is more than the radial displacement of hub in outward direction

Negative value of δ_r suggests improper fitting.

4.2.4 PRESS FIT CALUCATIONS – Recommended

Parameter	**Value**
Inner Diameter	109.8 mm
Outer Diameter	111.8 mm
Taper	1:50
Material	EN -24

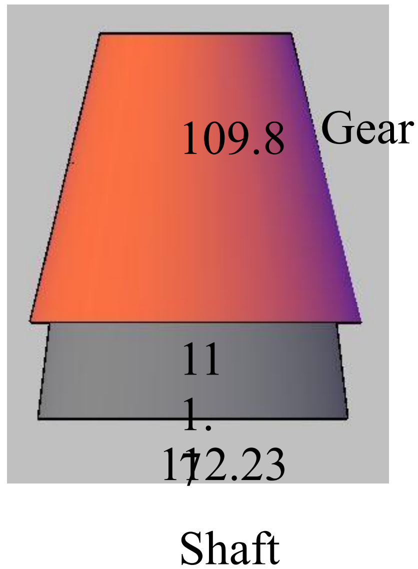

Figure 4.2.3 Taper 1:50

Hence the taper was changed. We decreased taper till we got a positive value of δ_r. It was taken care that the new taper could also be easily manufactured.

δ_{rs}	0.0021
δ_{rh}	0.0022
δ_r	+0.0001
Force	148.8 N
Torque	8246.24 Nm

Table 4.2.4 Recommended Interference Calculations

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4.3 HOLE

The 12.7 mm diameter hole on the back face of the gear was the origin of the crack. It was the weakest point on the gear. Post failure, we tried to approach the client regarding the significance of the hole so that alternatives could be suggested.

That, however, could not happen. The client simply informed us to remove the hole in the new design. Thus the 12.7 mm diameter hole was removed on the dictation of the client.

4.4 MATERIAL

EN 9 cracked under assembly. We needed a new material which will have higher *Sut* than EN 9 so that permissible stresses increase. At the same time its modulus of elasticity should be lesser than that of EN 9 so that δ_{rh} increases for better interference fit.

These two conditions were satisfied by EN 24

Change in stresses and internal pressure due to material change and other factors are studied in the stress analysis part of this chapter.

4.5 STRESS ANALYSIS

Stress analysis was done of the original design and then increase in strengths due to different design changes was shown. For this, the gear was considered as a thick cylinder, internally pressurized because of press fit by the shaft.

Lame’s equation for thick cylinders was used.

Here the circumferential stress is the max principle stress and is the criterion for design.

Different design changes included taper change, material change, with – hole condition and without- hole condition.

Strengths were calculated for *with hole* and *without hole* condition.

Stresses were also calculated post material change and taper change.

In the end, % increase in permissible stresses due to our recommendations was found out.

Factor of safety was taken as 10.

All values are in MPa	Original Design	+ Hole Removal	+ Material Change	+ Taper Change	% Increase
Interference Pressure	23.70	31.83	32.78	32.87	38.69 %
Radial Stress	-23.70	-31.83	-32.78	-32.87	38.69 %
Circumferent ial Stress	49.88	67.00	68.99	69.03	38.40 %
Longitudinal Stress	13.08	17.51	18.11	18.08	38.22 %

Table 4.5.1 Stress Analysis Calculations

4.6 CONCLUSION

Thus in this chapter we studied the original design and applying formulae of press fit, we changed the taper. Furthermore material was changed. Post these design changes stress analysis was done on the design.

CHAPTER FIVE
CONCLUSION

CHAPTER FIVE

CONCLUSION

5.1 CONCLUSION

In the end three design changes were recommended.

- 12.7 mm diameter Hole was removed
- Taper was changed to 1:50 for ease of assembly
- Material was changed to EN 24

The gear was manufactured accordingly and is said to have been working well.

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REFERENCES

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