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Simulation of Brake Pressure Multiplier (BPM) through ANSYS 14.0 For Effective Braking in ATV

Ronak Bandil[2], Anand Baghel[1], Akash Singh Parihar[2], Shubham Kumar Verma[2], Vikas Singh[2]

[1]Faculty, RJIT Tekanpur, [2]B. Tech Scholar, RJIT Tekanpur

ABSTRACT

The requirement of brake systems have become increasingly complex in recent years due to the expansion of global markets and diversification of the conditions under which vehicles are used in different parts of the world. It is also becoming increasingly important to ensure that vehicles offer the secure and better braking effect which are expected by drivers, but can also provide less aggressive feel to driver while severe braking. For providing these requirements, we have introduced a new brake booster which is completely hydraulic named **BRAKE PRESSURE MULTIPLIER**. We designed it with keeping all the requirements of brake system and driver in mind; it multiplies the driver force up to a limit which is required for the brakes to stop a vehicle in minimum distance and also with less aggressive feel to driver. This booster is completely hydraulic so no external sources are required to run this booster like conventional brake booster in which engine is used to produce vacuum or air pressure for working of booster. The chances of condition of brake fail is also reduced in ours design because of no hose pipes used in this like in conventional boosters, also the weight of whole assembly is very less than the conventional ones.

Keywords: **–** Brake Pressure Multiplier, booster, braking effort

I. Introduction

In today's conventional brake boosters we have to use engine for creating vacuum hence for overcome this effect We introduce a concept of *HYDRAULIC BRAKE BOOSTER* in which brake force is multiplied with the help of hydraulic mechanism not with the help of atmospheric pressure or vacuum generated by engine.

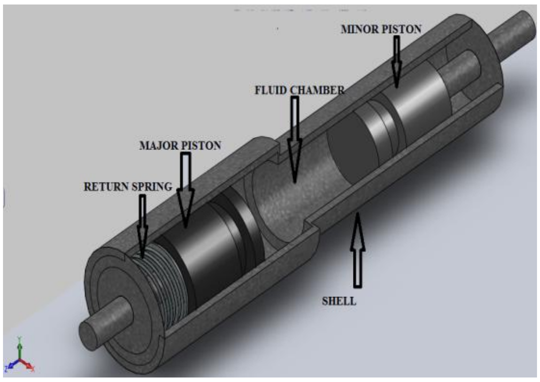

Fig 5.1 Brake pressure multiplier with all the major components

II. Principle

This works on general law of fluid mechanics PASCAL'S LAW which states that "Pressure exerted anywhere in a confined incompressible fluid is transmitted equally in all direction throughout the fluid such that the pressure variations (internal differences) remain the same". Also we know that the pressure is equal to the ratio of force to the area over which that force is distributed.

Hence

$$\boxed{\text{Pressure} = \text{Force/Area}}$$

III. Working

In the Brake Pressure Multiplier there is variable cross section shell made up of aluminium alloy inside which there are 2 pistons of different diameters. As this assembly is mounted between the brake pedal and master cylinder the 2 pistons are connected with these two, out of which the smaller diameter piston is connected with the brake pedal with the help of push rod and bigger piton is connected with the master cylinder by the push rod, and between these 2 pistons fluid (dot 3) is filled.

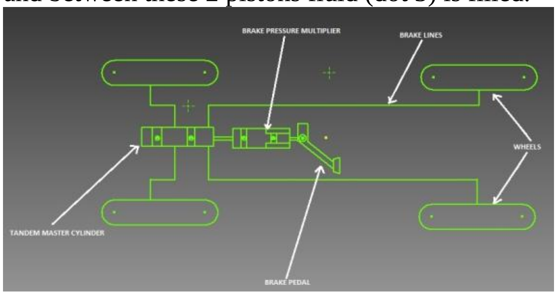

Fig 5.2 Brake hydraulic circuit along with brake pedal & brake pressure multiplier.

Hence when we applied force on the brake pedal the smaller piston displaces and forces the fluid and hence fluid displaces the bigger piston but due to difference in their areas the force transmitted by bigger piston to the master cylinder is more than the actual pedal force hence the force is multiplied. And the ratio of input and output forces is equal to the ratio of smaller and bigger diameter.

