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Design and Analysis of Damper Systems for Circuit Breaker

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ABSTRACT

In today's age, a damper is an integral part of design of the operating mechanism for absorbing impact force to control override and bounce during opening operation in a circuit breaker. We present, in this paper, design of two passive damper systems, rubber damper and hydraulic damper, for a circuit breaker during opening operation. The challenges in building the dampers due to the kinematics and dynamics of the complex transmission linkages and the calculation of impact force on the damper are discussed. An evaluation of the dampers based on analytical calculations for finding the maximum amplitude of vibration and simulation of static structural and explicit dynamics modules using ANSYS Workbench has been provided. The maximum equivalent stress and maximum deformation due to impact force and velocity are computed for the two dampers in each of the simulation cases.

Keywords- **Circuit breaker, rubber damper, hydraulic damper, shock absorption, impact, vibration.**

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I. Introduction

The damper system in the operating mechanism works to dampen the impact, so that the amount of rebound of the mechanical stop is limited to a set range and time span when the breaker is open. We discuss, in this paper, design of two passive damper systems, rubber damper and hydraulic damper, for a circuit breaker during opening operation.

I. Literature Review

N. B. Kate, T. A. Jadhav [1] presented a mathematical model for the damping force of the hydraulic shock absorber which is implemented to analyze the shock absorbers mounting brackets attached to the vehicle structure. Damping characteristics of automotive were analyzed by considering the performance of displacement-sensitive shock absorber (DSSA) for the ride comfort. The results obtained by experimental method using damper test rig are close to results obtained by analytical model of damping force with 10 % of error. Physical testing results thus indicate that the considered shock absorber mathematical model is reliable and can be used to calculate the durability target life of mounting brackets. Thus this presented methodology can be utilized as an effective way to reduce time and cost in design and development of automotive components.

Mr. Sudarshan Martande, Mr. Y. N. Jangale, Mr. N.S. Motgi[3] focus on developing new correlated methodologies that allow engineers to design components of shock absorbers, which are an essential component of an automobile, using FEM based tools. This paper uses a particular method to test it using Finite Element Analysis technique. The analytical calculations for the considered piston assembly in this paper were done using the basic design calculations of each part and the same were compared with the ANSYS results. The different stress and deflection values in shock absorber components were obtained using FEA tools and compared with analytical solutions. Percentage error was calculated and it was found that percentage error is less than 15%. Various stress results were below allowable limits of material thus proving successful use of the commercial FEA tool ANSYS in the design validation of shock absorber.

D. D. L. Chung [2] reviews the materials used for vibration damping, including metals, polymers, cement and their composites. Metals and polymers tend to be better than cement due to their viscoelasticity. The paper concludes that damping enhancement mainly involves micro-structural design for metals, interface design for polymers and admixture use for cement.

John Dixon's book [5] introduces the basics of the shock absorbers for novices and helps identify the different types of shock absorbers available in the industry. The steps required for designing an automotive hydraulic damper and the parameters to be considered are discussed.

Douglas P. Taylor [6] explained, in a very beautiful and easy to understand manner, the energy calculations and the industry methods to pick and select appropriate dampers based on their applications, types, efficiencies and cost.

Nitin S. Gokhale et al [7] provide a concise view of the practical finite element methods used for solving the complex dynamic problems involved in modern softwares.

II. Methodology

It was a challenge to decide on the inertial mass to be considered acting on the damper. Through brainstorming and discussions we decided on the entire mass assembly of the mechanism upper shaft about its axis for inertia. Through free body diagram and simple mechanics calculations of the mechanism linkages, we achieved the rotational speed of the shaft. The total energy to be absorbed by the damper was calculated to find the force acting on the damper for a considered stroke length and hitting link dimension which gave the linear velocity acting on the damper.

A simple mathematical model was considered with many assumptions discussed in the next section. The hydraulic damper was solved analytically for the maximum amplitude of vibration using this model.

