Repair of Large, Surface-Degraded Industrial Gears — a New Approach

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This paper presents a new approach to repair industrial gears by showing a case study where pressure angle modification is also considered, differently from the past repairing procedures that dealt only with the modification of the profile shift coefficient.

A computer program has been developed to automatically determine the repair alternatives under two goals: minimize the stock removal or maximize gear tooth strength.

It will also be shown that the refurbishment of industrial gears using these approaches can result in a 30–40% savings, compared to the cost of a new part, without reducing gear strength. This paper will also show that the whole repair procedure can be carried out in no more than two weeks, depending on the gear size.

Introduction

Heavy industry components are expensive for of two reasons: first, they are large and are manufactured in small batches, or even individually, and, second, their replacement is not easy. Usually (and for the same reasons), the machinery suppliers or the Original Equipment Manufacturer companies (OEMs) do not have spare parts available in enough time, thus the time spent from the order placement to delivery is long. Some of these companies may have ceased their activities, leaving their machinery users without any after-sales support. Despite this situation, the research in the field of overhauling and repairing used gears to bring them new life and longer durability is practically absent, while a great effort is spent to improve gear performances and production through new materials, heat treatments, manufacturing, and surface finishing processes.

Machine capabilities and tool manufacturing are among the reasons that strongly limited gear refurbishment and hindered the research in this field. For a long time, the repair of a gear was restricted to the modification of the profile shift coefficient, while the modification of the pressure angle was impracticable, due to the axis motion limitations of the conventional gear machinery and the cost of fabrication of the gear cutters. For example, the modification of the profile shift coefficient is a technique already applied some decades ago to repair a gearset, as can be seen in (Ref. 5). However, the possibility of changing the pressure angle from 20° to a fractioned angle was costly because of the machining tool's limitations. It was not worth to manufacture a gear cutter with a non-standard pressure angle to save a single damaged gear. The multi-axis CNC machine centers overcame this limitation by allowing modifications of both the pressure angle and the profile shift coefficient.

The following innovations are enabling new types of modification to be applied to gear refurbishment.

- Introduction of CNC machine centers with multi axis and CNC gear grinding machines
- Development of new cutting tool materials which are able to machine parts with a surface hardness of more than 60 HRC
- Advancements in reverse engineering
- Improvement of the central processor speed of computers

These modifications produce refurbishments with great accuracy, in a timely manner, and with cost savings.

From a sustainability point of view, the repair of gears reduces the waste of components, which could be in service condition, keeping them working up to their obsolescence. Moreover, a study of 200 gears, carried out by one of the authors during the last five years, shows that the repair cost of a gear part can correspond to an average value of 37% of the price of a new one. Sixty-three percent of the gears considered in this study were fabricated with quench and tempered materials, 32% were case-hardened, and 5% were normalized. It can therefore be concluded that the repairing of heavy components seems to be the more convenient way to fulfill the issues of maintenance deadlines and premature decommissions.

In a few words, there are two good reasons for determining if a repair job is desirable: time and cost.

Background

It is important to highlight that this paper is mainly focused on the repair of industrial gearboxes; hence, the procedures are focused on the common practice of this sector. The parameters (e.g., the minimum service factor of the analytical strength analysis) may need to be updated if the approach is applied in another sector; a "mild" damage in the steelmakers segment, in fact, can be considered "severe" in the wind power segment.

This research does not rely on the search of the root causes of the failures. Usually, a gear specialist had already done the failure analysis before the unit was sent to the repair shop. Moreover, the results of the failure analysis are not available to third parties, including the repair service suppliers. The duties of the companies that provide repair servicing of gearboxes are usually limited to equipment overhauling (knowing which parts will be replaced, repaired or reused), assembling, and testing.

Nevertheless, it is important for these companies to know the gear failure modes (Ref. 1) in order to determine if the gear design can be improved to avoid future premature failures. The repair service companies should also be skilled enough to know the manufacturing processes available, and their limits.

The proposed approach applies only to gears affected by mild damages. For example, gears that present cracking, fracture, bending fatigue, and severe plastic deformation cannot usually be repaired because of their heavily impaired geometry and mechanical properties. Gears affected by these types of failures can be easily identified and discarded by a simple visual inspection or other quick inspection methods, like a dye penetrant test (Fig. 1).

Gears that do not present severe failures may thus be repaired and put into service again. Wear, scuffing and Hertzian Fatigue are failure modes that the proposed method can potentially remove (Ref. 1).

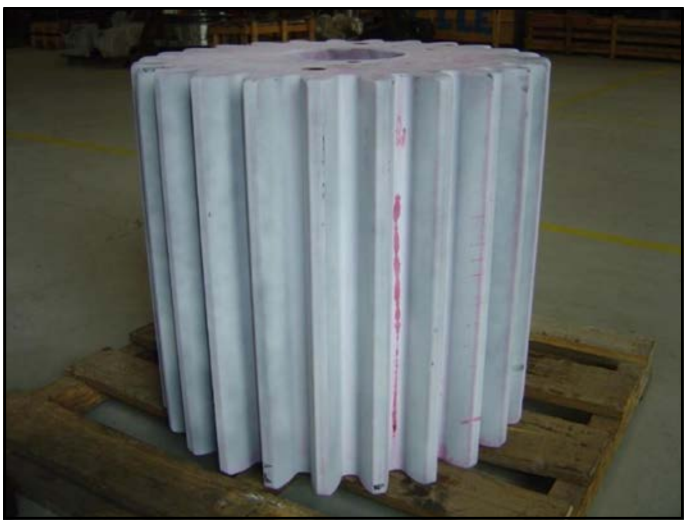


Figure 1 Crack revealed by dye penetrant test.

Methodology

The first step consists of data gathering. If either the users or the manufacturers do not provide the gear data, the gears have to be accurately measured in order to determine all the necessary input data to run an analytical strength calculation. The analytical strength is then determined by inputting the gear geometry (Ref. 12), materials, and application parameters. Available gear software (Ref. 23) can be used to calculate gear set strength against pitting, bending fatigue, scuffing, and micropitting.

In any case, it would be better to know the original strength of the gear set to avoid the design of a weaker pair.

Once the actual geometry and strength of the gears are known, the second step can begin. This step consists in evaluating which gear geometry modifications are possible in order to decide either how to remove the damage or also to guarantee an adequate strength of the gear. A *Matlab* script has been developed to assist the designers in this phase; the script is based on the procedure described earlier.