IV. Calculation

Let we assume the driver force or the force acting on small piston is, $F_1 = 100$ N
And diameter of small piston, $D_1 = 2.85$ cm
Diameter of large piston, $D_2 = 3.8$ cm
Hence area of smaller piston $A1 = \pi r^2$
$= 6.37\ cm^2$
Area of large piston $A_2 = \pi r^2$
$= 3.14*(3.8)^2/4$
$= 11.33\ cm^2$
We know that from Pascal's law the pressure applied on each piston is equal hence
$\mathbf{P_1 = P_2}$
And pressure P= F/A
So, $F_1/A_1 = F_2/A_2$
$100/6.37 = F_2/11.33$
$F_2 = 177.86$ N

Hence we get that the force applied on the master cylinder will be 177.86 N which **means that force is multiplied by 1.77 times or increased by 77%.**

As the only connection between the two pistons is only fluid hence the displacement of both pistons is different because of their different diameters, we can calculate the displacement by following method.
The maximum displacement of push rod in master cylinder is 3.5 cm hence the displacement of large piston should be equal to this i.e. 3.5 cm
Now we know that volume of fluid is constant throughout the time hence
$A_1*L_1 = A_2*L_2$
Where L_1& L_2 are the displacement of small and large piston respectively
$6.37*L_1 = 11.33*3.5$
$L_1 = 6.2$ cm
From above result we got that we displacement of small piston will be more than the large piston so we have to displace the small piston more for complete travel of push rod in master cylinder.

V. Design

During the operation of brake pressure multiplier the magnitude of acting forces and pressure is very high and fluctuating so to validate our design we have analysed all the main components of brake pressure multiplier in. ANSYS 14.0 work bench. Following are the results of simulation performed on different parts.

5.1 SHELL

Shell is not the working part of Brake Pressure Multiplier but still it is suffered from higher pressure and forces. During operating condition of brake pressure multiplier it is suffered from internal pressure of fluid which is inside the shell of brake pressure multiplier. The calculation of pressure and the simulation of shell on that pressure is given as follows
Maximum tensile strength of material (aluminium 1016) = 83 Mpa
Then circumferential stress σ_c = maximum tensile strength / factor of safety $\sigma_c = 83/7 = 11.85$ Mpa
We take the thickness of shell, t= 3 mm
Because the thickness of shell is less with respect to the diameter and length of shell we can consider this as thin cylinder, hence by the property of thin cylinder
$t = P*D_1 / 2\sigma_c$
$P = 2t\,\sigma_c / D_1$
Where P is the internal pressure in the shell. We calculate the maximum pressure from the smaller diameter section because the pressure in the large diameter section is less than the previous section.
P = 2.5 Mpa

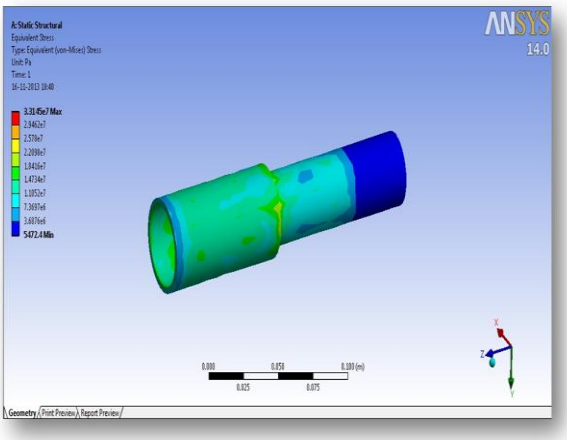

Fig 6.1 (a) Stress diagram of shell

This is the von mises stress diagram of shell. Here the max stress producing is 33 Mpa which is far below than ultimate tensile strength of our material. Hence we conclude that our design is safe & do not break or crack under such worst loading.

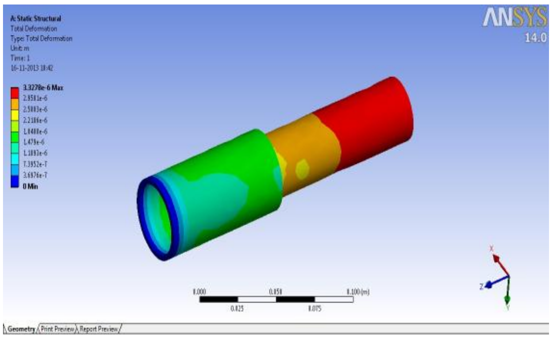

Fig 6.1 (b) Deformation diagram of shell

This is the displacement diagram of our shell. Here the max displacement produced is .0033mm which is very small in size & can easily be neglected. Therefore we conclude that our shell structure is capable of bearing such worst loading with a very small deformation.

