ISSN 2395-1621

Optimization of gear micro geometry for reducing gear whine

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ABSTRACT

In-cab noise reduction is important for passenger comfort and long life of any automotive design. The Drive train contributes more towards vehicle noise and vibration due to excitation provided during meshing of gear pair. Rolling contact noise of gear pair under load is called as "Gear whine". Transmission error and mesh stiffness variations are major sources of gear whine. If constraints are put so as to minimize Gear whine at the source, only option left is to change gear geometry. Gear micro geometry allows changes in geometry at micro level that causes significance reduction in gear whine without affecting other parameters. Magnitude of transmission error indicates severity of Gear whine. In this paper, for a given gearbox, analysis is carried out at defined duty cycle, contact stresses, mode shapes and transmission error is estimated. From the nature and the magnitude of the transmission error, set of micro geometry parameters like lead crowning, lead slope, involute barreling, involute slope etc. are modified. Variation of contact stresses and transmission error with change in given set of parameters are studied carefully. Set of parameters are selected after number of iterations in such a way that it reduces the gear whine. After comparing these iterations, a optimized micro geometry that reduces gear whine and satisfies functional requirements is selected.

Keywords: - Gear whine, Micro geometry, Transmission Error

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Gears are the key elements of power train as they transfer torque and speed in given quality and quantity as per requirement of driver. After World War II there is dynamic improvement in logistics that leads to inventible development in gears. Different types of gears are used according to application requirements e.g.for low speed and small distance power transfer between two parallel shaft spur gears are used. They are more efficient than helical gears of same size. Helical gears are same like spur gear but their teeth are inclined at an angle called "Helix Angle" longer teeth than spur gear increases load carrying capacity and simultaneously decreases efficiency; for smooth and quiet transmission they are good but generate high axial thrust. They can be used for parallel as well as perpendicular shafts and if application involves transfer of motion between two intersection shafts.At high speed they become noisy and cannot be applicable for parallel shafts but can be used for intersecting shaft angleup to 180^0. Hypoid gears are combination of rolling motion and high tooth contact pressure; due to this they need special lubrication requirements in the form of oiliness and anti-weld additives but they can be used for high gear ratio and large diameter shafts. They are different than all other types of gears and resemble with screw; called as worm gears.They allow non-intersecting skew shaft to engage thus, very high gear ratio can be obtained. Feature of anti-locking is available which is safe for operator, at the same time they have low efficiency, high friction losses, run at very high pressure which requires heavy lubrication. Each gear type has its own advantage and disadvantage and varies among the applications where they are used according to constraint of cost, performance and life. Almost all automobile vehicles from two wheelers to heavy vehicle usehelical gears in their transmission. Gear drivesare mechanical drives that transmit power by engagement. In engagement there is unavoidable contact, contact raises noise and vibration. Ideally, helical gears have gradual contact and no impact while transferring power and torque, but practically this do not happen because of shaft

