Design and Analysis of Spring-Ball Clutch Torque Limiter

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***Abstract* - The main purpose of self-releasing safety spring-ball clutch is to protect the system from overloads which causes the driving member to stall or get ruptured. This torque limiter provides self re-engagement after the removal of overload at the output which is not provided by conventional torque limiters. Design of this spring ball clutch is made for DC motor as the driving member with particular torque value which consists of the requirement of spring stiffness, design of input flange and cylindrical body for easy engagement and disengagement of clutch.ANSYS14.0 is used as analysis software**

***Index Terms* - Overload protection, spring ball safety clutch, Variable torque limit, Static Structural analysis, Ansys, SolidWorks**

I. INTRODUCTION

In an industry there is always need of more rapid, more rigid and precise equipment to increase capacity and productivity. Such requirement demands various mechanisms like gearing arrangements, high capacity motors and shaft drive mechanism. The output load on driving member exceeds in some of the applications like centrifugal pumps, grinders, ship propellers etc. When machine gets overloaded it results in failure of components such failure of shafts, burning of motor, gear teeth rupture [1]. In order to avoid overloading some preventive measures are incorporated between driving and driven mechanism, use of torque limiter is one of them.

This paper describes ball detent type torque limiter. Till date many type of torque limiters are made available and used. These come with various specifications, e.g. shear pin torque limiter which uses so called mechanical component designed to withstand specified shear load. It has disadvantage that it requires replacement of shear pin after each breakage. Another type is permanent magnet torque limiter, this type creates backlash problems. Third type is pawl-spring torque limiter, in which spring-loaded, cam follower or pawl-detent device is used but due to need of operator for re-engagement it is disadvantageous.

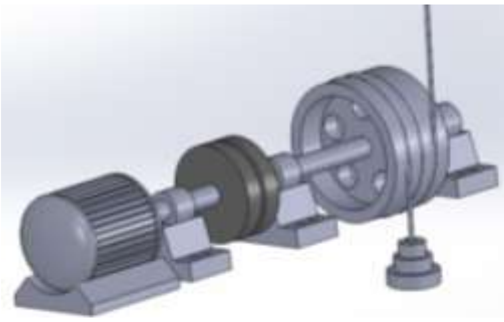
Considering the disadvantages we designed this torque limiter to protect against overload at particular torque limit which inherits following features

i. Variable torque limits
ii. Automatically re-engaged
iii. No-manual replacement needed
iv. No part worn-out
v. No backlash

II. OBJECTIVES

- Design & Development of V- profile ball detent three ball set overload Safety ball clutch with adjustable torque limit.
- Determination of angle of inclination of V-groove for easy disengagement & re-engagement.
- Calibration for limiting torque for given angular movement of locknut.

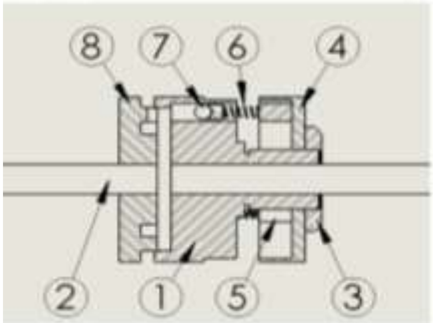
Figure 1Testing setup for the Torque Limiter



III. DESIGN

This section describes design of various components of toque limiter. The components are as shown in figure.2. The torque limiter is designed to sustain the toque up to 2 N-m and it is driven by 0.5 hp DC motor having speed 1500 rpm

Figure 2 Components of the Torque Limiter



ITEM NO.	PART NUMBER	QTY.
1	Cylindrical body	1
2	Shaft	2
3	Locknut	1
4	casing	1
5	plunger	3
6	spring	3
7	Ball	3
8	Input Flange	1

Input Shaft

The input shaft of the torque limiter is driven by a DC motor and it is carries an input flange on opposite side.

Input Flange

Input flange is one of the important components of torque limiter as shown fig.3. It carries 3 V-grooves as shown and the design procedure is described below.