5.2 SMALL PISTON

Pistons of brake pressure multiplier are also suffered from severe forces and pressure, we see that the shell of BPM is only suffered from pressure only but piston will suffered from both pressure and force, the magnitude of both forces and pressure can be calculated as follows
Let consider the driver force be 100 N
Since due to pedal ratio which is taken as 4:1,
hence the input force= 400 N
The piston is suffered by the driver force from one side and suffered from internal pressure of fluid from other side that pressure can be calculated as
Pressure P = Force /πR_1^2
P = 0.627 MPa
We analyse the piston by considering the force of 400 N on the outer side and pressure of 00 .627 MPa on the inner side of piston.
The material we chose for the piston is Aluminium alloy so that it will be of light weight and of good strength. The length of piston we take is 4 cm so that it can withstand severe forces and high pressure. A slot is cut in the piston for provide a better seal for preventing leakage.

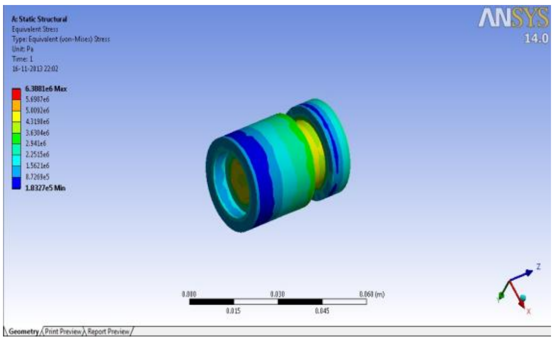

Fig 6.2 (a) stress diagram of minor piston

This is the stress diagram of our minor piston or sender piston. Here the max stress generated is 6.3Mpa. On dividing tensile strength with this stress we get appreciable factor of safety which proves that our design is safe.

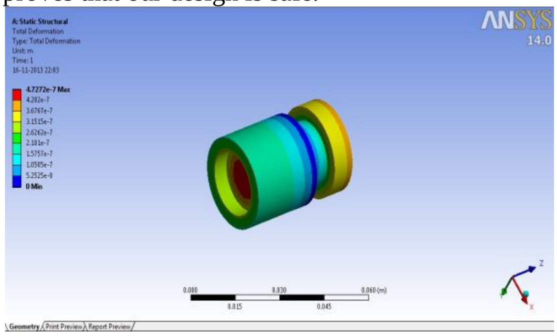

Fig 6.2 (b) Deformation diagram of minor piston

This is the deformation diagram of our minor piston. Here the max deformation produced is .0004mm. which is negligible & can easily be neglected & also the area where this deformation is produced is not much of greater use.

5.3 LARGE PISTON

Pistons of brake pressure multiplier are also suffered from severe forces and pressure, we see that the shell of BPM is only suffered from pressure only but piston will suffered from both pressure and force, the magnitude of both forces and pressure can be calculated as follows
Because the output force is increased by 77% hence the force on outer side of piston will be 708 N
The internal pressure will be the same as we calculated above i.e. .627 MPa hence we analyse the piston by considering the force of 708 N at outer surface and pressure of .627 MPa at inner surface.

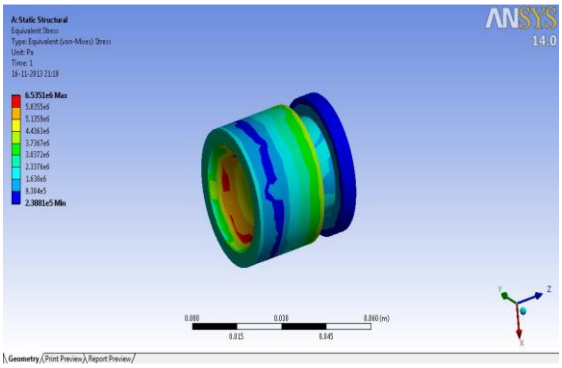

Fig 6.3 (a) Stress diagram of major piston

This is the equivalent stress diagram of our major piston .in this max stress generated is 6.35Mpa which is easily acceptable.
We get an appreciable factor of safety which concludes that our design is safe.

