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Design of a Diaphragm Spring Clutch

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***Abstract* – Clutch is a very essential element in torque transmission process. The purpose of a clutch is to initiate motion in the automobile generally by transferring kinetic energy from another moving here which is the Engine. Clutches are designed to transfer maximum Torque form engine with Minimum Heat Generation. The present work deals with designing of (Ø310) Diaphragm Spring Clutch for a GVW of 8330kg with input parameters provided by the customer company.**

Keywords – Diaphragm Spring Clutch, Torque, Stress Generated, Clamp Loads, Heat Generated.

I. INTRODUCTION

In the Automotive Power Transmission the Clutch is designed after the Engine and Gear Box characteristics is finalized. The Customer has provided Input data comprising of the Engine Parameters (i.e. Engine Power, Max Engine Torque, RPM and Number of Engine Cylinders), Gear Box Parameters (i.e. Gear Ratios and Final Reduction), Vehicle Dimensions(i.e. Clutch Housing Diameter and Vehicle Weight) to design the Single Plate Diaphragm type Clutch for Medium Duty Vehicle. During the Clutch Design is necessary to ensure that,

[1] The Contact surfaces develop the frictional force that may pick up and hold the load with reasonable low pressure between the contact surfaces.

[2] The Heat of friction should be rapidly dissipated and tendency to grab should be at minimum.

[3] The surface should be backed by the material stiff enough to ensure reasonable uniform distribution pressure.

[4] Suitable Friction material forming the contact surfaces is to be selected.

[5] Clutch should not require any external force to maintain contact and the projection parts must be guarded.

Clutch Components are to be designed using these all the necessary parameters.

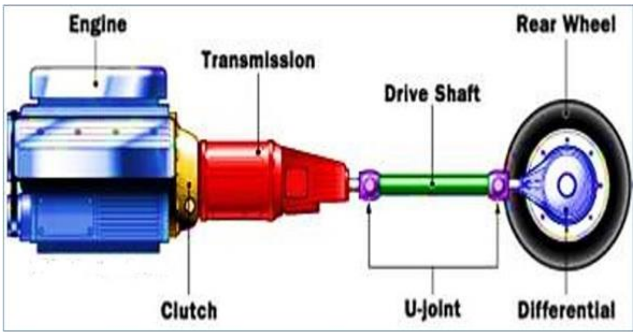

Fig.1. Automotive Transmission System Layout.

II. WORKING OF CLUTCH

The Diaphragm spring clutch uses a diaphragm or conical spring instead of coil spring to produce adequate pressure for engaging the clutch. The clutch cover is successfully secured to the engine flywheel. The pivot rings are held in clutch cover. The outer rim of the diaphragm spring is in contact with the pressure plate. In engaged position, the diaphragm spring keeps the pressure plate in firm contact with the flywheel.

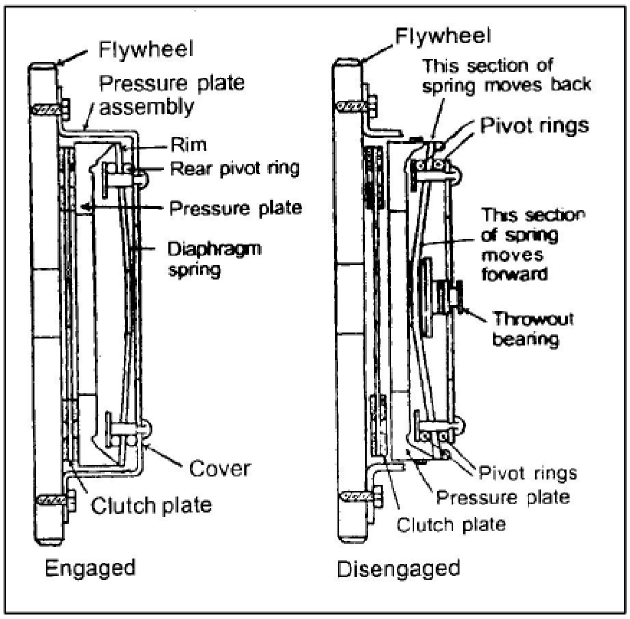

Fig.2. Working of Diaphragm Spring Clutch.

