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Research Paper

ESTIMATION OF SPUR GEAR TOOTH VIBRATION

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ABSTRACT

Through this paper an idea is proposed to estimate the vibration response of gear tooth during a period of contact. Hence a case study of spur gear pair is taken into consideration. The tooth force is obtained during the period of contact, which is then useful to estimate the vibration response of a gear tooth. A MATLAB program is prepared for this analysis.

It is seen that during a period of contact of a gear tooth, the tooth force varies with time. The time variant tooth force causes a tooth vibration during a period of contact. Hence the objective of a present work is to estimate the vibration response of a gear tooth during a period of contact, Firstly; a case study is taken of a gear pair considering power and speed. The tooth force is then estimated during the period of contact. The time period taken for one period of contact is also estimated.

The estimated force and time is then useful to draw a force pattern. Lastly, the tooth is considered as single degree of freedom system for estimation of vibration response. The force pattern is now useful for harmonic analysis. This harmonic analysis helps to estimate complete vibration response.

Keywords: Period of contact, Vibration response, Detoriation, harmonic analysis, MATLAB software.

1.General Discussion about Gear Vibration –

1.1. Effect of Vibration on the Performance of Gear Train

Vibrations of mechanical equipment is generally not good particularly in gears[4]. The excessive vibration in gear train causes wear of flanks of a gear tooth. This wear of gear tooth may leads to additional unanticipated load, which will be added in the tooth load. Sometimes, excessive tooth load may create additional bearing reactions. These bearing reactions are time variant in nature, which is the main source of increase in vibrations. In fact, due to time variant force systems the internal bearing parts may be detoriated over a period of time. This detoriation provides some unbalance force in the bearing shaft. In all these conditions affect the performance of gear, when it is in rotation.

1.2.Sources of Gear Vibration:

Gear blank, Misalignment Due To Bearings, Vibration of gear due to bad condition of bearings, Looseness, Porosity in casting, Non uniform density of material. Manufacturing tolerances, Machining, Maintenance actions like changing bearings or cleaning, etc, Couplings & Rotational mass distribution etc.

2.An Approach for Estimation of Gear Vibration during a Period of Contact

In previous article, the effect of vibration of a gear on its performance and so many sources of gear vibration are discussed[8]. Hence through this paper an attempt is being made towards estimation of gear vibration response. In view of this a case study of two gears is chosen for this work.

A special approach is being applied for this case study. The vibration response is calculated only for the period of contact, for this considering a torque, applied at the shaft of a gear, the tooth force is estimated. The variation of this force over a period of contact is then estimated by certain adequate steps. The estimated force over a period of contact is plotted against time. This forcing function is used for finding out the vibration response of a tooth, which is done by harmonic analysis.

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3.Estimation of Vibration Response of a Gear Tooth for one period of Contact

3.1.A Case Study of a Gear Train

In the present work, a gear train of following specification is considered, which is detailed below.

Two shafts with their axes parallel to each other at 60cm apart are connected by spur gears, which are shown in Figure 1. The driver speed is 400rpm. Speed reduction ratio is 2:1. Teeth have involutes profile with module 20mm; addendum is equal to module. Pressure angle is 200. The gears transmit 35HP power.

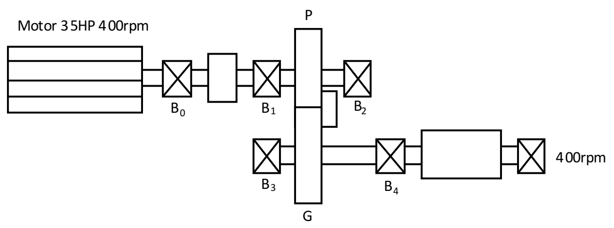

Figure 1 schematics of gear train transmitting power from a motor to process machine

Q Estimation of gear tooth load during one period of contact

In this section, the first step is to calculate circular pitch diameter (CPD) the calculation of CPD is discussed below –

Q Estimation of circular pitch diameter

As the speed reduction ratio is 2:1, from which one can estimate the value of O2 as 20cm. and the value of O3 as 40 cm., which forms pitch circle point 'P'.

In above discussion O2, O3 are the centre of pinion and gear.

Where D2 is pitch circle diameter of pinion and D1 is pitch circle diameter of gear.

$$D_2 = 2D_1$$

As centre distance between two shaft is 60cm.