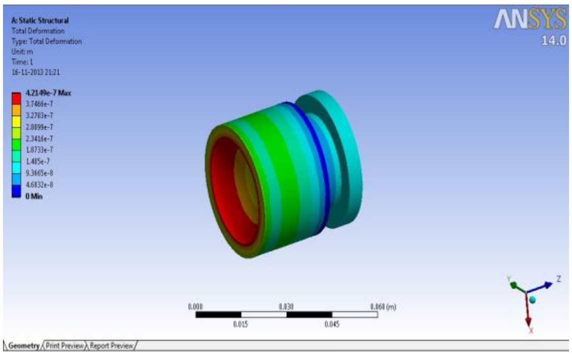

Fig 6.3 (b) Deformation diagram of major piston

This is the displacement diagram of our major piston. In this the max deformation produced is .00042mm. which is quite acceptable.

Its not only concludes that our design is safe, but also concludes that our design is capable in working such worst condition without any failure.

VI. Effect Of Brake Pressure Multiplier On Brakes

Consider a vehicle having mass of 800kg.so following are the initial data about this vehicle required for calculation.

Gross weight of vehicle = 800 *10
= 8000 N

Centre of gravity height from ground (h) = 40cm

Wheelbase (l) = 160cm

Deceleration rate (a) = 1g

Since the vehicle is rear engine rear wheel drive, so the weight on rear wheels is more than weight on front wheels.

Let us assume that weight distribution is 40:60.i-e 40% on front & 60% on rear.

So weight distribution is 3200 N on front wheels & 4800 N on rear wheels.

Now we calculate the DYNAMIC WEIGHT TRANSFER

For front wheels it is = $W_{front} + W_{total}*(h/l)*(a/g)$

For rear wheels it is = $W_{rear} - W_{total}*(h/l)*(a/g)$

On calculating dynamic weight transfer for front & rear wheels are as follows

For front wheels = 5200N

For rear wheels = 2800N

This is the required force that requires for a vehicle to stop.

Individual force required for each wheel is

For front wheel = 2600 N

For rear wheel = 1400 N

Since the tyre diameter is of 23 inch

Hence tyre radius is 11.5 inch or 29.21 cm

Brake torque required for front wheel= 759.46 N-m

Brake torque required for rear wheel= 408.94 N-m

This is the required brake force & brake torque requires stopping a vehicle.

Now generating brake force & torque without any external aid (i-e brake pressure multiplier or Hydraulic brake booster.)

In worst condition, when the brake is suddenly applied, it is assumed that the max. Input force which is applied by the driver is 100N

Since the pedal ratio is 4:1, the input force which is going to the master cylinder =400N

Since the master cylinder diameter is 19.05 mm. the pressure generated inside the master cylinder is
= force/ area
= 1.40 Mpa

Now the clamping force or generating force generated by the master cylinder is
= master cylinder pressure* calliper piston area
= 1939.61 N

Since the generated brake force is less than required brake force. Hence brake will not be applied or you can say brakes are failed.

Now brake force generated with the help of brake pressure multiplier.

Since we know that from the above experiment it is clear that it increases the efficiency up to 77%.

So the input force= 708 N

Now the brake pressure generated inside master cylinder = $708/(3.14*9.525^2)$
=2.49Mpa

Now clamping force generated = $2.49*3.14*(21)^2$
= 3449.75 N

Since the clamping force is more than required hence brakes are applied.

Brake torque will be generated by = 3449.75*.23
= 793.44 N-m

Since the clamping force & torque are more than required.

VII. Advantages Of Brake Pressure Multiplier

1. It increases the brake pressure by **77%** hence the stopping distance of vehicle is reduced.
2. Since due to hydraulic medium the power loss between pistons is very less i.e. 1-2%
3. Does not require any external power source for operating.
4. Constant power output throughout his operation.
5. Compact size.
6. Long lasting Since the factor of safety is more than 7 so its operating life will be more.
7. Because of having lees no of parts its manufacturing cost is low.
8. As the whole assembly is made up of Aluminium hence the weight of assembly is low.
9. Services & maintenance cost are very less.

10. Since the medium used between the pistons is hydraulic, so wear and tear of pistons is less.

VIII. Conclusion

Hence by calculation stated above we can conclude the result that without using the brake booster it is very difficult to stop the vehicle because the brake force generated by the master cylinder is not enough for stopping the vehicle because the force generated on each wheel while propulsion is much higher than the brake force. So the need of brake boosters is obvious in vehicles and the need is increased as the weight of the vehicle is increased.

But as we stated above the conventional brake boosters has their own disadvantages so we can’t use conventional brake boosters hence our design of brake pressure multiplier easily replace conventional brake boosters and fulfil the requirement of brake system.

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