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Resolution of micropitting in case-carburized helical gear pairs through tooth profile modification

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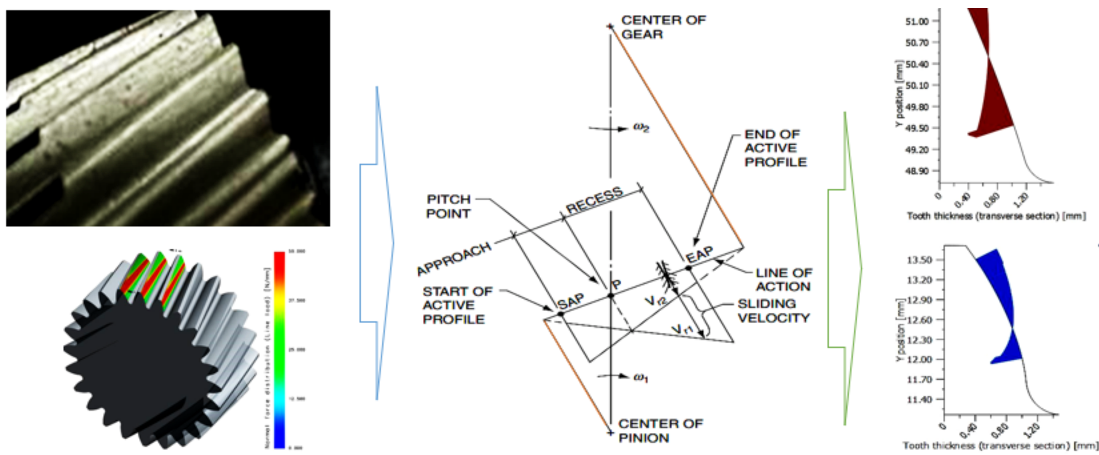

ABSTRACT

Helical gears are crucial for power transmission in parallel shaft systems, operating under various conditions such as load, speed, temperature, and environmental factors. These gears are typically case-hardened through carburization to handle higher torque transmission. However, this capability also introduces surface fatigue stresses, leading to potential surface-related failures. Micropitting is one such failure, characterized by changes in the tooth flank's surface texture, which can escalate and result in catastrophic damage. This research compares the outcomes of analytical and experimental investigations to identify potential zones where micropitting may initiate. The analytical approach was conducted using the KISSsys model, while the experimental work involved applying loads to the gearbox shaft using a dynamometer test rig under a defined load cycle. To assess changes in the gear surface, gear profile analysis was performed using a Zeiss CMM machine equipped with the GearPro program. It was evident that micropitting began below the pinion's pitch line. The tooth profile was subsequently modified with lead and crown adjustments. Following these modifications, micropitting was no longer observed during the same testing cycle. This finding was further validated through gear profile analysis.

Keywords: Micropitting, GearPro, Kisssys, Dynamometer, Gear Design,

INTRODUCTION

There has been much evidence showing the micropitting as one of the critical yet least known reason for gear failures. It is found in abundance in case hardened gears. Helical gears on account of helical contact and overlap ratio provide smooth meshing but is accompanied by frictional losses also. These losses become more evident in the case of low-speed pair. This research work is about the effect of tooth profile modification on micropitting. Both analytical and experimental analysis is performed to ensure the comparison and effect of modification on micropitting.

Specific Sliding

Gear pair while operating roll and slide over each other. Fig-1 shows the rolling and sliding parameter of pinion and gear with subscript1 and 2 respectively. Over rolling and sliding action rolling action is always preferred because it allows the entrance of lubricant between the meshing teeth which results in corresponding increase in oil film thickness. On the contrary sliding velocity is detrimental as it releases heat, lowers the gear pair performance, and increased the asperity distress. It also shears the oil film thickness thus reducing the effective lubricant film thickness in the contact area under sliding. As shown in the Figure-1 the rolling velocity magnitude increases from zero at interference point to max at the end of contact point shown as SAP and EAP for both gear

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and pinion the distance between the rolling velocity vector of Pinion and Gear represents the sliding velocity as shown in the Figure-1.The relative sliding velocity at any point of contact on contact line is given as a product of distance of the concerned point from pitch point to the sum of angular velocities of Pinion and gear.

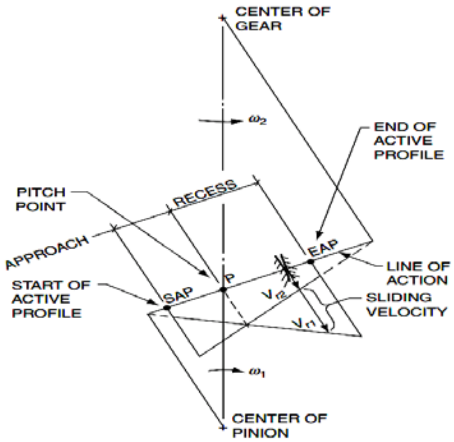


Figure 1 Rolling and sliding Velocity.

$$V_{r1} = \omega_1 * \rho_1 \quad (1)$$
$$V_{r2} = \omega_2 * \rho_2 \quad (2)$$
$$V_{s1} = V_{r1} - V_{r2} \quad (3)$$
$$V_{s2} = -V_{s1} \quad (4)$$
$$V_e = V_{r1} + V_{r2} \quad (5)$$

Similarly specific sliding ratio can be given as below:

$$V_{ss1} = \frac{V_{s1}}{V_{r1}} \quad (6)$$
$$V_{ss2} = \frac{V_{s2}}{V_{r2}} \quad (7)$$

Equation 6 and 7 provides the way to calculate the specific sliding ratio which is dimensionless parameter such that it is positive in addenda with a range from zero to +1. Its value id zero at pitch point whereas +1 is at the interference point with mating gear. Its value is negative for dedenda with zero value at pitch point and -∞ at interference point. Negative sliding refers to opposite direction for sliding to rolling direction. Negative sliding is important as it promotes Hertzian Fatigue.

Gear Accuracies

Generally, for the functional requirement, gear errors need to be controlled and is classified into four functional groups as shown in Table-1.

Table-1.- Function Group of Deviation

Functional Group		Important Deviations
G	Uniformity of the transmission of movement	F_i', f_i', F_i'', f_i'', F_r, F_p
L	Smooth running and dynamic loading capacity	f_i', $f_{H\beta}$, f_i'', F_r, F_p, F_f, f_p, f_{pe}
T	Static load Capacity	$f_{H\beta}$, f_{pe}, TRA
N	No indication of the function	F_i'', $f_{H\beta}$, F_f, f_i''

Whenever a particular gear required a general service property without taking special functional requirements in account, a single gear tooth quality is prescribed, for example quality 6. For other case the fulfilment is done by adding the specific functional group with gear quality like L6. L6 means gear with quality grade 6 and it need to be taken care for smooth running and dynamic loading capacity. To streamline the testing requirement these are further subcategorised in A, B and C. In our case it is B.

Present concern

The rigorous usage of gearbox for mixing application deals with continuous loaded duty for almost 12 hours per day basis. Such gears are subjected with surface related failure more as compared to root related fatigue bending failure. Based on the field experience, micropitting is observed on the tooth flank below the pitch circle but this behaviour is not common for all gears used in mixing industry. Gear pair considered in this research work are subjected to micropitting in the predicted zone and there needs a means to avoid it.

Table 2.-Gear Pair Data

Parameter	Pinion	Gear
No of teeth	24	97
Normal Module	1	
Helix Angle (degree)	13-24'-8"	
Pressure Angle(degree)	20	
Rpm	160	39.6
Material-Case hardened and Ground	20MnCr5	

To continue the research, work the most popular gear pair is considered as per Table-1

Gear and Pinion is Case hardened and ground. The quality grade is considered as Grade -6 as per ISO:1328. Gear pair need to analysed for various parameters based on past work done to find a way to avoid or reduce the micropitting.

Literature Review

To understand Micropitting various set of experimentation and related research has been carried since long till now. During investigation of past research work through Literature review it was found that the experimentations were considered on various testing setup starting from roller disc to actual gears in controlled environment. This section details some of the important literatures which has helped for the current research.

Heli Liu, Huaiju Liu, Caichao Zhu, Ye Zhou [1], presented a paper on A Review on micropitting studies of steel gears. In this paper authors have analyzed the various research work done to take care of micro pitting of steel gears from past data. Tooth contact pattern was analyzed between modified and unmodified gears to observe the stress pattern and their values. Various tooth surface with different roughness values were analyzed to observe the contact fatigue failure risk. One of the interesting analysis was to find the effect of surface wear on micropitting. Wear depth of gear and pinions were plotted along the line of action. As this was the review research work, it also involved the lubricant effect and various gear testing equipment like FZG, PCS instrument. It was concluded that micropitting can be reduced by microscopic and macroscopic geometry modification. It was also observed that micropitting initiated on account of contact pressure whereas its propagation is driven by specific sliding. Addition of the wear process to it was further deteriorating the condition and micropitting was more visible.

