

# Analysis and Vibration Behavior of a Medium Voltage Spring Operated Switch Gear Mechanism used for Circuit Breaker

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**Abstract**— In recent years, considerable attention has been given to the analysis of flexible mechanisms. For mechanisms which operate at high speeds, the inertial forces become so large and they cannot be ignored and the mechanism members undergo considerable deformation. Under these situations the rigid body assumption is no longer valid and hence the need arises for consideration of the flexibilities of the members of the mechanism. This paper investigates dynamic response of a spring operated mechanism of a 69-kilovolt SF6 (Sulphur hexa-Fluoride) circuit breaker during its opening operation. In the present work, a spring operated mechanism is analyzed using the multi body simulation.

The kinematic synthesis of the mechanism considering its links to be rigid is derived. In kinematic analysis the velocity and acceleration of each link is calculated and their responses are plotted with the time during opening operation of the mechanism. This method is particularly important in analyzing the dynamics of complicated mechanisms. The dynamic response of the circuit breaker with different spring constants was studied and its effect on the breaking time was also presented. The switchgear mechanism's link dimensions are verified by performing the mechanism motion simulation by using ADAMS software. The proto type model of the switch gear mechanism is fabricated and tested for studying the dynamic response of the links in the mechanism for its operation. The vibration characteristics by using LABVIEW software with the aid of FFT (Fast Fourier Transform) analyzer are presented.

**Keywords** — *Circuit breaker, circuit breaker testing, FFT analyzer, multi body dynamic analysis, switchgear.*

## I. INTRODUCTION

Automation has led to higher operating speeds and also high accuracy requirement by the industry led to use more sophisticated tools. The assumption of rigid members however is not justified always, in particular for the modern machinery operating at high speed, where due to

its inertia effect a linkage may experience a significant deformation which is adequate to disturb the performance of its intended function and actual operation. The high speed mechanism's functioning introduces vibrations, acoustical radiations, joint detritions and incorrect positioning due to elastic connections deformations. A circuit breaker is a device constructed to open an electrical circuit automatically on a predetermined overloading current. It can also be opened and closed by non-automatic means. The operating mechanisms of circuit breakers can be classified into three types, namely, pneumatic, hydraulic, and spring according to its actuation. Pneumatics and hydraulics are mainly used in circuit breakers with high voltage levels that require high operating force. However, for small or medium sized circuit breakers, springs are usually employed. The major aspect of the analysis is that the analysis is to be carried out during the time of its operation of about 80 milliseconds during opening operation, which makes even the measurements are difficult.

The basic design considerations of a circuit breaker are insulation, heat conduction, electrical conductivity and so on. For a gas insulated circuit breaker, the breaking time is another significant factor and is customarily designed in the range of milliseconds to avoid prolonged arcing time that may overheat and melt moving and fixed components. SF6 is an excellent gaseous dielectric for high voltage power applications. It has been used extensively in high voltage circuit breakers and other switchgears employed by the power industry. In recent years, improvements to the spring-operated circuit breaker in terms of using less operating power have been constantly in progress such that in time to come, these circuits will inevitably expand in applications [1-4]. The dynamic response of these mechanisms is generally analyzed and simulated using computer aided design

packages. Root and Ragsdell [5] presented a circuit breaker model that incorporates breaking time in the objective function with nonlinear inequality constraints, aided by the use of IMP. Jobes et al.[6] studied the structural synthesis of a residential circuit breaker by way of creative mechanism design [6] and an ADAMS model was created to simulate and optimize the mechanism. M-S. Claessens1, *et al*, presented advanced modeling methods in the design of switchgear mechanisms by considering flexibility [7] a multi-body system is an assemblage of rigid members that produce relative motions due to constraints. Multi-body dynamics [8] involves analyzing these systems with the equations of motion, which is usually derived using the Lagrange equation. This is particularly important for complicated mechanisms. One such system is the mechanism of a spring-actuated circuit breaker. The high-speed activation and rapid breaking of a circuit breaker demands that the members of the operating mechanism be tough, durable and safe to withstand incoming power supply so that in situations of system break down, they function as required.

This paper has been organized in four sections. This first section covers the introduction related to analysis of flexible mechanisms. The description of spring-type operating switch gear mechanism and its synthesis is given in the section-II. Kinematic analysis and the solution of general motion equation of mechanism is presented in the section-III. In the section-IV, the results from experiment and using ADAMS software are presented and are discussed and finally main conclusions are highlighted.

## II. DESCRIPTION OF SWITCH GEAR MECHANISM AND IT'S SYNTHESIS

### A. Description

A 69 KV switch gear mechanism is shown in Fig-1 [15]. The circuit breaker mechanism is shown at bottom of the figure.