The rubber and hydraulic dampers were solved in the static structural and explicit dynamics module of Ansys WorkBench to find the maximum stress and deformation due to impact using a crude tetrahedron mesh for the boundary conditions of force and velocity keeping the support fixed in case of rubber damper and the cylinder fixed in case of hydraulic damper.

III. Damper Designs

A. *Rubber Damper*

This concept introduces a rubber damper to cushion the vibrating linkages during tripping. This concept is chosen due to the space constraint of 65 mm involved in horizontal positioning of the damper and because it is the most cost effective solution.

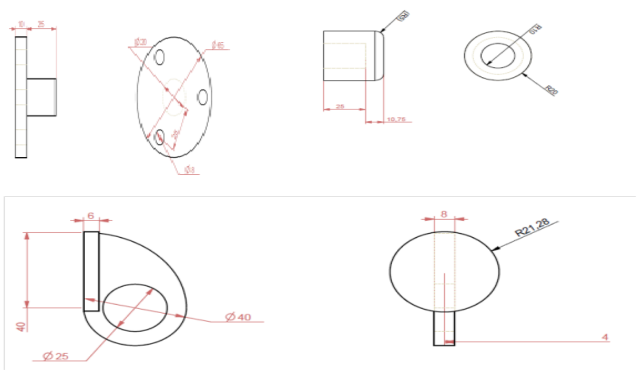

Fig 1: Rubber damper parts

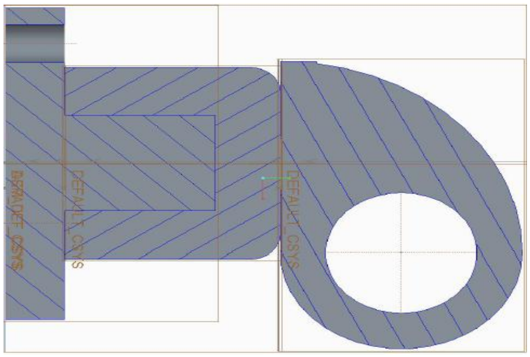

Fig 2: Rubber damper assembly cross section

B. *Hydraulic Damper*

This concept proposes a self designed spring and dashpot parallel arrangement to suit the requirements specific to our new mechanism during tripping. This concept is chosen as it gives better performance and longevity compared to the first concept. Modelling for this concept is done according to the calculations and simulation analysis of vibration during tripping time is done.

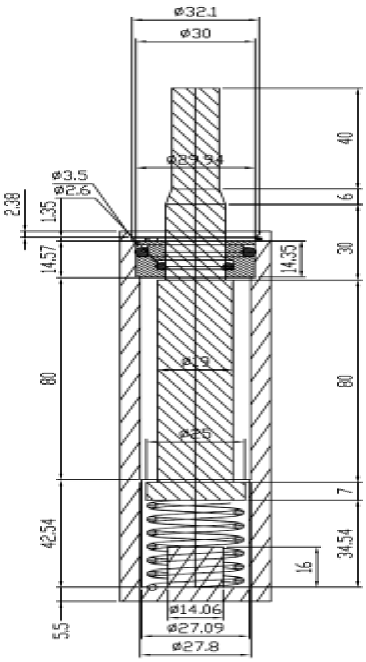

Fig 3: Hydraulic damper assembly cross section

IV. Design Calculations

A. *Total Energy and Output Force*

Mass moment of inertia about mechanism shaft axis I = 518 kg-mm^2

Angular speed at mechanism shaft ω_n = 62.267 rad/s

Kinetic Energy = 0.5 x I x ω_n^2 = 1.0042 Nm or J

Total energy lost by internal and external trip springs during α = 26.113° of rotation just before hitting the piston head E_{lost} = 15.3844 J

Drive Energy = Total energy of springs – E_{lost}

= 23.14 – 15.3844

= 7.7556 J

Total Energy Et = K.E + D.E = 1.0042 + 7.7556 = 8.76 J

Stroke s = 16mm (considered)

Output force F is assumed to be a square wave against displacement.
Et = F x s
So, Output Force F = 547.5 N

