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Design & Development of Parkinson Gear Tester for Spur Gear to Check the Flank Surface

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Abstract

In order to check the combined tooth error different types of gear testing machines are used. Various machines have its ability to check specified parameters only. Highly precise machine required special installation and space. For the purpose of checking gear in machine shop while performing machine required such an arrangement which is robust and quick one. This purpose can be solved using gear test rig. This type of gear test rig can be used for mass production of gears of a particular gear box.

Gear test rig is such arrangement which simplifies the measurement and saves the labourtime and labour cost with greater accuracy. In gear test rig all the gears will be mounted on a plate which may be fixed or stationary as per the requirement of the measurement. While measuring the one gear remaining will act as a master gear. This will help in finding the composite error. This test rig can be used in shop floor as it requires less space and operator can use it as per need without wasting much time. The test rig can be developed for different parameter as per measurement requirement. There are various test rigs which can be used for that particular designed condition.

Keywords

Gear,flank surface.

Introduction

Today world requires speed on each and every field. Hence rapidness and quick working is the most important. Now a day for achieving rapidness, man manufactures various machines and equipments. The engineer being constantly conformed to the challenges of bringing ideas and new design in to reality. New machines, equipments and the techniques are being developed continuously to manufacture various products at cheaper rates and high quality.

This paper"Design & Development of Parkinson gear tester for spur gear to Check the Flank Surface" being compact and Portable equipment, which is skillful and is having some thing precise in testing the gears being manufactured. Most of the material is made available by our college. The parts can be easily made in our college work-shop. It's price is also considerable. This project gives us knowledge, experience , skill and new ideas of manufacturing. It is a working project and having guarantee of success. This project is the equipment useful to improve the quality of the gear being manufactured and can be made in less time, hence we have selected this paper.

History

Gears have been in use for hundreds of years and it will be continue for few more years.Sometime these gears are manufactured in mass production like manufacturing the gears of a specific machine's gear box. Gear performance depends on various parameters such as material, design, manufacturing, operation and environment. Manufacturing of the gear is avery important step which decides the accuracy of the gear. This requires the inspection at various steps. Also this inspection should not consume too much money in terms of labourand time. That's why it should be easy to inspect and operate. Gear test rig is such arrangement which simplifies the measurement and saves the labourtime and labour cost with greater accuracy. In gear test rig all the gears will be mounted on a plate which may be fixed or stationary as per the requirement of the measurement. While measuring the one gear remaining will act as a master gear. This will help in finding the composite error. This test rig can be used in shop floor as it requires less space and operatorcan use it as per need without wasting much time. The test rig can be developed for different parameter as per measurement requirement. There are various test rigs which can be used for that particular designed condition.

From the above study it can be concluded that there are various test rig which are used formeasurement of particular parameters. Design of test rig differs as per the requirement of application and as per requirement of parameter to be tested.

Development of testrig

To operate the testing machine, electric motor (prime mover), which is torque motor having 20 N-m torque capacity, is used to rotate the master gear against the gear to be tested. Also an another motor of the same capacity is used to rotate the paper rolling drum to pass the recording paper against the vibrating pen and stylus due to the improper tooth geometry provided.

The gear to be tested is installed on the trolley gear shaft using the fasteners as the nut and bolts. The trolley being spring loaded is in continuous close contact with the master gear. The master gear shaft is extended and is coupled with the driving torque motor using a coupling.

When the pair of master gear and the gear to be tested is rotating and if there is any mis-run of the gear to be tested then the stylus and pen arrangement will deflect and the appropriate amount of variation in the graph which is recorded on the moving paper is being recorded. Thus the operation of gear testing machine is done.

Principle

It works on the principle of measurement of the mis-run of the smooth running of the precisely meshing gears (when rotated with respect to each other) with any variation in the geometry of the gear tooth profile due to the wear and tear by the periodic use or the faulty manufacturing.

Special Features

It is having the precisely measuring capacity and the reproducibility of Parkinson gear testing machine due to stylus and pen and spring arrangement is high. So however, a small variation in the flank of the gear tooth will deflect the stylus along with the pen. We get the consistent force on job. Previously the operations, which this machine does, were done individually in different comparing-machines such as optical profile protector or by testing manually, using gear tooth varniercalliper.

