International Journal of Engineering and Technical Research (IJETR)
ISSN: 2321-0869 (O) 2454-4698 (P), Volume-5, Issue-1, May 2016

Design and Optimization of Spiral Bevel Gear

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Abstract— **Wood working machines are used to cut wood work-piece in furniture making, Casting pattern making, wooden seat design, wood prototyping etc. They use a set of spiral bevel gears for transmission of power from motor to tool in the application. The weight of the hand held tools and subsequent vibrations makes it difficult to operate the machine for longer time and so also power consumption per unit cut has been found to be 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 gears namely plain(ie no weight reduction), secondly weight reduction done by providing recess on the face of the gear , an thirdly by providing even number , equispaced holes on the face. Comparative analysis of the performance of the gears by

Load so as to derive the optimal performance of the gears by optimization of spiral bevel gear we can reduce weight of bevel gear and thereby cost of product or gear is depending on manufacturing process & material used for it.

Index Terms—**Weight reduction, Face recess, Face holes, Vibration Analysis.**

I. INTRODUCTION

A gear is a mechanical device often used in transmission systems that allows rotational force to be transferred 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. Bevel gears are widely used because of their suitability towards transferring power between nonparallel shafts at almost any angle or speed.

II. PROCEDURE

A. Topology optimization

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

Topology optimization method was 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 in the initial stage of a structural design, thus improve design efficiency, improve product design quality, and reduce 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 so also power consumption per unit cut has been found to be very high , and vibrations lead to inaccuracy in cutting and error in profile shape.

Thus there is a need in 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 the gear , an thirdly by providing even number , 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 D^2 x f / 8

Here material to be machined is aluminum alloy where

Hb = 95 kgf/mm^2 for aluminum alloy

D= diameter of drill = 12 mm

F= in feed = 0.005 mm/rev

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Hence M = 95 x 12 ^{2}x 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 10^3/ 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/sec^2

Angular speed (ω) = 2 π N/ 60

(ω) = 2 π x 3000 / 60 =314.16 rad/sec

Let F_0=Force transmitted to machine handle / foundation

$F_0= m_0 e \omega^2$

(m_0 e) = Rotating imbalance owing to the dynamometer pulley action

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

Thus;

$F_0= m_0 e \omega^2$

F_0= 0.075 x 314.2^2

F_0=7.4x10^3 N

Considering the maximum transmitted ratio as ;

T = F_T / F_0

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

T = 3500 / 7.4 x10^3

T= 0.47

Now as T<1 ,

T = 1/(r^2-1)

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

Now,

Natural frequency (ω_n) = ω /r

(ω_n) = 314.2/ 1.76 = 178.5 rad /sec

Now Equivalent stiffness of the system is given by,

$K_{eq} = m \omega_n^2$

K_{eq} = 1.5 x 1178.5^2

$\boldsymbol{K_{eq}}$ **= 47.7x10^3 N/m**

Determination of maximum theoretical displacement of system:

K_{eq} = W/ δ

δ = W / K_{eq}

δ = (1.5) X 9.81 / 47.7 = 0.308 mm

Thus maximum displacement of the system δ =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

ā̃ = (c z + k z) / m

Where

ā̃ = acceleration transmitted to the machine m/sec^2

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/sec^2

Thus maximum acceleration of the machine at no load condition is

147.1 m m/sec^2

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 accelerationis well within control.

IV. EXPERIMENTATION

Fig 4.1. Experimental Setup

Rope brake dynamometer with effective diameter pulley is used= 100 mm

Procedure of trial:

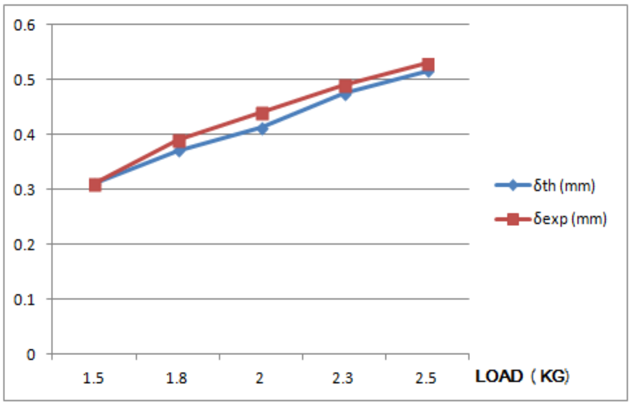
1. Add 1.5 kg load and take rpm reading using Tachometer
2. Note displacement reading on vibrometer
3. Note acceleration reading on vibrometer
4. Repeat procedure for 1.8, 2.0, 2.3, 2.5 kg load.
5. above same procedure for three different gear.

