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Design and Calculation of Brake Booster for Hydraulic Braking System

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Abstract: This paper describes the design of brake booster used in hydraulic system of SUZUKI car. The function of brake booster, master cylinder, the wheel cylinder, the brake is also important to get actuating force for the vehicle. When the brake pedal is pressed, the brake pedal force enters to the booster, the booster force enters to the master cylinder and the force enters to the caliper. The booster is reducing the driver force. A brake system starts from the brake pedal. So, in this design uses the brake pedal length 340mm and pedal ratio 5.66. The brake system is used the brake booster because it reduces the driver force 80N to 20N and the brake pedal movement 46mm to 20mm. While this reduces operation, master cylinder piston moves to 3.533mm. In this paper, when the brake booster establish which has diaphragm pressure 41kPa and diaphragm area $35.88\times 10^{-3}m^2$,the total driver applied force is increased in four times. This force enters to the master cylinder. Master cylinder in the piston is applied force 1255.96N and pressure 2.36MPa. The output force from the master cylinder applied to the caliper. Then, the caliper output force is 3931.5N. This force is better than without booster force. Finally, the output torque is 82.56Nm. This torque is four times of without booster braking system. In accordance with the braking force, the length of brake pedal and the size of booster and the force acting on the diaphragm. Calculation of brake pedal force, booster force, master cylinder force, caliper force and torque are presented in this paper.

Keywords: Brake Booster, Brake Pedal, Diaphragm, Cylinder, Torque, Caliper Force.

I. INTRODUCTION

The various system fitted to a motor vehicle to slow it down, stop or keep all stationary came from this heating. A moving vehicle processes kinetic energy which is converted into heat energy on the application of brake. This heat is transferred to the surrounding air. In the simplest form, a brake comprises a stationary brake shoe with a friction lining on it and brake drum. The heat generated due to braking action is proportional to the force which brings the shoe in contact with the drum brake, in general, required to slow.

1. Application of brakes should bring the vehicle to a relatively quick stop on any type of road-wet, dry, uphill or downhill.
2. A separate mechanical brake is required to hold vehicle in position on a gradient.
3. The braking system components require minimum maintenance.
4. The pedal effort required to produce maximum deceleration should be negligible and should not vary with the condition of the road.
5. The braking system should allow minimum time between application of pedal effort and actual braking effect on the drums.
6. The braking action should not involve any noise, or drift the vehicle away from its desired path.

A. Type of Braking Systems

This criterion gives the following braking system types.

- Hydraulic brakes.
- Mechanical brakes
- Electric brakes
- Grilling Mechanical brakes
- Engine Exhaust brakes
- Air brakes

II. DESIGN OF WITHOUT BRAKE BOOSTER FOR ONE POT CALIPER

Torque is a measure of how much force acting on an object to rotate. The object rotates about an axis, which called the pivot point and labeled 'O' called the force 'F'. Torque is defined as;

$$T = F_t \times r_e \quad (1)$$

Where,

T =Torque (Nm)
F_t= Transmitted Force (N)
r_e= Effective radius (m)

A. Design of Brake Pedal Model

The brake pedal model lever is the device for applying input force to the master cylinder initiating deceleration. The input force depends upon the pedal ratio and applied force. Therefore, the input force may be defined as;

$$F_{dapp} = L_p \times F_{app} \quad (2)$$

Where,

F_{dapp}= Driver applied force (N)
L_p= Brake pedal ratio
F_{app}= Applied force (N)

$$L_p = \frac{d_{app}}{d_{in}} \quad (3)$$

Where,s

d_{app}= Brake pedal and fulcrum distance (mm)
d_{in}= Fulcrum and push rod distance (mm)

B. Design of Master Cylinder Piston Movement

Operation of the master cylinder is simple. When the brake pedal is depressed, force is applied through the push rod to the master cylinder piston.

$$d_{cm} = \frac{d_{pm}}{L_p} \quad (4)$$

Where,

d_{cm}= Master cylinder piston movement (mm)
d_{pm}= Pedal movement (mm)

C. Design of Master Cylinder Piston Force

The piston moves along the bore of the master cylinder and this movement is transferred through the hydraulic fluid to result in a movement of the slave cylinder. Fig.1 shows the hydraulic pressure created by moving a piston.

