Design and Development Model of Spiral Bevel Gear with Minimized Weight

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***Abstract*—This paper, the spiral bevel gear (SBG) is a key component of the power transmission of non-parallel shafts.Considering example such as hand held tool wood working machines, these machines are used to cut the wood work-piece for the making furniture, Casting pattern, wooden seat design, wood prototyping etc. In that machine the set of spiral bevel gears used for power transmission from motor to tool. The hand held tools weight and continues vibrations makes it difficult to operate the machine for longer time and also power consumption per unit cut has been very high , and vibrations lead to inaccuracy in cutting and error in profile shape. Thus methodology used in study is to carry out test on three sets of bevel firstly plain gears (i.e. no weight reduction), secondly weight reduction done by face counter and thirdly by providing equidistance holes on the face. The development of spiral bevel gear we can reduce weight, material, process timing and cost of production.Comparative the performance analysis of the gears by load so as to derive the optimal performance of the gears.**

***IndexTerms*—Weight reduction, Face recess, Face holes, optimal performance characteristic.**

I. INTRODUCTION

The gear is use as a mechanical devicein transmission systems it allows to transfer the rotational force to another gear or device. The gear teeth, or cogs, allow force to be fully transmitted without slippage and depending on their configuration, can transmit forces at different speeds, torques, and even in a different direction. Throughout the mechanical industry, many types of gears exist with each type of gear possessing specific benefits for its intended applications.Mostly bevel gears are used for powertransmission between nonparallel shafts at almost any angle or speedbecause of their suitability. Spiral bevel gears have curved and sloped gear teeth in relation to the surface of the pitch cone. As a result, an oblique surface is formed during gear mesh which allows contact to begin at one end of the tooth (toe) and smoothly progress to the other end of the tooth (heel).

II. PROCEDURE

A. Topology optimization

Topology optimization of continuum structures is most challenging technically and rewarding economically. Rather than limiting the changes in the sizes of structural components, topology optimization provides much more freedom and allows to designer to create totally novel and highly efficient conceptual design for continuum structures. The stress level in all part of a structure can be determined by using a finite element analysis. The reliable indicator of inefficient use of material is low values of stress (or strain) in some parts of the structure. Ideally the stress in all part of the structure should be close to the same or safe level. This concept leads to the rejection criterion based on local stress. Where the low-stressed material is assumed to be under-utilized and is therefore eliminated subsequently. The removal of material can be conveniently undertaken by removing elements from the finite element model.

Topology optimization method has been used to optimize the structure of the gear. The minimum volume was set as the direct optimization goal. The topology optimization can provide designers with a conceptual design at the initial stage of a structural design, thus it improve the design efficiency, design quality, and reduce the development costs.

B. Problem Statement

The weight of the hand held tools and subsequent vibrations makes it difficult to operate the machine for longer time and also high power consumption, and vibrations lead to in accuracy in profile cutting and error in shape.

Thus it required study is to carry out test on three sets of bevel gears namely plain (i.e. no weight reduction), secondly weight reduction done by providing recess on the face of gear , an thirdly by providing even number and equi-spaced holes on the face.

C. Objective

Effect of weight reduction on vibration of gear through experimentation validation.

III. VIBRATION ANALYSIS-THEORETICALLY

Router machine is used to excavate the material as a cavity by plunging the router tool into the material, thus the router process takes place in two stages;

a) Drilling the given profile hole size into the material: This process consumes maximum power and accounts for maximum vibration.

b) Milling process involves the lateral movement of tool with reference to the work-piece to achieve the desired length of cut, this process accounts for lesser power consumption as compared to drilling process hence accounts to lesser vibration.

Thus the plunge drilling operation characteristic are used to determine the power requirements to account for maximum factor of safety.

Thus the analogy of drilling a maximum size of hole i.e. 12mm in aluminum alloy material.