Its special features are:

- It applies the accuracy up to 1 micron.
- It is light in weight and hence it is portable.
- Weight of machine is 30 kg.(approx.)
- It requires very low maintenance.
- Its setting time is less.
- It requires very low floor space area.
- Its manufacturing cost is also very low.
- No separate arrangement of drawing sketch is required.
- It requires low power for its operations hence it can be excited using d.c.
- Power and d.c. motor (also to be used in remote areas also).
- It is compact. Total length of machine is 1200 mm.

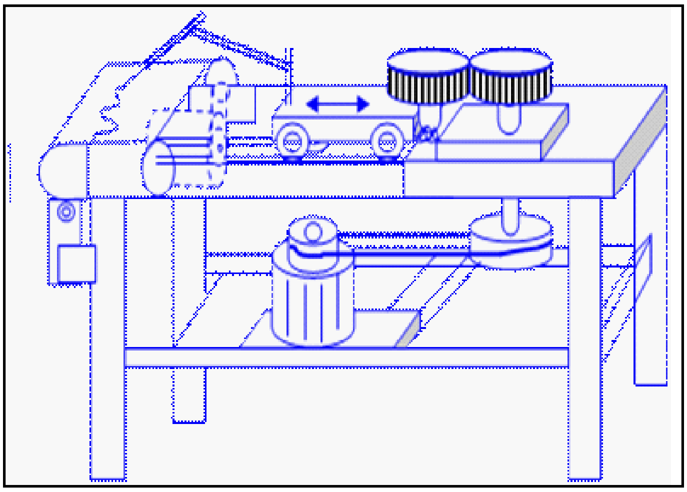

Fig. 1 : Testrig of parkinson Gear tester

Design Parameters &Dimentions

Design of Shaft :

The shaft may be designed on the basis of :

1) Strength 2) Rigidity

The following cases may be considered when shaft designing is on the strength basis:-

a. Shaft subjected to twisting movement or torque only.
b. Shaft subjected to bending moment only.
c. Shaft subjected to combined twisting & bending moment.
d. Shaft subjected to axial loading in addition to combined torsional & bending load.

Shaft Subjected To Combined Twisting Moment &Bending Moment

When the shaft is subjected to combined twisting and bending moment then the shaft must be designed on the basis of the two moments.

The following two theories are important from design point of view : 1) Max.Shear stress theory. It is used for ductile material such as M.S.

2) Max. Normal stress theory or Rankine's theory, it is used for brittle material such as C.I.

A. Twisting moment (T) may obtained by using the following relation.

$P = 2\Pi NT/60$

Where T = twisting moment

N= RPM of shaft

P = power transmitted in Watts

B.When the shaft is subjected to a twisting moment, then the d diameter of shaft is to be obtained by using torsion equation.

$$T/J = fs / r \qquad - (i)$$

Where, T = twisting moment acting upon the shaft

J = polar moment of inertia of shaft about the axis of relation.

Fs = Torsional shear stress

r = d/2

C) We know that for round solid shaft, polar moment of inertia

$$J = \Pi /32 \times d^4$$

The equation (i) may now be written as

$T/\Pi/32 \times d^4 = fs / r$

D) By using relation between driver and driven gear

$$N1/N2 = D2/D1$$

Where N1 & D1 = speed and diameter of shaft of the motor on driver gear.

Where N2 & D2 = speed and diameter of shaft of the motor on driven gear.

$$30 / N2 = (5 \times 25.4) / (1.5 \times 25.4)$$

$$N2 = 9 \text{ rpm}$$

E) Torque transmitted by shaft

$$T = P \times 60 / 2 \Pi N2$$
$$= 0.5 \times 1000 \times 60 / 2 \Pi 9$$
$$= 530$$

F) If instead of direct gear, we would have used pulleys then

$$T = (T1 - T2) \times r$$

Where T1 = tension on tight side of the belt

T2 = tension on slack side of belt on pulley

$$530 = (T1 - T2) \times 20$$

T1 – T2 = 26.5 Newtons

We assume that T1 /T2 = 2.2

$$T1 = 2.2\ T2$$

Therefore, T2 = 22.08 N & T_1 = 48.58 N

Now to calculate the diameter of shaft,

Take **Fs = 0.4 N/mm^2**

$d3 = T \times 16\ \pi\ fs$

$d3 = 530 \times 16 / \pi \times 0.4$

d = 10.65 mm

For factor of safety we have selected dia. more by 2 mm.

d = 12mm.

