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Parametric optimization of Seal groove for Brake Caliper of an ATV

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Abstract:-- **For any vehicle, safety of the driver is paramount thus there is a need to design and fabricate a safe and reliable braking system. Design of seal groove in the brake caliper significantly influences the braking performance. The deformation of the brake seal during the movement of piston affects the squeezing force, hence the sealing properties of the oil seal and the drag torque, which acts as a resisting force against the motion of the vehicle. One of the major drawbacks of using a floating type is the drag torque. Hence, improper design of the seal groove geometry can lead to problems such as friction losses, excessive pad wear, noise and vibrations. This paper focuses on optimising the three parameters, i.e., the Corner-break angle, Bottom angle and the Front angle to reduce drag torque and maximize piston retraction for a single piston floating type custom brake caliper.**

Index Terms:- Drag torque, Finite Element Analysis (FEA), Fluid displacement, Hyper-elastic materials, Oil seal.

I. INTRODUCTION

Baja SAE is an undergraduate collegiate competition organized by the Society of Automotive Engineers (SAE), in which students design and manufacture a single-seater All-terrain vehicle. The brakes setup in our ATV designed for the Baja SAE competition includes Pedal, Pushrod, Master cylinder, Proportioning valve, Brake lines, Front & Rear calipers, and Disc. This setup is a hydraulic braking system with a proportionating valve to control brake fluid pressure.

A custom floating type brake caliper is Designed, as it had the following advantages over an OEM brake caliper: Less weight, small size, and less drag torque. Less weight of caliper helped decrease the unsprung mass, while small dimensions allowed to reduce the packing distance of wheel assembly, hence increasing the safety factor of suspension components. Also, custom caliper helped bring down the overall cost of manufacturing.

Brake fluid displacement and drag torque are the quantities that need to be optimized while designing the brake caliper. Braking fluid displacement implies an additional volume of brake fluid required for the caliper to apply the brake. Drag torque is the residual torque on the disc after the brake is released. A minimal drag torque and maximum piston retraction can reduce frictional losses, excessive wear, and noise, thus improving the life of the components. The increase in piston retraction will reduce the brake fluid displacement at a static position. This will ensure that excessive pressure does not build up in the line. Hence the optimum design of the brake caliper is necessary to increase the piston retraction and minimize the drag torque.

Housing caliper fingers, brake lining, and seal groove geometry can be modified to reduce drag torque. Also, the material of the brake seal has a significant role in determining the performance of the caliper. However, based on actual design considerations such as weight and considerations for increased housing and caliper finger stiffness, easy material availability of higher stiffness pad lining, and limited options of brake seal material, it is more convenient to change the seal groove geometry to get the desired piston retraction for reducing drag torque. Iterative optimization and modification of the seal groove are feasible and the most economical options.

II. METHODOLOGY

It is necessary to determine the residual force and the maximum piston retraction to calculate the drag torque. To simulate the working of the brake caliper, a Finite Element Model of the caliper seal groove is modelled in ANSYS Workbench. Experimental data was used for modelling the non-linear behaviour of brake seal which increased the accuracy of the simulation. Accurate modelling of the oil seal is extremely essential because when the piston is pushed outwards, the oil seal deforms. As a result, strain energy is stored in a deformed oil seal which is released by retracting the piston. Due to hysteresis, the loading and the unloading curves of the hyper-elastic materials do not follow the same path. Hence, some amount of energy is lost due to which the piston does not retract back to its original position.

The contact between the brake seal and the brake caliper, and the brake seal and the piston are a non-linearity which can cause the same inaccuracy as it is very difficult to predict the exact coefficient of static and dynamic friction. For modelling the non-linear contact, the value of static friction, less than the actual value of friction coefficient for the material pair was used, as the brake oil acts as a lubricant reducing the resistance to the motion of the piston [1]. It is assumed that the piston is a perfectly rigid body, as young's

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modulus of hardened steel is significantly greater than young's modulus of Brake seal material (NBR) and Brake Caliper material (Aluminium 6065-T6). As residual pressure in the brake lines is also a contributing factor to the drag torque, it is considered that at the end of the braking cycle, the brake pedal returns to its original position, i.e., there is no excess pressure in the brake lines.

