Modifying Involute Profile of Gear to Reduce Noise and Vibration

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Abstract – Gear noise is the accepted principle cause of vibration excition in meshing gear pairs. As on root two principle source of this vibration one fluctuation in the load transmitted by gear mesh & other is transmission error in the meshing gear. In this dissertation by considering aggregate contributions of these excitation components of vibration during meshing of gears formulates by two ways in simpler in manner. Which is by using FEA analysis & Fourier's null matching technique. These imposing a constant value on the transmission error and requisite contact region on a tooth surface that provides a constant load transmitted by the gear mesh. For an example demonstrate we create steel material spur gear for proofing profile modification invention in by theoretical as well as practicle. The final gear tooth geometry of involute tooth form modified and controlled in such a way that account for deformation under load across a range of loadings. Overall this method should reduce vibration vice-versa noise.

Keywords— Fourier null matching technique & FEA analysis.

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INTRODUCTION

The inception of rotary machinery for transmitting the power gears have been used skillful in manner. After that moderation did in wooden gear involute drive trains have been integral to the development of machinery and power control technology. Most modern gearing is conjugate [1], i.e. it is designed to transmit a constant rotational velocity. The involute tooth form [2] is the most common example of conjugate gearing. Ideal involute gears transmit uniform rotational velocities without any error but such gears are practically not possible.

The displacement based exciter function known as the transmission error (T.E.) [3] it is the accepted principal source for noise in involute gearing. In most gear systems operating at speed, applied load, inertia forces and relative rotational position error of the meshing gears is directly correlated to any vibration caused by the system. Put another way, an ideal gear pair would be able to transmit a constant rotational velocity perfectly from the drive gear to the driven gear under constant load conditions. The difference in the position of the output gear when compared to its ideal analog is the transmission error. This transmission error can be directly traced to deviations from the perfect involute form due to geometric differences and deformation under load.

There are two manifest requirements imposed upon a gearing system (tooth profile) for transmission error fluctuations to be eliminated. One is the transmission error must be constant through the range of the gear rotation for any mesh loading and second one is the gear mesh must transmit a constant loading.

The transmission error as experienced in meshing gears arises because of two things one is components geometric deviation and other is deformations under load. Geometric deviations can either be intentional or unintentional. Intentional deviations arise from manufacturing modifications such as crowning or tip relief. Unintentional errors like scalloping or improper finishing are also geometric deviations. Deformation across the loaded tooth mesh also has two components. Both gross body compliance and Hertzian Compliance contributes to deformation under load. These deviations from the ideal involute gear tooth surface for loaded gears are the primary cause of transmission error. Eliminating or compensating for these deviation types can eliminate transmission error.

The purpose of this thesis is to reduce noise & vibration by modifying involute profile of gear by minimizing transmission error fluctuations during and the maintenance of constant transmitted gear mesh loading. This method is must accounted for when tooth geometry is modified to achieve the goal.

The development of precise compensatory gear geometry that when loaded accounts for all design and compliant variations in the tooth-to-tooth meshing of a rotating gear pair for deformation under load.

This design should approach the performance of a rigid, ideal involute drive train thereby transmitting a uniformly proportional rotational velocity and thereby no transmission error. The use of finite element analysis (FEA) & numerical optimization.

II. PREVIOUS RESEARCH

Gears can be traced to the earliest machines by Greeks [4].

Roman and Chinese [4] first define analytical construction in gear. The mathematical and geometrical tools invented by Renaissance engineers were simply insufficient to solve the problem of the optimal gear tooth profile [8]. Leonhard Euler was the first to successfully attack the problem by showing that uniform transfer of motion can be achieved by a conjugate and specifically, an involute profile [4], in 2003 [5] Invention did on noise of gear.

Henry Walker [7] was able to state that the gear noise will primary lead by transmission error. As gear analysis progressed slowly from basics that time author analyzed the transmission error became the accepted cause of gear noise [3].