Thus moment required by the machine to perform drilling operation on the given profile is given by following relations:

A) MOMENT (M) = Hb x D2 x f / 8

Here material to be machined is aluminum alloy where

Hb = 95 kgf/mm2 for aluminum alloy

D= diameter of drill = 12 mm

F= in feed = 0.005 mm/rev

Hence M = 95 x 12 2x 0.05 / 8 = 10.68 kgf –mm = 106.8 N-mm

M= 0.106.8 N-m

Thrust developed during the metal cutting operation is given by

T = 1.7 TO 3.5 (M/d) = 1.7 X (0.106 X 103/ 12) = 15.016N

T = 15 N =1.52 kg

This is the maximum force that is acting on the cutter in router operation vertically upward direction lading to vibration which needs to be damped by the damper.

Thus the loading of the system developed using a load dynamometer will carry load of 1.5 kg to 2.5 kg as in 1.5 kg, 1.8 kg, 2.0 kg, 2.3kg, 2.5kg, this is to check the frequency response of the system under various loading condition i.e. , 100% loading to 160% loading thereby ensuring the safe operation of the system for allowable acceleration in device not to exceed 7.7 m/sec2.

Theoretical Calculation for Displacement and acceleration-

Power input = 350 watt

Maximum acceleration = 7.7 m/sec2

Angular speed (ω) = 2 π N/ 60

(ω) = 2 π x 3000 / 60 =314.16 rad/sec

Let F0=Force transmitted to machine handle / foundation

F0= m0 e ω2

(m0 e) = Rotating imbalance owing to the dynamometer pulley action

= (1.5 x 0.1/2) = 0.075 kg-m

Thus;

F0= m0 e ω2

F0= 0.075 x 314.22

F0=7.4x103 N

Considering the maximum transmitted ratio as ;

T = FT / F0

The maximum permissible amplitude of force transmitted not to exceed 3500 N

T = 3500 / 7.4 x103

T= 0.47

Now as T<1 ,

T = 1/(r2-1)

r = Sq. rt (1 +1/0.47) = 1.76

Now,

Natural frequency (ωn) = ω /r

(ωn) = 314.2/ 1.76 = 178.5 rad /sec

Now Equivalent stiffness of the system is given by,

Keq = m ωn2

Keq = 1.5 x 1178.52

Keq = 47.7x103 N/m

Determination of maximum theoretical displacement of system:

Keq = W/ δ

δ = W / Keq

$\delta = (1.5) \text{ X } 9.81 / 47.7 = 0.308$ mm

Thus maximum displacement of the system $\delta = 0.308$ mm
δ =0.308mm-----rounded off to o.3 mm

Theoretical determination of acceleration under given system of forces:
Ft = kx + cx
Cx = Ft – kx=(1.5) – (47.7/9.81 x 0.3) =0.04
C= 0.04 / x = 0.04 /0.3 =0.133
If the base of the system is subject to displacement y(t)., then the acceleration transmitted to the machine of mass m is determined as
ā̄ = (c z + k z) / m
Where
ā̄ = acceleration transmitted to the machine m/sec2
z = displacement of the machine relative to its base and is equal to the total displacement of the isolator.
Ā = 0.133X 0.3 +4.77 X 0.3 = 1.471m/sec2
Thus maximum acceleration of the machine at no load condition is
147.1 m m/sec2
As the actual acceleration as effect of load of 1.5 kg on dynamometer pulley is 1.47m/sec2 < allowable 7.7 m/sec2 the system is safe for use in hand tools as the maximum accelerations well within control.

IV. EXPERIMENTATION

Rope brake dynamometer with effective diameter pulley is used= 100 mm
Procedure of trial:
1. Add 1.5 kg load and take rpm reading using Tachometer
2. Note displacement reading on vibrometer.
3. Note acceleration reading on vibrometer.
4. Repeat procedure for 1.8, 2.0, 2.3, 2.5 kg load.
5. above same procedure for three different gear.