The input of the script is the gear geometry data gathered from the reverse engineering (Ref. 26) or by the original drawings (if available). The program calculates the coordinates of the active profile of the pinion and the gear.

Tooth pointing must be avoided since the strength applied on the tip during the meshing can lead to a breakage. It is recommended that the tooth tip thickness shall not be less than 20% of the normal module (Ref. 25).

Then, the modification of the profile begins with the increasing of the pressure angle by an amount of 0.1° and the calculation of the new geometry is carried out.

The algorithm enters into a loop, increasing the profile shift coefficient of the pinion by an increment of 0.001 — and consequently decreasing the profile shift coefficient of the gear to keep the backlash — up to the point that the damage is completely removed from the tooth surface.

The new tooth profile with the changes of the pressure angle, profile shift coefficient and root fillet (if necessary) is verified against the "undercut" due to a possible interference (lack of material) of the original tooth on the tooth root of the new profile. The undercut above the active root diameter impairs the gear performance, leading to other issues like transmission errors and reduction of the strength against the bending fatigue.

The algorithm tests a new increment of the pressure angle, increasing it once more by an amount of 0.1° since either the root radius does not reach 0.2 times normal module (Ref. 25), or the new profile intersects the original profile on the active root diameter, or the pressure angle exceeds 25° — whichever occurs first.

This program provides two approaches: the first is aimed at minimizing the stock removal during the machining process, while the second maximizes the strength that a repaired gear could reach, taking into consideration the tooth actual geometry and the damage present.

Even if the procedure intends to save both pinion and gear (if the backlash between the pair is still in tolerance), in most of the circumstances the pinion, which is the less expensive among the pair elements, must be re-manufactured to obtain an adequate backlash. If the pressure angle is also changed, both members will need to be re-manufactured.

Apart from the macro tooth modification, the micro geometry modification can also be included to increase the gear strength. Tip and root profile modification, as well as flank modifications, are examples of modifications that may increase the resistance against failures like scuffing, pitting, and micropitting.

A second analytical strength calculation is then carried out using the modified parameters. These results are then compared with the strength of the original geometry. The result is accepted if the gear set safety factors are above the minimum required by the industrial application.

Finally, the modified gear set drawings are sent to the manufacturing department staff so that they can decide how to machine the parts.

Today this approach is usually convenient under the economical perspective, thanks to multi-axis CNC machine centers and the development of the new gear CAD/CAM software packages integrated in these machines.

According to (Ref. 8), the traditional gear machining methods, such as hobbing and shaping, pose geometric limitations on manufacturers' ability to produce gears in small and medium batches. The gears manufactured have the same geometric parameters of the tools, such as pressure angle, addendum, and dedendum height, and in some cases, the protuberance for a grinding process. Different parameters and sizes require different and expensive tools with long delivery time, especially for small batches or even individually. This is usually a time-consuming process.

In recent years new gear machining methods have been developed that allow the use of multi-axis CNC machines and standard tools like end-type or disk-type milling cutters. The tools are not affected by the limits described in the previous paragraph, and the milling using a multi-axis CNC machine is not only user-friendly but the final machining process is also in accordance to higher gear standard accuracies.

The tools are typically stocked-standards with simple shapes, enabling a reduction in the cost of consumable tooling per gear. The tooling used with these methods is solid carbide or inserted carbide tooling capable of hard cutting after heat treatment (up to 62 HRC). This type of tooling has predictable tool wear, which can be controlled. Hence, it is possible to get less part-to-part variation in the manufacturing process.

Procedure to remove damages from the gear tooth surface using the profile shift coefficient and the pressure angle. This section discusses in detail the technical aspect of the procedure, i.e. the rationale of the software code. For the sake of clarity, the procedure is described considering a single transverse section, even though the whole surface of a flank should be considered.

In order to better understand the procedure, it is worth noting that, in this paper, it has been assumed that the amount of material to be removed is not compatible with maintaining the original center distance together with an acceptable backlash, as instead could happen in the daily repair practice for less severe damages. Therefore, in order to keep the same center distance and an acceptable backlash, a new pinion has to be conceived, designed, and realized, so that it can mesh with the "new" mating gear that has been machined inside the "re-usable portion" of the "old" and damaged gear. Of course, the "new" pinion will be designed to correctly mesh with the gear, according to the basic laws of gearing (Ref. 25). In this paper in particular, it was decided to keep the normal module the same, so the pressure angle of the mating gear shall be modified to maintain the pitch base the same in both elements. Thus, unless the damage is not so deep that it enables the modification of both pinion and gear keeping the backlash in tolerance, one of the gears (usually the pinion, since it is the less expensive among the pair elements) must be manufactured.

Let us consider the section where the deepest superficial damage has been located. This damage can be treated as a profile deviation. The above introduced damages, i.e. large macropits and spalls, as well as profile deviations that are typically caused by wear or micropitting, can, in fact, be properly detected and measured by means of typical gear inspection procedures, since they have a typical surface extension of some millimeters and depths higher than some hundreds of micrometers. For initial pitting, in which the extension and the depth of the craters are smaller, the possible measurement problems can be overcome using non-contact measurement techniques, if available, or by estimating an upper limit for the depth of the craters. In this case, in fact, the depth of the craters is related to the region where the maximum stress level caused by the Hertzian pressure distribution occurs that can be evaluated by means of multiaxial fatigue criteria (Refs. 2–3). By definition these profile deviations are normal to tooth profiles in transverse plane. Nevertheless, deviation may be measured normal to tooth flank surfaces, and such measured values are to be converted before comparing them with limits of tolerances by dividing the values by $\cos\beta_b$ (Ref. 22).

The output of the inspection is the profile diagram, which includes the profile trace. Deviations of the curve (including the damage) from a straight line represent deviations of the profile from an involute curve (Ref. 22). For the purpose of gear tooth repair, it suffices to measure the "Total profile deviation" F_α (i.e. the maximum deviation) and the corresponding radius r_F on the tooth profile.

Let us assume that the maximum active tooth profile damage is equal to F_α and that it is located at radius r_F (Fig. 2). Point D is belonging to the original profile of the tooth and point F is located on the bottom of the damage. The segment DF is the vector N, the magnitude of which is F_α and the direction of which is the unit normal vector n to the tooth surface at point D.