The seal groove parameters which affect the piston retraction and drag torques are the Corner-break angle, the Bottom angle, and the Front angle. The width and depth of the seal groove are kept constant as the dimensions of the OEM oil seal were fixed for the given piston size. The values of the above-mentioned parameters are varied in fixed steps. The result data points obtained at the given values are extrapolated to obtain the nature of the graph over the range of values. The data obtained from Finite Element Analysis will be sorted and analysed, to obtain the optimum values of the selected parameters.

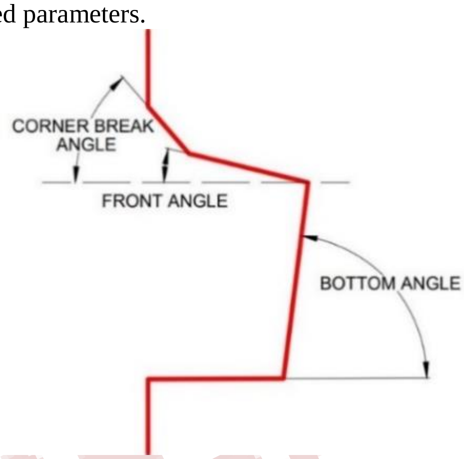

Fig-1. Nomenclature of groove angles

III. MODELLING OF OIL SEAL

The oil seal is made from **NBR** (Nitrile-Butadiene rubber) which is an oil resistant, synthetic rubber with hyper elastic material behaviour.

Stress vs % Elongation data of a uniaxial tensile test of NBR was obtained from the oil seal manufacturing company.

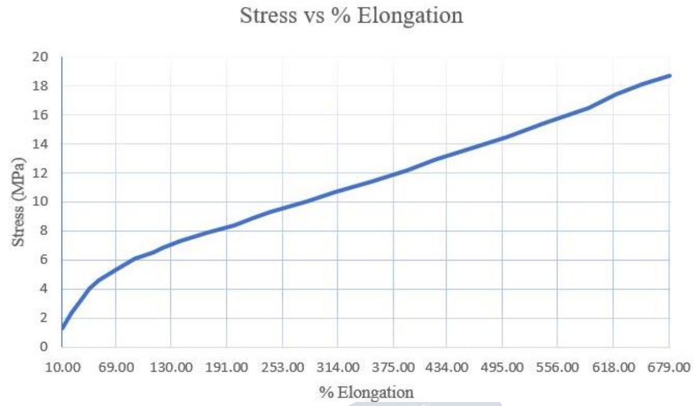

Fig-2. Stress vs Elongation graph for seal

A Mooney-Rivlin model for hyper-elastic material was choose as the it is the most accurate model to reproduce the mechanical behaviour of materials with properties similar to that of NBR [2]. As the nature of the graph is almost linear with no inflection points (First order equation), the 2 parameter Mooney-Rivlin for hyper-elastic material was chose. The Mooney-Rivlin model uses the strain energy to characterize mechanical properties of the material. Following is the Mooney-Rivlin model for strain energy potential.

$$W = C_{10}(\bar{I_1} - 3) + C_{01}(\bar{I_1} - 3) + \frac{1}{d}(J - 1)^2$$

Curve fitting was done on the Stress vs Strain graph of the uniaxial tensile test to obtain the values of the constants. Here, C01 and C10 are material constants and D1 is material incompressibility parameter.

Table 1. Constant values for NBR

C01	C10	D1
0.053MPa	0.21MPa	0.0021MPa

One of the limitations of Mooney-Rivlin model is, it is not suitable for deformation that exceed 150% elongation. For our case, maximum piston travel is 2 mm with a 4 mm thick brake seal. The maximum deformation that can be obtained for this scenario is 150% of original dimension. Hence, Mooney-Rivlin model can be adopted to model the brake seal.