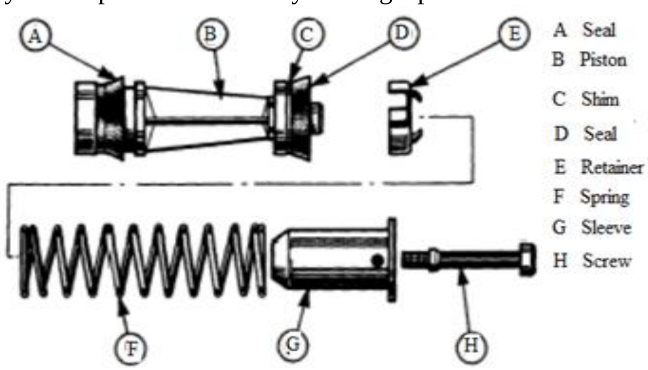

Fig.1 Master Cylinder Piston.

$$F_c = F_{dapp} - (F_{csp} + (K_{sc} \times d_{cm})) \quad (5)$$

Where,

F_{csp}= Master cylinder pre-tension force (N)
K_{sc}= Master cylinder rate of spring (N/mm)

D. Design of Master Cylinder Piston Pressure

A master cylinder is used to convert force from the brake pedal into the hydraulic pressure that operates the brake caliper. The master cylinder piston pressure is defined as;

$$P_c = \frac{F_c}{A_p} \quad (6)$$

Where,

F_c= Master cylinder piston force (N)
A_p= Area of piston (m^2)

E. Design of Caliper Force

Brake caliper being the heart of a braking system. Opposed piston designs are typically stiffer. But, if the input pressure and piston area is the same for both designs, the caliper will generate the same theoretical forces.

$$F_{cp} = P_c \times A_{cp} \quad (7)$$

Where,

F_{cp}= Caliper force (N)
P_c= Master cylinder piston pressure (Pa)
A_{cp}= Caliper area (m^2)

F. Design of Transmitted Force

Effective braking is a critical factor determining the performance of any vehicle. The clamping force and the coefficient of friction are on one of the equation.

$$F_t = F_{cp} \times \mu_c \times n \quad (8)$$

Where,

μ = Coefficient of friction
n = Number of pads

Table I. Design Specification of Without Brake Booster for One Pot Caliper

No.	Name	Data
1	Applied force, F_{app}	80 N
2	Brake pedal and fulcrum distance, d_{app}	340 mm
3	Fulcrum and push rod distance, d_{in}	60 mm
4	Pedal movement, d_{pm}	46 mm
5	Diameter of master cylinder, d_c	26 mm
6	Piston spring pre-tension, F_{csp}	15N
7	Piston spring rate, K_{sc}	8 N/mm
8	Wheel diameter, d_w	250 mm
10	Caliper piston, d_{cp}	46 mm
11	Number of pads	2
12	Coefficient of friction, μ_c	0.35

III. DESIGN SPECIFICATION OF WITH BRAKE BOOSTER FOR ONE POT CALIPER

The brake booster doesn't make any noise and it doesn't use any electricity or gasoline, but it insures that brake booster can stop the car with only a light touch of the brake pedal. Things weren't always like that, before the invention of the vacuum brake booster, cars still stopped. A light application of the brakes is translated by the car stops quickly.

$$P_D = P_{atm} - P_{vacuum} \quad (9)$$

Where,

P_D= Diaphragm pressure (Pa)
P_{atm}= Ambient pressure (Pa)
P_{vacuum}= Manifold pressure (Pa)

A. Design of Diaphragm Area

The amount of braking power from vacuum booster is directly related to the area of the diaphragm. The larger the diaphragm area the more "free pressure" the booster can provide. The booster diaphragm area is the different total area and valve body.

$$A_D = A_T \times A_v \quad (10)$$

Where,

A_D= Diaphragm area (m^2)
A_T= Total area (m^2)
A_v= Area of valve body (m^2)