On the basis of Livtin's equations (Ref. 24) (Fig. 3), the coordinates x_D and y_D of point D (i.e., point of the original profile at the diameter where the failure is measured) are given by Equations 1 and 2:

$$x_D = r_b \cdot \cos\left[\left(\frac{\tan\lambda_b}{p}\cdot\sqrt{r_F^2 - r_b^2}\right) - \eta\right] + \sqrt{r_F^2 - r_b^2}\cdot\sin\left[\left(\frac{\tan\lambda_b}{p}\cdot\sqrt{r_F^2 - r_b^2}\right) - \eta\right] \quad (1)$$

$$y_D = r_b \cdot \cos\left[\left(\frac{\tan\lambda_b}{p}\cdot\sqrt{r_F^2 - r_b^2}\right) - \eta\right] - \sqrt{r_F^2 - r_b^2}\cdot\cos\left[\left(\frac{\tan\lambda_b}{p}\cdot\sqrt{r_F^2 - r_b^2}\right) - \eta\right] \quad (2)$$

where:

$$\eta = \frac{s_{bn}}{d_b} \quad (3)$$

s_{bn} Base tooth thickness measured along the arc of the base cylinder of helical gears

d_b Base diameter

r_b Base radius

λ_b Lead angle on the base cylinder (which is the complementary angle of the helix angle at base cylinder)

p Screw parameter (Axial displacement in screw motion corresponding to rotational through the angle of one radian. The screw parameter is invariant with respect to the radius r of the cylinder that intersects the helicoid

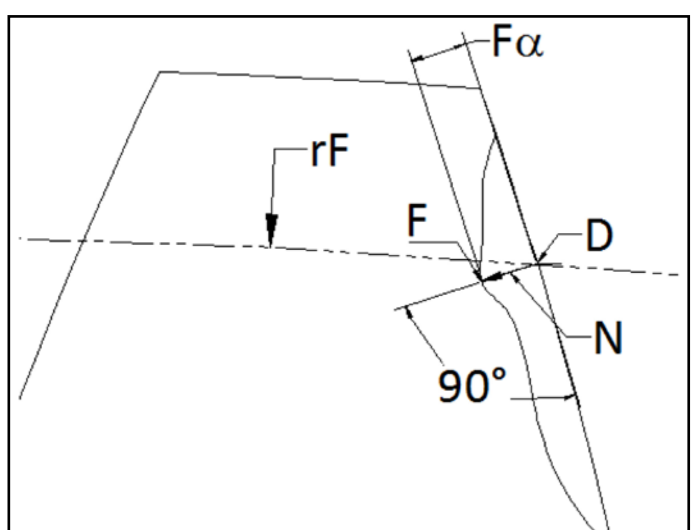


Figure 2 Normal vector of deepest failure.

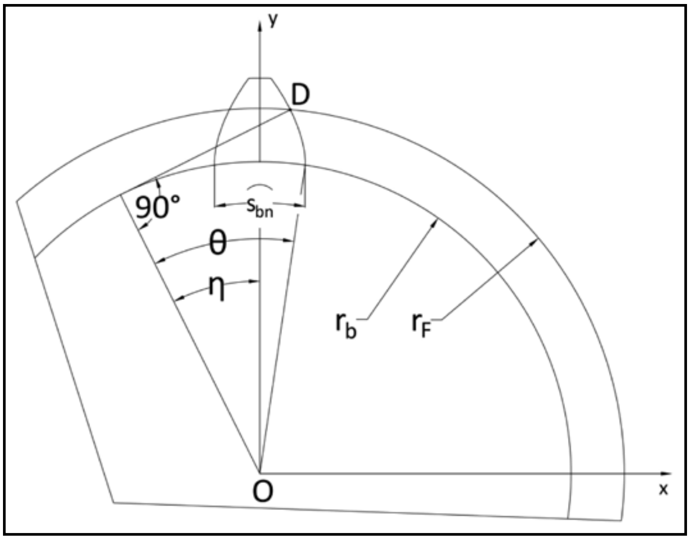


Figure 3 Gear profile generation (Ref. 22).

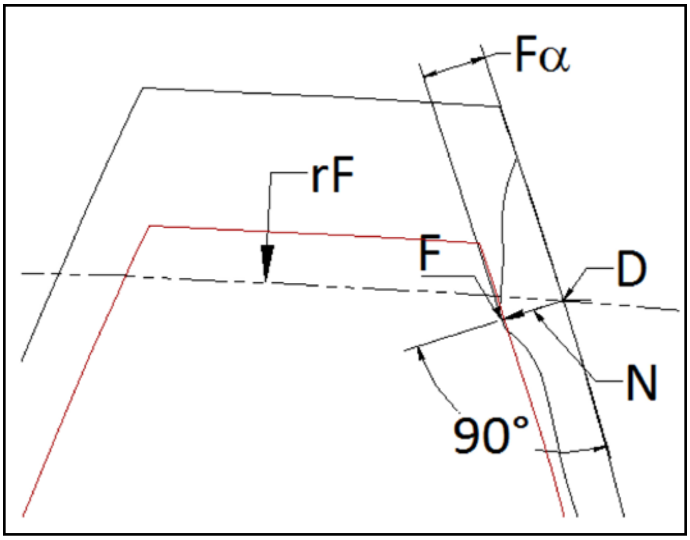


Figure 4 Modified profile, free from defects (red).

being considered, and is given by $p = r.tan\lambda$) (Ref. 24).

Flank normal transpose vector N at point D is given by Equation 4:

$$N = F_\alpha \cdot n = [\sin(\theta - \eta)\, F_\alpha - \cos(\theta - \eta) \cdot F_\alpha]^T \quad (4)$$

where:

$$\theta = \frac{\tan \lambda_b}{p} \cdot \sqrt{r_F^2 - r_b^2} \quad (5)$$

The coordinates x_F and y_F of point F (i.e. the "floor" of the defect) are therefore:

$$x_F = x_D - N_x \quad (6)$$
$$y_F = y_D - N_y \quad (7)$$

where:

N_x and N_y are the x and y components of vector N.

The aim of the repair procedure is to define a new gear tooth completely enclosed within the existing gear tooth (since it is only possible to remove material) and such that its external surface (free from defects) passes through point F. Figure 4 shows an example of the repaired tooth profile (in red).