Hence,

O2 + O3 = 60 … … … … 2

$$\frac{O_3}{O_2} = 2$$

O3 = 2O2 … … … … 3

By substituting the value of equation 3 in equation 2 one can get

O2 + O3 = 60

O2 + 2O2 = 60 … … … … 4

3O2 = 60

Thus,

= 20cm.

Hence, the centre O2 of pinion 'P' is 20cm. from the pitch point 'P'

$$\therefore O_3 = 2O_2$$

$$O_3 = 2 \times 20$$

Thus the centre O3 of gear 'G' is 40cm. from the pitch point 'P'.

Q Geometric construction of gear tooth profile

In this section construction of tooth profile and wheel geometrical construction are discussed.

First the point O2 is marked then the pitch circle point 'P' is marked at a distance of 20cm from point O2. Now the point O3 is marked at a distance of 40cm, from the pitch circle point 'P' kindly refer figure number 2.

Then one can find out the addendum circle for the driver gear i.e. distance O2P + module and addendum circle for driven gear, i.e. distance O3P+Module. Thus, one can estimate the pitch circle diameter (PCD) i.e. module multiplied by distance between the addendum circle and base circle.

Now, estimation of the number of teeth on pinion and number of teeth gear are done. Conversely the circular pitch PC is also calculated.

Angle of 200 is marked from the point O2, on the base circle. At the same time, angle of 200 is marked from the point O3 on another side.

Value for addendum circle of driver gear is calculated by adding the module in the distance of O2 and the value of addendum circle of driven gear is calculated by adding the module in the distance of O3. One will get the values of driver and driven wheels.

Addendum circle (driver)

= O2 + m

= 20 + 2 = 22 cm

Addendum circle (driven)

= O3 + m

= 40 + 2 = 42 cm

The point D (pitch circle diameter) is marked on the addendum circle by using the formula D = PCD = Module × distance between addendum circle and base circle, so

D = PCD = Module × distance between addendum circle and base circle

= 20 × 15

= 300 = 150 (half)

Then, the number of teeth on pinion will be calculated by using the below stated formula -

The teeth on pinion can be estimated as -

$$T_p = \frac{2A}{\left(\sqrt{1 + G(G+2)\sin ...}\right.}$$

$$T_p = 10.22 \approx 11 \text{ teeths}$$

Where A—Addendum, G—Gear, Pressure angle.

As the no. of teeths on pinion are 11, the no. of teeths on

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gear can be calculated by multiplying the number of teeth on pinion with velocity ratio.
So teeth on Gear
= V.R. × number of teeth of pinion
= 2 × 11 = 22 teeth
Now Circular pitch P C
P C = = 2 × 3.14
PC = 6.28cm … … … … 6

$$\therefore \frac{PC}{CPD} = \frac{6.28}{15} = 0.418\ rad$$

$$= 0.418 \times$$

= 23.980
Now by assuming efficiency to be 90% one can calculate the power
Power P =

$$\frac{35}{0.9} = \frac{2\times\pi\times400\times T}{4500}$$

$$\therefore T = 69.63 kgf m$$

$$= 69.63\times10^2 kgf\,cm$$

$$T = 6963$$

By using the torque one can estimate the force on the gear tooth by putting the value of torque 'T' in the following relation.

$$F = \frac{T}{O_2}$$

$$\therefore F = \frac{6963}{0.020}$$

F = 348150
F = 3481.50 kgf … … … … 8
From the above force the value of F1 is calculated

$$\frac{x}{o_2c^1}$$

F1 = 370.372 kgf … … … … 9
Now, from the Figure 2 major the distance between C and C' thus,
CC1 =18mm … … … … 10
And the distance
O2C' =188mm … … … … 11
Now by using the relation

