Dynamic Study and Stress Analysis of Mechanical Spring Drive(MSD) in Circuit Breaker

By

Joshi Ashish Balkrishna

(10MMCC02)

DEPARTMENT OF MECHANICAL ENGINEERING
INSTITUTE OF TECHNOLOGY
NIRMA UNIVERSITY
AHMEDABAD-382481

May 2012

Dynamic Study and Stress Analysis of Mechanical Spring Drive(MSD) in Circuit Breaker

Major Project

Submitted in partial fulfillment of the requirements

For the degree of

Master of Technology in Mechanical Engineering (CAD/CAM)

By

Joshi Ashish Balkrishna
(10MMCC02)

Guided By

Prof. Jatin M Dave
Mr. Sandeep Chandalia

DEPARTMENT OF MECHANICAL ENGINEERING
INSTITUTE OF TECHNOLOGY
NIRMA UNIVERSITY
AHMEDABAD-382481
May 2012

Declaration

This is to certify that

(i) The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering (CAD/CAM) at Nirma University and has not been submitted elsewhere for a degree.

(ii) Due acknowledgement has been made in the text to all other material used.

- Joshi Ashish Balkrishna
10MMCC02

Undertaking for Originality of the Work

I, **Joshi Ashish Balkrishna**, Roll. No.**10MMCC02** , give undertaking that the Major Project entitled "**Dynamic Study and Stress Analysis of Mechanical Spring Drive(MSD) in Circuit Breaker**" submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in **CAD/CAM** of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

____

Signature of Student

Date:_________________

Place: NU, Ahmedabad

Endorsed by

(Signature of Institute Guide)

(Signature of Industry Guide)

Certificate

This is to certify that the Major Project entitled **"Dynamic Study and Stress Analysis of Mechanical Spring Drive(MSD) in Circuit Breaker"** submitted by **Joshi Ashish Balkrishna(10MMCC02)**, towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (CAD/CAM) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in this major project, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

Prof J M Dave
Project Guide,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University, Ahmedabad

Mr Sandeep Chandalia
Sr Design Engineer,
PPHV- Technology Centre,
ABB Ltd.,
Vadodara

Dr R N Patel
Head of Department
Department of Mechanical Engineering,
Institute of Technology,
Nirma University, Ahmedabad

Dr K Kotecha
Director,
Institute of Technology,
Nirma University, Ahmedabad

Abstract

The need of electrical energy without interruption is becoming more and more important. To meet this requirement, the demand on reliability of each individual component is increasing. The circuit-breaker constitutes the last link in the chain of the protection equipment for a power supply system.

The dynamic analysis of drive mechanism, Enable industries to save cost and reduce time that spend on doing mechanical testing of actual prototype of mechanism. Due to the wide range of physical principles involved in high voltage circuit-breakers, different numeric tools have been employed to simulate and understand mechanics, hydrodynamics and electrical behaviour. In power transmission or distribution network, circuit breaker provide automatic protection in response to abnormal load conditions. The circuit breaker may therefore be called upon to perform a number of different operations such as interruption of terminal faults or short line faults, interruption of small inductive currents, interruption of capacitive currents, out-of-phase switching or no-load switching.

In circuit breaker, actual opening and closing operation is carried out by two contacts: **Fixed Contact** and **Moving Contact**

The moving contact is operated by drive mechanism, where linkage system connects the drive mechanism to the moving contact. In this work initially mathematical model for four bar mechanism is simulated in ADAMS. The results of software is compared and validated by analytical calculations. Further, Various analytical solutions are obtained for different components of **MSDx1 Drive (Mechanical Spring Drive)**. An assembly is prepared and its dynamic properties are modelled in **MSC ADAMS R3 (Automatic Dynamic Analysis of Mechanical System)** for dynamic analysis. Results are obtained by running simulation in ADAMS.

Acknowledgements

I found myself very fortunate when I have got the opportunity as Project Trainee at PPHV-Technology Centre at ABB Ltd., Vadodara

First of all I would like specially thank to Project Guide **Prof J M Dave (Asst Prof, Nirma Institute of Technology)** & Industry Guide **Mr Sandeep Chandalia (Sr Design Engineer, ABB Ltd.)** without their motivation, guidance and help it was very difficult to complete my project, which I have completed to put my all efforts.

My sincere thanks and gratitude to **Mr. Kalpesh Chauhan (Asst. Vice President, ABB ltd.)** and **Dr D S Sharma (Section Head, Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad)** and **Dr R N Patel (Head of Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad)** for their continual kind words of encouragement and motivation throughout the dissertation work. I would also like to thank **Dr K Kotecha (Director, Institute of Technology, Nirma University)** for allowing me to carry out my project work in industry. I am thankful to Nirma University for providing all kind of required resources.

My sincere thanks and gratitude to **Mr. Anupam Arya, (Vice President, Head-Power Product High Voltage, ABB Ltd.)** and his team for their support.

I would like to thank **Mr. Priyank Bhatt (Engineer, ABB Ltd.)** and **Mr. Varun Parker (Sr Engineer, ABB Ltd.)** for their significant support.

Finally, I would like to thank the almighty, my family members for supporting and encouraging me in all possible ways. I would also like to thank my friends who have provided continuous encouragement in making this dissertation work successful.

- Joshi Ashish Balkrishna
10MMCC02

Contents

Declaration iii

Undertaking for Originality of the Work iv

Certificate v

Abstract vi

Acknowledgements vii

List of Table xi

List of Figure xii

Nomenclatures xiv

1 Introduction 1

1.1 Preliminary Remark 1

1.2 Multi Body Mechanical System 2

1.3 Objective of MSD Drive Development 3

1.4 Design Features 3

1.5 Operating Principle of MSD Drive 5

1.5.1 Charging operation: 5

1.5.2 Closing Operation 6

1.5.3 Opening (tripping) operation 6

1.6 Problem Definition 7

1.7 Objective 7

1.8 Motivation 8

2 Literature Review **9**

2.1 Introduction 9

2.2 Spring operating mechanism and Circuit Breaker 10

3 Analytical Calculation **17**

3.1 Gear Box calculation 17

3.1.1 Epicyclic Gear Box 17

3.1.2 Bevel Gear Box 18

3.1.3 Technical Data for Speed reduction 19

3.2 Torsion Spring Calculation 19

3.3 Monotube Hydraulic Damper 22

3.4 Rotary damper 23

4 Dynamic Simulation in ADAMS **24**

4.1 Introduction of Simulation software MSC ADAMS 24

4.2 Four Bar Mechanism Kinematic and Dynamic Analysis 24

4.2.1 Input Parameters 24

4.2.2 Analytical Calculations 25

4.2.3 Comparison of Results between Analytical Calculation & ADAMS Software 29

4.3 Steps adopted for Simulation 31

5 Results and Discussion **34**

5.1 Steps adopted for simulation 34

5.2 Middle Spring Plot (Torque Vs. Time) 36

5.3 Inner Spring Plot (Torque Vs. Time) 36

5.4 Outer Spring Plot (Torque Vs. Time) 37

5.5 Damper Parameter 38

5.6 Closing catch Contact Force Magnitude on the closing latch 39

5.7 Cam and Roller Contact Force 39

5.8 Operating Time for Closing and Opening Operation 40

5.8.1 Closing Operation 41

5.8.2 Opening Operation 41

5.9 Stress Analysis of Drive Component 42

5.10 Cam and Roller Flexible analysis 43

5.11 SF_6 Puffer Circuit Breakers 44

5.11.1 Puffer Movement 44

6 Conclusion and Scope of Future Work **46**

6.1 Conclusion 46

6.2 Scope of Future Work 47

Appendix **48**

References **49**

List of Tables

1.1 Table of Components 5
1.2 Working Direction of Drive Mechanism 7

2.1 Legend 12

3.1 Analytical method for Epicyclic Gear Box 18
3.2 Torsion Spring Properties 21

4.1 Dimension of Various Links 26
4.2 Position Analysis of Four Bar Mechanism 26
4.3 Kinematic Analysis of Four Bar Mechanism 27
4.4 Inertia properties for Links 27
4.5 Materials of Different Components 32
4.6 Inertia of different components in ADAMS 32
4.7 Joint Details in ADAMS 33

List of Figures

1.1 245 kV Circuit Breaker 2
1.2 Cross Section of MSD Drive Mechanism 4

2.1 A schematic of a Spring Operating Mechanism 10
2.2 Model of Vacuum Circuit breaker in ADAMS 11
2.3 Construction and working principle of hydraulic damper 12
2.4 Part of Operation Mechanical System 13
2.5 Vector loop diagram of Operating Mechanism 14
2.6 A model of 126kV VCB 15

3.1 Planetary Epicyclic Gear Box 17
3.2 Bevel Gear Box 19
3.3 Torsion Spring End Condition 20
3.4 Mono-tube Hydraulic Damper 22

4.1 Four Bar Mechanism in ADAMS 25
4.2 Dynamic Force Analysis of a Four bar linkage 27
4.3 Analytical and ADAMS Results for F12 x & y 29
4.4 Analytical and ADAMS Results for Force Component 29
4.5 Analytical and ADAMS Results for Force Component 30
4.6 Analytical and ADAMS Results for Force Component 30

5.1 Simulation steps in ADAMS[1] 34

5.2 Drive Mechanism in ADAMS 35
5.3 Middle Spring Torque vs Time 36
5.4 Inner Spring Torque vs Time 36
5.5 Outer Spring Torque vs time 37
5.6 Hydraulic Damper Force 38
5.7 Contact Force of Catch on Latch Gear 39
5.8 Contact Force Between Cam and Roller 40
5.9 Closing Operation 41
5.10 Opening Operation 41
5.11 Steps Followed for Flexible Analysis in MD ADAMS 42
5.12 Mesh model of Cam in ANSYS 42
5.13 Flexible Cam in ADAMS 43
5.14 Cam Roller Stress versus Time 43
5.15 Cam Roller Strain versus Time 44
5.16 Main Components of the puffer interrupter 45
5.17 Puffer Pressure 45

6.1 Closing Operation reference curve 48
6.2 opening Operation reference curve 48

Nomenclature

N_1, T_1	Input Speed and Number of Teeth
N_2, T_2	Output Speed and Number of Teeth
T	Torque
M	Bending Moment
I	Moment of Inertia
E	Young‘s Modulus
R	Radius of Curvature
L	Length of Wire
k	Torsional Stiffness
D	Mean Diameter
d	Diameter of wire
n	Number of Turns
θ	Angular Deflection
σ	Bending Stress
K	Wahl‘s Stress Factor
Q	Flow Discharge
A_o	Orifice Area
C_d	Co-efficient of Discharge
Δp	Pressure Difference
ρ	Density
V_p	Piston Speed
A_p	Piston Area
A_i	Orifice Area

Chapter 1

Introduction

1.1 Preliminary Remark

A circuit breaker[2] shown in Figure 1.1 is an apparatus in electrical systems that has the capability to, in the shortest possible time, switch from being an ideal conductor to an ideal insulator and vice versa. It is a crucial component in the substation, where it is used for coupling of bus bars, transformer, transmission line, etc. The most important task of a circuit breaker is to interrupt fault currents and thus protect electric and electronic equipment. The interruption and the subsequent reconnection should be carried out in such a way that normal operation of the network is quickly restored, in order to maintain system stability. For maintenance or repair of electrical equipment and transmission lines, the circuit breakers together with the disconnectors, earthing switches will ensure personnel safety.

It can be manually opened and closed, as well as automatically opened to protect conductor or equipment from damage caused by excessive temperature from over current in the event of short circuit. For that important consideration to above said event is the breaking and making time which is customarily design for few milliseconds to avoid prolonged arcing and pre-arcing time that overheats and thus melting moving and fixed contacts.

Figure 1.1: 245 kV Circuit Breaker

Breaker is operated by spring operated mechanism which supplies required mechanical energy to it. The drive mechanism is major component of the circuit breaker. Within a fraction of milliseconds the drive mechanism has to supply the energy which will transform the circuit-breaker from a perfect conductor to a perfect insulator. A failure of the mechanism constitutes failure of whole breaking operation.

1.2 Multi Body Mechanical System

Mechanical system is simple (like slider-crank mechanism or four-bar linkage) to more complex system such as application in robotics or vehicle dynamics. However, each mechanical system can be considered as a collection of bodies in which some or all of the bodies can move relatively to one another. The actual shape or outline of a body may or may not be of concern in the process of analysis. The location of joints, type of force direction and direction are described by following assumptions:

a. Multi-body system consists of rigid bodies and ideal joints. A body may degenerate to a particle or a body without inertia. The ideal joints include the rigid joint, the joint with completely given motion (rheonomic constraint) and the vanishing joint (free motion).

b. The topology of a multi-body system is arbitrary. Chains, trees and closed loops are admitted.

c. Joints and actuators are summarized in open libraries of standard elements.

d. Subsystems may be added to existing components of a multi-body system.