This new tooth can be designed by reducing the profile shift coefficient and/or by changing the value of the pressure angle.

On the basis of the data of the original gear (gathered from reverse engineering or by the original drawings, if available) of the surface damage entity and on the actual loads, the developed script determines the pair of pressure angle and profile shift coefficient that remove the defect and maximizes tooth root strength.

However, several aspects are considered during these calculations.

The root radius fillet is usually changed, taking into account that the full fillet condition represents a limit.

Tooth pointing must also be avoided since a tip load application can result in a failure if the tip is completely hardened. It is recommended, in fact, that the tooth tip thickness shall not be less than 20% of the normal module. If the tip thickness condition is satisfied, the algorithm moves on the next check, which is the maximum radius fillet adopted on the tooth root diameter. If the radius fillet is less the maximum permitted, the original radius is kept; otherwise, the maximum fillet calculated is used.

The code plots two sets of graphics.

The first set of graphics consists in an overview of the area of the transversal section of the tooth (Fig. 8). The *y-axis* of these graphs shows the involute active profile in diameter units, and the *x-axis* shows the depth of the normal vector in respect of the involute curve (i.e. along unit normal vector direction $\boldsymbol{n} = \boldsymbol{N}/\|\boldsymbol{N}\|$). The shapes of the colored and the white area are the same for all the graphs of this output. The white area corresponds to the situations that, according to the developed method, are not possible to repair. This set is divided into two rows: the first row shows the parameters calculated to remove the damage with less stock removal as possible, and the second row shows the parameters that exploit the maximum strength the modified gear set can reach. The color represents the optimal value of the parameter, according to the color scale near each graph. More in detail, the first, second, and third columns represent the values of the pressure angle, the profile shift coefficient, and root fillet radius coefficient, respectively. The profile shift coefficient and the root fillet are consequences of the pressure angle adopted. The last graphic gives the reason why the code stops the loop — in other words, it shows that the new profile reaches the maximum pressure angle admissible (25°), or the minimum tooth tip thickness, or the new active profile will be undercut by the actual profile. The reader can see a vertical line, which marks the grinding limit for the production. Gears with damages beyond this limit must be firstly machined with a roughing process and then finished by grinding. There is also a horizontal line marked in these graphs. This line indicates the minimum diameter the gear can reach if the minimum profile shift coefficient is used (Fig. 8). It means that by turning the gear diameter, damages located above this line will be removed, regardless of the depth.

The second set of graphs (Fig. 12) takes into consideration only the measured damage and gives the original active profile of the gear-set, the damage depth, and the modified active profile calculated. Figure 12 also includes the values of the modified pressure angle, profile shift coefficient, and the tooth root fillet coefficient of both pinion and gear.

Figure 5 Refurbished gearbox.

Case Study — Quenched and Tempered Gearset

Figure 5 shows the case study gearbox. The user is a steelmaker, and this gearbox drives the wheel that moves the crane of either a 300/50 tons × 22 m hot metal charging crane or a 300/50 tons × 23.5 m teeming crane. This unit was first built in 1979, and some repairs have been carried out so far. The reason why the gearbox was removed from the crane is not known.

Table 1 summarizes the gear main data.

The gear geometry is calculated by gear calculation software (Ref. 23), and the profiles of the gear set are depicted in Figure 6. Gear set rating was calculated and the main results are summarized in Table 2. The minimum safety factors for these units are:

- Root safety: $S_{Fmin} \geq 1.5$
- Flank safety: $S_{Hmin} \geq 1.0$
- Safety against scuffing (integral temperature): $S_{int,min} \geq 2.0$
- Safety against scuffing (flash temperature): $S_{Bmin} \geq 2.0$
- Safety against micropitting: $S_{\lambda min} \geq 1.0$

It is worth noting that the safety against pitting is below the safety factor, which is supposed to be $S_{Hmin} \geq 1.0$. The pinion and the gear have safety factors equal to 0.77 and 0.80, respectively. This is evidence that the teeth may have their flanks damaged by pitting. Indeed, the inspection confirms the presence of pitting (Fig. 7).

The script developed is used, and the first information of the repair procedure is obtained. As Figure 8 shows, the repaired damage can be about 1.00 mm deep on the tooth tip and can linearly increase up to a depth of about 2.00 mm at the tooth root. The pressure angle can be increased up to 25° depending on the location and the depth of the damage. Deep damages located in an area near to the tooth root can be repaired using pressure angles from 15° to 20° (in blue). The modification of the profile shift coefficient is a consequence of the pressure angle calculated, and it can be seen that it varies from the original $x_2 = 0.4464$ to about $x_2 = -0.2$. The root radius fillet coefficient can be kept 0.3 mm as the original profile. The gear tip diameter can be reduced to a diameter below $d_{a2} < 645$ mm (the black horizontal line) if the minimum profile shift coefficient is used. In the second row, the graph shows the values for maximum strength, which means that the modified profile is the borderline from the colored area (maximum possible depth).

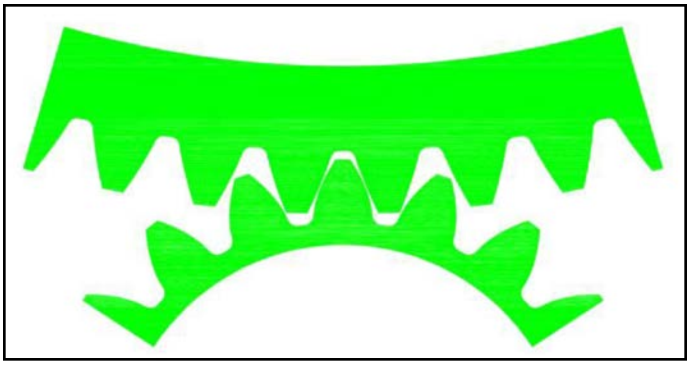


Figure 6 Profile of original pair.