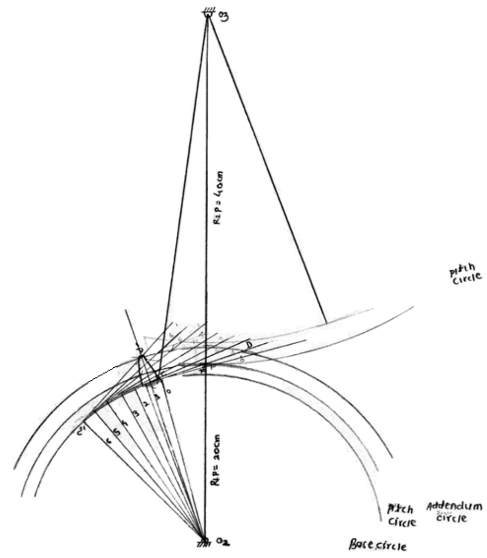
$$\frac{CC^1}{O_2C^1} = \frac{18}{188}$$

$$= 0.095\ rad$$

$$= 0.095\times\frac{180}{\pi} = 5.443^0$$

The tangent is drawn from the point C and extend that upto the addendum circle then mark the point D on the addendum circle then measure the distance CPD.

$$Scale: 1cm = 2cm$$



For this one can select the scale as follows –

$$CPD\ actual = 9.6cm$$

$$angle\ \theta^1 = \frac{CPD}{O2C^1} = \frac{9.6}{18.8} = 0.51 rad$$

$$= 0.510\times\frac{180}{\pi} = 29.25^0$$

Then mark the angle, which will come on the base circle and label it as C" then distribute that angle into six equal part and draw the tangent from each point.

$$Angle\ \theta^1 = \frac{DPCC^1}{O_2C^1} = \frac{11.4}{18.8}$$

$$= 0.606\ rad$$

$$= 0.606\times\frac{180}{\pi} = 34.73^0$$

Now again calculate circular pitch

$$= \frac{\pi\times2O_2}{no.\ of\ teeth}$$

$$= \frac{\pi\times2\times40}{22} = 11.42mm$$

Thus C D'C' forms a tooth profile in Figure 2.
The estimated tooth force for the position of six different angles, which is shown in Table 1. The corresponding tooth force pattern is shown in Figure 3.
Conversely, using the equivalent force, which is coming on the tooth a moment is estimated. This moment is calculated by multiplying the force with the length of link arm moment.
Total angle is of 34.73 is divided into six parts and each angle is of 1.64o alterwords, the equivalent force is calculated, which is shown in Table 2.

Table - 1
Tooth force for different angles of a gear pair.

Angle (	1.64	3.28	4.92	6.56	8.20	9.84
Force kgf	185.18	185.18	370.37	370.37	185.18	185.18

Table - 2
Equivalent force, moment, force, length for different angle of a gear pair.

Sr. No.	Force (kgf)	Length (mm)	Moment (kgfmm)	Equivalent Force at tip (kgf)
1	185.18	1.0mm on drg = 2mm on real	370.36	12.340
2	185.18	2.0mm on drg = 4mm on real	740.72	24.690
3	370.37	3.0mm on drg = 6mm on real	2222.22	74.070
4	370.37	3.5mm on drg = 7mm on real	2592.59	86.410
5	370.37	6.0mm on drg = 12mm on real	4444.44	148.148
6	185.18	7.5mm on drg = 15mm on real	2777.77	92.590
7	185.18	13mm on drg = 26mm on real	4814.68	160.480

Time required for one revolution is calculated and for each angle the time is calculated as –
As
N = 400rpm
400rpm=60sec.

For

$$1.64^{o} = \frac{0.15\times1.64}{360} = 0.000$$

$$3.28^{o} = \frac{0.15\times3.28}{360} = 0.000$$

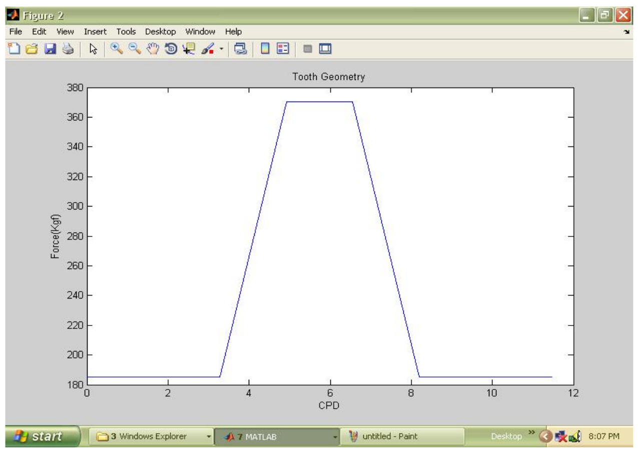
$$4.92^{o} = \frac{0.15\times4.92}{360} = 0.0$$

$$6.25^{o} = \frac{0.15\times6.25}{360} = 0.0$$

$$8.24^{o} = \frac{0.15\times8.24}{360} = 0.0$$

$$9.84^{o} = \frac{0.15\times9.84}{360} = 0.0$$

By using above time the force pattern is estimated, which is shown in Figure 2.



