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Design of Machine Shaft in Fatigue Loading by Using C++ Programming Language

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Abstract

In the case of power transmission sometimes a fluctuating or repetitive load comes on drive shaft or driven shaft in working condition. At that time shaft as well as gear or pulley which is mounted on the gear shaft also comes under the fatigue loading. To determine the shaft diameter under dynamic condition we should consider length of shaft, diameter of pulley and gear, tension on pulley, rpm of shaft, distance between gear and pulley, pulley and bearing on which shaft is mounted, permissible stress of shaft material and fatigue load. The main purpose of writing of this paper is to determine shaft diameter with different material and under the different permissible stress.

Keywords: *Shaft, fatigue loading, Shear Stress, tangential force*

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INTRODUCTION

Types of Shaft

Shaft can be classified mainly in to two parts.

a.) Transmission shaft
b.) Machine Shaft

- Transmission shaft

The shaft which transmits power between one source and the rotation machine element is called transmission shaft (Figure 1).

Fig. 1: *Transmission Shaft.*

- Machine shaft

The shaft, an integral part of any machine assembly is called the machine shaft (Figure 2).

Fig. 2: *Machine Shaft.*

Stress in the Shaft

In actual practice mainly three kinds of stress are induced in it.

a.) Shear stress – due to the transmission of the torque.
b.) Bending stress – due to force acting upon machine element, and weight of shaft itself.
c.) Stress due to combined torsional and bending loads.

Maximum Permissible Working Stress for Transmission Shaft

According to American Society of Mechanical Engineers (ASME) code for the design of transmission shaft, the maximum permissible working stress in tension or compression may be taken as;

- For shaft purchased under definite physical specification, the permissible tensile stress (σ_t) may be taken as 60% of the elastic limit in tension (σ_{el}), but not more than 36% of the ultimate tensile strength (σ_u) in short the permissible tensile stress.
- σ_t= 0.6 σ_{el}or 0.36 σ_u, whichever is less.
- For shaft purchased under definite physical specification, the permissible shear stress (τ) may be taken as 30% of the elastic limit in tension (σ_{el}), but not more than 18% of the ultimate tensile strength (σ_u) in short the permissible shear stress.
- τ =0.36 σ_{el}or0.18$\sigma_u$whichever is less.

In many cases; a machine shaft contains a multiple machine component like; gear, pulley, belt, etc. In our case we consider a shaft which contains a bearing at both end, and a pulley and gear mounted on the shaft at specific distance [1–3].

In steady condition, shaft is subjected in bending only. But in actual case when shaft is rotating or working with this kind of component it will be subjected in combined stress because of rotational motion, weight of pulley and gear and in tension due to power transmission through belt. This rotating shaft comes under fluctuating loading condition while it will run. In short shaft will come under the fatigue loading. Our aim to write this paper is to find a suitable diameter of shaft for given specific stress limit, or fatigue limit for which shaft will work without failure.

THEORY

Maximum Shear Stress Theory

According to this theory, the failure occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a sample tension test [4].

- τ_{max}=τ_{yt} / FS

τ_{max} = Maximum shear stress in system
τ_{yt}= Shear stress at yield point
FS = Factor of safety

Normal Stress Theory

According to this theory, the failure occurs at a point in a member when the maximum principal or normal stress in a bi-axial system reaches the limiting strength of the material in simple tension test [5].

- $\sigma_{t1}= \frac{\sigma yt}{FS}$for ductile material

- $\sigma_{t1}= \frac{\sigma u}{FS}$for brittle material

Design of Shaft

The shafts are designed on the basis of strength, rigidity and stiffness.

In design of shaft base on strength following main cases are considered:

a.) Shaft subjected to twisting moment only.
b.) Shaft subjected to bending moment only.
c.) Shaft subjected to combined twisting and bending moment.
d.) Shaft subjected to axial loads in addition to combined torsional and bending loads.

ANALYSIS OF PROBLEM

We will use basic two kinds of theory to solve and analyse the problem

1. Maximum shear stress theory
2. Maximum normal stress theory

To find a suitable shaft diameter for given working condition.

Problem definition

We consider a shaft supported on bearings A and B, 1000 mm between centers. A 20° tooth straight tooth spur gear having 400 mm pitch diameter is located 200 mm to the right of the left hand bearing A and a 600 mm diameter pulley is mounted 300 mm towards the left bearing B. The gear is driven by a pinion with a downward tangential force while the pulley drives a horizontal belt having 180° angle of wrap. The pulley also serves as a fly wheel and wait 6000N the maximum belt tension is 6000N and the tension ratio is 3:1. Determine the maximum bending moment and the necessary shaft diameter if allowable shear stress of the material is 60 MPa (Figure 3).