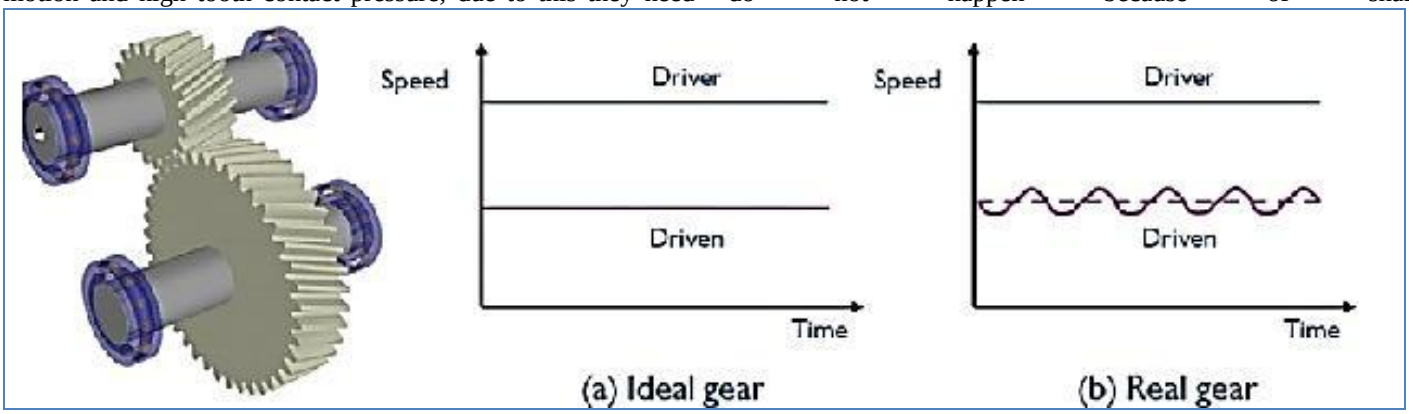

Fig.1 Speed vs. Time graph of (a) Ideal gear (b) Real gear

. Backlash; manufacturing errors and dynamics of gearbox also contribute for noise and vibration. Due to all these factors there is generation of tonal noise called "gear whine".Whengears are loaded and unloaded, sinusoidal vibrationsthat are generated are called as "gear rattle". Transmission Error is route cause of gear whine. Transmission Error is deviation from conjugate profile. If it is assumed that both driver and driving gear are perfectly rotating with conjugate motion and are perfect involute, their speed time graph is as shown in Fig.1.Driven gear rotated with constant speed is determined by gear ratio. On the other hand real gear do not exhibit similar behavior. They are not perfect, have manufacturing errors and their teeth deflect under load. Bearings and shafts on which they are mounted deflect under load and misalign; due to this real gear drive behavior is different from ideal one as shown in Fig.1.Transmission error not only induces gear whine but it deviates line of power transmission from center of gear to either end along face width. This causes load carried by gear localized to small portion of face width and large portion of face width unused, this lead to premature tooth failure, stress concentration on small portion, decreased durability and increased tooth root stresses and decrease in overall gear life. Fundamental principal to reduce noise and vibration is to reduce it at its source. Here Source is TE in gear transmission; reduce TE Gear whine proportionally. TE is generated due to improper contact between driving and driven gear. Improvement in contact pattern is done by modifying micro geometry of gear at Lead and Involute sections. At lead this is done by following ways

- Lead Slope
- Lead Crowning
- Linear relief at top and start
- Linear relief at bottom and End

At involute this is done by following ways

- Involute Barreling
- Involute Slope
- Linear relief root start
- Linear relief tip start

Number of geometrical combinations formed for different values of above parameters, out of them which minimized stress, reduced gear whine and improved contact pattern is selected and is optimized one. In subsequent chapters it has been learned about literature review, simulation in FEA software, Study of Results obtained, Discussions of results and conclusion drawn from it.