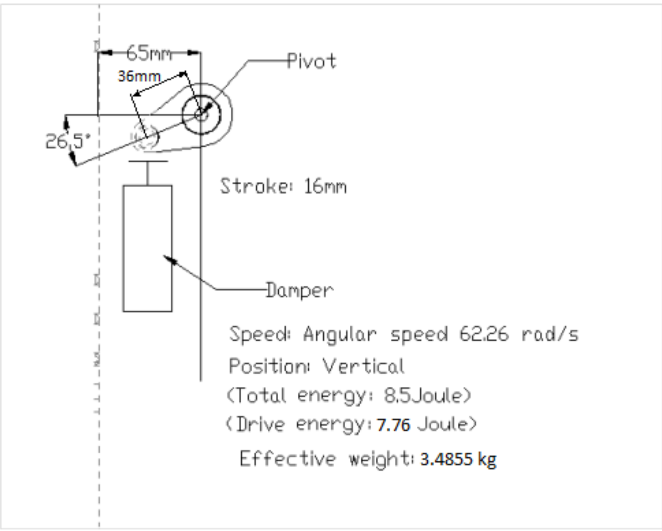

Fig 4: Damper position line diagram

B. Spring, Piston and Cylinder Design for Hydraulic Damper

1) Spring Design:
Spring Material = EN 10279-1 Drawn
End Type = Closed and Ground
End Condition = Guided
Wire diameter d = 2.34mm SWG 13
Spring stiffness k = 6.65 N/mm
No. of coils n = 6.4
No. of active coils na = 4.4
Free length Lf = 35mm
Outer Diameter of spring = 24.2mm
Helix angle α = 5.73°
Static compressed length dst = 0.46mm
Assembled length = 34.54mm
Compressed length = 18.54mm
Stroke s = 16mm

2) Dashpot Design:

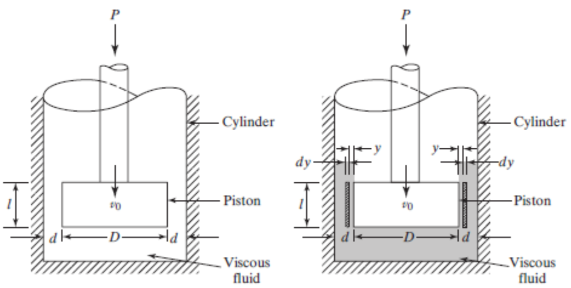

Fig 5: Dashpot line diagram with parameters

Considering
The piston and cylinder are made of high tensile steel hardened and corrosion resistant.
Cylinder bore diameter Di = 27.092mm
Cylinder outer diameter Do = 38mm
Piston head diameter D = 25mm
Piston length l = 7mm
Mass of piston mp = 0.312 kg
Distance between piston head and cylinder bore d = 1.046mm
Taking damper oil grade to be SAE Monograde summer oil SAE 40
Volume of oil V = 79.644 ml (approx.)
Dynamic viscosity μ = 0.013 N-s/m^2
Now, we know the damping coefficient is given by [4],

$$c = \mu\left[\frac{3\pi D^3 l}{4d^3}\left(1+\frac{2d}{D}\right)\right]$$

= 3.175 N-s/m
The critical damping coefficient $c_c = 2\sqrt{(k.mp)}$ = 91.1 N-s/m
Therefore, the damping ratio $\zeta = c / c_c = 0.035$
The natural frequency of the system ω_n = 146 rad/s
The damped natural frequency of the system $\omega_d = \omega_n.\sqrt{(1-\zeta^2)}$ = 145.91 rad/s
$\Phi = \tan^{-1} \zeta/\sqrt{(1-\zeta^2)}$ = 0.035 rad

C. *Mathematical Vibration Model*

Assumptions
- Underdamped vibration is assumed with inelastic collision between tripping mechanism and damper system.
- Mass is assumed to be the mass of piston rod linkage.
- Impact loading conditions are assumed as the opening operation lasts for a very short time of the order of milliseconds with the first highest amplitude at approximately 5 ms from the time of impact.
- It is assumed that the support for damper system is rigid.
- Spring force is assumed to be linear to displacement and damping force is assumed to be linear to velocity.