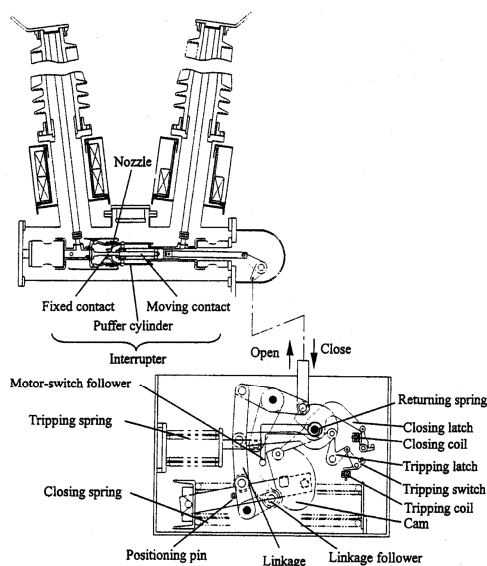


Fig.1: Circuit breaker [15]

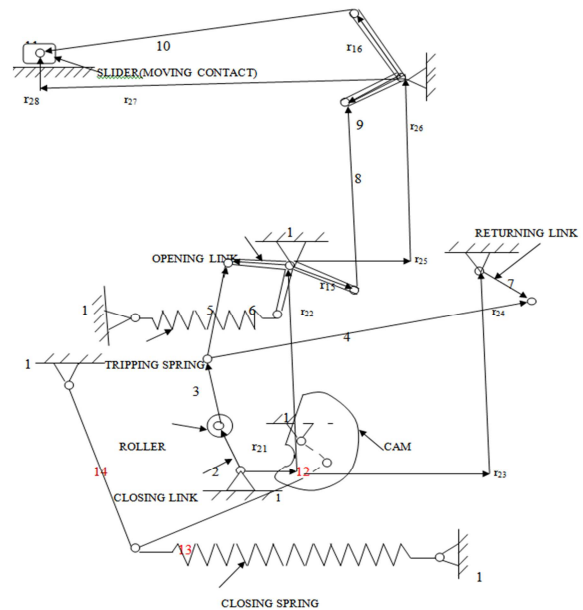


Fig.2: Vector loop diagram of switch gear mechanism

This circuit breaker has the following topological structure characteristics:

1. The controlling mechanism consists of eleven links and fourteen revolute joints, two of which form a multiple joint and the mechanism is thus considered as having two degrees-of-freedom (*dof*)
2. In each of the three basic operations, by fixing an arbitrary link that adjoins the fixed link, the mechanism is reduced to a ten-member linkage consisting of thirteen revolute joints, two of which form a multiple joint. The mechanism has thus a single *dof*.
3. The controlling mechanism has a closing link 2. During closing, this acts as the input link which compresses the tripping spring and drives the moving contact to form a loop.
4. An opening link 6, in the mechanism acts as the input link during opening. It separates the fixed and moving contacts of the interrupter and opens up the circuit.
5. The returning link 7 in the controlling mechanism acts as the input in the return operation and functions mainly to return the mechanism to the initial position in readiness for closing.

In the close, open, and return operations, different links are fixed to achieve the desired motions, which is analogous to the fixing of different links in the planetary gear train to attain the desired speed ratio [4].

For the close operation, closing spring drives the cam via a link and cam in turn drives the follower, to close up the moving and the fixed contact while compressing the tripping and contact springs. The cam rotates about  $180^\circ$ . The oscillating roller follower is followed the cycloid motion, which is being mostly used for high speed applications

### B. Synthesis of mechanism

With the presence of offset, the motion of the crank and connecting arm is no longer symmetric about the sliding axis. Therefore the crank angle required to execute the

forward stroke is different from the crank angle required for the return stroke. An offset slider crank mechanism Fig.3 provides a quick return when a slower working stroke is needed or when one stroke requires less time of operation.

A measure of the quick return action of a mechanism is the time ratio, Q, which is defined as follows,

$$Q = \text{Time of quicker stroke} / \text{Time of slower stroke.}$$

An imbalance angle,  $\beta$  can be related to the time ratio, Q by

$$Q = (180^\circ + \beta) / (180^\circ - \beta); \text{ and } \beta = 180^\circ(Q-1) / (Q+1)$$

Thus, in the design of Offset slider crank mechanism the desired time ratio can be used to determine the required imbalance angle. In this mechanism the time ratio is taken as 1.5. Hence  $\beta$  is given by  $180^\circ(1.5-1)/(1.5+1)=36^\circ$ . It should be noted that the stroke of an offset slider crank mechanism is always greater than twice the crank length. As the offset distance increases the stroke also becomes larger. The feasible range for the offset distance is  $L_1 < L_3 - L_2$ . The design of slider crank mechanism involves determining an appropriate offset distance,  $L_1$ , and the two link lengths  $L_2$  and  $L_3$  to achieve the desired stroke,  $S_{\max}$  and imbalance angle  $\beta$ .