III. DESIGN DETAILS

Table I: Input for Ø 310 Clutch Design for TD 3250

Sr. No.	Particulars	Specification
1.	Vehicle : TD 3250 CR	
2.	GVW (kg)	8330
3.	RAW (kg)	1710
4	Tyre	3540
5	Dy. Radius (m)	0.39
6.	Engine	TD 3250 5CYL
7	Engine Power (kW)	90
8	Max Engine Torque Nm	320
9	Max Engine RPM	2800
10	Max Torque RPM	1800-2400
11	Gear Box Ratios	

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	First	5.053
	Second	2.601
	Third	1.512
	Fourth	1
	Fifth	0.784
	Reverse	4.756
	Final Reduction	4.375
12	Set Height	62
13	Clutch Actuation	Hydraulic
14	Pot Depth of Flywheel	Flat
15	Clutch Mounting PCD	314
16	Clutch Facing Inner Diameter (mm)	170
17	Clutch Facing Outer Diameter (mm)	310

Table III: Nomenclature and Symbols

Sr. No.	Parameter	Symbols
1	Gross Vehicle Weight	GVW
2	Dynamic rolling radius	m
3	Engine Torque	T
4	Engine Speed	N
5	Clutch Plate Inner Radius	R_i
6	Clutch Plate Outer Radius	R_O
7	Coefficient of friction	μ
8	Outside diameter of clutch	D
9	Inside diameter of clutch	D_i
10	Slip Torque	T_S
11	Clamp load	W
12	Mean radius	R_m
13	Diameter ratio	D_r
14	Load on flat	P_{flat}
15	Thickness of diaphragm spring	t
16	Height of diaphragm spring	h
17	Stress on diaphragm spring	σ
18	Deflection of spring	δ
19	Spring rate	K
20	Working load	F
21	Working stress	τ
22	Solid length of spring	L_S
23	Block load of spring	F_B
24	Block stress of spring	τ_B
25	Total torque	T_T
26	Total load	F_T
27	Wire diameter	d
28	Outer diameter of spring	D_O
29	Mean diameter of spring	d_m
30	Modulus of rigidity	E
31	Torsional radius	R
32	Torsional angle	β
33	No. of total coils of spring	n
34	No. of active coils of spring	n_a
35	Acceleration	A
36	Force	F
37	Velocity	V
38	Slip time	t_s
39	Heat generated	q
40	Specific heat	Q
41	Thermal stress	σ_T
42	Outside diameter of casing	D_C
43	Maximum diameter for bolt	d_b
44	Spline diameter	D_S
45	Maximum length required for spring	L_{S1}
46	Length of slot for 1st stage	L_{S2}
47	Outside diameter of hub	D_h
48	Diameter of stop pin	d_s
49	Outside diameter of diaphragm	D_a
50	Outer diameter of slot	D_i
51	Outside diameter of fulcrum	D_{ap}
52	Inside diameter of fulcrum	D_{ip}
53	Spring index	C
54	Deflection Factor	K_1, k_2, k_3
55	Diaphragm angle	A

- ***Material Selection: 50CrV4***

3.1 Clamp load (W) Calculation:

- ***The given values as per problem statement are:***
  - Max Engine Torque (T) = 320 Nm
  - Outer radius (R_o) = 155mm
  - Inner radius (R_i) = 87.5mm

- ***To find***

Clamp load
Stresses at various points

- ***Slip torque calculations:***

Max Engine Torque (T) = 320 Nm

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Assume Slip Torque (T_s) is 1.7 times T
Therefore T_s = (T) * (1.7)
= 320*1.7
T_s =544 Nm

- ***To calculate mean radius (R_m):***

Mean Radius (R_m) $= \left(\frac{2}{3}\right) * \left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$ (According to uniform Pressure Theory)