Before rigorous analysis of involute gearing was available gear noise was addressed during modification of the available gear involute tooth surface by basic principle. This procedure came to be known as crowning [6] or relief. Tip relief made proper tooth clearance and nominal base plane action which are critical to gear noise. In modern crowning is typically combination of single lead and a single profile modification are combined and applied to the gear tooth surface. Axial and profile crowning together have been shown to reduce gear noise but that addressed some issues like addition of load computation, mesh analysis and line of contact constraints [7] after all outcome results was robust and manufacturable. [8] September/October 2002–The author modifies modern gear design is generally based on standard tools reducing tooling expenses and inventory. [9] Bending Stress Minimization (September/October 2003): Author works on Lewis equation and find out the bending stresses developed in the gear. [10] In 2013 author works on high speed non lubricated gear: These type of gear design technique introduce and applied to reduce tooth contact temperature and noise excitation of a high speed spur gear pair running without lubricant.

III. PROPOSED METHOD

We can find the solution by two methods they are:

A. By using finite element analysis (FEA) software.

B. By using mathematical expression of Fourier's transformer

Before implementing any technique we must have basic information of gear such as material, Number of teeth, face width, pitch radius, helix angle, pressure angle, Young's modulus & poison's ratio.

By using finite element analysis (FEA) software:

1. Make a model for FEA analysis by available input data.
2. Applying meshing on model for analysis..
3. After applying the load of conjugate teeth we will get the stress affected region it may be line contact type or point contact type.
4. Analysis results we will show maximum stress affected or constant deformation regions exact co-ordinates. Fig.4 shows the stress impact zone.

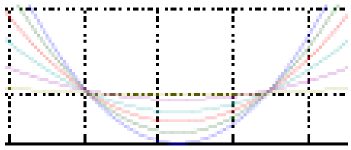

Fig. 4 Stresses on tooth profile.

5. After getting these results we need to modifying stress affected co-ordinates, by making the new model for analysis by removing material from affected stressed zone co-ordinates.
6. Rate of material removing is start from 1micron and gradually increases up to 25 micron.
7. Plot result sheet in the form of length of linearly increment portion of convolution precursor (∆) & Deflection of gear tooth line contact (s).
8. After getting these full results tabulated data now we ready to select optimum solution which will give minimization transmission error, contact loading stress point, vibration excitation and noise.
9. We got these data in form of s & ∆ means actual co-ordinates on gear. Accordingly we will practically remove the material from involute profile by grinding machine.

By Experimental modification results

Gear grinding equipment can typically reach a geometric accuracy of approximately 1 µm. With a mean step distance on the order of 0.5 mm, it is realistic to attempt to manufacture the negative s portion of contact region for the deformation ranges of 5 µm to 25 µm. Arranging the set up as per below picture on machine Holroyd GTG2 grinding machine.

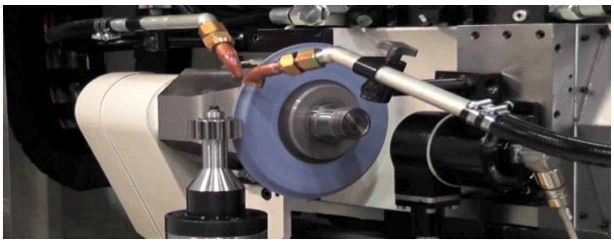

Photo: Gear grinding machine arrangement

Before proceeding actual manufacturing the set up approval required by quality assurance person. In that process parameters are verified in case of these grinding of special gear the parameters set as per below,

Machine on auto mode inspection

Grinding wheel RPM is 1200.

Grade of grinding wheel: Fine 2.0

Coolant used which is MMT80.

First of all machine operator sets zero – zero value of x and y co-ordinate then as per our resultant value 15 micron set in Y co-ordinate direction and start machine.

Due to CNC control machine removed exact 15 micron material from invoute profile of gear. Same was verified by quality inspector which is on video-measuring machine. After that the same modified gear is ready to deliver by production department.