Webster & Norbart [2], presented a paper on Experimental investigation of micropitting using a roller disk machine. This investigation was mainly concentrated on the factors influencing the micropitting enhancement. Various operating parameters were reviewed. To perform the experiment roller disc machine setup was used. Different geometrical modification like flat and chamfered profile were used with material 17CrMo6 having hardness 58HRC and 60HRC respectively. Both items were grounded on OD. Roller disc machine as expected has shown the micropitting in the dedendum zone. It was observed that the microprinting occurred on surfaces having relatively low speed. Increasing the surface finish caused the increase in specific film thickness and was the useful method to avoid the micropitting. Slide to roll ratio was also reviewed that reducing its magnitude retards micropitting as well. Overall, it weas the slide to roll ratio, surface finish and load were the potential reason for micropitting and taking necessary action to them could help in reducing the micropitting.

Dario Croccolo, Massimiliano De Agostinis, Giorgio Olmi,Nicolo Vincenzi [3], presented a paper on a practical approach to gear design and Lubrication. To meet the current competitive environment of better efficiency, torque transmission this research work is an important reference. It has been a detailed work of data collection from practical examples to define the process and relevant parameters for gear design. Spur gear pair was the subject of this research work. Two different materials are considered to investigate the hardening effect as 18NiCrMo5 & 41CrAlMo7.Tooth pairs were with Asymmetric and symmetric construction. Methododlogy of gear design started from initial steps from start of module selection to system design. Apart from gear design the lubrication system and its application was the important area of research work. This also detailed the way of preparing the drawings and technical parameters required to manufacture a gear.

Hasan Ozturk, Mustafa Sabuncu, Isa Yesilyurt [4], presented a paper on the Early detection of pitting Damages in gears using Mean Frequency of Scalogram. This research work includes topics related to the gear defects alongwith contact ratio with its effect and way to measurement of effect. Time and Frequency domain and related research are the priority of this research work. As the work was related to detection of pitting Test rig used was with pitted gears in a two-stage helical gearbox, Input and output of the gearbox was equipped with Induction motor and DC motor. Induction motor was being as a source of power input whereas the DC motor was acting as a source of load bank. DC motor which was acting as a load bank , because of it power capacity the face width of the gear pair were reduced to 4mm from 12mm width. Accelerometer in perpendicular position was used to each other to capture the vibration signals. To examine the response of healthy gears with pitted gears various curves were drawn. Frequency signals were considered as a mean to know the pitting in gears. Amplitude of vibration was considered as in indicative of the pitting growth.

A Oila , B A Shaw,C J Aylott, S J Bull [5], presented a paper on martensite decay in micropitted gears. Investigative work of this research work was alligned for the micropitted gears. Idea was to know how the martensitic layer alters during the micropitting process. Case carburized in gaseous medium helical gears were considered with material as 16MnCr5 steel. The gears were having the grain size limited to size of 7 as per ASTM standard. Back-to-Back test rig was the experimental test rig. Loading were considered from 5 million cycles to 50 million cycles. Lubricating oil was used was Mineral oil. Scanning electrom microscope was used to determine the surface condition after testing. Dark etching regions, white etching bands and mechanical properties were the important analysis parameter. Martensitic decay in gears were very much like the same parameters in bearing materials.

B R Hohn , K. Michaelis [6], presented a paper om Influence of oil temperature on gear failures. This research work provided a good information about the behavior of oil at high temperature and its associativity with the gear material. Oil's viscosity gets reduced as its temperature rises, this reduces the thickness of oil which results in the creation of thin film between the meshing gear flanks. Elevated temperatures sometimes cause detrimental effect in lubricating oil, allowing their chemical elements to react with adjacent materials which sometimes affect the endurance limit of the gears also. Lubricating oils must be selected based on the temperature condition and the environment for which it is subjected to. Thermal calculation and contact analysis is required to know the temperature rise and so that the lubricating oil can be selected accordingly.

L. Winkelmann, O. El-Saeed, M. Bell [7], presented a paper on the effect of Superfinishing on Gear Micropitting. This research work was on the gear surface finish. Experimental kit used was Standard FZG gear test setup. Baseline and superfinished gears were used in this experimental testing. Baseline gears which were unmodified version of specification of FVA information sheet 54 and superfinished gears were also as per FVA information sheet but were finished using chemically accelerated vibratory finishing method. Case Carburized gears were considered with material of 16MnCr5 and having quality grade 5. Base line gears were unmodified and having the surface finish of 0.47Ra on average basis. Superfinish gears were having the surface finish with average value of 0.1 Ra. Load stage test and Endurance test were considered for both pair of gears with defined load cycles. Visual effects of micropitting were considered as one of the result analyses whereas the profile deviation was the other parameter which was considered as checking points. Profile deviation value was observed more in base line gears than the superfinished gears.

Marco Antonio Muraro, Fabio Koda, Urbano Reisdorfer Jr., Carlos Henrique da Silva[8], presented a paper on the influence of contact stress distribution and specific film thickness on the wear of Spur gears During Pitting tests. Various parameters relative speed, surface finish, lubrication condition and temperature for their effect on the gears during the testing. FZG test setup was used as testing setup for this research work. AISI 8620 steel was used as Gear material. It was manufactured by shaving and milling operation. Two Torque values of 135Nm and steady state Torque value of 302 Nm with corresponding test temperatures of 60 degree and 90 degrees Celsius respectively were considered for Whole experiment. Gear image was analyzed to find the wear levels on the gear surfaces. Wear was found to have a strong relation with lubricant film thickness and was more on milled gear. Load sharing functions has huge influence on gear surface deterioration. Resistance to Wear was found inversely proportional with high surface finish value.

I.S. Al-Tubi, H.Long, J. Zhang, B. Shaw [9], presented a paper on Experimental and analytical study of gear micropitting initiation and propagation under varying loading conditions. Analytical and experimental approach is used here to continue and compare the investigation. Strong initiative was there to find the relation between varying load and micropitting initiation. ISO TC 15144-1 was used to investigate the analytical approach. Initial pit formation was analyzed while initial load cycles along with its propagation. This finding was compared with the analytical approach. Pinion addendum zone was the point where micropitting initiated initially. Pitting was getting engraved towards addendum over a period of time. Analytical approach identified lubricant film thickness and contact stress values as the responsible term for micropitting. Along with these other parameters were identified as surface finish of gears.

Michael Hein, Thomas Tobie, Karsten Stahl [10], presented a paper for Parametric study on the calculated risk of tooth flank fracture of case-hardened gears. This investigation gives an important insight for crack propagation below tooth surface. Analytical approach was carried through the ISO standard technical report after considering various parameters as guided by the standard document. Due to recent development and high surface finish condition, it was found that it is quite possible to control the micropitting but it may lead to other type of failure like sub surface crack initiation. To find the correlation of analytical work experimentation was done on FZG test setup. Numerous parameters were considered for the experimental analysis which included both external factor like load cycle, lubricant to internal geometric factor like tooth profile etc. This research work included various parameters as per FZG test which included material parameters, tooth flank geometry, external loads, local hertzian contact stress, normal radius of relative curvature and miscellaneous parameters. ISO technical document was very much in line with FZG test, and the results were also satisfactory.

Aleks Vrcek, Tobias Hultqvist, Tomas Johannesson, Par marklund, Roland Larsson [11], presented a paper on micropitting and wear characterization for different rolling bearing steels. This research work is useful for the work related to different surface hardness and methodologies followed on the surfaces. Bearing steels are under consideration for this research work. Discs are formed in the form these materials are subjected to high contact stress with lower lubricant film thickness. Hardening of these discs were done by different hardness process. Asperities plasticity was evaluated using Tribo testing. Surface failure was an important function of surface hardness variation among discs. Plain medium carbon steel behaved better than alloy steel from surface fatigue point of view. Surface roughness was the other parameter which was observed that a harder surface with poor surface finish was getting deteriorated earlier than comparatively softer surface with lower hardness.

Robin Olson, Mark Michaud, Jonathan Keller [12], presented a paper on case study of ISO/TS 6336-22 Micropitting calculation. High speed gear set test was first initiated with centrifugal compressor. Micropitting was observed in Pinion dedendum which later travelled to the addendum. Same gear set was tested through ISO standard analytically and the charts were plotted against the lubricant film thickness and stress on the gear tooth surface. The results were in comparison with the experimental work. The second work was done for the investigation of wind turbine gear test setup. Similar set work was performed with analytical and experimental approach an the results were compared to each other. In this part also the graphs were plotted with specific film thickness at two temperatures of 50 degree and 70 degree Celsius. All three cases were considered with helical gears. Geometries of the gears were arranged in such a way that different slide to roll ratio can be availed. For all the there cases, different Torque ratings were considered as 265Nm, 300Nm and 400Nm respectively. Lubricant oil M-200EP was considered for each case and specific film thickness was determines too. IN first two cases the specific film thickness was more than unity still there was micropitting which should not be there as per ISO standard recommendation. This released a conclusion that for higher specific film thickness it is difficult to estimate the micropitting.

Nadine Sagraloff, Thomas Tobie, Karstem Stahl [13], presented a paper on Suitability of the test results of micropitting tests according to FVA 54/7 for modern practical gear applications. In this research work profile modification and its effects are analyzed experimentally. Test gears were developed for the micropitting test. 16MnCr5 and 18CrNiMo7-6 were considered for the test gears with micro geometry, macro geometry and profile grinding method. For experimental part FZG back-to-back setup was used for this experiment. Test rig included both test gears and slave gears. Both gears were arranged as per standard practice. Test gear and slave gears were case hardened. Different graphs were plotted with respect to profile deviation and load stage and endurance test. New approach usage and the methods followed were almost similar there were no significant changes in the results as per FVA 54/7 as compared to the new process.