From the three position synthesis for the Chebyshev spacings, the link lengths can be obtained for Off-set slider crank mechanism.

Similarly, the four bar mechanism of double lever type using three position synthesis, can be obtained for the required output link rotations that can produce the required slider movements.

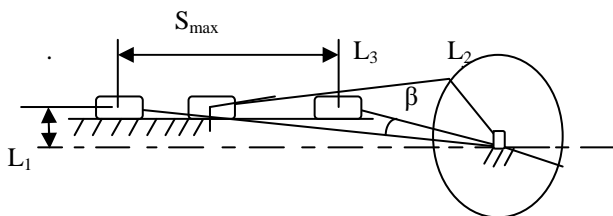


Fig.3: Offset slider crank mechanism

### III. KINEMATIC AND DYNAMIC ANALYSIS OF MECHANISM

#### A. Equation of motion

In the open operation, the action of the tripping spring rotates link-6 in the counter clockwise direction, thus causing the interrupter to separate the moving contact from the fixed contact. Therefore, in this operation, link 6 is the input link.

For all members of mechanism are rigid, their motion can be expressed by using second order non-linear differential equation as

$$I(\ddot{\theta}) + C(\dot{\theta})^2 = M(\theta) \quad (1) [15]$$

This equation is also called general equation of motion. Where  $I(\theta)$  is the inertia moment,  $C(\dot{\theta})$  is

$$(1/2)(d[I(\theta)]/d\theta)$$

and  $M(\theta)$  is the generalized moment. This equation can be solved either direct or inverse approach. In direct approach for the given  $M(\theta)$ , the parameter  $\theta$  will be found, consequently it's derivatives to find velocity and accelerations. Where as in Inverse approach the required displacement, velocity and accelerations of the slider motion, Input torque,  $M(\theta)$  values can be found. Here we are considering for the later method of solution as the time of operation of slider movement is crucial here.

If the number of links in the mechanism is  $n$ , then the  $I(\theta)$  and  $M(\theta)$  are found by using

$$I(\theta) = \sum_{i=1}^n [m_i(jh_{gix}^2 + jh_{giy}^2) + I_i] jh_i^2$$

$$M(\theta) = \sum_{i=1}^n [m_i(jh_{gix}jh'_{gix} + jh_{giy}jh'_{giy}) + I_i] jh_i h'_i$$

Where  $m_i$  is the mass of link  $i$ ,  $I_i$  is the mass moment of inertia of link  $i$ ,  $jh_i$  is the kinematic coefficient

of link  $i$  relative to the input link (link  $j$ ) and  $jh_{gix}$  and  $jh_{giy}$  are the kinematic coefficients of the centre of gravity of link  $i$  in the X and Y axis respectively relative to the input link (link  $j$ ). The sign ( $'$ ) represents the derivative of the variable. If one of the members in the mechanism is allowed only to translate, then the order of its kinematic coefficient is different to that of a member which rotates.

This coefficient is often represented by  $jf_i$ , the relative coefficient of link  $i$  to the input link (link  $j$ ).  $jf_{gix}$  and  $jf_{giy}$  are the kinematic coefficients of the centre of gravity of link  $i$  in the X and Y axes respectively relative to the input link (link  $j$ ) and  $jf'_{gix}$   $jf'_{giy}$  are their derivatives respectively.

When using the equation of motion (1), to analyze the dynamic response of a spring-type operating mechanism circuit breaker, the kinematic coefficients of the members and their centers of gravity must first be derived to obtain the coefficients in the equation of motion.

From the vector loop diagram shown in Fig-2 of linkage assembly, in open operation the following four vector loop equations can be derived

$$\vec{r}_2 + \vec{r}_3 - \vec{r}_4 - \vec{r}_7 - \vec{r}_{24} - \vec{r}_{23} = 0 \quad (2)$$

$$\vec{r}_2 + \vec{r}_3 + \vec{r}_5 - \vec{r}_6 - \vec{r}_{22} - \vec{r}_{21} = 0 \quad (3)$$

$$\vec{r}_{15} + \vec{r}_8 - \vec{r}_9 - \vec{r}_{26} - \vec{r}_{25} = 0 \quad (4)$$

$$\vec{r}_{16} + \vec{r}_{10} - \vec{r}_{28} - \vec{r}_{27} = 0 \quad (5)$$

#### B. Kinematic coefficients

These are the coefficients that are used for finding the required link-  $i$  motion characteristics whose motions are caused by input link  $j$ . This method is used to provide insight in to the motion of the linkage in which the X and Y- components of the loop-closure equation is differentiated with respect to the input position variable, rather than differentiating directly with respect to time.