III. VALIDATION

Mathematical validation:

Before putting any values in the equation we consider the face width material reduction which is 15 micron from involute side,

Before modification of face width = 10mm

After FEA results modified face width is = 10mm – 0.015 x 2 …. (For both sides).

Now final face width is 9.97

After addendum module which is 6mm then,

Face width,

$$9.97 \times 6 = 59.82\ mm$$

Theoretical Analysis of Contact Stress using Hertz Equations: Earle Buckingham used the Hertz theory to determine the contact stress between a pair of teeth while transmitting power by treating the pair of teeth in contact as cylinders of radii equal to the radii of curvature of the mating involutes at the pitch point. According to Hertz theory as given in when two cylinders are pressed together, the contact stress is given by,

$$\sigma c = \frac{2P}{\pi BL}$$

$$\sigma c = 205.7387\ MPa$$

As per mathematical formulae & calculation we found that acting stresses is 205.7387MPa.

Stress testing validation:

Refer below test report table which was conducted at external lab. Refer annexure for testing report.

Sample Identification Mark	Load applied	Compressive stress (MPa)
Gear	1300 N	207.13

Table: Stress testing report

IV. RESULTS AND DISCUSSION

After modifying involute profile of spur gear tooth effect as,

Parameter	Without profile modified earlier gear data	Profile modified gear data
Noise	15dB	9 dB
Vibration	3.2 mm at32 Hz	2.4 mm at 25 Hz
Stress	248.453MPa	FEA Analysis : 205.778MPa Mathematical :205.73 MPa Actual on gear stress testing in lab : 207.13 MPa
Life of mating gear average	145 days	Till 200 days and more no issue
Surface finish	Ra 1.2	Ra 0.4

Table 8.1Modifying involute profile of spur gear tooth effect.

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Vibration:

1. Without profile modified : Before modifying involute profile of gear tooth maximum peak value point is 3.2 mm at 32 Hz.

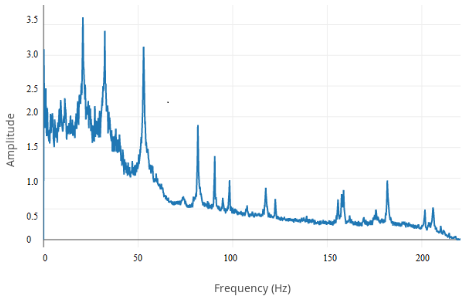

Fig.8.1 Without profile modified vibration graph

2. After modification of involuteprofile: After modifying involute profile of gear tooth maximum peak value point is 2.4 mm at 25 Hz

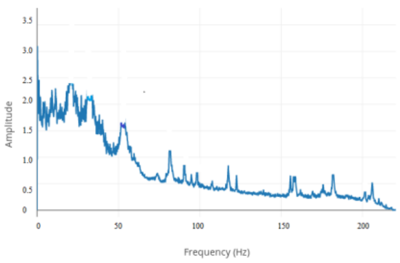

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Fig.8.2 after modifying involue profile of gear vibration graph

DISCUSSION:

In case of validation the results obtained from software analysis and results from testing are compared and they should be nearly equals and for our case the results are nearly equals. So from above test report it is clear that the method we chose for analysis as well as manufacturing modification is right and gives satisfactory results and can predict the outcomes from software analysis are nearly equivalent to actual results obtained from testing.

LIMITATION

1) Young's modulus and Poisson's ratio adds variability and uncertainty to the full technique.

2) Material properties are crucial to accurate finite element computation.

3) Friction and lubrication properties are small and do not lay in the plane of action.

CONCLUSION

It is an optimal gear design method by modifications on an involute tooth working surface for minimizing error fluctuation, vibration excitation & nose including all Hertzian contributions. The results of this procedure is most beneficiaries and it is practically possible invention. Hope it's real revolution in gear design & manufacturing industries.

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