ADAMS (Automatic Dynamic Analysis of Mechanical Systems) is a CAD package for mechanical system analysis and design. It is very comprehensive, and is currently used by professional engineers. It is representative of this class of CAD tools.

1.3 Objective of MSD Drive Development

To overcome drawbacks related to conventional spring operated actuators with regard to the damping. The drive is to provide a damper for the closing that requires small space and few components, which is reliable and precise. MSD stands for Mechanical Spring Drive.

1.4 Design Features

The operating mechanism consists of three torsion springs. The inner and middle spring generate the required driving force to close the circuit breaker and charge the opening spring. The opening spring is directly connected to the circuit breaker link system. The mechanical energy in the opening spring is always stored for the opening operation when circuit breaker in closed position.

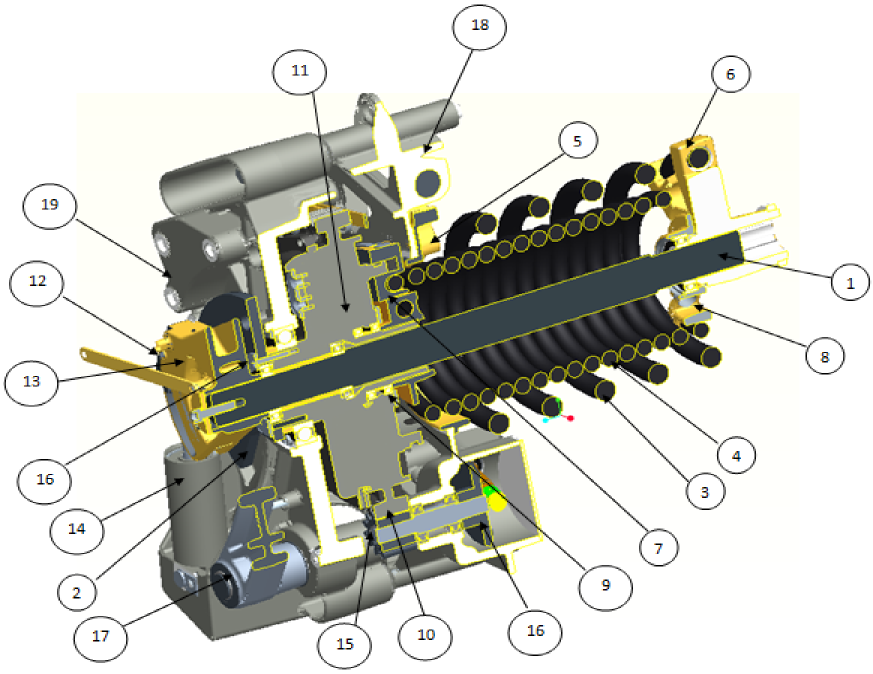

Figure 1.2: Cross Section of MSD Drive Mechanism

The springs are charging in the unwinding direction and the residual stresses are removed completely during manufacturing. So, the spring failure is reduced.

When the fault condition appeared the circuit breaker is ready for tripping operation. The closing springs again charge from the motor. The springs are kept in charged state by a latch that is released when the circuit breaker is being closed.

Table 1.1: Table of Components

Sr. No.	**Component Name**	**Sr. No.**	**Component Name**
1	Shaft	11	Damper Sidewall
2	Cam	12	Catch Operating Lever
3	Outer Opening Spring	13	Operating Lever
4	Middle Closing Spring	14	Hydraulic Damper
5	Outer Spring Support	15	Spur Pinion
6	Outer Spring Bracket	16	Bevel Gear
7	Middle Spring Support	17	Lever Assembly
8	Middle Spring Bracket	18	Cover
9	Roller Bearing	19	Casting Frame
10	Gear Wheel		

1.5 Operating Principle of MSD Drive

There are three operations carried out by drive mechanism as under[3]:

a. Charging operation,

b. Closing operation and

c. Opening operation.

1.5.1 Charging operation:

The Charging operation of MSD drive is described as under:

a. Charging operation is integrated with the closing damper. Damper is shaped as a gear wheel with external radially projecting teeth.

b. The gear wheel cooperates with a pinion driven by an electric motor via gear box.

c. During charging, the pinion drives the gear wheel about a complete turn.

d. Initially the middle spring will be charged up to 4 secs.

e. After that time with the help of spring bracket the inner spring also will be charged.

f. The cam part contacts with the roller of the limit switch due to that it cuts off the electric supply of the motor mean while the Inner spring stops the charging.

Thus the winges on both gear wheel and damper sidewall will reach a position relative to each other as described above when the closing movement starts.

1.5.2 Closing Operation

The Closing operation of MSD drive is described as under:

a. In closing operation, coil gives command to actuate closing latch.

b. The closing damper sidewall rotates in clock wise direction and breaker pole comes to the close position. The high impact of the moving contact with the fixed contact of the breaker pole is reduced by the rotary air damper.

c. Simultaneously the tripping spring gets charged from closing spring. Again the closing spring gets charged from the motor (Auto Recoiling Mechanism) and the operating lever coupled with hydraulic damper and the tripping catch.

d. Finally, Tripping catch comes into trigger position on the tripping latch.

1.5.3 Opening (tripping) operation

The tripping operation in drive mechanism will start as follows:

a. In tripping operation, coil gives command to actuate the tripping latch.

b. Tripping spring will be discharge.

c. When the moving contact disconnects from the breaker pole at that time the external force is damped by mono tube hydraulic damper.

d. Tripping operation gets over, after removal of fault condition again the closing operation starts.

The main shaft is assembled with the transmission rod of the breaker pole. The rotation direction of the main shaft for above three operations is shown in Table 5.17.

Table 1.2: Working Direction of Drive Mechanism

Component Name	**Operation Direction**
Charging operation	Clockwise Direction
Closing operation	Clockwise Direction
Opening (Tripping) operation	Counter Clockwise Direction

1.6 Problem Definition

To analyze the drive mechanism in the ADAMS software and find the contact forces of various components. Prepare and validate the model of the hydraulic and rotary air damper.

1.7 Objective

The objective of this project is to improve and optimize the MSD drive components such a way that

- It reduce mechanism response time
- It improve reliability of the mechanical component

1.8 Motivation

From past few years, The need of electrical energy without interruption is becoming more and more important. The most important task of a circuit breaker is to interrupt fault currents and thus protect electric and electronic equipment.

Much research has been undertaken concerning the dynamic behaviour of power devices like switchgears, transformers and bus bars systems. The drive mechanism is the system that stores the energy required for the circuit breaker operation. Therefore it is very important to understand and identify different modeling parameters of a new product to help smooth introduction to new circuit breaker products by means of simulations. This leads to improved products and reduces the need for time consuming and costly tests and design. Since the early 1990's, ABB has been benefiting from this research to improve and expand products in its electrical power equipment portfolio.

Chapter 2

Literature Review

2.1 Introduction

The operating mechanism is major component of the circuit-breaker and safety device of power supply system. Within a few milliseconds the operating mechanism has to supply the energy which will transform the circuit-breaker from a perfect conductor to a perfect insulator. The energy required for interrupting short circuit currents is partly taken from the arc itself, significantly reducing the energy required from the operating mechanism. Lower operating energy inherently reduces mechanical stresses, on the circuit breaker itself as well as on the foundation, and increases circuit breaker reliability. However, with the introduction of self-blast circuit breakers, the requirement of high energy for operation is decreasing and the hydraulic mechanisms are losing ground to spring-operated mechanisms. Yongpeng [4] has investigated main failure of SF_6 circuit breaker from 1988 to 1991 and found that 44 percent of all failures in high voltage circuit-breakers were due to mechanical reasons. Therefore, to achieve highest operational reliability, the mechanical design should be as simple and robust as possible.

2.2 Spring operating mechanism and Circuit Breaker

Wan-Suk Yoo and et al.[5] has presented a spring operating mechanism (SOM) is a dynamic system to open and close the circuit breaker in a voltage controlling system. For a high speed action of opening and closing within a few mili-seconds, a SOM consists of many links, joints, chains and cams. Thus, various dynamic characteristics are occurred, especially large contact forces between the cam and the roller, the shaft and the stopper.

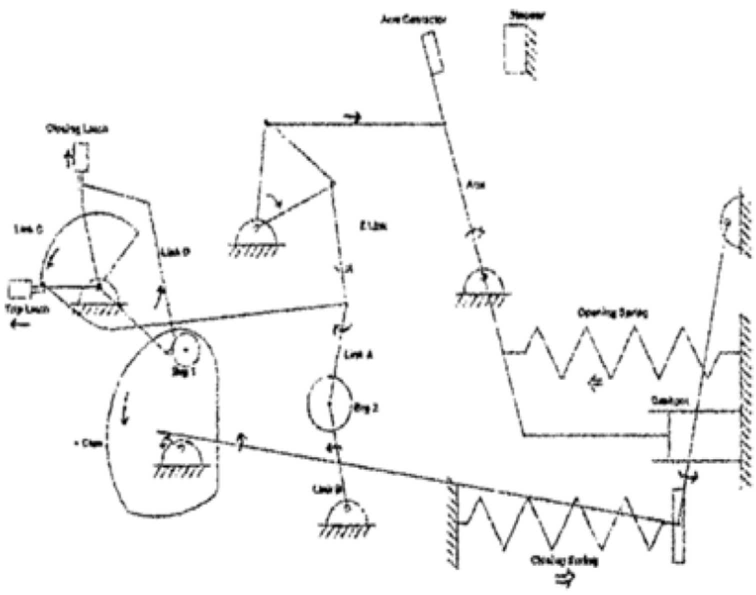

Figure 2.1: A schematic of a Spring Operating Mechanism

A circuit breaker is installed in a power system to open and close the circuit repeatedly in scores of milliseconds. Since the operating force reaches scores of tons, it is essential to calculate contact forces.

Tao Liu and Zhiying Ma [6] presented their paper on the dynamic simulation of Medium Voltage VCB (Vacuum Circuit Breaker) using ADAMS software to solve complex equation of moving characteristic of circuit breaker. ADAMS is applied in the dynamic simulation to solve the complex calculation of moving characteristics.

Satisfactory simulation results can be obtained by iterative modifying parameters of the model, and it is shown that model can be used to study the dynamic features of VCB and to provide a good basis for circuit breaker optimum design. The Figure 2.2 shows the operating mechanism after joints was added and motion and forces were applied. In this paper, the optimum moving characteristics will be obtained mainly through changing parameters of three set of springs or the orifice of hydraulic damper.

Dynamic simulation is performed to solve complex calculation of moving characteristic of circuit breaker. ADAMS is applied in the dynamic simulation of full motion, to view opening and closing characteristics, easily modify parameters and evaluate its functionality on the computer.

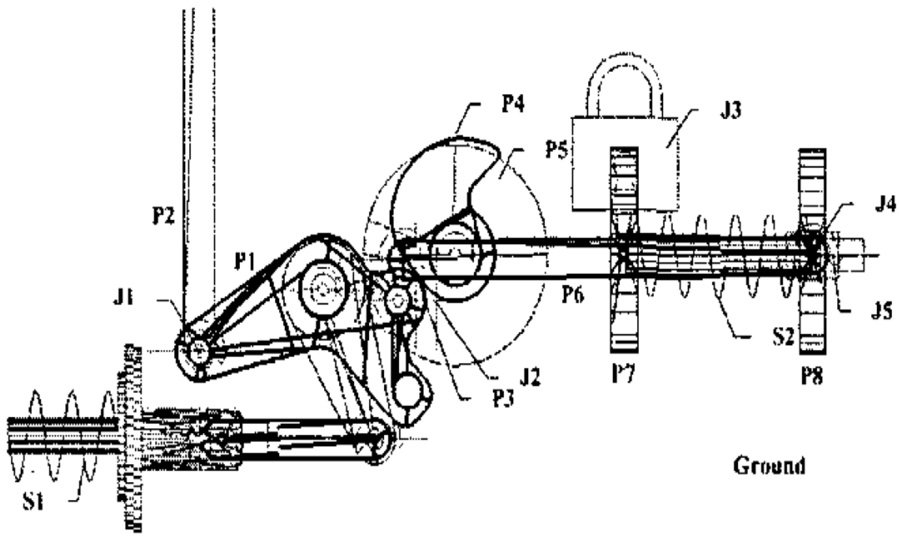

Figure 2.2: Model of Vacuum Circuit breaker in ADAMS

The moving contact's rebound travel at the end of opening process can be reduced by adjusting the position and area of orifice appropriately at the damper. Optimization is done by mainly through changing the parameters of spring or orifice of hydraulic damper.