The estimated force pattern is now useful for the estimation of vibration response of a gear tooth. Hence, in view of this, the total tooth is considered as single mass, single stiffness and single damper unit[9]. The mass of gear tooth is designated as m, while stiffness is designated as K, and damper is designated as C. The derivation of governing equations for such a system is detailed[10].
The complete steady state solution is given by

$$x_p(t) = \frac{a_0}{2k} + \sum_{j=1}^{\infty} \frac{aj/k}{\sqrt{[1-j^2r^2]^2 + (2\xi ir)^2}} \cos(jw - \phi_j) +$$
$$\sum_{j=1}^{\infty} \frac{aj/k}{\sqrt{[1-j^2r^2]^2 + (2\xi ir)^2}} \sin(jwt - \phi_j)$$

PROGRAM OF MATLAB

```
%Estimation on spur gear tooth vibration
clc
clear all
close all

theta = 20;
module= 20;%mm
N=input('Enter the value of Rotation/min (rpm): ');
%rpm
Power=input('Enter power (Hp): ');
tp=(2*2)/(1+2*(2+2)*(sin(theta))^2-1)^0.5;
tp=ceil(tp);
disp('Tooth on pinion = ')
tp
disp('Tooth on Gear = ')
tp*2
disp('Circular Pitch (cm) = ')
module*3.14/10

disp('Ratio PC/PCD (Degree)= ')
180*module*3.14/10*15*3.14

disp('Torque in(Kgfcm)= ')
Power*4500/(0.9*2*3.14*400)

disp('Force at O2(Kgf)= ')
(Power*4500/(0.9*2*3.14*400))/0.020

disp('Force at O2_C(Kgf)= ')
```

```
F1=100*(Power*4500/(0.9*2*3.14*400))/18.8

disp('Theta_CPD (Degree)= ')
(9.6/18.8)*180/3.14

disp('Theta_DPC_C (Degree)= ')
(11.4/18.8)*180/3.14

disp('Circular Pitch (mm)= ')
3.14*2*40/22

angle=[0 1.64 3.28 4.92 6.56 8.20 9.84 11.48];
force=[F1/2,F1/2,F1/2,F1,F1,F1/2,F1/2,F1/2];

figure(1)
subplot(2,3,1)
plot(angle,force)
title('Tooth Geometry')
xlabel('CPD')
ylabel('Force(Kgf)')
%%%%% Part2
subplot(2,3,2)
stem(angle,force)
hold
plot(angle,force,'-r')
title('Tooth Geometry')
xlabel('CPD')
ylabel('Force(Kgf)')

%should be edited with formula(relation)
angle_1=[1.64 3.28 4.92 6.56 8.20 9.84 11.48];
Length=[2,4,6,7,12,15,26];
force_1=[F1/2,F1/2,F1,F1,F1,F1/2,F1/2];
Moment=zeros(1,7);
for i=1:1:7
   Moment(i)=Length(i)*force_1(i);
end
subplot(2,3,3)
plot(angle_1,Moment)
title('Tooth Profile')
xlabel('Angle(theta)')
ylabel('Moment(Kgfmm)')
%%%%%Part3
disp('One Revolution time(sec)= ')
R1=60/N
Time=zeros(1,7);
Equforce=zeros(1,7);
for i=1:1:7
   Time(i)=angle(i)*R1/360;
   Equforce(i)=Moment(i)/30;
end

subplot(2,3,4)
plot(Time,Equforce)
title('Vibration Response')
xlabel('Time(sec)')
ylabel('Equivalent Force(Kgf)')

subplot(2,3,[5 6])
eqF=fft(Equforce,7);
eqF=fftshift(eqF);
Mag_eqF=abs(eqF);
plot(Time,Mag_eqF)
title('Vibration frequency Response')
xlabel('Time(sec)')
ylabel('F(t)')

%%%%%%%%%%%%%%%%%%%%%%
%%individual Plots(graphs)
figure(2)
plot(angle,force)
title('Tooth Geometry')
xlabel('CPD')
ylabel('Force(Kgf)')

figure(3)
stem(angle,force)
hold
plot(angle,force,'-r')
title('Tooth Geometry')
xlabel('CPD')
ylabel('Force(Kgf)')
figure(4)
plot(angle_1,Moment)
title('Tooth Profile')
xlabel('Angle(theta)')
ylabel('Moment(Kgfmm)')

figure(5)
plot(Time,Equforce)
title('Vibration Response')
xlabel('Time(sec)')
ylabel('Equivalent Force(Kgf)')

figure(6)
plot(Time,Mag_eqF)
title('Vibration frequency Response')
xlabel('Time(sec)')
ylabel('F(t)')
```