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IV. FEA OF SEAL GROOVE GEOMETRY MESHING –

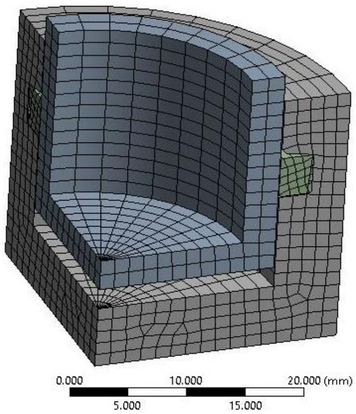

Fig-3. Mesh for seal groove

Table 2. No. of Nodes & Elements

Nodes	**948**
Elements	**295**

A quad mesh was selected for meshing the brake seal, Selected PLANE183 and CONTACT172 elements type represents elements that have plasticity, hyper elasticity, creep, stress stiffening, large deflection, large strain capabilities and interface delamination. Also, a non-linear adaptive mesh region (Shape criterion with maximum corner angle of 150 degrees) was assigned to the brake seal geometry to aid convergence of solution.

Boundary conditions -

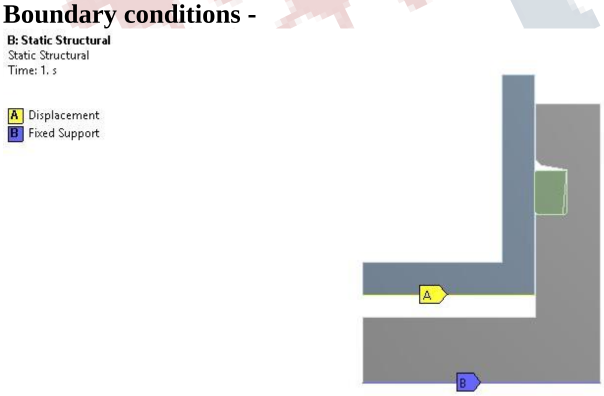

Fig-4. Boundary Conditions for seal groove

An axis-symmetric model of caliper is used for seal groove analysis. Caliper is given a fixed support and remote displacement of 2 mm was assigned to the caliper, which is the maximum clearance between the brake pads and the brake disc. Only degree of freedom assigned to the piston is translation is y direction. The interference treatment for the contact region between the brake seal and the caliper was seat 'Add offset, Ramped effect', as there is interference fit between the brake seal and the caliper. The first sub step was used to simulate the interference fit of brake seal. Then maximum brake fluid pressure (6.2 MPa) was applied to the bottom face of the piston which is gradually reduced to zero in the following 10 sub steps.

The Corner break angle, bottom angle and the front angle were marked as parameters in the Design modeller. The Corner break angle is varied in steps of 5 degrees, from 45 degrees to 65 degrees (45^{0}, 50^{0}, 55^{0}, 60^{0}, 65^{0}). The bottom angle is varied from 88 degrees to 91 degrees in intervals of 0.25 degrees (91^{0}, 90.75^{0}, 90.5^{0}, 90.25^{0}, 90^{0}, 89.75^{0}, 89.50^{0}, 89.25^{0}, 89^{0}, 89.75^{0}, 89.5^{0}, 89.25^{0}, 89^{0}, 88.75^{0}, 88.5^{0}, 88.25^{0}, 89^{0}). The values of front angle varied in intervals of 4 degrees starting from 6 degrees up to 14 degrees (6^{0}, 10^{0}, 14^{0}).