Mean Radius (R_m) $= \left(\frac{2}{3}\right) * \left(\frac{0.140^3 - 0.0825^3}{0.140^2 - 0.0825^2}\right)$

R_m=0.12438 m

3.1.1 To calculate the clamp load (W):

Slip Torque T_s= 2 * μ* W* R_m
544= 2*0.27*W*0.1136
W = 900 Kg
Clamp Load (W) = 825.63 kg

- ***To determine the stresses induced in the diaphragm spring***

Input Parameters
Thickness of diaphragm (t): 3.65mm
Outside diameter of diaphragm (Da): 289mm
Slot outer diameter (Di'): 240mm
Diaphragm angle (A): 13.833º
Height of diaphragm (h): 5.857mm
Outer fulcrum diameter (Dap): 284.5mm
Inner fulcrum diameter (Dip): 236mm
Poisons ratio (μ): 0.27
Modulus of elasticity (E): 20500kgf/mm2

To determine:

1. Actual Diameter (D_i) = (D_i')-(2 * S * Sin(A))
=217 – 2(3 x sin (14))
D_i= 238.34 mm

2. Diameter Ratio (D_r) $= \frac{D_a}{D_i}$

$= \left(\frac{263}{215.548}\right)$

D_r =1.2130

3. Alpha (α) $= \frac{\left(\frac{1}{\pi}\right) * \left(\frac{Dr+1}{Dr-1}\right)^2 * (1-\mu^2)}{\left[\frac{Dr+1}{Dr-1} - \frac{2}{lnD_r}\right] * 4 * 21000}$

$\alpha = \frac{(0.3183) * (0.03271) * (0.91)}{(10.0579 - 10.0247) * 84000}$

$\alpha = 3.3673 * 10^{-6}$

4. Load on flat (P_{flat}) $= \frac{(h) * (t^3)}{(\alpha) * D_a^2}$

$= \frac{(5.6) * (3.25^3)}{(3.3973 * 10^{-6}) * (263^2)}$

P_{flat}= 1014.8897 N

5. Load on flat with D_{ai} & D_{ip} P_f $= (P_{flat}) * \left[\frac{(D_a - D_i)}{(D_{ap} - D_{ip})}\right]$

$= (818.07) * \frac{(263 - 215.548)}{(258 - 216)}$

P_f=1025.3524 N

3.1.2 Stress Calculations:

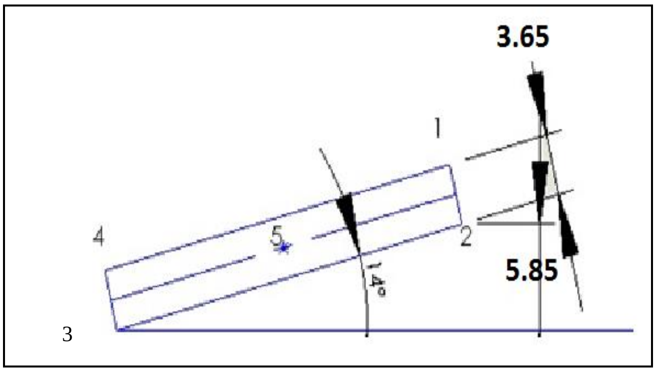

Fig.3. Stresses on Diaphragm Spring

Thickness as, t =3.65mm
Consider Deflection h = 5.857mm

Calculation of deflection factor k_1, k_2, k_3

1. $k_1 = \frac{\left(\frac{1}{\pi}\right) * \left(\frac{Dr-1}{Dr}\right)^2}{\left[\frac{Dr+1}{Dr-1} - \frac{2}{\ln Dr}\right]}$

k_1= 0.3050

2. $k_2 = \left(\frac{1}{\pi}\right) * \left(\frac{Dr}{\ln(Dr)}\right) * \left[\frac{Dr-1}{\ln(Dr)} - 1\right]$

k_2= 1.096

3. $k_3 = \left(\frac{1}{\pi}\right) * \left(\frac{6}{\ln(Dr)}\right) * \left(\frac{Dr-1}{2}\right)$

k_3= 1.0535

4. Stress at point 5:

$$\sigma_5 = \left(\frac{-3}{\Pi}\right) * \left(\frac{4E}{(1-\mu^2)}\right) * \left(\frac{h * t}{k_1 * {D_a}^2}\right)$$