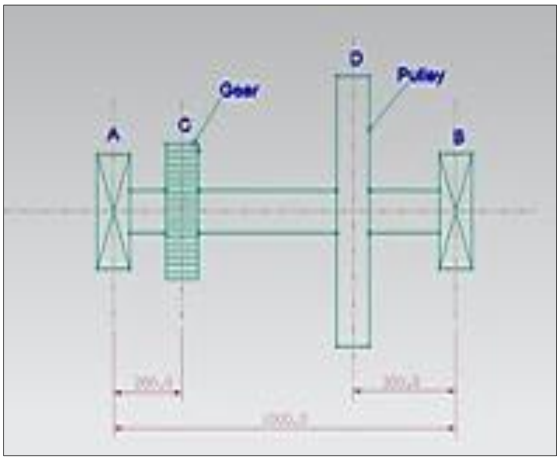

Fig. 3: *Maximum Bending Moment.*

DESIGN PROCEDURE

- Torque on the shaft

$T = (T_1 - T_2) \cdot R_D$ (1)

$T = 1200 \times 10^3$ N·mm

- Assume torque at D and C is same so tangential force on gear C

$F_{tc} = \frac{T}{Rc}$ (2)

$F_{tc} = 6000$N

- Normal load on the gear

$W_c = \frac{Ftc}{cos\alpha c}$ (3)

$W_c = 6385.06$ N

- Resolving the force

Vertical component of W_c at C

$W_{cv} = W_c \cos 20°$ (4)

$W_{cv} = 6000$ N

Horizontal component of W_c at C

$W_{ch} = W_c \sin 20°$ (5)

$W_{ch} = 2183.81$ N

- Tension ratio 3:1

$\frac{T1}{T2} = 3$ and $T_1 = 6000$ N (6)

$T_2 = 2000$N

- Vertical Load at D

$W_{dv} = W$ (7)

$W_{dv} = 6000$ N

- Horizontal Load at D

$W_{dh} = T_1 + T_2$ (8)

$W_{dh} = 6000 + 2000$

- Let's consider reaction at both bearing (Vertical Loading)

$R_{AV} + R_{BV} = W_{CV} + W_{DV}$ (9)

$R_{AV} + R_{BV} = 12000$ N

Take moment about A so

$R_{BV} = 5400$ N

$R_{AV} = 6600$ N (10)

- Bending moment at A and B

$M_{AV} = M_{BV} = 0$

- Bending moment at C

$M_{CV} = R_{AV} x\ 200$ (11)

$M_{CV} = 1320 x 10^3$N·mm

- Bending moment at D

$M_{DV} = R_{BV} x300$ (12)

$M_{DV} = 1620 X 10^3$N·mm

- Let's consider reaction at both bearing (Horizontal Loading)

$R_{AH} + R_{BH} = W_{CH} + W_{DH}$ (13)

$R_{AH} + R_{Bh} = 10183$ N

Take moment about A so

$R_{BH} = 6036.6$ N

$R_{AH} = 4146.4$ N

- Bending moment at A and B

$M_{AH} = M_{BH} = 0$

- Bending moment at C

$M_{CH} = R_{AH} x\ 200$ (14)

$M_{CH} = 829280$ N·mm

- Bending moment at D

$M_{DH} = R_{BH} x300$ (15)

$M_{DH} = 1810980$N·mm

- Resultant Bending Moment

$M_C = \sqrt{(Mcv)^2} + \sqrt{(Mch)^2}$ (16)

$M_C = 1558879.50$ N·mm

$M_D = \sqrt{(Mdv)^2} + \sqrt{(Mdh)^2}$ (17)

$M_D = 2429765.175$ N·mm

- Maximum bending moment

$M_D > M_C$

$M_{max} = M_D = 2429765.175$ N·mm

- K_m = combine shock and fatigue factor for bending

- K_t = combine shock and fatigue factor for torsional

K_m = 1.5 to 2 (From Appendix I)
K_t = 1.5 to 2 (From Appendix I)

- Equivalent Twisting moment

$T_e = \sqrt{(Km * M)^2} + \sqrt{(Kt * T)^2}$
T_e= 5419874.008N·mm (18)

- $T_e = \frac{\pi}{16} * 60 * D^3$ (19)

D = 80 mm (under twisting moment)

- Equivalent Bending moment

$M_e = \frac{1}{2}[Km * M + \sqrt{(Km * M)^2} + \sqrt{(Kt * T)^2}]$
M_e = 513672.174 N·mm (20)

- $M_e = \frac{\pi}{32} * \sigma b * d^3$ (21)

D = 90 mm (under bending moment)

- $D_{Bending} > D_{Twisting}$

- **Shaft Diameter is $D_{Bending}$= 90 mm**

RESULTS

Experimental method showed that 90 mm is the shaft diameter. We should take diameter of shaft as maximum from its bending moment as well as twisting moment for better safety.

CONCLUSION

From this experimental design calculation results, we can conclude that diameter of shaft will be decided on the basis of stress induced in it at the time of fatigue loading. With the help of C++ programming language; design of shaft with different load condition is very easy and less time consuming.