V. RESULTS

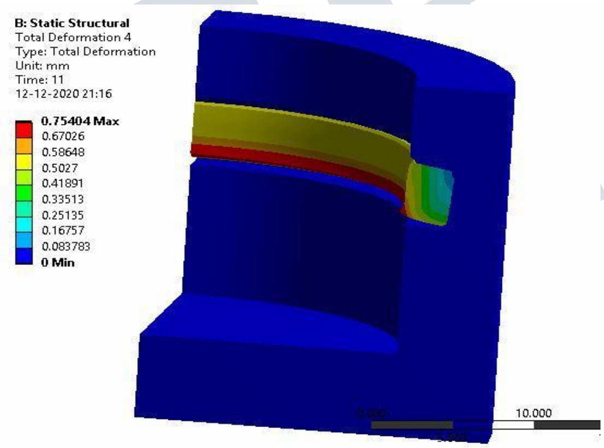

Fig-5. Deformation of seal

Corner break angle = 45°

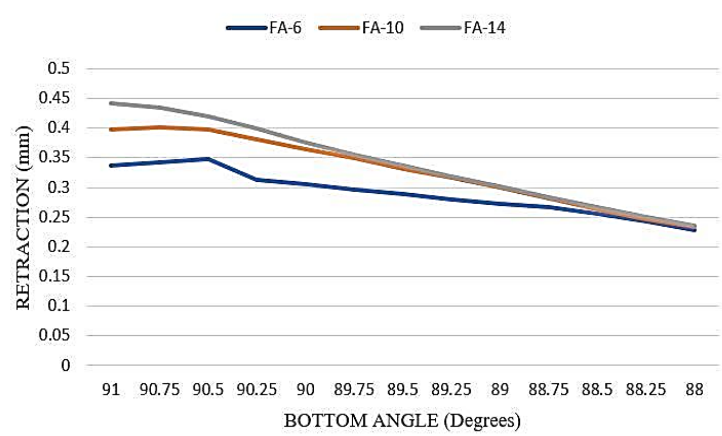

Fig-6. Retraction vs Bottom Angle for 45° Corner Break Angle

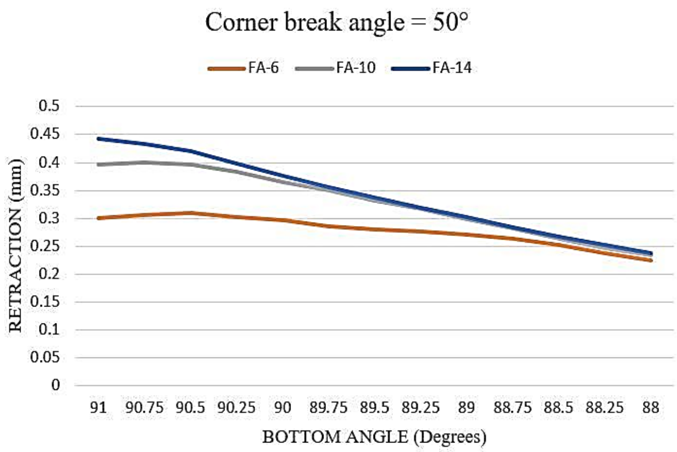

Fig-7. Retraction vs Bottom Angle for 50° Corner Break Angle

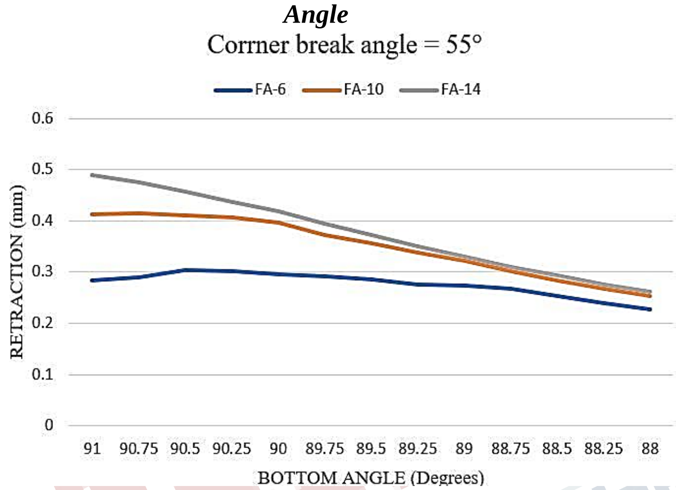

Fig-8. Retraction vs Bottom Angle for 55° Corner Break Angle

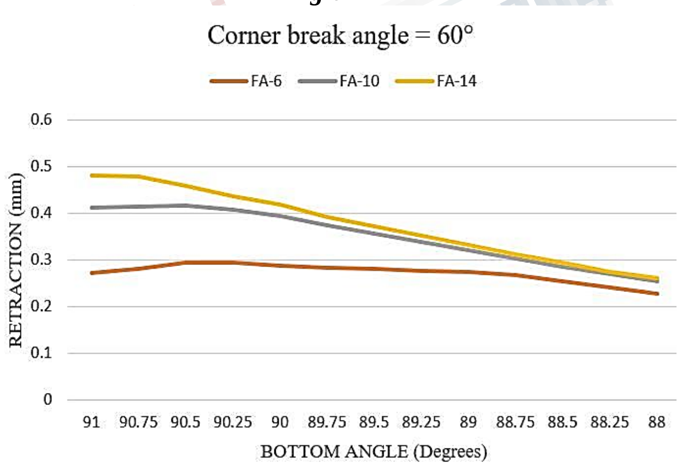

Fig-9. Retraction vs Bottom Angle for 60° Corner Break Angle

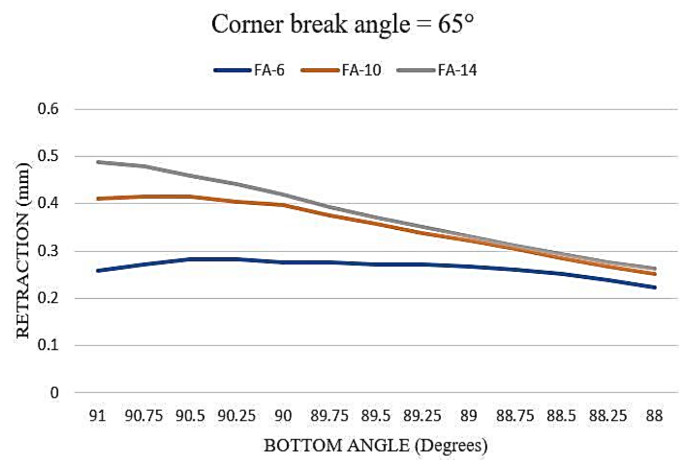

Fig-10. Retraction vs Bottom Angle for 65° Corner Break Angle

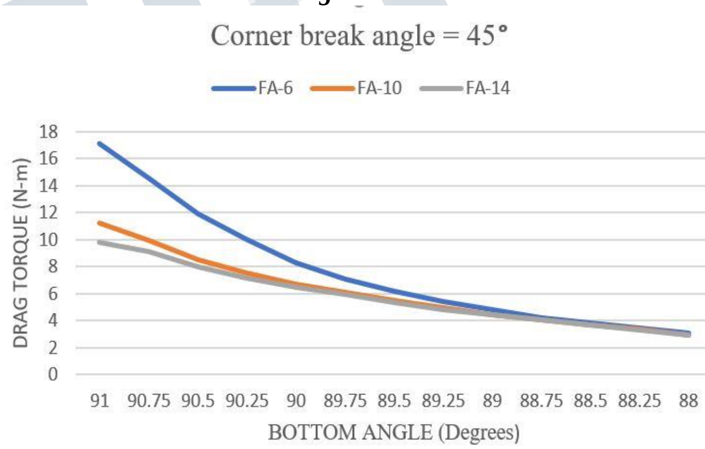

Fig-11. Drag torque vs Bottom Angle for 45° Corner Break Angle