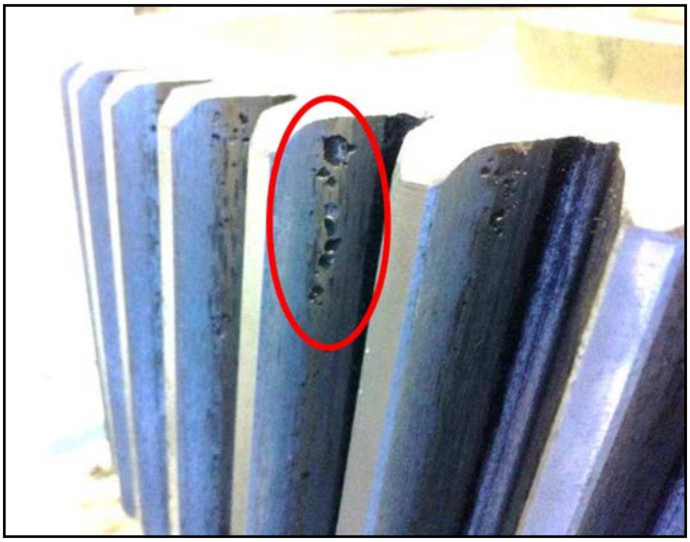


Figure 7 Pitting detail.

Table 1 Gear set — main data

	Gear 1	Gear 2
Power (kW)	71.5	
Speed (1/min)	142.91	34.81
Torque (Nm)	2522.5	10355.4
Application factor	2	
Required service life	20000	
Center distance (mm)	400	
Center distance tolerance	ISO 286 Measure js7	
Normal module (mm)	8.0	
Pressure angle at normal section (°)	20.0000	
Helix angle at reference circle (°)	9.0375°	
Number of teeth	19	78
Face width (mm)	146.00	140.00
Profile shift coefficient	0.5000	0.4464
Material	34 CrNiMo 6	42CrMo 4
Surface hardness (HB)	300	260
Addendum coefficient	1.000	1.000
Dedendum coefficient	1.250	1.250
Root radius factor	0.300	0.300
Type of profile modification	None	None
Tip relief (μm)	8.50	9.20
Lubrication type	Oil bath lubrication	
Type of oil	Mobilgear 600 XP 320	
Oil temperature (°C)	70	

Table 2 Rating results of pair (9); (11); (13); (14); (15); (16); (17); (18); (19)

Contact ratio (Transverse/Overlap/Total)	1.4267/0.8750/2.3017	
	Gear 1	Gear 2
Actual tip circle $d_{a,e}$ (mm)	177.015	654.090
Root safety	2.09	1.84
Flank safety	0.77	0.80
Safety against scuffing (integral temperature)	4.03	
Safety against scuffing (flash temperature)	5.40	
Safety against micro-pitting	0.96	

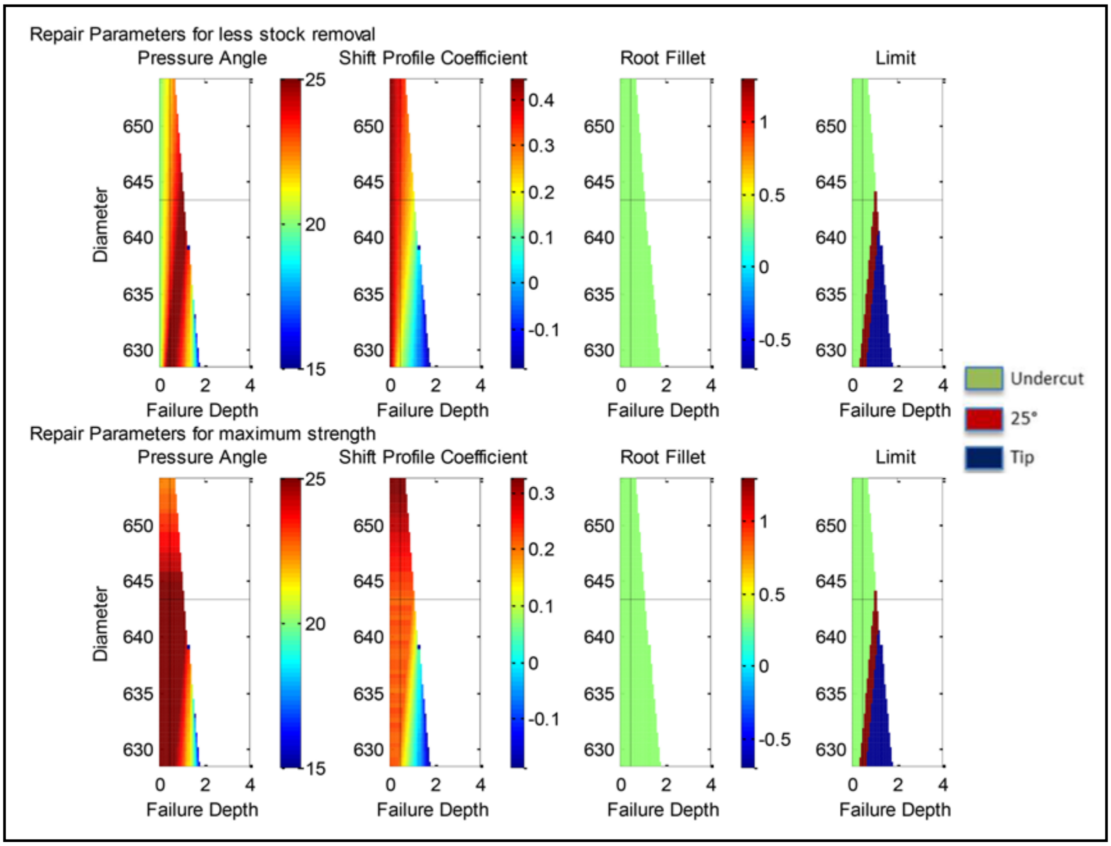


Figure 8 Modification of damaged gear.

The gear profile has also been inspected (Fig. 9a) (Ref. 22). The tooth of the gear profile surface that has the deepest pits is marked (Fig. 7); the profile chart is shown in Figure 9b. The depth and diameter location of the deepest pit are, respectively, F_α and $D1\_2$ of the left profile.

As can be seen, the pit is about 0.4 mm deep (the value in the chart is in microns) and it is located at diameter 634.710 mm. Despite the damage on the left flank, the right involute profile is in tolerance (DIN quality 9). The light blue arrow in Figure 10 shows the diameter of the deepest pit.

The coordinates of point D are calculated using Equations 1 and 2:

$$x_D = 317.277 \text{ mm} \quad (8)$$

$$y_D = -7.035 \text{ mm} \quad (9)$$

Applying Equation 4, the normal vector N is:

$$N = [0.122 \quad 0.342]^T \quad (10)$$

Thus the coordinates of the damage depth (Point F), using Equations 6 and 7 for x and y, respectively are:

$$x_F = 317.155 \text{ mm} \quad (11)$$

$$x_F = -6.693 \text{ mm} \quad (12)$$

The hypothesis for repair is to change the pressure angle to the maximum and the profile shift coefficient to the minimum in order to remove the damage and minimize the stock removal, avoiding tip pointing and undercut of the active profile of the repaired gear.