The hydraulic damper force is a reactive force and act at the end of the travel which is often used to absorb kinetic energy of the moving contact in order to withstand impact to other parts of mechanism. It depends on the size and position of the orifice on the piston of the damper. Based on Bernoulli equation and liquid flow continuity law, buffer force can be described as follows:

The construction and working principle of hydraulic damper is as shown in Figure 2.3.

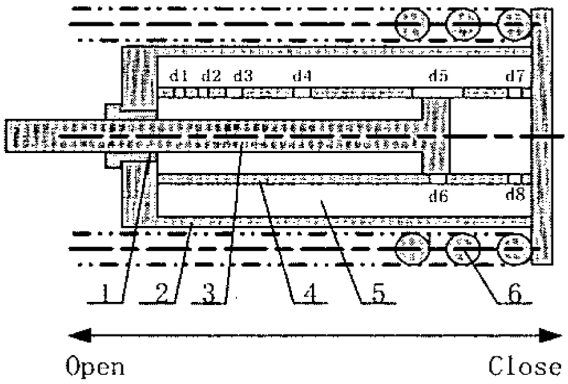

Figure 2.3: Construction and working principle of hydraulic damper

Table 2.1: Legend

1	Sealing Plate	4	Cylinder
2	Buffer Tank	5	Hydraulic Oil
3	Piston	6	Opening Spring

Dr. Sami Kotilainen and et al.[7] has presented model was created to optimize, simulate and analyze a mechanism. The drive simulation is done using MSC ADAMS dynamic software. The drive is the system that stores the energy required for the circuit breaker operation including mechanical motion when triggered by an external control system. In most modern circuit breakers, the energy in the drive is stored in mechanical springs. The mechanical transmission is a linkage that transmits and transforms the output motion of the drive into the motion of the electrical contacts in the current interrupting units. The interrupter consists typically of two sets of electrical contacts. Specifically, one is designed to conduct the nominal load current and therefore has very low resistance and thus low losses. The other is intended to withstand the very high temperature plasma arc which takes place during current interruption. In addition, the interrupter unit incorporates various nozzles, diffusers, valves and pistons which provide sufficient gas pressure, density and cooling to ensure that the current can be interrupted under all of the required operating conditions.

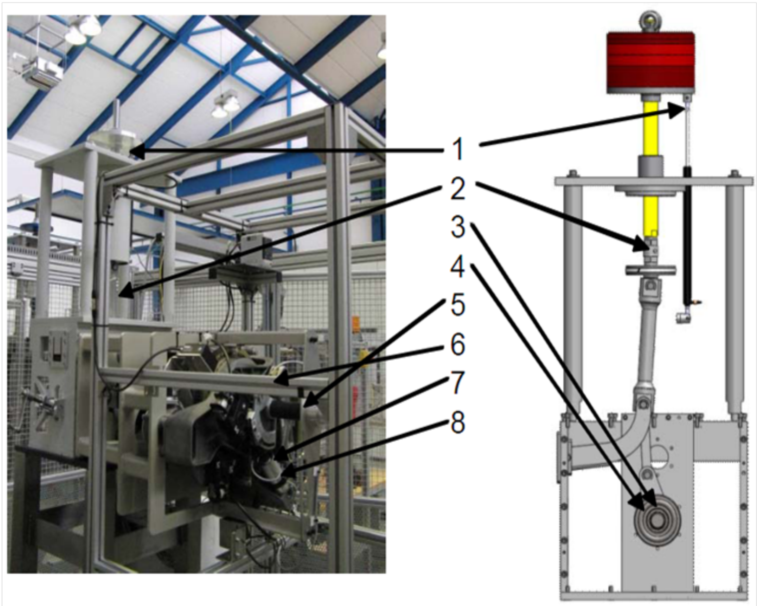

Figure 2.4: Part of Operation Mechanical System

The following states of the system are measured during the operation; the position (1) and force at the test mass (2) , rotational angle (3) and torque between the drive output and linkage (4), angle at the opposite end of the drive shaft (5), force in one of the drive links (6) and force in a drive internal damper (7) as well as position (8), voltage and current at both open and close latches.

Fu-Chen and Yih-Fong[8] have investigated the the dynamic response of a spring-type operating mechanism of a SF6 gas insulated circuit breaker in open operation. They have adopted the vector-loop method is employed to derive the kinematic coefficients of all the members of the spring-type operating mechanism and their centres of gravity. The equation of motion is then solved using the fourth order RungeKutta method. The analytical results revealed that the breaking time of the circuit breaker was 0.078 s, a mere 5% difference from the experimental result. The dynamic response of the circuit breaker with different spring constants was studied and its effect on the breaking time was also assessed. With the dynamic response known, the input driving torque of the circuit breaker can also be computed from the equation of motion.

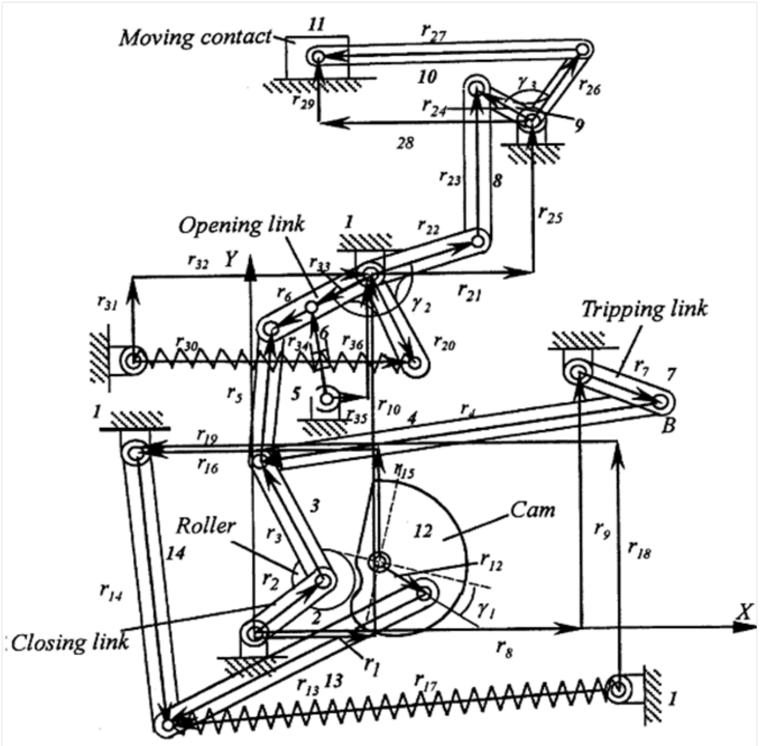

Figure 2.5: Vector loop diagram of Operating Mechanism

Li Yu and et al.[9] have presented the characteristic of damper and spring for opening and closing process. The Spring type operating mechanism is widely used in high voltage circuit breakers. Dynamics models were set up to demonstrate the relationship between displacement characteristics including initial opening speed, average opening speed, average closing speed and design parameters of operating mechanism including contact spring, over-travel, opening spring, cam and damper. Their simulation results show that the initial velocity increased significantly as over-travel and preload of contact spring increased. The damper and the cam can be designed to get a suitable displacement characteristic of movable contact.

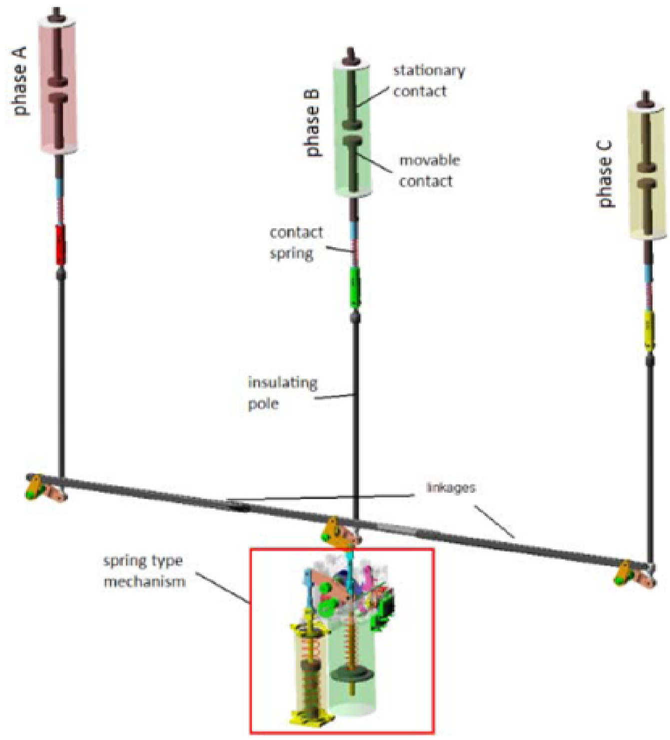

Figure 2.6: A model of 126kV VCB

Yongpeng and et al.[4] has presented Mechanism of circuit breaker is various and complex, so it is difficult to monitor the condition of all mechanical parts by limited sensors. In fact we cannot install sensor in every part of mechanism we are interested in, and it is still difficult to differentiate all conditions of circuit breaker according to sampling signals of sensors. Although we can simulate the possible failures of each interested parts by experiment and inversely get the condition of them from the field data, it is still unable to simulate all failure cases practically and to get more accurate condition parameters. To solve this problem, a software simulation method is developed in our research, which helps us investigate mechanism performance deeply.

The appearance and widely use of ADAMS (Automatic Dynamic Analysis Mechanical System) supply a powerful means for researching mechanical characteristic of circuit breaker. It has interactive graphic environment and abundant library, which make it easily to constitute three dimensional parameter models.

ADAMS provide many modules for user to simulate their mechanical system including view, solver, FEA (Finite Element Analysis), Animation, Postprocessor, etc. The basic module ADAMS/View is used to create model and simulation. Other module also has specific functions.

Chapter 3

Analytical Calculation

3.1 Gear Box calculation

3.1.1 Epicyclic Gear Box

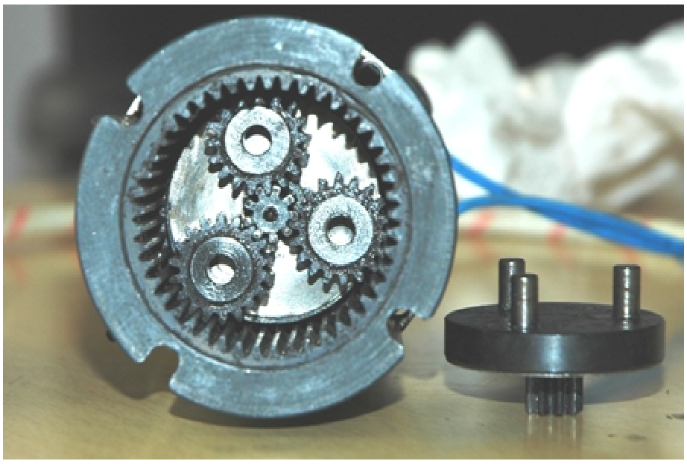

Figure 3.1: Planetary Epicyclic Gear Box

The heart of the planetary gearbox is the reduction group composed by the sun pinion and three or more planets mounted on pins which are supported by the spider. The sun pinion transmits it's motion to the planets. The planets then turn inside the ring gear which is static. As consequence the spider rotates. The motion of the spider can be transmitted to an output shaft or to another reduction group. The reduction obtained is determined by the relation between the number of teeth on the sun pinion (Z_1) and the number of teeth on the ring gear (Z_2).The ratio can be calculated using the shown formula i = Z_2/Z_1+ 1. The epicyclic gear trains are useful for transmitting high velocity ratios with gears of moderate size in a comparatively lesser space.

Table 3.1: Analytical method for Epicyclic Gear Box

Step No.	Condition of Motion	Revolution of elements			
		Spider A	Sun Pinion B	Planet Wheel C	Internal Gear D
1	Spider A fixed, wheel B rotates through +1 revolutions (anticlockwise)	0	1	$-\frac{T_b}{T_c}$	$-\frac{T_b}{T_d}$
2	Spider A fixed, sun wheel B rotates through +x revolutions (anticlockwise)	0	1	$-\frac{T_b}{T_c}$	$-\frac{T_b}{T_d}$
3	Add +y revolutions to all elements	0	1	$-\frac{T_b}{T_c}$	$-\frac{T_b}{T_d}$
4	Total Motion	y	x + y	$y-\frac{T_b}{T_c}$	$y-\frac{T_b}{T_d}$

The epicyclic gear trains are useful for transmitting high velocity ratios with gears of moderate size in a comparatively lesser space.

