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Modeling and Analysis of Metallic Sintered Spur gear.

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ABSTRACT
Gears are one of the most critical components in mechanical power transmission systems. This is because they are machine elements used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth. This research focuses on modeling and analysis of sintered spur gears which are used to transmit motion and power between parallel shafts. Sintered spur gears are manufactured through powder metallurgy processes. One of the challenges faced in using sintered spur gears is interference in the involutes system of gearing. In this research a C-program is developed for the modeling of a sintered spur gear for both standard and profile corrected tooth. This model is then imported to an analysis software ANSYS for carrying out the static analysis. Finally the comparisons for stress is carried out for both the standard sintered spur gear and profile corrected sintered spur gear. The results are tabulated and graphically shown. It was seen that the stresses due to interference for the profile corrected sintered spur gear was considerably less as compared to the standard sintered spur gear. Another contribution of this research also provides a platform or method for mechanical engineers to asses and evaluates performance of gears.

Keywords- *Gears, Sintered gears,Interference & Involute systems,Powder metallurgy.*

I. INTRODUCTION

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. The rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined application of gear technology. Gears are machine elements used to transmit rotary motion between two shafts, usually with a constant speed ratio. The pinion is the name given to the smaller of the two mating gears; the larger is often called gear or the wheel. Fig1.1 indicates a typical gear profile.

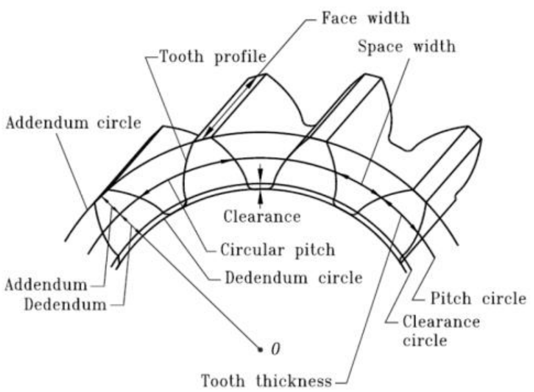

Fig.1 Gear Tooth Profile

The noise reduction in gear pairs is especially critical in the rapidly growing today's technology since the working environment is badly influenced by noise. The most successful way of gear noise reduction is attained by decreasing of vibration related with them. Thus studying how to reduce noise by reducing controlling vibration assumes significance. Spur gears are currently being extensively used increasingly as a power transmitting gear owing to their relatively smooth and silent operation, large load carrying capacity and higher operating speed. Designing highly loaded spur gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods that can easily be put into practice and also give useful information on contact and bending stresses. The major cause of vibration and noise in a spur gear system is the transmission error or interference between the meshing gears. According to Sfantos et al., Generally transmission error is due to two major causes. The first cause is production incorrectness as well as mounting mistake and the second one is caused by elastic deflection at the time of loading. Transmission error is considered as one of the main contributor to noise and vibration in a gear system. If a pinion and a gear have ideal involutes profiles running with no loading they should theoretically run with zero transmission error. Even the slight variations can cause noise at a frequency which matches a resonance of the shafts or the gear housing causes noise to be enhanced. Hence its needs to put in place production systems that are capable of producing spur gears that transmit without much interference.

II. METHODOLOGY

Powder Metallurgy (P/M) has been defined as the art and science of producing metal powders and making semi-finished and finished objects from individual, mixed or alloyed powders with or without the addition of non-metallic constituents. The basic PM process can be seen in Fig.2

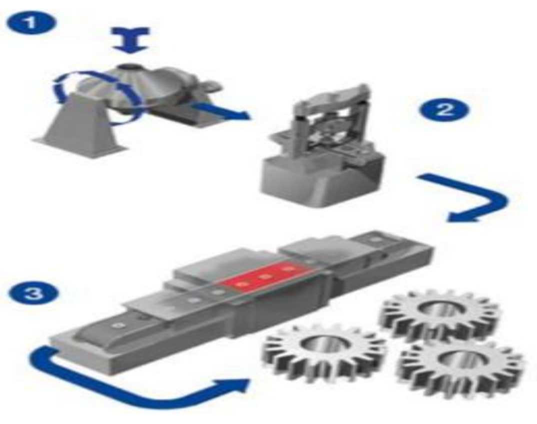

Fig.2 Manufacturing of Gears

The basic steps in PM production of gears are shown in the above Fig.2,(a) Powder mixing,(b) Compaction,(c) Heat treatment. At the mixing stage powder is mixed to ensure a homogeneous mixture of the metal powder and from here the powder is fed onto a single or multiple cavities.
A hydraulic or mechanical piston then compresses the powder into the cavity. After the green part is ejected from the cavity and transported to the sintering furnace, here the part is heat treated.

