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Design and Analysis of a Novel Y-Type Hydraulic Braking System for Implementation in ATVs

Kunal Mathur
Mechanical Department, Delhi Technological University (DTU), Delhi, India

Lalit Kumar Choudhary
Mechanical Department, Delhi Technological University (DTU), Delhi, India

Krovvidi Srinivas
Mechanical Department, Delhi Technological University (DTU), Delhi, India

***Abstract* - The aim of this paper is to design, develop and validate a 'Y-type' dual split hydraulic braking system for proper implementation in certain production based ATVs and other vehicles with similar specifications. This novel braking system is supposed to be inherently safer than the pre-existing ones being used by different ATVs. Various design approaches are followed in a calculative fashion along with some sound market selection strategies to develop and implement this braking system. Some braking components are designed and analysed using CAD and CAE software respectively while other components are selected from market to enhance standardization in the braking system. Components have been designed against static, thermal and fatigue loading. The entire braking system is validated for multiple ATV models manufactured by different OEMs using vehicle dynamics equations followed by systematic torque comparison and conformance with the government safety regulations.**

Index Terms – Ansys, ATV, FEA, NPT, SolidWorks, Y-type dual split hydraulic braking system.

INTRODUCTION

ATVs (All Terrain Vehicles) are used for multiple purposes such as recreation, motorsports or as commercial passenger vehicles. For such a wide range of customers of this product, occupant safety during dynamic conditions such as braking is a crucial need. This braking system is capable of locking a minimum of three wheels during a single circuit failure. Hence the brake circuit design provides a much safer and a reliable method of locking the wheels during a single brake circuit failure by reducing the chances of vehicle skid considerably.

In this paper, a Y-type dual split hydraulic braking circuit has been designed and analysed for failure in order to install in a particular group of ATVs. Unlike the common X-type or the horizontal split type circuit, this braking circuit locks three wheels in case of a single circuit failure, hence reducing the chances of skidding while braking. This sole factor makes the Y-type circuit a safer and a reliable option. A typical Y-type braking system involves each circuit to connect with three wheels/calipers at a time when two of the wheels/calipers belong on one axle (Figure 2). Specific ATVs are selected with similar specifications in this paper. The proposed Y-type dual split hydraulic braking system is designed for easy implementation, quick assembly/disassembly in such ATVs using standard available market brake components and connectors which are thoroughly selected according to their functions and compatibilities. This makes the system design highly modular, standardised and reduces the overall cost during a mass production process. The compatibilities of these components were confirmed after a series of calculations. The custom components are designed in SolidWorks® and analysed for the dynamic environment that the vehicle would be subjected to using CAE (Computer Aided Engineering) tools of Ansys 17.1®. These components are analysed for deformations, strength, fatigue life, thermal stresses and thermal fatigue. Possible alterations in the system that could occur in the implementation of this circuit design into ATVs not belonging to the group mentioned above are also discussed in this study. The entire developed braking system is also tested for its conformability with the Indian government CMV Rules and standards.

Various research papers in the similar field are studied and referred for this study. The study by Jain PK et al. in [1] is about the development of a racing Go-kart in which the design and analysis of the braking system has been done. Multiple braking factors such as aerodynamic drag and rolling resistance are not taken into account for the deceleration effect. Moreover, the design of the pedal is not analysed for lateral deformation, which is required as the driver exerts an appreciable amount of force on the pedal with his/her foot in the lateral direction while exiting the vehicle. This factor has been taken into account in the current study. The selection of driver pedal effort is abstract in the studies of Jain PK et al. in [1] and Sharma A et al. in [2] and are not according to any anthropological survey standards. Mishra D et al. in [3] describes the design and analysis of the braking system of an ATV for Baja SAE. The use of a balance bar has been made which requires critical adjustment and its implementation according to the desired brake bias is often time consuming. We have deployed the standard proportioning valves that can be purchased as a whole unit in bulk for industrial mass production needs. These are relatively easy to operate and set. Moreover, the use of four different calipers has been done in [3] which offers difficulty in tuning the brakes for all wheels to lock simultaneously without skidding. Hence, the braking system developed by us has deployed a total of four disc brakes with same calipers. Shrivastava D. [4] and Sharma A et al. [2] have implemented the use of drum brakes in an ATV for Baja SAE, which are bulkier and have a lot of drawbacks as compared to the disc brakes. The work of Gupta US et al. [5] is based on the same study as that of Shrivastava D. [4] but the braking system of the ATV in [5] has been designed setting the frictional and the braking torques equal in magnitude. In practice, the braking torque would always be lesser in magnitude as compared to its computed value, owing to the irregularities in the braking fluid that violate Pascal’s Law. Due to this reason, the braking torque has been kept always greater than the frictional torque in this paper.