3.1.2 Bevel Gear Box

Bevel gears are used mostly in situations that require power to be transmitted at right angles (or applications that are not parallel). Bevel gears can have different angles of application but tend to be 90.

$$N_1 * T_1 = N_2 * T_2 \tag{3.1}$$

So that for reduced output speed is calculated by $N_2 = \frac{N_1 * T_1}{T_2}$

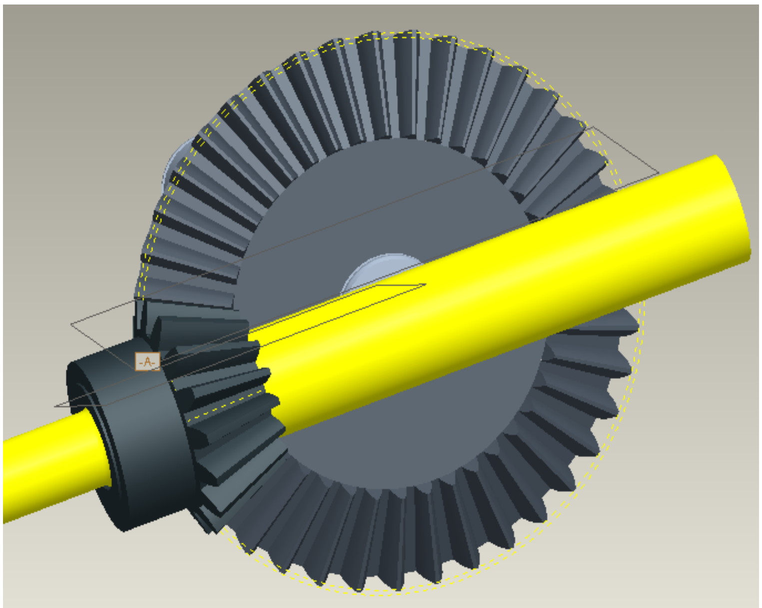

Figure 3.2: Bevel Gear Box

3.1.3 Technical Data for Speed reduction

This calculation provides the input speed to the main gear wheel as an input motion. The Equation of motion in ADAMS is,

$$\textbf{Speed} = 15.15(\text{deg}) \ / \ \text{time}(\text{sec})$$

As per the input time to dynamic simulation time the revolutions is supplied.

3.2 Torsion Spring Calculation

Helical torsion springs have essentially the same shape as helical compression, or tension, springs except that the ends are formed in such a way that the spring may be loaded by a torque about the coil axis. If the direction of loading is such as to unwind the spring, it is advisable to heat treat by means of a stress relieving treatment in order to remove residual stresses.[10]

The primary stress in helical torsion springs is bending stress whereas in compression or tension springs, the stresses are torsion shear stresses. A little consideration will

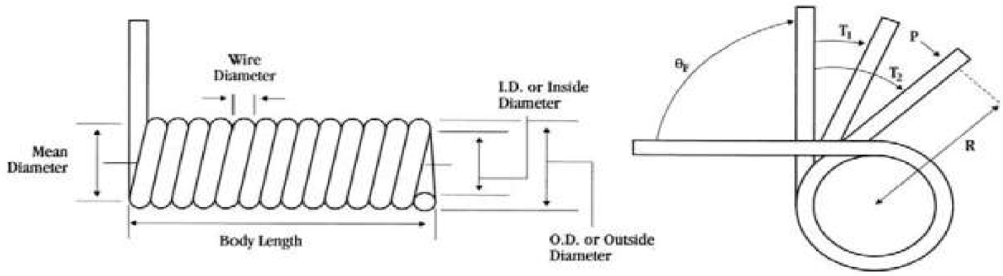

Figure 3.3: Torsion Spring End Condition

show that the twisting moment is applied to the spring. Thus, the wire is under pure bending.

The Bending equation is given by,

$$\frac{M}{I} = \frac{E}{R} \tag{3.2}$$

The radius of curvature,

$$R = \frac{L}{\theta} \tag{3.3}$$

Substituting in Equation3.2,

$$\begin{aligned} \frac{M}{I} &= \frac{E * \theta}{L} \\ \frac{M}{\theta} &= \frac{E * I}{L} \end{aligned} \tag{3.4}$$

For Circular wire moment of inertia,

$$I = \frac{\pi * d^4}{64} \tag{3.5}$$

Length of the wire,

$$L = \pi * D * N \tag{3.6}$$

The stiffness of the spring is defined as under:

$$k = \frac{M}{\theta} \tag{3.7}$$

$$k = \frac{Ed^4}{64Dn} \qquad (\frac{N-mm}{rad}) \tag{3.8}$$

When, the torsion helical spring is being wound, residual stresses are locked in the wire of the spring due to strain hardening. Residual stresses are opposite to working stresses if the moment is always applied in the winding sense. According to A. M. Wahl[[10]], the bending stress in a helical torsion spring of round wire Bending Stress,

$$\sigma = K\frac{32 * M}{\pi * d^3} \tag{3.9}$$

Wahl‘s stress factor,

$$K = \frac{4c^2 - c - 1}{4c^2 - 4c} \tag{3.10}$$

For above values of stiffness, the torque can be calculated for the input angle. The values are shown in the Table 3.2. The stiffness data is supplied to the ADAMS software as an input for Inner, Middle and Tripping Spring.

Table 3.2: Torsion Spring Properties

Description	*Tripping Spring*	*Middle Spring*	*Inner Spring*
spring index	9.5833	8.7027	5.2432
Torsional Stiffness, K (θ) (N mm/deg)	15953.3783	2544.5138	4223.3683
wahl stress factor	1.0843	1.0936	1.1655
Bending Stress, σ_b (N/mm^2)	915	863	920
Torque Applied (N mm)	1141320	491420	494320

The stiffness data is supplied to the ADAMS software as an input for inner, middle and tripping spring.

3.3 Monotube Hydraulic Damper

The hydraulic damper function is to absorb or dissipate energy. One design consideration, when designing or choosing a shock absorber, is where that energy will go. In most dashpots, energy is converted to heat inside the viscous fluid. In hydraulic cylinders, the hydraulic fluid heats up, while in air cylinders, the hot air is usually exhausted to the atmosphere. In other types of dashpots, such as electromagnetic types, the dissipated energy can be stored and used later.

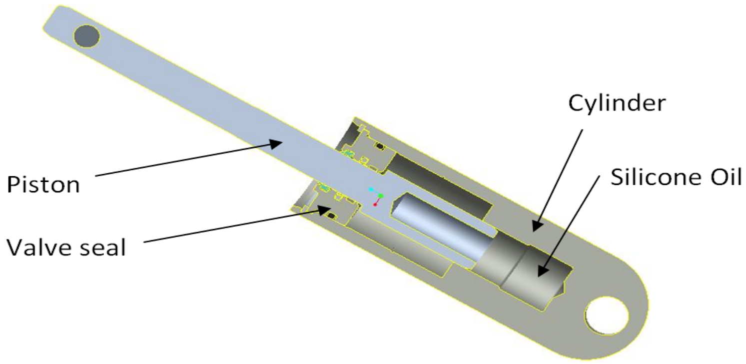

Figure 3.4: Mono-tube Hydraulic Damper

The relationship between flow rate and pressure drop can be determined using Bernoulli's Equation 3.11.

$$Q = A_o * C_d * \sqrt{\frac{2 * \Delta p}{\rho}} \tag{3.11}$$

$$\Delta p = \frac{Q^2 * \rho}{2 * A_o^2 * C_d^2} \tag{3.12}$$

The damping force is applied in the ADAMS as a function which is varying with stroke position to satisfy the variable damping.

$$F = \frac{\rho * V_p^2 * A_p^3}{2 * C_d^2 * \sum_{i=1}^{n} A_i^2} \tag{3.13}$$

$$F = C * v^2 \tag{3.14}$$

There are orifices at different position of the piston. Based on which the variation in the damping occur. In User Defined Function of the ADAMS the stroke is divided in the coding of the function. The input velocity is calculated using run-time expression and the stroke will be changed and the necessary damping will compute. The runtime expression in ADAMS provides the facility to compute the required damping force.

3.4 Rotary damper

A rotary damper is also commonly known by the term "Rotary Dashpot". The air medium is used for the purpose of damping. In present work, torque is applied as resistive force. The run time expression is used in ADAMS for the simulation of the model. The rotary air damper provide damping of relative low kinetic energy per unit time in comparison with hydraulic damper. The damper includes housing walls enclosing a circular working chamber and further includes a radial end wall and a rotatable radial displacement body within the chamber, which radial wall and displacement body sealingly cooperate with the housing walls. The walls have one outlet orifice for air displaced by displacement body.

The displacement body displaces the air out through the air outlet during the main part of its movement and then, after the displacement body has passed the air outlet, compresses the air between itself and the radial end wall. During compression stage, the rotation is damped.

$$T = [\frac{A_{cyl} * b}{360° * A_o * C_d}]^2 * \frac{\rho}{2} * \theta^2 \tag{3.15}$$

The Equation 3.15 is applied using user defined function in model and the θ is measured using run time expression in simulation.

Chapter 4

Dynamic Simulation in ADAMS

4.1 Introduction of Simulation software MSC ADAMS

Adams is the most widely used multi body dynamics and motion analysis software in the world. Adams helps engineers to study the dynamics of moving parts, how loads and forces are distributed throughout mechanical systems, and to improve and optimize the performance of their products.

4.2 Four Bar Mechanism Kinematic and Dynamic Analysis

The objective of this analysis is to simulate the four-bar mechanism using ADAMS and compare the analysis results with the analytical calculations.[11]

4.2.1 Input Parameters

The dimensions of the Four bar mechanism of interest are shown in the Table 4.1.

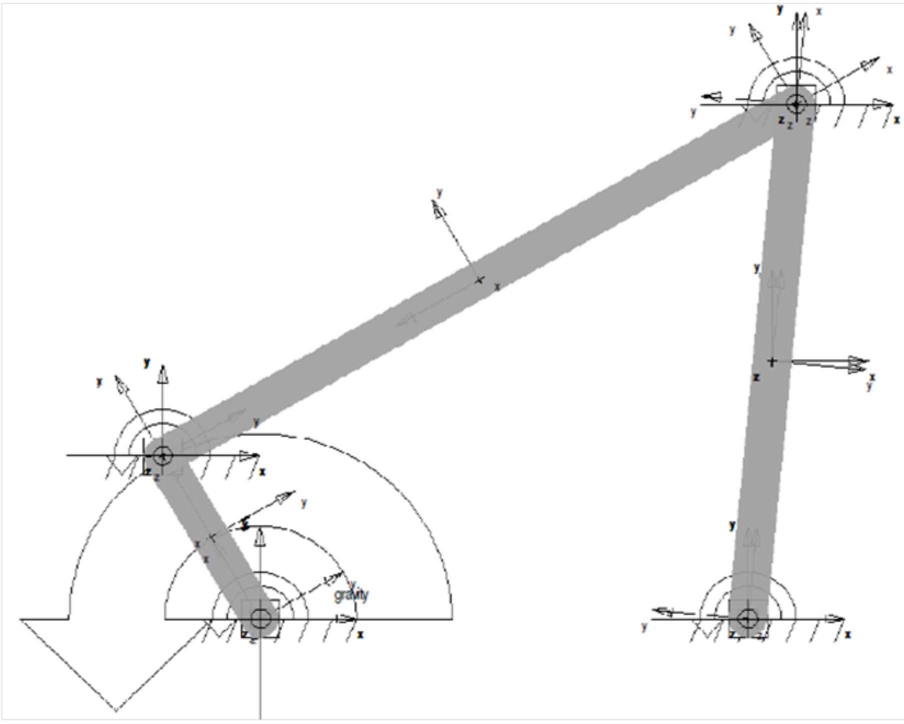

Figure 4.1: Four Bar Mechanism in ADAMS

4.2.2 Analytical Calculations

Position Analysis

The position vectors are calculated analytically from Equation 4.1, 4.2, 4.3 and 4.4

$$s = [r_1^2 + r_2^2 - 2 * r_1 * r_2 * \cos(\theta_2)]^{1/2} \tag{4.1}$$

$$\beta = \arccos \frac{r_1^2 + s^2 - r_2^2}{2 * r_1 * s} \tag{4.2}$$

$$\psi = \arccos \frac{r_3^2 + s^2 - r_4^2}{2 * r_3 * s} \tag{4.3}$$

$$\lambda = \arccos \frac{r_4^2 + s^2 - r_3^2}{2 * r_4 * s} \tag{4.4}$$

Table 4.1: Dimension of Various Links

Sr. No.	Description	Designation	Value
1	Crank	$r_2(m)$	0.1016
2	Coupler	$r_3(m)$	0.3810
3	Rocker	$r_4(m)$	0.2794
4	Fixed link	$r_1(m)$	0.254
5	Crank Angle	$\theta_2(rad)$	2.0944 (120^o)
6	Input Ang. Velocity	$\omega_2(rad/sec)$	94.2
7	Ang. Acceleration	$\alpha_2(rad/sec2)$	0