Figure 11 shows a detail of the gear profile and the location of the pitting.

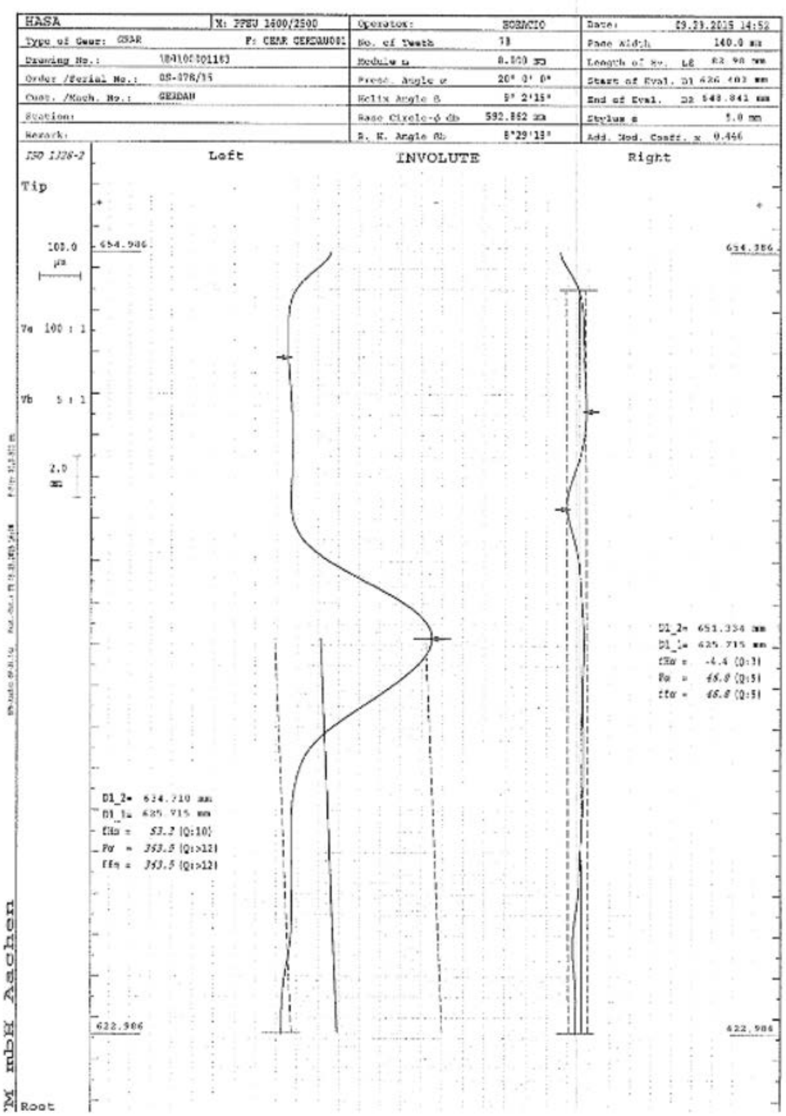
The second output of the script gives the profile of the original and modified teeth of the pinion and the gear, as well as the position and the depth of the damage. The second output also gives the nominal values of the parameters necessary to remove the damage considering the premises above. Normal pressure angle α_n, the profile shift coefficient x_2 and x_1, and the root fillet coefficient ρ are shown (Fig. 12).

Figure 13 shows a comparison between the new profile and the previous one. The red area represents the stock removal of the gear, and the yellow area is the complement of the profile of the new pinion that might be manufactured.

Table 3 summarizes the results of the calculations of the strength of the refurbished gear. Even if the software did not consider gear strength, since the decision was to minimize the stock removal, the root safety factors of the revised gear set are higher than the original gear pair. In particular, the safety against micropitting has been greatly improved. Even if pitting safety is still unsatisfactory, this result was accepted since the original gear pair has operated satisfactorily for about 30 years in similar conditions. It is worth noting that this gear has not been surface heat treated, hence its pitting resistance can still be increased by means of a flame or induction hardening treatment prior to finish grinding. Practically, these considerations demonstrate how the results of this approach can also supply useful hints to define a proper refurbishment procedure.

Using this procedure, the refurbishment of this case study may take about five days to be executed, considering in addition the machining process (turning, milling, and grinding). The machining and grinding processes were estimated and shall be evaluated since the production stages were not done. Figure 14 displays the Gantt chart from time elapsed up to the drawings' elaboration and the time estimated in the production to conclude process.

(A)



(B)

Figure 9 Damaged gear placed in gear-testing machine; a) — result of inspection; b) — pitting is about 0.4 mm deep and located in diameter of 634.710 mm.

Finally, an estimated cost to repair this gear is about 35% of the cost of a new one, apart from the cost to produce a new pinion.

Discussion and Future Works

The strength against micropitting greatly increased. However, this verification is of scarce relevance for industrial gears, while it is very important for wind power and turbo machinery gearboxes.

The pitting resistance is a critical factor for the quenched and tempered gears used in this study. This is due to the fact that these gears were through-hardened. The pitting resistance will increase greatly if their surfaces are nitride- or induction/flame-hardened. However, the increase in safety is noticed in comparison from the actual profile to the new profile. Consideration should be given to part distortion and original material selection if surface hardening processes are to be used.

The methodology tries to save at least one of the two mating gears (usually the more expensive one). This is the drawback of the methodology, unless the necessary intervention to repair is so superficial that the backlash between the pair is still in the tolerance, one of the gears of the pair (the pinion or the gear) might be sent to the scrap heap.

Delivery time is an important variable in repair procedures. As can be seen on the case study, the time spent (less than a week) in repairing is quite satisfactory if the meshing member does not need to be manufactured. However, even if a pinion is needed to be manufactured, the time spent can be less than the manufacturing of the whole gear set.

The cost is also relevant, and according to the author records, the cost of a gear repair is about 35% to 45% of the cost of a new part (not including the manufacturing of the mating gear, if necessary), depending on which material is used.