B. Design of Diaphragm Force

The force at which the higher pressure moves toward the lower pressure is determined by the difference in pressures. When the pressure is slightly lower than atmospheric, the force is low. When there is a large difference, the higher pressure will rush into the lower and the force will be great.

$$F_{atm} = P_{atm} \times A_D \quad (11)$$

Where,

F_{atm}= Atmospheric force (N)

$$F_{vacuum} = P_{vacuum} \times A_D \quad (12)$$

Where,

F_{vacuum}= Intake manifold force (N)
P_{vacuum}= Intake manifold pressure (Pa)

$$F_D = F_{atm} - F_{vacuum} \quad (13)$$

Where,

F_D= Diaphragm force (N)

C. Design of Booster Output Force

The feedback force applied as pedal feel is always less than booster output force, but it also is always proportional. The booster output force depended on the diaphragm force, spring pre-tension, rate of spring and master cylinder piston movement.

$$F_B = F_D - (F_{Bsp} + (K_{sB} \times d_{cm})) \quad (14)$$

Where,

F_B= Booster output force (N)
F_{Bsp}= Booster pre-tension force (N)
K_{sB}= Diaphragm rate of spring (N/mm)

D. Design of Driver Applied Force

The driver applied force depended on the applied force, brake pedal ratio and booster output force.

$$F_{dapp} = (F_{app} \times L_p) + F_B \quad (15)$$

Where,

F_{app}= Applied force (N)
L_p= Brake pedal ratio
F_B= Booster output force (N)

Table II. Design Specification of with Brake Booster for One Pot Caliper

No.	Name	Value
1	Applied force, F_{app}	20 N
2	Brake pedal and fulcrum distance, d_{app}	340 mm
3	Fulcrum and push rod distance, d_{in}	60 mm
4	Pedal movement, d_{pm}	20 mm
5	Diameter of master cylinder, d_c	26 mm
6	Piston spring pre-tension, F_{csp}	15N
7	Piston spring rate, K_{sc}	8 N/mm
8	Wheel diameter, d_w	625 mm
9	Caliper piston, d_{cp}	46 mm
10	Number of pads	2
11	Coefficient of friction, μ_c	0.35
12	Diaphragm diameter, d_D	220 mm
13	Valve body diameter, d_v	52 mm
14	Diaphragm spring pre-tension force, F_{Bsp}	50 N
15	Ambient pressure, P_{atm}	82 kPa
16	Manifold pressure, P_{vacuum}	41 kPa

IV. DESIGN OF THE DIAPHRAGM SPRING

A single diaphragm spring in the diaphragm spring clutch cover replaces a multitude of components from its immediate predecessor, the coil spring and lever –type clutch cover in figure2. This reduced complexity gives the diaphragm spring clutch a cost, manufacturing and weight advantage over previous types.

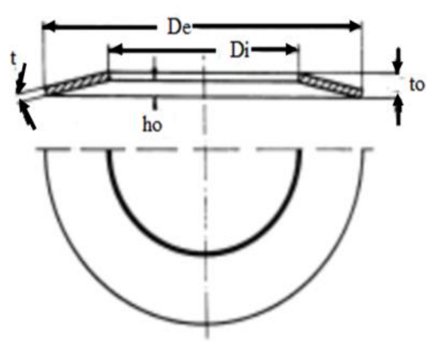

Fig2. Diaphragm Spring.

Also the diaphragm spring's characteristic, shallow cone shape and high loads generated through small displacements make possible a compact package with high torque transmitting capabilities. This configuration is known as a diaphragm spring.

$$D_{actual} = D_i - (2 \times s \times \sin\theta) \quad (16)$$

Where,

D_{actual}= Actual inner diameter (mm)
D_i= Inside diameter (mm)
s= Deflection (mm)

$$\delta = \frac{D_a}{D_{actual}} \quad (17)$$

Where,

δ= Diametric ratio
D_a= Outside diameter (mm)
D_{actual}= Actual diameter (mm)

$$P_{flat} = \alpha \frac{st^3}{D_a^2} \quad (18)$$

Where,

P_{flat}= Load at flat condition (N)
α= Coefficient (N/mm^2)
t= Thickness of individual spring (mm)

$$P_{ful} = P_{flat} \times \frac{D_a - D_i}{D_{a1} - D_{i1}} \quad (19)$$

Where,

P_{ful}= Load at flat with fulcrum (N)
D_{a1}= Outer diameter of clutch point relying (mm)
D_{i1}= Internal diameter of clutch point relying (mm)