II. LITERATURE REVIEW

Deepak and Pushpendra defined Transmission Error as "The difference between actual position of the output gear and the position it would occupy if the gear drive were perfectly conjugate". Study of only gears was not sufficient to determine amount of Transmission Error. Gear noise is generated due to different reason as mentioned earlier out of about manufacturing errors. This Static Transmission Error can be overcome by proper mountings for gear meshing, while Dynamic Transmission Error can be controlled by detailed FEA techniques. There are different modeling choices for analysis like simple dynamic factor model, tooth compliance model, torsional model and gear rotor dynamic model. Mats focused his work upon sources of Transmission Error and measurement methods implemented. Tooth mesh frequency and its harmonics causes gear whine. Phantom frequencies arises from cyclic errors of worm wheel drive in grinding machine of gear and this is imparted to gears in the form of helix undulation and sometimes on each tooth they become major source of Transmission Error. Ghost frequencies originates from dressing wheel used for dressing grinding wheel. This causes undulation of gear flank. Due to undulation of grinding wheel this have wavelength of 0.5 mm and amplitude of approximately 4 µm. Pitch error also generates transmission error at tooth contact frequency and corresponding harmonics. Use of small size encoder of typically 90mm diameter and 9000rpm are suitable for measuring transmission error. The total transmission error is not equal to summation of individual element transmission error. But knowledge of elemental gear error leads to transmission error but vice versa is not possible. Braun, Walsh, Horner and Chuter carried out pass by noise test according to modified ISO 362 standard in Europe that revels Power Train has highest ranking in noise generation. This implies torque from engine that transmits to tiers by continuous engagement of gear teeth causes vibration and noise; this is a typical source-path-receiver problem. Transfer of motion through engagement causes lots of vibration and noise like in gear meshing. Peter and Bernd established ranking of tribological and geometrical parameters that affect gear whine. They adopted approach of multi body simulation that includes oil film thickness and elastic deformation directly. Simulation of all parts including bearing, shafts and housing has been done; this helps to predict overall effect of all operational and geometrical parameters of all transmission components on gear whine. This approach enables determination of structure borne noise of transmission and its dependency on other parameters. This analysis shows that only geometrical parameters have influence on gear whine at given torque. If constraint are put such that only way to minimize gear whine is to change geometry of gear without affecting its functional parameters like load carrying capacity, efficiency and gear ratioR. Tharmakulasingam, Dr. G. Alfano, Dr. M Atherton Studied effect of tooth profile modifications on transmission error. Given model of gear pair simulated and analyzed under static condition. It has been concluded that transmission error is a function of tooth profile modification. Another cause of gear noise is mesh stiffness variation with number of gear pair in contact, it of cyclic in nature and has typical rectangular waveform. Macro and micro geometry are two approaches to which Transmission Error and mesh stiffness variation were major causes. In some analysis mesh stiffness variation is assumed to be rectangular waveform of magnitude 10^{+08}. Transmission Error for load at low speed is considered as Static Transmission Error and for load at high speed it is termed as Dynamic Transmission Error. Static unloaded Transmission Error predicts information change gear geometry for reducing Gear whine, macro have restrictions that it can be optimized at design step only. Also it involves highly accurate tolerance which results into high cost and comparatively low noise reduction. Micro geometry approach is feasible with low cost and high result in noise reduction as compare to macro approach.Palermo, Mundo, Lentini, Hadjit ,Mas, W. Desmet proposed in their research work that gear bearing forces has been measured for noise analysis of gearbox. All forces in gearbox that contributes towards noise and vibrations including shaft deflections, transmission error and tooth deflection passes through bearing to housing .Dynamic transmission error is used for analyzing effect of these forces on bearing forces also inertial contribution taken into account. In particular, the peak to peak value of transmission error can be used as an indicator of the TE variability. A constant TE, in fact, would not cause vibrations or noise.Platten, Blockley, James, Prabhakaran and Scott considered micro geometrical modification are to be implemented after design procedure in numerical model rather than traditional experimental basis model. In Traditional approach micro geometry considered as secondary treatment and its nature is of trial and error, but optimization needs much faster and accurate approach. For this they used Romax Designer software which includes all aspects of gearbox like gear rating, shaft material, bearing stiffness, housing stiffness, inertia of synchronizer and implementation of duty cycle. This enables replication of actual working environment effectively which in turns increases accuracy of results. Advantage of this software is micro geometrical modifications can easily be implemented and its effect on gear whine can be analyzed. According to Amol and Vijay it is most effective option to deal with gear micro geometry. There are different sources of noise generation out of which transmission error is major cause. Several methods are there to measure it. Different simulation approach used for controlling effect of transmission error; micro geometry optimization is best suitable option.

III. MODELLING ANDSIMULATION

Automotive driveline includes housing, bearings, shafts and main transmission element gear. Here gearbox is designed without housing so housing stiffness is neglected. Layout of gearbox under consideration is shown in Fig 2 below

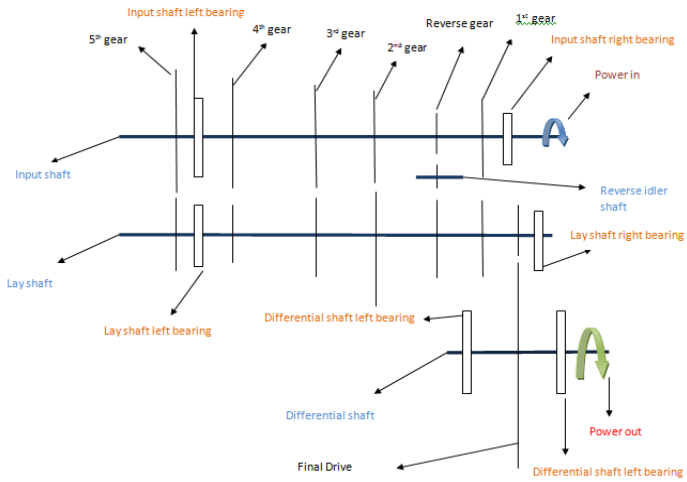

Fig. 2 Layout of gearbox under consideration

On input shaft all gear blanks are model as solid integral with shaft except the 5th gear which is modeled as simple carrier, in other way it is freely rotating. All the gear blanks on lay shaft are simple carrier type except final drive pinion blank which is modeled as integral with shaft. Separate blank has been designed for differential gear, which is also modeled as simple carrier type. Various component are modeled as per given description

A. Shafts

Shafts are modeled as per the drawings. It also includes type of heat treatment given to the shaft and material from which it has been manufactured.