The effect of the profile and lead modifications were not evaluated in this research, so the software should take into consideration the micro-modifications, which improve the gear resistance to pitting, micropitting and scuffing.

In this work, the damages were inspected in a two-dimensional gear-testing machine. The damage can be topologically mapped in a three-dimensional machine, so the repair method can be further optimized.

In future works, to perform a better repair procedure an amount of stock removal must be considered and included in the script to improve the accuracy of the damage depth due to:

- The portion of the damage below the probe that is not reached, a consequence of its diameter

Table 3 Rating results of modified profile (9); (11); (13); (14); (15); (16); (17); (18); (19)

Contact ratio (Transverse/Overlap/Total)	1.3312/0.8750/2.2062	
	Gear 1	Gear 2
Actual tip circle da,e (mm)	179.397	651.946
Root safety	2.13	1.90
Flank safety	0.79	0.81
Safety against scuffing (integral temperature)	4.08	
Safety against scuffing (flash temperature)	5.56	
Safety against micro-pitting	1.35	

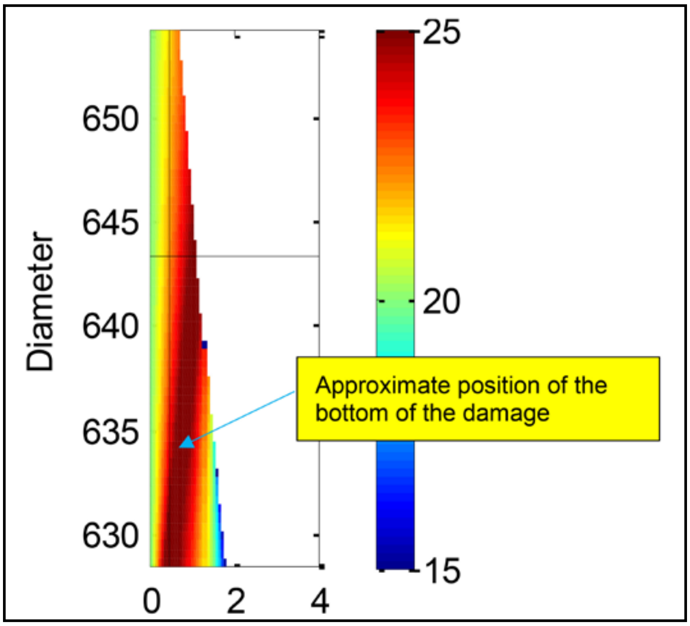


Figure 10 **Position of bottom of damage inspected.**

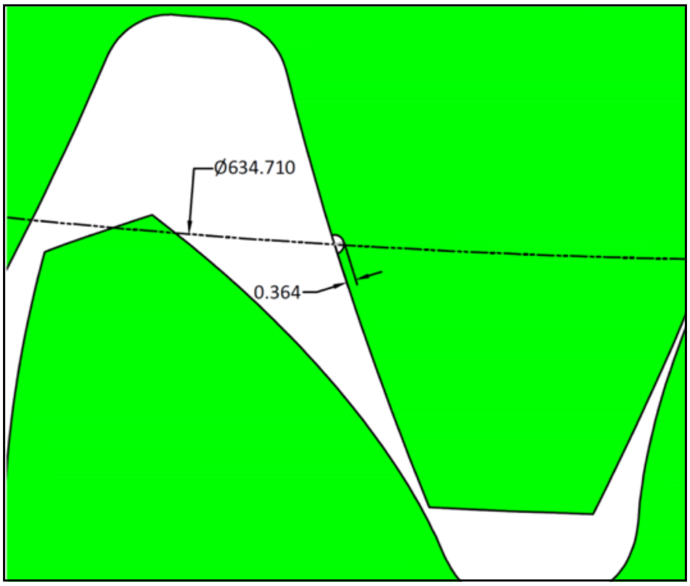


Figure 11 **Detail of depth and position of pitting.**

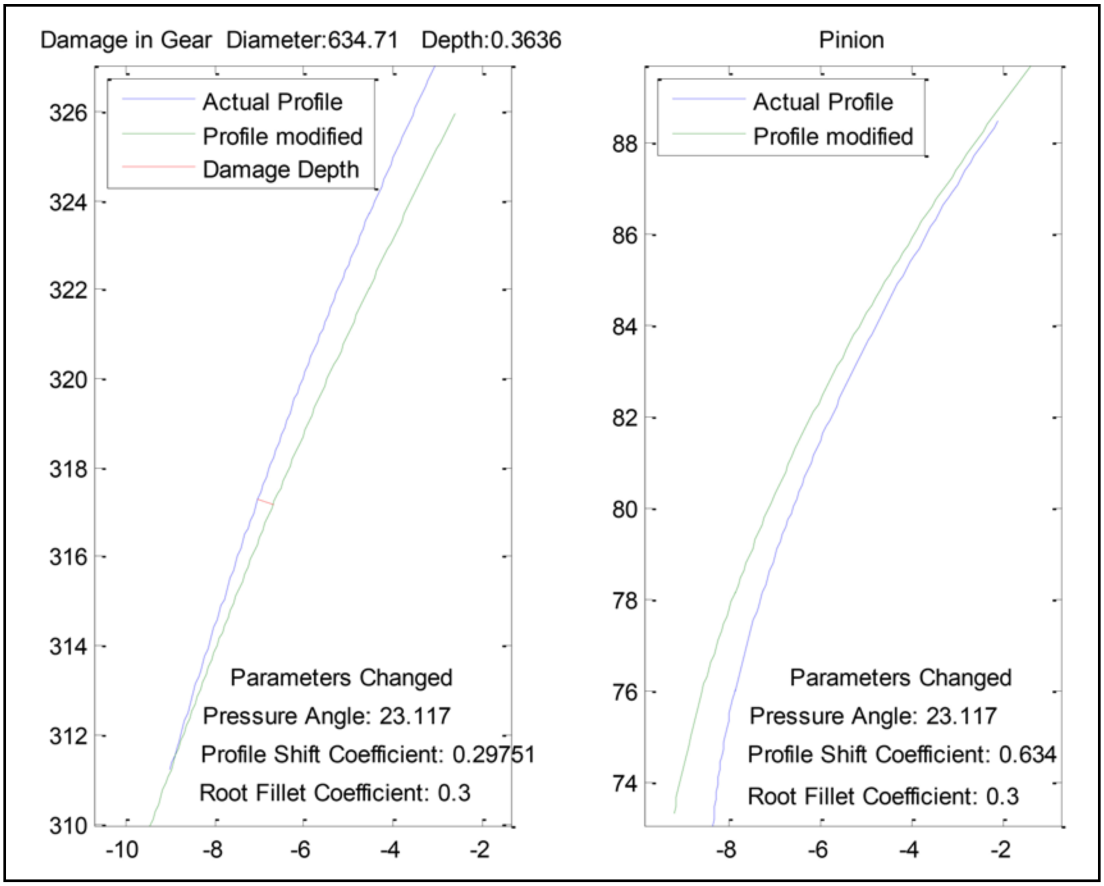


Figure 12 **Modification chart.**

- Small cracks typically associated with surface degradations

Gears can run in only one way; or they run loaded in one way and unloaded in the other way. Thus the flanks of the tooth can present different degrees of failure, and the gear flank wear is asymmetric. The approach can be optimized, taking into consideration the degree of damage from one flank to another, and less stock can be removed, or deeper failures can be removed, if an asymmetrical repair procedure is carried out. Moreover, the approach could even consider a refurbishment of the gears proposing an asymmetrical profile, i.e. — the right and left flanks having different tooth geometries.

The final product of the approach proposed in this article is a non-standard gear. It is important to evaluate the accuracy of the CAD/CAM software and the multi-axis CNC machinery in producing nonstandard gear sets.

Regarding surface-hardened gears, such as case-carburized and nitrided, the approach does not take into account that the hard layer of these gears could have been totally or partially removed during the machining process, due to uneven stock removal. For this reason, at the moment, the described procedure for gears is applicable to through-hardened gears, and additional investigations should be made on heat treatment processes in order to also apply it to surface-hardened gears. The potential for gear blank distortion must be carefully considered in light of dealing with a finished machine part that needs to keep its interface dimensions (e.g., bore and keyway) unchanged. This is a field for future research.