III. PROFILE CORECTION OF GEARS

Research conducted by Osman et al, showed that the interference may be avoided if the path of contact does not extend beyond interference points. According to Weber et al, the modern trend in gear technology envisages the use of corrected gears in most of the power transmitting and other areas, to avoid undercutting or to achieve a predetermined center distance. In positive corrected gear the addendum is increased by an amount of xm and the deddendum is correspondingly decreased by the same amount of xm. The reverse is true in the case of negatively corrected gears. The following are the characteristics will effects when positively corrected gears are used:

- Avoidance of undercutting
- To increase the strength of the root and the flank of the tooth. It will be shown that due to positive correction; the thickness of the tooth at the root is increased, resulting in a greater load capacity of the teeth. By choosing the proper amount of correction, the designer is in a position to specify gear sets of higher capacity without entailing the corresponding cost increase for materials of higher strength.
- Betterment of sliding and contact relations.
- In a corrected gear, the following parameters remain changes: Tip circle diameter becomes bigger by an amount of +2xm in the case of S-plus gears and smaller by-2xm in case of S-minus gears.
- It is customary to express the amount of correction in terms of 'm' since x is dimensionless. xm is in millimeters. The magnitude by which gear blank is to be made bigger or smaller radially from the center of the blank, depends on whether x is positive or negative.

Table 1 Dimensions of Standard Gear
(PCD = 200mm, ϕ=20^o, x=0)

Module	7	8	9	10
No of Teeth	29	25	22	20
Addendum Dia, d_a,mm	214	216	218	220
Deddendum Dia, d_d,mm	183.53	180.23	179.14	176.86
Base Circle Dia, d_b,mm	188	188	188	188
Tooth Thickness t,mm	11.4	12.5	14.28	15.7

Table 2 Dimensions of corrected Gear
(PCD = 200mm, ϕ=20^o, x=0.62)

Module	7	8	9	10
No of Teeth	29	25	22	20
Addendum Dia, d_a,mm	225	226	227.2	232.4
Deddendum Dia, d_d,mm	184.18	185.5	186.7	187.4
Base Circle Dia, d_b,mm	188	188	188	188

Tooth Thickness t,mm	14.2	16.2	18.2	

IV. GEOMETRICAL MODEL OF GEARS

Modeling of entire gears in mesh would significantly increase the complexity and size of geometric and numerical model which would, in turn, result in prolonged calculation time. Thus, already in modeling phase certain simplifications have been made. Only parts of the rims of the wheel and the pinion have been modeled Fig. 3, both with two whole teeth and two teeth segments.

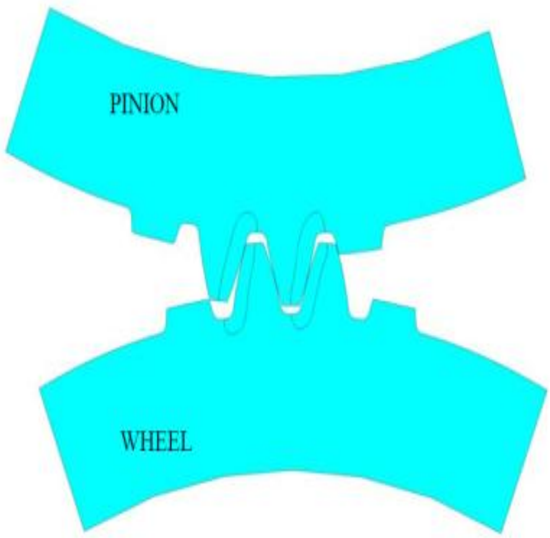

Fig.3Geometry of gears in mesh

V. MESHING OF GEAR MODEL

Three types of finite elements have been used for meshing of gear models. Gear models have been divided in areas and they have been mashed with elements PLANE183. These elements are defined by 8 nodes, having two degrees of freedom at each node: translations in the nodal x and y directions and are well suited to modeling irregular meshes.