Figure 3 Input Flange

Selecting Mild Steel-Checking for shear failure at where shaft is attached to flange

$[\tau]$ =64.8 N/mm^2

Since d=20 mm

$$(\tau)_{ind} = \frac{16\left[Mt\right]}{\pi\ d^{\ 3}}$$

$$(\tau)_{ind} = \frac{16\times 2\times 10^3}{\pi\times 20^3}$$

= 1.27 N/mm^2

Here $(\tau)_{ind} < [\tau]$

Hence the design is safe.

Cylindrical Body

Figure 4 Cylindrical Body

Cylindrical body is having three concentric holes as shown in fig.4 which carry three spring-ball-plunger assemblies. Holes are accurately drilled and reamed passing axially through the body, the holes are spaced exactly 120^0 apart around the same 90mm PCD.

The right hand end of the body is reduced in diameter and threads to receive the casing.

Design of Cylindrical Body

σ_t = 400 N/mm^2

σ_u = 480 N/mm^2

D_o = 40

D_i = 20

$F_{s\,max}$ = 66 N/mm^2

$$T = \left[\left(\frac{\pi \times F_{sact}}{16}\right) \times \left(\frac{\left(D_o^4 - D_i^4\right)}{20}\right)\right]$$

$$2 \times 10^3 = \left[\frac{\left(\pi \times F_{sact}\right)}{16}\right] \times \left[\frac{\left(40^4 - 20^4\right)}{20}\right]$$

F_{sact} = 0.08488 N/mm^2

$F_{sact} < f_{s\,max}$

Cylindrical body is safe under tensional load.

Ball and spring

Ball clutch nomenclature:

d = diameter of ball, mm
D =pitch circle diameter of groove, mm
Ft= total tangential force on balls, n
Fs=total spring force, n
F=spring force on each ball, n
A=angle of inclination of groove
Ks=spring stiffness, n/mm
Lf=free length of spring, mm
Mt=torque transmitted, n mm
N=number of turns in the spring
P=pitch of spring coil, mm
Z_b=number of balls in the clutch
M=coefficient of friction
K_1=stiffness per turn n /mm
A2=movement of ball while clutch is slipping, mm

a) Calculation of tangential force on balls

T = 2 Nm

$$Ft = \frac{\left(2 \times M_t\right)}{D}$$

$$= \frac{\left(2 \times 2 \times 10^3\right)}{90}$$

(Assuming pitch circle diameter of groove D=90 mm)

= 44.44 N

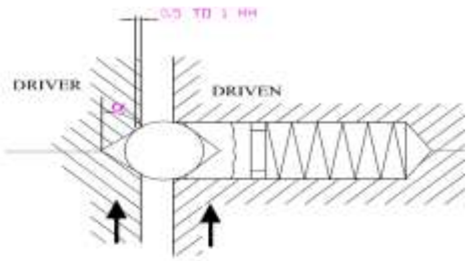


Figure 5 Spring ball arrangement

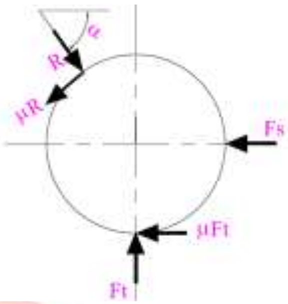


Figure 6 FBD of ball

b) Calculation of total spring force on balls (F_s)

$$Fs = Ft \times \left[\left\{ \frac{\cos\alpha - \mu \sin\alpha}{\sin\alpha + \mu\cos\alpha} \right\} - \mu \right]$$

Where,
μ=coefficient of friction between ball and body of clutch=0.08
α=Angle of inclination of groove=45^0

$$F_s = 44.44 \times \left[\left\{ \frac{(\cos 45 - 0.08 \times \sin 45)}{(\sin 45 + 0.08 \cos 45)} \right\} - 0.08 \right]$$

F_s = 34.30 N

c) Calculation of force on each spring

$$F = \frac{F_s}{Z_b}$$

Where,

Zb = No. of balls in clutch
= 3

$$F = \frac{34.30}{3}$$

=11.43 N

This is the static load acting on spring so taking dynamic load = F×1.75 = 20.00 N
Now by choosing the value of spring stiffness from PSG for the Dynamic load of 20 N

Table 1 Stiffness and permissible static and dynamic loads for helical compression springs