B. Gear

Gear profile is modeled as per basic geometry, teeth profile related parameters, gear blank details, tip and root diameter, gear material, gear rating etc. This initial gear geometry inputs are considered as base of micro geometry. All these power transmitting components are assigned with proper finite stiffness and assumed to be flexible bodies.

C. Bearings

Bearings are mounted at given place according to drawing parameters like no of rollers, bearing dimensions, static and dynamic capacity are given as input. According to these input provided bearing stiffness is calculated.

D. Optimization

Gear geometry changes at micro level have been done on both profile and lead of gear tooth. There is a variation of contact patch pattern, transmission error and mesh misalignment error with change in micro geometrical parameters. This arise multiple geometrical combinations with different value of above mentioned parameters. Optimization here refers to selection of one combination out of all resulting combination so that there is minimum transmission error thus resulting in reduction of gear whine.

After modeling; given gearbox is ready to expose for duty cycle. Duty cycle is set of constraint that imposed on gearbox in the form of RPM, temperature, torque and time duration for which particular gear exposed to load. Contact between gear teeth assumed as nonlinear dynamic contact for analysis purpose and shaft are considered as Timoshenko beam. Transmission error is not dependent on gears only but it depends on mesh misalignment on which influence of other member is significant like bearing stiffness. It is necessary to model entire transmission. The following methodology adopted for solving this problem

- Consider all possible elastic deflection so as to replicate mesh misalignment value of practical gearbox into modeled one
- Optimization of gear micro geometry according to loads and mesh misalignment

Ideally line of power transmission pass through center of gear teeth along face width, but practically due to transmission error this is do not happen; power transmission line is shifted to either end of face. This results into reduced load carrying capacity of gear tooth and other part of face remains unutilized. Excessive load on small area causes severe damage to gear teeth and may cause breakage of gear tooth. For given duty cycle contact patch pattern report gives load distribution of gear teeth in contact; such contact patch report for 1st speed gear pair along with gear pair is shown in Fig 3 and Fig 4 respectively.

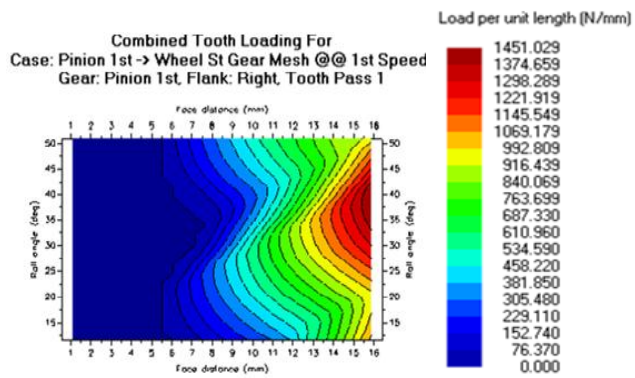

Fig. 3 Contact Patch Pattern for 1st speed Gear Pair

Contact patch report shows that gear pair is heavily loaded at one side of face shown by red color area in Fig 3 of normal load of 1451N/mm and at another side there is no load shearing, so load distribution is unequal on face along face width. This cause reduced life of gear teeth as it is exposed to severe load.

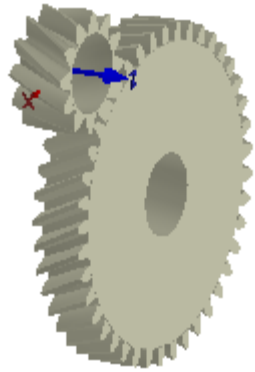

Fig. 4 1st Speed Gear Pair

As discussed earlier, transmission error classified into static and dynamic transmission error. Static transmission error was calculated by considering inner and outer race deflection of bearings, deflection of shafts and gear deflection. Static deflection is shown in Fig 5 below

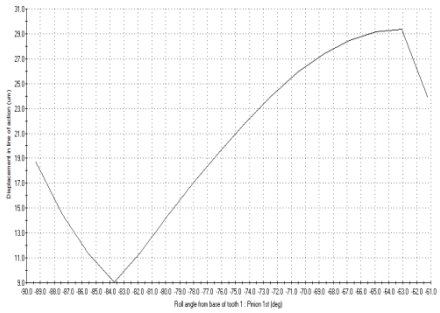

Fig 5. Static Transmission Error for 1st speed gear pair

Dynamic transmission error can be determined by adding static transmission error in displacement due to frequency of gear mesh because of force. It is given by equation below

$$TE_{dynamic} = D + TE_{static}$$

Where,

D= displacement cause due to difference resulted from constant force applied and force applied at given frequency.