Table 4.2: Position Analysis of Four Bar Mechanism

$s(m)$	β	ψ	λ	γ
0.3172	0.2810	0.8039	1.3804	0.9573

Velocity & Acceleration Analysis

The angular velocity on coupler and rocker is calculated using following Equations 4.5, 4.6

$$\omega_3 = \omega_2 * \frac{r_2 * \sin(\theta_2 - \theta_4)}{r_3 * \sin(\theta_4 - \theta_3)} \tag{4.5}$$

$$\omega_4 = \omega_2 * \frac{r_2 * \sin(\theta_2 - \theta_3)}{r_4 * \sin(\theta_4 - \theta_3)} \tag{4.6}$$

The angular accelerations on coupler and rocker is calculated using following Equations 4.7, 4.8

$$\alpha_3 = \frac{-r_2 * \alpha_2 * \sin(\theta_4 - \theta_2) + r_2 * \omega_2^2 * \cos(\theta_4 - \theta_2) + r_3 * \omega_3^2 * \cos(\theta_4 - \theta_3) - r_4 * \omega_4^2}{r_3 * \sin(\theta_4 - \theta_3)} \tag{4.7}$$

$$\alpha_4 = \frac{r_2 * \alpha_2 * \sin(\theta_3 - \theta_2) - r_2 * \omega_2^2 * \cos(\theta_3 - \theta_2) + r_4 * \omega_4^2 * \cos(\theta_3 - \theta_4) - r_3 * \omega_3^2}{r_4 * \sin(\theta_3 - \theta_4)} \tag{4.8}$$

Table 4.3: Kinematic Analysis of Four Bar Mechanism

$\theta_3(rad)$	$\theta_4(rad)$	$\omega_3(rad/sec)$	$\omega_4(rad/sec)$	$\alpha_3(rad/s^2)$	$\alpha_4(rad/s^2)$
0.522886	1.48015	17.70707	41.8954	1011.6253	-715.8928

Force Analysis

All dimensions of link lengths, link positions, locations of the link's CGs, linear accelerations of those CGs, and link angular accelerations and velocities have been previously determined from a kinematic analysis.

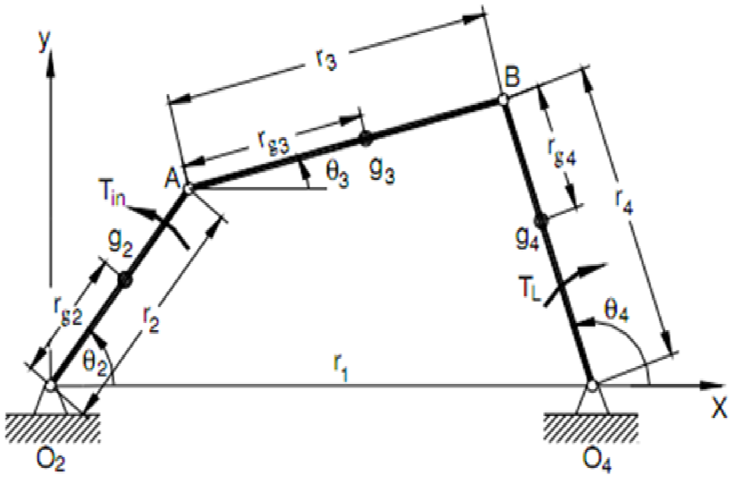

Figure 4.2: Dynamic Force Analysis of a Four bar linkage

Table 4.4: Inertia properties for Links

Link	Mass(kg)	Moment of Inertia $I_{xx}(kgm^2)$	Location of CG of each link
Crank	0.18262	0.00021534	0.0508, 0.0 deg
Coupler	0.61854	0.00812523	0.1905, 0.0 deg
Rocker	0.46002	0.00335034	0.1397, 0.0 deg

The linkage kinematic parameters are defined with respect to a global XY system (GCS) whose origin is at the driver pivot O2 and whose X axis goes through link 4's fixed pivot O4. The mass (m) of each link, the location of its CG, and its mass moment of inertia (IG) about that CG are also needed. The CG of each link is initially defined within each link with respect to a local moving and rotating axis system (LRCS) embedded in the link because the CG is an unchanging physical property of the link. The CG orientation within the link is defined by a position vector in this LRCS.

The instantaneous orientation of the CG can easily be determined for each dynamic link position by adding the angle of the internal CG position vector to the current GCS angle of the link. The position of each link‘s dynamic parameters and force locations with respect to a local, moving, but non rotating axis system (LNCS) located at its CG. These kinematic and applied force data differ for each position of the linkage.

$$\begin{bmatrix} 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ -R_{12y} & R_{12x} & -R_{32y} & R_{32x} & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & -1 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -1 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & R_{23y} & -R_{23x} & -R_{43y} & R_{43x} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -1 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & R_{34y} & -R_{34x} & -R_{14y} & R_{14x} & 0 \end{bmatrix} * \begin{bmatrix} F_{12x} \\ F_{12y} \\ F_{32x} \\ F_{32y} \\ F_{43x} \\ F_{43y} \\ F_{14x} \\ F_{14y} \\ T_{12} \end{bmatrix} =$$

$$\begin{bmatrix} m_2 a_{G2x} \\ m_2 a_{G2y} \\ I_{G2}\alpha_2 \\ m_3 a_{G3x} - F_{px} \\ m_3 a_{G3y} - F_{py} \\ I_{G3}\alpha_3 - R_{px}F_{py} - R_{py}F_{px} \\ m_4 a_{G4x} \\ m_4 a_{G4y} \\ I_{G4}\alpha_4 - T_4 \end{bmatrix} \tag{4.9}$$

4.2.3 Comparison of Results between Analytical Calculation & ADAMS Software

Force of link 1 on 2 (F12x & F12y):

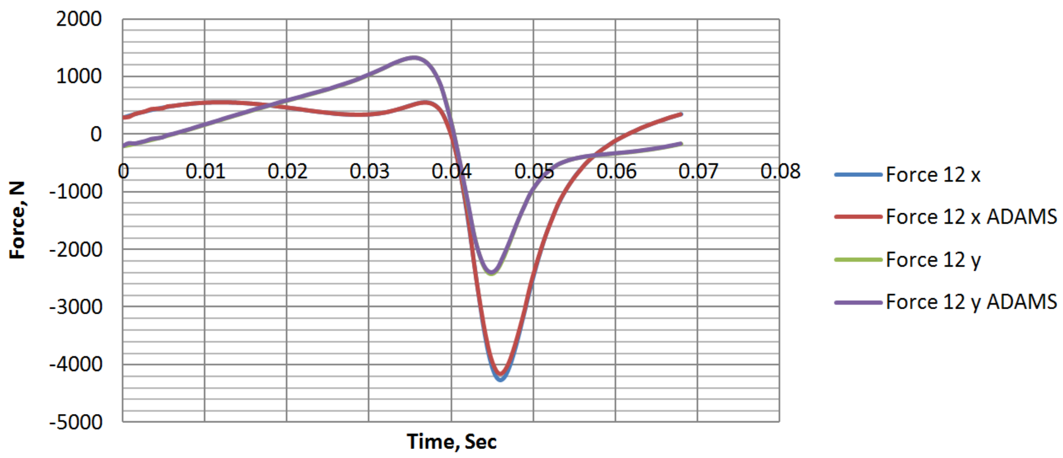

Figure 4.3: Analytical and ADAMS Results for F12 x & y

Force of link 3 on 2 (F32x & F32y):

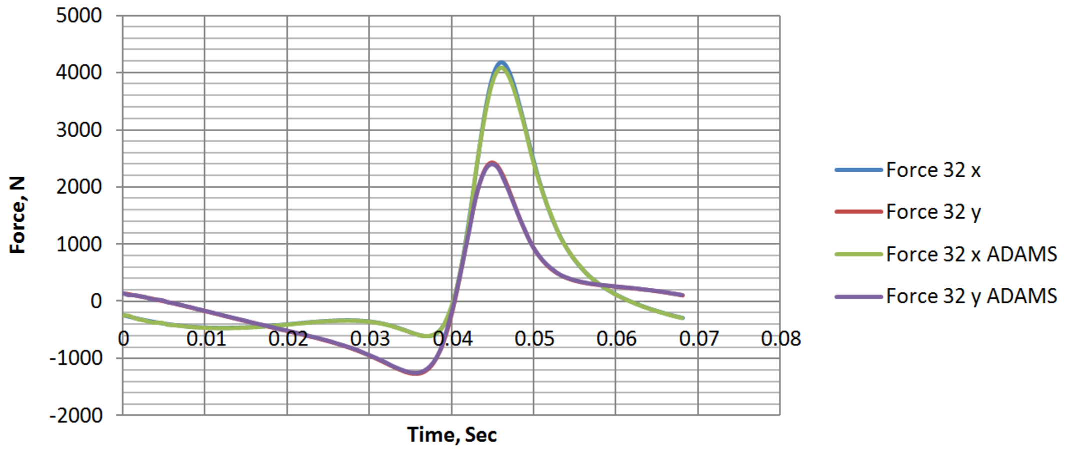

Figure 4.4: Analytical and ADAMS Results for Force Component

Force of link 4 on 3 (F43x & F43y):

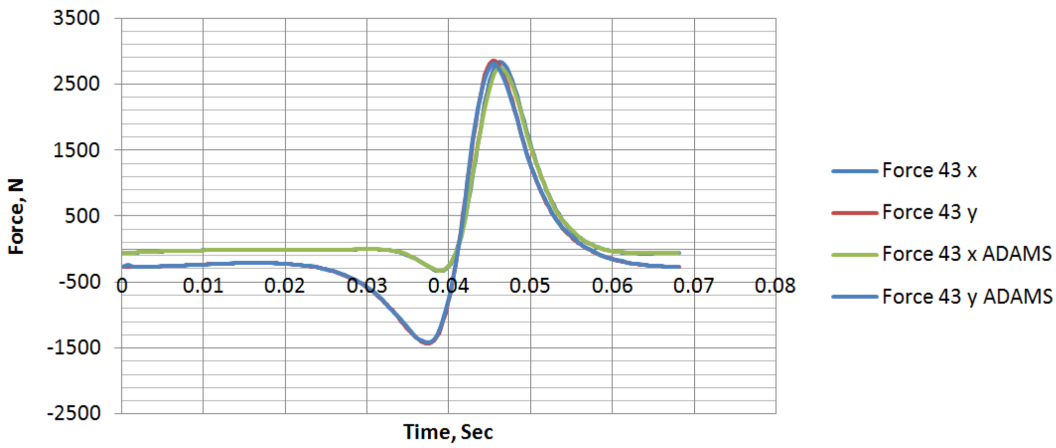

Figure 4.5: Analytical and ADAMS Results for Force Component

Force of link 1 on 4 (F14x & F14y):

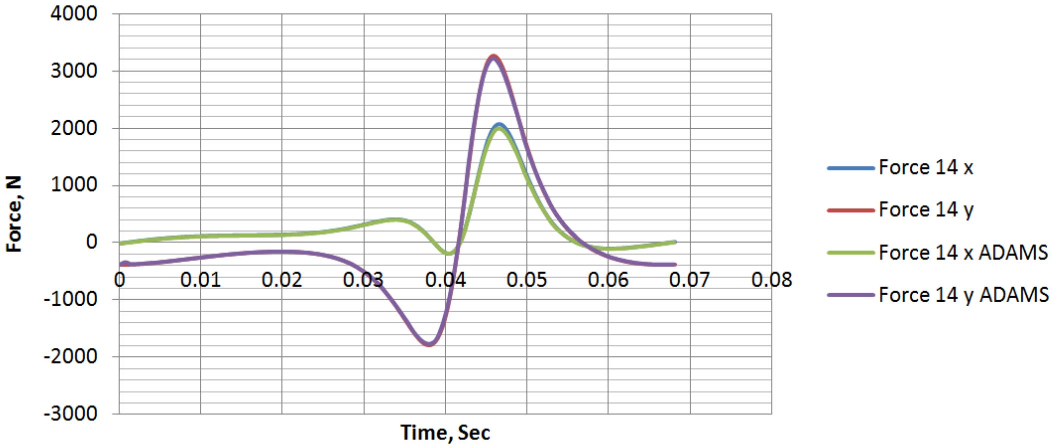

Figure 4.6: Analytical and ADAMS Results for Force Component

Figure 4.3 to 4.6 shows the results for analytical solution and solution for ADAMS for four bar mechanism. The results are found to be same for all parameters.

4.3 Steps adopted for Simulation

The steps mentioned as under

- Import geometry from Pro-E
- Define Material
- Define joints and motion to individual components
- Define flexible Connectors to measure dynamic characteristic
- Post processing in ADAMS

The geometry of MSD drive refines in to PRO-E (solid modelling software). The unnecessary parts are suppressed or removed from the assembly after refine. The refined assembly is exported into parasolid file format.