Mao Ueda, Benjamin Wainwright, Hugh Spikes, Amir Kadiric[14], presented a paper on effect of friction on micropitting. This investigation founded the effect of friction on micropitting. Ball on disc test rig was considered. This test rig was having an arrangement to trace the lubricant film thickness while experimentation. Friction was varied by using oil with different content. Other different variant of lubricating oils were explored for this experiment to analyze the results. Graphs were prepared between Friction parameter with respect to the load cycles during experimentation. 8 million cycles were considered as target load cycle. Ball's surface got damaged due to friction and it was visible under 200-micron vision. Defects were found as the function of the load cycles which were directly proportional to each other. Side by side surface finish of disc was also getting affected. Various graphs of different oil film thickness was clearly showing that lower the oil film thickness and more was the surface defects.

Outcome of Literature survey

Literature reviews were related to wide range of problems with different variant of problem ranging from disc to gears. The study of these research paper has provided the direction to way ahead for the current experimental work. The finding is detailed as below:

All the forces, sliding and rolling is occurring online of contact and is an important parameter for analytical as well as experimental work.

Lubricants must be selected according to the operating temperature it is subjected to along with environmental condition.

Gear surface failure is more affected by varying load acting on it even for the same gear speed.

Contact line includes the complete meshing cycle load and shall be analyzed carefully.

Gearbox must be designed as per international standard to maintain uniformity and rule out unseen load effects.

Lubricant film thickness below 1 micrometer has good correspondence standard material whereas for more than 1 micrometer micropitting may occur and there is no suitable guideline for it.

Specific sliding produces friction and has detrimental effect on gear surface and must be reduced as small as possible.

Surface finish of the gear pair has inverse proportional relation with miropitting growth.

Very fewer research is available with material 20MnCr5 as compared to 18CrNiMo-7-6 or 16MnCr material[15.

None of the research work were found for gears in mixing applications.

Actual research work with helical gears for similar scope as of this research work were not available.

Gear profile modification can be useful in gear surface related failure issues.

Research parameter for experimental work.

From the last section of literature review it was clear that Loads, surface finish, Lubricant, surface hardness, friction on account of sliding speed are the main parameter responsible for micropitting. In the current research work load and specific sliding are considered as the main parameter whereas other parameters like lubricant, surface finish, surface hardness are considered as non-affecting parameters because they are considered in the best possible form and achievable from cost and engineering process point of view. Out of these parameters lubricant parameter is considered constant and hence the lubricant is selected suitably. Surface hardness and surface finish are material property and is almost frozen once 20MnCr5 material is selected with manufacturing quality grade 5. Lubricant is also selected based on the maximum usage and availability in the current market scenario[16].

Gear tooth modification to reduce the sliding speed.

Gear pair while meshing with each other possess rolling as well as sliding velocity. Rolling velocity is desirable while sliding velocity is not. However, it is not possible to eliminate the sliding velocity. We can control the sliding velocity to minimum value and balance it on both gear and pinion end. Exhaustive analysis is performed in KISSOFT to get the desired profile shift value to have balanced specific sliding ratio.

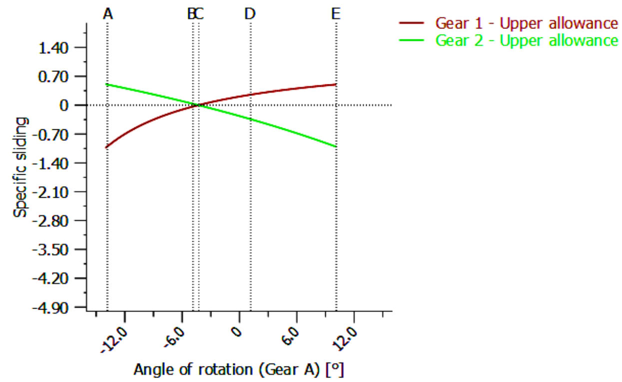


Figure 2 Balance Specific sliding

Figure-2 shows the graph showing the specific sliding values for the gear pair. Gear 1 refers to the pinion and Gear 2 refers to the Gear. In the Figure 2, upper allowance s are shown with minimum value as -1.041629 and maximum value as 0.502704 for Pinion whereas the same values for gear are -1.010876 and 0.510195 respectively. Lower allowance value for both Pinion and gear are very close to the upper allowance as -0.929121 minimum and 0.497717 maximum for Pinion whereas -0.990910 minimum and 0.481629 maximum for Gear. The value with limits looks very closer for both Pinion and Gear and con be considered for balance specific sliding condition. This value is achieved by statistical analysis following for various profile shifts, profile modification, and crowning effect applied to gear and pinion both[17].

Software used in this Research.

Kissys 2020 is used, for gear and shaft calculation, Finsap is used for the gear casing analysis.

Gear Pair Analysis

Gear and Pinion are re calculated as existing in the KISSYS 2020 without any change in any parameter to understand the its behaviour for various parameters. After doing this analysis gear pair are subjected to geometric modification based on the detail's statistical analysis so that the detrimental effect of actual gear pair can be reduced.

Specific sliding of gear and Pinion

Figure 3 and Figure 4 shows the specific sliding of gear and Pinion before and after modification of gear tooth. Because of Balance specific sliding and tooth modification the same gear pair is having the difference in their sliding value zone. Post modification the sliding speed value has improved for both Pinion and Gear.

Gear Pair Mesh

Figure 5 and Figure 6 shows the mesh pattern of gear pair pre modification and post modification.

There is marginal or no difference in between the speed and Torque carrying perspective of the gear pair in pre and post modification status. The green dotted line shows the line of contact and there is a difference on account of tooth modification in Figure 6 whereas the contact ratio is still above 1.4

Transmission error

Figure 7 and Figure 8 shows the effect of transmission error of the gear pair pre and post modification.