σ_5 = -72.523 N/mm^2

5. Stress at point 1:

$$\sigma_1 = \left(\frac{-4 * E}{1 - \mu^2}\right) * \left(\frac{h * t}{k_1 * D_a^2}\right) * \left(k_2 * \left(\left(\frac{h}{t}\right) - \frac{0.5 * h}{t}\right) + k_3\right)$$

σ_1= -146.7N/mm^2

6. Stress at Point 2:

$$\sigma_2 = \left(\frac{4 * E}{1 - \mu^2}\right) * \left(\frac{h * t}{k_1 * D_a^2}\right) * \left(-k_2 * \left(\left(\frac{h}{t}\right) - \frac{0.5 * h}{t}\right) + k_3\right)$$

σ_2 =13.364 N/mm^2

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7. Stress at Point 3:

$$\sigma_3 = \left(\frac{4*E}{1-\mu^2}\right)*\left(\frac{h*t}{k_1*Dr*D_a^2}\right)*\left((2k_3-k_2)*\left(\left(\frac{h}{t}\right)-\frac{0.5*h}{t}\right)+k_3\right)$$

$\boldsymbol{\sigma_3}$ **= 123.66N/mm²**

8. Stress at Point 4:

$$\sigma_4 = \left(\frac{4*E}{1-\mu^2}\right)*\left(\frac{s*t}{k_1*Dr*D_a^2}\right)*\left((2k_3-k_2)*\left(\left(\frac{h}{t}\right)-\frac{0.5*h}{t}\right)-k_3\right)$$

$\boldsymbol{\sigma_4}$ **= -7.536N/mm²**

Safe Stress = 0.8* UTS

= 0.8 * 180

Safe Stress = 144 N/mm²

As the Max Stress, 138.914 N/mm² < Safe Stress 144 N/mm² our design is safe

Factor of safety = $\frac{UTS}{Max.Stress}$

Factor of Safety = 1.295

3.2 Design of Damper Spring

As per the given data of damper dimensions we find out the deflections, spring rate, stress and torque on the spring.

Material for damper spring: Spring Steel as per AS 4454 1975 Grade2D

Damper Torque Calculation:

Assume that Damper Torque is 1.3 to 1.45 times engine torque

Damper Torque= 1.4* Engine Torque

= 1.4*320

Damper Torque = 448 N

Now we find out deflection to determine working load and spring rate for inner and outer springs

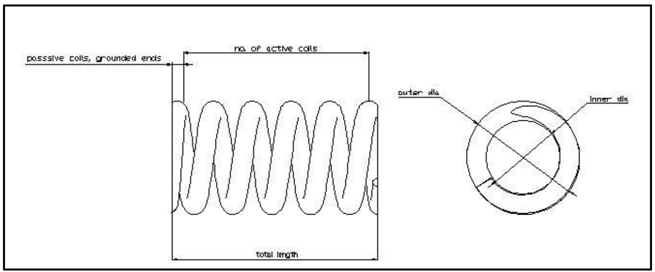

Fig.4. Inner and Outer Damper Spring

Table III: Input parameters for outer spring

PARAMETER	UNIT	SPECIFICATION
Wire Dia	Mm	4.5
No. of Total Coils		5.5
No. of Active Coils		3.5
O. D.		23
Material Grade		2D
UTS	Kgf/mm2	185
Mean Dia(dm)		18.5
Length		28
Modulus Of Rigidity (G)	Kg/mm2	7500
Number of springs		8
Engine Torque	Nm	320
Torsion Angle	Degree	5.5
Torsional Radius	Mm	53.3