V. TEST AND TRIAL

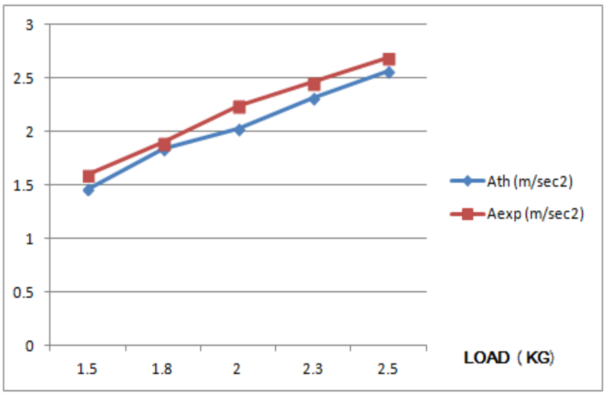
A. On plain spiral bevel gear

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	0.31	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

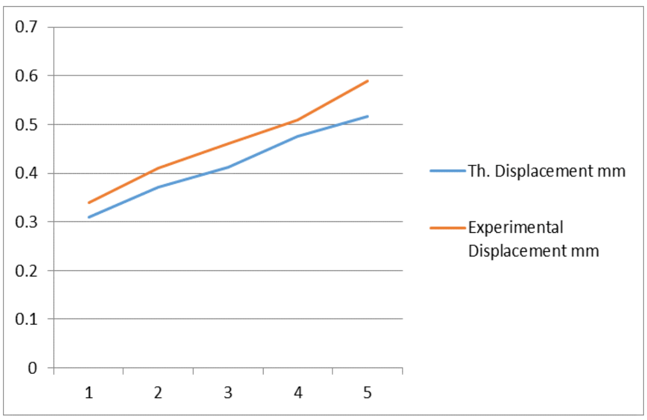


Graph 2. Load vs Acceleration

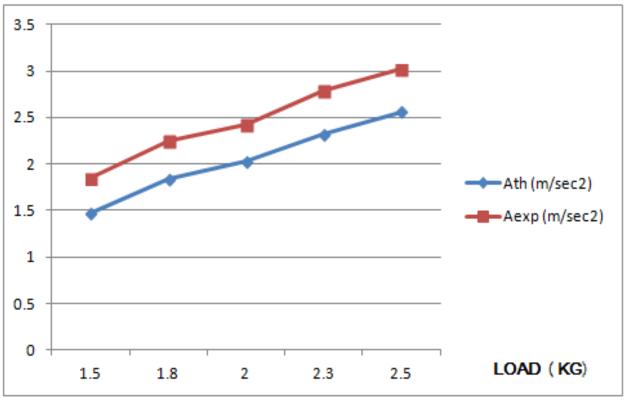
B. *On plain spiral bevel gear –with hole reduction*

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

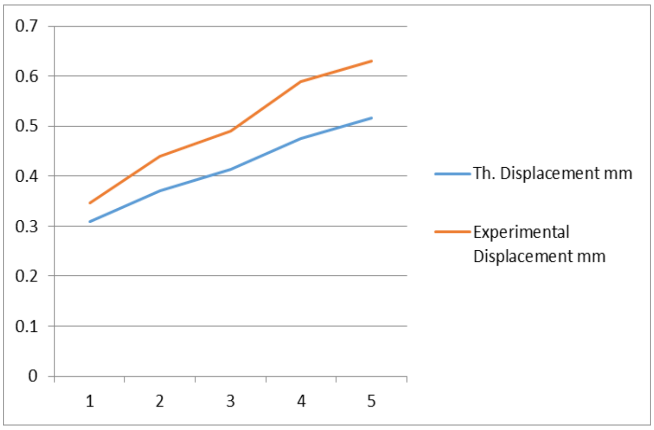


Graph 4 Load vs Acceleration

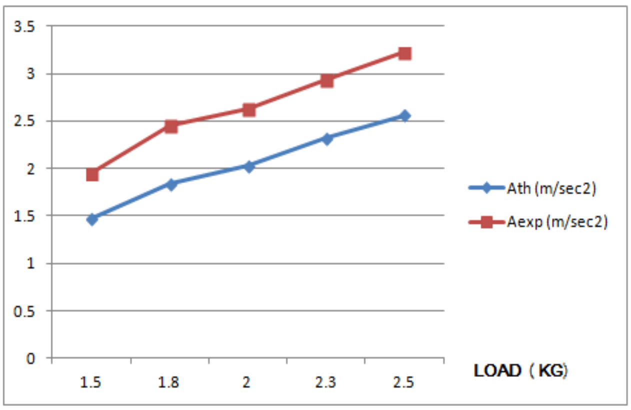
C. *On spiral bevel gear –with face counter reduction*

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 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 m/sec2 of bevel gear with hole reduction and 2.69 m/sec2 with plain bevel gear)thus the bevel gear with face counter reduction can be recommended over other two gears..

Acknowledgment

My profound thanks to my guide Prof. N.D.Padwale sir, I am thankful to Prof. S.K. Bhor sir Head of Department of Mechanical Engineering for his invaluable advice and constant encouragement to complete this Paper in a successful manner. I am thankful to our ME coordinator Prof. P R Sonawane sir for his kind support and providing all facilities And academic environment for my Research paper work. I would like to express my gratitude to our esteemed Principal Dr. R J Patil sir for his encouragement.

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