Consider the response of a damped single-degree-of-freedom system subjected to a step impact force. For an underdamped system with $F(t-t_0) = F_0$, the solution of the equation of motion can be written as [13]

$$x(t) = \frac{F_0}{k.\sqrt{1-\zeta^2}}\left[\sqrt{1-\zeta^2} - e^{-\zeta\omega_n(t-t_0)}.\cos\{\omega_d(t-t_0)-\phi\}\right]$$
(eqn. 1)

This describes the vibration response of the damper system in our circuit breaker mechanism.

Putting the dashpot design values in eqn. 1 for first maximum amplitude for time $(t-t_0) = 5 \times 10^{-3}$ secs we get,
x(t) = 20.63 mm
Thus, the maximum amplitude is,
A = x(t) – stroke(s) = 20.63 – 16 = 4.63mm on the damper hitting link of length
R = 36mm

Now, this vibration is transmitted to the upper shaft link which can be approximated as,

4.63x24/36 = 3.0867mm

So, the lever link receives approximately the same vibration on the mechanism side. And the vibration on the pole side link can be approximated as,

3.0867x85/109 = 2.407 mm < 3mm

Thus, this damper design falls within the permitted maximum range of vibration of 3mm for the pole side link of the circuit breaker.

V. SIMULATIONS

A. Boundary Conditions and Material Properties

1) *For rubber damper:* The material used for the rubber was silicone and that for the support and hitting link was mild steel MS. The support was fixed and a rotational velocity of 62.267 rad/s was given to the hitting link. A pressure of 0.078 MPa was considered to be acting on the rubber surface.
2) *For hydraulic damper:* The piston and cylinder are made of high tensile steel hardened and corrosion resistant. Damper oil grade is SAE Monograde summer oil SAE 40. The cylinder was fixed and static structural analysis of spring for 16 mm of stroke travel was done for a force of 547.5 N. For the explicit dynamics module, a velocity of 2.242 m/s was given to the piston to get spring deformation.

B. Analysis Results

The FEM analysis of the rubber and hydraulic dampers are carried out in Ansys Workbench for the static structural and explicit dynamics module. The results are tabulated in the next section. The equivalent (Von Mises) stresses and the total deformation values as obtained from the software are shown below in the pictures.

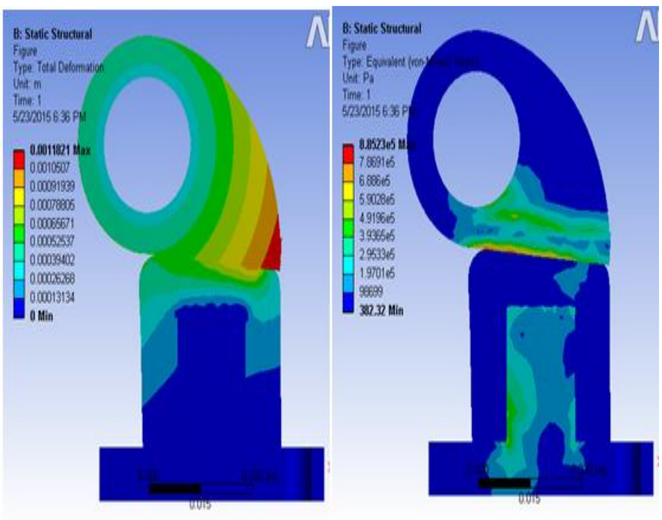

Fig 6: Static structural analysis for total deformation and equivalent stresses in rubber damper