3.2.1 Calculation for outer spring

1. Deflection (δ) = Angle x radius(r)

$= \frac{5.5}{180}$ x Π x 57.3

δ = 4.9087 mm

2. $\frac{F}{\delta} = \frac{G*d}{(8*C^3*n)}$

$$F = \frac{5.1164*8300*2.8}{8*(4.628)^3*4.66}$$

Working Load (F) = 85.1538kgf

3. Spring rate (K) = $\frac{F}{\delta}$ **= 17.35kg/mm**

4. Working Stress (τ) = $\frac{8 \text{ x } F \text{ x } (D_0-d)}{\pi d^3}$

τ= 44.022kgf/mm²

5. Solid Length (L_S) = (n) x (d) – $\frac{(d)^2}{d_m}$

= (6.16 x 2.8) – $\frac{2.8^2}{13}$

$\mathbf{L_S}$**= 23.65 mm**

6. Spare Length ={Length - Solid Length} – {Deflection}

= 28 – 23.65 - 4.9087

Spare Length= -0.563 mm

7. Block Load (F_B) = Block Length x K

$\mathbf{F_B}$**=78.386kgf**

8. Block Stress (τ_B) = $\frac{(8 \text{ x } F \text{ x } (Do-d))}{\Pi * d^3}$

$= \frac{(8 \text{ x } 73.8447 \text{ x } (13-2.8))}{\Pi \text{ x } (2.8)3}$

$\boldsymbol{\tau_B}$**=38.97kgf/mm²**

9. Total Load (F_T) = No. Of springs x Working Load

$\mathbf{F_T}$**=631.2034kgf**

10. Total Torque (T_T) = Load x Torsional Radius x 9.81

=631.2034 x 53.3 x 9.81

$\mathbf{T_T}$**=417.69 Nm**

3.2.2 Calculations for Inner Spring

Table IV Input parameters for Inner Spring

PARAMETER	**UNIT**	**SPECIFICATION**
Wire Dia		2.5
No. of Total Coils		9.5
No. of Active Coils		7.5
O. D.		13.5
Material Grade		2D

UTS	Kgf/m m2	185
Mean Dia (d_m)		12
Length		28
Modulus Of Rigidity (G)	Kg/m m2	7500
Number of springs		8
Engine Torque	Nm	320
Torsion Angle	Degre e	5.5
Torsional Radius	Mm	53.3

1. Deflection (δ) = Angle (in radian) x radius(r)

$= \frac{5.5}{180} \times \Pi \times 57.3$

δ = 4.9087 mm

2. $\frac{F}{\delta} = \frac{G * d}{(8 * C^3 * n)}$

$F = \frac{5.1164*8300*2.8}{8*(4.628)^3*4.66}$

Working Load (F) = 85.1538kgf

3. Spring rate (K) $= \frac{F}{\delta}$ **= 17.35kg/mm**

4. Working Stress (τ) $= \frac{8 \times F \times (D_0 - d)}{\pi d^3}$

τ= 44.022kgf/mm²

5. Solid Length (L_S) $= (n) \times (d) - \frac{(d)^2}{d_m}$

$= (6.16 \times 2.8) - \frac{2.8^2}{13}$

L_S= 23.65 mm

6. Spare Length ={Length - Solid Length} – {Deflection}

= 28 – 23.65 - 4.9087

Spare Length= -0.563 mm

7. Block Load (F_B) = Block Length x K

F_B=78.386kgf

8. Block Stress (τ_B) $= \frac{(8 \times F \times (Do - d))}{\Pi * d^3}$

$= \frac{(8 \times 73.8447 \times (13 - 2.8))}{\Pi \times (2.8)3}$

τ_B=38.97kgf/mm²

9. Total Load (F_T) = No. Of springs x Working Load

F_T = 631.2034kgf

10. Total Torque (T_T) = Load x Torsional Radius x 9.81

=631.2034 x 53.3 x 9.81

T_T = 417.69 Nm

3.3 Calculations of Slip Time for 1st& 2ndGear :

Table IV Input parameters for Slip Time Calculations

Sr. No.	Particulars	Specification
1	Gradability (%)	15
2	GVW (kg)	8330
3	Dy. Radius (m)	0.339
4	Max Engine Torque Nm	320
5	Max Torque RPM	1400
6	Gear Box Ratios	
	First	5.053
	Second	2.601
7	Differential ratio	4.375
8	η of transaxle (%)	85
9	Road resistance constant (kg/ ton weight)	20