Table III. Design Specification of the Diaphragm Spring

No.	Name	Value
1	Outside diameter, D_a	200 mm
2	Inside diameter, D_i	160 mm
3	The thickness of individual spring, t	2.5 mm
4	Free one height of the unloaded individual spring, h	5.670 mm
5	Deflection, s	5.573 mm
6	Angle, θ	45 degree
7	Outer diameter of clutch point relying, D_{a1}	196.05 mm
8	Internal diameter of clutch point relying, D_{i1}	163.94m

A. Design of K_1,K_2 and K_3

The stiffness k, of a body is a measure of the resistance offered by an elastic body to deformation. Every object in this universe has some stiffness. Generally for spring the spring stiffness is the force required to cause unit deflection.

$$K_1 = \frac{1}{\pi} \frac{\left(\frac{\delta - 1}{\delta}\right)^2}{\left(\frac{\delta + 1}{\delta - 1}\right) - \frac{2}{\ln(\delta)}} \quad (20)$$

Where,

K_1= Constant point (1)
δ= Diametric ratio

$$K_2 = \frac{1}{\pi} \times \frac{6}{Ln(\delta)} \times \left[\frac{(\delta - 1)}{Ln(\delta)} - 1\right] \quad (21)$$

Where,

K_2 = Constant point (2)

$$K_3 = \frac{1}{\pi} \times \frac{6}{\ln(\delta)} \times \left(\frac{\delta - 1}{2}\right) \quad (22)$$

Where,

K_3= Constant point (3)

B. Design of Spring Stress

Most compression spring is a straight cylindrical spring made of round wire. Spring stress levels are determined by the dimensional limits together with the load and deflection requirements.

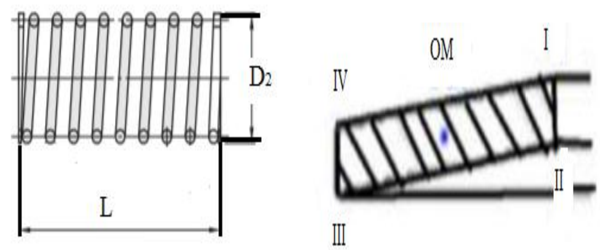

Fig 3: Cross-Section of Diaphragm Spring.

The compression springs are stress relieved to remove residual bending stresses produced by the coiling operation. These are springs designed with torsional stress levels when compressed solid that the minimum tensile strength of the material. As shown in figure 3.

$$\sigma_{OM} = \frac{-3}{\pi} \times \frac{4E}{(1 - \mu^2)} \times \frac{s \times t}{K_1 \times D_a^2} \quad (23)$$

Where,

σ_{OM}= Spring stress at point OM (N/mm^2)
E = Young's modulus (G Pa)
D_a= Outside diameter (mm)

$$\sigma_I, \sigma_{II} = \frac{-4E}{(1 - \mu^2)} \times \frac{s \times t}{K_1 \times D_a^2}\left[K_2\left(\frac{h}{t} - 0.5 \times \frac{s}{t}\right) + K_3\right] \quad (24)$$

Where,

σ_I= Spring stress at point (I)
h = Free once height of the unloaded individual spring (mm)
σ_{II}= Spring stress at point (II)

$$\sigma_{III} = \frac{-4E}{(1 - \mu^2)} \times \frac{s \times t}{\delta K_1 \times D_a^2}\left[(2K_3 - K_2)\left(\frac{h}{t} - 0.5 \times \frac{s}{t}\right) + K_3\right] \quad (25)$$

Where,

σ_{III}= Spring stress at point (III)

$$\sigma_{IV} = \frac{-4E}{(1 - \mu^2)} \times \frac{s \times t}{\delta K_1 \times D_a^2}\left[(2K_3 - K_2)\left(\frac{h}{t} - 0.5 \times \frac{s}{t}\right) - K_3\right] \quad (26)$$