APENDIX I

NOMENCLATURE

AB = Distance between two bearings
AC = Distance between bearing and gear
CD = Distance between gear and pulley
DB = Distance between pulley and bearing
Rc = Radius of gear
Rd = Radius of pulley
Ɵ = 180°
W = Weight of pulley

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T = Tension on belt
τ = allowable shear stress
N = shaft speed (rpm)

APENDIX II

Nature of Load		**K_m**	**K_t**
1.	*Stationary shaft*		
-	Gradually applied load	1.0	1.0
-	Suddenly applied load	1.5 to 2	1.5 to 2
2.	*Rotating shaft*		
-	Gradually applied load	1.5	1.0
-	Suddenly applied load with minor shocks only	1.5 to 2	1.5 to 2
-	Suddenly applied load with minor shocks	2 to 3	1.5 to 2

APENDIX III

C++ Program for designing shaft by combined loading:

```
//Program for designing shaft by combined loading
#include<iostream.h>
#include<conio.h>
#include<stdio.h>
#include<math.h>

void main()
{
        clrscr();
        long int d1,d2,T1,T2,L,Fs,L1,L2,L3,W,N;
        double  Dg,Dp,T,M,Te,Me,A,B,C,D,E,F,G,H;
         float M1,T11,TR,Pow,Ftg,Wg,Wgv,Wgh,Wp,Wpv,Wph,Te1,Rav,Rah,Rbv,Rbh,Rph,Mp
,Mg,Mgh,Mph,Mgv,Mpv,Mg1,Mp1,Km,Kt;
        cout<<"\n Distance between two shaft bearing L(mm)=";
        cin>>L;
        cout<<"\n Distance between bearing and gear L1(mm)= ";
        cin>>L1;
        cout<<"\n Distance between gear and pulley L2(mm)=";
        cin>>L2;
        cout<<"\n Distance between pulley and bearing L3(mm)=";
        cin>>L3;
        cout<<"\n Diameter of pulley Dp(mm)=";
        cin>>Dp;
        cout<<"\n Diameter of gear Dg(mm)=";
        cin>>Dg;
        cout<<"\n load W(Newton)=";
        cin>>W;
        cout<<"\n Tension Ratio TR(Newton)= ";
        cin>>TR;
        cout<<"\n For shaft material Permissible stress Fs(MPa)=";
        cin>>Fs;
        cout<<"\n Shaft speed N(RPM)=";
        cin>>N;
        cout<<"\n Max. rest torsion T1(Newton)=";
        cin>>T1;
        T2=T1/TR;
```

```
T=((T1-T2)*(Dp/2));
cout<<"\n T="<<T;
Pow=(2*3.14*N*T*1000)/60;
//For Gear
Ftg=(2*T)/Dg;
Wg=Ftg/0.939;
Wgv=Wg*0.939;
Wgh=Wg*0.342;
//For Pulley
Wpv=W;
Wph=T1+T2;
cout<<"\n Wgv="<<Wgv;
//For shaft Vertical loading
Rbv=((Wgv*L1)+Wpv*(L1+L2))/L;
Rav=Wgv+Wpv-Rbv;
cout<<"\n Rav="<<Rav;
//Bending moment of Gear and Pulley (vertical)
Mgv=Rav*L1;
Mpv=Rbv*L3;
cout<<"\n Mgv="<<Mgv;
cout<<"\n Mpv="<<Mpv;
//For shaft Horizontal Loading
Rbh=(Wph*(L1+L2)+(Wgh*L1))/L;
Rah=Wgh+Wph-Rbh;
cout<<"\n Rbh="<<Rbh;
//Bending moment of Gear and Pulley (Horizontal)
Mgh=Rah*L1;
Mph=Rbh*L3;
cout<<"\n Rah="<<Rah;
//Resultant Bending moment
A=pow(Mgv,2);
B=pow(Mgh,2);
C=pow(Mpv,2);
D=pow(Mph,2);
Mg1=A+B;
Mp1=C+D;
Mg=pow(Mg1,0.5);
Mp=pow(Mp1,0.5);
if(Mp>Mg)
{
        M=Mp;
}
else
{
        M=Mg;
}
cout<<"\n M="<<M;
//Twisting moment
cout<<"\n Enter combined shock of fatigue factor for bending Km=";
cin>>Km;
cout<<"\n Enter combined shock of fatigue factor for torsion Kt=";
cin>>Kt;
M1=M*Km;
T11=T*Kt;
```

```
        E=pow(M1,2);
        F=pow(T11,2);
        Te1=E+F;
        Te=pow(Te1,0.5);
        G=((Te*16)/(3.14*Fs));
        d1=pow(G,0.333);
        Me=0.5*((M*Km)+Te);
        H=(Me*32)/(3.14*Fs);
        d2=pow(H,0.333);
        cout<<"\n\n Using max. shearstree theory calculate dia(mm)= "<<d1;
        cout<<"\n\n Using max. Normal stree theory calculate dia(mm)= "<<d2;
        if(d1>d2)
         {
                cout<<" \n\n Selected diameter  = "<< d1 <<"  (mm) ";
         }
        else
        {
                cout<<"  \n\n Selected diameter = "<< d2 <<"  (mm) ";
        }
        getch();
}
```