Practical measurement of dynamic transmission error is very difficult hence sophisticated optical encoders used to measure it as discussed in earlier section. Dynamic transmission error is shown in Fig 6 for 1st speed gear pair

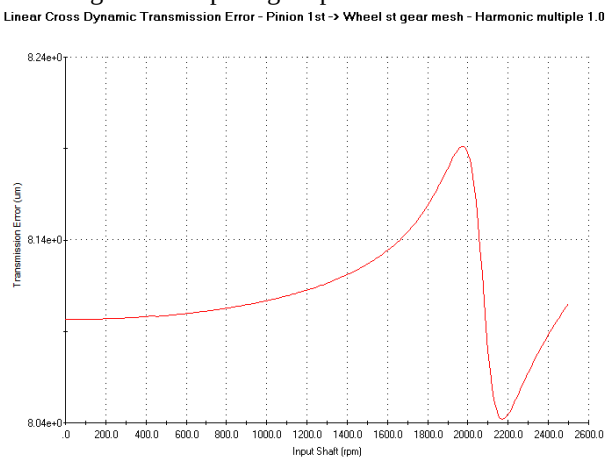

Fig. 6 Dynamic Transmission Error for 1st speed gear pair

IV. RESULTS AND DISCUSSIONS

Now objective is to minimize this transmission error, as stated earlier micro geometry modification is effective way to reduce gear whine. Number of iterations has been carried out by changing different micro geometrical parameters like lead slope, linear tip relief, lead crown along lead and involute barreling, and involute slope along involute profile. Out of this iteration which reduces gear whine to maximum extent without affecting functional parameters is selected as optimized gear geometry. Three iterations have been carried out by changing micro geometrical parameters for those variation change in mesh misalignment for different speed observed according to table I Careful study of mesh misalignment value shown in table above predicts that there is huge reduction in magnitude of mesh misalignment value from original to iteration 1, this reduction is though significant for 1st speed and final drive but not significant for other speed

TABLE I

Mesh Misalignment for different speed for three iterations

Speed	Original	Iteration 1	Iteration 2	Iteration 3
1st speed	-77.90	-57.93	-58.23	-58.33
2nd speed	10.29	9.72	9.72	9.72
3rd speed	17.02	14.20	14.20	14.20
4th speed	21.27	17.15	17.15	17.15
5th speed	-22.06	-11.78	-11.78	-11.78
Final drive	-151.67	-121.12	-121.12	-121.12

.

Mesh misalignment value variation from iteration 1 to iteration 3 is constant for all other speeds except 1st speed. For 1st speed there is no significant variation in magnitude. This variation is because here only 1st speed gear pair is exposed to micro geometry modification. This effect will help to minimize its contribution towards misalignment of other gears. Due to this there is only slight variation in magnitude of other gear pair and significant in 1st speed.