Where,

σ_{IV}= Spring stress at point (IV)

Table IV. Result Data of Without Brake Booster for One Port Caliper

Descriptions	Symbols	Values	Units
Pedal ratio	L_p	5.66	-
Master cylinder piston movement	d_{cm}	8.127	mm
Driver applied force	F_{dapp}	452.8	N
Master cylinder piston force	F_c	372.784	N
Master cylinder piston pressure	P_c	0.702	MPa
Caliper force	F_{cp}	1166.88	N
Transmitted force	F_t	816.81	N
Torque	T	245.04	Nm

TABLE V. RESULT DATA OF DIAPHRAGM SPRING

Descriptions	Symbols	Values	Units
Actual inner diameter	$\mathrm{D_{actual}}$	152.11	mm
Diametric ratio	δ	1.3148	-
Load at flat condition	$\mathrm{P_{flat}}$	195.153	kgF
Load at flat with fulcrum	$\mathrm{P_{ful}}$	243.1	kgF
Constant point 1	$\mathrm{K_1}$	0.67	-
Constant point 2	$\mathrm{K_2}$	1.07	-
Constant point 3	$\mathrm{K_3}$	1.10	-
Spring stress at point OM	σ_{OM}	-44.5	$\mathrm{kgF/mm^2}$
Spring stress at point I	σ_{I}	-108.58	$\mathrm{kgF/mm^2}$
Spring stress at point II	σ_{II}	-108.58	$\mathrm{kgF/mm^2}$
Spring stress at point III	σ_{III}	-85.06	$\mathrm{kgF/mm^2}$
Spring stress at point IV	σ_{IV}	-7.0	$\mathrm{kgF/mm^2}$

TABLE VI. RESULT DATA OF WITH BRAKE BOOSTER FOR ONE PORT CALIPER

Descriptions	Symbols	Values	Units
Pedal ratio	$\mathrm{L_p}$	5.66	-
Master cylinder piston movement	$\mathrm{d_{cm}}$	3.533	mm
Diaphragm pressure	$\mathrm{P_D}$	41	kPa
Diaphragm area	$\mathrm{A_D}$	35.88×10^{-3}	$\mathrm{m^2}$
Atmospheric force	$\mathrm{F_{atm}}$	2942.16	N
Intake manifold force	$\mathrm{F_{vacuum}}$	1471.08	N
Diaphragm force	$\mathrm{F_D}$	1471.08	N
Diaphragm spring rate	$\mathrm{K_{sB}}$	58	N /mm
Driver applied force	$\mathrm{F_{dapp}}$	1299.36	N
Master cylinder piston force	$\mathrm{F_c}$	1255.96	N
Master cylinder piston pressure	$\mathrm{P_c}$	2.36	MPa
Caliper force	$\mathrm{F_{cp}}$	3931.5	N
Transmitted force	$\mathrm{F_t}$	2752	N
Booster output force	$\mathrm{F_B}$	1186.16	N
Torque	T	825.6	Nm

VII. CONCLUSION

The hydraulic brake system is important for the high speed car. The master cylinder is the heart for the hydraulic brake system. The brake pedal actually operates four separate hydraulic lines running to all four wheels. Instead of a single cylinder, there's usually one main cylinder operated by driver foot and the brake pedal and then one secondary cylinder on each wheel. The brake booster is very important in defining the brake pedal force characteristics. The booster internal component characteristics can be turned to create significant effects on brake pedal 'feel'. So, the brake booster reduces driver applied force. When the comparison of the without brake booster and with brake booster in one car, without the brake booster is less brake power than with the brake booster. The brake booster reduces the driver applied force 80N to 20N and the pedal movement 46mm to 20mm.The brake booster is good for the brake power. The brake booster force depends on the diaphragm area. If the diaphragm area is wide, the different between atmospheric pressure and vacuum pressure more apply on the diaphragm. So, the diaphragm wide area booster is better than the diaphragm less area booster for the brake booster force.

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