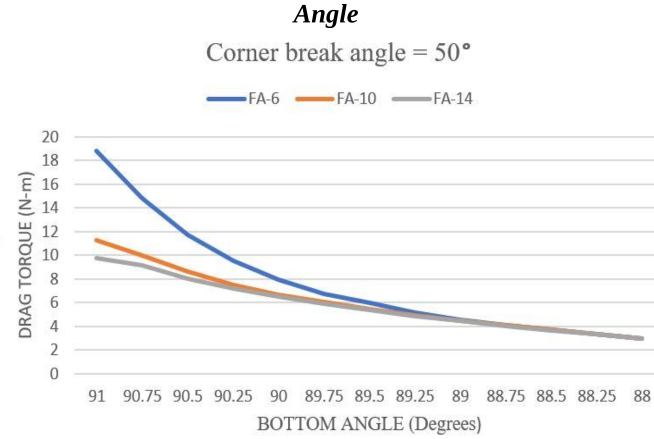

Fig-12. Drag torque vs Bottom Angle for 50° Corner Break Angle

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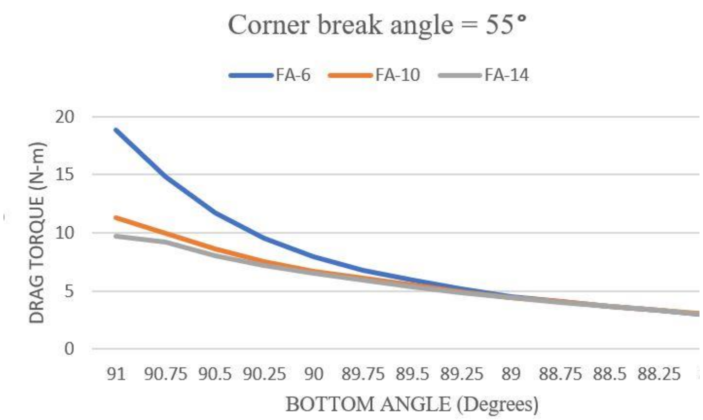

Fig-13. Drag torque vs Bottom Angle for 55° Corner Break Angle

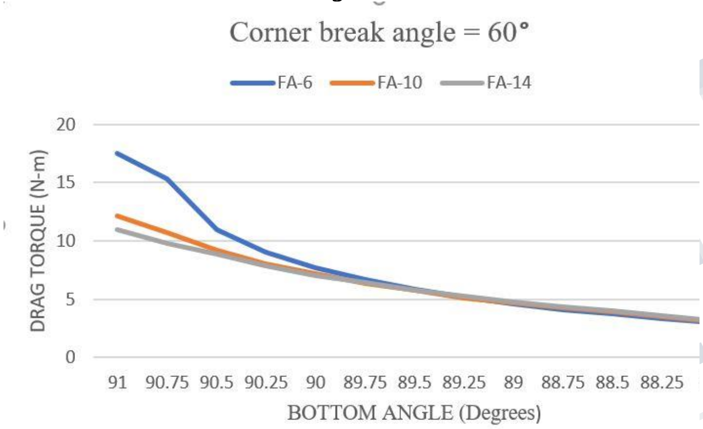

Fig-14. Drag torque vs Bottom Angle for 60° Corner Break Angle

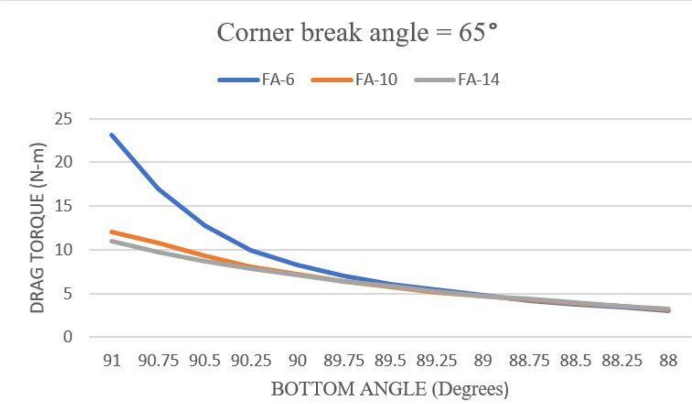

Fig-15. Drag torque vs Bottom Angle for 65° Corner Break Angle

The finite element analysis of the seal groove provides the value of residual force in Y-direction or the residual axial force. This force is then multiplied with effective radius of brake disc (0.077 mm) to obtain the value of drag torque.

With change in bottom angle for each corner break angle different values of piston retraction and drag torques are obtained. From the graphs it can be seen that drag torque and piston retraction varies significantly for different front angle when bottom angle lies between 91 degrees to 89.5 degrees. For the values of bottom angle below 89.5 degrees, both drag torque and piston retraction exhibit a linear relation. Also, for same Corner break angle and Bottom angle, data set with lower value of front angle exhibit a greater drag torque, but lower piston retraction. This validates the basic hypothesis that piston retraction varies inversely with drag torque.