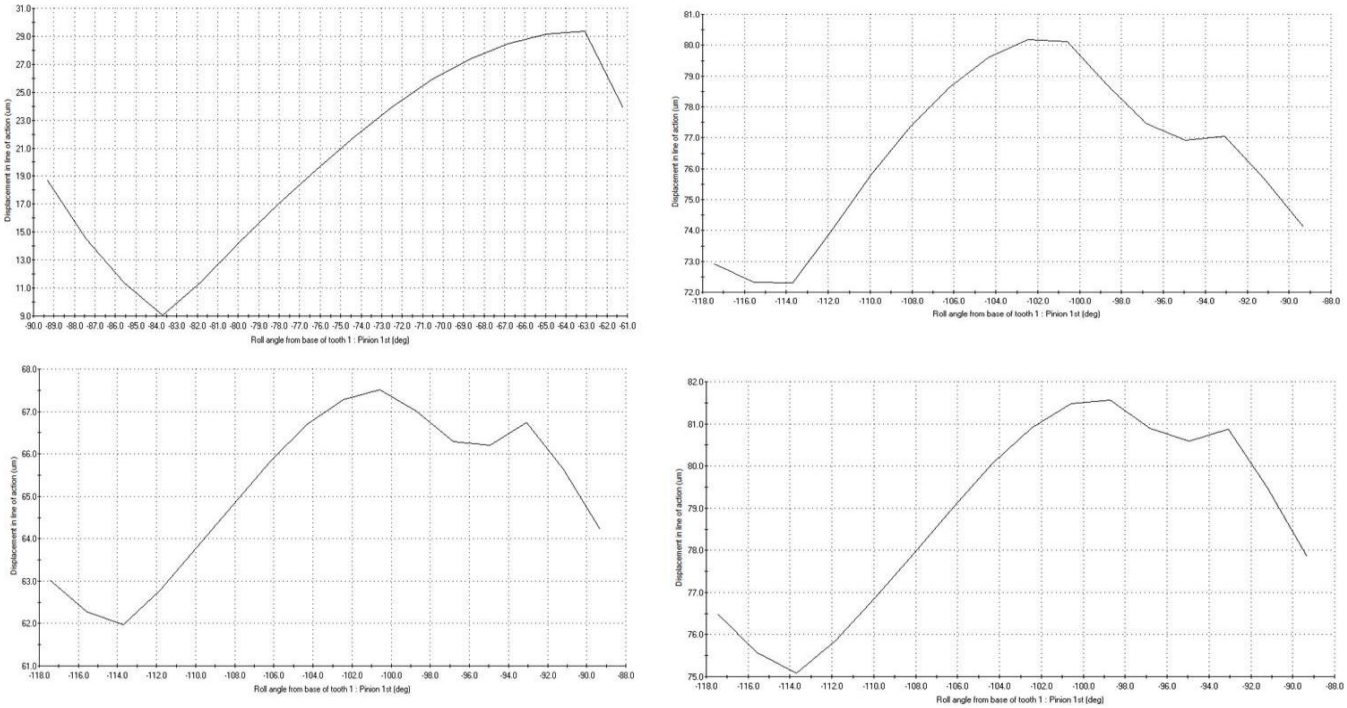

Fig. 7 Static Transmission variation for 1st speed gear pair from clockwise sense without any modification to iteration3

Static transmission error is expressed by plotting graph of displacement in the line of action against roll angle as shown in Fig 7. From left hand corner clockwise orientation shows that there is initial sharp plot but after modifications there is reduction in sharp variation. This predicts that there is reduction in static transmission error considerably. Effect of modifications on gear face studied with the help of gear contact patch pattern. Contact patch pattern is very important for predicting gear durability, as gear durability depends upon maximum utilization of gear face for loading. Equally distributed load along face N/mm for 1st iteration, 1030 N/mm for 2nd and 977 N/mm for 3rd iteration. It can also be minimized further but it include few constraints on manufacturing of given micro geometry that covers cost too. Also profile modification Now to see the effect of all these parameters on gear whine, we need to see variation causes less exposure of only one side of gear tooth that results into long life of gear teeth. This is shown in Fig 8 below; from left hand side corner in clockwise sense it shows variation of contact patch pattern for normal load by plotting it along roll angle against face distance. Careful observation shows that modification in micro geometry causes shifting of contact pattern towards center of gear face along face width, it results into reduction in normal load and increase in durability of gear pair. Micro geometry has significant effect on normal load as it was initially 1443 N/mm after modifications it was 1193 of dynamic transmission error with respect to change in gear micro geometry. Not only magnitude of dynamic transmission error but also difference between its two peak values affect gear whine. The minimum difference between two peak minimum will be whine generated. This variation is shown in Fig 9 below.