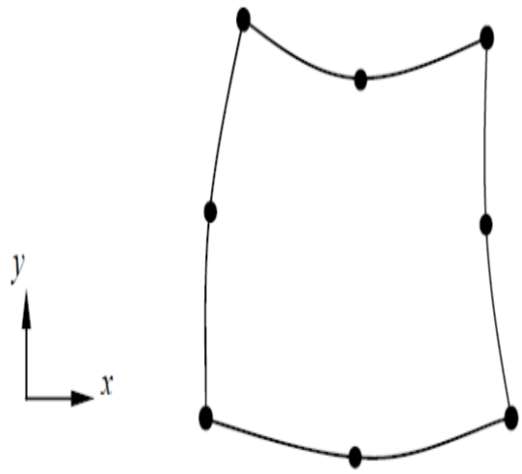

Fig.4, PLANE 183 Finite Elements

Parts of teeth flanks in contact have been meshed with contact elements TARGE169 and CONTA172. These parabolic elements (Fig.3) with two nodes on end and one midside node each with two degrees of freedom (translations in the nodal x and y directions) are very suitable for analysis of problems with states of plane stress and plane strain.

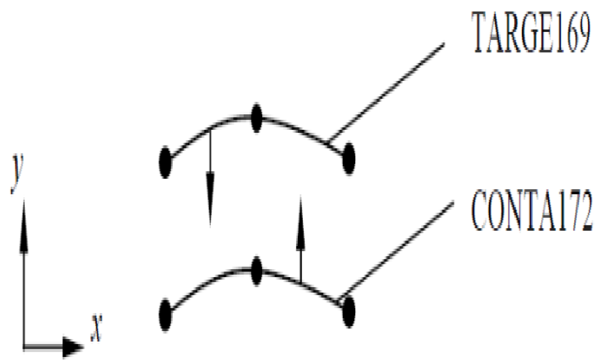

Fig. 5, Contact finite elements

In order to further decrease calculation time, finite element mesh has been adapted as well. Areas around contacting surfaces have been meshed with larger density of finite elements mesh because these areas are crucial for results accuracy. Coarser finite elements have been used in areas of less significance such as gear rim and parts of gear teeth that are not in the contact. Meshed gear model is shown in Fig. 6

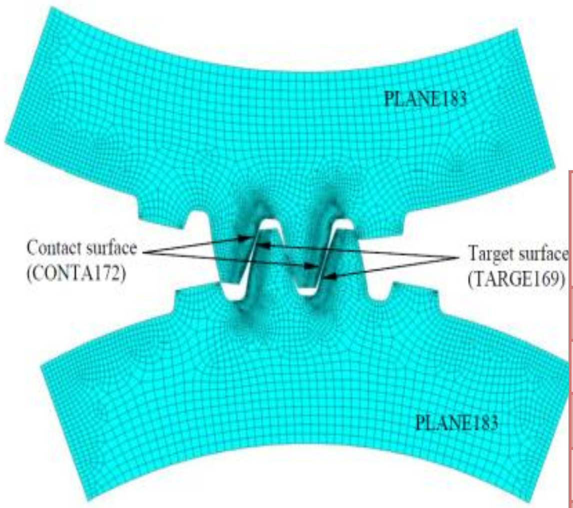

Fig. 6 Meshed Gear Model

VI. BOUNDRY CONDITIONS

Modeling of entire gears in mesh would significantly increase the complexity and size of geometric and numerical model which would, in turn, result in prolonged calculation time. Thus, already in modeling phase certain simplifications have been made. Only parts of the rims of the wheel and the pinion have been modeled Fig. 3, both with two whole teeth and two teeth segments. The gears have been loaded by positioning mating teeth i.e. their flanks into contact due to inadequacy of other loading models . Namely, concentrated force couldn't be applied due to high local deformation of the material which takes place near point of force action and significant influence on the results. After positioning the mating teeth in desired position the boundary conditions have been applied. The wheels' nodes placed on inner rim radius and on the ends of rim have been constrained in global Cartesian coordinate system (x, y) in all directions i.e. the movements in directions of both axis have been disabled ($\Delta x=0$, $\Delta y=0$). The pinions nodes placed on inner rim radius have been constrained in global cylindrical coordinate system (r, φ) in a way: $\Delta r=0$. Centre of both mentioned coordinate systems have been centre of rotation of the pinion.