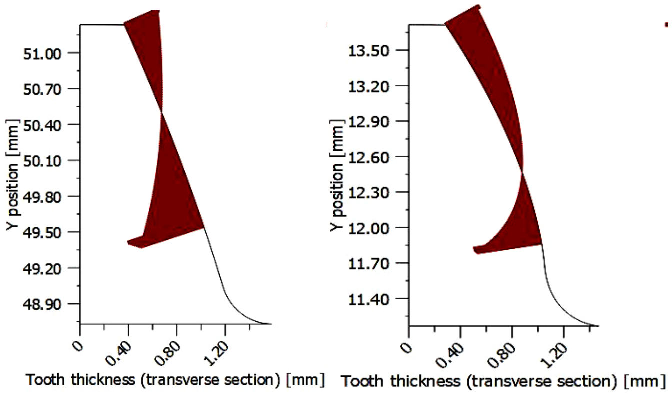


Figure 3 Specific sliding of Gear and Pinion Premodification

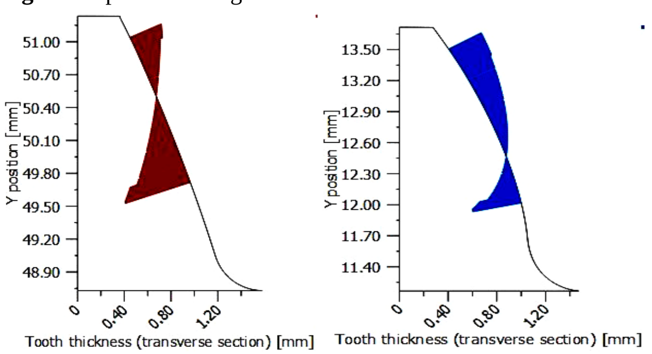


Figure 4 Specific sliding of Gear and Pinion Post modification

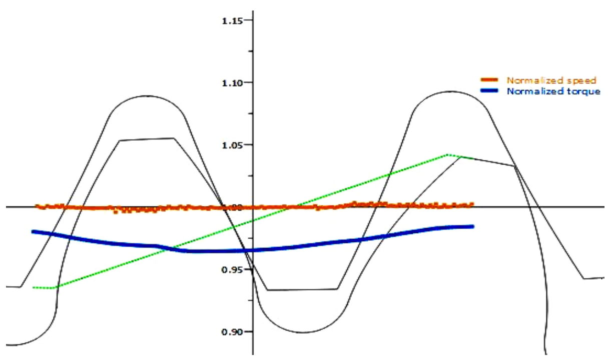


Figure 5 Mesh Pattern Pre modification

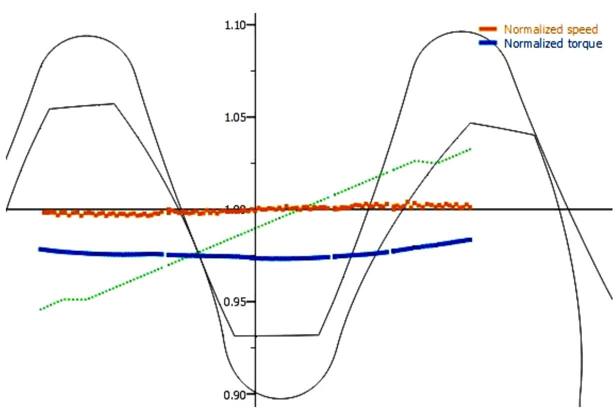


Figure 6 Mesh Pattern Post modification

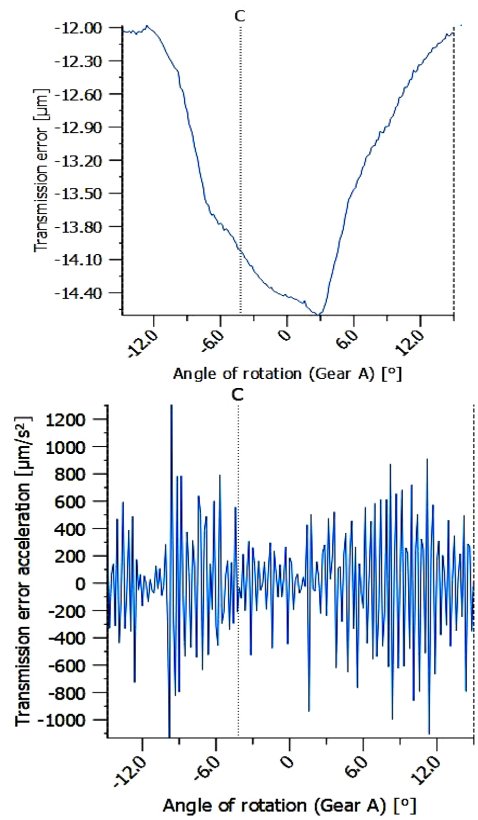


Figure 7 Transmission Error Pre modification

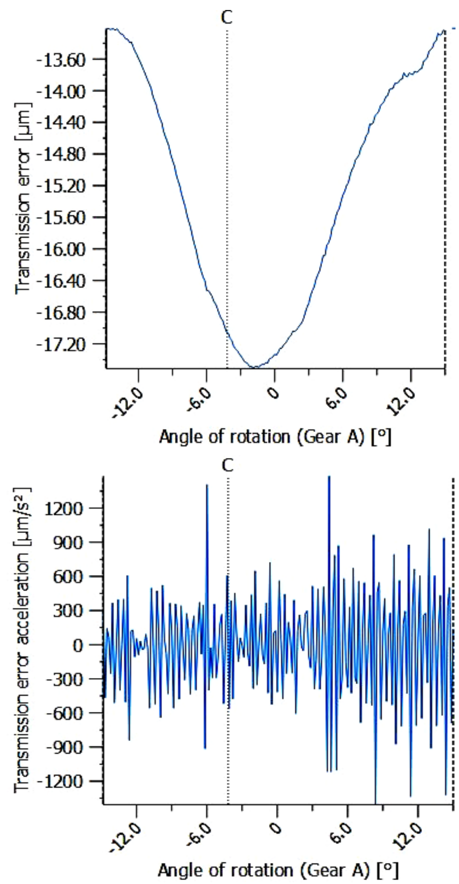


Figure 8 Transmission Error Post modification

There is a slight improvement in the transmission error over the rotation of pinion is observed Transmission error looks to be shifted towards the pitch point and trying to maintain the equilibrium rather than being concentrated in one zone. However, it is not completely resolved. Post modification there is slight uprise in magnitude at the end and start of contact of pinion, but it is marginal and has no counter effect. Acceleration graph shows clear shift of error in the addendum zone as compared to dedendum zone of pinion. The maximum magnitude of acceleration is however showing same as peak point in both pre and post modification.

Single Contact Stiffness value

Figure 9 and Figure 10 shows the difference in the single contact stiffness value of gear pair pre and post modification.

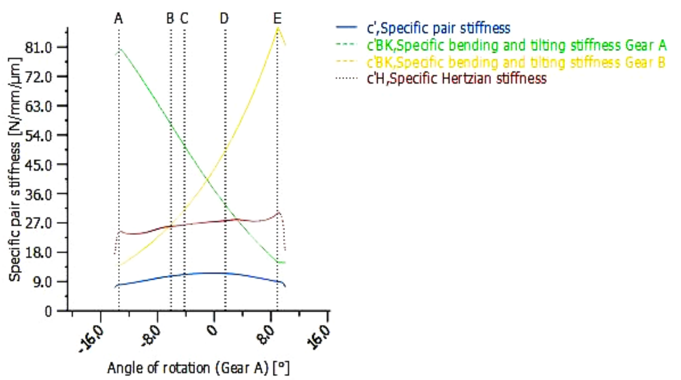


Figure 9 Single contact stiffness value pre modification

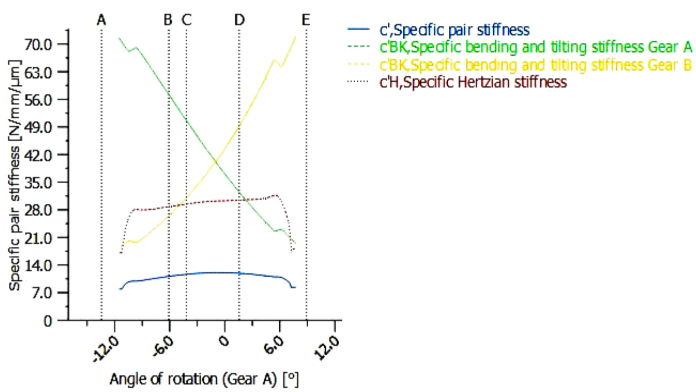


Figure 10 Single contact stiffness value post modification

From the above shown Figures specific pair stiffness has improved in post modification stage as compared to pre modification stage. Tilting stiffness value has gone down as the maximum load is applicable at the centre of tooth in modified version as compared to unmodified version. Other parameters are also showing substantial improvement.

Torque and Speed curve

Figure 11 and Figure 12 shows the Torque and speed curve of the gear pair in pre modified condition and post modified condition.

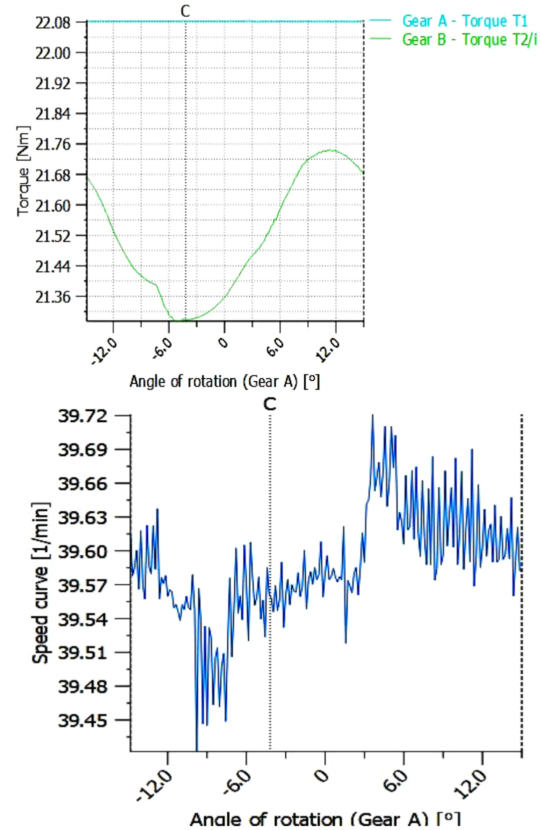


Figure 11 Torque and Speed curve Pre modification

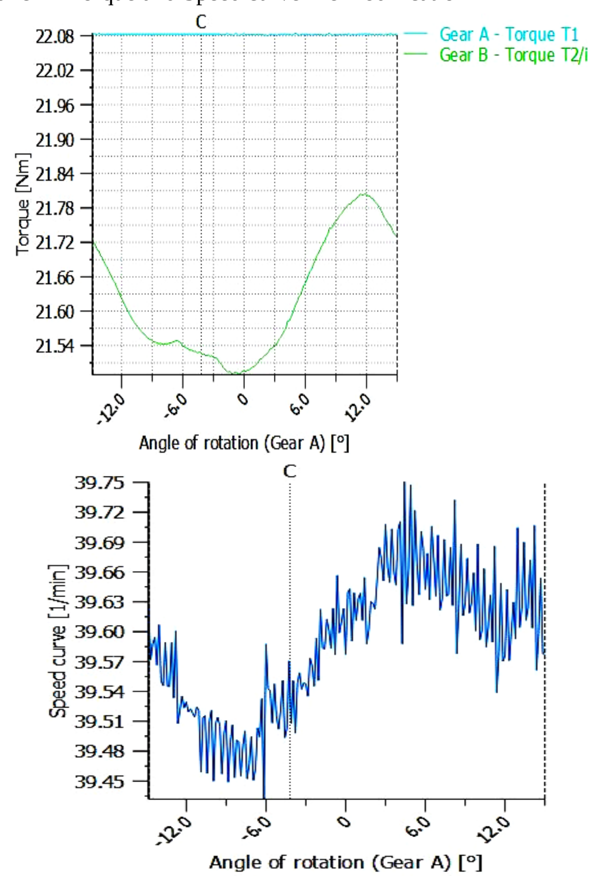


Figure 12 Torque and Speed Curve Post modification

Torque Curve in Pre modified pinion is much steeper than what it is shown in post modified condition. Sudden decrease in or increase in speed brings sliding speed also in picture. This nature of speed change cannot be eliminated however can be improved as it is totally dependent on the geometry of revolution and varying radius of curvature throughout the flank. This behaviour can be easily viewed in the Speed curve where the curve is not smooth line but a curvy line with many peaks which shoes huge variation of speed over the angle of rotation. Both Torque and speed curve has improved in the dedendum zone. The addendum zone is more or less similar in both pre and post modified condition. Both Torque and speed peaks have increased in the post modified condition of gear pair.