DESIGN METHODOLOGY AND COMPONENT SELECTION

I. Selection of ATVs

The following ATVs, having similar GVW (Gross Vehicle Weight), axle weight distribution, tire diameters, maximum velocity, hydraulic disc brakes and maximum number of passengers (between one and two) have been selected.

- **ATV-1:** Polaris ACE 500®
- **ATV-2:** 2011 Polaris Ranger 500 EPI®
- **ATV-3:** Polaris ACE 570 SP®
- **ATV-4:** 2018 Kymco UXV 450i CAMO®
- **ATV-5:** 2018 Bennche Spire 150®

FIGURE 1
ATV-1 (POLARIS ACE 500) [6]

II. Brake circuit type

For modern front-wheel-drive cars the X-split circuit appears to have become an established practice, while the simple front-rear split circuit is still generally used on rear wheel drive cars. In off road vehicles usually ATVs or All-Terrain Vehicles, Y-split circuit should be used with the help of four disc brakes for efficient braking. This system is widely used nowadays in racing arena and SAE events for students. Y-split circuit is best suited for the ATVs [7]. Hence Y-type split hydraulic dual circuit is chosen for implementation in the selected ATVs. In this type of circuit design, two completely independent brake line circuits operate simultaneously, each having their own separate reservoirs. One circuit is attached to a disc D1 at front and both the discs (D3 and D4) at the rear of the vehicle. The other circuit connects D4 with both the discs at front (D1 and D2) as shown in figure 2. Hence in case of a single brake circuit failure due to leakage or contamination of brake fluid, at least three wheels would get locked, thereby diminishing the chances of vehicle skidding while braking.

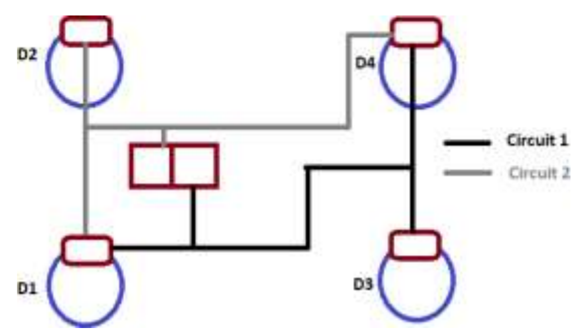

FIGURE 2
Y-TYPE DUAL SPLIT BRAKE CIRCUIT

III. Brake system components

The braking system for the ATVs will contain the following elements:

- Master cylinder and reservoirs (selected)
- Brake calipers (selected)
- Proportioning valves (selected)
- Unions/connectors (selected)
- Brake fluid (selected)

- Hoses (designed; only length)
- Brake pedal (designed)
- Brake pads (selected)
- Brake discs (designed)

At first, the selection of the standard selected components that are used in the braking circuit have been discussed below.

(a) Master Cylinder

Tandem master cylinders are preferred over single master cylinders for the following reasons. First, the tandem master cylinders have two separate piston chambers with two different inlet and outlet ports. Secondly, in order to implement a dual hydraulic braking system with independent circuits, one would have to use two single master cylinders which would have acquired space. This factor is eliminated by having one tandem master cylinder.

Wilwood's TM-1 tandem master cylinder is selected for its following features [8]:

- Lower cost
- Light weight since it is made of aluminium
- It comes with two separate reservoirs with respective mount holes. Both can be mounted on the cylinder itself.
- It can be mounted horizontally, thereby decreasing vertical space at the front of the vehicle.
- It comes with an attached push rod with fork ends for holding the pedal and transferring the pedal effort to the cylinder easily.
- It has a suitable stroke length that would easily accommodate the pedal travel by the driver while braking.

FIGURE 3
WILWOOD TM-1 MASTER CYLINDER [8]

(b) Brake Calipers

Calipers are used to support the brake pads that arrest the motion of the rotor to cause braking. They also support the pistons that transmit the force from the fluid to the pads. Wilwood's PS-1 caliper with one inch pistons is selected for the following reasons [8]:

- Compact body with appreciable amount of clamping force
- Light weight (aluminium body)
- Lower cost
- Different designs of the same product are available both for left and right wheels of the vehicle simplifying brake bleeding. Hence the use of different calipers for different wheels is not required. For this reason, PS-1 is selected for all the four wheels.
- Utilizes two stainless steel deep cup pistons to minimize heat transfer from the pads to the brake fluid.
- Only one bleeding valve is present on the caliper. The lesser number of leakage points decreases the chances of brake fluid leakage.
- Bleeding valve faces upwards for easy bleeding [3].
- The allowed thickness of the disc is lesser, which simplifies the manufacturing of the brake disc.