VI. Optimization using Linear programming

The data obtained from the ANSYS Workbench was exported into MS Excel. For the drag torques, values for bottom angle greater than 90 degrees are considerably high and values for bottom angle less than 90 degrees vary in a quite linear fashion. Hence, a regression line was fitted for the Drag torque and piston retraction against the inputs of Corner-Break angle, Bottom angle and Front angle.

Equation of the line obtained for drag torque is,

Drag torque = 0.012*CBA + 2.045*BA - 0.027*FA – 177.52

Equation of the line obtained for retraction is,

Deformation = 0.002*CBA + 0.052*BA -0.008*FA – 4.33

An objective function was defined in order to maximize piston retraction and minimize drag torque. Equal weights of 0.5 was assigned and multiplied with normalized drag torque and retraction. Torque is multiplied with -0.5 as it is to be minimized. The normalized values were calculated using the following formula –

$$Normalized\ Value = \frac{Value - Mean\ of\ the\ Values}{Standard\ Deviation\ of\ the\ Values}$$

The objective function is maximized and solved using Simplex method of linear programming using MS excel solver, subject to the input constraints of corner-break angle ($35 \leq CBA \leq 75$), Bottom angle ($87 \leq CBA \leq 93$), and Front Angle ($5 \leq CBA \leq 10$) and output constraints Drag torque ($0 \leq$ Drag torque), and piston retraction ($0 \leq$ Retraction).

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Torque Regression:			
FA	BA	CBA	CONSTANT
-0.027	2.045	0.012	-177.523
0.008	0.041	0.004	3.613
0.951	0.304	#N/A	#N/A
856.324	131.000	#N/A	#N/A
237.221	12.097	#N/A	#N/A
Mean	4.853		
Std. Dev.	1.364		
'orque = 0.012*CBA + 2.045*BA - 0.027*FA - 177.5			

Deformation Regression:			
FA	BA	CBA	CONSTANT
0.008	0.052	-0.002	-4.330
0.001	0.005	0.000	0.469
0.557	0.039	#N/A	#N/A
54.926	131.000	#N/A	#N/A
0.256	0.204	#N/A	#N/A
Mean	0.288		
Std. Dev.	0.059		
formation = -0.002*CBA + 0.052*BA -0.008*FA - 4			

CBA	BA	FA
45	87	14

	Torque	Retraction
Optimized Value	0.538	0.231
Normalized Value	-3.164	-0.973

Objective Function
1.095

Fig-16. Excel datasheet for formulating Linear programming model

Table 3. Optimized values of variables/parameters

Variable/Parameter	Final Value
Corner-break Angle	45 deg.
Bottom Angle	87 deg.
Front Angle	14 deg.
Drag torque	0.538 N-m
Piston Retraction	0.23 mm

Fluid Displacement Calculations:

The deflection was used to find the fluid displacement required for the calipers.

Effective piston travel = caliper deflection + Piston clearance + Line expansion + MC losses + Other losses

Not considering losses,

Piston clearance = Maximum Piston deflection – Piston Retraction = 2 – 0.23 = 1.77 mm

Travel = (0.442) × 2 + 1.77 × 2 = **4.42 mm**

$Volume\ of\ fluid\ displaced = \frac{\pi}{4} \times (1.25 \times 25.4)^2 \times 4.42 = \mathbf{3502.62\,mm^3}$

$$Pushrod\ Travel = \frac{3502.62}{\left(\frac{\pi}{4} \times 19.04 \times 19.04\right)}$$

$Pushrod\ Travel =$ **12.03 mm**

Hence, for the Master cylinder with bore diameter of ¾ inch or 19.04 mm, pushrod travel of minimum 12.03 mm is required to generate the desired clamping force.

VII. CONCLUSIONS

1. Minimum drag torque of 0.989 N-m was from the optimised values.
2. OEM master cylinder has a minimum push road travel of ½ in or 12.7 mm, which is greater than required pushrod travel.
3. Optimal values of seal groove angles were estimated using linear programming model.

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