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.

I. 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 manipulating 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.

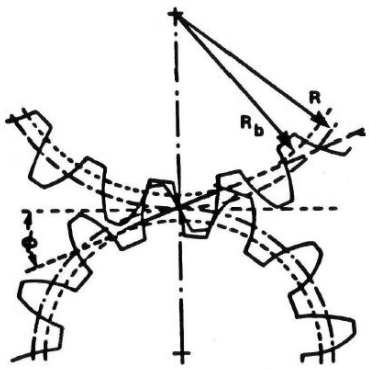

Fig.1. an example of meshing gear teeth is shown. The global pressure angle, Φ, the base radius, Rb, and the pitch radius, R are indicated.

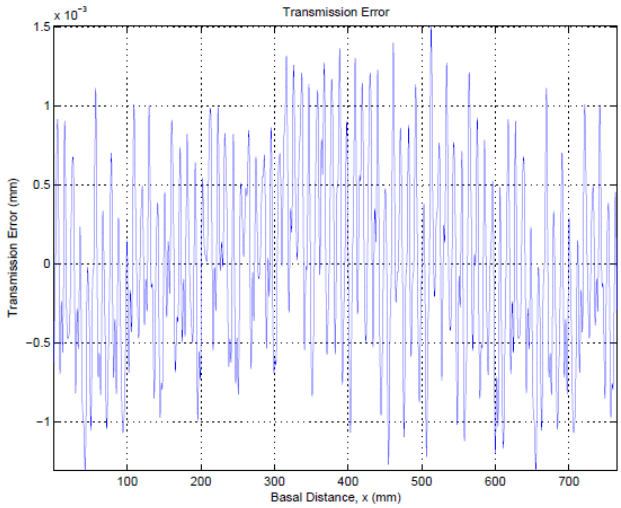

Fig. 2. An example transmission error profile is shown. The horizontal coordinate can be thought of as roll distance of gear. The vertical coordinate is the deviation from gear rolling perfect transfer position expressed at the tooth surface (Around 1.5 cycles of gear rotation are shown).

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.

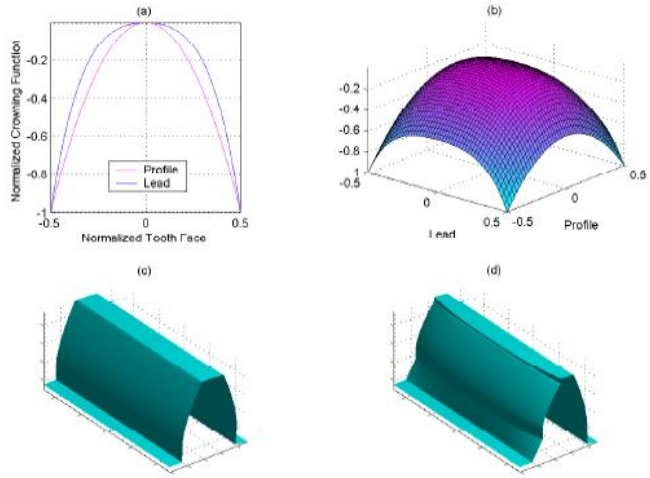

Fig.3. Crowning: Typically a single lead modification and a single profile modification: (a & b) Shows the crowned perfect involute form of gear. Material as described in this deviation from involute is removed from the perfect involute tooth (c). The crowning on normal tooth surface and the direction shown in (d).

[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.

A. 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.

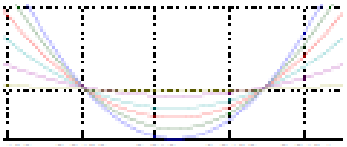

Fig. 4 Stresses on tooth profile.

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 (Δ) & 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 & Δ means actual co-ordinates on gear. Accordingly we will practically remove the material from involute profile by grinding machine.

B. By using mathematical expression of Fourier's transformer:

Refer Fig 5. Shows standard gear diagram of mating gear. From Mark [5].

Refer Fig.6 Shows: Involute curve I wrapped with radius Rb. The roll angle € is the angle swept by the point of tangency T. The pressure angle Φ. The construction pressure angle Φi is equal to Φ when the string is unwrapped to the pitch point P.