FIGURE 4
WILWOOD PS-1 CALIPER (RIGHT SIDED) [8]

(c) Proportioning Valves

Proportioning valves are selected over a balance bar to provide brake bias for the following reasons:

- Easy to use and install
- Balance bar requires the use of two single master cylinders.
- Balance bar implementation is not possible in a Y-type brake circuit since each circuit is connected to the wheels of both the axles at a time.

Wilwood's proportioning valve with M10x1 NPT (National Pipe Thread) inlet and outlet is selected for the following reasons [8]:

- They are much more compact, reliable and cheaper as compared to other companies.
- Ports have fine and tapered threads that prevent leakage of the pressurized fluid.
- Ports with M10x1 NPT are also used in the selected TM-1. Hence union connections of the same type could be used throughout the braking circuit. This reduces the number of different components used in the system, making it much more economical for bulk orders in a mass production process.
- Control in the amount of brake bias is achieved by just turning the knob of the proportioning valve (easy to use).
- Two lateral bolt holes are available for easy mounting.

FIGURE 5
WILWOOD M10X1 PROPORTIONING VALVES [8]

(d) Unions

Four types of unions/connectors are used to connect the different braking components of the circuit. They are as follows:

- **Simple unions:** have one inlet and outlet for connecting two brake components together. An M10x1 NPT union with both male threads is used here to join brake hoses with proportioning valves and master cylinders. Another union having one end as above and the other as male 1/8-27 NPT is used to join one brake hose and one caliper.
- **Tee unions:** are used to connect three brake circuit components for the transfer of pressurized brake fluid. One type of Tee connector has all three male M10x1 NPT connections for joining the corresponding female thread hoses. The other type of Tee union has one male 1/8-27 NPT and rest two M10x1 NPT male threads for connecting two brake hoses with a caliper (figure 2).
- **Cross unions:** used to connect four braking components for proper brake fluid transfer. Cross unions having four M10x1 NPT male threads have been used to connect four female thread brake hoses with each other.

FIGURE 6(A)
SIMPLE UNION [8]

FIGURE 6(B)
TEE UNION [8]

(e) Brake fluid

Brake fluids are used to transmit pressure effectively throughout the brake hoses and connectors all the way up to the caliper pistons which arrest the motion of the brake disc by pushing the pads against it. Following are the required qualities of brake fluid [9]:

- High boiling
- Low freezing point
- Ability to not deteriorate metal or rubber brake parts
- Ability to absorb moisture

TABLE I
COMPARISON OF BRAKE FLUIDS [9]

DOT 3	DOT 4	DOT 5
Absorbs moisture the quickest	Lower absorbing power than DOT 3	Does not absorb moisture
Lowest boiling point of 205°C	Boiling point 230°C	Highest boiling point 260°C
Restricted for road use only	High temperature tolerance	Used in heavy vehicles
Lowest cost	Moderate cost	Highest cost

DOT 3, DOT 4 and DOT 5 are compared for the above properties. DOT 4 is chosen for the braking system after comparison on the basis of the above properties.

(f) Brake hoses

Standard flexible steel braided rubber hoses are to be used in our system. This prevents physical damage to the hose to a great extent and imparts flexibility to the brake lines for allowing them to connect at intricate corners in the vehicle. The length of these hoses can be customized by the hose supplier easily. Hoses with M10x1 NPT female threads would be used all across our circuit.

(g) Brake Pedal

It is a linkage that is used to provide mechanical advantage to the driver's pedal effort and transfers the force. The pedal force multiplied by the leverage ratio is applied to the master cylinder to produce a hydraulic pressure [10].

Class-2 lever mechanism (where the loads are at the same side of the pivot point) is chosen so as to push the pushrod in the same direction as the pedal travel. This would enable the OEM to install the master cylinder assembly behind the pedal, giving the driver room for his feet, hence preserving the ergonomic aspect of the vehicle. The design of the pedal is made in SolidWorks 2017®. The pedal would be pivoted on an extruded boss attached to its bracket and retracted with the help of a torsion spring. A small mount for the stop plug would be welded on the arm of the pedal so that the stop plug would restrict the pedal travel according to the pedal ratio and the maximum stroke of the master cylinder. The stop plug of the pedal arm would interfere with that on the bracket which would arrest further travel of the pedal. Figure 7(a) depicts the CAD model of the pedal.