Design of Gear

Max rpm = 20

Material selected C45 with properties.

$[\sigma_b]$ = 1400 kgf/cm^2

= 140 N/mm^2

$[\sigma_c]$ = 5000kgf/cm^2

= 500 N/mm^2

As the application is a gear tester & we are designing master gear, we will design for wear & bending.

Designing for wear,

From (p.s.g) pg 8.13

We have,

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$$A >= (I + 1)\ \frac{0.74}{[6c]}\ \frac{E[Mt]}{.i\ \psi}$$

$$.i = \frac{60}{51} = 1.76 \text{ approx} = 1.2$$

let us take I = 5.5 as we design for max conditions & spur gear reduction is only possible to 5.5 calculation of torque power = 27W

$$P = \frac{2\Pi NT}{60}$$

$$27 = \frac{2*\Pi*20*T}{60}$$

Mt = 13.217
Now (from psgpg 8.15)
[Mt] = Mt k kd
= 3.217*1.5
assume K kd = 1.5 (from psgpg 8.15)
[Mt] = 19.736 N-m
= 19736 N-mm

assume $\psi = \frac{b}{a}$

$\psi = 0.060$
$\psi = 0.1$

as it is very low & it should be between 0.1 – 0.3 we take = 0.1
Now taking E = 2.1 *10^5 N/mm^2 (from psgpg 1.1)
For steel we have selected C 45 steel
Putting all these in above formulae We get,
a = m(Z 1 + Z 2)
= m(51 + 280)
a = 165.5m
putting all these in above formulae

$$165.5m = (5.5 + 1)^3 \left[\frac{(0.74)^2}{500} \times \frac{2.1 \times 10^5 \times 19736.37}{5.5 \times 0.1}\right]$$

165.5m = 6.5 *33.1 ; m = 1.3mm
let us design for bending
we have (from psgpg 8.13)

$$m >= 1.26^3\ \frac{[Mt]}{y\ [6b]\ ymZ1}$$

we have y = form factor,

From (psgpg 8.18) from table

For 51 teeth we have
Y = 0.477.
$\psi = 10$
Z 1 = 51
[6b] = 140 N/mm^2
[Mt] = 19.736 N-mm

$$m = 1.26\ ^3 \frac{[19.736 \times 10^3}{0.477 \times 140 \times 10 \times 51]}$$

= 1.05 mm
selecting higher module from wear considerations
m = 1.3 mm
selectingpreffered std. Module of 1.5 from (psgpg 8.2 table 1)

p.c.d. = mZ = 1.5*51 =76.5 ≈ 77

O.D. = p.c.d. + 2m

= 77 + 2*1.5

= 80 mm

I.D. = p.c.d. – 2.5m

= 73.25mm

b = 10m

= 10*1.5
= 15mm.

1) **Check for safe working of bolts**:-
Material of bolts -> plain carbon steel
Specification :- M8
Select permissible value of shear stress from data book.(PSG)
Fs(permissible) = 38N/mm^2

Considering shear failure of bolt

$$Fs = \frac{4 \times W}{\Pi\ dc^2}$$

Here dc = 6.72 mm. And W = 30 kg = 300N

$$Fs = \frac{4 \times 300}{3.14 \times (6.72)^2}$$

Fs= 8.462 N/mm^2.
This Fs is very less than Fs (permissible)

Therefore dimension of screw is safe.

Design of Springs:

<u>Statement-</u>
Spring required to with stand the pull of 20 kgf (max) = axial thrust.
G = 0.84 x 10^6 kg / cm².
Number of springs = 2
Coefficient of friction = μ =0.6
Fs = 4200 kg / cm² for spring steel.
To find the diameter of spring wire,
Torque = T = w x D/2
= 20 x 1.2 / 2
= 20 x 0.6 = 12 kg – cm.
Torque = 12 = (Π / 16) x 4200 x d³
d³ = 12 x 16 / (3.14 x 4200)

$d = \sqrt[3]{0.01455}$

d = 0.244 cm.
d = 2.4 mm
Hence we have taken the spring wire of diameter as 2.5 mm.
To find the number of turns;
Deflection of spring = $\delta = 8\ w\ D^3\ n / (d^4\ G)$
For 20 mm deflection calculate 'n' no of turns.