The machining process is an endless field to be explored. The cutting tools optimization focused in the repair of damaged gears is also a theme to be observed.

This approach could be applied not only in spur and helical gears but also in planetary gears, bevel gears, worm gears, asymmetrical gears, and other types of gears not mentioned.

Conclusion

The repair of mechanical components — especially gears — is a field with a lack of studies in the scientific community, but it is an active practice in the gear sector by the companies that provide maintenance servicing and by the after-sales of manufacturers. The cost of repair can be approximately 40% of the cost of a new gear.

This research has intended to provide a new approach to repair gears, creating criteria of acceptance between those gears that cannot be useful anymore and those that can be reused. The first stage of the method eliminates gears that present deep damage to the tooth flanks, and the successive stages focus on superficial failure modes that can be removed by the modification of the gear tooth profile geometry.

The method shown is effective in repairing failures like wear, macro-pitting, micropitting, and scuffing. Among the superficial failures, the method has graphically shown how deep the damage could be, and it has shown that the repairable damage depth is dependent on its position on the tooth profile.

The software developed gives the user the two alternatives — if it is intended only to remove the damage with less stock removal during the machining/grinding process, or if it is also mandatory to have the gear tooth strength increased. For the last option the code exploits the maximum strength that the repaired gear can reach, without extrapolating the geometrical limits of tooth pointing, undercut, and pressure angles beyond 25°.

It is important to note that this methodology is effective because of the coming of the modern multi-axis CNC machinery with integrated CAD/CAM gear software and the improvement of cutting tools.

The methodology takes about one day to be executed from the visual inspection to the development of the new drawings, regardless of gear set size. Most of the time is spent in the machining shop, turning, milling, and grinding the parts. The time spent in production of the gears shall be more thoroughly studied in future works.

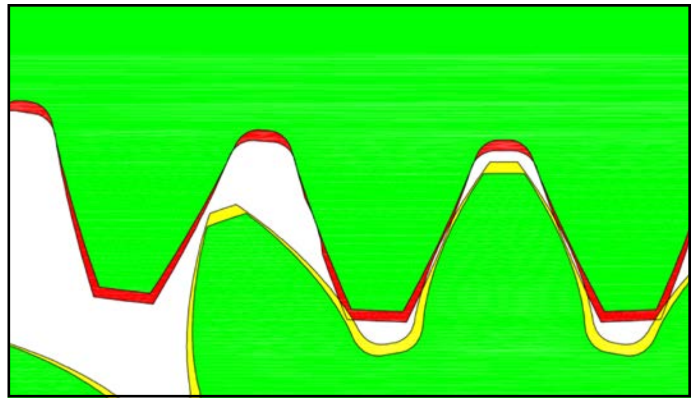


Figure 13 Comparative chart of actual and modified pair — red area is stock removal and yellow area is amount increased.

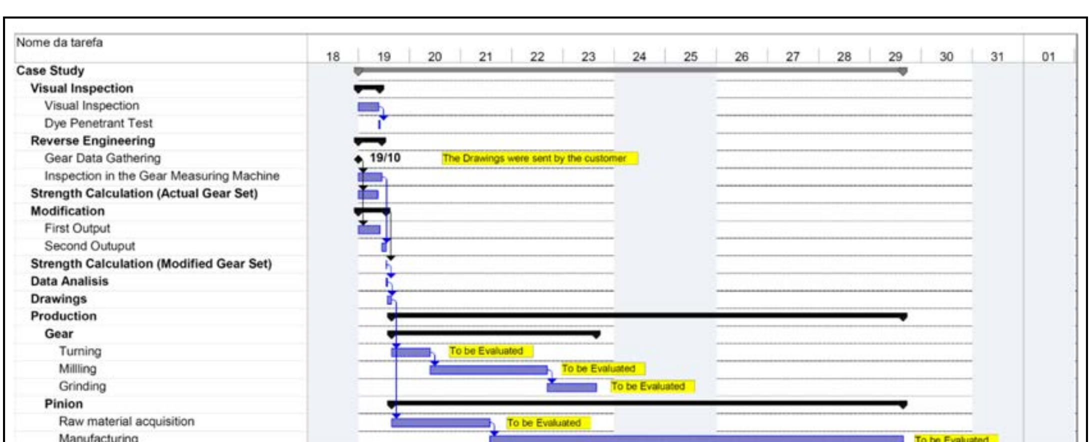


Figure 14 Time spent to complete repair.

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