Fig 4.1. Experimental Setup

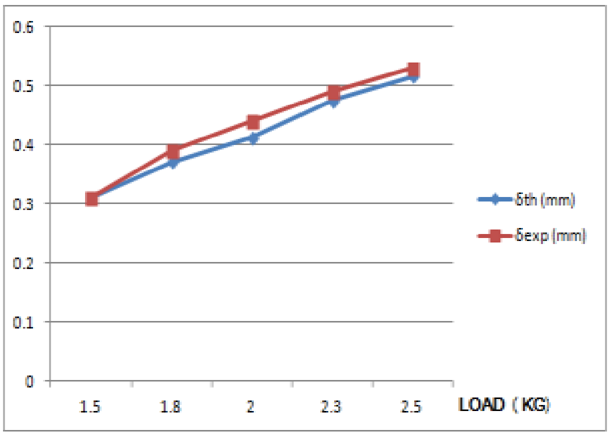
A. Plain Bevel Gear *B. With hole C. Face Counter reduction*

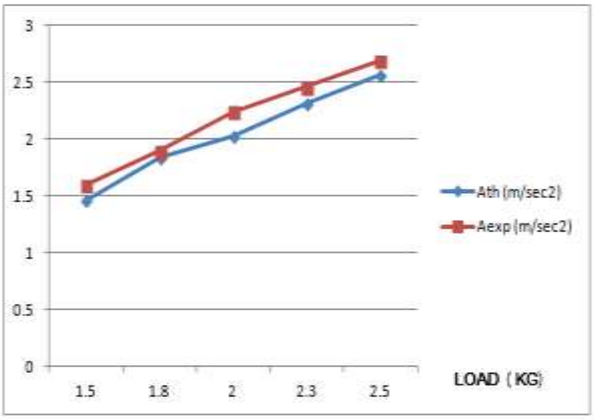
V. TEST AND RESULT

A. On plain spiral bevel gear

Result table for theoretical displacement and acceleration

Sr. no	Load (kg)	Theoretical		Experimental	
		Displacement (mm)	*Acceleration (m/sec2)*	*Displacement (mm)*	*Acceleration (m/sec2)*
1	1.5	0.30958	1.47	0.31	1.6
2	1.8	0.37150	1.835	0.39	1.9
3	2.0	0.41277	2.034	0.44	2.24
4	2.3	0.47469	2.315	0.49	2.46
5	2.5	0.51597	2.56	0.53	2.69



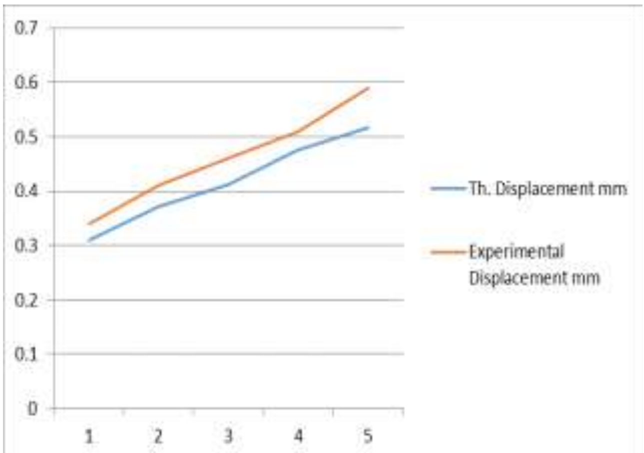


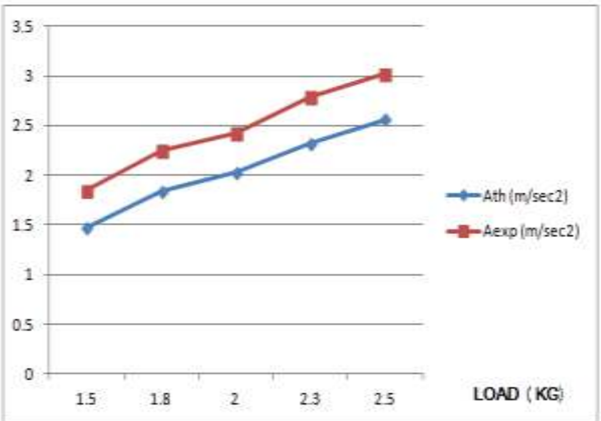
Graph 1. Load vs. Displacement**Graph 2.** Load vs Acceleration