$2 = [\ 8 \times 20 \times 1.2^3 \times n\ ] / [\ (0.2)^4\ 0.84 \times 10]$
$n = [\ 2 \times (0.0256)\ 840 \times 10^3\ ] / [\ 8 \times 20 \times 1.2^3\ ]$
n = 43008 / 227.96 = 156 turns.
When two springs are installed
The no of turns will be = 156 / 2 = 78.

Design of The Bearing:-

He ball bearing are selected for radial load of 20 kg. during 90% of time & 80 kg load during remaining 10%. The shaft rotates at 1000 rpm. We have to determine the value of dynamic load rating for 5000hrs of operation with not more than 10% of failure
W1 = 20 kg

W2 = 80 kg

N = 1000 rpm

Therefore no. of revolution during 90% of time,

L1 = 0.9 x 1000 x 60 x 5000

$L1 = 270 \times 10^6$ min

Number of revolution during 10% of time

L2 = 0.1 x 1000 x 60 x 5000

$L2 = 30 \times 10^6$ min.

Basic dynamic load rating = C

$$C = \left(\frac{L1W1^3 + L2W2^3}{10^6} \right)^{1/3}$$

$$C = \left(\frac{(270 \times 10^6 \times 20^3) + (30 \times 10^6 \times 80^3)}{10^6} \right)^{1/3}$$

$C = [\ 2160000 + 15360000]^{1/3}$
$C = [17520000]^{1/3}$
C = 259.28 kg.
Now using reference table, for static load of 259.28 kg bearing no 201 is used. SKF 6201

Design of Roller Shaft :-

The shaft carries the roller along with the pad belt. It is supported at its ends. It transmits ¼ h.p. at 60 rpm. The tension on tight side is 20 kg and on that of slack side is 10 kg. Hence we have to calculate suitable diameter.
Material :- M.S. fs= 420kg/cm^2 diameter of roller = 28 mm
So, torque acting on the shaft = force x perpendicular distance
T = (20 –10) x 2.8/2 = 18.0 kg-cm.
Also bending moment at point where belt is resting
M = 10 x 5.0 = 50 kg-cm
To find the equivalent torque = $Te = \sqrt{T2 + M2}$
Therefore $Te = \sqrt{18^2 + 50^2} = 53.14$
T find the diameter of the shaft, $Te = \Pi / 16\ fs\ d^3$
$Te = 3.14 / 16 \times 420 \times d^3$
$d^3 = 53.14 \times 16 / (3.14 \times 420) = 0.644$
Therefore d = 0.866 cm = 9mm.
Hence for the factor of safety we have selected the shaft diameter of 12 mm.

Design Of Welded Joint :-

A 25 x 25 x 5 mm angle is to be welded to by fillet weld . The angle is subjected to 30 kg load maximum

The allowable shearstrss for static loading = fs = 750 kg/cm^2
Let total length of the weld = l = la + lb
We know that for a single parallel fillet weld,
$P = (t \times l / \sqrt{2}) \times fs$
$L = \sqrt{2}\ P / (t \times fs)$
$L = \sqrt{2} \times 30 / (0.5 \times 750)$
L = 0.113 cm.
L = la + lb = 0.113
Now let us find the distance of centroidal axis from the bottom of the angle.

$$B = \frac{(2.5-0.5) \times 0.5 \times 2 + 2.5 \times 0.5 \times 0.25}{2.0 + 2.5}$$

= 2.31 / 4.5 = 0.51 cm.
A = 2.5- 0.51 = 1.99 cm.
La = l x b/ a + b
La = 0.113 x 0.51/(1.99 + 0.51) = 0.023 cm.
Lb = l – la = 0.113 – 0.023 = 0.0899 cm.