$\theta_i$  is the angle of vector  $\vec{r}_i$

$$\text{and } \phi_{n_3} = (d\theta_3/d\theta_6), \phi_{n_4} = (d\theta_4/d\theta_6),$$

$$\phi_{n_5} = (d\theta_5/d\theta_6), \phi_{n_7} = (d\theta_7/d\theta_6).$$

$6_{h_3}, 6_{h_4}, 6_{h_5}$  and  $6_{h_7}$  in the open operation can be obtained as

Where  $6_{h_3}' = (d^2\theta_3/d\theta_6^2)$ ,  $6_{h_4}' = (d^2/d\theta_6^2)$   
 $6_{h_5}' = (d^2/d\theta_6^2)$  and  $6_{h_7}' = (d^2/d\theta_6^2)$ .

$$6_{h_{10}} = 6_{h_9} 9_{h_{10}} \quad \text{and} \quad 6_{h_{10}}' = 6_{h_9}' 9_{h_{10}} + 6_{h_9} 9_{h_{10}}'$$

Therefore, if the angular velocity  $\omega_6$ , and the angular acceleration  $\alpha_6$ , of the link 6 are known, then the angular velocity  $\omega_i$  and the angular acceleration  $\alpha_i$  of link I can be obtained as

$$\omega_i = 6_{h_i} \omega_6 \quad \text{and} \quad \alpha_i = 6_{h_i} \alpha_6 + 6_{h_i}' \omega_6^2$$

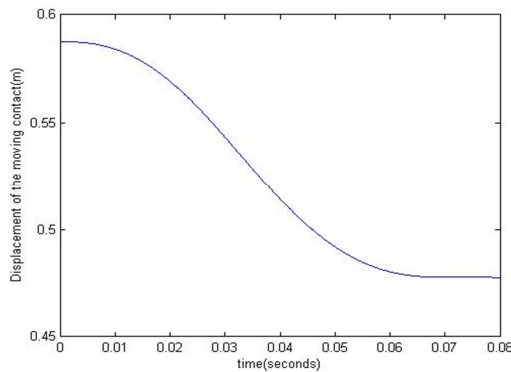
The linear velocity  $v_{27}$  and the linear acceleration  $a_{27}$  of the moving contact (*link-11*) can then be expressed as

$$v_{27} = 6_{f_{27}} \omega_6$$

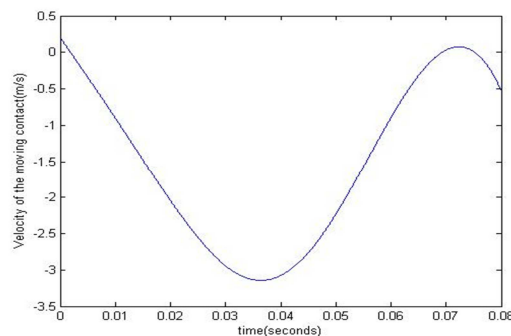
$$a_{27} = 6_{f_{27}} \alpha_6 + 6_{f_{27}}' \omega_6^2$$

Where,  $6_{f_{27}} = 6_{h_9} 9_{f_{27}}$  and  $6_{f_{27}}' = 6_{h_9}' 9_{f_{27}} + 6_{h_9} 9_{f_{27}}'$

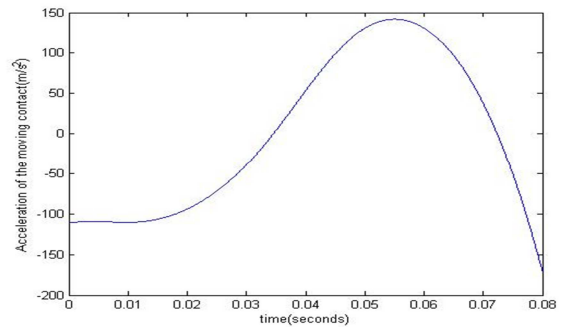
[Note:  $6_{h_9}$  indicates kinematic coefficient of *link-9*;  $6_{h_9}'$  is derivative of kinematic coefficient of *link-9*]  
 From the above equations the kinematic analysis results can be obtained as shown in *Fig-4*. The equation-(1) can then be solved. For getting the values of  $M(\mathbb{Z})$  by numerical methods after applying the initial conditions and the results obtained from the positional analysis. Here we are not solving the equations rather analyzed by using Multi body dynamic analysis software ADAMS.