When import Parasolid geometry, Adams/Exchange reads the file and converts the geometry into a set of MD Adams geometric elements. By importing geometry from standard CAD packages, reduce the need to recreate geometry primitives within MD Adams, and ability to realistically view the behaviour of complicated mechanical systems. It is important the imported geometry is currently open and displayed in MD Adams.

Then Import the geometry in the ADAMS View software and again check the orientation & position of the assembled components. Once it is imported without error the material property is applied as an input.

Table 4.5: Materials of Different Components

	Component Name	Material Name	Young's Modulus, N/mm^2,	Poisson's Ratio	Density, kg/m^3
Casted Components	Spring Support	SG Cast Iron	169000	0.275	7100
	Lever				
	Operating Lever				
	Opening Spring Bracket				
	Spring Support				
Flexible Connectors	Outer Close Spring	Hot Rolled Steel	206000	0.3	7850
	Middle Close Spring				
	Inner Close Spring				

Joint selection is very important step to apply as a constraint to the model. The behavior of the system during simulation is depending on the selection of the joints. The following joints have been define between different components are list out in the Table 4.7.

The forth step is to define flexible connectors in which includes spring and damper. Both mass and mass less characteristic we can include during the simulation. In present simulation, mass-less spring is considered.

Table 4.6: Inertia of different components in ADAMS

Part Name	Mass kg	Ixx $kg - mm^2$	Iyy $kg - mm^2$	Izz $kg - mm^2$
Outer spring bracket	3.12E+00	1.06E+04	4.70E+03	1.51E+04
Inner & middle spring bracket	8.72E-01	2.61E+03	1.03E+03	1.94E+03
Gear	5.13E+00	4.93E+04	4.96E+04	9.51E+04
Damper side wall	3.48E+00	2.17E+04	2.08E+04	3.52E+04
Inner Spring Support	1.06E+00	1.21E+03	1.26E+03	1.94E+03
Middle Spring Support	9.67E-01	9.99E+03	3.55E+03	7.20E+03
Main Shaft	7.21E+00	9.48E+05	9.48E+05	1.73E+03
Cam	2.53E+00	7.90E+03	1.19E+04	1.97E+04
Lever Assembly	3.75E+00	2.78E+03	6.96E+04	7.01E+04
Operating Lever	3.27E+00	8.10E+03	9.40E+03	1.61E+04

The different components have the different inertia as described above based on imported file format the ADAMS generates the inertia values for different components are described Table 4.6.

Table 4.7: Joint Details in ADAMS

Types of Joint	**Joint Between Components**	
Revolute	Lever assembly	Pin
	Spring bracket	Shaft
	Link	Lever assembly
	Operating lever	Link
	Drive shaft	Ground
	Gear	Damper Sidewall
	Drive Shaft	Pinion
	Closing Latch Roller	Closing latch pin
	Lever assembly	Main Roller
Fixed	Wing Machined(2)	Sealing Plate(2)
	Limit Switch Arm(1)	Connection Link
	Limit Switch Arm(2)	Connection Link
	Limit Switch Arm(1)	Limit Switch Wedge
	Ground	Connection Link
Translational	Piston Hydraulic Damper	Cylinder Hydraulic Damper
Contact	Catch Operating Lever	Tripping Latch Roller
	Catch	Closing Latch Roller
	Cam	Main Roller
	Closing Latch Bearing	Closing Latch Roller
	Operating Lever	Pin Hydraulic Damper
	Damper Sidewall	Bolt Gear
	Gear	Pinion
	Operating Lever	Pin Hydraulic Damper

The joints defined in ADAMS are shown in Table 4.7. The proper orientation of joint is required for simulation otherwise the mechanism will be lock up position or failed to simulate.

Chapter 5

Results and Discussion

Spring Operating Mechanism(SOM) is a dynamic system to open and close the circuit breaker in a voltage controlling system. Dynamic analysis is carried out using multi-body dynamic simulation tool (ADAMS).

5.1 Steps adopted for simulation

Steps involved in the dynamic simulation is shown in Figure 5.1.

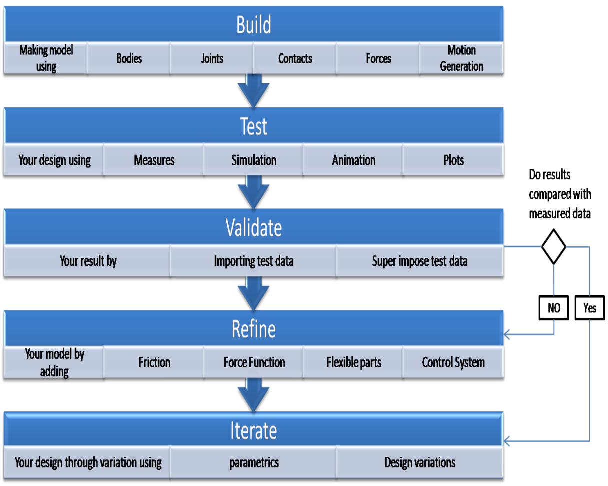

Figure 5.1: Simulation steps in ADAMS[1]

The drive mechanism is imported in ADAMS using parasolid file format. The imported model is shown in Figure 5.2.

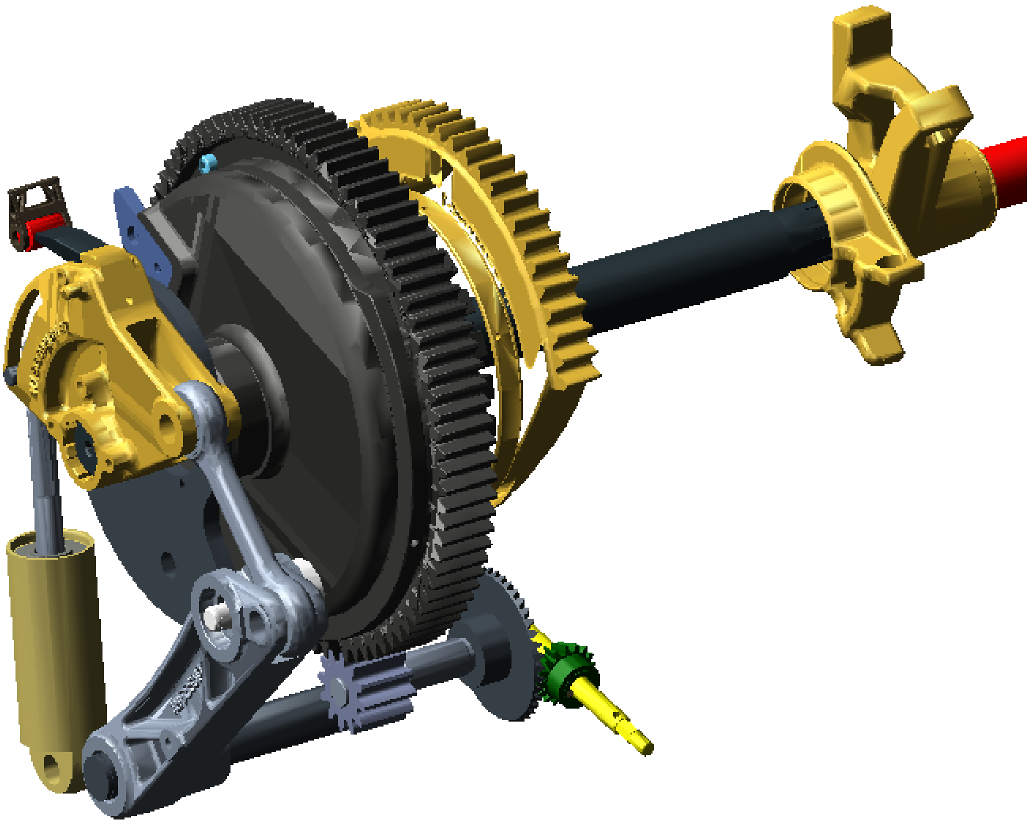

Figure 5.2: Drive Mechanism in ADAMS

The spring and damper characteristic are arrived either experimentally or by theoretical calculations. In this present simulation massless linear springs are considered. Spring torque versus time is plotted in Figure 5.3, 5.4 and 5.5.

5.2 Middle Spring Plot (Torque Vs. Time)

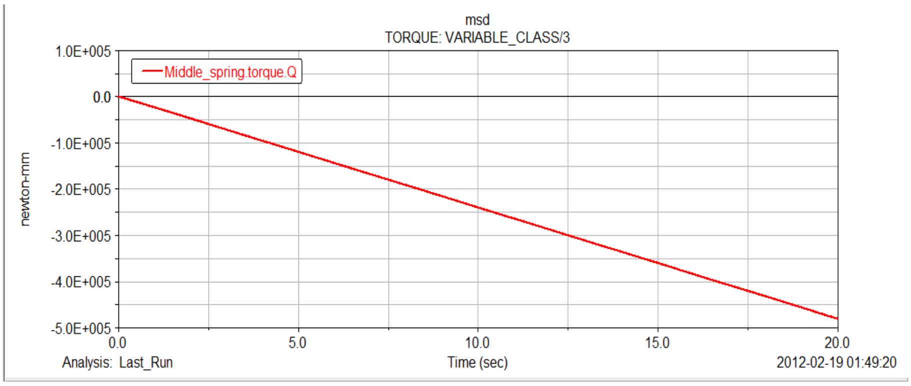

Figure 5.3: Middle Spring Torque vs Time

Middle Spring	ADAMS Results
Torsional Stiffness $(N - mm/deg)$	2544.5138

5.3 Inner Spring Plot (Torque Vs. Time)

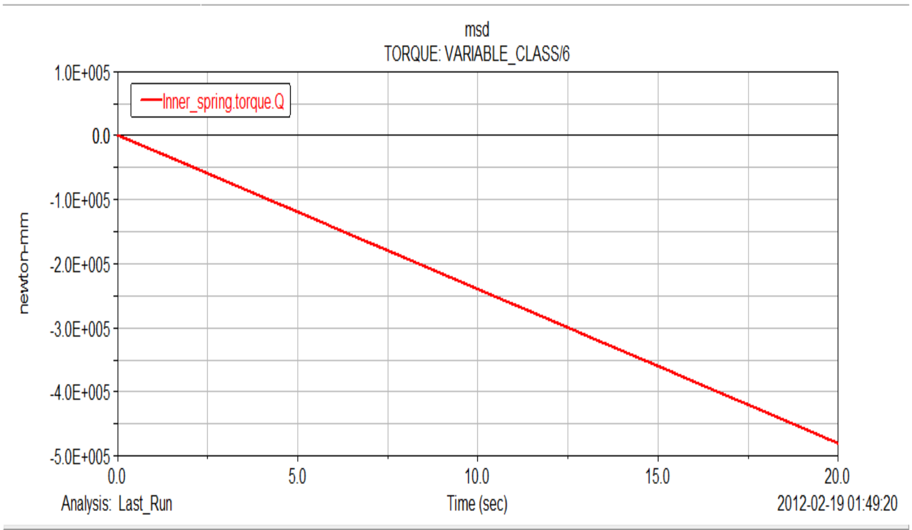

Figure 5.4: Inner Spring Torque vs Time

Inner Spring	ADAMS Results
Torsional Stiffness $(N - mm/deg)$	4223.3683

5.4 Outer Spring Plot (Torque Vs. Time)

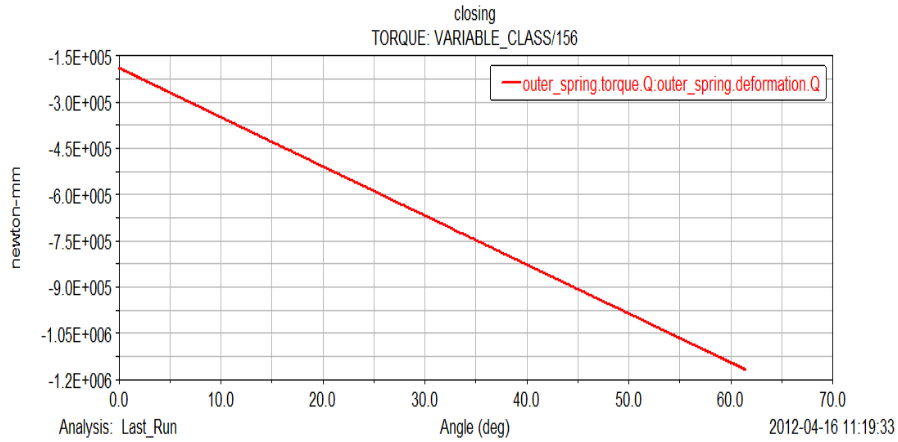

Figure 5.5: Outer Spring Torque vs time

Outer Spring	ADAMS Results
Torsional Stiffness $(N - mm/deg)$	15953.3783

The Figure 5.3 and 5.4 shows applied torque is store in inner and middle spring in clockwise direction. The torque stored is shown in Figure 5.5 by outer spring.