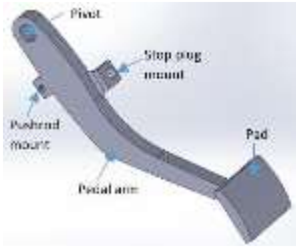

FIGURE 7(A)
CAD MODEL OF BRAKE PEDAL

Results based on FEA (Finite Element Analysis) using Ansys 17.1® helped to modify the design according to the software simulated conditions which are made very similar to that of the real dynamic environment of the vehicle.

For the 3D model of the pedal, two of the dimensions are very large in comparison to the third one, hence 2D meshing is carried out in the software [11]. The mesh model contains a fine mesh with 'quad' and 'tria' elements for accurate results [11]. The types of analysis performed on the mesh model are listed below:

- Deformation and vonMises stress on pedal effort
- Deformation and vonMises stress on lateral loading
- Fatigue analysis

The above analysis are completed after the input of required boundary conditions that are computed in the calculations section. Moreover, the pedal analysis are done when the pedal is in its maximum travel position (fully depressed).

Deformation and vonMises stress on pedal effort: Pedal effort (PE) is the normal force exerted by the driver on the pedal while braking. PE equals to 850N (calculated in the next section). The pivot hole of the pedal is set as a constraint while the load of 850N is applied normal to the pad of the pedal.

Aluminium-6061 is selected as the material of the pedal (owing to its light weight and satisfactory strength) and corresponding values are entered in the software. After the generation of the FE model, the software displays the deformation and stress contours.

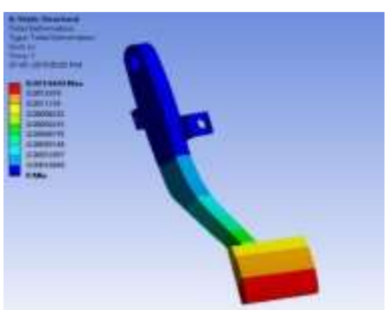

FIGURE 7(B)
TOTAL DEFORMATION UNDER NORMAL LOAD

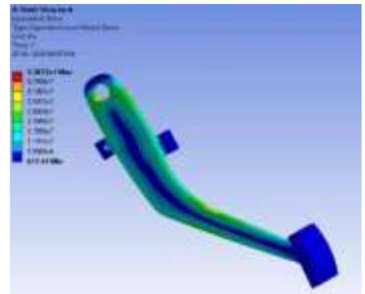

FIGURE 7(C)
VONMISES STRESS DISTRIBUTION FOR NORMAL LOAD

From figure 7(c), it can be concluded that the maximum equivalent stress comes out to be around 54MPa, whereas the yield strength of aluminium-6061 is 240MPa. Hence factor of safety (FOS) equals 4.44 which is suitable indeed.

Deformation and vonMises stress on lateral force: When the driver exits a vehicle, he/she might accidently apply some force on the lateral edge of the pedal. Hence the pedal is analysed for failure under lateral load. A lateral load of 300N is entered at the left edge of the pedal (assuming a right-hand drive vehicle). The constraint and material are same as above and are the boundary conditions of this FE model.

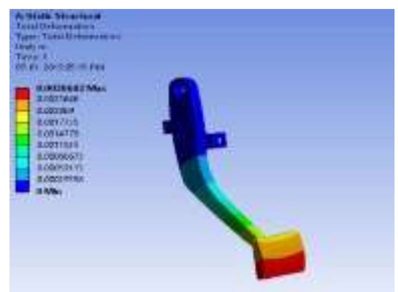

FIGURE 7(D)
TOTAL DEFORMATION FOR LATERAL LOAD

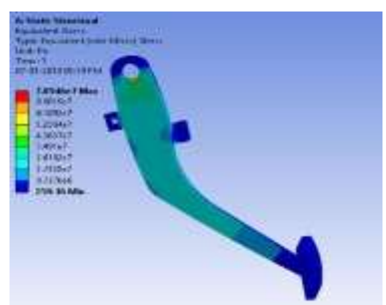

FIGURE 7(E)
VONMISES STRESS DISTRIBUTION FOR LATERAL LOAD

From figure 7(e), it is evident that the maximum vonMises stress in the pedal design upon lateral loading comes out to be around 78.6MPa. Hence the factor of safety FOS equals 3.0 which is acceptable.

Fatigue analysis: Since the proposed braking system has been designed for ATVs for public use, fatigue analysis is done to ensure the desired life of such components under prolonged load applications. Although the S-N curve of the chosen material does not have a well-defined knee point, after the input of the boundary conditions the design is found to be safe till 10^8 load cycles. The loading conditions entered are 0-850N of alternating loads and same constraints as those in the previous cases. S-N curve data of Al-6061 is also entered. The design is found to have a factor of safety (FOS) of 1.427 which is the ratio of the endurance strength according to 10^6 cycles and the maximum equivalent alternating stress. This suggests that the pedal design would not fail at given alternating loads before 10^8 such cycles which is more than favourable.