Contact Temperature

Figure 13 and Figure 14 shows the contact temperature distribution in gears while meshing.

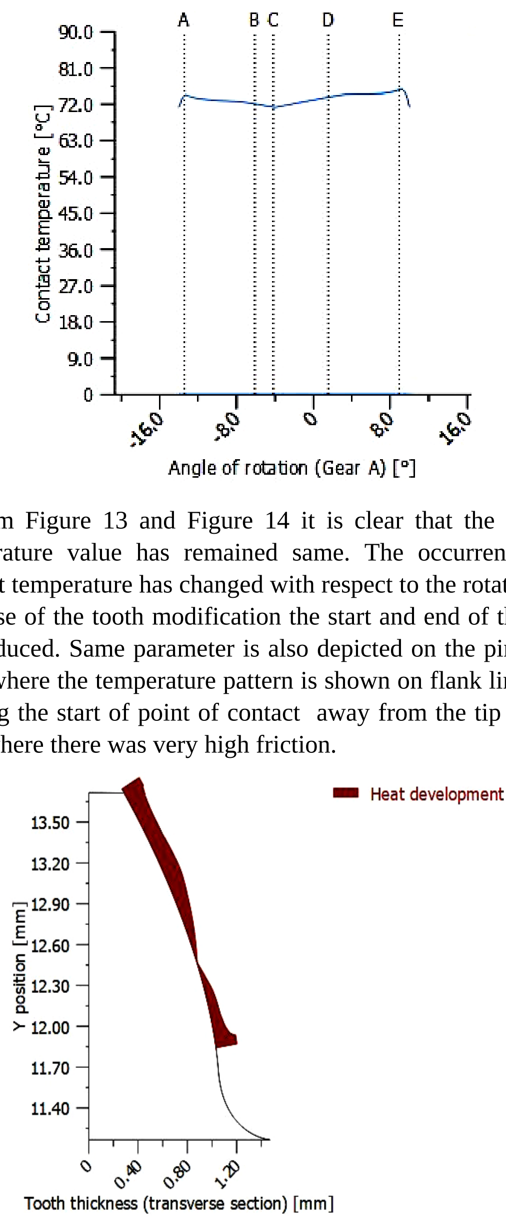


Figure 13 Contact Temperature Pre modification

From Figure 13 and Figure 14 it is clear that the maximum temperature value has remained same. The occurrence of the contact temperature has changed with respect to the rotation angle. Because of the tooth modification the start and end of the contact has reduced. Same parameter is also depicted on the pinion tooth view where the temperature pattern is shown on flank line. This is moving the start of point of contact away from the tip of mating gear where there was very high friction.

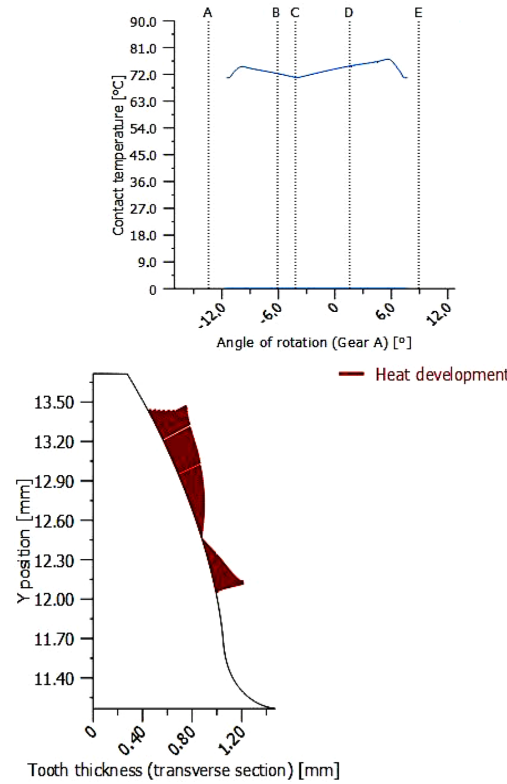


Figure 14 Contact Temperature Post modification

Specific Film Thickness

Figure 15 shows the specific lubricant film thickness variation over the entire contact line. It is the property of lube film thickness and surface finish of material. The pattern of this property is same for both pre and post modification.

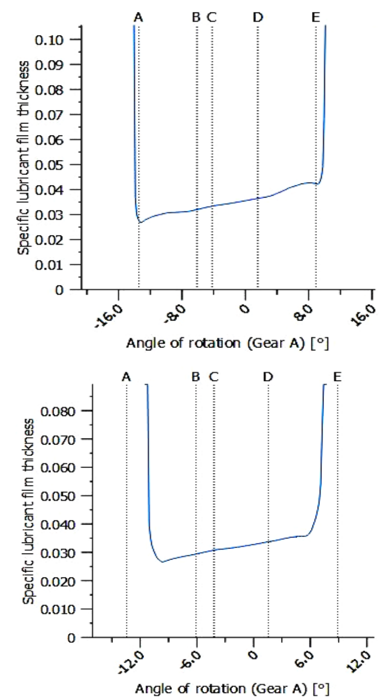


Figure 15 Specific Film Thickness Pre and Post modification

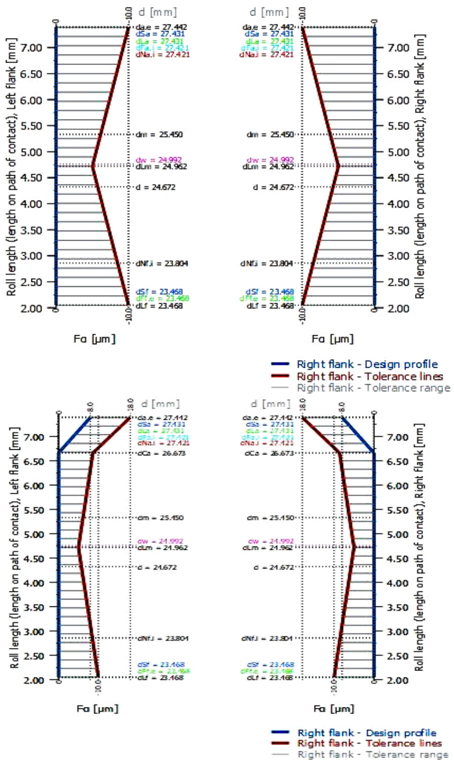


Figure 16 Pinion Profile diagram Pre and Post modification

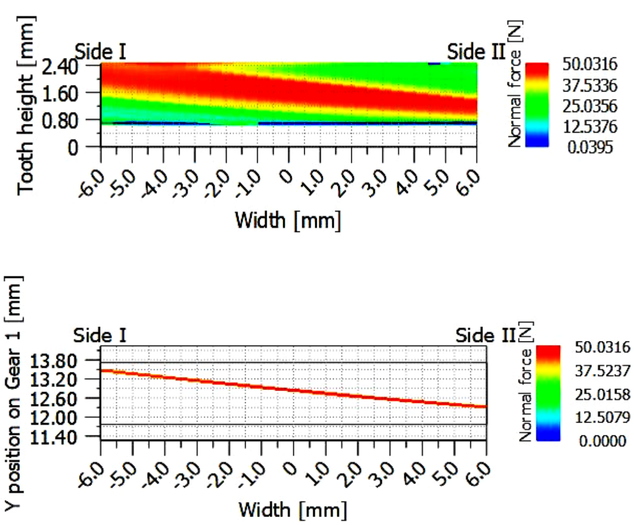
Profile Diagram

Figure 16 shows the profile diagram with both modified and unmodified status of Pinion.

Pre and post modification of profile diagram clearly shows the difference between the teeth profile and the modification offered in the post modification status. The manufactured ands used profile shall be in between the blue and red curve.

Contact line and Contact pattern

Figure 17 and Figure 18 shows the contact line difference in Pre and post modification version.



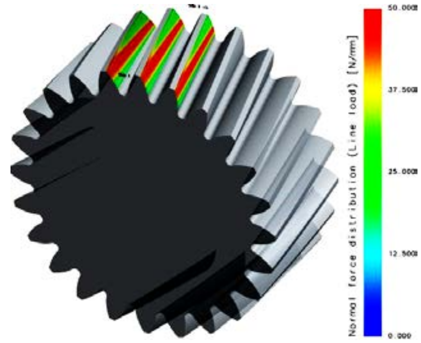


Figure 17 Pinion Contact line diagram Pre modification

As per Figure 17 the loading pattern is starting from Side II and ending to Side I of the face width. The ends of face width are deflecting because of the cantilever load. Any lateral deflection in the tooth is deteriorating the contact pattern of gear set. This will damage the gear surface. To take care of this tooth surface is modified with crowning so that the load gets concentrated at centre as shown in post modification.