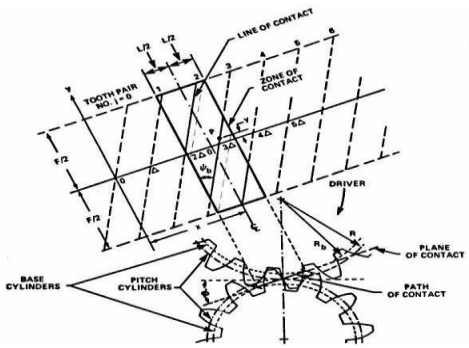

Fig.5

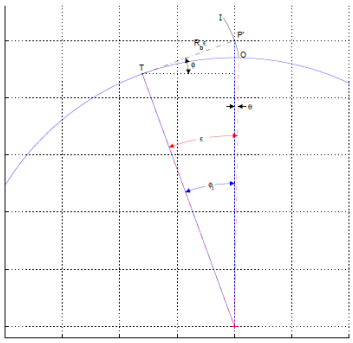

Fig.6

C. The main procedure:

Fourier stated that it is equivalent to describe a time-based function as a sum of frequency components. The Fourier transform,

$$x(f)\int_{-\infty}^{\infty} x(t)e^{-i2\pi ft}\,dt,$$

Allows a function to be analyzed in the frequency domain and to move back and forth to the time domain with its inverse,

$$x(t)\int_{-\infty}^{\infty} x(f)\,e^{-i2\pi jt}\,df,$$

It is standard notation to denote Fourier frequency domain representations with capital letters and time domain function with lowercase letters.

For the computational methods included here, the discrete Fourier transform is more appropriate:
The discrete Fourier transform also has an inverse:

$$\mathrm{x}_n = \frac{1}{N}\sum_{k=0}^{n-1}\binom{n}{k} xk^{e^{-i2\pi kn/N}}, \qquad \mathrm{n} = 0, \ldots., \mathrm{N}-1$$

The complex Fourier series of the transmission error represents the behavior of gear noise. The fundamental tooth-meshing harmonic frequency of n is 1/Δ meaning that there is one Fourier coefficient per harmonic of tooth-to-tooth contact this is shown by,

$$a_n = \frac{1}{\Delta}\sum_{-\Delta/2}^{\Delta/2} \tau(s) e^{-\frac{i2\pi ns}{\Delta}} ds.,$$

The Fourier Uncertainty Principle becomes important in our ability to limit the transmission error of a gear system. The Uncertainty Principle is most famous for describing the inability to know both the location and velocity of an electron at the same time. This idea has a counterpart in Fourier analysis. The more localized or shorter in duration a function of time is, the more broad its frequency distribution.

Similarly, the more tightly the frequency components of a signal are clustered and the more broad the temporal profile. There is a tradeoff between the compaction. and frequency components of a function. Kammler [7] bounds the uncertainty.

$$\Delta s \Delta t \geq \frac{1}{4\pi}$$

This bears on the reduction of noise in gears in that for a function of finite duration there can be no "zeroing" of the frequency components for all frequencies. For gears, Each tooth is in contact for a finite duration; therefore, the Uncertainty Principle directly applies. However, one can eliminate the frequency component for a single frequency or Integer series of frequencies, e.g. f, 2f, 3f, etc. To minimize the noise in gears we "zero" the harmonics of the tooth meshing fundamental (the frequency of tooth-to-tooth Contact) while maximizing the falloff of the remaining frequency components. Side bands of the tooth meshing harmonics, which are caused by shaft rate imbalances, shaft rate is alignment, etc., are minimized as well as these are also multiples of the tooth meshing fundamental. The algorithms behind the Fourier Null Matching Technique are the lifeblood of the procedure and they demonstrate how to implement the above relations to determine a quiet gear design.

At each line of loading, starting with s = 0, a finite element model of the gear is created. An estimate for the initial load is made. The element sizes on the line of contact are found from the Zero Error Cylinder Relationship, The compliance curves for each finite element within the line of contact are determined. The compliance formulation is then solved and the load for the constant deformation step is determined. Using this load the iteration is repeated until the load converges. This gives an accurate relation for the total gear compliance at the constant deformation in that step.

IV. PRACTICAL FLOW

By using any one from an above proposed method we can find out the exact stressing point co-ordinates of meshing gear after that we will get theme for production for modifying this involute profile by machining route.

The capability of gear grinding machine is to be removing minimum material is 1micron. So these type of gear grinding machine are capable to produce as per our resulted range of material removal which is 1micron to 25 micron. Also we indirectly improve the surface finish of contact region of gear will lead reduction in wear tear we will get maximum gear tooth life. After these material removal we achieve our desired target of minimization vibration excitation and gear noise.

V. 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.

VI. 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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