Result Analysis

Fig. 2 : Variation In Flank Surface For Model 1

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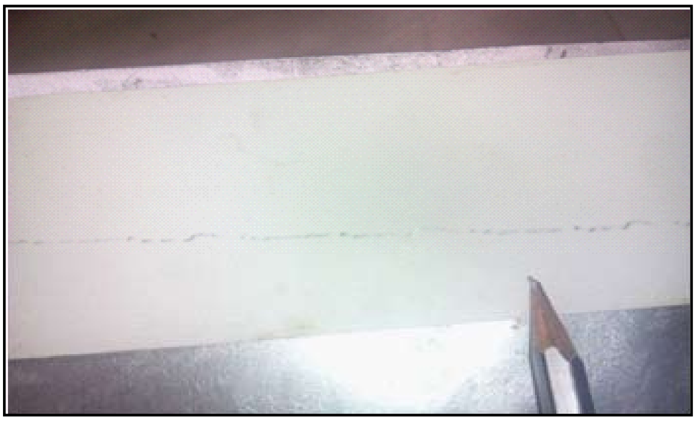
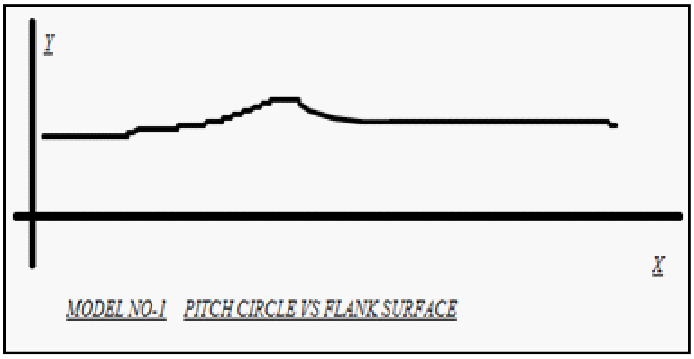
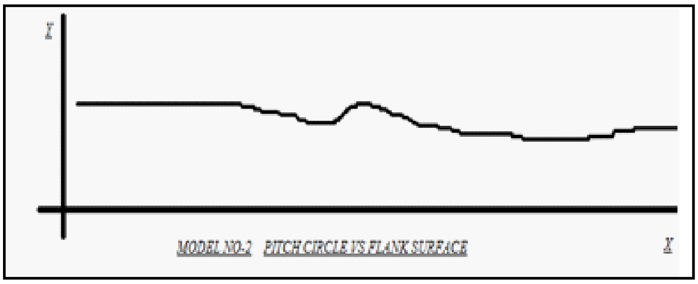


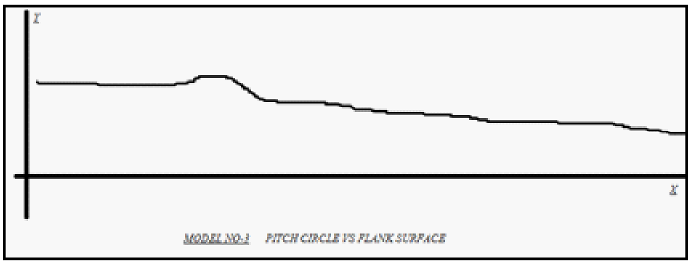
Fig. 3 : Variation In Flank Surface For Model 2



a)



b)



c)

Fig. 4 : Variation In flank Surface for different models

Conclusion

While concluding this part, we fill quite contended in having completed the project assignment well on time. We had enormous practical experience on the manufacturing schedules of the working project model. We are therefore, happy to state that the inculcation of mechanical aptitude proved to be a very useful purpose. We are as such overwhelmingly elated in the arriving at the targeted mission.

Undoubtedly the joint venture has had all the merits of interest and zeal shown by all of us the credit goes to the healthy co-ordination of our batch colleague in bringing out a resourceful fulfillment of our assignment described by the university.

The design criterion imposed challenging problems which however were welcome by us due to availability of good reference books. The selection of choice of raw materials helped us in machining of the various components to very close tolerances and thereby minimizing the level of wear and tear.

In this paper, we developed a branch and bound approach which is coupled with quick, gear testing in mass production requirement within a manufacturing cell.

The design of control architecture was an important aspect of study because a strong interaction between the many different parts was needed. We are testing gear with low running cost.. So we are satisfied with our project.

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