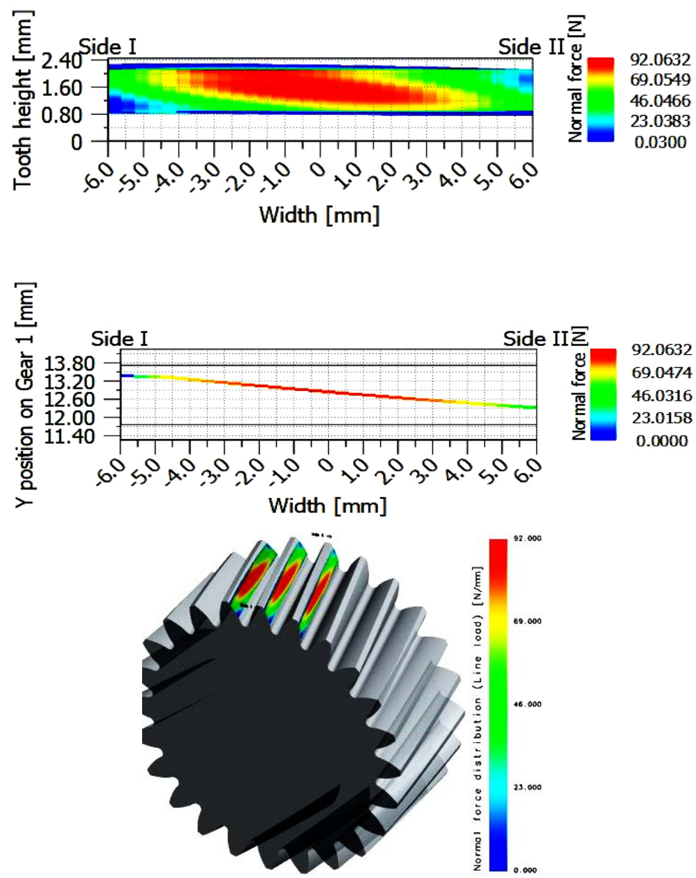


Figure 18 Pinion contact line diagram Post modification

Crowning of teeth at centre provides the increase in the stiffness at centre of the teeth also. Parallelly since the teeth profile is curved so the contact area is also reduced causing the high load concentration at centre of face width as shown in the Figure -18. Because of low contact area which is taking the complete load the magnitude of load has also increased as compared to pre modified version. But this is not affecting the tooth deflection as at centre of tooth the stiffness is very high which can easily take care of this load. Thus, the tooth contact pattern remains unaffected even after high load as shown in post modified version.

Contact line is also showing the same pattern which contact pattern is shown. Contact line is the line of contact about which all the forces are getting transmitted and contact pattern represents the contact over wide face width along the contact line.

Tooth Trace diagram

Tooth trace diagram shows the tooth modification along the face width. Figure 19 and Figure 20 clearly shows the tooth modification of pre and post modification.

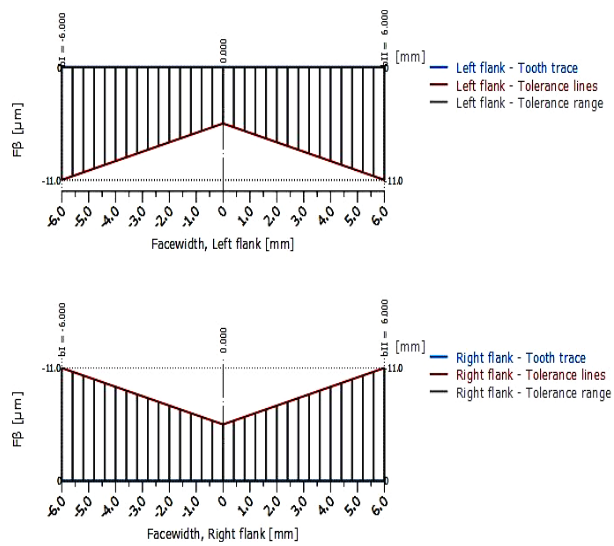


Figure 19 Contact Pattern pre modification

The above figure shows that there is no modification in the tooth form and the blue line along with red line shows the boundary limit. The grey line shows the tolerance range. This is purely defined as per the accuracy grade of the gears as defined. Manufacturing must be in between the red and blue line within tolerance zone.

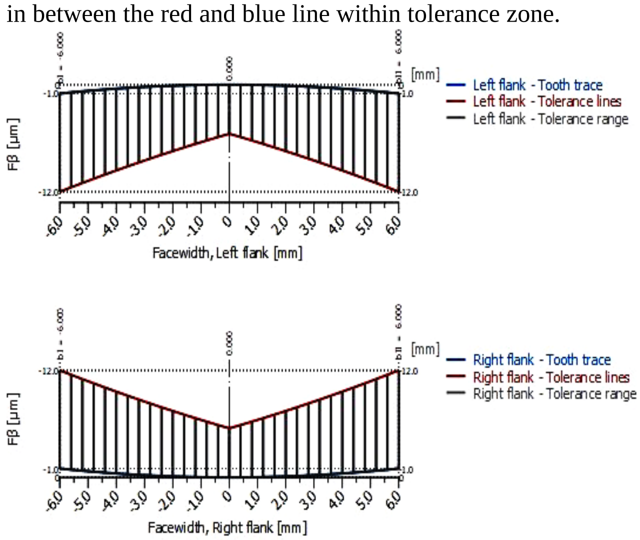


Figure 20 Contact Pattern post modification

The above figure shows that there is modification in the tooth form and the blue line along with red line shows the boundary limit. This is usually done for the mating flank but here it is done for both the flank. Also known as crowning effect help in increasing the stiffness of gear at mid of face width.

Hertzian Pressure Distribution pattern

Figure 21 shows the Hertzian tress distribution pre and post modification on the gear pair contact line.

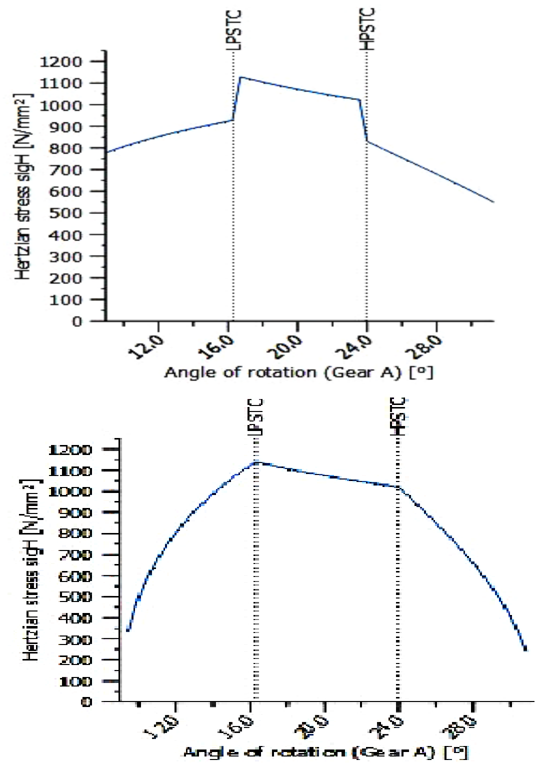


Figure 21 Hertzian stress distribution pre and post modification

As shown in the above figure the stress distribution over the contact line has changed to smooth curve from sharp peaks as shown in pre modified version. This help in gradual stress distribution over the contact zone helps in avoiding high stress concentration zone.

EXPERIMENTAL SETUP

Experimental setup consists of a powder dynamometer as a source of applying the required torque at the gearbox output shaft. Powder Dynamometer was equipped with torque meter to measure and ensure the torque to which the gearbox is subjected to. Gearbox was connected to the dynamometer through its low-speed shaft connected via torque meter. Gearbox input shaft relates to the input power source to provide the rpm and input torque as per need. Vibration sensor, oil sensor, current sensors were mounted to gearbox and power source.

Load cycle consideration

Micropitting is Fatigue related phenomena. It is necessary to take care of the loading parameters such that in a short period of testing time similar loading environment can be created as applicable to Fatigue load testing. Gear pair is designed at .5 Hp for 20000hours. A reverse calculation is done with varying load cycle to ensure that the gear is subjected to similar environment to which it is subjected during its actual working life.

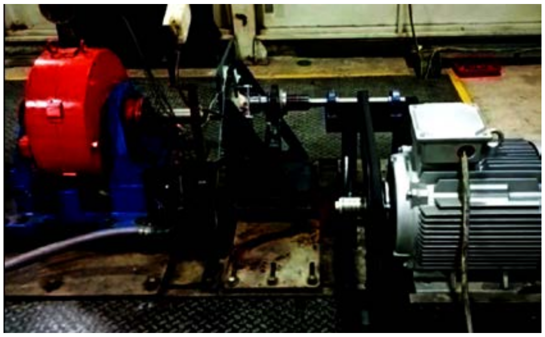


Figure 22 Testing Setup to measure and ensure the control environment condition.