*Fig.4a.* Displacement



*Fig.4b.* Velocity



*Fig.4c.* Acceleration

*Fig.4:* Kinematic analysis of the slider during open operation of 69 KV SF-6 switch gear mechanism using MATLAB program

#### IV. EXPERIMENTAL ANALYSIS

##### A. Experimental model

As the opening operation is more critical as it's time of operation is less when compared to closing and returning operations of the switch gear mechanism, in this paper we considered the analysis for opening operation of the mechanism. The model for opening operation of the switch gear mechanism is fabricated by taking all members of the mechanism are aluminium material with rectangular cross section of 25 x 3 mm where as slider is taken with plastic nylon material. As shown in *Fig-5*. This model is taken for lengths in half scale.



*Fig.5.* Fabricated model of the switchgear mechanism

##### B. Video graphic Analysis

The mechanism works when the charged spring is released, the *link-6* rotates in counter clock wise direction that causes the moving contact *link-11* moved to the right to disconnect the fixed contact. The half scale model is video graphed during the opening operation and replayed using any one of the video players like Windows media player, Real player and VLC player etc. For the opening operation, the motion of the slider *link-11* and the rotation of input link-6 are recorded with the help of video camera. The video is converted into a suitable number of frames. The frame by frame analysis at those frames where there is no distortion of the image due to high speed dynamics is chosen and snapshot is made at that time instant. The displacement curve obtained is put in mathematical cubic equation form with the application of MATLAB basic fitting. Then by differentiating that

equation, the velocity and accelerations of the input link and slider are obtained.

### C. Multi Body Dynamic Analysis

A multi-body system is an assemblage of rigid members that produce relative motions due to constraints. Multi-body dynamics involves analyzing these systems with the equations of motion, which is usually derived using the Lagrange equation. This is particularly important for complicated mechanisms. One such system is the mechanism of a spring-actuated circuit breaker. The high-speed activation and rapid breaking of a circuit breaker demands that the members of the operating mechanism be tough, durable and safe to withstand incoming power supply so that in situations of system break down, they function as required.

Multi body systems have conventionally been modeled as rigid body systems with superimposed elastic effects of one or more components. A major limitation of these methods is that nonlinear large-deformation, finite strain effects, or nonlinear material cannot be incorporated completely into model. The finite element (FE) method is used for modeling a multi body system can be employed to get better results.

The dynamic response of circuit breakers is generally analyzed and simulated using computer aided design packages. ADAMS software is a multi body dynamic analysis software that is being used for analysis of dynamic systems extensively. The switch gear mechanism elements are considered as beams, the connections between them as revolute pairs and the moving contact as a concentrated mass, the whole system as multi body system has been analyzed by using ADAMS software by considering members of mechanism both rigid and flexible.

The mechanism with same dimensions used in experiment is created in ADAMS software, the motion simulation is performed by giving input to the rotating link-6 that is obtained in the experimental analysis in the form of cubic polynomial equation. It is clear that the results of kinematic analysis obtained from the experiment and ADAMS software are appropriately equal and those are listed in table-2 and shown in Figures. 6 to 9.

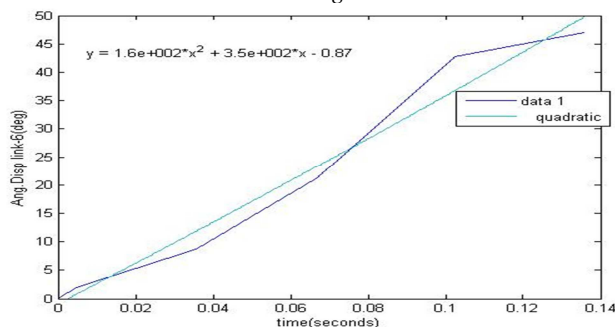


Fig.6a. (From Experiment)

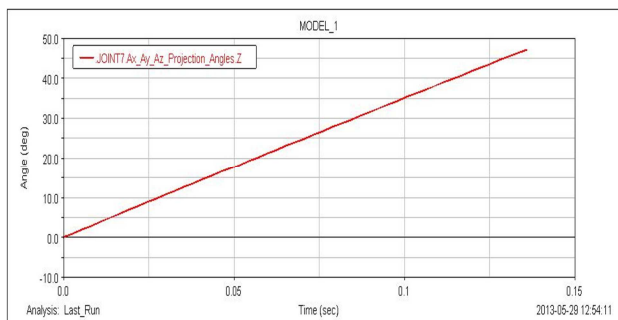


Fig.6b. (From ADAMS)

Fig.6: Angular displacement of input link-6

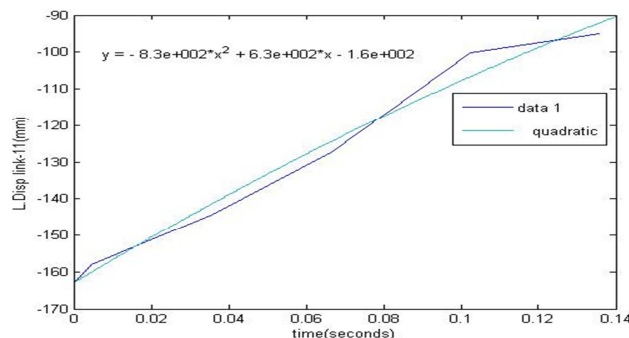


Fig.7a. (From Experiment)