Wire diameter (mm)	**Outer diameter Mm**	**Stiffness of spring per turn,k1 N/mm**	**Permissible loads-Static N**	**Permissible load – Dynamic N**
1.0	8.0	30.98	49.6	21.3

d) Stiffness of spring

$$K_s = \frac{K_1}{n}$$

Where,

K1=Stiffness of spring per turn (N/mm)

n = No. of turns of spring=6

Stiffness and permissible static and dynamic loads for helical compression springs

$$K_s = \frac{K_1}{n}$$

$$= \frac{30.98}{6}$$

= 5.16 N/mm

e) Compression of spring to exert a force (δ_1)

$$\delta_1 = \frac{F}{K_s}$$

$$= \frac{11.43}{5.16}$$

= 2.21 mm

f) Movement of ball while clutch is slipping (δ_2)

$$\delta_2 = \frac{d}{2} \times (1 - \cos\alpha)$$

Where,

d is diameter of ball=10 mm

$$\delta_2 = \frac{10}{2} \times (1 - \cos 45)$$

= 1.46 mm

g) Maximum deflection of spring

$$\delta_{\max} = \delta_1 + \delta_2$$

$$= 2.1 + 1.46$$

= 3.67 mm

h) Free length of spring

Lf = Solid length + Max. Deflection+ Clearance between adjacent coils

$$L_f = n'd + \delta_{\max} + (n' - 1)$$

$$n' = n + 2$$

= 6+2

= 8

$$L_f = 8 \times 1 + 3.67 + (8 - 1)$$

= 18.67 mm

i) Pitch of spring (P)

$$P = \frac{L_f}{(n-1)}$$

$$= \frac{18.67}{(6-1)}$$

= 3.734 mm

Table 2 Estimated Parameters of Spring Ball Clutch

Sr. No.	Parameters	Notation	Value
1	Diameter of ball	d	12 mm
2	PCD of groove	D	90 mm
3	Angle of inclination of groove	A	45^0
4	Rod diameter of spring	$d^|$	1 mm
5	Outside diameter of spring	$D^|$	8 mm
6	Pitch of coil	P	4 mm
7	Free length of spring	Lf	21 mm

Casing

Hardened steel casing is of the same outside diameter as the front end of the body. The sleeve is deeply bored at one side to be close fit over the reduced portion on the outside of the body. By fitting the casing over the body at that point its correct and accurate location relative to the body is not determined by the fit in the threads. The three plungers bear simultaneously against the inner left hand face of the sleeve; thus as that members advanced longitudinally, all springs will be compressed or expanded by an equal amount.

Locknut

A threaded lock nut is screwed on the body behind the sleeve for locking the sleeve in any desired setting.

$[\tau] = 64.8$ N/mm^2

r = 20mm &

R = 30mm

Checking for crushing failure:

No. of threads:

$P \times n = H$

n=8

$$A = \frac{\pi}{4} \times (42^2 - 40^2) = 3.14 \text{ mm}^2$$

Total area= $8 \times A = 25.12mm^2$

Maximum force= 34.3×3 N

$$\sigma_c = \frac{f}{A} = \frac{(34.3 \times 3)}{25.12} = 4.09 \text{ N/mm}^2$$

$[\sigma_c] = 96$ N/mm^2

$(\sigma_c) < [\sigma_c]$

Hence locknut is safe under crushing failure.

Output Shaft

The output shaft carries the cylindrical body which is keyed to it at one end; whereas on the other end the output shaft carries the dynamometer brake pulley.