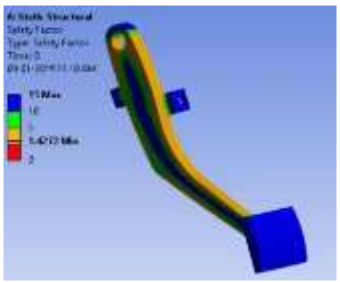

FIGURE 7(F)
FACTOR OF SAFETY AT 10^6 LOAD CYCLES

(h) Brake pads

The required properties of the brake pad here are long service life, high coefficient of friction between brake disc and the pad and the ability to recover easily from high temperatures and moisture [12].

The selected material is semi metallic brake pads, since they have a satisfactory service life and lesser noise generation than the metallic brake pads. They also do not cause thermal warpage unlike the ceramic brake pads [12].

(i) Brake disc

Discs over drum brakes are chosen for the current braking system because of the following factors:

- Lesser heat generated than drum brakes [5]
- Lesser vehicle stopping time than drum brakes [5]
- Lighter in weight and occupy lesser space
- Show lesser torque variation in the course of a stop as compared to the drum brakes which often exhibit a ‘sag’ in its torque curve [13]
- Have the ability to operate with little fade at high temperatures [14]
- Offer linear relationship between brake torque and pad/rotor friction coefficient [14].

Float type disc arrangement of the brake disc involves the outer brake disc riveted to a separate inner disc. On the other hand, the fixed type brake disc is a single brake disc in action. Float type discs are preferred over fixed type discs for the following reasons.

- They prevent the chances of thermal warpage of the brake disc during rubbing of the pads (unlike fixed discs).
- The floating brake disc can transmit mechanical vibrations to the riveted inner disc. Hence the brake disc is free from deformation due to the vibrations.

Figure 8(a) shows the CAD model of the brake disc that is made using SolidWorks 2017®. The model shows the outer brake discs riveted to an inner disc that has bolt holes for mounting on the hub. Slots and holes are present across the design for effective heat dissipation and weight reduction. The dimensions are taken in accordance with the standard design of the PS-1 caliper [8].

Thickness of disc = 0.19 inch = 0.004826m
Outer diameter of disc = 9 inch = 0.2286m
Brake pad width = 1 inch = 0.0254m
Effective radius of disc (r_{eff}):

$$r_{eff} = (9/2) - (1/2) = 4 \text{ inch} = 0.1016m \qquad (1)$$

FIGURE 8(A)
CAD MODEL OF BRAKE DISC

Results based on FEA using Ansys 17.1® are helpful for improving the design according to the software simulated version of the real dynamic conditions that the design would be subjected to. Hence the following CAE simulations were done:

- Thermal analysis
- Fatigue analysis

According to the material data given in [15] for brake discs, out of Gray cast iron (GCI), Ti-alloy (Ti-6Al-4V), Ti-composite, Al-composite and Al-Cu alloy; the Al-Cu alloy has the best properties followed by GCI. The material properties considered are compressive strength, friction coefficient, wear resistance, thermal capacity and specific gravity. Since Al-Cu alloy has a higher cost, GCI is finally selected for our study due to economic factors and yet reasonable material properties.

Thermal analysis: This is done to ensure that the temperature of the rotor does not go above its melting point. Heat flux is calculated by converting the linear kinetic energy of the vehicle and the rotational kinetic energy of the wheels into heat energy. This heat energy is then divided by the net disc-pad interface area and the time required to stop the vehicle. This time is obtained by using the parameters having the maximum value given in Table II to simulate for the 'worst case scenario'. Hence the total heat flux is obtained (597.5 kWm^{-2}) and entered into the software along with the initial temperature as 22ºC.

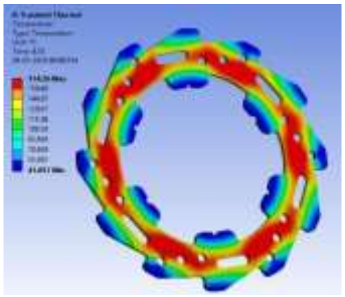

FIGURE 8(B)
TEMPERATURE DISTRIBUTION ACROSS THE BRAKE DISC

As shown in figure 8(b), the maximum temperature of the disc reaches 174.3ºC during the stopping time i.e. 4.53 seconds. This temperature is below the melting point of the rotor. The FE model of only the outer disc is generated as the inner disc has been considered as a heat sink. This would give a slightly larger temperature than the actual one.