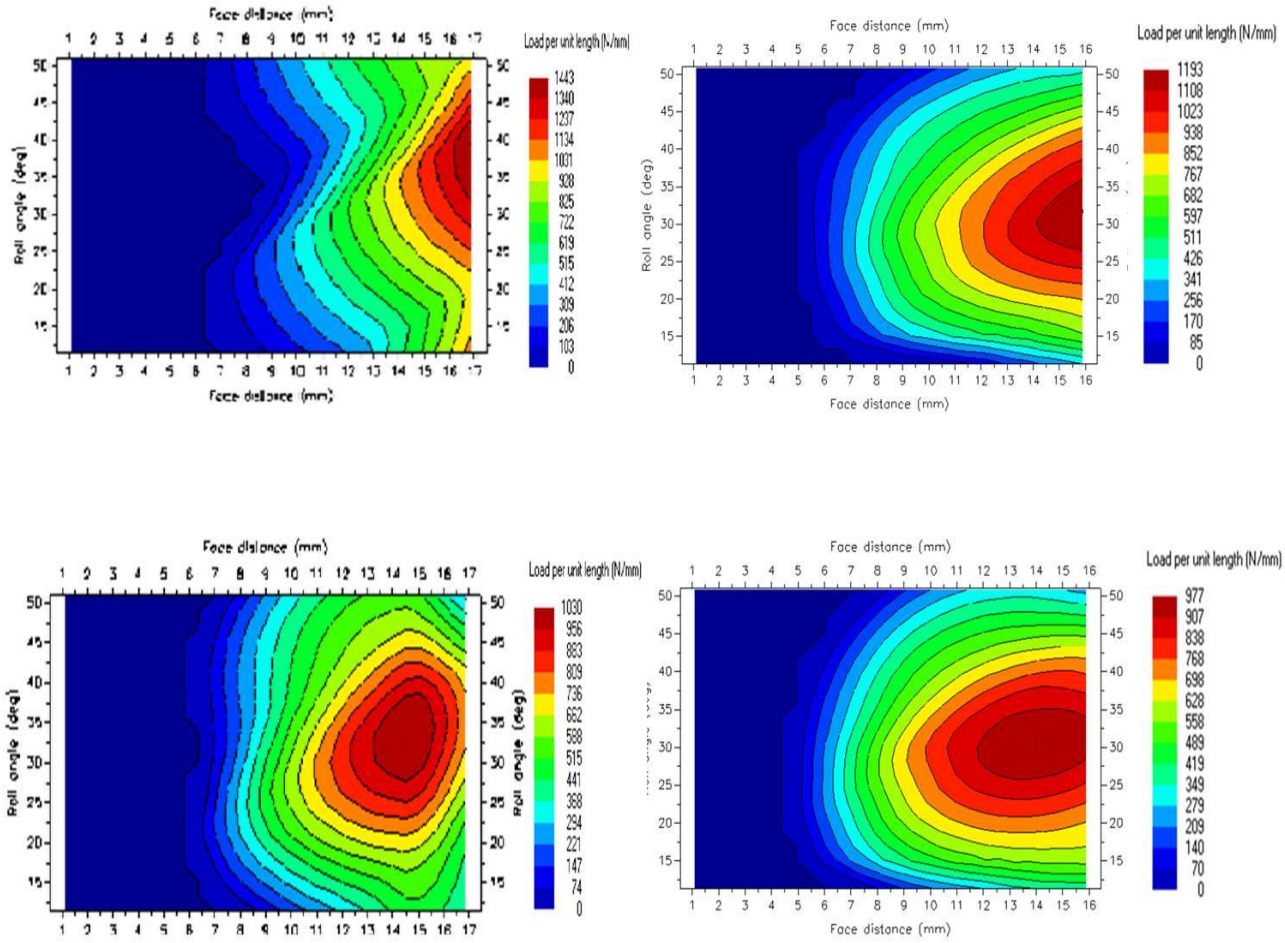

Fig. 8 Contact Patch Pattern for 1speed Gear Pair in clockwise sense original to iteration 3 variations

V. CONCLUSION

From comparative study of three iterations with original geometry for mesh misalignment, contact patch pattern, static transmission error and dynamic transmission error it can be concluded that iteration 3rd with 977 N/mm normal loads at contact and dynamic transmission error difference between two peaks as 0.06µm is optimized geometry. Though in case of iteration 2, magnitude of dynamic transmission error is less than iteration 3, but peak to peak difference is dominant in case of transmission error. This is because no matter how much is the magnitude but difference between positioning of two gears in mesh decides amount of gear whine. Thus iteration 3 is best suited from all aspect of gear design which has minimum load on gear face, maximum durability and quiet operation.