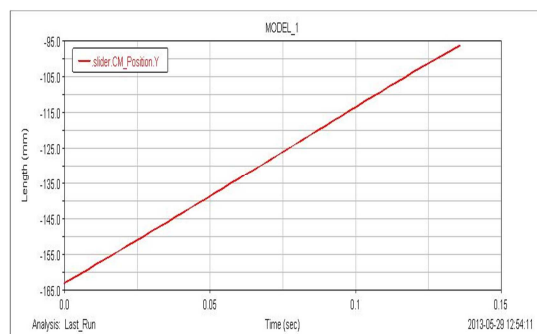


Fig.7b. (From ADAMS when links are rigid)

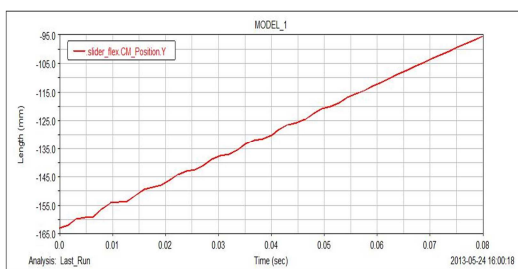


Fig.7a. (From ADAMS when links are flexible)

Fig.7: Linear displacement of slider link-11



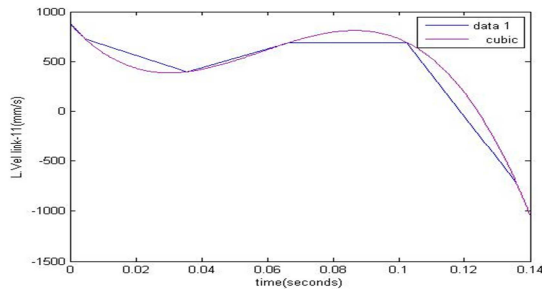


Fig.8a (From Experiment)

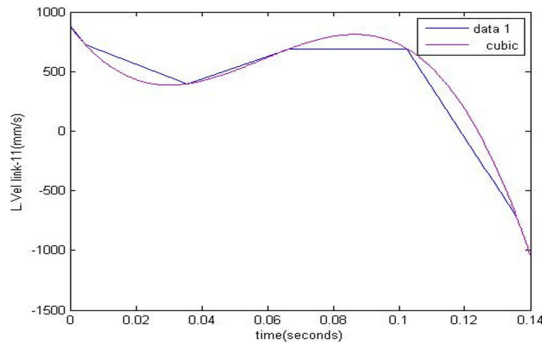


Fig.8b. (From ADAMS when links are rigid)

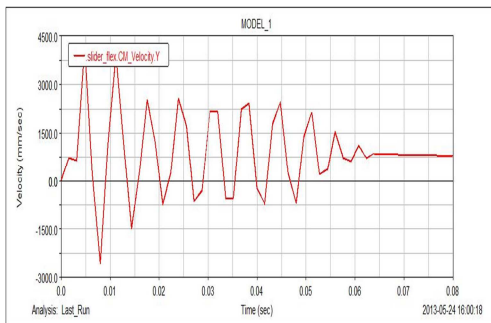


Fig.8c. (From ADAMS when links are flexible)

Fig.8: Linear velocity of slider link-11

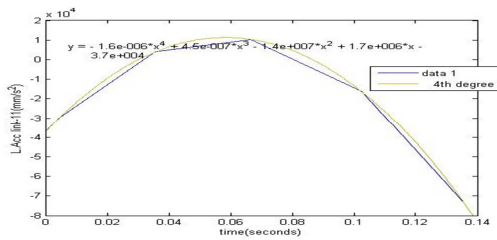


Fig.9a. (From experiment)

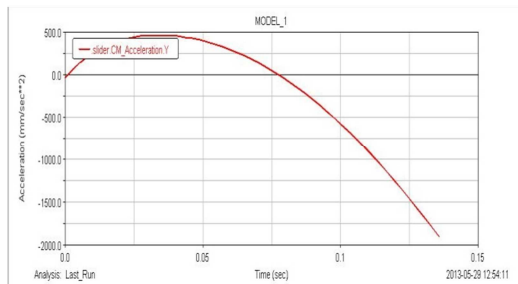


Fig. 9b (From ADAMS when links are rigid)