B. On plain spiral bevel gear –with hole reduction

Result table for theoretical displacement and acceleration

Sr. no	Load (kg)	Theoretical		Experimental	
		Displacement (mm)	*Acceleration (m/sec^2)*	*Displacement (mm)*	*Acceleration (m/sec^2)*
1	1.5	0.30958	1.47	0.34	1.85
2	1.8	0.37150	1.83	0.41	2.25
3	2.0	0.41277	2.03	0.46	2.43
4	2.3	0.47469	2.31	0.51	2.79
5	2.5	0.51597	2.56	0.59	3.02



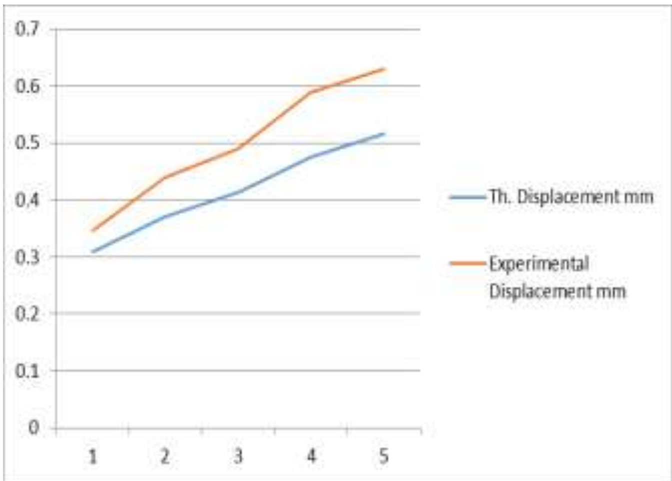


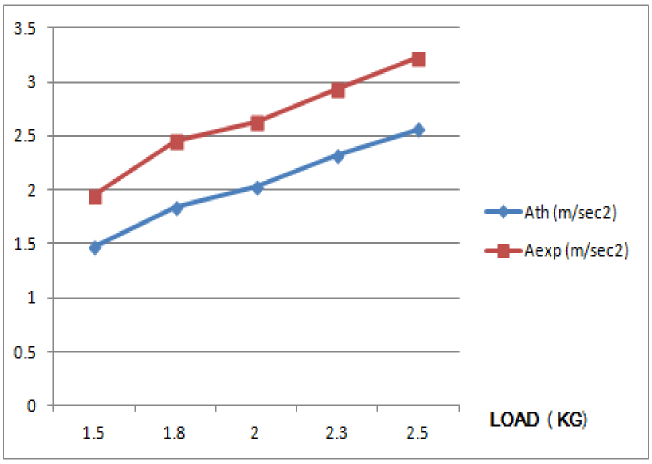
Graph 3. Load vs. Displacement**Graph 4.** Load vs Acceleration

C. On spiral bevel gear –with face counter reduction

Result table for theoretical displacement and acceleration

Sr. no	Load (kg)	Theoretical		Experimental	
		Displacement (mm)	*Acceleration (m/sec^2)*	*Displacement (mm)*	*Acceleration (m/sec^2)*
1	1.5	0.30958	1.47	0.347	1.95
2	1.8	0.37150	1.83	0.44	2.45
3	2.0	0.41277	2.03	0.49	2.63
4	2.3	0.47469	2.31	0.59	2.94
5	2.5	0.51597	2.56	0.63	3.22





Graph 5. Load vs Displacement Graph **Graph 6**. Load vs. Acceleration

VI. CONCLUSION

Maximum acceleration in case of the bevel gear with face reduction is 3.22 m/sec2 which is slightly higher as compared to (3.02 of bevel gear with hole reduction and 2.69 with plain bevel gear) …thus the bevel gear with face counter reduction can be recommended over other two gears.

Maximum weight reduction is achieved by face counter reduction 14.15 %.3. Maximum stress in all conditions is well below the allowable limit hence weight reduction by either methods is recommended Weight reduction by the face counter method is recommended as it offers minimal deformation and only marginal increased stress as compared to the hole reduction method.

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