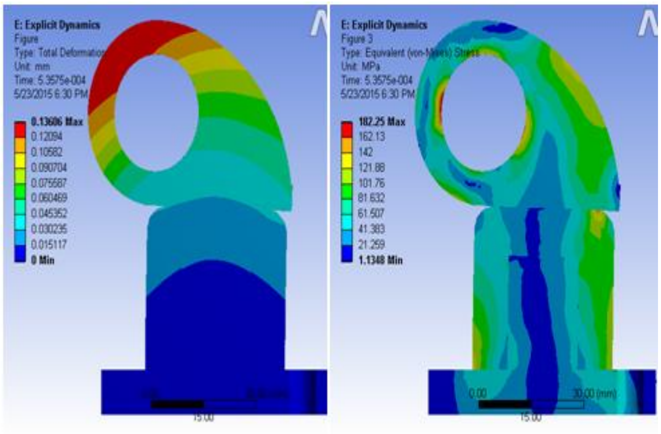

Fig 7: Explicit dynamics analysis for total deformation and equivalent stresses in rubber damper

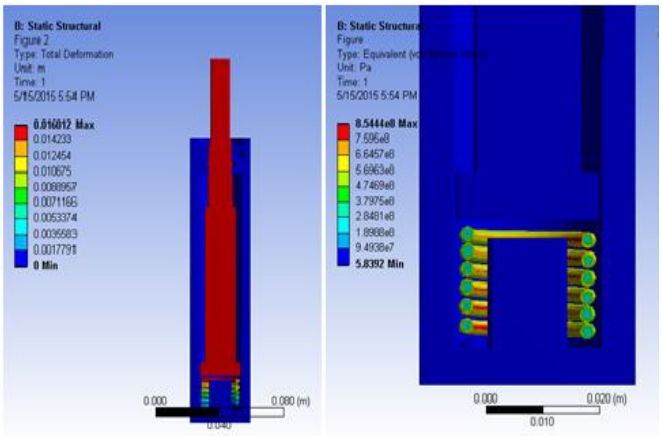

Fig 8: Static structural analysis for total deformation and equivalent stresses in hydraulic damper

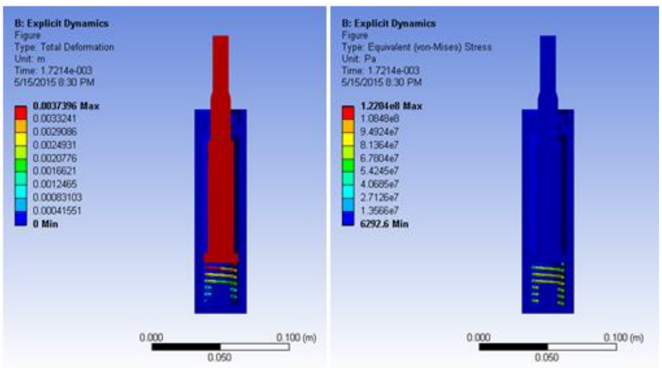

Fig 9: Explicit dynamics analysis for total deformation and equivalent stresses in hydraulic damper

VII. RESULTS AND CONCLUSIONS

Rubber Damper:

The maximum equivalent stress in static structural for rubber damper was found to be very low compared to the maximum stress limit for the damper. However the total deformation was higher compared to explicit dynamics. The maximum equivalent stress in explicit dynamics module was found to be higher than the maximum stress limit for the rubber damper.

Hydraulic Damper:

In case of hydraulic damper, the equivalent stress was higher in static structural module than the maximum stress limit for the damper material. The equivalent stress was lower than the maximum stress limit for the damper material in the explicit dynamics module. The spring deflection was performed for full 16mm stroke of damper in case of static structural module. It was deformed for only 1.7214mm in case of explicit dynamics module.

The maximum amplitude of vibration was found to be 2.407 mm which is less than 3mm. Thus it is within the set limits for the designed damper in the circuit breaker as seen through analytical calculations.

I TABLE
Analysis Results for Dampers

Damper Type		**Value**	**Static Structural**	**Explicit Dynamics**
Rubber Damper	**Total Deformation (mm)**	**Min.**	0	0
		Max.	1.1821	0.13606
	Equivalent Stress (Mpa)	**Min.**	0.000382	1.1348
		Max.	0.88523	182.25
Hydraulic Damper	**Total Displacement (mm)**	**Min.**	0	0
		Max.	0.012	1.7214
	Equivalent Stress (MPa)	**Min.**	0.0000058	0.006293
		Max.	854.44	122.04

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