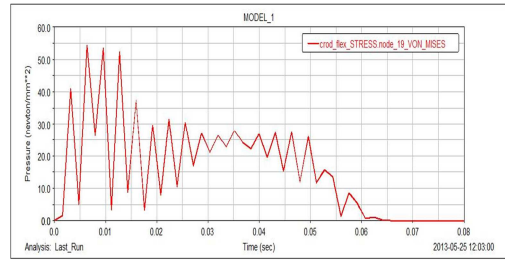


Fig.9c. (From ADAMS when links are flexible)

Fig.9: Linear acceleration of slider link-11

VON MISES Hot Spots for crod_flex Date= 2013-05-25 12:03:00						
Model= MODEL_1		Analysis= Last_Run		Time = 0 to 0.08 sec		
Top 10 Hot Spots		Abs	Radius= 0.0 mm			
Hot Spot #	Stress (newton/mm^2)	Node id	Time (sec)	Location wrt LPRF (mm)		
				X	Y	Z
1	54.4659	19	0.0064	-325.748	83.3228	12
2	53.4975	23	0.0064	-325.938	77.3258	12
3	52.9937	24	0.0064	-328.842	80.4193	12
4	51.3488	20	0.0064	-322.845	80.2293	12
5	19.5058	25	0.0064	-328.842	80.4193	6
6	19.0173	12	0.0048	-333.65	-166.052	0
7	18.0043	16	0.0048	-333.46	-160.055	0
8	16.4285	21	0.0064	-322.845	80.2293	6
9	14.5184	11	0.0048	-330.557	-163.149	0
10	13.7832	62	0.0064	-316.144	70.6652	6

Fig.10a. Vonmises stress at different nodes on connecting link-10

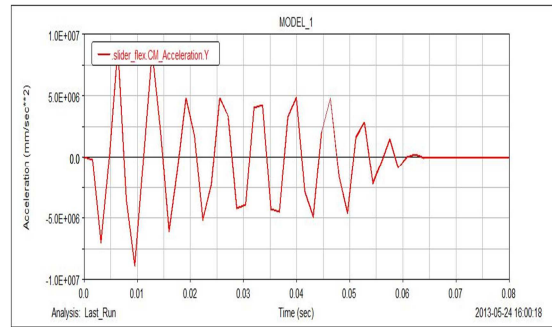


Fig.10b. Vonmises stress variation at node-14 where maximum stress developed

Fig.10: Stress plot for connecting rod link-10

TABLE 1. Results from experimental and ADAMS

Results from	Input link-6	Output Slider link-11		
	Rotation (deg)	Linear Displacement (mm)	Linear Velocity y (mm/sec)	Linear Acceleration (mm/sec <sup>2</sup> )
Experiment	47	68	550	45000
ADAMS software	47.2	70	700	50000

As the results from Experiment and ADAMS software are approximately equal, (The variations in the accelerations could be because of not consideration of damping effect) The flexible multi body dynamic analysis is further carried out in ADAMS with the use of finite element

method for all members of the mechanism, and stresses are found in all members and stresses in connecting rod link-10 are shown in the Fig-10.

To study the vibration characteristics of the mechanism (Fig.11) by using accelerometer and FFT analyzer with the help of LABVIEW software, acceleration of connecting link was measured along its length i.e along X-axis, Y-axis and Z-axis. In each case with different springs, the amplitude, damping factor, time period and natural frequency are listed in table-3. Results for 8 mm mean diameter of spring from FFT analyzer by using LABVIEW software are shown in Fig.12.

TABLE 2. Results from FFT (LABVIEW software)

S.No	Description of the arrangement	Amplitude of vibration(mm)	Damping time period(sec)	Damping factor	Natural frequency (HZ)
		In X-direction	In X-direction	In X-direction	In X-direction
1	With 8 mm dia spring(stiffness 0.2 N/mm)	21	0.9	0.08	1.12
2	With 10 mm spring(stiffness 0.1 N/mm)	13	0.8	0.07	1.26

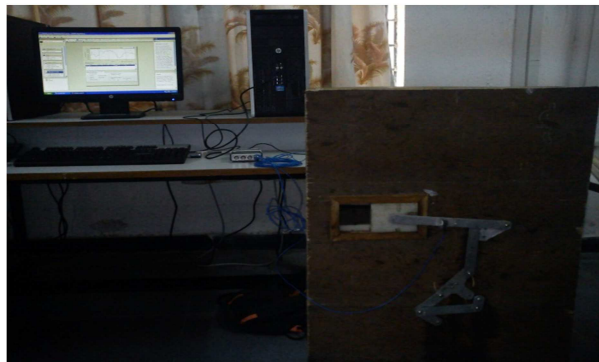


Fig.11. Experimental setup for FFT Analyzer

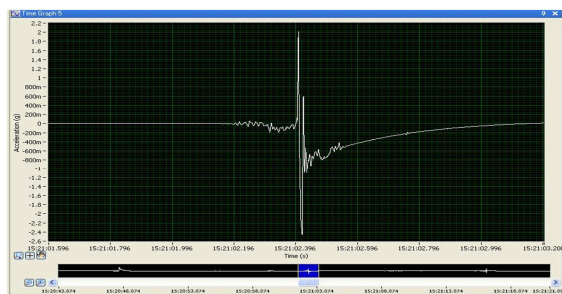


Fig.12a. Filtered Acceleration plot obtained from LABVIEW for connecting rod along its length i.e along X-axis