Fatigue analysis: As shown in figure 8(b), the maximum temperature is very much less than the melting point of grey cast iron (GCI). It is due to this reason that there is no need to perform the creep analysis. Moreover the compressive forces by pads will not lead to fatigue failure.

(j) Final assembly of the braking system

All the above selected and designed components of the braking system are finally assembled in order to form an effective 'Y-type' dual split hydraulic braking system. Figure 9 shows the entire braking circuit. A total of eight simple unions, four tee unions, one cross union, two proportioning valves, four PS-1 calipers, four brake discs (same design), eleven separate brake hoses, two reservoirs and one master cylinder are used, as depicted in figure 9.

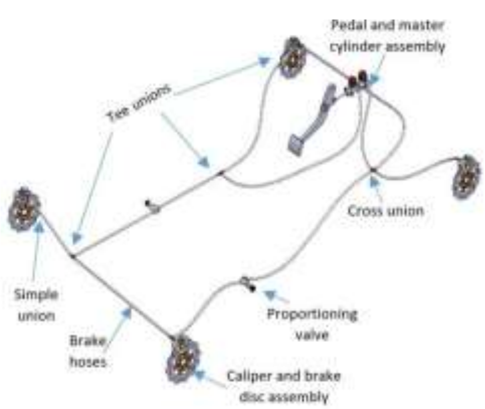

FIGURE 9
FULLY ASSEMBLED Y-TYPE BRAKING SYSTEM

IV. Validation of the braking circuit

The validity of the braking system formed by assembling the selected/designed components is tested for successful implementation in the above mentioned five ATVs.

The aim of the following calculative approach is for checking the validity or compatibility of the braking system with the selected ATVs in terms of performance and vehicle safety during dynamic conditions. This was done by:

- Ensuring compatibility with CMV (central motor vehicle) rules of the Indian government.
- Ensuring that the front and rear braking torques are greater than the respective front and rear frictional torques. They should be greater by a marginal amount, so that the wheels get locked at the required right time (not too soon for avoiding skidding and not too late).

TABLE II
LIST OF ATVS AND THEIR PARAMETERS [16] [6]

Parameters	ATV-1	ATV-2	ATV-3	ATV-4	ATV-5
x	0.4	0.4	0.4	0.45	0.4
y	0.6	0.6	0.6	0.55	0.6
W (N)	4498.2	4690	4429.6	4501	5311.6
h (m)	0.6985	0.762	0.6985	0.6985	0.7366
L (m)	1.5621	1.8288	1.5621	1.8288	1.651
r_f (m)	0.3048	0.3175	0.3175	0.3175	0.2413
r_r (m)	0.3048	0.3175	0.3175	0.3175	0.254
GC (m)	0.254	0.2667	0.2604	0.2667	0.2667
V (ms^{-1})	15.278	15.278	16.67	13.89	15.278

Physical quantities and their symbols to be used for the vehicle dynamics equations for system validation are as follows:

x: portion of weight at the front axle of the vehicle.
y: portion of weight at the front axle of the vehicle.
W: weight of the vehicle including the driver
h: height of the CG of vehicle from wheelbase
L: wheelbase of the vehicle
r_f, r_r: radius of front and rear tires respectively.
GC: ground clearance (used for checking with the rotor diameter)
V: velocity at which the vehicle brakes
μ: coefficient of friction between road and tire
μ= 0.7 [1]
Braking system parameters are as follows:
g = 9.8ms^{-2}
PR: pedal ratio = 6:1
d_{mc}: Diameter of master cylinder piston = 15.875mm = 0.015875m [8]
A_{mc}= Area of master cylinder piston

$$A_{mc}= \pi\, d_{mc}^2/4 \quad (2)$$

PE: Pedal effort of the driver
The values of the Knee Angle **(β)** and the Thigh Angle **(α)** are measured according to the vehicle. Figure 10 shows the

ninety fifth percentile male data chart from NASA [17] that gives the approximate value of the pedal effort. Hence PE equals to 850N.