Since the application to which this gearbox is used have constant speed so speed of input and output shall remain same throughout the testing period. Power source to this gearbox is of 1 HP and is very much eligible to deliver high torque as per the consumed torque. The Torque value can be varied at the output shaft with the help of Powder dynamometer and the same can be recorded as well. To ensure the control environment the temperature parameter of oils is also monitored throughout the testing cycle.

Table 3.-Load Cycle data

Step	Time(hours)	Power (Kw)	Input Rpm	Output rpm	Torque (Nm) @ Input	Torque (Nm) @ Output	Load cycle-Pinion	Load cycle-Gear
1	51	0.75	160	40	45	180	489600	122400
2	336	0.5	160	40	30	120	3225600	806400

Based on the detailed calculation above data was showing the similar effect on gear pair which in actual the gear pair would be subjected in its whole actual life. Reason for two rating was to ensure the high-power requirement while mixing different density and viscosity fluid in actual working life.

Actual Manufacturing of components

Gear housing

Gear housing is manufactured with Aluminium, AL-6061 T6 material. The T6 refers to the temper or degree of hardness, which is achieved by precipitation hardening. Machining process is carried over the vertical machining centre machine BFW BMV 50 TC24.

Gear Pair manufacturing and assembly

Gear and Pinion was manufactured with 20MnCr5 material. Conventional hobing machine is used to cut the gear and later the grinding machine was used to provide the ground surface as per need after case hardening. Images while manufacturing is shown below in Figure 16. Bearings used in the assembly was 6006-2z and 61904-2z which are life long lubricated bearings and don't need external lubricants to work.

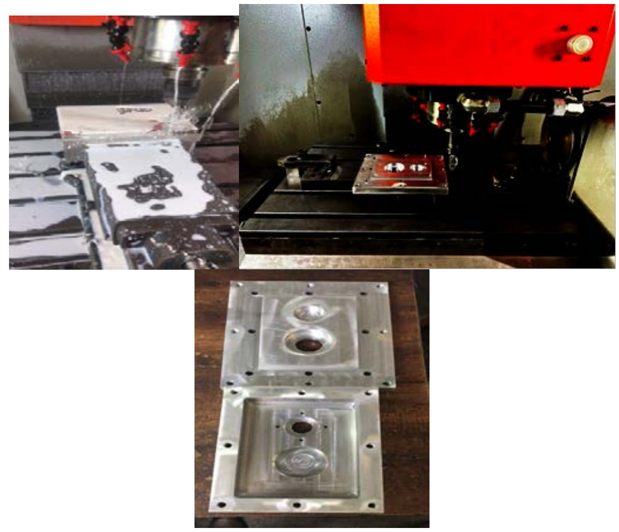


Figure 23 Gear casing

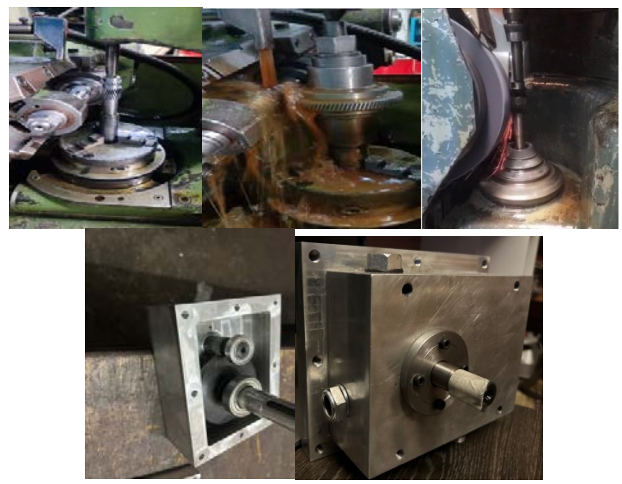


Figure 24 Gear pair manufacturing and assembly

Results and Finding

a. *Vibration reading and finding.*

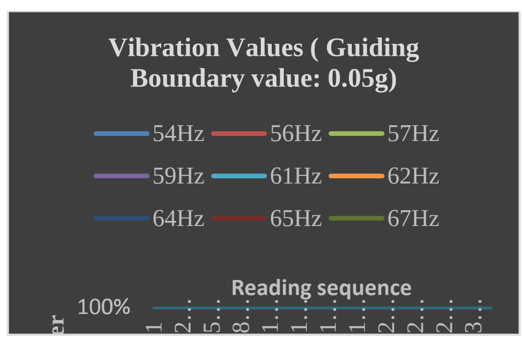


Figure 25 Accelerometer reading pre and Post modification.

Accelerometer reading is plotted above for the worst values recorded over a period of testing. As against the limit value a sufficient margin was observed between the limit and actual value. It was observed that the assembly couldn't get damaged to pass the limit of vibration. Some peaks were observed which was because of external effects. The nature of curve for both pre and post modification was found similar.

Gear flank Surface Visualization

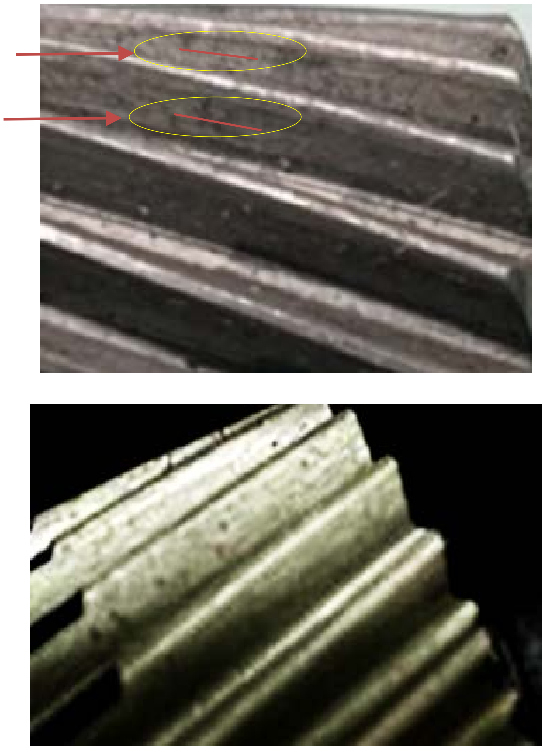


Figure 26 Gear Surface snap pre and post modification

Experimentation was run for two sets one with gears without modification and other with modifications. After the testing the gears were cleaned, and their snaps were taken. The highlighted zone in the Figure shows the signs of micro pitting. The figure shown on the right-hand side shows no such changes on the flank surface for the same duty cycles. The micropitting was observed in Pinion only and that also in dedendum zone.

Pinion Profile variation.

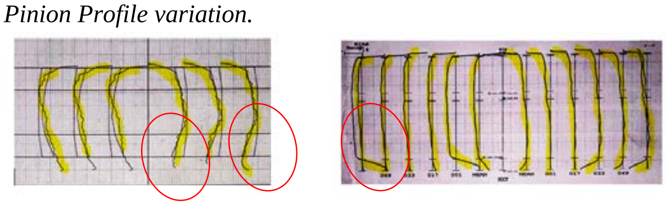


Figure 27 Profile diagram of Pinion pre and post modification after experimentation

Gear profile measurement was done post experimentation for both the pinion which get micropitted and one which was not having any sign of micropitting. As shown in the picture the profile diagram of mating flank was totally damaged as compared to non-mating flank of same gear. On the contrary the modified pinion was still able to maintain the good tooth profile as required. This shows that the profile was getting deteriorated if not properly defined and modified and finally causing the micropiting.

Microstructure comparison

Spare piece of unmodified pinion and micropitted piece was sent to lab for microstructural analysis. The magnification used here was 1000x. From the available Figure no clear distinction was observed and hence it became difficult to identify if microstructure changes on account of micropitting.

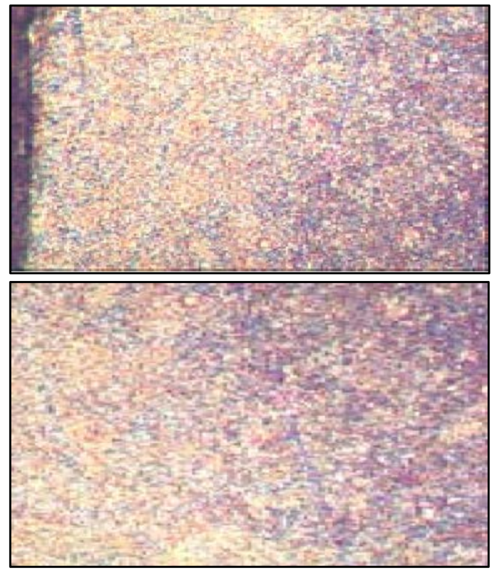


Figure 28 Microstructure comparison of micropitted pinion and new pinion.

Oil and Temperature Data

Oil and temperature data of lubricating oil was captured for 320 times the data were plotted as below in Figure 29. TDN no kept on decreasing over a period whereas there was change in temperature also. The pattern remained same for both the experiments. TDN no kept on decreasing over days and maintained the constant value of 900 which can be considered as good value yet on lower side. Similarly, the oil temperature also increased to the maximum limit of 80 degree Celsius as against the allowable limit of oil temperature as 120 degree Celsius.