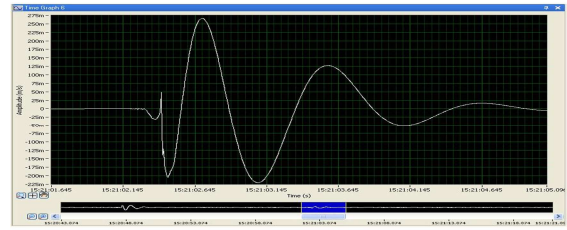


Fig.12b. Velocity plot obtained from LABVIEW for connecting rod along its length i.e. along X-axis

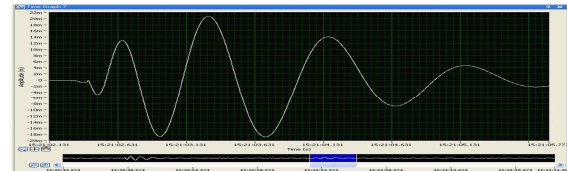


Fig.12c. Displacement plot obtained from LABVIEW for connecting rod along its length i.e. along X-axis

Fig.12. Results from LABVIEW software

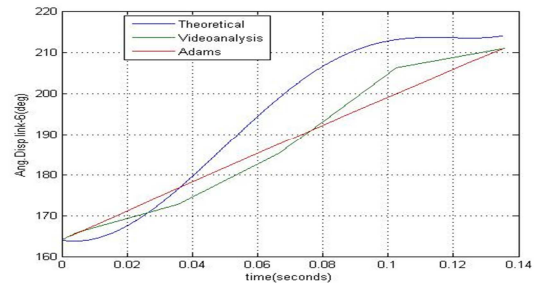


Fig.13. Comparison of Ang. Disp. Of input link-6

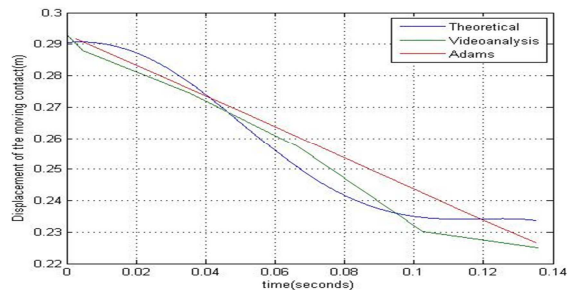


Fig.14. Comparison of Linear Disp. of slider link-11(moving contact)

## CONCLUSIONS

The main function of a circuit breaker is to open up a circuit for a pre-defined time so as to prevent a sudden surge in current that could damage the equipment due to high heat. The kinematics and dynamics of the mechanism are analyzed with the vector-loop method and the Lagrange equation. The numerical results, solved using the MATLAB shown in Fig.4 are appropriately same as the results shown by the Fu-Chen Chen [8].

The mechanism model is fabricated and made it to operate in similar way of switch gear mechanism. The model (half scale dimensions of a standard mechanism of 69 KV circuit breaker mechanism which is being in service) is tested for its dynamic response and the results of the displacement versus time are plotted with the help of

video graphic analysis. It is found that that the input rotation given by spring release for this model is exactly following the similar curve that was obtained through MATLAB program for 69 KV circuit breaker mechanisms.

As the kinematic analysis results obtained (e.g. Displacements shown in Fig.13&14) from both experimental and by using ADAMS software are matching, the analysis is further extended to carry out flexible multi body dynamic analysis to find stresses in members using ADAMS software and found that the stresses obtained are well within their safe values. To study vibration characteristics, FFT analyzer is used with the help of LABVIEW software and results are plotted and tabulated in TABLE-2. Although the results reported herein are for the dynamic response of spring-type operating circuit breakers in open operation, in similar way, the analysis can be applied for close and return operations.

This open operation duration can also be designed according to requirements by means of adjusting the stiffness of the springs. It is taken two springs for tripping with different stiffness values of 0.1 N/mm and 0.2 N/mm and it is observed that by increasing the stiffness value of the spring, the operation time can be reduced.

Therefore, the results from this research could be used as a basis for designing mechanisms with multi-degrees-of-freedom as well as for other types of spring-actuated mechanisms e.g. gunnery mechanisms.

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