3.3.1 Slip Time for 1st Gear:

1. Consider 15%Gradability with Indian roads

$\tan \theta = (\frac{15}{100}) * \frac{\pi}{180}$

θ= 0.148radian

2. Force (F) =B- ((Road resistant const) * (GVW) + (θ)(GVW))

$B = \frac{(\eta \text{ of transaxle}) * (\text{Engine toruqe}) * (\text{1st gear ratio}) * (\text{differential ratio})}{\text{Dyanamic rolling radius}}$

Force (F) = 298.687kg

3. Force = (mass) * (Acclⁿ)

$Accl^n = \frac{298.687}{8330}$

a = 0.35 m/s²

4. $Velocity(v) = (\frac{2\pi N}{60}) * (\frac{\text{Dyanamic rolling radius}}{(\text{1st gear ratio}) * (\text{differential gear ratio})})$

$= (\frac{(2) * (\pi) * (1400)}{60}) * (\frac{0.339}{(5.053) * (4.375)})$

v= 2.393 m/s

5. V = u + at (By Newton's Laws of motion)

2.393 = 0 + (1.7305) * (t_{slip})

Slip time (t_{slip}) = 6.8175 sec

3.3.2 Slip Time for 2ndGear:

1. Force(F)=B- ((Road resistant const) * (GVW) + (θ)(GVW))

$B = \frac{(\eta \text{ of transaxle}) * (\text{Engine toruqe}) * (\text{2nd gear ratio}) * (\text{differential ratio})}{\text{Dyanamic rolling radius}}$

F= 665.85kgf

2. Force = (mass) * (Accl^n)

$$\text{Accl}^n = \frac{665.85}{8330}$$

a = 0.7842 m/s^2

3. Velocity(v)=$\left(\frac{2\pi N}{60}\right) * \frac{\text{Dyanamic rolling radius}}{(\text{2nd gear ratio}) * (\text{differential gear ratio})}$

$$= \left(\frac{(2)*(\pi)*(2000)}{60}\right) * \left(\frac{0.339}{(2.601)*(4.375)}\right)$$

v = 6.6435 m/s

4. V = u + at

6.6435= 0 + (0.0910) * (t)

t_{slip} = 8.4772 sec

As the slip time value of 2nd gear is Large, vehicle will not start in 2nd gear.

❖ Heat generated for 1st gear:

$$q = \frac{2\pi NT}{60000}$$

=46.98**kJ/s**

❖ Specific Heat for 1st gear (Q) = (q) * (t_{slip})

= (46.98) * (6.8175)

Q = 0.319 MJ

Thermal stress (σ_T) $= \frac{123920}{40192.75}$

σ_T= 1.5508 J/mm^2

*Maximum permissible thermal stress is 2 **J/mm^2**

As 1.5508J/mm^2<2J/mm^2 the Design is Safe.

IV. CONCLUSION

The Diaphragm type clutch is designed for TD 3250 as per the customer requirements. All the parts were modeled using CATIA and Finite Element Analysis in ANSYS is conducted to verify the results. All the results were found to be within the permissible limits. New designs of Diaphragm Spring have been suggested and should be considered for further investigation. Detailed study of manufacturing processes of Diaphragm spring was done and manufacturing process was suggested. All the parts were procured and assembled in the industry premises. Strict Quality Control and Inspection processes were devised and implemented on the clutch manufacturing.

V. FUTURE SCOPE

The Diaphragm spring Design can be optimized to reduce its weight. A modified design has been suggested, but further study and analysis needs to be done to checks its feasibility. The deflection of the Diaphragm clutch can be reduced by introducing a strap plate. However more study and research needs to be conducted. Pin fin can be used to increase heat transfer rate and prevent the overheating of the clutch. Clutch housing can be provided with a window to enhance air circulation for heat dissipation.

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