RESPONSE OF PROGRAM FOR FOLLOWING PARAMETERS

1) Enter the value of Rotation/min (rpm): 400
2) Enter power (Hp): 35

```
Tooth on pinion =
tp =  2
Tooth on Gear =
ans =  4
Circular Pitch (cm) =
ans =   6.2800
Ratio PC/PCD (Degree)=
ans =  5.3242e+004
Torque in(Kgfcm)=
ans =   69.6656
Force at O2(Kgf)=
ans =  3.4833e+003
Force at O2_C(Kgf)=
F1 =  370.5617
Theta_CPD (Degree)=
ans =  29.2723
Theta_DPC_C (Degree)=
```

ans = 34.7608
Circular Pitch (mm)=
ans = 11.4182
Current plot held
One Revolution time(sec)=
R1 = 0.1500

1. Discussion About the Vibration Response Obtained for a Period of Contact.

A vibration response is re-produced here for the sake of explanation which is shown in Figure 4. Now, the discussion is descritised into different stages.

Phase A-B: in this phase, the force on gear tooth changes from 12.34 kgf. to 24.69 kgf., while the displacement value at point 'A' is 0.00218mm and the displacement value at point 'B' is 2.165 × 10-3mm. During this phase, the gear tooth complete the time of 0.68 × 10-3sec. In this section the increment in displacement is seen.

Phase B-C: In this phase, 'C' is the third position point in a graph where the force is 74.70kgf. and the amplitude is 2.176 × 10-3mm. To reach the point 'C' the tooth completes a time 1.36 × 10-3sec. This section shows the increment in force value from point B to C as well as the amplitudes increases at these point.

Phase C-D: In this phase, 'D' is the fourth position point in a graph where the applied force at the tooth is 86.410kgf. The amplitude at this point is obtained as 2.178 × 10-3mm. it is seen that, during this phase C-D, the force has changed conversely the amplitude also changes.

Phase D-E: In this phase, 'E' is the fifth position point in a graph. The force and amplitude at this point are 148.148kgf. and 2.160 × 10-3mm. it is cognised that, although there is an increment in force, the amplitude has reduced.

Phase E-F: In this phase, 'F' is the last and sixth position point in a graph. At this point the force and amplitude are 92.59kgf. and 2.135×10-3 mm. the force is now changed from its maximum value to minimum value. Corresponding reduction in the amplitude is seen.

If one refers a vibration response of a gear tooth for a period of contact which is shown in Figure 4, then he finds the maximum amplitude is at point D, whose magnitude is 2.178 ×10-3mm. the force at this stage is 86.410kgf. It is observed that from point D the displacement graph falls downward. Whereas the minimum displacement is obtained at point A which has a value of minimum displacement is 2.118×10-3mm.

Response of program for Speed = 400rpm & Power = 35HP

Table – 3 Showing force, time & amplitude

Position	Force in KGF	Time (t) in second	Amplitude (mm)
A	12.340	0.68×10-3	2.118×10-3
B	24.690	1.36×10-3	2.165×10-3
C	74.070	2×10-3	2.176×10-3
D	86.410	2.6×10-3	2.178×10-3
E	148.148	3.4×10-3	2.160×10-3
F	92.590	4.1×10-3	2.135×10-3