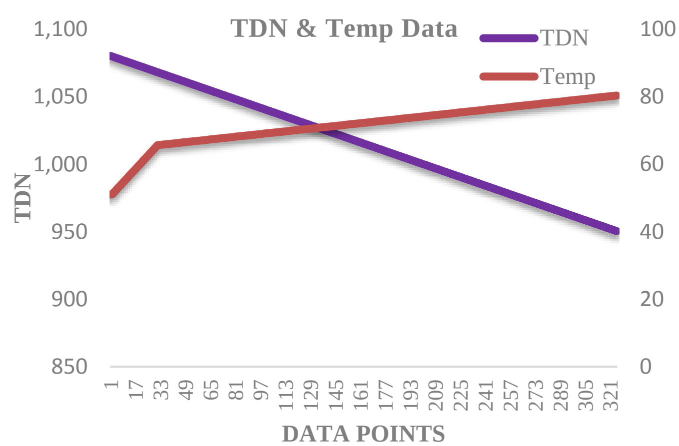


Figure 29 Oil temperature and TDN no details

FEA approach

Micropitted surface was analyzed for surface deterioration and the same was modeled in KISSYS for surface deterioration to find out the effect on its performance. Data from the actual deviation s in profile were fed to the software to generate the tooth surface. Following points were further observed.

Meshing pattern

After modeling the pinion and gear the meshing pattern was investigated. The meshing pattern is shown as below in Figure 30.

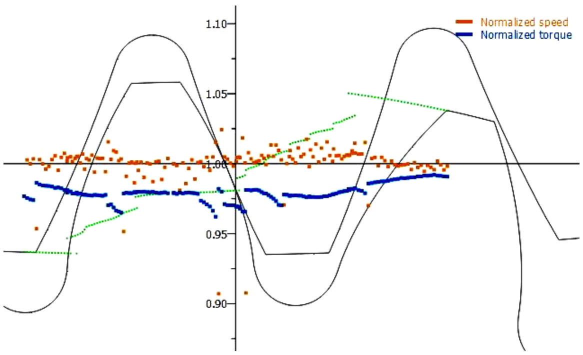


Figure 30 Mesh Pattern Deteriorated pair.

Because of the deterioration of the tooth profile the meshing pattern was found completely deteriorated. Green dotted line shows the line of contact which also look like to be completely disturbed. Speed values looks to be less distorted as compared to Torque cycle. Availability of local stress point on gear flank also validate this point.

Transmission Error

Figure 31 and Figure 32 shows a comparison of Transmission error with the modified gear having no defects with the defective gear.

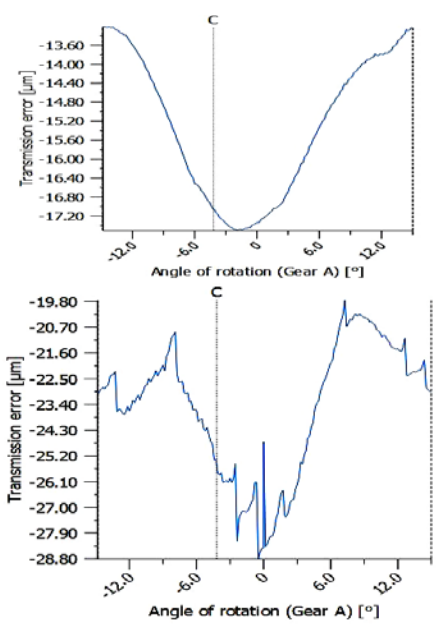


Figure 31 Transmission error as expected vs actual after surface deterioration

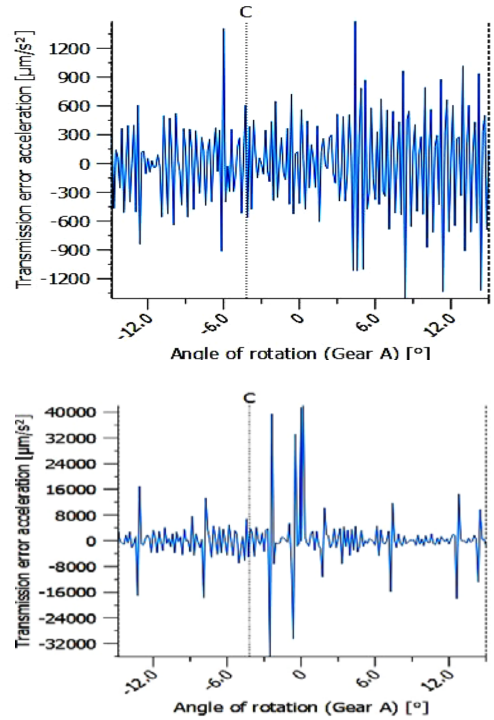


Figure 32 Transmission error acceleration as expected vs actual after surface deterioration

Left Figure in the above Figure-31 and Figure 32, shows the condition with a good condition new pinion. The transmission error curve looks to be smooth and follow a mirror pattern like. For the transmission error acceleration curve there are some spikes and mostly coming due to manufacturing deviations. Same values are shown on right figure of Figure 31 and Figure 32 and they are with big differences as compared to the un defected pinion. The transmission error range has increased for the micropitted gear model as compared to new manufactured pinion model. The values of Transmission error acceleration shows a huge spike in the values and the same effect can be seen on gear surface with some marks after completing the testing.

Lubricant film thickness comparison

Lubricant film thickness was calculated for both actual manufactured gear model and micropitted gear model. Figure 33 shows the lubricant film thickness for actual manufactured gear model(left side) and micropitted gear model(right side).

The minimum possible lube film thickness for the manufactured gear without any micropitting on surface was 0.015 micrometer as compared to the micropitted gear model where the thickness of oil film near the pitch line has become almost negligible and thus raising the concerned point for micropitting.

Because of deteriorated tooth profile the micropitting model is showing no oil film thickness near the pitch line in dedendum area. This is purely defined based on the tooth deviation values observed from the measurement of micropitted gear. Marks of micropitting was visible near the pitch line in dedendum zone after actual experimentation for pinion.

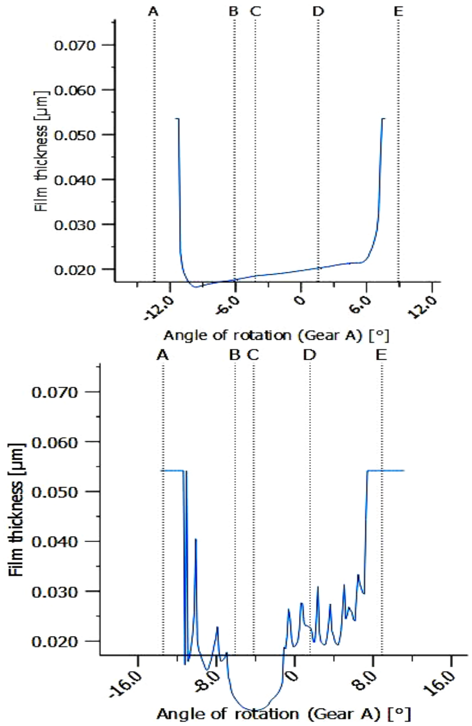


Figure 33 Lube Film Thickness for undefected gear and micropitted gear model

Contact Pattern change

Figure 34 shows the comparison of contact pattern of a healthy pinion used in this experiment with the surface deteriorated pinion.

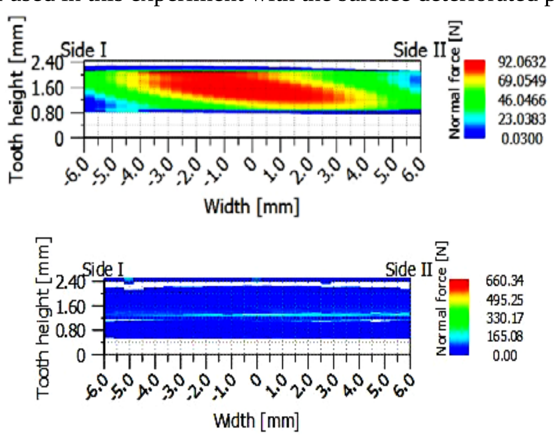


Figure 34 Contact pattern for new modified pinion(left) and wear pinion model (right)

Contact pattern shown on the left-hand side is for new modified pinion which even after the complete experimentation still maintain the same contact pattern where the pinion model on right hand side shows the contact pattern has completely deteriorated. Based on the transmission error accuracies error it was clear that the amplitude has increased drastically and the same is visible in terms of the forces as shown in the above Figure 34 with red circle. There are certain spot of 500N force which is causing to damage the pinion flank. This was clearly visible in the actual product also. Results observed indicate that Friction is an important parameter for micropitting in the concerned area. Lubricant film thickness is lower in the zone where the temperature is high.

Conclusion and Future scope

The experimental investigation aligned with the analytical findings. Micropitting was observed in the lower dedendum zone of the pinion. There was no change in the microstructure before and after the micropitting, indicating no correlation with micropitting. The vibration signals related to micropitting were insignificant. However, if the experiments had continued to a more severe level of deterioration, the vibration signals might have been useful for analysis.A good Future scope could be to perform experimental analysis with variable speed and variable load condition and to compare the finding of the result of this research work..

Conflict of Interest Statement

Authors declare that there is not conflict of interest for publication of this work. No financial aid received for this work.

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