IV. Analysis

Body

Figure 7 Total Deformation result on ANSYS for Cylindrical Body
Maximum deformation = 0.0013249 mm

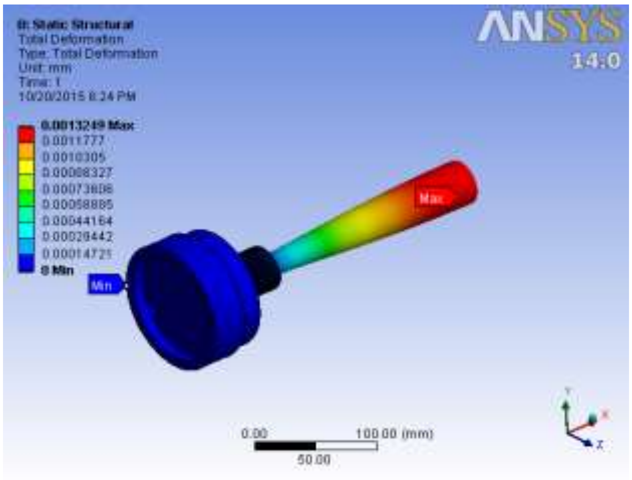
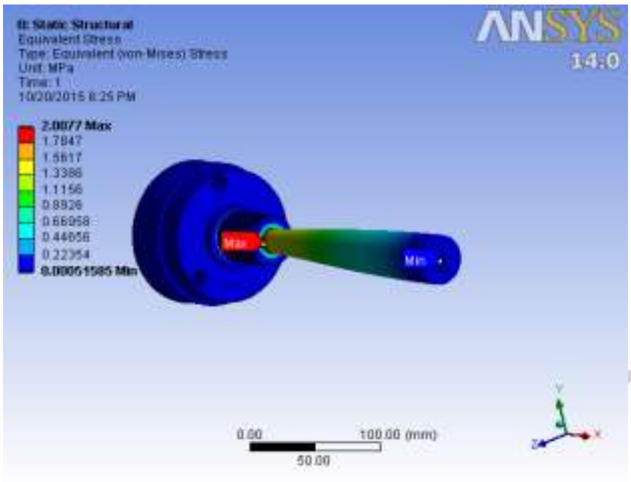


Figure 8 Total Deformation result on ANSYS for Cylindrical Body
Maximum Equivalent Stress = 2.0077 MPa, Allowable stress = 129.6 MPa



Plunger Spring Ball Assembly

Figure 9 Total Deformation result on ANSYS for ball-spring assembly
Total Maximum Deflection =6.7699 mm

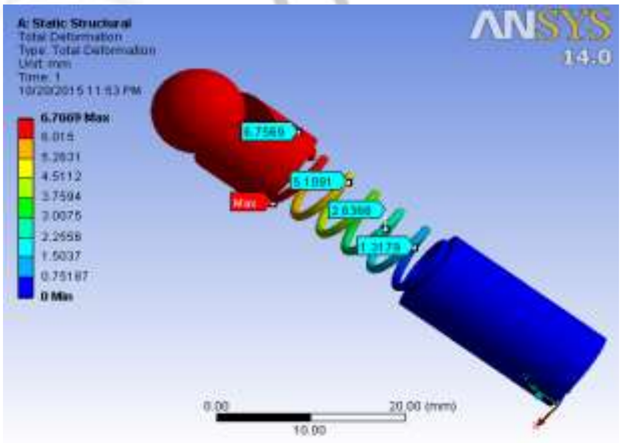


Figure 10 Stress Intensity result on ANSYS for ball-spring assembly

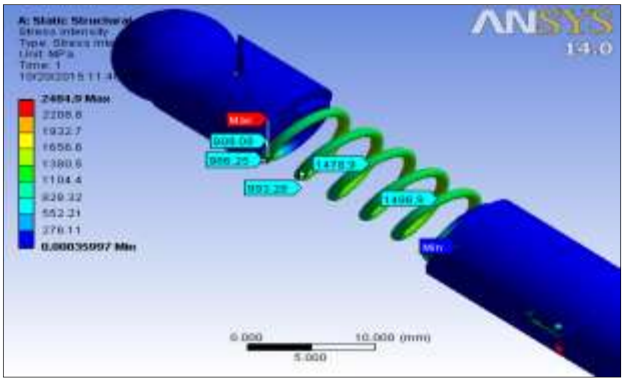
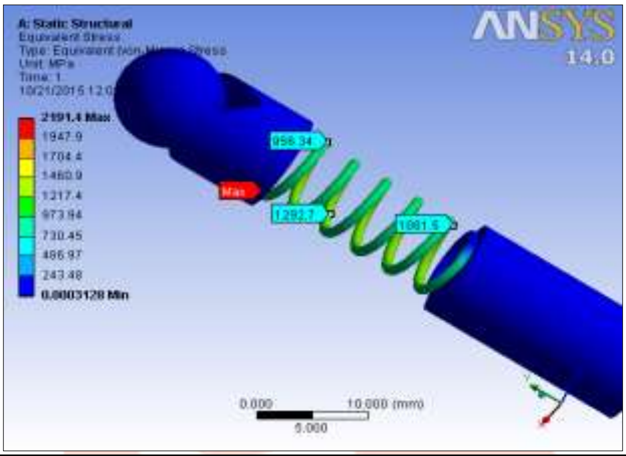