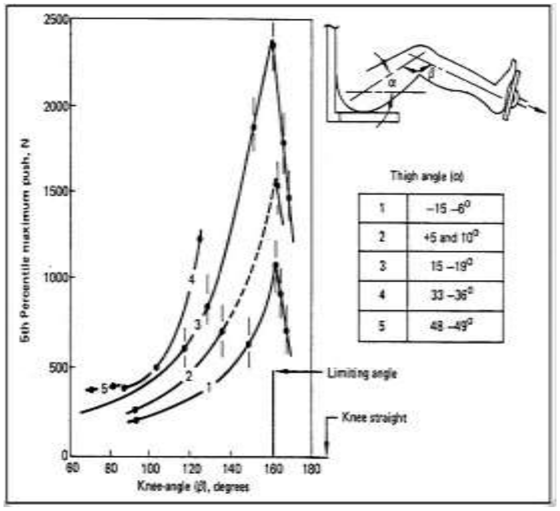

FIGURE 10
NINETY FIFTH PERCENTILE MALE DATA CHART FROM NASA [17]

F_P: Pushrod force

$$F_P = PR*PE \tag{3}$$

Where PR mechanical advantage by pedal or the pedal ratio

$$P = F_P/A_{mc} \tag{4}$$

$F_P/A_{mc} = P$
The above equation depicts Pascal's law.
P: brake line pressure
A_{cal}: Net caliper piston area
$A_{cal} = 0.79in^2 = 0.00051m^2$ (PS-1 properties) [8]
F_{cal}: Caliper clamping force

$$F_{cal\text{-}front} = P*A_{cal} \text{ [18]} \tag{5}$$

$$F_{cal\text{-}rear} = P*BR*A_{cal} \text{ [18]} \tag{6}$$

The above two equations are for front and rear calipers respectively.

BR is biasing ratio. Biasing ratio is the ratio between the brake line pressure towards the rear wheels and the front wheels. Since the front wheels require more braking force than the rear (due to dynamic load shift during braking towards front), the biasing ratio is introduced.

BR = 43/100 = 0.43 is taken since it is the maximum biasing limit of the selected proportioning valve [8].
μ_P : coefficient of friction between pad and brake disc
$\mu_P = 0.3$ [1]
T_{bf}: braking torque at the front axle
T_{br}: braking torque at the rear axle

$$T_{bf} = 2*F_{cal\text{-}front}*r_{eff}*\mu_P \tag{7}$$

$$T_{br} = 2*F_{cal\text{-}rear}*r_{eff}*\mu_P \tag{8}$$

F_{bf}: braking force at the front wheels
F_{br}: braking force at the rear wheels

$$F_{bf} = T_{bf}/r_f \tag{9}$$

$$F_{br} = T_{br}/r_r \tag{10}$$

F_b: total braking force

$$F_b = F_{br} + F_{bf} + F_a \text{ [13]} \tag{11}$$

Where F_a is the aerodynamic drag force.

$$F_a = D_a V^2 \tag{12}$$

D_a: aerodynamic drag coefficient
$D_a = 0.35$ referring to vehicle parameters in [19].
d: deceleration of the vehicle

$$d = F_b*g/W \text{ [13]} \tag{13}$$

SD: stopping distance of the vehicle

$$SD = V^2/2d \tag{14}$$

In the above equation, time required for pedal travel and hydraulic pressure build up is neglected. The conformity of the stopping distance and the deceleration of the vehicle with the CMV (central motor vehicle) rules of the Indian government are also checked by relations given below:

$$SD_{max} = 0.15*V_o + V_o^2/130 \text{ [20]} \tag{15}$$

$$V_o = 0.8*V_{max}*5/18 \tag{16}$$

This is the maximum permitted stopping distance for a hydrostatic braking vehicle under the CMV Rules [20]. Hence if $SD < SD_{max}$ and if $d > d_{min}$; then the braking design is compatible with the vehicle with respect to the CMV rules.

$$d_{min} = V^2/2SD_{max} \tag{17}$$

W_d: dynamic load shift while braking.

$$W_d = (d/g)*(Wh/L) \text{ [21]} \tag{18}$$

Value of d is determined after brake torque calculations.

$$W_f = W*x + W_d \tag{19}$$

$$W_r = W*y - W_d \tag{20}$$

The above equations compute the net normal loads on front and rear axles respectively while braking.
Calculating the frictional forces:

$$F_f = W_f * \mu \tag{21}$$

$$F_r = W_r * \mu \tag{22}$$

Frictional torques at respective axles:

$$T_f = F_f * r_f \tag{23}$$

$$T_r = F_r * r_r \tag{24}$$

If $T_f < T_{bf}$ and $T_r < T_{br}$; then the compatibility of the developed braking system with the vehicle requirements is ensured since the braking torques are greater than the frictional torques of the vehicle by a marginal amount for the wheels to lock.