5.5 Damper Parameter

The damping force is linearly proportional to the square of velocity. The velocity of the mechanism is defined using run-time functions. Here it is required to specify mathematical relationships between the subsequent instant during simulation that directly define the behavior of the system.

The following Equation shows the damping force syntax applied in ADAMS using user defined function.

Damper Force = IF[VARVAL(.tripping.d)-(x_1) : IF[VARVAL(.tripping.d)-(x_2) : IF[VARVAL(.tripping.d)-(x_3) : (k_3)*VARVAL(v)^2 , (k_3)*VARVAL(v)^2 , (k_2)*VARVAL(v)^2], (k_2)*VARVAL(v)^2 , (k_1)*VARVAL(v)^2], (k_1)*VARVAL(v)^2 , (k_1)*VARVAL(v)^2]

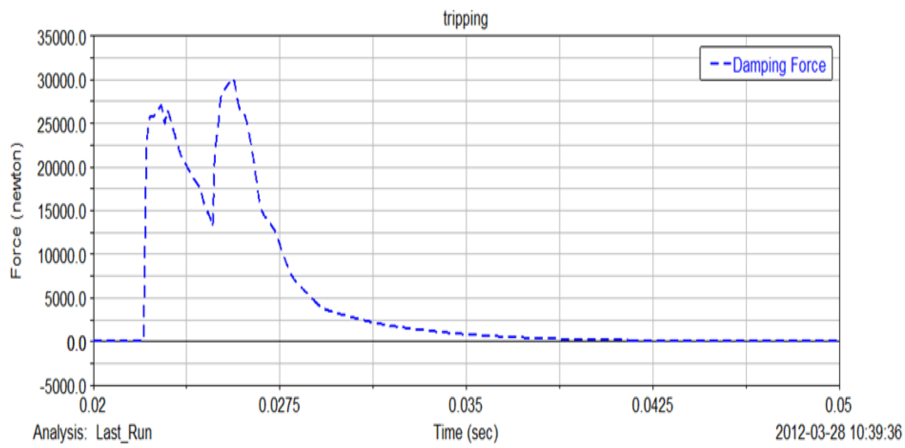

Figure 5.6: Hydraulic Damper Force

The Figure 5.6 shows the variable damping characteristic of Hydraulic damper. The Figure 5.6 shows the damping force during tripping operation. The damping force reach first high value of 27500 N at the first orifice. Between location of two orifice the force value is reduced. At second orifice the highest value of 32000 N obtained and then reduces.

5.6 Closing catch Contact Force Magnitude on the closing latch

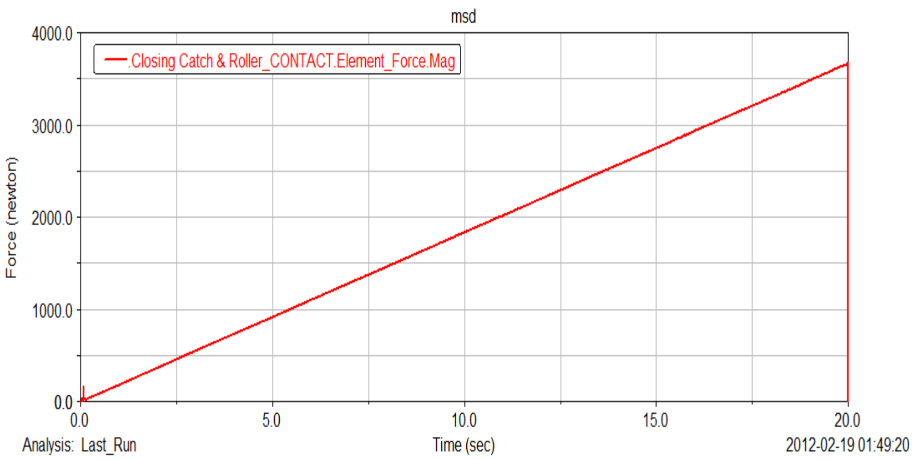

Figure 5.7: Contact Force of Catch on Latch Gear

The maximum force of catch on the closing latch roller is 3666 N.This force is maximum at the end of charging operation.

During the simulation maximum force at particular time is exerted on components is list-out in the Table:

Sr. No.	**Part Name**	**Maximum Force (N)**	**Time (Sec)**
1	Spring Bracket	31.5404	3.2522
2	Catch	197.2599	1.7699
3	Middle Spring Support	11.7987	1.7699
4	gear and pinion	108.0072	4.8658

5.7 Cam and Roller Contact Force

The contact force between Cam and Roller is shown in Figure 5.8.

The contact force on the cam and roller is 72000 N.

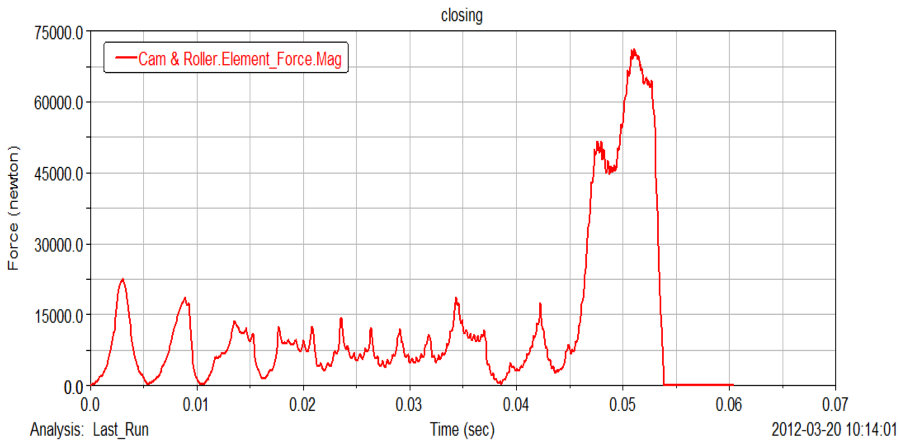

Figure 5.8: Contact Force Between Cam and Roller

5.8 Operating Time for Closing and Opening Operation

In circuit breaker the closing and opening time are very important.

Closing Time: The closing time is the interval of time from energizing of the closing release (e.g. closing coil) for a circuit breaker being in open position and the instant when the (arcing) contacts touch in all poles.

Opening Time: The opening time is the interval of time from energizing of the opening release (e.g. opening coil) for a circuit breaker being in closed position and the instant when the (arcing) contacts have separated in all poles.

5.8.1 Closing Operation

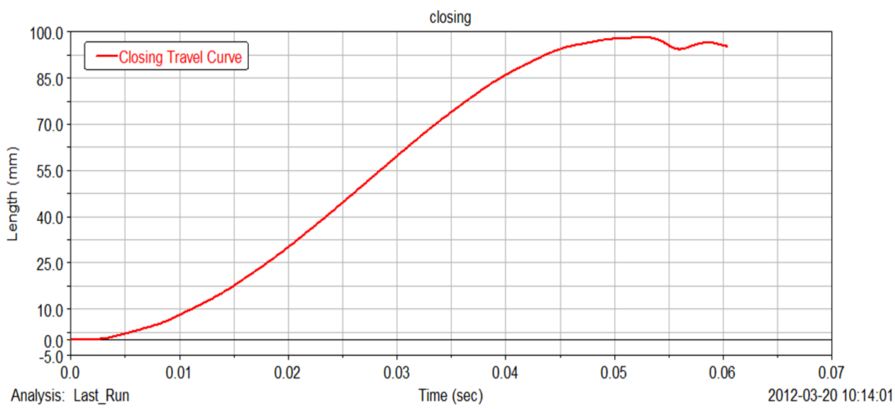

Figure 5.9: Closing Operation

During this operation the The total time taken by moving contact while extending from its bottom position to extreme position takes 55 millisecond is shown in Figure 5.9.

5.8.2 Opening Operation

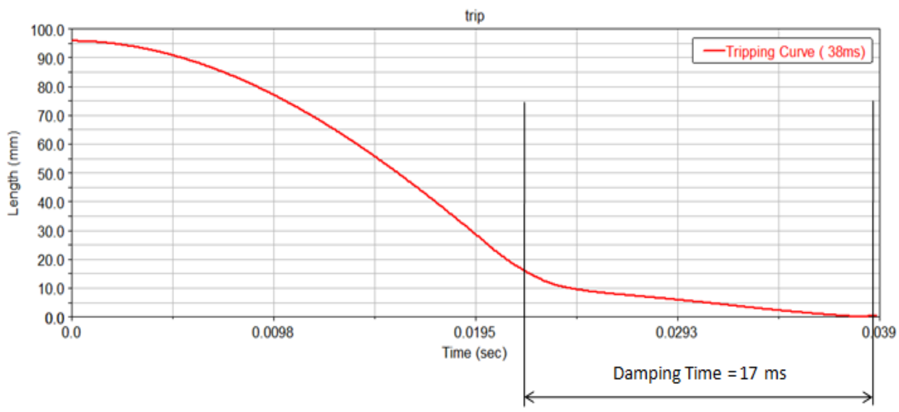

Figure 5.10: Opening Operation

The total time taken by moving contact while extending from its extreme upper position to bottom position takes around 38 millisecond shown in Figure 5.10.

5.9 Stress Analysis of Drive Component

Following Figure 5.11 are the steps to carry out flexible analysis in ADAMS. Many engineering components are rigid because their deformations and distortions are negligible in comparison with their relative movements.

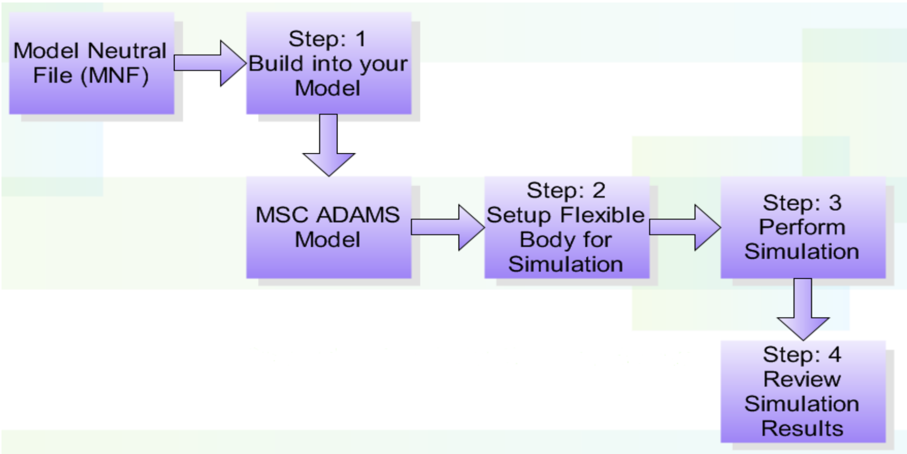

Figure 5.11: Steps Followed for Flexible Analysis in MD ADAMS

To convert flexible part, meshing of the model in ANSYS is carried out as shown in Figure 5.12. To define joint in ADAMS, rigid coupling is required in mesh model. Mass 21 element is used for rigid coupling to transfer all joint load to part surface.

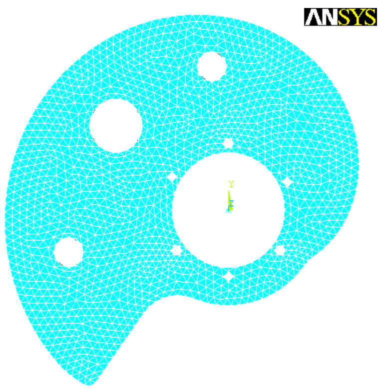

Figure 5.12: Mesh model of Cam in ANSYS

Now mesh model is used to create a modal neutral file (.mnf) and import it to ADAMS environment. After importing mnf file shown in Figure, Apply the joint at

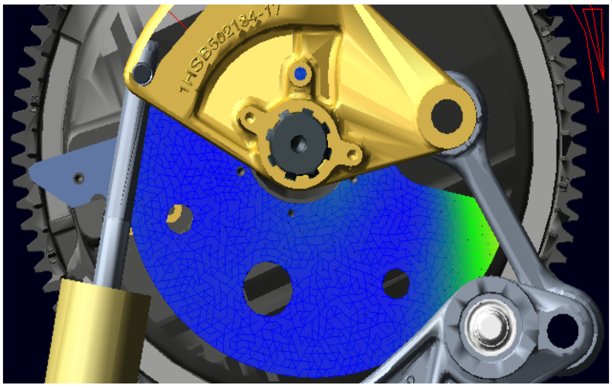

Figure 5.13: Flexible Cam in ADAMS

the rigid coupling and simulate. The hot spots are locations of high stress or strain on a flexible body or rigid stress object.