VII. RESULTS & DISCUSSION

Modeling of entire gears in mesh would significantly increase the complexity and size of geometric and numerical model which would, in turn, result in prolonged calculation time. Thus, already in modeling phase certain simplifications have been made. Only parts of the rims of the wheel and the pinion have been modeled Fig. 3, both with two whole teeth and two teeth segments. Gear tooth root stresses along the path of contact in standard model have been calculated and then compared to the stresses in modified one to present the influence of determined profile modification on gear tooth root stresses. The results of FEM analysis for pinion and wheel are shown in Fig. 7. For standard unmodified model when double contact exceeds into single contact (point B on path of contact) and reverse (point D on path of contact) gear tooth root stress changes rapidly i.e., the wheel speed changes at two shifting points and causes the additional dynamic load as visible in Fig.3

Table 3. Comparison of standard and corrected gear stresses.

Helix Angle(β)	σ $_{(Exp)}$ MPa	σ $_{(ANSYS)}$ MPa	Difference (%)
15^0	56.94	58.17	2.11
20^0	55.98	57.70	2.98
25^0	53.15	54.87	3.15
30^0	49.95	52.33	4.54
35^0	44.68	47.07	5.02

Instead of the first contact between meshing gears with linear tip relief profile modification on the pinion tooth tip (point A on path of contact), it occurs lower on tooth flank (point A' on path of contact). The same situation appears at point E. Gear tooth root stress increment between points A' and B' (double contact) and decrement between points D' and E' (double contact) are almost linear. There aren't rapid stress changes at the shifting points so gears run smoother than standard gear pair without additional dynamic load.

The analysis also showed that the highest values of the tooth root stresses ppear in point B on path of contact for standard and in point B' for modified model.

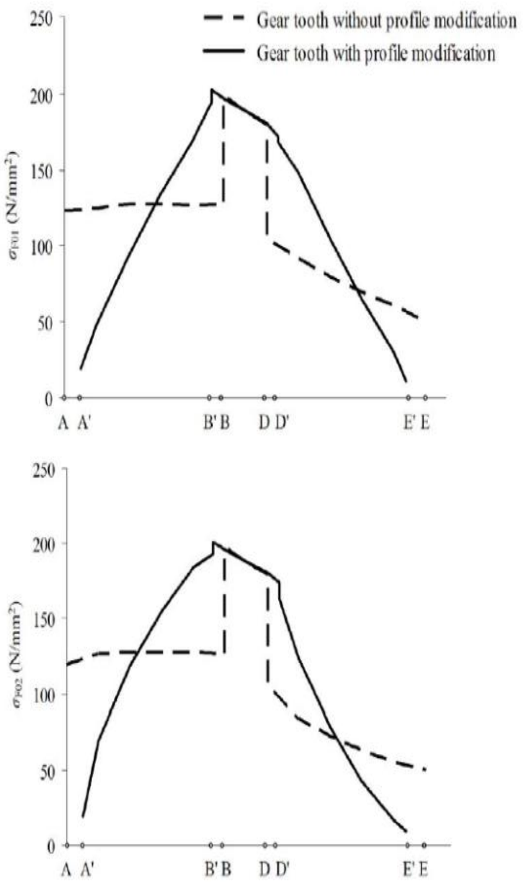

Fig.7, Tooth root stress for pinion and wheel

VIII. CONCLUSION

The standard gear model and also modified one have been developed and analyzed by using finite element method. Nonlinear analysis has been used because it gives the most accurate results. Numerical calculation methods such as finite element method provides easier stress calculations on teeth with no limits in gears' geometrical specifications and also allows determination of stress distribution on whole path of contact. Obtained results show that in case of standard unmodified model when double contact exceeds into single contact and reverse gear tooth root stress changes rapidly i.e., the wheel speed changes at two shifting points, and causes the additional dynamic load, unlike, in case of modified model wheel speed don't change rapidly so there aren't rapid stress changes at the shifting points. Also, instead of the first contact between meshing gears with linear tip relief profile modification on the wheel tooth tip it occurs lower on tooth flank. The same situation appears at the end of contact between meshing gears with linear tip relief profile modification. This phenomenon results in a way that gear tooth root stress increment and decrement on double contact zones are almost linear so gear pair with linear tip relief profile modification runs smoother than standard gear pair.

XI. ACKNOWLEDGEMENTS

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