Figure 11 Equivalent Stress result on ANSYS for ball-spring assembly



Flange

Figure 12 Equivalent Stress result for Input Flange on ANSYS
Maximum Equivalent Stress = 2.1355 MPa, Allowable stress = 129.6 MPa

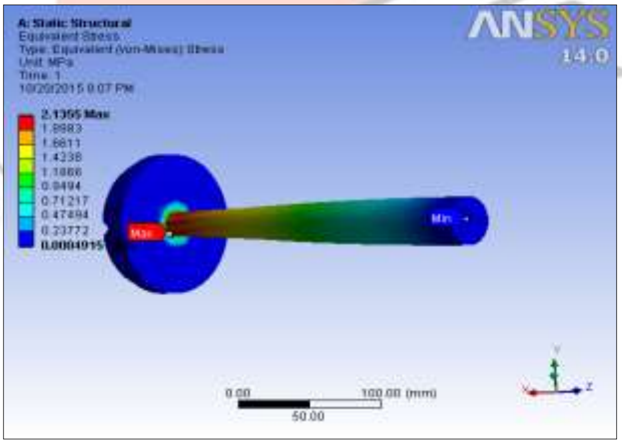


Figure 13 Total deformation result for Input Flange on ANSYS
Maximum Total Deflection= 0.0021669 mm

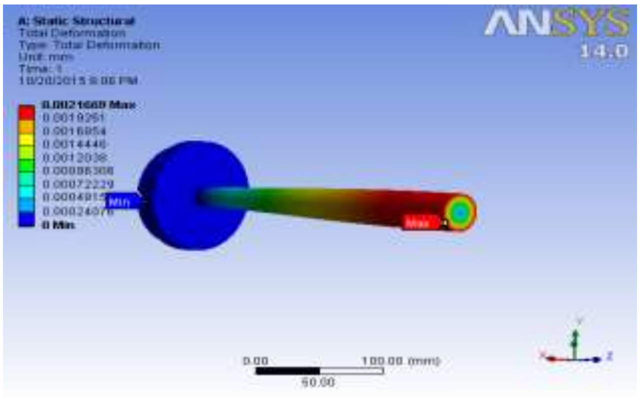
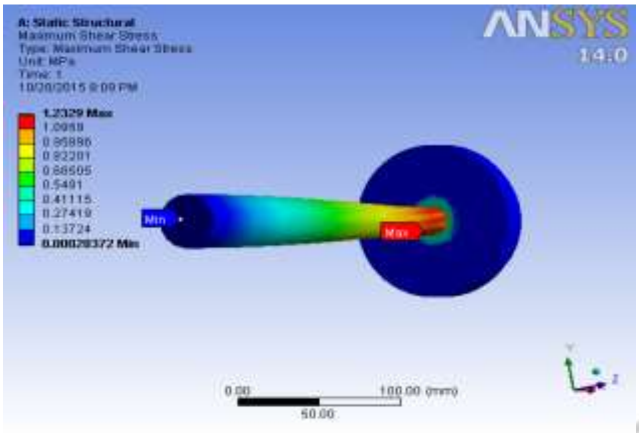


Figure 14 Max shear stress result on ANSYS for Input Flange
Maximum Shear Stress = 1.2329 MPa, Allowable stress = 64.8 MPa



V. CONCLUSION

The ball detent type of torque limiter with adjustable torque limit provides a range of torque that can be limited to protect the driver i.e. DC motor.

Mechanical design of individual component considers the material to be used, factor of safety, cost and availability of the material and feasibility of fabrication.

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