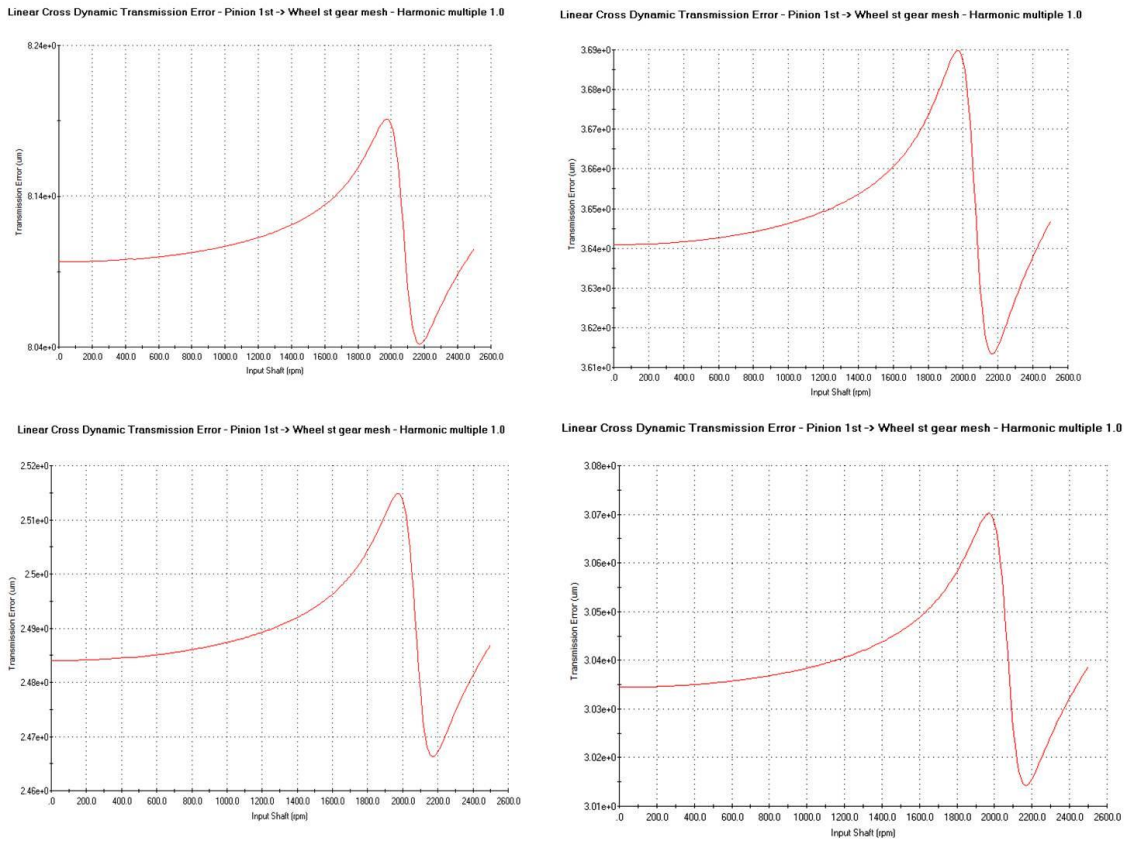

Fig. 9 Variation of dynamic transmission error with respect to shaft RPM

ACKNOWLEDGEMENT

Thanks to Dr. J.L. Minase for his valuable guidance and constant support for preparing this paper. Also thanks Mr. S. Rakesh, Senior Manager of Mahindra Truck and Bus Division, Chinchwad-Pune for providing his technical expertise and sharing his experience.

REFERENCES

[1] Deepak Malviya and Pushpendra Kumar Sharma, Transmission Error in Gear, IJMER, January 2014, Vol 4.

[2]Mats Akerblom, Gear Noise and Vibration- Literature Survey, Volvo Construction Equipment Components AB.

[3] M.E. Braun, S.J. Walsh, J.L. Horner, R. Chuter, Noise Source Characteristics in the ISO 362 Vehicle Pass-By Noise test: Literature Review, ELSEVIER, Applied Acoustics, June 2013

[4] Peter Fietkau, Bernd Bertsche, Influence of Tribological and Geometrical Parameters on Lubrication Conditions and Noise of Gear Transmission, ELSEVIER, Mechanism and Machine Theory, July 2013

[5]AmolKorde, Predicting Gear Noise due to Mesh Transmission Error, LMS users Conference, September 2007

[6] AmolKorde and Vijay Mahendrakar, Driveline Noise Reduction by Predicting Mesh Transmission Error and Micro Geometry Optimization, SAE Paper No 2009-26-052

[7] R. Tharmakulasingam, Dr. G. Alfano, Dr. M Atherton, "Reduction of Gear Pair Transmission Error with Tooth Profile Modification",Brunel University, School of Engineering and Design Uxbridge, Middlesex, UB8 3PH, United Kingdom

[8]A. Palermo, D. Mundo, A.S.Lentini, R.Hadjit , P.Mas, W. Desmet, "Gear noise evaluation through multibody TE-based simulations", University of Calabria, Department Mechanical Engineering Ponte PietroBucci, 87036, Rende, Italy

[9] M.Platten, C.Blockley, B.James, S.Prabhakaran and D.Scott, "Optimization of Gear Micro geometry for Durability and NVH", Romax Technology Ltd., Nottingham, United Kingdom