Computed Constants		
μ	0.7	
g	9.8	m/s2
PE	850	N
PR	6	
F_p	5100	N
d_{mc}	15.875	mm
P	25.78	MPa
A_{cal}	509.6764	mm2
F_{cal}	13139.15	N
μ_p	0.3	
r_{eff}	0.1016	m
T_{bf}	800.955	Nm
BR	0.43	
D_a	0.35	
T_{br}	344.4107	Nm

FIGURE 11(A)
LIST OF BRAKING SYSTEM PARAMETERS

Computed Variables	ATV-1	ATV-2	ATV-3	ATV-4	ATV-5
F_{bf} (N)	2627.805	2522.693	2522.693	2522.693	3319.333
F_{br} (N)	1129.956	1084.758	1084.758	1084.758	1355.947
d (m/s2)	8.365	7.709	8.196	8.002	8.777
T_f(Nm)	750.197	758.571	761.966	762.115	717.358
T_r (Nm)	209.539	283.781	222.513	238.233	189.289
SD (m)	13.952	15.140	16.952	12.056	13.298
SD_{max} (m)	21.49305	21.49305	24.93186	18.31032	21.49305
d_{min} (m/s2)	5.430064	5.430064	5.572967	5.268398	5.430064

FIGURE 11(B)
LIST OF COMPUTED VARIABLES ACCORDING TO DIFFERENT ATVS

Data for Chart					
ATVs	ATV-1	ATV-2	ATV-3	ATV-4	ATV-5
T_f(Nm)	750.197	758.571	761.966	762.115	717.358
T_r (Nm)	209.539	283.781	222.513	238.233	189.289
T_{bf}(Nm)	800.955	800.955	800.955	800.955	800.955
T_{br} (Nm)	344.4107	344.4107	344.4107	344.4107	344.4107

FIGURE 11(C)
COMPARISON OF FRICTIONAL AND BRAKING TORQUE VALUES FOR BOTH THE AXLES

These calculations are solved in Microsoft Excel for the five selected ATVs, shown above. Figure 12 shows the marginal difference between the frictional and the braking torques with respect to both the axles. The red and the green lines show the braking torques of the braking system that remain constant throughout the five ATVs. The blue and the yellow lines show the frictional torques for both the axles for the five ATVs.

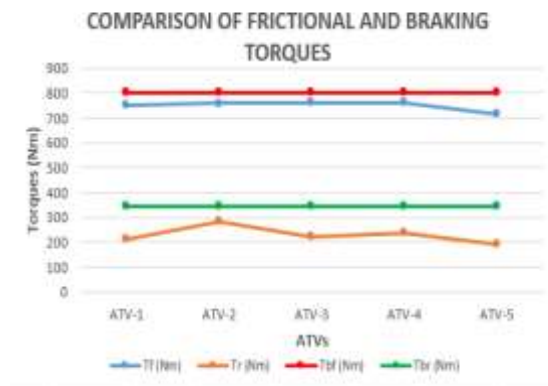

FIGURE 12
GRAPHICAL COMPARISON BETWEEN FRICTIONAL AND BRAKING TORQUES

Moreover, from figure 11(b) it is inferred that the stopping distance and the computed decelerations of the respective vehicles equipped with the new braking system fall within the limitations set by the CMV rules [20].

V. Limitations of the Braking System

The following are the projected limitations of the above designed braking system:

- The selected components will only work on the five ATVs and other ATVs with similar specifications. For heavier ATVs with larger tires and greater top speed, calipers with higher clamping forces can be used. Eg- Wilwood's DH4 dual inlet caliper [8] has two inlet ports that could eliminate the need of Tee unions attached to the caliper. In order to balance the braking force among wheels on the same axle, separate calipers would be used for those wheels where only one inlet pipe is required. The same strategies can be used to select/design the braking components for ATVs with different specifications, if not the same braking components.
- The 'Y-type' braking system involves the use of a larger number of components and a lot of hoses.
- Factors such as rolling resistance of the wheels, pressure build up time towards the caliper, pedal travel time, road gradient, pressure losses inside braking components, coefficient of friction on wet roads have been neglected in the vehicle dynamics calculations.

CONCLUSION

The components of the braking system are finally selected/designed and then assembled in order to form an effective 'Y-type' dual split hydraulic braking system. The designs of these components ware later analysed with the help of FEA using Ansys 17.1® for the dynamic conditions and prolonged application they would be subjected to while the vehicle operates. The compatibility of this braking system is ensured with the five selected ATVs (and similar such vehicles) by following a detailed calculative approach and checking the system's conformability with the Indian government CMV regulations for the dynamic safety of the vehicle. Moreover, the circuit design for the ATVs that do not fall into the above category is also discussed briefly. Overall it can be concluded that the selected five ATVs and vehicles with similar specifications can be fitted with the designed 'Y-type' braking system that would offer a safer braking in case of a single brake circuit failure by locking three wheels at a time.

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