Mini project on

Design Of Drum Brake for Two Axle Vehicle

Vishal Singh 1401ME49

IIT Patna

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Objective

Design and analysis of internal expanding drum brakes for a two axle vehicle to achieve required stopping distance at given maximum velocity of the vehicle under normal conditions.

Design Parameters

Consider the Baja vehicle travelling at maximum speed of V_{max} =50 km/h, which is required to stop at a minimum stopping distance of 10m by braking. The details of the vehicle required for the design of drum brakes are mentioned below.

• Total mass (vehicle + driver): M = 300 kg	• Wheel base : WB = 1.47 m
• Height of CG from ground : H = 0.56 m	• Max speed of vehicle : V_{max} = u = 50km/h
• Static front axle load : W_f = 1236N	• Static rear axle load : W_r =1764 N
• Max stopping distance : s = 10m	• Radius of tyre : R_{tyre} = 0.279m

Braking Torque Calculation

- Deceleration during braking, $a_{max} = (V_{max})^2/(2*s) = (50*1000/3600)^2/(2*10) = 9.64\ m/s^2$.
- Weight transfer during braking : WT =(M*a_{max}*H) / WB = 1101.7 N
- Dynamic load on the front wheels during braking=W_{fd}= Static front load + weight transfer = 2337.7N
- Braking force on each front wheel: $F_{bf} = (W_{fd}*a_{max}) /(2*g)$ = 1148.6N
- Braking torque on each front wheel : $T = F_{bf} * R_{tyre}$ = 320.9 N.m

Parameters Of the Drum Brake System

Following standard terminology of drum brake system have been followed for remaining work:

r = inner radius of the drum a = distance b/w the hinge & the center of drum b = width of the drum and pad P_{aR} = max pressure on right pad θ_a = angle of max pressure wrt hinge	θ_1 = angular position of friction pad start wrt hinge θ_2 = angular position of friction pad end wrt hinge f = coefficient of friction b/w friction pad & drum. P_{aL} = max pressure on left pad

And the corresponding values are as given below

r = 10 cm $\theta_1 = 10^0$ f = 0.35	a = 7.5 cm $\theta_2 = 120^0$	b = 3.75cm $\theta_a = 90^0$

Calculation

Total moment on right side pad due to frictional force about hinge : M_f

$$M_f = \int f dN(r - a\cos\theta) = \frac{fP_{aR}br}{\sin\theta_a}\int_{\theta_1}^{\theta_2} \sin\theta(r - a\cos\theta)d\theta = 2.3327\times10^{-4}\times P_{aR}$$

Total moment on right side pad due to normal force about hinge : M_N

$$M_N = \frac{P_{aR}bar}{\sin\theta_a}\int_{\theta_1}^{\theta_2} \sin^2\theta\, d\theta = 3.55\times10^{-4}\times P_{aR}$$

$$F = \frac{M_N - M_f}{c} = 1.22\times10^{-4}\times\frac{P_{aR}}{c}$$

$$T_r = \frac{fP_{aR}br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2} \sin\theta\, d\theta = 1.9488\times10^{-4}\times P_{aR}$$

Total moment on left side pad due to frictional force and normal force respectively:

$M'_f = M_f * P_{aL} / P_{aR}$ and $M'_N = M_N * P_{aL} / P_{aR}$

Since the actuating force on both pads are equal, we have

$$F = \frac{M'_N + M'_f}{c} = 1.22\times10^{-4}\times\frac{P_{aR}}{c}$$

On substituting the expressions for M'N and M'_f we get,

$P_{aR} = 4.82 * P_{Al}$ ……(1)

$$T_L = \frac{fP_{aR}br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2} \sin\theta\, d\theta = 1.9488\times10^{-4}\times P_{aL}$$

Braking torque , **$T = T_L + T_R$**

$P_{aR} + P_{aL} = 82.33 * 10^4$ ……(2)

From (1) and (2) we will get : **P_{aL} = 141* 10^3 Pa = 141 kPa**

P_{aR} = 681.8 * 10^3 Pa = 681.8 KPa

Actuating force: F =(M_N - M_f)/c = 3.6 kN

Results Obtained

Max pressure on right pad, Par = 681.8 kPa

Max pressure on left pad, Pal = 141 kPa

Actuating force on each pad , F = 3.6kN

Discussion

Max(Par, Pal) < Pallowe indicating the safe operation of the pad.

Material selection

Brake Drum	**Friction pad**
Gray Cast Iron Grade250 K = 0.0544 w/mmoC Y = 1e05 MPa Cp = 410 J / kg- K	Rigid molded asbestos f : 0.35 -0.41 Pmax = 0.75 Mpa Tmax = 4000 C

Comparison with other brakes

- Effective in heavy vehicles
- Problem of centrifugal force
- unable to release heat out
- High wear rate

What improvements can be done???

- Cooling process of the brake drum can be improved by extending fins on outer surface of the drum
- For same amount of actuating force, braking torque can be increased by increasing no of pads.