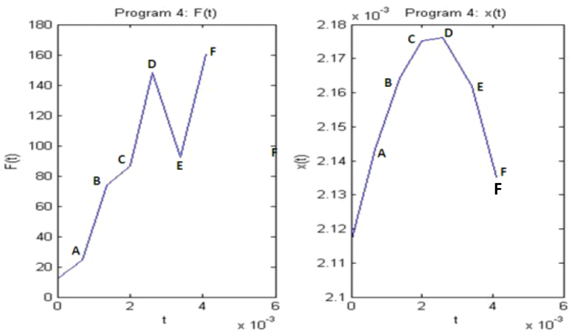


Figure 4 Schematics of equivalent force variation of a tooth load and vibration response of a tooth

1. Discussion about the Vibration Response Obtained for Different Speed of Gears

In this section the response of gear tooth for various speed are obtained. The power of a system keeps constant for this analysis. The intensive of this section is to see the variation in amplitude with regard to the change in speed of a system. If one refers the Figure 5 and 6 of the vibration response of a gear tooth for different speed of a system, then he finds that the maximum amplitude is arrived at common point 'C' only.

Response of program for Speed = 500rpm & Power = 35HP

Table - 4 Showing Force, Time & Amplitude

Position	Force in KGF	Time (t) in second	Amplitude (mm)
A	17.23	0.63×10^{-3}	2.487×10^{-3}
B	74.22	0.9×10^{-3}	2.498×10^{-3}
C	82.30	2.1×10^{-3}	2.507×10^{-3}
D	149.44	1.2×10^{-3}	2.505×10^{-3}
E	90.00	2.5×10^{-3}	2.493×10^{-3}
F	162.23	2.7×10^{-3}	2.482×10^{-3}

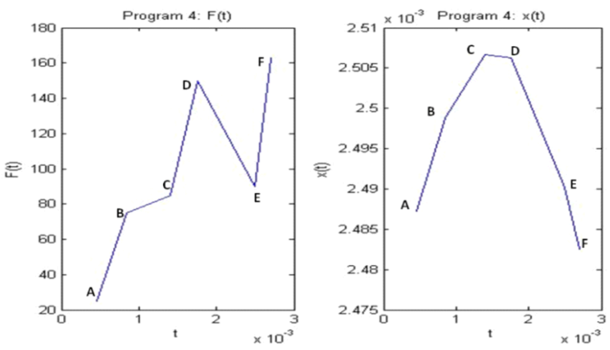


Figure 5 Schematic of Equivalent force variation of a tooth for 35HP, 500rpm.

Response of program for Speed = 600rpm & Power = 35HP

Table 5 Showing Force, Time & Amplitude

Position	Force in KGF	Time (t) in second	Amplitude (mm)
A	25	0.45×10^{-3}	2.488×10^{-3}
B	75	0.85×10^{-3}	2.498×10^{-3}
C	85	1.4×10^{-3}	2.507×10^{-3}
D	150	1.7×10^{-3}	2.505×10^{-3}
E	90	2.5×10^{-3}	2.487×10^{-3}
F	163	2.7×10^{-3}	2.482×10^{-3}

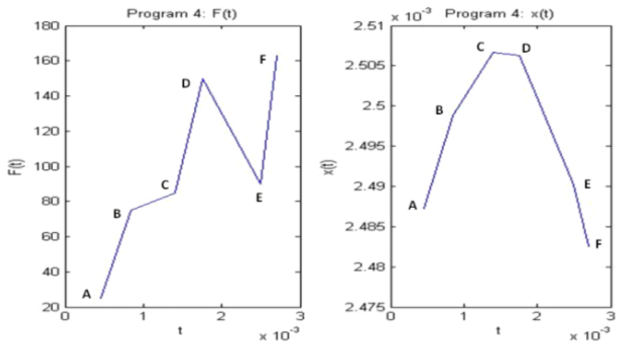


Figure 6 Schematic of Equivalent force variation of a tooth for 35HP, 600rpm.

CONCLUSION:

Through this present investigation following some important conclusions are made, which are discussed below -

1. Considering the present case study, the equivalent tooth force is estimated for one period of contact. This tooth force is then plotted against the time. The pattern of this equivalent tooth force shows non-linearity in nature.
2. Using a force pattern, a vibration response is estimated for a gear tooth. In this vibration response, the maximum amplitude is obtained as 2.178×10-3mm, which is at time 2.6 × 10-3sec. At this point the force is obtained as 86.410kgf.
3. The different vibrations responses are obtained for different speed of gears, keeping power as constant. These vibration responses, shows a common point 'C' at, which the maximum amplitude is obtained.

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D

E

F