5.10 Cam and Roller Flexible analysis

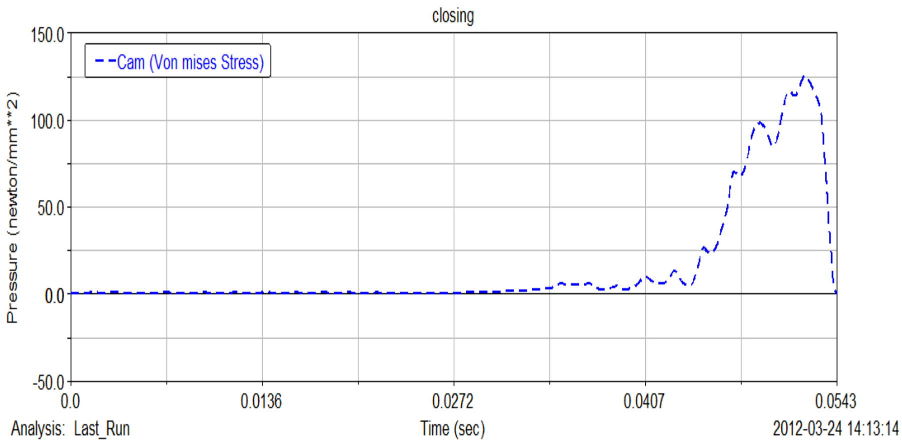

Figure 5.14: Cam Roller Stress versus Time

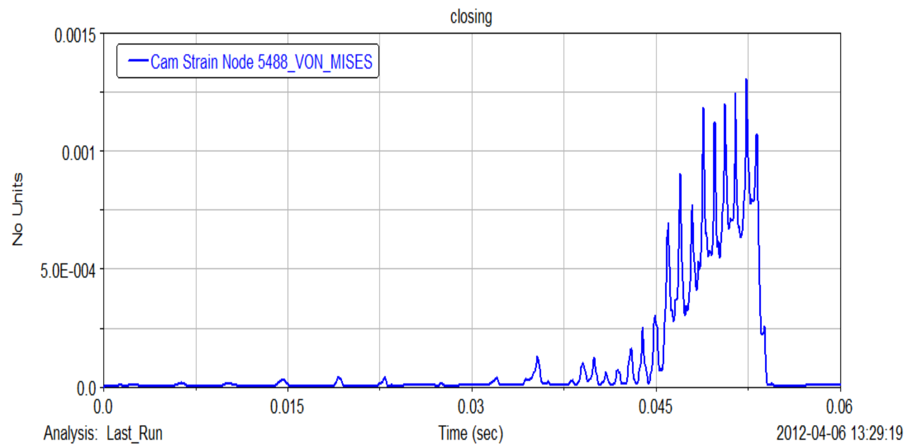

Figure 5.15: Cam Roller Strain versus Time

5.11 SF_6 Puffer Circuit Breakers

In the SF_6 puffer, the gas pressure for the cooling blast is created during the opening stroke in a compression cylinder. In the opening operation, the compression of the gas will start at the same time as the contacts start their motion. The compressed gas is blown out through an insulating nozzle in which the arc is burning. Figure 5.16 shows the function of a puffer interrupter. The insulating nozzle is normally made of PTFE (Teflon).

5.11.1 Puffer Movement

The puffer pressure during the opening operation can be seen in Figure 5.17. The puffer pressure is measured. The stroke for the first case is 25 mm, 35 mm for the second case, 45 mm for the third case. The locking pressure will found out by the simulation.

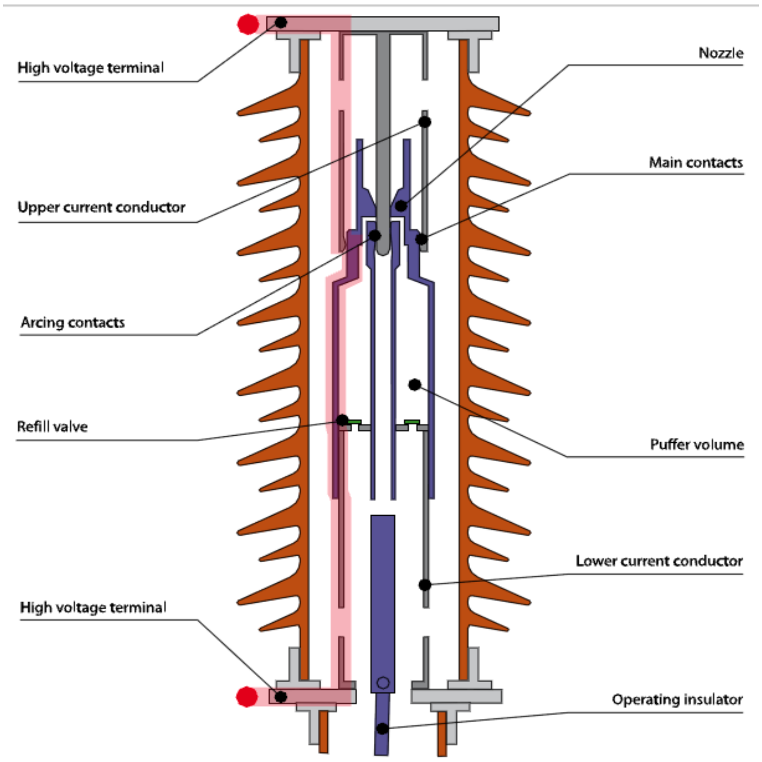

Figure 5.16: Main Components of the puffer interrupter

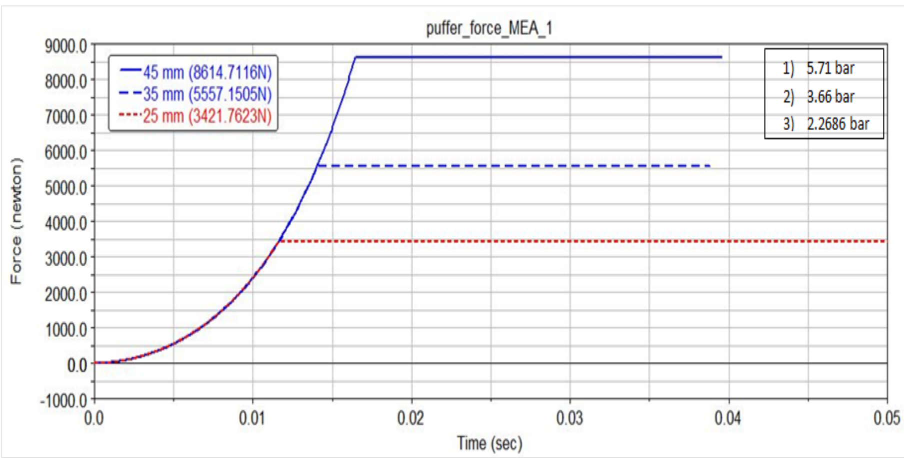

Figure 5.17: Puffer Pressure

Chapter 6

Conclusion and Scope of Future Work

6.1 Conclusion

- Input data charging time and torque is obtained by calculation for gears and spring.
- Damper is mathematically modeled for variable orifice diameter and its location. An equation to find force is obtained and it is converted into algorithm for software calculation and results are within limits.
- Multibody dynamic simulation is carried out for MSD (Mechanical Spring Drive) mechanism.
- The results for spring from analytical solution are found to be same.
- As cam-roller and catch assembly is important for tripping operation the forces are obtained for these components. Further FEA is carried for cam by defining flexible body in software. The stresses are within limit.
- The hydraulic damper and rotary air damper are modeled in the ADAMS. The pressure generated inside the hydraulic cylinder is verified.

- The displacement of moving contact is for both opening and closing is obtained from software. It is found to be very close with reference curve.

6.2 Scope of Future Work

- Fatigue analysis of drive components can be carry out to evaluate reliability of components.
- Optimization of the drive performance by changing the spring and damper orifice position.
- Detail mathematical modeling and programming of Rotary air damper in ADAMS.

Appendix A

245 kV CB Reference Curve (Closing Operation)

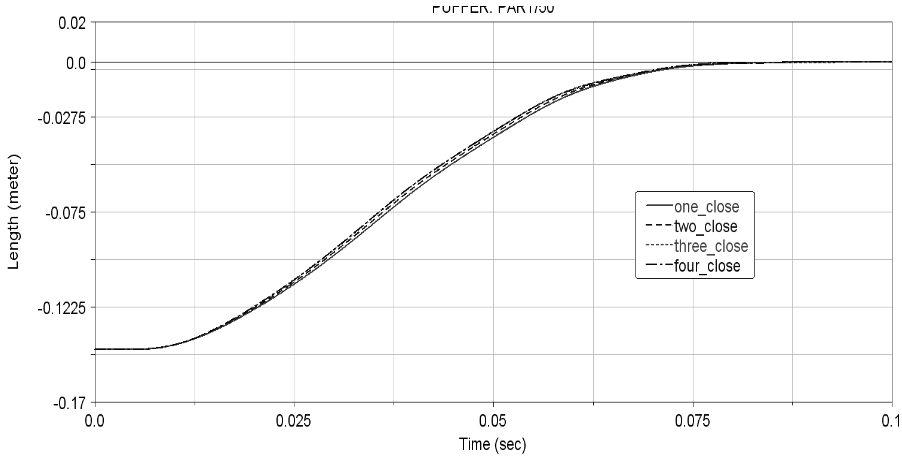

Figure 6.1: Closing Operation reference curve

245 kV CB Reference Curve (Opening Operation)

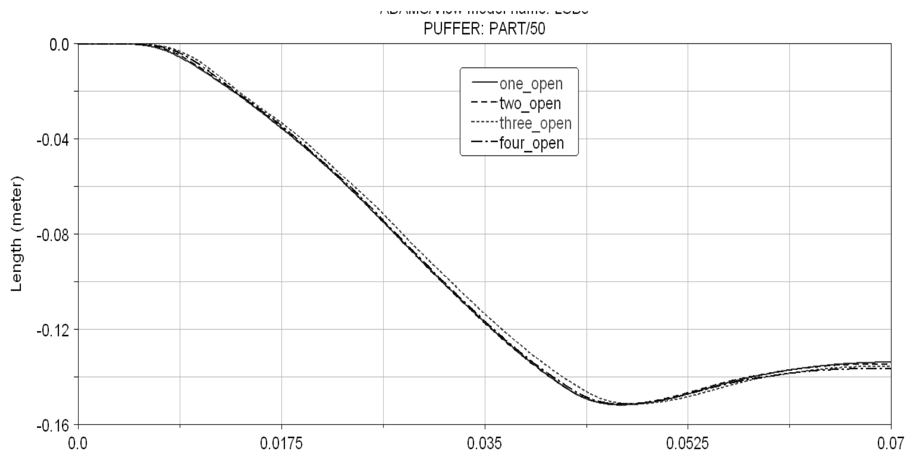

Figure 6.2: opening Operation reference curve

References

[1] MSC Software Corporation. *ADAMS Student Guide 12.0.*

[2] 1HSM9543 23 02en LiveTank Circuit Breaker Application Guide. *www.abb.com/highvoltage/1hsm954323-02en live tank circuit breakers-application guide.pdf.*

[3] ABB Technology. Msd drive apparatus patent. *International Publication Number WO 2011/054737 A1*, pages pages 137–140, May 2011.

[4] Mingzhe Rong Wu Yang Yongpeng Meng, Shenli Jia and Zongqian Shi. Mechanism dynamic simulation and analysis of medium voltage circuit breaker,. *IEEE, pages 1619-1623*, 2002.

[5] Seung-Oh Kim Wan-Suk Yoo and Jeong-Hyun Sohn. Dynamic analysis and design of a high voltage circuit breaker with spring operating mechanism. *Journal of Mechanical Science and Technology, Vol. 21, pages 2101-2107.*

[6] Tao Liu and Zhiying Ma. Design and dynamic simulation of moving characteristics of 126 kv vacuum circuit breaker. *IEEEpages 307-310*, pages 307–310, 2004.

[7] Kotilainen Dr. Sami. Chladny Dr. Ryan. and Budde Dr. Christoph. Multi objective optimal model validation of multi body system in circuit breaker application.

[8] Fu-Chen Chen and Yih-Fong Tzeng. On the dynamics of spring-type operating mechanism for 69 kv sf6 gas insulated circuit breaker in open operation. June 2001.

[9] Zhiyuan Liu Liqiong Sun Li Yu, Yingsan Geng and Jianjun Yao. Relationship between displacement characteristics of a 126 kv vacuum circuit breaker and spring type operating mechanism. 2008.

[10] Wahl A. M. *Mechanical Springs.* Mc Graw Hill Book Company, chap-11, pages 133-143.

[11] Norton L. Robert. *Design of Machinery,.* Mc Graw Hill Book Company.